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CONS/NSF/42-1

PRELIMINARY FEASIBILITY EVALUATION OF COMPRESSED AIR STORAGE POWER SYSTEMS

Volume I

FINAL REPORT PERIOD
JUNE, 1975 — DECEMBER, 1976

Date Published — December, 1976

Work Performed Under a
Non-ERDA Contract R76-952161-5

UNITED TECHNOLOGIES
RESEARCH CENTER
EAST HARTFORD, CONNECTICUT

MASTER

ENERGY RESEARCH & DEVELOPMENT ADMINISTRATION USA

ENERGY RESEARCH AND
DEVELOPMENT ADMINISTRATION
Division of Electric Energy Systems

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PREPARED FOR THE
ENERGY RESEARCH AND DEVELOPMENT ADMINISTRATION
Office of the Assistant Administrator for Conservation

Division of Electric Energy Systems

SUPPORTED BY
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FOREWORD

This preliminary feasibility evaluation of compressed air storage power systems was initiated by the National Science Foundation as part of its Research Applied to National Needs (RANN) program. Midway through the program, responsibility for technical oversight was transferred to the Energy Research and Development Administration. The program was supported, in part, under NSF Grant No. AER 74-00242 to the United Technologies Research Center. The Commercial Products Division of the Pratt & Whitney Aircraft Group and the Power Systems Division of United Technologies Corporation contributed to the program, as did two subcontractors, Acres American Incorporated and Oswald C. Farquhar. The work was initiated in June 1975 and completed in December 1976. The final results are presented in two volumes comprising:

Volume I - Final Technical Report
Volume II - Appendixes

The United Technologies Research Center provided overall program management and made major contributions in the areas of power generation equipment, auxiliary equipment and special design considerations, environmental considerations, parametric performance analyses and system optimization. This effort was performed under the general guidance of F. L. Robson, Chief, Utility Power Systems and involved the following people:

A. J. Giramonti, Program Director
W. A. Blecher
W. R. Davison
D. R. Dohrmann
S. J. Lehman
R. D. Lessard
R. L. Sadala
E. B. Smith

The effort of the Commercial Products Division of the Pratt & Whitney Aircraft Group of United Technologies Corporation dealt with turbomachinery design and layout and involved the following people:

W. T. Dennison
B. A. Stauffer

Mr. S. Hamilton of the Power Systems Division of United Technologies Corporation provided interdivisional liaison and design review.

The effort of Acres American Incorporated was in the areas of general siting criteria, north central region survey, cavern excavation technology, special aspects of compressed air systems, cavern layout and costs, environmental considerations, plant design and cost estimates, and generation alternatives. This work was performed under the general guidance of J. L. Haydock, Vice President

Special Projects and involved the following people:

I. A. Milne, Program Manager, June 1975 to March 1976
M. J. Hobson, Program Manager, April 1976 to Dec. 1976
C. L. Driggs
D. R. McCreath
R. J. Pine
M. L. Walia

The effort of O. C. Farquhar dealt with the section on the northeast region survey.

The program team also included three electric utilities, Northeast Utilities Service Company (NUSCO), New England Electric System (NEES), and Potomac Electric Power Company (PEPCO). NUSCO and NEES participated in most program review and overview meetings and were instrumental in securing the cooperation of NEPLAN to provide appropriate utility data for New England. NUSCO also provided informal consulting assistance on several utility-related matters and made available to the program the results of a comprehensive geological study of prospective areas for underground caverns in New England. PEPCO provided appropriate utility and geological data for their service area. These efforts involved the following people:

For NUSCO:
B. E. Curry
A. Ferreira
R. W. Goodrich
R. H. Meyer

For NEES:
S. K. Batra
R. O. Bigelow
D. R. Campbell
H. Lurie

For PEPCO:
A. J. Como
P. W. Robb
J. H. Rumbaugh

New England Power Planning (NEPLAN) made available to the project appropriate utility data for New England. This effort involved the following people:

J. R. Smith
A. W. Barstow
W. S. Ng

An independent technical overview committee comprising utility, industry, and government organizations was established to periodically review progress, critique results, and make recommendations for subsequent work. The committee membership

varied, but include at one time or other the following people:

W. L. McGaw, Chairman
R. Beckwith, Commonwealth Edison
R. Bishop, Connecticut Department of Planning and Policy
J. Belanger, Connecticut Department of Planning and Policy
C. Calabrese, Consolidated Edison
H. Curran, Hittman Associates
S. Jansson, Stal-Laval
E. Loane, GPU Service Corporation
W. Messner, Consolidated Edison
J. Pepper, EPRI
H. Thomas, Connecticut Department of Environmental Protection
P. Torpey, Consolidated Edison

While under the technical supervision of NSF, program management was provided by:

Dr. Richard I. Schoen, Deputy Director
Division of Advanced Energy and Resources
National Science Foundation

Dr. Thomas P. Anderson
Division of Intergovernmental Science and Public Technology
National Science Foundation

Dr. Aly Mahmoud
Division of Advanced Energy and Resources
Research and Technology
National Science Foundation

After the transfer of technical oversight responsibility to ERDA, program management was provided by:

Mr. J. Charles Smith
Division of Electric Energy Systems
Energy Research and Development Administration

The support and assistance of all the aforementioned people and organizations in contributing to the successful completion of this program is acknowledged with deep appreciation.

PRELIMINARY FEASIBILITY EVALUATION OF COMPRESSED AIR STORAGE POWER SYSTEMS

TABLE OF CONTENTS
VOLUME I

TABLE OF CONTENTS (Cont'd)

TABLE OF CONTENTS (Cont'd)

TABLE OF CONTENTS (Cont'd)

TABLE OF CONTENTS (Cont'd)

TABLE OF CONTENTS (Cont'd)

PRELIMINARY FEASIBILITY EVALUATION OF COMPRESSED AIR STORAGE POWER SYSTEMS

PART I

SUMMARY

The object of this program was to conduct a preliminary technical, economic, and environmental feasibility evaluation of generating peak power with a compressed air power system incorporating a modified state-of-the-art gas turbine and an hydraulically—compensated, mined, hard-rock cavern. The compressor and turbine sections of the gas turbine would be alternately coupled to a motor/generator for operation during different time periods. During nighttime and weekend off-peak periods, low-cost power would be used to compress air which would be stored in the underground cavern. During subsequent daytime peak-load periods the compressed air would be withdrawn from storage, mixed with fuel, burned and expanded through the turbines to generate peak power.

This preliminary evaluation program consisted of a three-phase analytical study effort which provided an in-depth, engineering investigation into the uncertainties associated with the air storage concept. Results are presented covering the siting potential and economics for hard-rock storage caverns, the types of aboveground equipment which could be used with suitable modifications, system performance and economics, and the potential for electric utility application.

The technical approach was based on technology currently available, although in some cases not yet reduced to commercial practice. By focussing on state-of-the-art technology it was possible to identify a low-risk design approach capable of producing about 250 MW which could be considered for near-term commercial application. The design approach has the distinct advantage of providing considerable flexibility with respect to storage conditions without altering the basic configuration or expensive low-pressure components. This practical approach to design was augmented by inclusion of three electric utilities on the project team to infuse utility needs, objectives, background, and experience into the program.

This program was supported, in part, by National Science Foundation Grant No. AER 74-00242. Responsibility for technical oversight was subsequently transferred to the Energy Research and Development Administration, Division of Electric Energy Systems.

CONCLUSIONS

1. Compressed air power systems are technically feasible and potentially attractive for future peak-load power generation. These systems are useful for load-leveling purposes and could be used in a daily cycle, which would charge and discharge each day, or a weekly cycle, which could make extensive use of low-cost weekend electricity.

2. By utilizing off-peak electricity derived from coal or nuclear resources for compression, the consumption of premium petroleum fuels would be only 40 to 50 percent of that consumed by conventional fossil peaking plants. The fuel consumption could be further reduced to only 30 to 40 percent by the use of high effectiveness recuperators.

3. Underground caverns excavated in large, hard-rock formations and hydraulically compensated by small surface reservoirs are one viable means of storing high-pressure compressed air. The storage caverns would comprise gallery systems excavated by conventional mining techniques in massive granite, shale, or limestone formations. The economics of this mode of storage improve significantly as storage pressure (and cavern depth) increase to values above 50 atm (1700 ft).

4. Compressed air power systems are not subject to the same degree of siting difficulties faced by conventional pumped hydro plants. Sufficient siting opportunities exist for mined caverns in the north central region comprising Illinois, Indiana, Ohio and Southern Michigan and the northeast region comprising a corridor 100-miles wide from Boston to Washington, D.C. Approximately 20 percent of the area of these two regions is underlain by suitable rock formations. All major cities, and many secondary ones, in these regions are within 50 miles of a suitable formation. Similar siting opportunities should exist throughout most of the United States.

5. Low-risk designs for the aboveground facilities are feasible using state-of-the-art equipment with relatively modest modifications. Standard designs for the relatively expensive low-pressure (below 16 atm) equipment could be prepared. The high-pressure booster compressor and expansion turbine could be selected to provide the specific overall pressure ratio corresponding to the air storage requirements dictated by local geological conditions or specific utility optimization.

6. A prudently designed compressed air power system should have little long-term adverse impact on the environment. The combustion systems can be expected to meet or exceed the anticipated air emission regulations and the hydraulic compensation system would not significantly affect ambient water quality. Any impact would be highly localized and associated with the construction or mere presence of the facility. The use of ring dikes with proper landscaping could make the plant aesthetically pleasing and provide proper noise abatement.

7. The champagne effect associated with air dissolution in the vertical water shaft and loss of pressure for a hydraulically compensated cavern can be controlled by: a) using a U-tube, or water seal, extending about 10 percent below the cavern floor, b) enlarging the cavern volume by about 10 percent to allow the cavern pressure to drop by simple air expansion, c) circulating fresh water through the cavern to prevent the water from becoming saturated, or d) suitable combinations of the above.

8. The competitive position of compressed air power systems is strongly dependent on utility-specific factors such as: existing generation mix, site availability for conventional pumped hydro, fuel costs, load factor, weekly load distribution, utility carrying charges, and projections for the future costs and performance of alternative peaking plants.

RECOMMENDATIONS

The compressed air power system concept has not matured to the point where it could be considered as a viable commercial alternative to conventional power generation methods. Several key issues dealing with station siting, cavern integrity, machinery design, system optimization, and cost estimates remain unresolved, especially for US applications. The next phase in the development of compressed air storage should be a comprehensive demonstration program in which the concept is reduced to practice by one or more utilities. There is much latitude in the formation of a demonstration program, encompassing as it can a variety of plant complexities, different underground storage types, parallel technical studies, and other supporting studies. The various technical issues have been sufficiently defined that a comprehensive program can be outlined which would demonstrate the basic technical feasibility of compressed air storage while providing the kind of information which would enhance its acceptability by electric utilities.

Demonstration Program

The demonstration project should be kept simple to insure high probability of success. The aboveground equipment should reflect first generation or state-of-the-art technology. Specifically, the rotating machinery should be modifications of commercial or nearly developed equipment. In the interest of simplicity, and to keep capital costs at a minimum, it might be desirable to omit the recuperator and high-pressure expansion turbine.

At least one, but hopefully three, compressed air power system demonstration projects should be pursued in which a mined rock cavern, aquifer, and salt cavity would be used for storage. Low-temperature air storage should be used. The initial mined-rock cavern should be hydraulically compensated, but eventually a dry rock cavern should also be demonstrated.

Since underground storage represents the largest unknown, especially in aquifer storage, design of demonstration plants should be preceded by intensive geological exploration. Site development work by the utilities and aboveground equipment development should not be finalized until questions of storability and deliverability are answered. Problems of carryover, both physical and chemical, in the air exhaust might affect aboveground plant design. Candidate storage volumes should also be tested for mechanical response to thermal and pressure cycling. This applies particularly to salt cavities.

Supportative Needs

In addition to the main-line tasks associated with the demonstration program, a variety of technology support tasks are essential to show that air storage systems have wide application in the US utility industry. For example, precious little information is available on specific siting opportunities. Salt beds have been fairly well mapped, but the individual formations have not been reviewed for their air storage potential. Similarly, aquifers are known to cover the entire central US, but the majority of these do not have the necessary characteristics. Also, hard rock exists almost everywhere, but less than 10 percent of the US has been surveyed for air storage potential. In most cases, the basic geological data needed for an air storage assessment exists, but a comprehensive geological review and evaluation of this data is needed to remove some of the uncertainty associated with siting.

A program to develop and verify mathematical models of underground storage should be instituted. These models should be concerned with underground fluid flows, thermal cycling, pressure cycling, the champagne effect, and, if possible, contaminant carryover. Verification of analytical models would permit design of future storage facilities and preliminary screening of alternative sites without extensive and costly field experiments.

High-pressure systems which promise attractive economic gains, especially for hard-rock storage systems, should be pursued. State-of-the-art industrial compressors and turbines manufactured in the US are limited to pressure ratios of 16:1. Presently commercial industrial compressors and expansion turbines could be used to provide the additional pressure ratio, but these machines are relatively small, inefficient, and costly. Therefore, advanced designs of high-pressure, high-temperature axial-flow rotating equipment should be pursued.

All power systems have some environmentally undesirable features. Compressed air systems appear to have less overall environmental impact than others, but several key environmental questions should be carefully reviewed. For example, compressed air systems would consume less than half the amount of fuel (for a given power output) than alternative fossil-fueled peaking stations, so air pollution should be substantially reduced. However, pollutant formation (especially oxides of nitrogen) is pressure dependent. Consequently, further investigation of emission characteristics of high-pressure combustors is advisable. Several environmental questions pertaining to geology should also be addressed. For example, air leakage from the cavern could have detrimental effects on surface vegetation and the behavior of geological faults. Although highly site specific, the land use question (e.g., surface reservoirs for compensated caverns and surface manifold systems for aquifer wells) should be investigated. All of these environmental issues deserve careful consideration to put them into proper perspective and to identify critical problems.

Since the competitive position of compressed air power systems is strongly dependent on utility-specific factors, a systematic review and analysis of major utilities and/or power pools should be made to estimate the prospective market and identify an economic incentive to stimulate industry support of the air storage concept.

Advanced Concepts

Last, but certainly not least, it must be recognized that for any new concept to make a significant contribution to power generation it must have long-term growth options and no major drawbacks. The conventional compressed air power system represents a practical compromise between low equipment cost, relatively small storage volume, and low fuel consumption. Even though this type of system relies on clean petroleum fuels which are in short supply, a substantial number of future installations are deemed likely for the foreseeable future. Since this reliance on petroleum is a potential long-term weakness of the concept, it seems appropriate to investigate advanced concepts designed to completely eliminate consumption of petroleum during the generation cycle.

A number of advanced concepts have been identified which are worthy of further investigation. For the most part, they involve substitution of coal, nuclear, or stored thermal energy for petroleum fuel. Examples include: a coal-fired fluid-bed air heater cycle, a continuous coal gasification cycle in which low-Btu fuel gas provides the energy for both off-peak compression and generation, a nuclear-fueled cycle in which reactor heat is diverted from the steam turbine to preheat air for greater power output, and a variety of thermal storage schemes.

INTRODUCTION

The load demands experienced by electric utilities undergo wide daily, weekly, and seasonal fluctuations. Because of these fluctuations, a very significant fraction of installed generating capacity is required to serve load demands of relatively short duration. During the past decade, new steam turbine and gas turbine units have been installed to provide peaking power. In addition, many utilities have been turning to energy storage as a means of leveling out load demand (see Fig. I-1). Pumped water storage is an economical and flexible energy storage method which has gained widespread acceptance, although some conflict has arisen recently over the land use for large pumped storage reservoirs and the need to construct transmission lines through remote, unspoiled countryside. Other energy storage schemes being researched today include high-energy storage batteries, electrolysis of water to produce hydrogen and oxygen which could be recombined in fuel cells, magnets, flywheels, and thermal storage.

An alternative energy storage system which could be attractive for future peak-power applications is a modified gas turbine power system utilizing underground storage of compressed air. This new power system is termed herein CAPS referring to Compressed Air Power System. The novel features of CAPS (see Fig. I-2) are the uncoupling of the compressor and turbine, so that they operate during different time intervals, and the incorporation of intermediate storage of the compressed air. During the low-load, off-peak periods the turbine clutch would be disengaged, the compressor clutch engaged, and the electric generator used as a motor to drive the compressor. The compressed air would be stored in a mined, underground cavern which could, if desired, be maintained at constant pressure by a water-filled shaft connected to a surface water reservoir (see Fig. I-3). During subsequent peak-load periods the air would be withdrawn from storage, mixed with fuel, burned, and expanded through the turbine to generate power. During this expansion process the compressor clutch would be disengaged and the entire output from the turbine would be available to drive the generator.

Since a major part of the gross power developed by the expansion turbine in a conventional gas turbine cycle is consumed to compress incoming air, uncoupling the compressor from the turbine as in CAPS would increase net turbine output by two to three times that obtainable from a conventional gas turbine with the same size components. Another advantage of CAPS stems from the use of off-peak electricity for compression. This power would presumably be generated from coal or nuclear power stations. Since 50 to 70 percent of the basic energy consumption would be for compression, this means that 50 to 70 percent of the basic energy requirement would be transferred from imported petroleum products to low-cost indigenous energy resources.

The CAPS concept was first conceived and patented during the 1940s. Since that time, the CAPS concept has been considered for use in Sweden (Ref. I-1), Great Britain (Ref. I-2), France (Ref. I-3), Finland (Ref. I-4), and other countries. Stal-Laval Turbin AB has continually revised and improved their CAPS designs based on the latest gas turbine technology (Ref. I-5). The first CAPS installation is scheduled for operation in mid-1977 (Ref. I-6). The plant will be rated at 290 MW and located in Huntorf, near Hamburg, Germany. It will be owned by Nordwestdeutsche Kraftwerke, A. G., a major West Germany utility. The turbomachinery equipment is being designed by Brown Boveri and Company, while the cavern is being created from salt domes by Kavernen Bau-und Betriebsgellshaft.

The United Technologies Research Center performed its first exploratory evaluation and preliminary design of CAPS for US applications in 1968. The updated results of that study, which are summarized in Ref. I-7, indicate that the air storage concept appears to be both technically and economically feasible in the US. Compressed air systems could be located almost anywhere that suitable rock can be found at the desired depth (see Fig. I-4) and would use only about one-tenth of the land surface that pumped water storage systems would use. The land requirement (and cost) could be reduced still further if a natural body of water (ocean, bay, lake, river, etc.) could be used to maintain hydrostatic pressure on the cavern. Air storage systems would consume less than half the amount of fuel consumed by alternative fossil-fueled peaking stations resulting in a significant reduction in atmospheric pollution. In addition, the peak energy output from an air storage plant, per unit of off-peak pumping or compression energy consumed, could be approximately twice the output from a pumped water storage system. The round-trip heat rate (including the heat rate of the station generating the electric power for compression and the CAPS fossil heat rate) of an air storage plant would be somewhat higher than the heat rate for a conventional gas turbine engine but significantly less than that for a pumped water storage system.

Despite the potential advantages of CAPS and the extensive technical information prepared for European applications, considerable evaluation design and optimization remains to be done for US applications. Several electric utilities have considered installation of CAPS, but the concept has not matured to the point where it could be considered as a viable alternative to conventional power generation methods. Additional work is required to resolve uncertainties which still exist regarding station siting, cavern integrity, turbomachinery design, overall system optimization, and cost estimates.

As a result, two major federally-supported programs were commissioned to investigate the viability of CAPS for domestic applications. The Energy Research and Development Agency sponsored a study by the team of General Electric, Fenix and Scisson, and United Engineers and Constructors (Ref. I-8). That study dealt primarily with aquifer and hard rock storage, but an offshoot program (Ref. I-9) was concerned with salt storage. The second major program, reported herein, was supported by the National Science Foundation and directed by the Energy Research and Development Administration. Many other smaller scale feasibility studies have also been supported by private and government funds.

The purpose of the present evaluation program was to attempt to resolve the aforementioned uncertainties by conducting selected siting studies, conceptual system design and cost studies, and utility application studies. The siting studies involved establishing general siting criteria and conducting geological surveys of two selected populous regions of the US to identify prospective areas suitable for excavating underground air storage caverns in hard rock. The results of this phase are contained in Part II. The conceptual system design studies consisted of preliminary evaluations of the technical, economic, and environmental feasibility and dealt with critical problems and uncertainties such as cavern integrity, turbomachinery design, and cost. The results of this phase are contained in Parts III, IV, and V. The third phase covered utility application studies and concerned the competitive aspects of compressed air power systems relative to alternative peaking and energy storage systems. These results are presented in Part VI. Parts I through VI comprise the main technical report and are contained in Volume I. Supporting appendixes are contained in Volume II.

REFERENCES FOR PART I

I-1. Swedish Patent No. 138026, application made January 19, 1949.

I-2. Brown, F. H. S.: The Prospects for Alternative Methods of Generation of Electric Power: A Comprehensive Review. Combustion, May 1967.

I-3. Description of et Devis de Realisation de l'Usine Avant Project Rentabilite et Perspectives d'Avenir. July 7, 1967.

I-4. Keskinen, R., H. Haavisto, and J. E. Kilpelainen: Sub-Surface Power Plants - A Study Performed in Finland on the Pumped-Storage and Gas Turbine Sub-Surface Plants. U.N. Symposium on Hydro-Electric Pumped Storage Schemes, Athens, Greece, 1972, ATH/SYMP/EP/13.7.

I-5. Robertsson, H.: Stal-Laval's Activities in the Field of Compressed Air Energy Storage. Proceedings of the Workshop on Compressed Air Energy Storage System, Airlie House, Virginia, December 1975.

I-6. Mattick, W., O. Weber, Z. S. Stys, and H. Haddenhorst: Huntorf - The World's First 290-MW Gas Turbine Air Storage Peaking Plant. American Power Conference, Chicago, Illinois, April 1975.

I-7. Giramonti, A. J. and R. D. Lessard: Exploratory Evaluation of Compressed Air Storage Peak-Power Systems. Energy Sources, Vol. I, No. 3, 1974.

I-8. Bush, J. B., Jr., et al.: Economic and Technical Feasibility Study of Compressed Air Storage. General Electric Report to the Energy Research and Development Administration, ERDA 76-76, Contract E(11-1)-2559, March 1976.

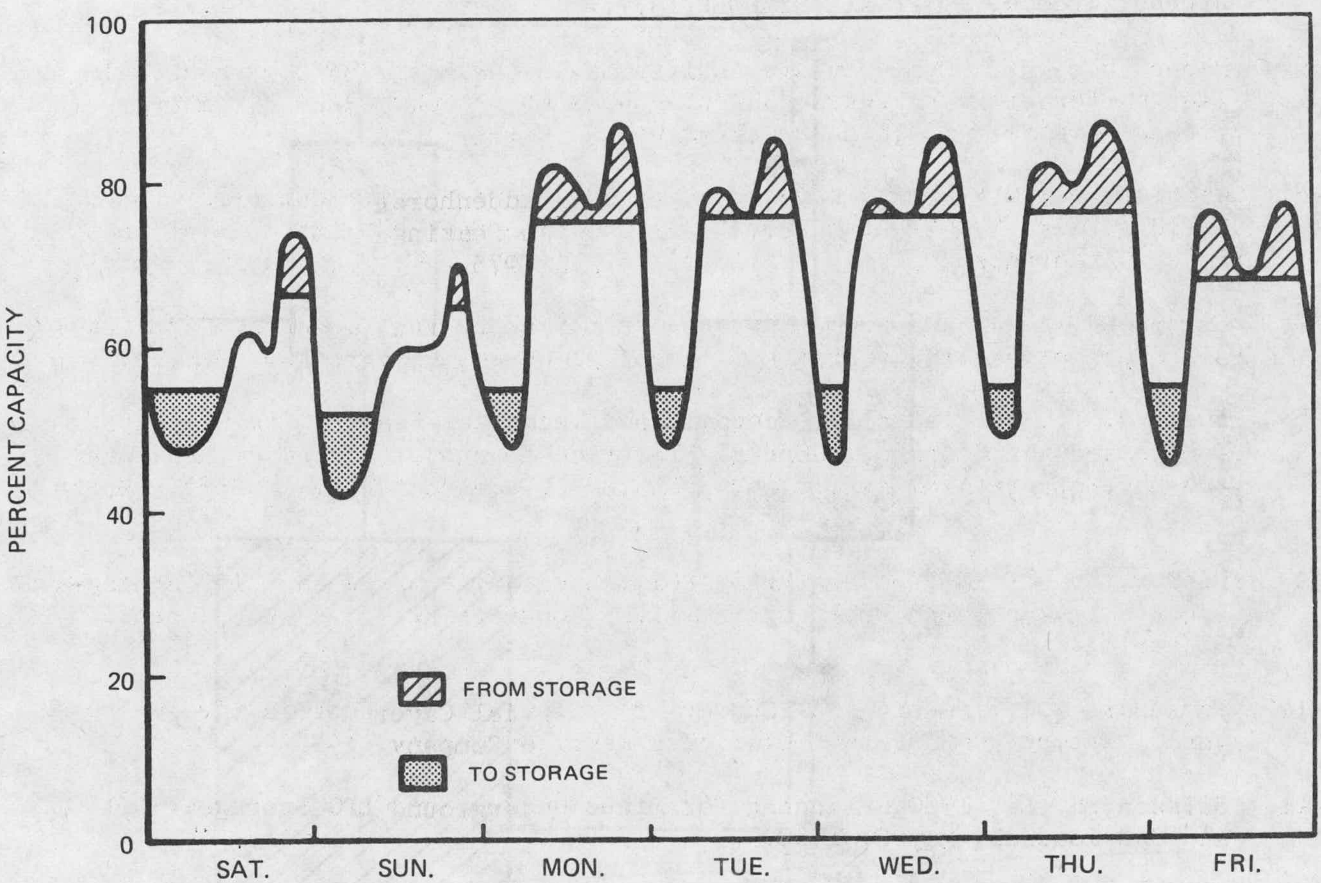
I-9. Design for a Pilot/Demonstration Compressed Air Storage Facility Employing a Solution-Mined Salt Cavern. EPRI Contract RP 737-1 with General Electric.

I-10. Farquhar, O.C.: Geological Survey of Potential Cavern Areas in New England. Report to Northeast Utilities Service Company, 1974.

I-11. Scisson, S. E., 1960, Planning for Mined Underground LPG Storage. Oil and Gas Journal, May 2, 1960.

I-12. American Gas Association, 1973, The Underground Storage of Gas in the United States and Canada. Twenty-Third Annual Report of Statistics Committee on Underground Storage. Arlington, Virginia.

WEEKLY ELECTRIC UTILITY LOAD CURVE



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SIMPLIFIED COMPRESSED AIR POWER SYSTEM

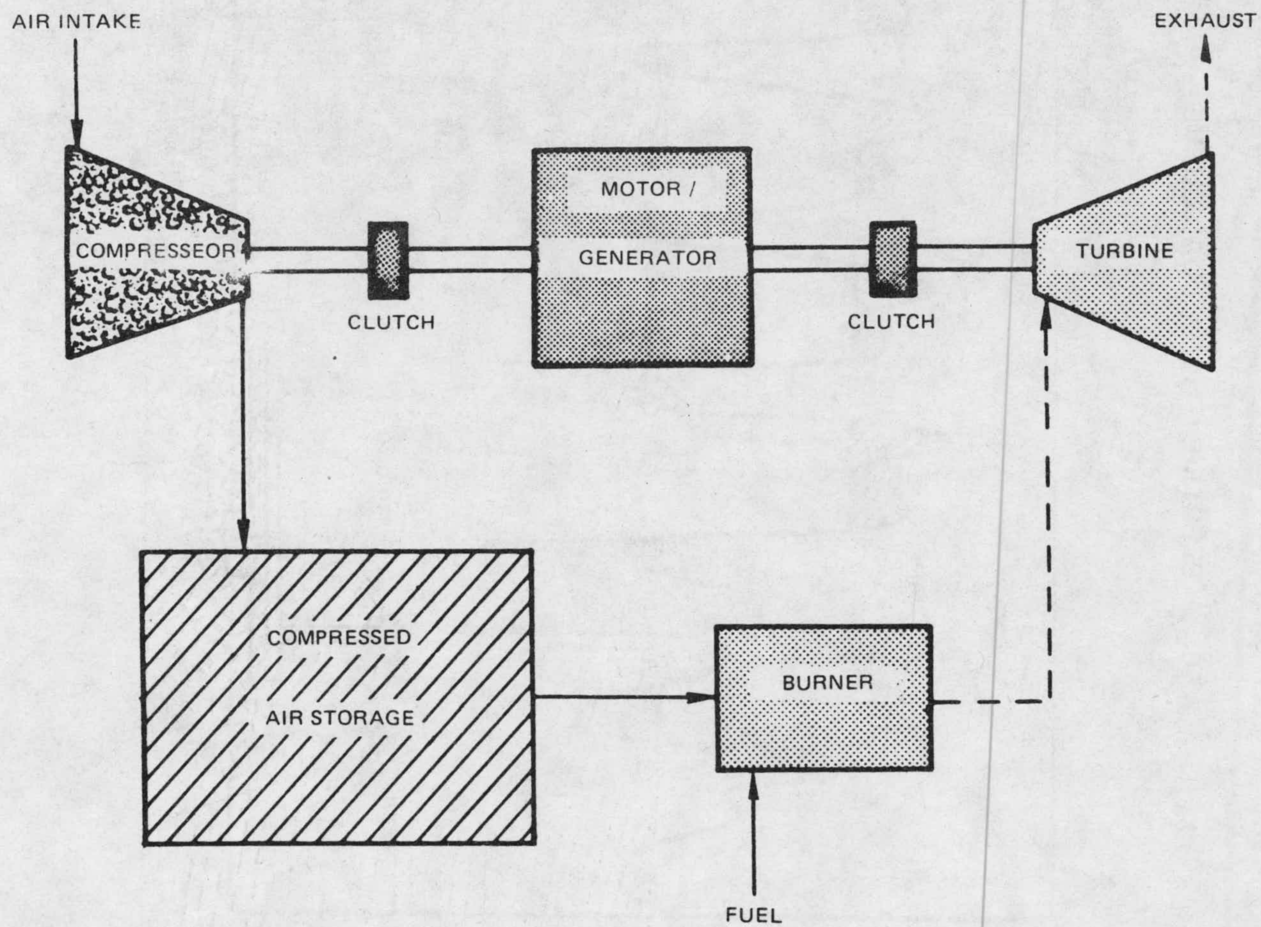

FIG. I–2

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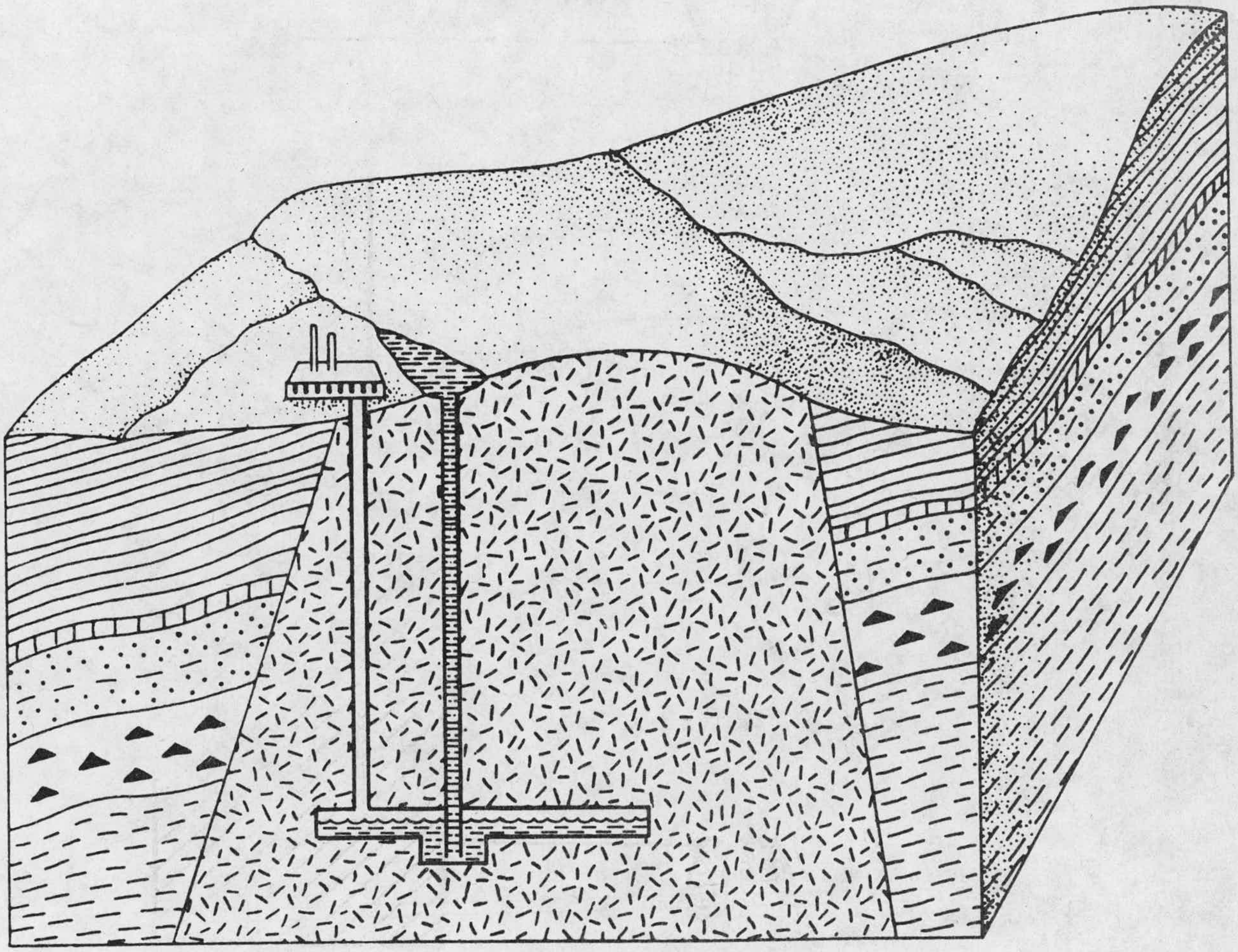
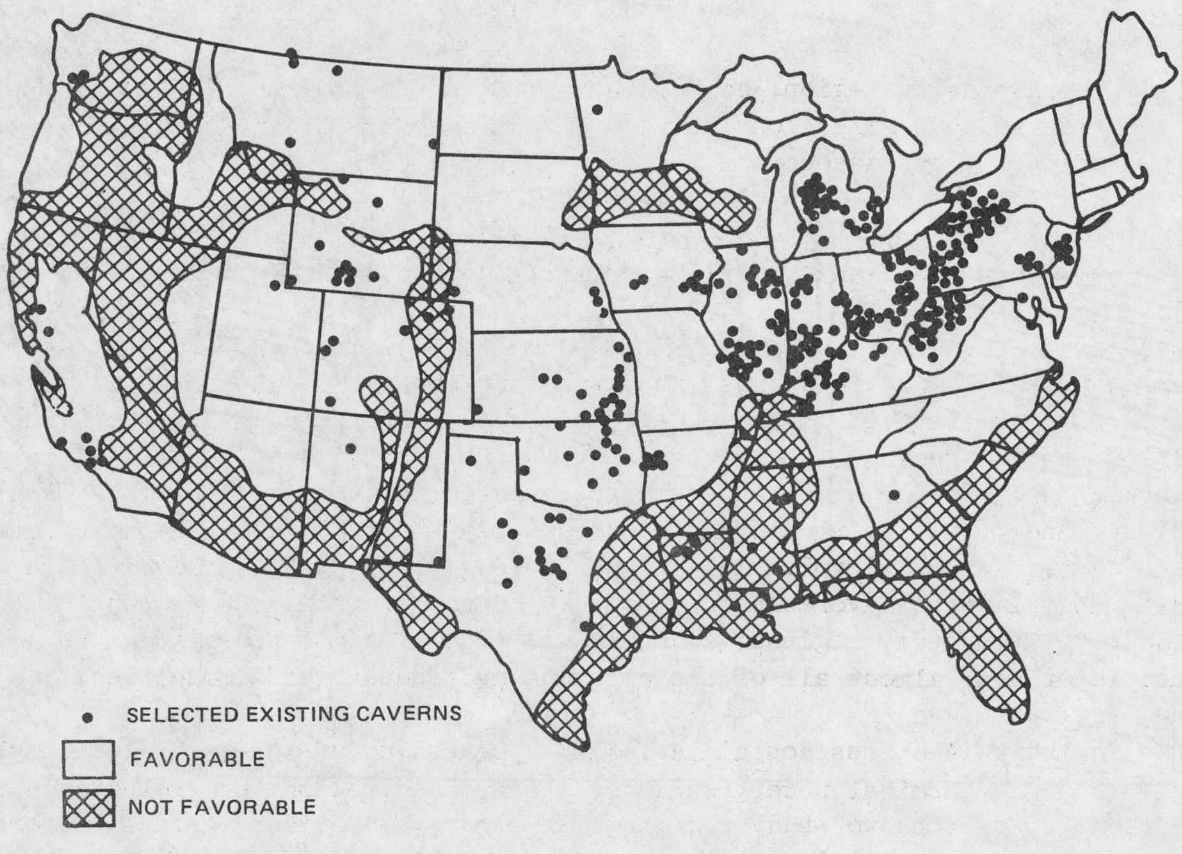
HARD ROCK STORAGE CAVERN WITH HYDROSTATIC COMPENSATION

(FROM REF. I–10, WITH MODIFICATIONS)



PROSPECTIVE AREAS FOR HARD ROCK CAVERN STORAGE

(FROM REFS. I–11 AND I–12)



PART II

GEOLOGICAL SURVEYS

Several of the rock formations in the US are potentially suitable for underground compressed air storage. In particular, the carbonate rocks (e.g., limestones and dolomites) in the sedimentary sequences, plutonic rocks (e.g., granites and grabbros), and some metamorphic rocks appear to offer the most advantageous geological and geotechnical conditions for storage. Although specific potential sites would need considerable investigation effort to establish feasibility, it is possible to make a generalized assessment of the geology region by region in order to establish potentially favorable areas within those regions. For this study, two specific populous regions of the US with different geological conditions have been selected for such an assessment. These are (see Fig. II-I):

(a) The northeast region, comprising a corridor approximately 100 miles wide and 400 miles long between Boston and Washington (underlain primarily by igneous and metamorphic rocks), and

(b) The north central region, comprising Illinois, Indiana, Ohio and Southern Michigan (underlain primarily by sedimentary rocks).

The general objective of this phase of the investigation was to consider whether enough geologic data exist to pursue the concept of cavern storage. Underground information is in three categories: 1) rock mechanics; 2) engineering and siting; and 3) geology and geophysics. The present state of knowledge in each category is adequate for major subsurface excavation. It seems clear that small areas where uncertainties remain can be recognized and avoided. The next level of investigation should involve local studies (e.g., deep drill-holes) within the areas underlain by favorable formations. There are numerous options throughout the two study regions, and suitable sites for air storage can be selected in or near almost all of the populous and industrial communities.

The following sections contain detailed discussions of the general siting criteria for an economical underground air storage cavern and the geological survey results for the two study regions. In order to provide a guideline for the geological surveys, siting criteria and parametric cost estimates covered in Parts II and III, preliminary assumptions were made regarding the general characteristics for a reference CAPS installation. The assumed characteristics for the reference plant design are given in Table II-1. These characteristics were modified, as a result of an overall system economic optimizations, for the final system design discussed in Parts IV and V.

GENERAL SITING CRITERIA

The process of location and assessment of general areas and, later, specific sites with a generally favorable geology will require consideration of three major elements of the CAPS plant - the air storage cavern, the shafts, and the upper (ground surface) reservoir. Of these elements, the geology surrounding the air storage cavern will have the greatest impact on the plant, with stringent requirements regarding air leakage and, to an extent, cavern stability. The geology of the soil and rock overburden in which the shafts must be sunk (or raised) will have an important effect in terms of costs, especially in the north central region, but in most cases it will not present problems regarding technical feasibility. For the upper reservoir, the topography will generally be more significant than the geology. However, the rather small land area requirements (in comparison with other man-made reservoirs) permit the economic utilization of artifical lining, thereby reducing the overall importance of the geographical aspect.

Air Storage Cavern Technical Criteria

The purpose of this section is to describe the technical criteria which should be applied to identify suitable sites and underground rock masses for underground storage of compressed air. The key technical criteria pertain to the geotechnical aspects of constructing the cavern, shafts, and upper reservoir. These criteria are incorporated into a numerical ranking method which is described in a subsequent section entitled Numerical Ranking Methodology. Items not included are: connecting costs to transmission line corridors, disposal or sale of excavated rock, and the costs of satisfying environmental constraints.

Air Leakage and Rock Permeability

The most important single aspect with regard to the geology of the air storage cavern is to select a rock mass of sufficiently low permeability that air leakage to the ground surface can be kept to an economically and ecologically tolerable minimum. This factor has been emphasized in available literature on the theory of air transport through rock. The importance of this factor was also made clear from information obtained during a visit to Scandinavia during the first six months of the study to collect and assess European cavern excavation technology. Because of the importance of the air leakage aspect, it has been identified as a "special requirement" and is dealt with in detail in Part III in the Section entitled Air Leakage . In summary, that section concludes that rock with an overall mass permeability in the order of 10^{-6} cm/sec (for water, roughly equal to 10^{-4} cm/sec for air) or less is necessary for a technically and economically feasible plant. Higher permeabilities than this could be encountered at specific fault or crush zones, if present, and these would have to be grouted to prevent the formation of leakage "pipes". Should the rock mass chosen for the plant have a permeability (for water) between 10^{-6} cm/sec and 10^{-4} cm/sec, it should be possible to use the site in combination with a water curtain to prevent air leakage. (Water curtains are discussed in a subsequent section entitled, Numerical Ranking Methodology and in Part III in the section entitled Air Leakage.) Although it is probable that areas of high permeability could be avoided by careful site selection, the state of the art of determining rock mass permeability at low values of permeability is not well advanced, and there is the possibility that, on opening the air storage caverns, the mass permeability would prove to be higher than geological and geotechnical investigations had indicated. Thus, the use of special sealing techniques such as pressurized water curtains and grouting might be necessary in local zones, although the general intent of the site selection process would be to minimize the need for such special measures.

The required condition of low permeability of the rock mass is most likely to be encountered in high-quality metamorphic or plutonic igneous rocks, or in a limited range of sedimentary rock types, such as shales or limestones and dolomites, which are free of solution cavities. The igneous and metamorphic rocks typify the most favorable available rocks in the northeast region and the limestones and dolomites in the north central region.

As part of the low air permeability requirement, it is essential that the chosen rock mass be saturated, that the groundwater table be near the ground surface, and that this condition not be drastically altered during the construction of the caverns.

Cavern Stability

During Excavation

The stability of the air storage cavern is mainly a question of the size of cavern which will be stable in a given rock with the minimum of reinforcement measures such as rock bolting and shotcreting. Keeping these measures to a minimum will have a significant effect on overall cavern costs.

In recent years there have been several attempts at classifying case histories and formalizing the overall approach to designing appropriate support measures. The most recent and comprehensive classification is due to Barton (Ref. II-1) which provides a convenient framework for estimating the type of reinforcement measures which will be necessary, based on preliminary geological interpretations. Parameters which are important in determining the overall rock mass quality, Q , which in turn determines the necessary reinforcement measures and costs, include:

(i) Rock Quality Designation (RQD) - A measure of the extent of fracturing in drill core

(ii) The number of joint sets (systematic discontinuities)

(iii) The nature of the joint surfaces (rough or smooth, clay filled or unaltered, etc.)

(iv) Permeability

(v) Rock strength and in-situ stress conditions.

These parameters are dealt with in considerable detail in Barton's paper, as is the selection of appropriate reinforcement measures. The paper is based on some 200 case histories and is in good agreement with experience in underground hydroelectric and other facilities.

With the exception of shales, all of the rock types cited as favorable for minimizing air leakages will also be of favorable strength and structure for cavern stability. Based on Barton's approach, the minimum rock mass quality, Q, for a limestone or dolomite would be in the order of 15 (RQD 90, three joint sets, rough or irregular, planar joints, unaltered joint walls, minor water inflow, medium stress). This would classify the rock as "good", needing systematic bolting, and allowing a maximum span of about 50 feet. In igneous and metamorphic rocks the rock mass quality would probably increase to one hundred or more, placing the rock in the "very good" to "exceptional" categories, requiring spot bolting only and allowing spans of approximately 80 feet.

Morfeldt (Ref. II-2) quotes typical dimensions for oil storage caverns built in Scandinavia at depths generally in the order of 300 feet. For good quality igneous and metamorphic rocks, spans have been in the range of 65 to 80 feet and heights up to 115 feet. Where such rocks are schistose or have other important structural controls, spans and heights have been reduced to 60 and 80 feet, respectively. In sedimentary limestone, the spans have been reduced to 25 to 40 feet and the heights to 50 to 65 feet with favorable bedding plane dips. These dimensions can be considered typical of economic cross sections for excavations in Scandinavia and should be directly applicable in North America, where excavation technology is essentially similar.

Shales, however, rank very low on rock mass quality (Q less than one), requiring considerable reinforcement measures to sustain even relatively modest spans. This leads to a high cost per unit volume excavated, adding a large economic penalty. Shales should therefore be discounted as direct air storage containment if other, more economic alternatives exist, but may merit consideration as an impermeable cap rock to a limestone or dolomite containment, a situation which could well arise in the north central region.

During Operation

The nature of a constant pressure air storage peaking plant is such that the storage cavern will be subject to cyclic variations (generally on a 24-hour basis) of the position of the air/water interface, air and water temperature, air humidity, and dissolved air concentration in the water. In addition, the cavern excavated at atmospheric pressure will ultimately be pressurized to 50 atmospheres or more.

For shales, the economic penalty due to the requirement to use relatively small spans and major support installation has already been mentioned. In addition, the cyclic wetting and drying of a shale containment during operation, with associated temperature variations might lead to disintegration of the shale with time.

As far as the other rocks cited are concerned, the only serious geological/geotechnical problems the above conditions could pose are the temperature effects. These effects are negligible provided that the air temperature is reduced to about 120°F before introduction to the cavern. This topic is covered in more detail in the Part III section entitled Temperature Effects . Possible solutioning of limestone will not be of significance, provided that the balancing water is not rich in CO_2 or acidic for other reasons. This is not likely to occur. If the same water is recycled from upper to lower storage levels, it would tend to become saturated with dissolved carbonates and become progressively less aggressive to the limestone.

Shaft Access Technical Criteria

The depth of the air storage cavern below ground surface (1500-2500 feet) is such that a vertical shaft access with high-speed hoisting would be more economical than an inclined access tunnel using diesel trucks to remove the excavated rock. Using a single 16 ft diameter access shaft, it is possible to remove all the excavated rock within 6 months to 1 year. In addition to the access shaft at least one additional shaft about 6 feet in diameter is required for ventilation and emergency access during construction, and for the air or water column in the plant operational mode. The air-pressure shaft could be in the form of a steel pipe within the main access shaft. Alternatively, the air-pressure shaft could be separate, with the access shaft converted to the water-pressure shaft.

It can therefore be assumed that two shafts will be required; one 16 ft and one 6 ft in diameter. The logistics of the excavation operation will be such that the 16 ft shaft must be sunk and the 6 ft shaft could be sunk or raised. It is also possible that the access shaft could be significantly smaller and still allow a reasonable construction schedule.

At possible sites within the northeast region, it can be assumed that the shafts will be wholly within a hard igneous or metamorphic rock, except for relatively minor depths of surficial soil overburden. Under these conditions, only minor water in-flow problems should be encountered and the thickness of the shaft lining could be kept to a minimum. Drilling, blasting and time-related costs might be relatively high due to the hardness of the rock, but grouting costs would probably be very low.

The sites within the north central region will be mostly in rocks of sedimentary origin, and consequently the variation in shaft costs could be considerable. Under good conditions, drilling and blasting costs might be substantially less than in the igneous and metamorphic rocks, due to the generally softer nature of the sedimentary rocks. However, there could be much higher costs for the grouting and lining operations, particularly within a major sandstone aquifer, and under such conditions time-related costs could be substantially higher than in a harder dry rock.

Upper Reservoir Technical Criteria

The upper reservior is required to maintain the air storage cavern under constant pressure as air is extracted. A volume of water is required daily, approximately equal to the volume of stored air. It can be assumed that this volume is in the order of 250,000 cu yards for the reference installation (Table II-1).

The upper reservoir could be the ocean, an existing lake or reservoir, a new reservoir or a river. In the case of the ocean there is obviously an abundant source of water and the only potential technical problem is the acceptability of salt water within the plant system.

An existing lake or reservoir would offer an attractive source of supply provided that possible environmental and aestetic problems can be resolved. The main technical problem would be the fluctuation of water level and its effect on the lakeshore and on existing usage of the lake. For a body of water with a 1-sq mi surface area (640 acres), the fluctuation would be about 3 in., which would be negligible compared with wave action and natural changes of water level. A smaller body of water would obviously be subject to greater fluctuation. This is a rather site-specific problem.

If a new reservoir were to be created behind a single small dam, the surface area would depend on the topography, i.e., the steepness of valley sides and slope of the valley; if built on flat terrain, a ring dike would be required. Assuming a maximum water level fluctuation of 20 ft, the reservoir area would be about 12 acres, including the dike. The volume of rock fill from the air storage cavern excavatior would be more than adequate for the body of such a ring dike.

Sealing the dam or ring dike could be achieved with impervious natural fill and filter zones within the rock fill, or by means of asphaltic or HDPE (polyethylene) lining. If the surface soils were favorable, only the rockfill dikes would need to be made impermeable. If foundation conditions were unfavorably permeable, the complete reservoir would need to be lined.

The use of a river as a water source without reservoir storage is feasible provided that the fluctuation in the river level and flow rate caused by the daily pumping cycle can be tolerated. The use of potentially polluted river water may pose other environmental problems.

Numerical Ranking Methodology

In concert with the search for geologically favorable locations for siting, it is desirable to numerically rank the favorability of alternative sites. The preceeding discussion of technical criteria identified the most significant geological and geotechnical factors which were included in the ranking methodology. The ranking was given economic significance by assessing the probable incremental construction cost relative to a nominal base case, associated with each significant geological and geotechnical factor. For example, the variation in rock strength and continuity will cause a variation in the cost of roof and wall reinforcement items such as rock bolts and chain link mesh. The incremental construction costs are expressed in terms of the incremental capital cost, in cents per kilowatt of generating capacity. The individual incremental capital costs are summarized in Table II-2. The sum of the incremental costs for each ranking factor leads to the final numerical ranking (in cents per kilowatt) for the site conditions considered. The lower this number, the more favorable will be the relative ranking.

This numerical ranking methodology was applied to the results of the geological survey of the north central region at the end of the next section entitled, North Central Region Survey. This method was not used for the northeast region because of the large number of rock formations in that region. Instead a more general ranking procedure was used.

Each ranking factor and the rationale behind establishing the incremental cost ranges are discussed in the following paragraphs.

Rock Treatment to Reduce Air Leakage

For the high-quality rock masses which are considered suitable for air containment, systematic grouting will not be required, other than for treatment of specific fault or crush zones. It has been assumed that in the worst case, bearing in mind the preselection of favorable sites, a major feature might occur every 1,000 ft and a minor feature every 100 ft, requiring expenditures of about \$100,000 and \$10,000, respectively, for remedial grouting of each feature. For this study the total length of tunnel is in the order of 1,500 ft, requiring a total expenditure of about \$300,000. The maximum cost penalty for the grouting would therefore be in the order of 107 cents per kilowatt. Including some allowance for schedule delays or other contractual disturbance caused by the grouting, a range of 0 to 120 cents per kilowatt has been allowed.

Water Curtain to Reduce Air Leakage

The cost of a water curtain required to prevent leakage from a 250,000 cu yd storage volume at a depth of 1,500 ft has been estimated, making the following assumptions:

- The air storage consists of three parallel caverns 60 ft wide and 500 ft long, spaced at 180 ft from center to center. Directly above these caverns run three parallel water-pressure galleries from which approximately horizontal holes are drilled at right angles to the cavern axes. The galleries are assumed to be approximately 8 ft square, and the drill holes spaced at 12 ft centers. The holes extend completely across the plan area of the caverns and extend 90 ft beyond the center line of the outer caverns.

- The three galleries are brought together near the access shaft, and water pressure is applied to the galleries and the drill holes through a steel pipe running up the access shaft to the ground surface. Pressure is provided by a water tower approximately 70 ft high, of sufficient capacity to contain a one-day supply, assuming a rock-mass permeability of 10^{-5} cm/sec (water). The tower is filled from the surface reservoir by means of two pumps - one active, one on standby.

The total cost of this system has been estimated to be \$547,000 or 195 cents per kilowatt. The cost would not be significantly greater for a rock with a permeability as high as 10^{-4} cm/sec, provided that a reduced water tower storage time could be tolerated. The running costs of the system will be very low in comparison to other running costs of the plant. For a rock mass permeability of 10^{-5} cm/sec, the total quantity of water pumped into the curtain is approximately 0.1 percent of the air storage volume per day.

A lump-sum allowance of 200 cents per kilowatt has been made to be applied to reservoir sites housed within rock units which have known high permeability values, or for which there is substantial uncertainty concerning permeability due to lack of data. The occurrence of an impermeable capping layer such as shale relatively close to the storage caverns would be sufficient reason at this stage to exclude the requirement for a water curtain.

Roof and Wall Reinforcement

Spot bolting would be required only in the high-quality, igneous and metamorphic rocks (rock mass quality, Q, greater than one hundred, see previous section entitled Air Storage Cavern Technical Criteria). This could be conservatively translated into untensioned grouted bolts (1-inch diameter), 15 ft long on a 10-ft by 10-ft pattern in the cavern roof only. The total cost of this "minimum" support would be approximately $250,000.

Recalling that careful preselection of sites is assumed, then the worst rock conditions that the caverns could be built in might approximate to a rock mass quality of ten. For this condition, the quantity of bolting is assumed to increase to tensioned grouted bolts 1-3/8 in. in diameter, 20 ft long on a 5-ft by 5-ft grid in the cavern roof, and on a 7-ft by 7-ft grid in the upper two thirds of the walls. In addition, chain link mesh would be fixed in the roof only. The total cost of this "maximum" support would be $3,150,000. From minimum to maximum support the total cost differential is $2,900,000 or 1,040 cents per kilowatt. Allowing for schedule delays, the range of incremental capital costs is taken to be 0 to 1,150 cents per kilowatt.

Cavern Excavation

The excavation cost for the cavern including labor, plant and materials for drilling, blasting and mucking to ground surface, excluding grouting with support costs, is unlikely to vary significantly from site to site within the rocks specified herein as favorable for air storage. Recently, in Scandinavia, the total costs for excavation of oil storage caverns of similar geometry in favorable geology, with comparable labor and plant unit costs, were in the order of $10 to $15 per cu yd. Very little data are available on recent costs for similar caverns excavated in North America. However, based on escalation of previous North American projects in conjunction with recent analyses of similar, large-scale excavations for pumped water storage, it is unlikely that the cost would be in excess of $15 to $20 per cu yd, excluding the cost of roof and wall reinforcement. This gives a cost of 1,350 to 1,800 cents per kilowatt, which is obviously a very large cost in comparison with the other variables. Small percentage variations in this cost could be important in the geological ranking. At this stage it is not feasible to estimate this possible variation. Detailed site exploration would be required, as well as detailed assessment of such variables as labor cost and availability within a specific area, etc. The excavation costs are considered in more detail in the Part III section entitled Cavern Layout and Costs. Excavation techniques are considered in more detail in the Part III section entitled Cavern Excavation Methods.

Shafts

For the case of a 16 ft diameter shaft sunk by conventional techniques to a depth of 1,600 ft, it has been estimated that the following approximate costs would apply.

- In a high-quality igneous or metamorphic rock to the full depth, the total cost inclusive of mobilization, demobilization, excavation and lining will be approximately $2,800,000.

- In a limestone/shale/sandstone sedimentary sequence with no unusually bad groundwater problems, the total cost will be approximately $3,400,000. If approximately 100 ft of very poor-quality overburden (saturated silts, loose sands, soft clays, etc.) have to be penetrated, the latter cost will increase to $3,500,000.

- If, in addition to the above conditions in the sedimentary sequence, it proves necessary to penetrate a major sandstone aquifer (for example, 200 ft thick at a depth of 500 ft), the total cost will be approximately $3,850,000. It is unlikely that conditions worse than this will be selected.

The range in cost for conditions is therefore $3,850,000 - $2,800,000 = $1,050,000. For a facility of 280,000 kilowatt generation capacity, this cost range is equivalent to 0 to 375 cents per kilowatt.

For a 6 ft diameter drilled shaft, the costs for shafts drilled wholly within granite, or wholly within sedimentary rocks have been respectively estimated to be $600,000 and $400,000 respectively. These figures make no allowance for water control or lining costs, which are likely to be more for the sedimentary rocks. This will tend to narrow the differential, effectively diminishing the cost index range to small value.

The total cost of a raise-bored shaft, 7 to 9 ft in diameter, through sedimentary rocks and without a major aquifer to penetrate, has been estimated to be approximately $1,050,000, inclusive of water control and lining costs. This is comparable to the costs estimated for the drilled shaft of similar diameter.

The smaller shafts, therefore, will have much less impact in terms of cost variation and so an allowance for a range of 0 to 400 cents per kilowatt (Table II-2) is considered adequate to cover all costs of varying the shaft conditions from site to site.

Surface Reservoir

The maximum cost of providing a surface reservoir would result from having to build a ring dike (with a complete lining) on permeable soils. For a 12-acre reservoir, 20 ft deep, the cost of the lining and preparation would be in the order of $750,000; placing the rock fill dikes would be $200,000. The combined total of $950,000 results in a unit cost in the order of 350 cents per kilowatt.

The minimum cost might be represented by the use of an existing reservoir or lake, which in comparison with providing ring dikes and lining would be very low. The ancillary civil works connected with the surface reservoir can be assumed to be constant for all types of water source within the sensitivity of these ranking figures.

Therefore, for the purpose of site ranking, allow an incremental cost range of 0 to 350 cents per kilowatt.

NORTH CENTRAL REGION SURVEY

The North Central region (see Fig. II-1) comprises the states of Illinois, Indiana, Ohio, and the southern end of Michigan. This region is underlain with sedimentary rocks which are almost three miles thick at some points. The essential requirements for an economical excavated cavity are low permeability, good minability, and permanent stability. The operating conditions which would have the greatest influence on the choice of suitable rock types are the relatively minor cyclic variations in pressure, temperature, relative humidity of air within the chamber, and the level of the water in contact with the rock walls. Rocks which are sensitive to such conditions should be avoided.

For example, bentonitic clays or shales are subject to swelling and shrinking tendencies in environments where temperature, ambient pressure, and relative humidity are not constant. The evaporites (rock salt, gypsum, and anhydrite) are soluble enough to cause concern and should be avoided in the chamber itself as well as in any areas where the existing groundwater regime might be changed by the installation of a water curtain. However, the carbonates (limestone and dolomite) are effectively insoluble within the service life of the structures. Beds rich in combustible organic materials (such as coal and oil shale) must also be avoided in the host rock of the cavity, although they can be present in the overlying strata through which the conduits will pass, since these are to be lined. Minor amounts of disseminate sulfides, which commonly occur in shales and carbonate sedimentary rocks, can be tolerated in spite of their low resistance to oxidation, provided that the quantities are so small as not to provide a source of appreciable amounts of acid solutions. Shales should be avoided in the first instance because they commonly lose physical strength during cyclic wetting and drying, and special tests are required to determine whether this phenomenon will occur in a particular shale stratum.

The requirement for structural stability at minimum cost dictates the avoidance of thin-bedded formations, at least in the immediate environment of the chamber. Areas where the rocks are likely to contain more than the average abundance of fractures transverse to the bedding should also be avoided in the first instance. Abundant fractures are often found in the vicinity of faults and of sharp variations in attitude of the strata, where such flexures have been caused by deformation subsequent to the time when the original soft sediments hardened into brittle rocks. In the north central region it appears that faults and secondary deformation have affected several small areas, but these are of such limited extent that they can be avoided when choosing the precise setting for a particular chamber.

The presence of oil, gas, or oil-field brines may inhibit the construction of a chamber and the necessary vertical conduits. The difficulties may be due either to the dangers and costs associated with their presence or to prohibitive administrative or legal problems, and such areas should be avoided. In the first instance it can be assumed that all known oil and gas fields are suspect, even though the individual producing pools may not be continuous under the full surface area of the field. Even beyond the limits of known fields, small amounts of natural gas and oil might be present in the rocks through which the shafts would be driven or the chamber excavated. It is assumed that such sites would be detected by exploratory drilling performed for the investigation of a particular proposed site. In the present study, only areas of known oil and gas fields have been eliminated.

Many of the deep groundwaters of the region contain various dissolved materials, including hydrogen sulfide, which is released from solution in the form of a noxious and potentially dangerous gas. In sufficient concentration in a working space, this gas can be poisonous or inflammable. It can also cause corrosion of metals and may oxidize to yield acidic sulfate solutions which can cause deterioration of concrete or solution of carbonate rocks. The presence or absense of significant concentrations of hydrogen sulfide in groundwater can be determined by sampling in boreholes during the investigations of individual potential structures.

Because of the layered strata in this region, it is necessary to focus the geological survey on the approximate depth below ground surface which seems most appropriate for air storage caverns. This situation is quite different from the geology in the northeast where single formations frequently extend from near the surface to thousands of feet below the surface. Early CAPS studies indicated that the 1500-2000 ft depth would be of most interest. To provide a preliminary design basis for the present study, a reference system was selected which would require a cavern at approximately 1600 ft (corresponding to an overall pressure ratio of almost 50, as noted in Table II-1.) Later studies extended the depth to the 2200-2400 ft range to correspond with the storage pressure selected in Part V. Most of the subsequent discussions for the north central region are keyed to the 1500-1600 ft and 2200-2400 ft ranges.

General Geologic Framework

The sedimentary rocks in the north central region were deposited during a very long period of time under conditions which ranged from deep sea immersion to arid desert. Successive changes occurred in sea level and depositional environment, and there were also long periods of erosion. As a result, the beds display lateral changes in their physical properties (facies changes), and it is only with some difficulty that contemporaneously formed strata in different parts of the region can be correlated. In the vertical sense (transverse to the bedding), even more abrupt and numerous variations occur.

These beds were originally deposited in the form of muds and sands which were progressively buried under increasing thicknesses of sediment. They were compressed and lithified during physicochemical adjustment to this new environment. The more ancient surface on which they were deposited was a rolling landscape which had been developed by erosion on top of the rocks of Precambrian age. These Precambrian rocks are now collectively referred to as the basement, and the ancient land surface upon which the younger sediments were deposited is the most prominent unconformity in the stratigraphic column.

The basement surface consisted of broad ridges and basins which can still be detected by drilling and geophysical investigation of this unconformity. Figure II-2 shows the present general configuration of that surface and some of the names which are applied to its larger features. The Cincinnati arch system in Indiana and western Ohio was a positive topographic feature before the beginning of deposition of the sediments and it remained a relatively elevated area while most of the sediments were being deposited and lithified. Many of the formations are thinner where they pass over the top of this rise, and some are discontinuous and occur in only one of the flanking basins. A representative cross section taken through Springfield, Indianapolis, and Dayton (Fig. II-3) depicts the major carbonate strata and basement surface.

On the upper surface of the thick sedimentary rock sequence is a relatively thin veneer of soils. These include many different types, primarily of glacial origin but ranging in origin from lacustrine to aeolian and in grain size from clays to gravels (Ref. II-3).

The relevant characteristics of the various rock formations are described below.

Stratigraphic Section

In describing the geologic conditions in the North Central region, reference will be made to the broad features shown in Fig. II-2, which includes parts of three major basin-shaped structures. In general, the favorable rock formations occur at appropriate elevation on the flanks of these basins, since in the central portions they are too deep to be of interest in this application. The rocks are described in groupings based on rock types (i. e., lithologic units), rather than on the basis of time-stratigraphic units. Therefore, the designations do not necessarily correspond with groupings developed for other purposes. In most cases the terminology adopted by the American Association of Petroleum Geologists (Refs. II-4 and II-5) has been used to designate the larger rock units which can be traced over long distances.

The oldest rocks (Precambrian) are described first, followed by the groups of Paleozoic age, from oldest to youngest.

Precambrian

Since the Precambrian cannot be expected to produce oil and gas, relatively few holes have been drilled into it. However, there are sufficient data, both from drilling and from geophysical surveys, to allow the generalized contours in Figure

II-2 to be drawn. The upper surface of the basement varies in elevation from -1,500 feet (with respect to sea level) at the Illinois-Wisconsin border to below -12,000 feet in the central part of the Illinois Basin and in eastern Ohio. The basement is thus too deep almost everywhere to be a candidate in the context of the present study.

The geologic conditions within the basement appear to vary considerably from place to place (Refs. II-6 through II-10). Samples from deep drill holes are of many different rock types, including metamorphic amphibolite, hornfels and marble, as well as granite and biotite gneiss which may be of metamorphic or igneous origin. West of the Cincinnati Arch most of the samples are of granitic and low-grade metamorphic rocks about 1.2 to 1.4 billion years old (Ref. II-4), which probably correspond to the Superior province of the exposed Precambrian in the Canadian Shield. To the east of the Cincinnati Arch the rocks are somewhat younger and include plutonic igneous and high-grade metamorphic types, similar to those of the Grenville Province of the Shield.

Cambrian

The rocks of the Cambrian period form the oldest and lowest of the sedimentary strata of the Paleozoic era. The term "Potsdam Sandstone" is used here to designate the predominantly sandstone formations which overlie the Precambrian basement, including (among others) the Mount Simon and Eau Claire of Indiana, Ohio, Michigan and Illinois; the Franconia of Indiana, Ohio and Michigan, and the Jacobsville and Dresbach of Michigan (Refs. II-5 and II-11). The upper part of the Potsdam includes shaly and dolomitic-calcareous units and becomes predominantly dolomite in the southern part of the Illinois basin.

The thickness of the Potsdam is quite variable. In northern Illinois as much as 3,000 feet of sedimentary rock is included, most of it belonging to the Mount Simon sandstone. In general, however, individual beds are thin and there are likely to be many permeable and porous strata rendering much of the volumes of Potsdam rocks relatively unsuitable for economical excavation of large chambers for storage of compressed air.

Cambrian-Ordovician

The early Ordovician or late Cambrian (Ref. II-10) Canadian Series (including the Beekmantown Dolomite of Ohio) and the uppermost Cambrian formations are grouped under the designation of Knox Dolomite. In addition to dolomites, some sandstones are present, as in the New Richmond Formation of the Prairie du Chien Group of Michigan and Illinois, and the Rose Run in Ohio. There are small percentages of chert and shale and little or no limestone (except for the basal Knox Dolomite in a single well in Ohio (Ref. II-9). The shale is commonly present as thin partings.

The thickness of the Knox Dolomite varies progressively from as much as 7,000 feet in the southern Illinois basin to as little as 500 feet along the Michigan-Ohio border (Ref. II-5). This thinning in the northerly and easterly directions is due primarily to progressive truncation of the upper portions by an old erosion surface (an unconformity), as well as to progressive thinning of individual units in that direction. A third factor is that the base of this unit is considered to be the bottom of the predominantly dolomitic rocks, so that a progressive change to a sandy facies causes a raising of the bottom boundary of the Knox on progressing northeastward.

The Knox Dolomite is known to be porous in some areas in Ohio. In Morrow County, the Knox porosity is associated with oil production (Ref. II-9). In general, however, the Knox Dolomite is associated with moderate to high electrical resistivity and sonic velocity (Ref. II-5), which suggests that these beds are relatively massive and dense and are more likely than the underlying Potsdam to provide opportunities for storage of compressed air in mined caverns. However, this must be considered unproven in view of the scarcity of data on the continuity of individual beds of proven lithologic and physical properties. For the purposes of the present study, the Knox Dolomite should be considered of second priority importance.

Middle Ordovician

The Ottawa Megagroup, as defined by Bond et al (Ref. II-5), includes the Trenton-Black River beds of Michigan, Indiana, and Ohio, the equivalent Platteville and Galena of Illinois, and the Eggleston of Ohio, among others. Because of its economic importance in the oil and gas industry and its moderate depth in most of the area of interest, the Ottawa has been penetrated by many drill holes and its extent and properties are relatively well known. At least one large series of caverns has been excavated in the Black River member near Cincinnati, by the Cincinnati Gas and Electric Company, for shallow underground storage of liquefied propane. The thickness, continuity, and properties of the carbonate formations of the Ottawa beds are such that many sites suitable for underground storage of compressed air can probably be found. Figures II-4 and II-5 show the locations at which these Middle Ordovician beds occur at depts of approximately 1500-1600 ft and 2200-2400 ft, respectively.

The Trenton-Black River are both composed almost entirely of carbonates, including both limestone and dolomite. In Ohio, the composition varies from primarily dolomite in the northwest to primarily limestone in the southwest and east. The Black River is the older and deeper of the two major carbonate members, and is generally thicker than the Trenton in those cases where the two can be separately identified in drill holes. The average thickness of the Black River in Ohio is more than 400 feet. Although primarily a fine-grained (lithographic) limestone, it contains shaly members near its base in some areas as well as several bentonite shale horizons in its uppermost member (equivalent to the Eggleston limestone of Ohio). The Trenton is commonly dense and fossiliferous to crystalline. In the oil and gas producing areas of northwestern Ohio it is porous (Ref. II-12.) In the area between the Cincinnati Arch and the Appalachian basin the upper portion of the Trenton contains thin, shaly partings, possibly related to a transition to the overlying Cynthiana shales. In the Chicago area, where considerable investigation has been done in connection with the deep sewer project (Ref. II-13), the Trenton is between 170 and 210 feet thick and is mostly dolomite, but with some shaly and calcareous portions.

Both the Black River and the Trenton are likely to exhibit very low permeabilities in areas of normal lithology outside of oil and gas fields.

The thickness of the whole Ottawa Megagroup varies within the north central region from more than 1,300 ft in the deeper part of the Illinois basin to somewhat less than 500 ft over portions of the Cincinnati Arch complex (Ref. II-5). It is probable that sufficiently massive beds of the required thickness (about 150 ft) can be found at almost any point. For example, a series of wells in Ohio described by Shearrow (Ref. II-12) gives the following information (depths below ground surface, in ft):

<u>Wood County (Plain Township)</u>

Trenton	1,115 to 1,225 (110 ft) dolomites and limestones
Black River	1,225 to 1,790 (565 ft) limestones- including
	1,475 to 1,790 (315 ft) almost pure lithographic limestone.

<u>Wyandot County (Crawford Township)</u>

Trenton	1,321 to 1,391 (70 ft) limestones
Black River	1,393 to 1,885 (492 ft) limestones - including
	1,500 to 1,800 (300 ft) almost pure lithographic limestone.

<u>Wyandot County (Antrim Township)</u>

Trenton	1,755 to k.825 (70 ft) limestones
Black River	1,825 to 2,335 (510 ft) limestones - including
	1,865 to 2,165 (300 ft) almost pure lithographic limestone.

<u>Marion County (Claridon Township)</u>

Trenton	2,073 to 2,136 (63 ft) limestones
Black River	2,136 to 2,655 (519 ft) limestones including
	2,345 to 2,655 (310 ft) almost pure lithographic to crystalline limestone.

For the purpose of this study, it is concluded that the Ottawa Megagroup should be considered to have good siting potential.

Ordovocian - Silurian

The remainder of the Ordovician, together with the lower Silurian rocks, is less promising. Although sandstones, shales, and carbonates are all present, calcareous shale is perhaps the most abundant single rock type. Some horizons in some areas are likely to be sound and durable, and potentially of use as hosts for large caverns. However, there are many lateral facies changes in these beds, the formations are generally thin, and it is not known whether massive beds with the required thickness are present over large areas. For lack of adequate data, therefore, the Upper Ordovician - Lower Silurian beds cannot be included at this time among those offering good siting opportunities.

Silurian - Devonian

Beginning with the Niagaran Series of Middle Silurian age, and extending upward through the Middle Devonian rocks, is a thick sequence containing many carbonate members. This is referred to by drillers in Ohio as "Niagaran Big Lime" or simply, "Big Lime". It is approximately equivalent to the Hunton Megagroup of Bond et al (Ref. II-5).

The Hunton attains a maximum thickness (Ref. II-5) of more than 1,800 feet in the deepest part of the Illinois basin, and thins progressively toward the north, northeast, and east. It reaches the surface (under glacial deposits) along parts of the Cincinnati, Kankakee, and Findlay arches and has been thinned or removed by erosion in such areas. The thickness along the southern border of Michigan ranges from 600 to over 1,000 feet.

Within the Hunton Megagroup there are several distinct formations or members at different stratigraphic horizons which offer opportunities for siting large underground cavities. Because of the complex history of deposition and erosion in the large area under consideration, the properties, compositions, and thicknesses of these formations vary from place to place, so that a formation which is suitable in Illinois may not be suitable where encountered on the east side of the Cincinnati Arch in the Appalachian basin. It will also usually be called by a different local name. However, it is very likely that one or other of the thick carbonate units within the Hunton will prove to be an acceptable host at most points where Silurian-Devonian carbonates occur at suitable depth, as shown in Figs. II-4 and II-5.

The lowest part of the Hunton is formed by the Lockport Dolomite of Ohio and its equivalents in other areas. The Lockport is a group of formations which can be separated into Gasport, Goat Island and Guelph Dolomite units when examined in the outcrop areas, although these distinctions cannot normally be made on the basis of evidence available from drill holes. The Lockport is the upper part of the Niagaran Series of Middle Silurian age and is the basal unit of the Big Lime, immediately overlying the Rochester Shale Formation.

In northeastern Ohio (Ref. II-14) the Lockport consists entirely of dolomite, with no limestone or anhydrite. However, near its lower contact with the underlying shale, there is locally an interbedded sequence of shale and dolomite, in some places as much as 80 ft thick. A crystalline dolomite horizon, called "Newburg Sand" by drillers, occurs locally and may be as much as 40 ft thick. Some fossil reef structures are present in Ohio, Indiana and Illinois. The thickness of the Lockport is quite variable, even within a relatively small area. In northeastern Ohio for example, it varies from 55 ft in Licking County to 521 feet in Morrow County (Ref. III-14). In the region immediately east and southeast of Cleveland, it is between 200 and 300 ft thick.

The Salina Group of the Cayugan Silurian Series is the next prominent unit overlying the Lockport Dolomite within the Big Lime. Of variable lithology and thickness, it is unlikely to contain beds favorable for the development of storage facilities. Although large volumes consist of dolomite, the beds are likely to be thin. There are many prominent evaporite units, including both salt (halite) (Ref. II-15) and anhydrite in different places, as well as shale and dolomitic or limey shale. The Salina Group is contained within the Hunton Megagroup.

Overlying the Salina (but still within the drillers' Big Lime) is a series of favorable formations which are primarily carbonates. In Ohio the names (Ref. II-16) of these formations, from oldest to youngest, are: Bass Islands Dolomite, Helderberg Limestone, Oriskany Sandstone, Bois Blanc Formation, Columbus Limestone and Delaware Limestone. Not all of these are present in any one section, and there are facies variations from place to place. The equivalents in other areas have been given other names, in most cases, but are similarly composed primarily of dolomite, with lesser amounts of sandstones, shales, etc. In Michigan, the Detroit River Group, which is near the top of this sequence, contains some salt and gypsum or anhydrite.

The major geological time break between the Silurian and Devonian systems can be detected in most of the area in the form of an unconformity which represents a period of erosion between the deposition of the Bass Islands Dolomite and Helderberg Limestone Formations (or their equivalents). A thin sandstone member often occurs here, and the underlying Bass Islands Formation is locally thinned or absent because of erosion. In eastern Ohio, however, the Silurian-Devonian contact is a transitional one (Ref. II-14), at least locally. In any case, this unconformity does not usually introduce a sharp enough discontinuity in the rock properties to be significant for present purposes. It appears to be equally likely that favorable rock can be found within a short distance either above or below this unconformity.

Although the overall thickness of the Hunton Megagroup is as much as 1,800 ft in part of the north central region, the average thickness is much less. Some data are quoted in Table II-3 for the Big Lime sequence extending south from the Cleveland area through central Ohio to the Ohio River (Figs. II-4 and II-5). Only one well has been chosen in each county, although more data are available (Ref. II-17). The progressive thinning in a southerly direction will be noted. This is contrary to the situation on the opposite (west) flank of the Cincinnati Arch, where the approximately equivalent Hunton Megagroup thickens toward the south.

Mississippian - Pennsylvanian and Younger

Above the top of the Big Lime (or approximately equivalent Hunton Megagroup) are the upper Devonian, Mississippian, Pennsylvanian, and post-Paleozoic rocks. In total, these represent a very long interval of time and many thousands of feet of sedimentation. Nevertheless, the probability of encountering suitable formations at the required depths is thought to be considerably lower than in the older rocks described previously. In the Illinois basin, the Upper Devonian and Lower Mississippian New Albany and Borden units are predominantly shale, with some sandstones. The equivalent beds in Ohio and Michigan are generally similar.

The next overlying beds in Illinois and Indiana include a considerable thickness of siliceous limestone with some anhydrite and dolomite. This sequence comprises the "Mississippi Lime" of the drillers' nomenclature. The approximately equivalent formations in Ohio and southern Michigan are primarily shales and some sandstones (including the economically important Berea Sandstone) with lesser amounts of limestone. It is possible that suitably massive beds are present within this Lower Mississippian sequence, although the available information is scanty. This portion of the stratigraphic column should therefore be considered as a second priority target.

Most of the remainder of the Mississippian and the bulk of the overlying Pennsylvanian sequences consist of alternating beds of shale and sandstone or limestone, with important coal beds in places. Although there are some substantial thicknesses of limestone locally, there appear to be a few or no carbonate formations which are likely to provide favorable conditions at suitable depth under large areas.

In most of the north central region, rocks younger than the Pennsylvanian were either not deposited or have been removed by erosion. However, there are local substantial thicknesses of the Permian (uppermost Paleozoic era) and Jurassic Systems. Where these occur, they are both primarily sandstones with some shales, evaporites, and coal beds.

Assessment of Survey Results

The preceding discussion shows that, on the basis of the evidence available at present, the most favorable rock conditions are likely to be found in the Trenton-Black River carbonates of Middle Ordovician age and the thick carbonate members of the Silurian-Devonian Big Lime and its equivalents. Figures II-4 and II-5 show approximately where these two major sequences can be expected to occur at about 1,500-1,600 ft and 2200-2400 ft depth in the north central region. Two sequences of second priority, the Knox Dolomite and the lower Mississippian carbonates, are not shown.

The information on which Figs. II-4 and II-5 are based comes from several sources in Refs. II-4, II-5, II-9, II-10, II-12, II-14, and II-17 through II-24. There are some differences between the compilations provided by the various authors. Probably the most important reason for this is that the authors have used different amounts of detailed data in compiling their maps and reports (i.e., different well spacings, different groupings of formations in any one well log). In the time since the oldest of these references (Ref. II-22) was originally compiled in 1920, a great deal of geological mapping has been done, many more wells have been drilled, more advanced methods of logging have been used, and more detailed subdivision of the subsurface stratigraphy has become feasible. Interpretation of correlation has evolved, as well as newer understanding of the relative importance of the various units for different purposes. It is assumed that any further work on an individual potential site would be preceded by a detailed review of individual wells in the vicinity. Well logs and samples are available for examination at the Geological Survey offices of the several states or, in some cases, at the State universities. Until such examination has been done for a particular site, Figs. II-4 and II-5 can be used as a general guide.

Excluded Areas

As was previously noted, there are certain conditions which, if present, might excessively increase the cost of construction or operation. Of particular importance are oil and natural gas under pressure in the host rock of the chamber or in the overlying strata through which the shafts must penetrate. In several areas, members of the Ottawa Megagroup, including the Trenton-Black River, are exploited for oil and gas production. In Illinois (Ref. II-25), many small and medium-sized oil pools are known in the Trenton, along with one gas pool (Ref. II-25). Virtually all the Ordovician oil and gas of Michigan is produced from the Mohawkian Series, which is the Trenton-Black River equivalent (Ref. II-5). The Black River produces oil in Ohio (Ref. II-9), as does the Trenton (Ref. II-9), and both are producers in Indiana (Ref. II-24). Important gas production comes from the Devonian of Ohio (Ref. II-26).

Figures II-4 and II-5 show the areas remaining after gas and oil producing fields have been excluded. The most important single area is the major part of the central Ohio band underlain at 1,500 to 1,600 feet by the Silurian-Devonian carbonates. This corresponds in part with the Devonian gas fields. However, the remaining portions of this band along the shore of Lake Erie (in the immediate vicinity of Cleveland) and in the southernmost counties of Vinton, Jackson and Scioto, appear to be free of such interference. Many other smaller areas not shown in Figs. II-4 and II-5 could also be found between or to the west of known pools in this band, particularly in Ashland and Knox counties. Some of the Trenton-Black River (Ordovician) of northwestern Ohio has also had to be eliminated, since oil is produced from it in some places. Account has also been taken of the occurrences of natural gas in Pleistocene deposits of glacial drift in central Illinois (Ref. II-27), since excavating to bedrock in such areas might be unnecessarily hazardous and expensive.

It must be understood that these procedures have not eliminated all potential problems which might be caused by the presence of natural hydrocarbons. Known producing areas have been excluded, but as yet undiscovered oil and gas resources no doubt exist at many points (Ref. II-5), and there are known to be many "shows" (or occurrences of oil and gas which cannot be exploited profitably for various reasons). Some indication of such conditions can be obtained by detailed review of the logs of any exploratory drilling which may have been done in the vicinity of a proposed underground storage development. Nevertheless, it will inevitably be essential to drill and obtain information on these and other geotechnical conditions on the precise site of any such proposal.

Similar comments apply to the evaluation of groundwater conditions. Most of the developed water supplies of the north central region are obtained from streams, rivers and lakes, and from shallow wells in overburden. Groundwater in deep rock formations has been exploited only locally. Therefore there has been little economic incentive to evaluate the groundwater potential at depths which would be relevant to this study, although some systematic work has been done (Refs. II-28 through II-30), at shallow depths. Oil exploration has shown that in many areas the water which is found at such depths is chemically unsuitable for most uses, so that the incentive for deep exploration and development of water will probably not be sufficient to produce much new evidence in the foreseeable future. Fortunately, it is unlikely that groundwater conditions could prove so unfavorable as to prohibit the development of a site. At worst, an expensive grouting or freezing operation would be required ahead of limited portions of the shaft to prevent excessive inflows, or lowering of the surrounding groundwater table and pore pressures during construction. A program of investigations would be required at each specific site to provide data from which such possibilities could be evaluated. At the same time, the required information on rock permeabilities and other geotechnical characteristics would be obtained.

Location of Suitable Units

A selection has been made of the rock bodies which are most likely to provide the conditions required for economical structures, judged on the criteria of relative freedom from bedding surfaces and other discontinuities, low permeability, and adequate physical properties of the intact rock. Eliminating further consideration of the basement rocks because of their excessive depth within the study area, the remaining most favorable known units are the Trenton-Black River carbonate beds and their equivalents of Middle Ordovician age and the Silurian-Devonian carbonates which form part of the Big Lime of Ohio and their equivalents in other states. Large areas of Illinois, Indiana and Ohio and small areas of southern Michigan are underlain by one or other of these bodies at depths suitable for air storage caverns. After eliminating from further consideration those parts of the region where oil or gas are known to be present, there still remain large volumes of suitable rock in each of the four states, including potential areas within 50 miles of all the major load centers. The results of this selection are summarized in Figs. II-4 and II-5.

Geological Ranking

The aforementioned areas are those judged, on the basis of present knowledge of the geological conditions, to have the greatest potential capability of meeting the criteria for economical storage of compressed air at suitable depth.

Assumptions

For any specific site which is later selected as desirable, it is assumed that adequate investigation by drilling and other methods will be undertaken, and that sites with exceptionally unfavorable conditions would be rejected a priori. Such unfavorable conditions might include:

- closely spaced and open jointing (fracturing) resulting in high permeability or questionable stability of the proposed chamber excavation;

- closely spaced and abundant bedding surfaces or shale partings near the proposed chamber ceiling level;

- evaporite beds, bentonite seams, or other impurities which would be sensitive to decay or alteration under ambient operating conditions if exposed in the chamber;

- combustible materials such as coal, oil, shale, or asphaltic compounds at chamber elevation; and

- appreciable amounts of natural gas or excessive amounts of groundwater, particularly if hydrogen sulfide is present, at or above the chamber level.

Provided that individual sites with these or similar unusually unfavorable conditions are rejected, it can be assumed that the cavern and shafts will be excavated in rocks which approximate the average conditions in each of the areas of first priority. These average conditions vary from one area to another, and the extent of deviation from them will also vary among individual sites. An assessment of the probable range of variation is required for the purpose of ranking the different areas with respect to the probable resulting cost of development.

The estimated ranges of incremental capital costs used to rank individual sites were summarized in Table II-2. For each criterion listed in the table, the probable range of expected incremental cost due to siting the cavern in less than optimum conditions is given. The choice of a zero incremental cost signifies that the conditions, with respect to that particular criterion, are expected to be as favorable at the chosen site as could reasonably be expected anywhere in the northeast or north central regions of the United States. In the case of the water curtain criterion, the statement of a unique cost value, rather than a range of values, signifies that essentially the total cost would be incurred by the provision of the minimum water curtain system. This is therefore an all-or-nothing item.

From a study of the north central geology, it has been concluded that none of the favored areas would be likely to require a water curtain, as a relatively impervious shale cap rock can be expected to occur within a short distance above all of them. It is judged that the cost of roof and wall reinforcement will, in general, be somewhat less in the Black River formation than in the Big Lime. It is also assumed that grouting around the chamber will be required primarily to seal excessively permeable fracture zones, and that somewhat more grouting will be needed in the Silurian-Devonian rocks than in the Middle Ordovician, in which a larger proportion of the fractures may have been naturally healed by vein deposits.

For potential sites which have the option of utilizing one of the Great Lakes for an upper reservoir, a value of zero is assigned to the index for the upper reservoir criterion. In other cases the true value would be very much dependent on the specific site, and it is necessary to generalize to a high degree in choosing a suitable number at the present time. The choice is based primarily on what is known of the surficial geology, particularly whether the soil is of glacial origin (with the consequent probable availability of suitable dike construction materials- particularly impervious core materials- within short distances).

In the case of the shaft criterion, a higher cost is assumed where the average depth of soil is relatively thick, and some consideration is also given to the probable nature of the rock to be penetrated in terms of anticipated shaft sinking problems.

Numerical Site Ranking

The various zones are rated as shown in Table II-4. Designations of the areas are as shown in Figs. II-4 and II-5. Because of the large size of each individual zone, some of the assumed "average" conditions are applicable only in portions of it. The numerical ratings are therefore only a very general indication of the relative merit of each zone. A specific site in a low-ranking area may prove to be as economical as a specific site in a high-ranking area. It is suggested that all of the areas shown in Figs. II-4 and II-5 should be retained for further consideration and that the initial choice of sites within these areas should be based on other than geological considerations. It will then be possible to compare individual sites on the basis of site-specific geotechnical information, rather than having to generalize over large areas as is the case at present. For specific sites which are selected for further study, detailed review would be undertaken of drill hole records in the vicinity of the site. From this work, together with surficial examination of the terrain, more site-specific rankings would result.

In addition to the areas rated in Table II-4, it should be remembered that many other parts of the north central region could probably provide suitable sites for underground storage of compressed air if required. In particular, the beds of the Knox Dolomite and the Lower Mississippian are likely to contain suitable massive members, and these could be considered as second priority formations. These areas have been excluded as generalized first priority formations due to the combination of scarcity of data and known variability from area to area. However, in the event that a specific portion of one of these formations was judged to be particularly desirable from a locational standpoint, intensive investigation of that specific area may indicate that suitable potential exists.

NORTHEAST REGION SURVEY

The Northeast region (see Fig. II-1) consists of a corridor from Washington to Boston, a distance of about 400 mi. The corridor is almost 100 mi wide and covers an area of approximately 40,000 sq mi. Another 5,000 sq mi is included within a radius of about 50 mi from Washington and Boston at the two ends of the corridor. In addition to the District of Columbia, states included are northeast Virginia, Maryland, Delaware, southeast Pennsylvania, New Jersey, southeast New York, Connecticut, Rhode Island, and Massachusetts. The effective width of the southern sector is reduced by large bodies of water including Chesapeake Bay, Delaware Bay, and the adjacent ocean.

Geological Survey

The survey of the three New England states represents an extension of previous work (from Ref. II-31). Additional emphasis during the current study was placed on the section between Virginia and New York. Appropriate geological and geophysical records have been examined wherever possible. These items include maps, reports, well logs, seismic data, and many types of information on bedrock properties. The records have been analyzed for ranking purposes according to the general siting criteria previously discussed. In addition, discussions have been held with many government, industry, academic, and utility representatives.

At least 500 formations of bedrock have been recognized in the region. Most of these are identified in Appendix A. About 10 percent of the formations are mainly good and considered potentially suitable. The remainder are eliminated from further consideration. The sketch map in Fig. II-6 shows only the gross locations of potential cavern sites. More detail is available in a working map 11 ft long. The working map was reduced in size and is reproduced in six segments, Figs. III-7 through III-12. The working map was prepared from numerous sources as a transparent overlay using as a base the 1:250,000 USGS topographical maps. Twelve of these maps are needed to cover the Northeast Corridor. On this scale, which is about four mi to the in., the general position of suitable formations is shown. In many locations, two or more formations occupy adjacent areas but do not overlie one another in the vertical direction.

Broad questions about the feasibility of air storage depend first upon a small scale regional view, which the working overlay is intended to provide. It has been prepared to show the distribution of ares with the necessary siting characteristics.

Using the topographical maps for instance, the extent to which suitable formations coincide with available water supplies is apparent. In many of these areas, geologic information is available on guadrangle maps of more than 10 times the scale (1:24,000). For detailed site planning such larger maps would be essential.

One of the main features of the northeast region is the fall line, which runs roughly through Washington, Baltimore, Philadelphia, Trenton, and New York. This line separates two areas with marked contrasts in regard to underground siting. Most of the area on the northwest side is underlain by crystalline bedrock, and several formations are suitable for deep excavation. In places, there is a relatively thin cover of alluvium and other near-surface materials.

But across the fall line on the southeast side, the surface of the bedrock formations is uniformly overlain by incompetent materials of the Atlantic coastal plain. The inclination of this surface is about 60 ft/mi so that near Atlantic City, almost 100 mi southeast of Philadelphia, hard rock lies about one mi below the surface. Shafts could be constructed through the overlying materials, but only at greatly added cost compared with sound rock. It has been generally agreed that a thickness of 200 ft of these materials is the limit beyond which shafts would not be practical. This 200 ft line is marked on the working map. It is a few miles seaward of the cities named above.

The general criteria used in ranking prospective formations were previously discussed. One restriction adopted at an early stage for the northeast survey was to select only single rock formations large enough for both the cavern and the shaft. The effect has been to eliminate some areas where one good formation overlies another. Thus, Northfield Mountain in Massachusetts, although the site of a perfectly sound underground power house, does not appear in the map of suitable areas because more than one formation is involved. Each, though thin, is suitable. In fact, however, even together they may not extend deep enough for the air storage project, which would go several hundred feet lower, including the U-tube below the cavern.

This restriction to just one formation at each point could probably be relaxed when more detailed siting evaluations are undertaken, although it has been found appropriate in the broad treatment of this region of igneous and metamorphic rocks. There are enough formations that extend from the surface to far below the depth of the projected storage works. So far it has not seemed necessary to look beyond them to mixed strata.

In this respect, the north central region presents substantially different conditions. A parallel survey previously discussed has confirmed that air storage facilities there could be located in the available layered strata. In order to reach the cavern depth each shaft would have to go through varied rocks rather than single formations.

Ranking of Bedrock Formations in Northeast Region

The general siting criteria previously described were used to rank the principal bedrock formations in the northeast region. Four general ranking categories were used. They are:

1. Mainly good, or potentially suitable
2. Marginal or doubtful
3. Doubtful on two or more counts
4. Poor, rejected

The complete ranking of about 500 bedrock formations in the northeast region is given in Appendix A. The potentially suitable formations identified, 54 in all, are listed in Table II-5. These suitable rock formations include granite, gabbro, gneiss, anorthosite, massive schist, diorite, and amphibolite. They are all large, normally hard, moderately uniform, and virtually impermeable. Any reasonable difficulties with joints or fractures could be overcome by conventional methods at varying cost.

The mainly good and suitable formations occupy about one-fifth of the study region (Figs. II-6 through II-12). Many of them are found west of the fall line, others are in southern New England. They are distributed fairly evenly except for the lowland areas of New Jersey. Note in Fig. II-6 that all major cities in this region, and most secondary ones, are within 5 to 40 miles of a suitable formation.

About 450 formations are either doubtful or poor. The former include greenstone, diabase, quartzite, and serpentinite. The latter include poorly consolidated units of the coastal plain, thin sedimentary and volcanic rocks of the Triassic basins, and other formations that are strongly jointed, inclined to be permeable, too thin, and deformed or deformable. Examples include argillite, sandstone, mixed strata, shale, limestone, conglomerate, and mudstone.

REFERENCES FOR PART II

II-1. Barton, N., R. Lien and J. Lunde: Engineering Classification of Rock Masses for the Design of Tunnel Support. Rock Mechanics 6, pp. 189-236, 1974.

II-2. Morfeldt, C. O.: Storage of Oil in Unlined Caverns in Different Types of Rock. ISRM Fourteenth Symposium on Rock Mechanics, Pennsylvania, 1972.

II-3. Wayne, W. J.: Glacial Geology of Indiana. Indiana Geological Survey, 1958.

II-4. Illinois Geological Survey: Background Materials for Symposium on Future Petroleum Potential of NPC Region 9. Illinois Basin, Cincinnati Arch, and Northern Part of Mississippi Embayment, Illinois Petroleum 96, 1971.

II-5. Cram, I. H., ed.: Future Petroleum Provinces of the United States - Their Geology and Potential. American Association of Petroleum Geologists Mem. 15, 1971.

II-6. Bradbury, I. C. and E. Atherton: The Precambrian Basement of Illinois. Illinois Geological Survey Circular 382, 1965.

II-7. Heigold, P. C.: Notes on the Earthquake of September 5, 1972 in Northern Illinois. Illinois Geological Survey Environmental Geology Notes 59, 1972.

II-8. Owens, G. L.: The Precambrian Surface of Ohio. Ohio Geological Survey Report of Investigations 64, 1967.

II-9. Janssens, A.: Stratigraphy of the Cambrian and Lower Ordovician Rocks in Ohio. Ohio Geological Survey Bulletin 64, 1973.

II-10. Calvert, W. L.: Sub-Trenton Rocks from Fayette County, Ohio to Brant County, Ontario. Ohio Geological Survey Report of Investigations 52, 1964.

II-11. Michigan Geological Survey: Stratigraphic Succession in Michigan, Paleozoic Through Recent. Chart No. 1, 1964.

II-12. Shearrow, G. G.: Geologic Cross Section of the Paleozoic Rocks from Northwestern to Southeastern Ohio. Ohio Geological Survey Report of Investigations 33, 1957.

II-13. Buschback, T. C. and G. E. Heim: Preliminary Geologic Investigations of Rock Tunnel Sites for Flood and Pollution Control in the Greater Chicago Area. Illinois Geological Survey, Environmental Geology Notes 52, 1972.

II-14. Ulteig, J. R.: Upper Niagaran and Cayugan Stritigraphy of Northeastern Ohio and Adjacent Areas. Ohio Geological Survey Report of Investigations 51, 1964.

REFERENCES (Cont'd)

II-15. Clifford, M. J.: Silurian Rock Salt of Ohio. Ohio Geological Survey Report of Investigations 90, 1973.

II-16. Ohio Geological Survey: Generalized Column of Rocks in Ohio. Revised 1972.

II-17. Owens, G. L.: The Subsurface Silurian-Devonian "Big Lime" of Ohio. Ohio Geological Survey, Report of Investigations 75, 1970.

II-18. Mozola, A. J.: Geology for Environmental Planning in Monroe County, Michigan. Michigan Geological Survey Report of Investigation 13, 1970.

II-19. Illinois Geological Survey: Geologic Map of Illinois. 1967.

II-20. Indiana Geological Survey: Geologic Map of Indiana. 1956.

II-21. Michigan Geological Survey: The Centennial Geological Map of the Southern Peninsula of Michigan. Publication 39, Geological Series 33, 1936.

II-22. Ohio Geological Survey: Geologic Map of Ohio. 1965.

II-23. Mozola, A. J.: Geology for Land and Groundwater Development in Wayne County, Michigan. Michigan Geological Survey, Report of Investigation 3, 1969.

II-24. Landes, K. K.: Petroleum Geology of the United States. Wiley-Interscience, 1970.

II-25. Illinois Geological Survey: Petroleum Industry in Illinois, 1973. Illinois Petroleum 103, 1975.

II-26. Ohio Geological Survey: Oil and Gas Fields of Ohio. 1975.

II-27. Beebe, B. W., ed.: Natural Gas of North America. American Association of Petroleum Geologists Mem. 9, Vol. 2, 1968.

II-28. Stout, W., K. V. Steeg and G. F. Lamb: Geology of Water in Ohio. Ohio Geological Survey Bulletin 44, 1943.

II-29. Ohio Department of Natural Resources, Division of Water: Ground-Water Resources in Ohio. Undated.

II-30. Smith, W. H. and J. B. Stall: Coal and Water Resources for Coal Conversion in Illinois. Illinois Water Survey and Illinois Geological Survey, Cooperative Resources Report 4, 1975.

II-31. Farquhar, O. C.: Geological Survey of Potential Cavern Areas in New England. Report to Northeast Utilities Service Company, 1974.

TABLE II-1

PRELIMINARY CHARACTERISTICS FOR
REFERENCE CAPS DESIGN

Generation Capacity	280 MW
Generation Time	10 hr
Compression Time	10 hr
Overall Pressure Ratio	50:1
Cavern Storage Volume	250,000 cu yd
with 10% Margin	275,000 cu yd

TABLE II-2

SUMMARY OF INCREMENTAL CAPITAL COSTS
FOR NUMERICAL RANKING

Ranking Factor	Range of Incremental Capital Cost, ¢/kW
Rock treatment to reduce air leakage (grouting)	0 - 120
Water curtain*	200
Roof and wall reinforcement (bolts and mesh)	0 - 1,150
Shafts	0 - 400
Upper reservoir	0 - 350

*Optional item, single cost, no range.

TABLE II-3

BIG LIME SEQUENCE IN CENTRAL OHIO

County	Top Depth, ft	Bottom Depth, ft	Thickness, ft
Cuyahoga	1,390	3,066	1,676
Lorain	1,278	2,460	1,182
Ashland	1,375	2,374	999
Richland	1,428	2,315	887
Knox	1,418	2,231	813
Licking	1,386	2,117	731
Fairfield	1,430	2,070	640
Hocking	1,204	1,845	641
Vinton	1,468	2,050	582
Jackson	1,435	1,945	510
Scioto	1,459	1,900	441

TABLE II-4

GEOLOGICAL RANKING FOR POTENTIAL CAVERN AREAS IN NORTH CENTRAL REGION

Zone Designation*	Incremental Capital Costs ¢/kW						Relative Ranking
	Rock Treatment	Water Curtain	Reinforcement	Shafts	Reservoir	Total	
Ohio A	100	0	400	220	0	720	3
Ohio B	100	0	400	260	300	1,060	7
Ohio C	50	0	200	260	200	710	2
Indiana - Ohio (central part)	50	0	200	260	200	710	2
Indiana - Ohio (lakeshore)	50	0	200	260	0	510	1
Michigan	100	0	400	260	200	960	5
Indiana A	50	0	200	220	250	720	3
Indiana B	100	0	400	220	300	1,020	6
Illinois A	100	0	400	220	200	920	4
Illinois B	50	0	200	260	200	710	2

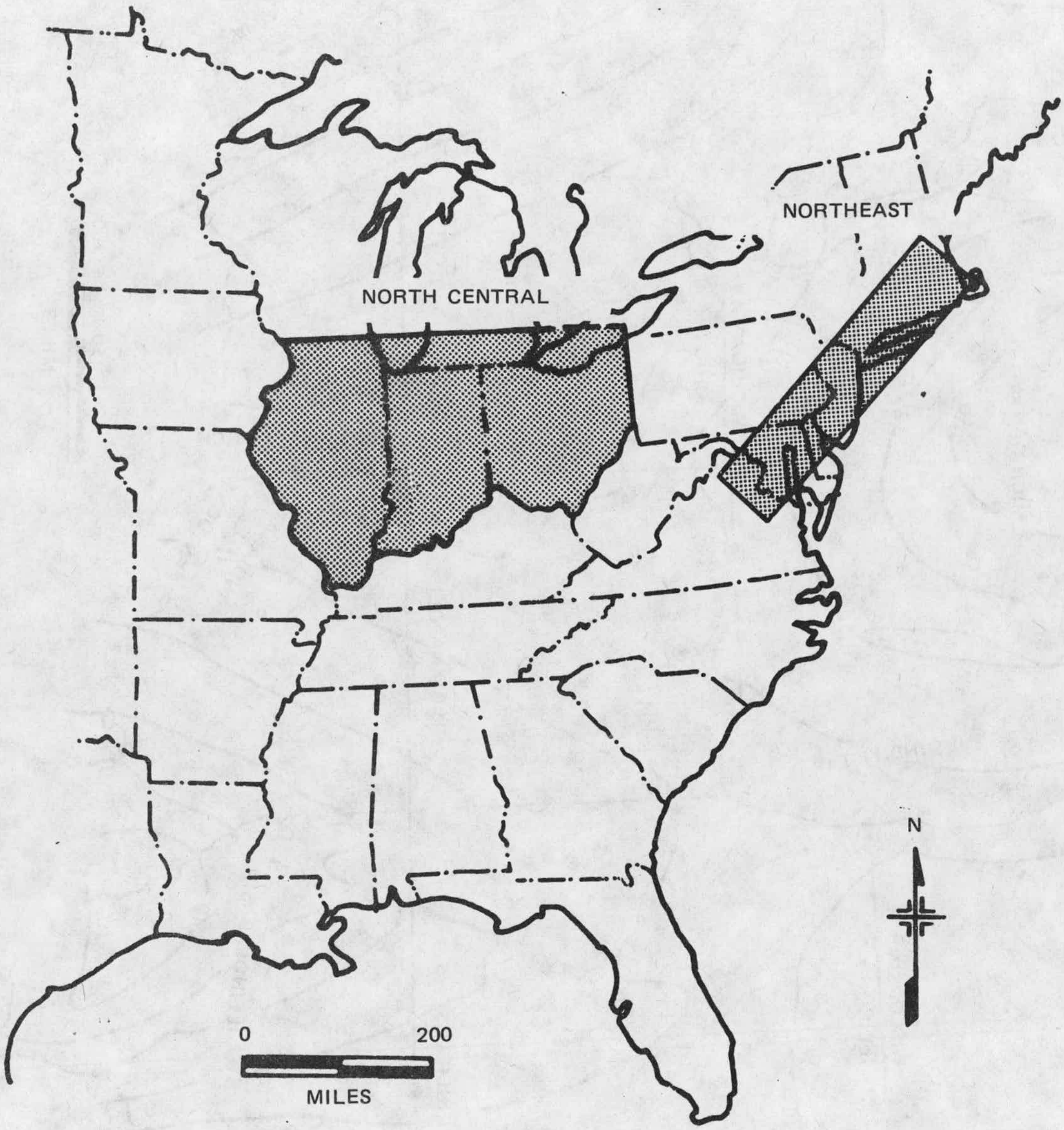
* See Figures III-4 and III-5 for locations of zones.

TABLE II-5

Guide to Favorable Rock Formations in the Northeast Region

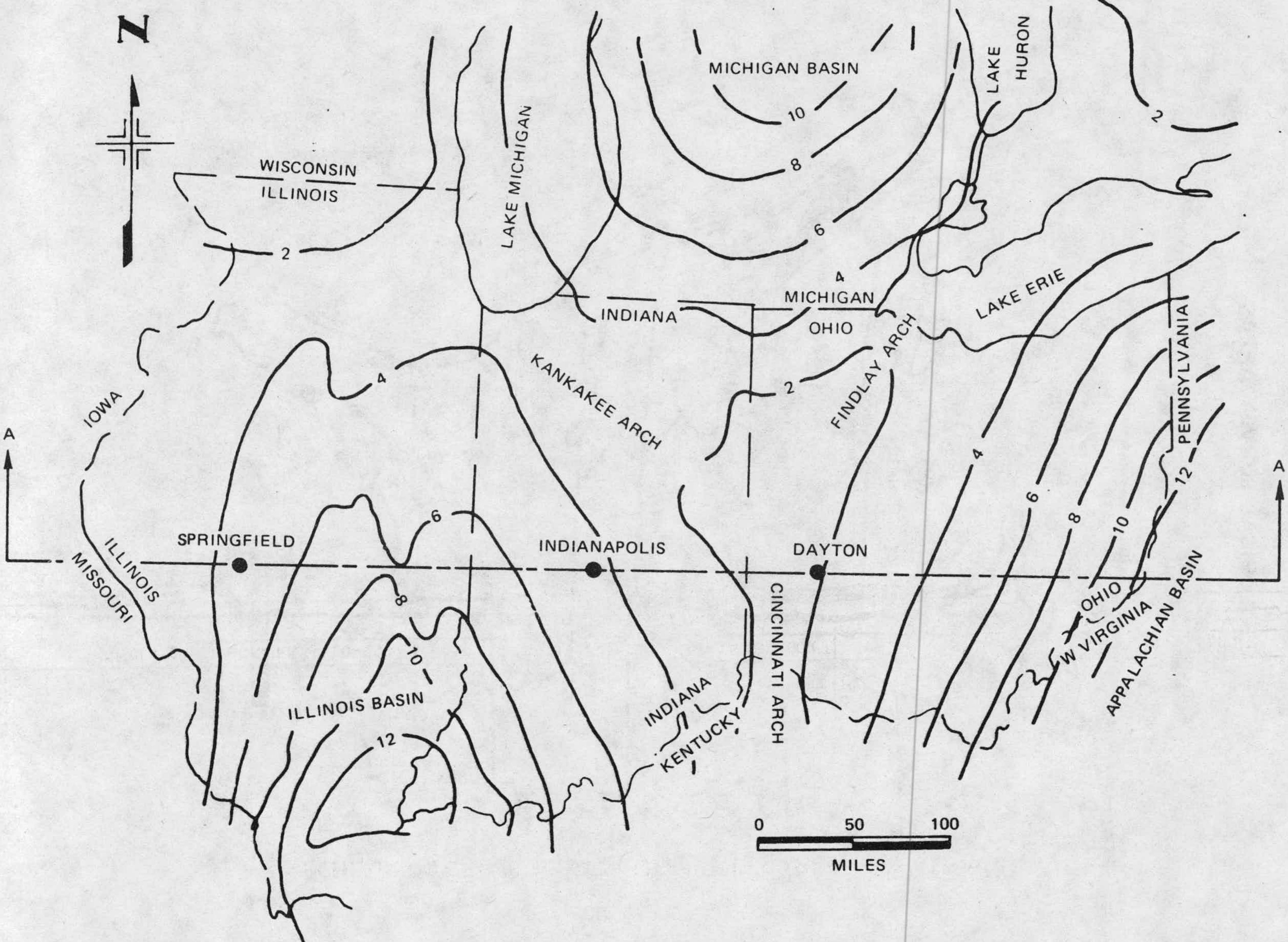
1 Andover Granite, Ma
2 Anorthosite near Honeybrook, Pa
3 Ansonia Granite, Ct
4 Arden Granite, De
5 Assabet Quartz Diorite, Ma
6 Ayer Granite, Ma
7 Baltimore Gabbro, Md
8 Baltimore Gneiss, Md
9 Beverly Syenite, Ma
10 Bulgarmarsh Granite, RI Ma
11 Cannan Mountain Schist, Ct
12 Cape Ann Granite, Ma
13 Cherry Hill Granite, Ma
14 Dedham Granodiorite, Ma
15 Edison Gneiss, NJ
16 Ellicott City Granite, Md
17 Escoheag Quartz Diorite Gneiss, Ct RI
18 Esmond Granite, RI
19 Glastonbury Granite Gneiss, Ct
20 Gospel Hill Gneiss, Ma
21 Haddam (& Monson) Gneiss, Ct
22 Hardwick Granite, Ma
23 Harrison Gneiss, NY Ct
24 Hope Valley Alaskite Gneiss, RI
25 Laurel Migmatite, Md
26 Marshall Gneiss, Va
27 Middlefield Granite, Ma
28 Milford Granite, Ma
29 Mount Prospect Complex, Ct
30 (Newark) Gabbro, De
31 Newburyport Quartz Diorite, Ma
32 Nonewaug Granite, Ct
33 Norbeck Quartz Diorite, Md
34 Peabody Granite, Ma
35 Peekskill Norite etc., NY
36 Pinewood Adamellite, Ct
37 Port Deposit Granodiorite, Md De Pa
38 Poundridge Granite, NY
39 Preston Gabbro, Ct
40 Prospect Gneiss, Ct
41 Quincy Granite, Ma (& RI)
42 Salem Gabbro-Diorite, Ma
43 Sharpners Pond Tonalite, Ma
44 Sterling Granite Gneiss, Ct RI
45 Stony Creek Granite Gneiss, Ct
46 Straits Schist, Ct
47 Sykesville Granite, Md
48 Topsfield Granodiorite, Ma
49 Tyler Lake Granite, Ct
50 Wenham Monzonite, Ma
51 Westwood Granite, Ma
52 Willimantic Gneiss, Ct
53 Wilminton Complex, Md De Pa
54 Woodstock Granite, Md

GEOLOGICAL SURVEY AREAS



R76-952161-5

NORTH CENTRAL REGION MAJOR STRUCTURAL ELEMENTS



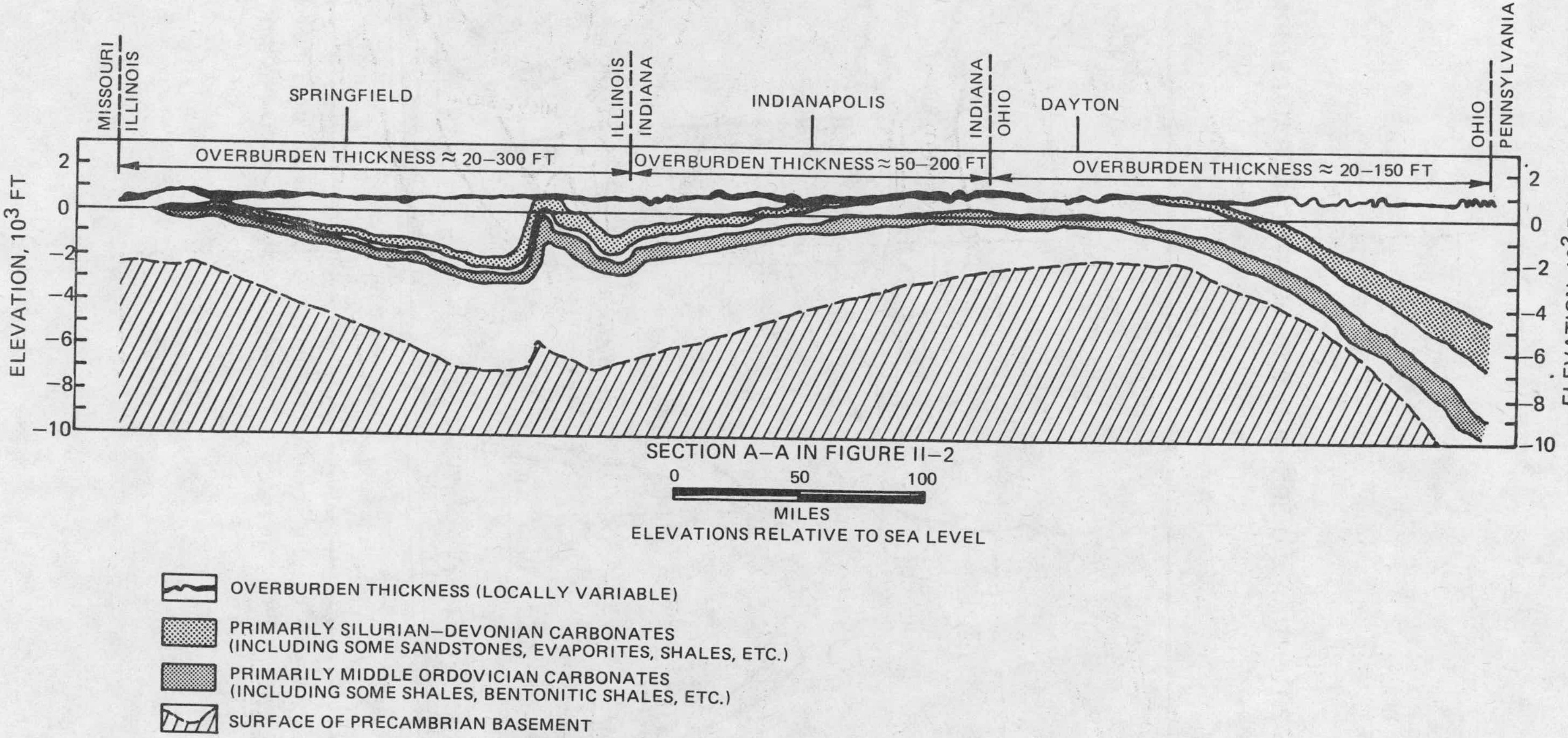
ELEVATIONS OF TOP OF BASEMENT IN FEET BELOW SEA LEVEL (x1000)

FIG. II-2

76-05-93-3

R76–952161–5

SCHEMATIC CROSS SECTION OF MAJOR CARBONATE STRATA

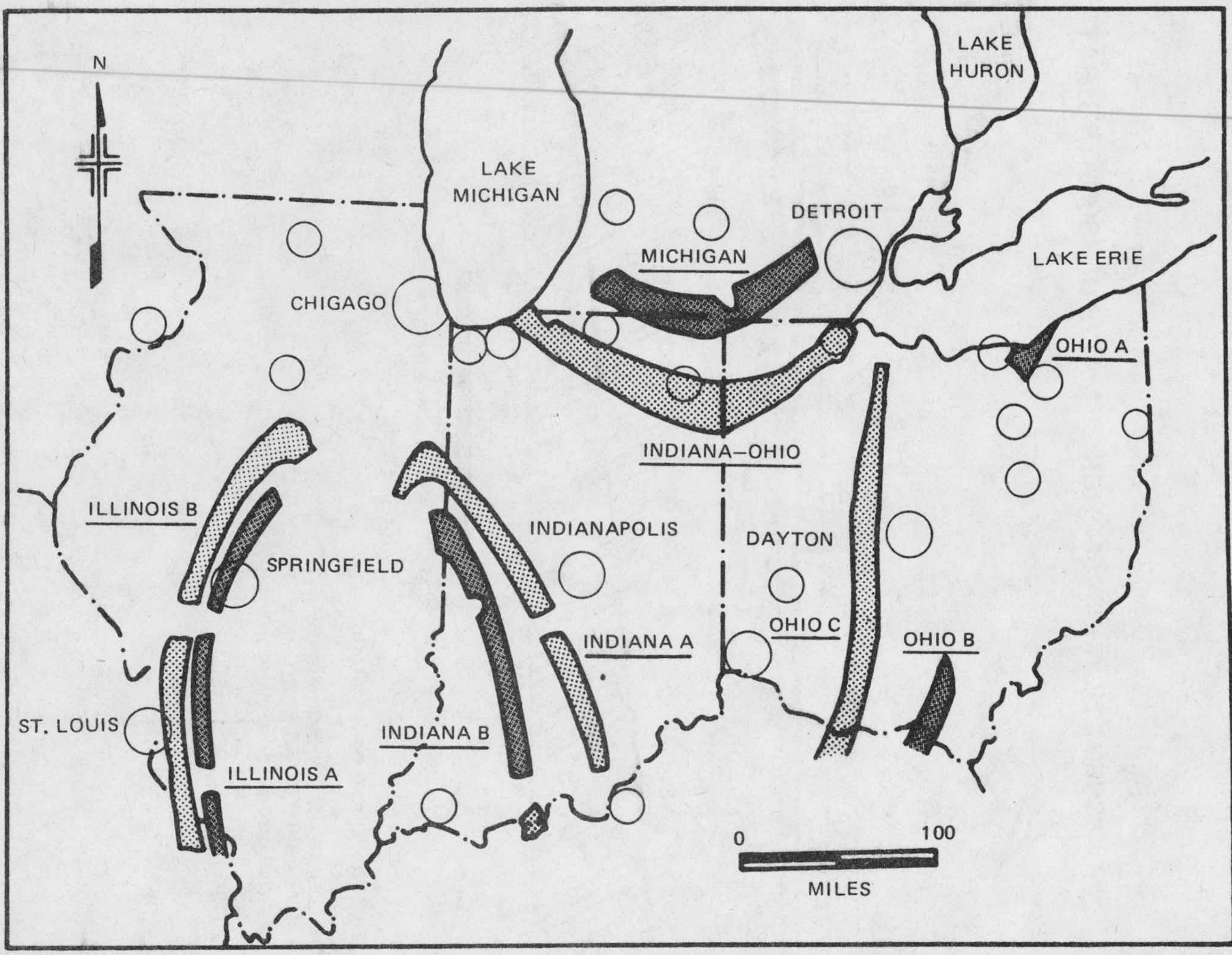


76–11–219–1

FIG. II–3

POTENTIAL CAVERN AREAS IN NORTH CENTRAL REGION

1500–1600 FT DEPTH

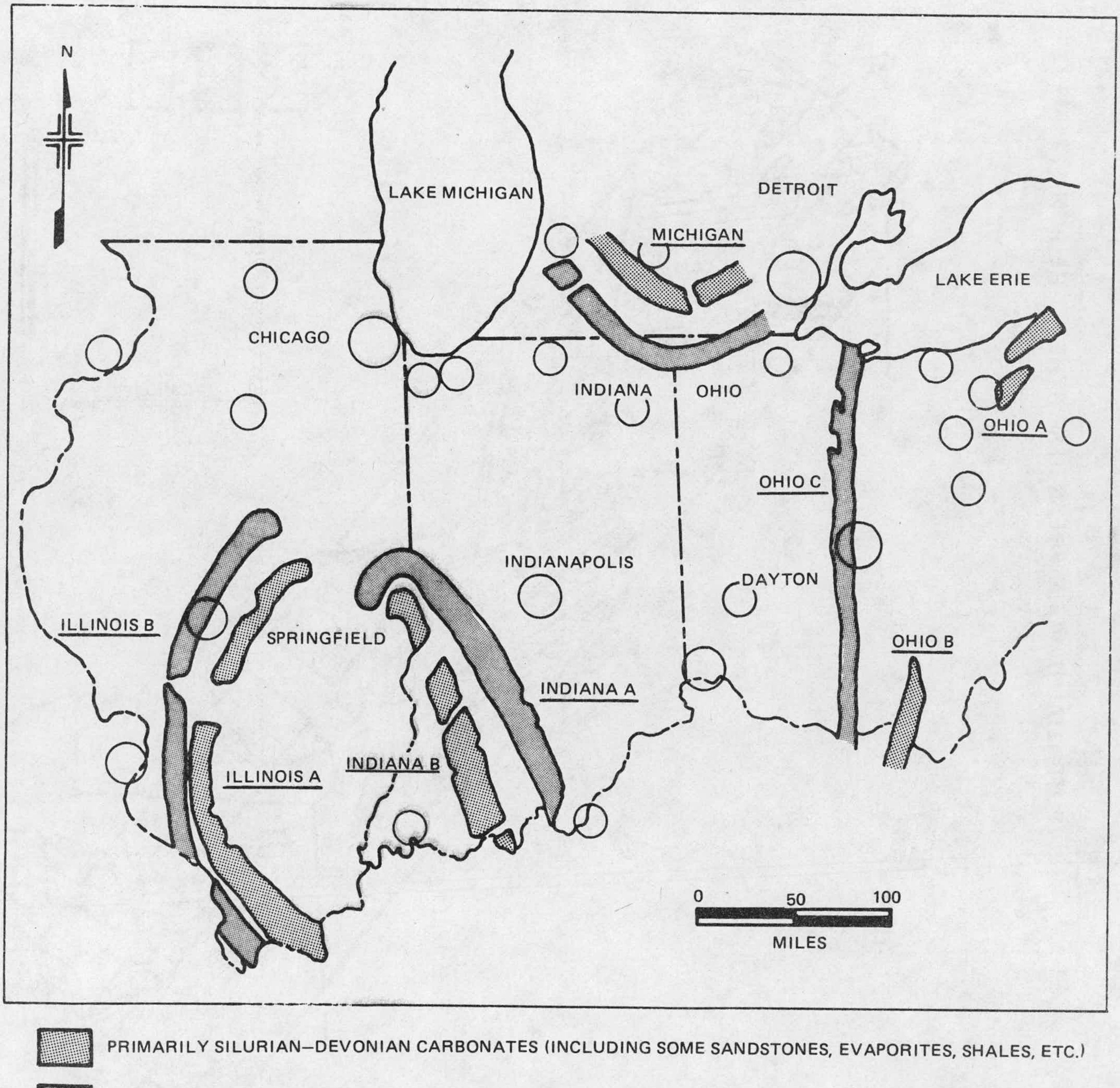


PRIMARILY SILURIAN–DEVONIAN CARBONATES (INCLUDING SOME SANDSTONES, EVAPORITES, SHALES, ETC.)

PRIMARILY MIDDLE ORDOVICIAN CARBONATES (INCLUDING SOME SHALES, BENTONITIC SHALES, ETC.)

FIG. II– 5

POTENTIAL CAVERN AREAS IN NORTH CENTRAL REGION
2200–2400 FT DEPTH

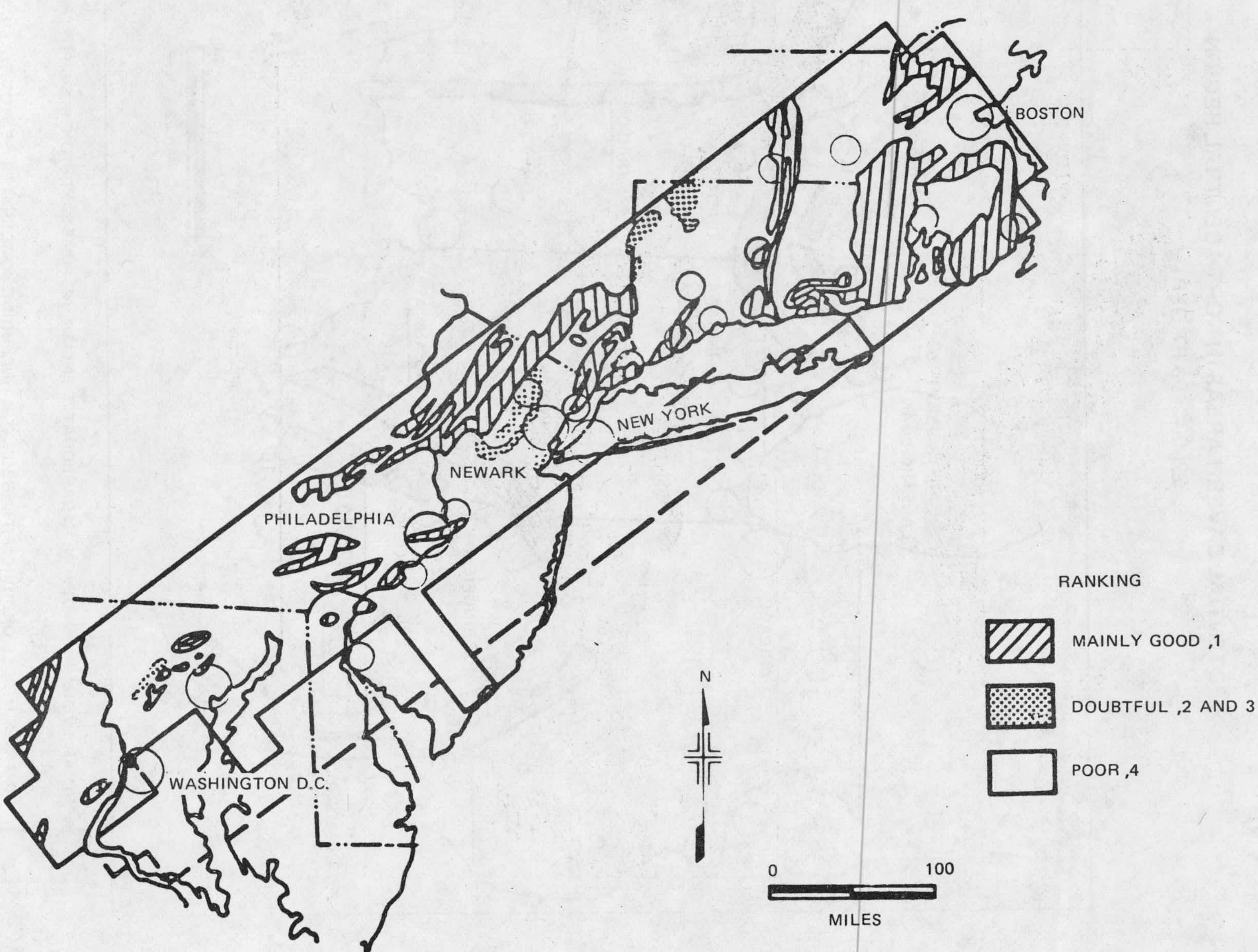


PRIMARILY SILURIAN–DEVONIAN CARBONATES (INCLUDING SOME SANDSTONES, EVAPORITES, SHALES, ETC.)

PRIMARILY MIDDLE ORDOVICIAN CARBONATES (INCLUDING SOME SHALES, BENTONITIC SHALES, ETC.)

R76-951261-5

POTENTIAL CAVERN AREAS IN NORTHEAST REGION



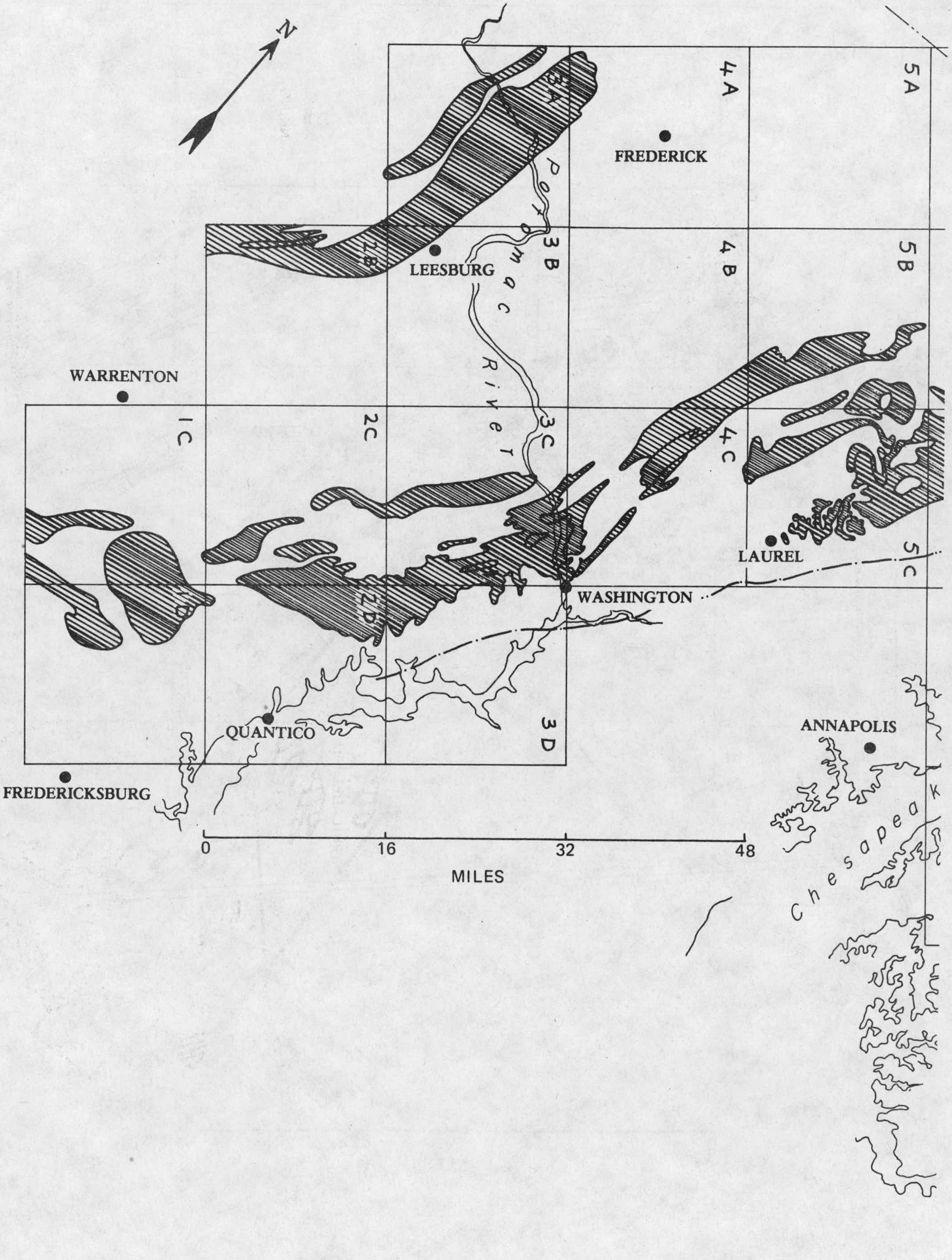
76-05-93-2

FIG. II-6

FIG. II–7

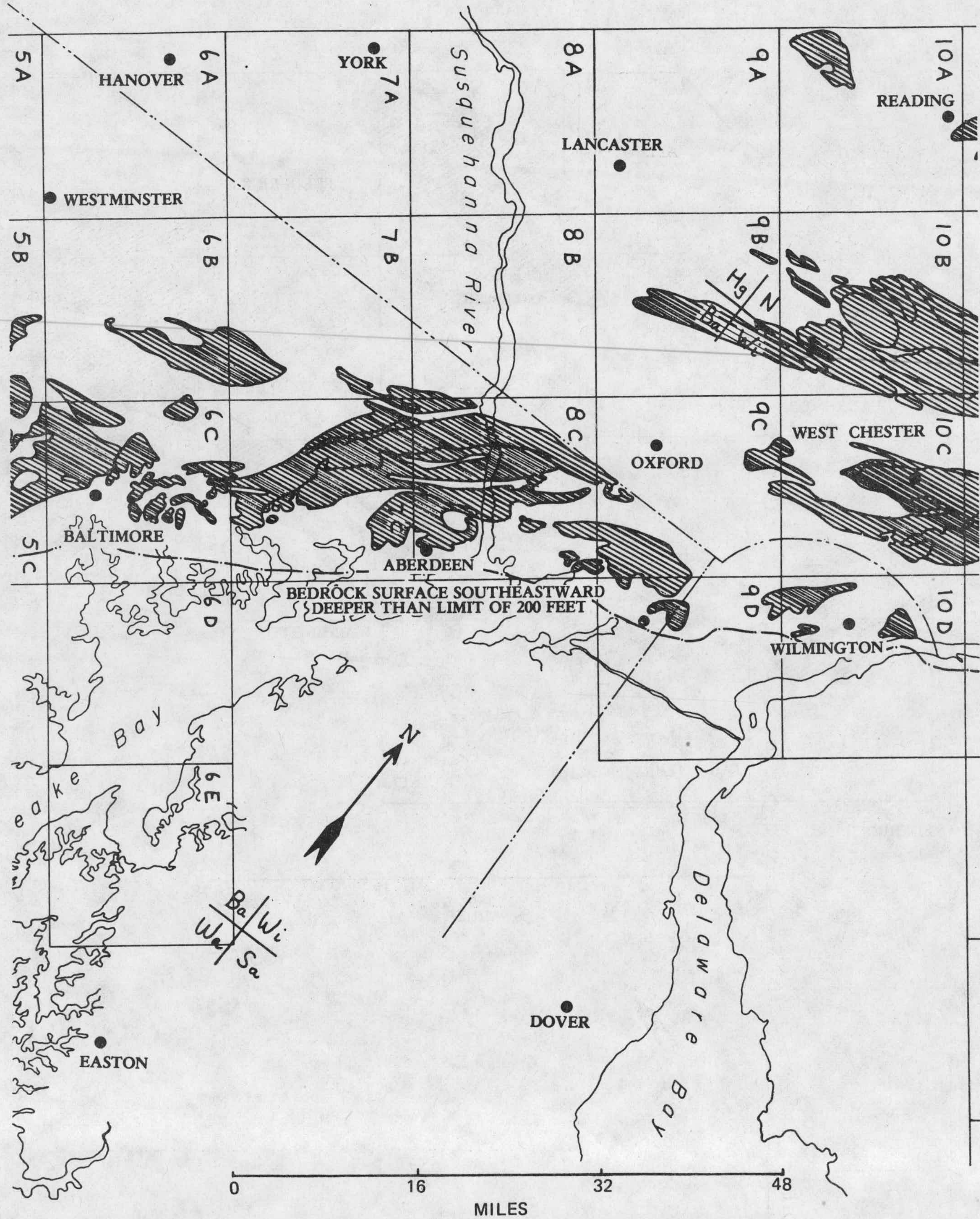
POTENTIAL CAVERN AREAS NORTHEAST REGION

MAP 1 OF 6



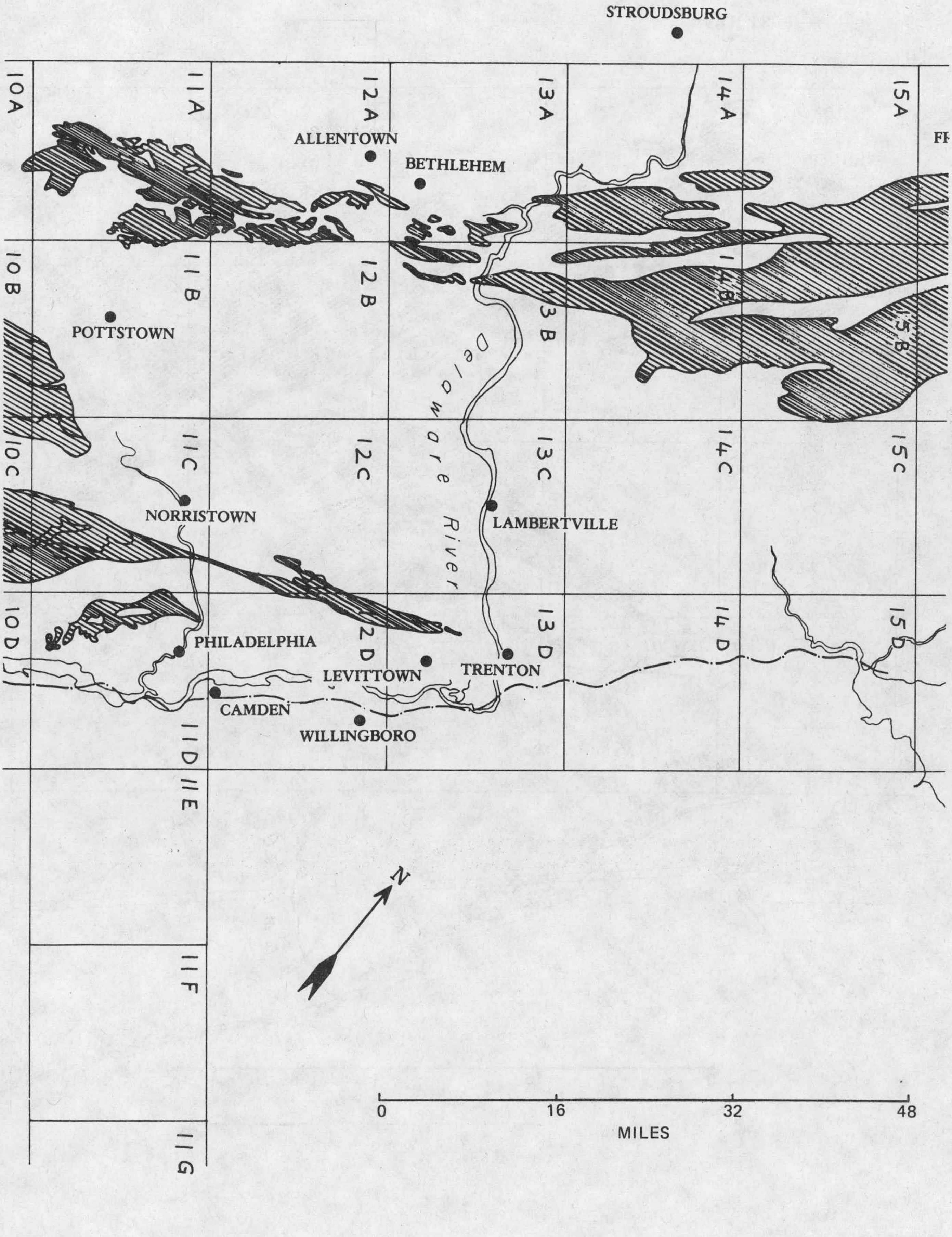
POTENTIAL CAVERN AREAS NORTHEAST REGION

MAP 2 OF 6



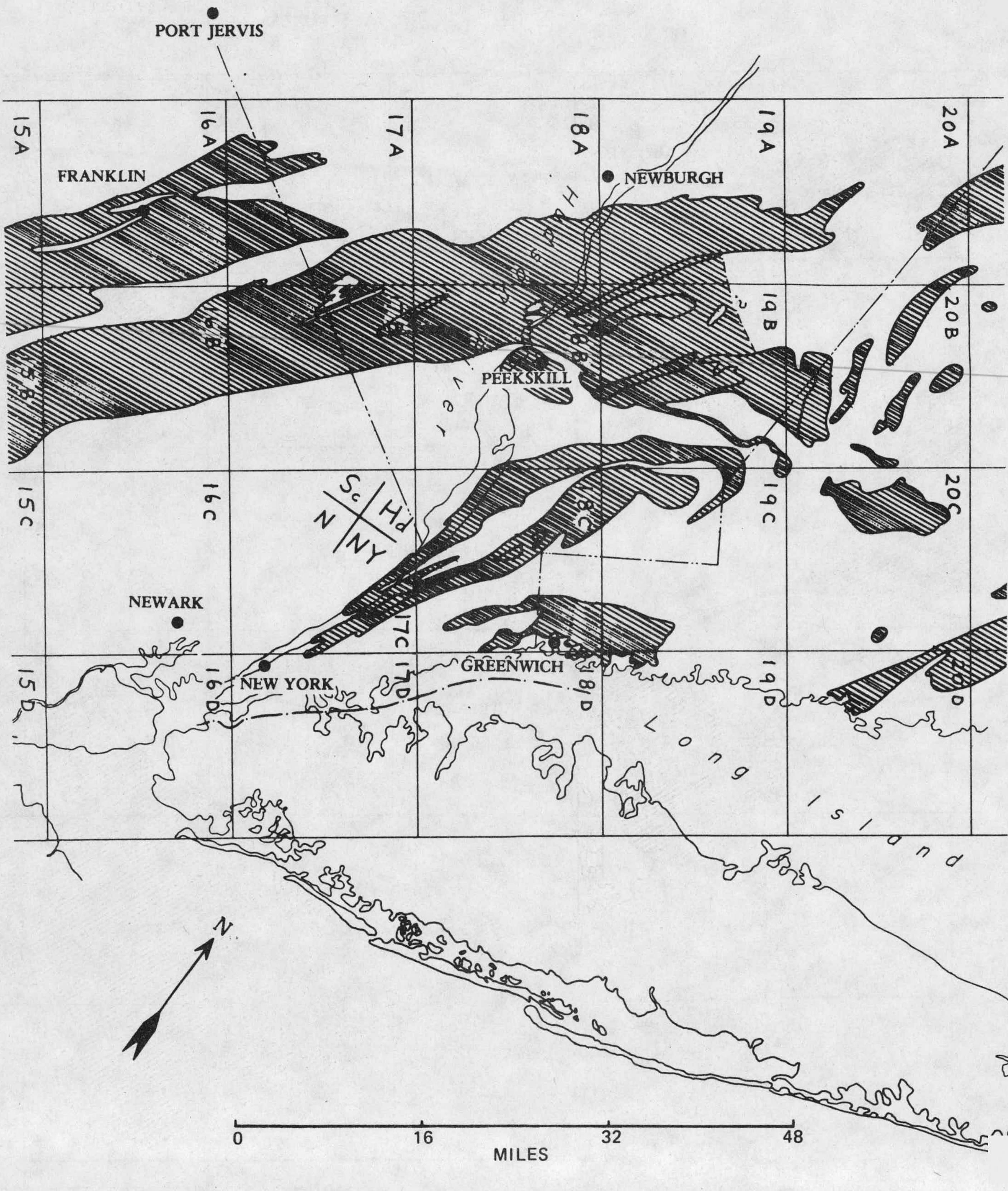
POTENTIAL CAVERN AREAS NORTHEAST REGION

MAP 3 OF 6



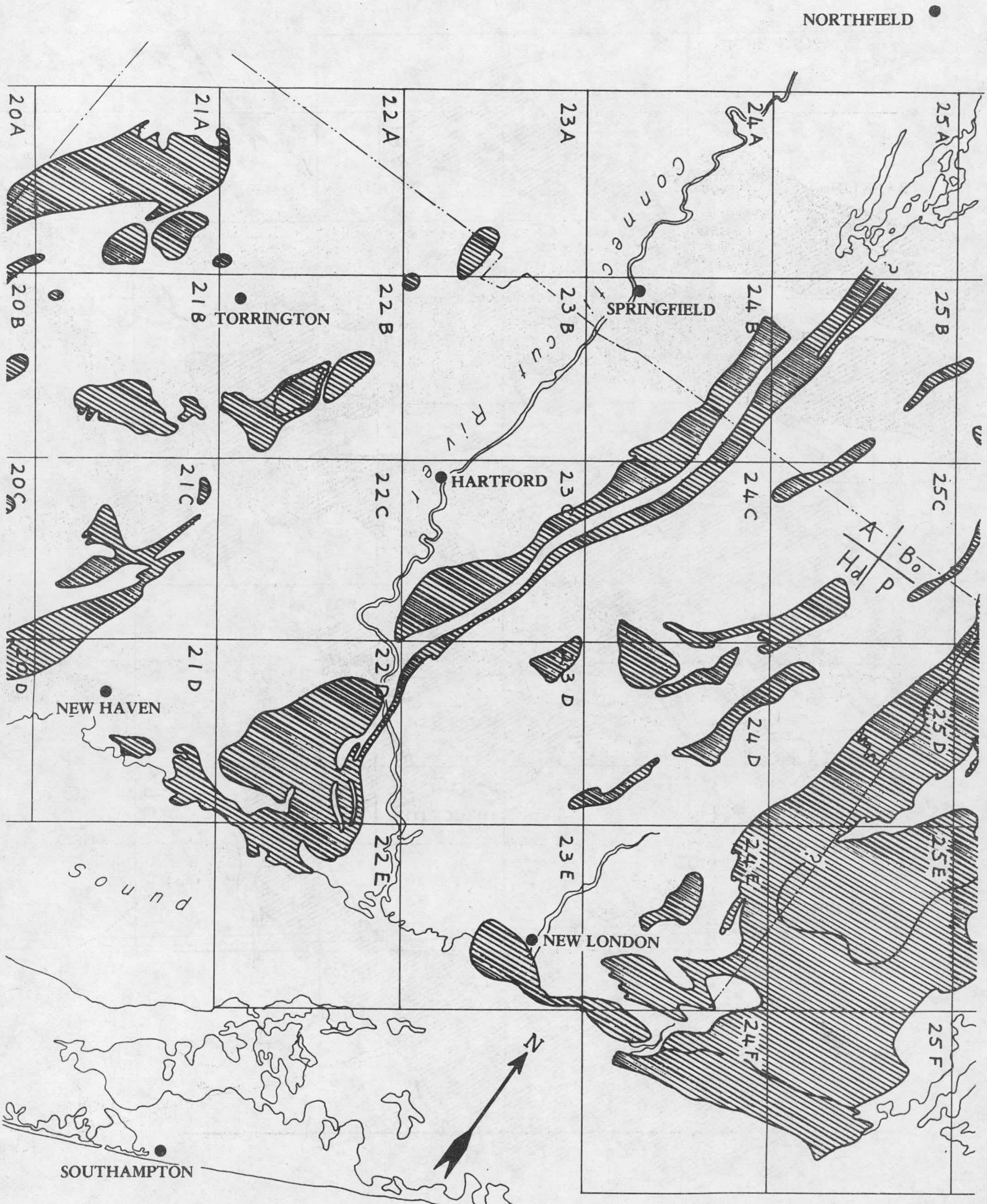
POTENTIAL CAVERN AREAS NORTHEAST REGION

MAP 4 OF 6



POTENTIAL CAVERN AREAS NORTHEAST REGION

MAP 5 OF 6

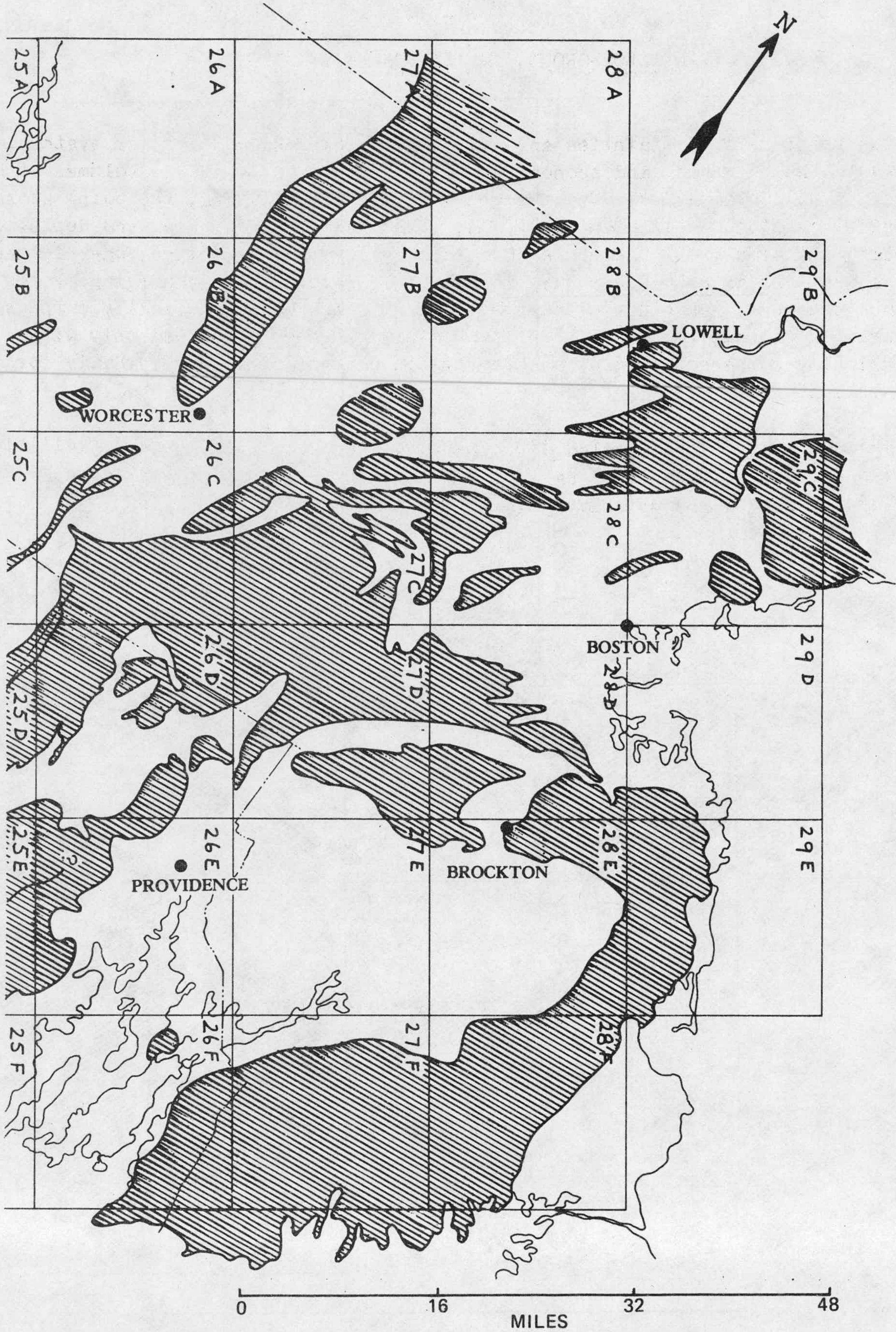


R76–952161–5

FIG. II–12

POTENTIAL CAVERN AREAS NORTHEAST REGION

MAP 6 OF 6



PART III

BELOW-GROUND FACILITIES EVALUATION

The principal uncertainties associated with compressed air power systems are related to the technical and economic feasibility of storing large volumes of high-pressure air. Several modes of storage appear to be promising, including storage in aquifers, solution-mined salt caverns, mined hard-rock caverns, and depleted natural gas fields. Storage can also be either at constant volume, wherein the storage pressure is allowed to vary during the charging and discharging cycles, or constant pressure, wherein the pressure is maintained by hydraulically compensating the variable storage volume. The present study has been concerned only with hydraulically compensated hard-rock storage in caverns mined specifically for the CAPS application.

This part contains detailed discussions of the main below-ground facilities. Included are commentaries on: cavern excavation technology, special aspects of compressed air storage, and cavern layout and costs.

CAVERN EXCAVATION TECHNOLOGY

This section contains a review of technology for constructing underground caverns in hard rock, including sinking of access shafts, spoil handling and ventilation.

Cavern Excavation Methods

The basic requirement for air storage is to produce a relatively large storage volume at depth with minimum cost. This has to be compatible with other requirements such as stability of roof and walls during construction and operation, minimization of air leakage, and hydraulic efficiency. The possible excavation methods can be conveniently classified under three headings:

- mining methods,
- machine-bored tunnels, and
- conventional drill-and-blast methods.

All three classifications are discussed below. However, based on a review of these alternative excavation technologies, the conventional drill-and-blast methods were considered most appropriate for CAPS storage facilities. The excavation process would proceed on a drill, blast, muck, and hoist sequence using multiple headings and benches. Rock bolts and chain link mesh would be used as reinforcement, and cement grouting would be used to prevent excessive water inflow and, subsequently, air leakage.

Mining Methods

Mining methods of excavation are directed toward obtaining the maximum volume of material from an ore body. Stability is usually of importance only for the safety of the miners and equipment in the immediate vicinity of the working faces. Collapse of the mined-out space is common and, in most cases, deliberate in the long term. This can be accompanied by extensive fracturing of the rock above the mining zone, leading to high permeability and subsidence at the ground surface. Methods of this type, despite a (sometimes) rapid rate of removal of material, are obviously unsuitable for the creation of an air storage.

Some of the mining methods have been developed for steeply dipping ore bodies. The resulting excavation may be stable, but is very high (up to several hundred feet) in comparison with its width. This configuration is again unsuitable for a hydraulically pressure-compensated air storage because of the implied variation in air pressure. Some of the more common mining methods (Ref. III-1) are briefly evaluated below:

(a) Sublevel Stoping - Stoping is a mining term for the loosening and and removal of ore. A stope is an excavation from which the ore has been extracted. Sublevel stoping requires a drawpoint loading system and an extensive network of haulage drifts, cross cuts, and raises for access to sublevels. Results in a high, narrow cavern. Unsuitable.

(b) Shrinkage Stoping - Same as for sublevel stoping except that rock ore is removed from the roof instead of the wall of the stope. Unsuitable.

(c) Cut and Fill - Another method used in vertical or steeply dipping ore bodies. Ore is removed from the stope roof and removed with a combination of lateral movement and vertical ore passes. The excavation is usually from bottom to top of ore body with the floor being continually filled with waste material, hence no real increase in storage volume. Unsuitable.

(d) Sublevel Caving - Used in vertical or steeply dipping ore bodies working from top to bottom. Can be highly mechanized. Results in high, narrow cavern. Unsuitable.

(e) Block Caving and Long Wall Mining - Both of these methods result in the loss of the excavated space due to massive subsidence. Unsuitable.

(f) Room and Pillar - This mining method is frequently used in ores with a horizontal or flat dip. Work is possible on several faces at one time and a combination of heading and benching can be used. The resulting space consists of a regular array of square rooms and pillars. In a limestone mine the rooms could have typically a 30-foot clear opening separated by 20-foot square pillars. The relative sizes of the rooms and pillars depend on the strength and continuity of the rock in the roof and in the pillars. Potentially suitable.

Methods (a) through (e) above have been discarded as unsuitable because of geometrical considerations and risk of surface subsidence. Only the room and pillar method shows any potential in the creation of an air storage cavern.

Machine-Bored Tunnels

Present-day tunneling machines are capable of excavation in materials of practically any hardness from clay to extremely hard rocks of over 50,000 psi compressive strength. Increasing attention is being paid to the economic use of these machines and extensive research programs are being carried out by public bodies such as the United States Bureau of Reclamation, Advanced Research Projects Agency (ARPA, Ref. III-2), National Science Foundation (Ref. III-3), and by industry. Pressure on the environment has prompted the federal government to establish a National Committee on Tunneling Research and Technology, charged to act as a focal point for academic, industrial and professional skills in this area. This body has stated that new techniques may reduce the cost of tunneling, including boring, supports and linings, by as much as 30 percent in the next 10 years; continuous excavation is fundamental to the achievement of such economics and is centered on the use of the tunneling machine or mole.

Many types of machines available are capable of drilling horizontal and inclined tunnels and vertical raises (Ref. III-4). Designs vary quite widely from full-face machines favored by the majority of manufactures to the pilot-hole and ream approach similar to that employed in raise boring; to the multiple rotating cutting head; and to the "Unicorn" single cutter developed in Germany for soft rock mining.

The principle advantages of using a tunneling machine are:

- the machine produces a smooth profile compared to conventional methods; rock disturbances and, therefore, cost of temporary supports are minimized;

- excavation is carried out on a continuous basis, readily adaptable to high-production rock mucking and hoisting systems; and

- excavated material is of small size, requiring no further crushing for removal and hoisting (though this might turn out to be a disadvantage according to the use which is ultimately found for the rock at the surface).

The further advantage which the machine offers in normal tunneling practice, that of accuracy of profile and lack of overbreak and waste excavation, is not an advantage in the air storage caverns where the object is to create useful volume, and the accuracy of the cavern profile is of little account. Distinct disadvantages in the use of the machine, aside from the question of cost, would be its slowness in terms of rate of removal of rock compared with conventional methods, its general lack of adaptability and, on present standards, its low reliability.

In a recent comparison of the performance of raise and tunnel-boring machines (Ref. III-5) the best performances were as follows:

Case	Date and Machine	Diameter, ft	Rock Strength, psi	Rate of Penetration, ft/hr
A	1965 Hughes-Betti	20-21	5,000-6,000	10-17
B	1970 Robbins 61R	6	29,000	11.5

Assuming a high level of utilization (400 hours per month) the volume of material excavated in Case A was of the order of 86,000 cubic yards per month, and in Case B it was 4,800 cubic yards per month. It should be noted that in Case A, although the excavation rate was high, the rock strength was low, and the rock probably offered a low resistance to the machine. In Case B the rock strength was comparable to strengths expected in the igneous and metamorphic rocks suitable for air storage. The rate of extraction for creating a storage volume of the size needed for CAPS would be too low and the diameter too small. The comparison also showed that for all rocks with strengths greater than 12,000 psi, the maximum tunnel size diameter was 11.5 feet and maximum rate of advance 11.5 feet per hour. It was concluded that even in situations where a tunnel rather than a storage volume is required, in comparison with drill and blast methods ". . . . it seems unlikely that mechanical boring machines can be developed to deal with the hardest and abrasive rocks economically".

In a report on the status of the ARPA program (Ref. III-2), similar conclusions were made. The main cause for the poor performance of hard rock tunneling machines was identified as the inability to reliably produce a sufficiently high thrust on the cutters necessary to break the hard rock at high advance rates. The machines were stated to be noncompetitive with drill-and-blast techniques in rocks with compressive strengths greater than 20,000 psi. However, in the limestones and dolomites which might have compressive strengths less than 20,000 psi and which could be anticipated in the north central study region, it is possible to envisage heading work performed by a tunneling machine, followed by conventional benching. The optimum diameter of a machine working in limestone is probably less than 20 ft which, along with the circular shape of the bore, would cause inefficiencies in the subsequent benching operations.

Recent research with full-scale water jet-assisted tunneling machines has shown a measured average improvement in tunneling rates of 45 to 50 percent, with much greater improvements in the harder rocks. With a machine designed and built to work specifically with water jet assistance it is felt that the improvement in tunneling rates in hard rocks could be in the order of 200 percent (Ref. III-6). Such a machine might be available in the late 1970's and might have an application in a compressed air power systems project. However, the capital cost of a tunneling machine would probably be too high to be written off in the project and still show economic excavation unit rates.

Conventional Drill-and-Blast Methods

The conventional drill-and-blast method can be considered as the state-of-the-art approach to excavating large caverns underground. Figure III-1 shows typical cross sections of a large cavern excavation, with a top heading and three benches. The heading is analagous to full-face tunneling where there is considerable confinement of the rock, therefore requiring more drilling and larger quantities of powder for blasting than the benching operation.

Figure III-2 shows a typical drill-and-blast pattern in a heading, including provision for a pilot heading which could be excavated first to provide information on the rock conditions ahead of the work face. Drilling would be performed typically by a multidrill "jumbo" which is a self-propelled rig (air power or electric/hydraulic power) carrying several highly maneuverable drill booms. The numbers adjacent to each drill hole represent the order in which the charges in those holes are ignited, with delays between charges measured in milliseconds. This type of blasting pattern reduces the ground vibrations because of the staggered ignition times, and provides an orderly decrease in the confinement of the rock from the center of the heading outward. It should be noted that the charges at the perimeter of the excavation are lightly loaded and closely spaced to give a "smooth-wall" blast.

Figure III-3 shows a typical bench drilling and mucking operation. The drill holes are usually subvertical (as shown) and drilled in rows parallel to the bench face. Several rows can be blasted at one time, again using short delays to provide an orderly release of confinement without disconnecting the later charges. Confinement is much less in benches, which therefore require less drilling and less powder than headings. A smooth wall can be achieved in both benches and headings by drilling closely spaced holes around the perimeter of the excavation and igniting the perimeter blast first (presplitting), or with a short delay after the main blasts (smooth-wall blasting).

Mucking (removal of blasted rock) can be by means of diesel-powered, front-end loaders and haul trucks or by air or diesel-powered, self-loading vehicles. Ventilation is of utmost importance for the diesel-powered equipment. Although in mining and tunneling the use of rail (tracked) or conveyor belt equipment is common, the use of rubber-tired equipment is now the most common in civil engineering applications for large cross-sectional caverns. In relatively shallow excavations (no deeper than 600 to 800 feet) it is common for the excavated rock to be hauled to the ground surface by diesel truck, through an inclined access tunnel or drift. At the depths contemplated for the air storage cavern, the length of tunnel would be excessive, and the rock could be more economically removed by means of a high-speed electric hoist installed in a vertical shaft. The hoist would probably be loaded via a loading pocket excavated in the rock which would be fed by the front-end loaders, self-loading haul units or other haul units.

Figure III-4 shows a typical bolt-drilling operation and bolt pattern for roof reinforcement. Rock bolts are typically 10 to 20 feet long in the size of opening contemplated for the air storage caverns. They may be untensioned (grouted or ungrouted) or post-tensioned (grouted or ungrouted). Chain link mesh is sometimes installed to retain small rock fragments, and is usually attached to the rock-bolt heads via plate washers. Drilling for grouting holes is performed in a similar manner, although in some cases the drilling for grouting would be performed ahead of the main excavations. In very permeable zones, grouting is generally much more successfully done ahead of the excavation. The better confinement allows higher grouting pressures in comparison with grouting a permeable zone intersecting an excavated cavern.

The drill-and-blast methods have been used extensively in North and South America, Europe, South Africa, and Australia for excavating underground hydroelectric powerhouses. In Scandinavia some two hundred oil storage caverns have been built of which a dozen were built for oil storage volumes of 0.6 to 1.3 million cubic yards (Ref. III-7). A summary of some of the excavations made in North America is included in Table III-1. In general, the unit costs of these excavations, which are of comparable volume to the required air storage caverns, have been considerably lower than the unit costs of machine tunneling. This situation is likely to continue for the foreseeable future, particularly with regard to the provision of air storage caverns, because of the following advantages of the drill-and-blast method:

- The method is basically very flexible and the cavern cross section can be readily adapted to suit geological discontinuities, strength deficiencies, or difficult in situ stress conditions.

- The plant involved, although specialized, is used extensively in the mining industry, is of proven reliability, and is readily obtainable.

- In comparison to the high cost of a tunneling machine, the capital investment in the plant is lower and would probably have a resale value after completion of the excavation.

- If mining labor rather than construction labor could be used for the excavation work, considerable economies would result because of the more flexible approach of mining unions.

- A tunneling machine would certainly come under the jurisdiction of construction unions, whereas this need not be the case with the conventional drill-and-blast equipment and operations.

Because of the high cost of the tunneling machines it is unlikely that more than single-face working would be economic. Drill-and-blast methods permit multiface working, performing different operations at each face, allowing continuous utilization of equipment. Although the tunneling machine may work continuously, it has been unable to match the volume rate of extraction of material achieved by the drill-and-blast methods, particularly in hard rock. This could be the case even comparing machine tunneling to the drilling and blasting of small headings, which is considerably slower than the benching operation. In comparison with the machine tunneling excavation figure of 4,800 cubic yards per month in hard rock, given in the previous section, recent drill-and-blast excavation rates achieved in hard rock oil storage caverns in Sweden are 32,000 cubic yards per month in heading and 65,000 cubic yards per month in benching, in a multiface operation.

Finally, the specific energy (which is the quantity of energy required to remove a unit volume from the rock face) consumed in machine tunneling is almost one hundred times greater than the specific energy consumed in the drill-and-blast method (Ref. III-5). Energy consumed in the mucking and hoisting operation is similar for either method. The high energy cost of machine tunneling in hard rock is readily apparent.

Innovative Methods

In attempts to improve the excavation rates currently achieved by tunneling machines in a variety of different rock types, several research programs (Refs. III-2 and III-3) have been instituted in recent years. The ARPA program has concentrated on research methods which show some promise in improving performance in hard rocks. The classification of hard rock adopted for the program is the one due to Bruce and Morrel (Ref. III-8). Their classification considered sediments and metasediments (metamorphosed sedimentary rocks) having uniaxial compressive strengths greater than 20,000 psi as "hard" rocks. For metamorphic and igneous rocks they considered uniaxial compressive strengths greater than 10,000 psi to be indicative of "hard" rocks. This classification includes most of the rocks considered suitable for air storage caverns in the current study, with the exception of some limestones and dolomites. The uniaxial compressive strength does not reflect completely the resistance of a rock to drilling or machine tunneling as the above classification suggests. Abrasiveness and indentation hardness are other factors involved. However, most performance data, both on laboratory and field scales, are related to the uniaxial compressive strength and it is convenient in this study to use it as a "hardness" scale.

Closely related to the performance of tunneling machine and drill and blast cycles is the use of innovative drilling techniques. Maurer (Ref. III-9) gives a review of more than twenty-five innovative drilling techniques, many of Russian origin, only a few of which show any real promise for successful use in the field. He concludes that some of the more successful techniques such as thermal spalling, spark drilling, chemical drilling and high-pressure water jets will find increasing applications in the next few years.

Both Olson and Atchison (Ref. III-2) and Cook and Harvey (Ref. III-5) conclude that at present none of the innovative techniques can make a significant impact on current machine tunneling or drill-and-blast excavation rates. The predominant problem is that most of the techniques require high specific energy values. This means that the techniques have to be limited to low-volume excavation (e.g., kerf cutting), or that very high energy consumption and slow excavation rates or unacceptably hot working environments result near the work face. Some of the techniques do, however, show some promise for the not too distant future and, depending on the starting date, may have a beneficial impact on such projects as excavating caverns for air storage. The techniques can be broadly classified as follows:

- Thermal Energy Processes
- Hydraulic Methods
- Combination Processes
- Improved Explosives Fragmentation
- Improved Mechanical Fragmentation

Thermal Energy Methods

These methods include the electron beam gun, laser, flame jet piercing and the "Subterrene". Both the electron beam gun and the laser provide a high-energy beam suitable for cutting narrow kerfs (slots) by fusion within the rock face. Because of the very high specific energies required, it is not feasible to think in terms of complete rock removal with the devices. However, the cutting of kerfs to weaken the rock face shows some promise for use in combination with a mechanical tunneling machine. At present the size of these devices is not large enough and development is not sufficiently advanced for production tunneling. There are also potential space problems because of the bulk and complexity of the equipment needed to produce the laser or electron beam power (Ref. III-5).

Flame jet piercing depends on a heat source (e.g., propane or acetylene torch) creating sufficiently high thermal stresses so that spalling occurs. Some rocks (mafic, basic rocks containing dark minerals) are not susceptible to spalling because of higher values of thermal conductivity leading to lower temperature gradients (and, therefore, stress gradients) near the rock surface, and fusion may then occur before spalling. Even in the susceptible rocks the specific energy required is very high, leading to high energy costs and cooling problems in the tunnel.

The rock melting Subterrene developed by Los Alamos Scientific Laboratories advances through rock using a high-temperature penetrator with a molybdenum body, which leaves behind a smooth wall of rock "glass". It thus leaves a stable hole requiring no lining (Ref. III-10). It has been used successfully in the field to drill holes 2 inches in diameter (Ref. III-11) and has been proposed in several forms for large-diameter tunnel excavation (Ref. III-12). Currently most effort is being made in producing a Subterrene suitable for drilling deep vertical holes for geothermal energy exploitation. The use of the Subterrene in the field for producing large-diameter tunnels is not likely in the near future (Ref. III-3).

Hydraulic Methods

Hydraulic impact methods include rock disintegration by low-velocity water slugs, high-pressure continuous jets, and ultra high-pressure pulse jets (Ref. III-2). The low-velocity water slug appears to have a disappointing performance with very high specific energy values (energy required per unit volume of rock removed).

The high-pressure continuous jet has been used to cut kerfs at high specific energy values (Ref. III-5). It has the drawback of requiring the nozzle to be close to the rock face which, on a full-scale project, would be impacticable because of the rock surface roughness and undulation, unless used with a tunneling machine producing a smooth face.

The ultra high-pulse jet method uses pulse pressures of the order of 500,000 psi. Specific energy values less than those achieved by conventional mechanical tunneling machines have been achieved (Ref. III-2), the fracturing mechanism being one of cratering. With potentially higher pressure utilizable, the method could become feasible for rapid hard-rock excavation, but is not feasible at present.

Combination Methods

These methods include a combination of rock weakening by kerf cutting, using one of the methods discussed above, or by the use of chemical rock weakening agents in conjunction with a mechanical tunneling device. Other combination processes which have been studied are thermomechanical breakage, chemomechanical drilling, sonic power for rock drilling, and the use of steel and concrete projectiles for initial fragmentation.

Probably the most promising combination process is the water jet-assisted tunnel boring machine which has reached the position of being usable on a full-scale tunneling contract (Refs. III-3 and III-6). The considerable potential improvement in excavation rates for tunneling machines using water jet assistance has been detailed in a previous section entitled, Machine-Bored Tunnels.

There have also been some very promising laboratory scale experiments using chemicals to enhance drilling rates (Ref. III-13). The effect of the chemicals is to change the surface potential of the rock being drilled. Some chemicals lead to a large positive or negative charge, both of which make brittle materials more brittle and, therefore, more susceptible to percussion or diamond bit drilling. Other chemicals lead to a near zero charge which makes softer materials softer still, and allows faster drilling rates with bits having a ploughing or gouging action, e.g., drag and spade bits. Drilling rates, using a specific chemical for a specific rock type, have shown improvements by a factor of three or four, and bit wear has been reduced by a factor of two. An optimistic assessment for reliable field scale use of a chemically assisted drill is approximately three years (Ref. III-13).

Improved Explosives Fragmentation

Explosives fragmentation, as exemplified by the drill-and-blast method, has reached a fairly refined state and gives, in hard rock, the best excavation rates and lowest specific energies of all practical methods. Unless a major research effort is directed toward improving the method, excavation rates can be expected to improve at a maximum rate of about 2 percent per year (Ref. III-2).

Advances could perhaps be made using a combination of kerf cutting and drill-and-blast methods or by the use of explosives to weaken the rock face for tunneling on a continuous basis.

Improved Mechanical Fragmentation

The state of the art of tunneling-machine excavation in comparison to drill-and-blast methodshas already been discussed. In the past 10 to 15 years there have been rapid advances in the technology with a five to tenfold increase in tunneling rates, but only in susceptible rock types. It has been concluded (Ref. III-2) that improvements will still occur, but only at a relatively modest rate, with possible considerable improvement using water jet assistance.

In hard-rock excavation, tunneling machines exhibit a combination of a high thrust requirement, rapid cutter and cutter bearing wear, relatively low specific power, and moderate reliability. Even substantial improvements are unlikely to give them a position competitive with drill-and-blast methods for the purpose of producing storage volume (Refs. III-2 and III-5).

A mechanical breakage system which is not tied to a rotary tunneling machine and shows some promise is hydraulic splitting in series with percussion (Ref. III-14). Hydraulic splitters force a metal plug into a predrilled hole and split rock toward an unconfined face in a manner similar to conventional blasting. The vibrations associated with blasting, however, do not occur with splitting, and this has enabled the hydraulic splitter to be used recently on tunneling projects in downtown areas of San Francisco, St. Louis, and Washington.

Coupling the splitter with a percussive thrust (e.g., from a jackhammer) gives the splitter an enhanced rate of production. In large headings a multiple splitter has the potential to be twice as fast as conventional drill-and-blast methods, partly by virtue of needing fewer drill holes. Given sufficient research funding, the multiple splitter could become commercially viable by about 1979 (Ref. III-14).

Shaft Excavation Methods

Shaft excavation methods can be divided into three types:

- conventional shaft sinking
- shaft drilling
- shaft raising.

Both the conventional sinking and drilling methods can be used to create the first shaft in a project. Raising requires an existing cavity below ground from which to start the raise, and a separate shaft for spoil removal and ventilation. All three techniques might have an application in the CAPS project, depending on the rock conditions. Good rock conditions favor drilling and raising small-diameter shafts, although poor conditions can be catered for with these methods at some extra cost for pregrouting and considerable extra cost for heavy steel linings. In very poor rock conditions there may be no choice but to sink a larger shaft by conventional methods.

Conventional Shaft Sinking

Conventional shaft sinking methods involve a drill, blast, and muck cycle similar to the cycle used in drill-and-blast excavated headings. A stage from which the drilling, mucking and lining operations are performed is winched down the shaft on cables as excavation proceeds. Drilling at the bottom of the shaft is by hand-held machines or a drill jumbo especially designed for shaft work.

Figure III-5 shows two of the more common methods of excavating large-diameter shafts. In the first method the shaft is sunk, either in full-face or half-face rounds, using similar techniques to those used in heading excavations. Muck is removed and hoisted upward. In the second method a pilot shaft is drilled from the ground surface to an existing cavern, or raised to the ground surface from an existing cavern by either raise-boring drills or self-propelled raise climbers. This method requires that prior access has been gained to the underground by some method. The pilot shaft can then be enlarged to the required size by a form of bench blasting from the ground surface down (known as slashing), with the excavated material allowed to drop to the existing cavern for mucking. The second method is usually cheaper than the first, and is suitable for provision of additional shafts after some underground development has taken place.

Where water-bearing strata are anticipated ahead of the work face (based on test boring information), these strata are drilled and grouted before the blast holes are drilled. After blasting, the broken rock is loaded into muck skips, typically using a Cryderman mucking unit, which is a hydraulic grab mounted on cables suspended from the stage. The muck skips are removed using an independent mucking hoist.

A poured concerete shaft lining (typical one foot thick) can be installed from the stage in approximately 20-ft sections, keeping approximately 60 ft above the shaft bottom. Care has to be exercised during the blasting to prevent excessive ground vibrations. These vibrations can damage the concrete lining and fracture pregrouted strata, causing an increase in ground water inflow.

If the shaft has to start in poor overburden conditions, it is usualto either freeze or grout ahead from the ground surface. A convenient practical limit for this operations is about 200 ft. Beyond that depth the grouting or freezing might have to be performed from the shaft, and even then might not be feasible due to high groundwater pressures and an unstable workface in weak materials.

The optimum shaft diameter for minimizing the time and cost per foot of depth of construction is about 15 ft. Below this size, and particularly below 8 ft in diameter, the reduction in working space slows work considerably. Above 15 ft in diameter, and particularly above 25 ft, the increased diameter leads to ground support problems and to considerably more expense, due to the volume of rock to be drilled, blasted and removed.

Typical rates of progress are about 40 ft per week, although higher rates are possible, given favorable conditions.

Shaft Drilling

Shaft drilling, using either fixed or mobile rigs similar to those used for drilling oil wells, is well established in connection with mining LPG storage caverns (Refs. III-15 and III-16). For these caverns the shaft depths have ranged typically from 300 to 2,000 ft, with a grouted steel lining of 42 in. in diameter inside a drilled hole of 52 in. in diameter. An extreme example of the scale of drilled shafts is one constructed for the United States Atomic Energy Nevada Test Site. This shaft was drilled in 1963 with a diameter of 10 ft to a depth of 6,000 ft, and lined with a ribbed steel lining (Ref. III-15).

The drilling, lining and grouting operations have usually been performed with drilling mud in the hole to ground level. Mud is circulated in the hole and rock cuttings removed from the mud at ground level using special separation equipment. The mud is returned to the hole. The lining is lowered into the hole and welded together in sections to give a full length of lining suspended from ground level. Grouting is performed from the bottom of the shaft upward, between the lining and the rock walls, displacing the drilling mud upward. When the grout has set, the drilling is then pumped out.

The maximum size of shaft which could be drilled to the depths required for CAPS application is approximately 10 ft in diameter. Drilled shafts with a smooth finished diameter of about 6 ft would probably be sufficiently large for the air discharge line. Depending on rock and groundwater conditions, this could be achieved in a number of ways.

In hard competent igneous or metamorphic rock of low permeability, the shaft could be drilled at 6 ft in diameter and have smooth walls (1/4- to 1/2-in. roughness and no blast damage) with no lining required. The drilling time for a 6-ft shaft through granite to approximately 1600 ft would be approximately two months, making allowance for plant setup and minor delays. However, even in such a case a lining might be required to satisfy safety regulations. If the rock were of low permeability throughout the entire depth (this would have to be proved by means of an exploratory borehole and packer testing), a thin steel lining with grouting between the lining and the rock wall might be adequate. A grout annulus of 6 to 12 in. would be required to allow enough space to grout pipes 1,600-ft long. The hole would therefore have to be drilled about 8 ft in diameter. The major uncertainties with this approach are how much water pressure would build up behind the lining and whether sufficient water could pass through the rock to cause the lining to buckle. Drilling the 8-ft shaft is estimated to take about three months, and lining and grouting about two weeks.

A more certain approach in competent low-permeability rock is to install a concrete lining about 12 in. thick by slip forming or jump forming, working from the bottom of the shaft upward. A precedent for this method is the installation of a 6-in. lining in a 12-ft shaft, 600 ft deep, at Lancashire No. 20 Mine, Carrolltown Pa. (Ref. III-17). This particular installation proceeded at an average rate of 18 ft per day. At this rate, a 1,600-ft shaft would take 88 working days or about four months. The concrete lining could be allowed to crack under the water pressure which might build up without any major dislocation in the lining surface. Given the low-permeability condition assumed, the rate of groundwater inflow would be low.

It is possible to conceive of installing a thick steel lining in the same way as the thin lining, taking about the same time. This lining could be thick enough to resist the full possible groundwater pressures. However, the cost of the lining for a shaft 1,600 ft deep would be extremely high.

For a shaft drilled in sedimentary rocks, a lining would be required either to retain and protect the weaker rock types (e.g., shale, coal evaporites) or to retain or control groundwater, oil, or gas inflow from aquifers. Where specific thin aquifers have been identified by the test drilling, the use of a thin steel lining could be considered. After installation and grouting, the lining could be drilled using oil well perforating equipment, under mud, in the aquifers allowing relief of water pressure. However, there would be risk of considerable water inflow upon removing the mud, and the amount might be unacceptable. To overcome this problem the shaft could be pregrouted throughout from the ground surface. This was done in the case of the 600-ft shaft described above, in which a concrete lining was installed. This could well be a time-consuming and expensive operation. Again, given pregrouting, a concrete lining could be installed as described above. Finally, the ultimate solution of an expensive thick-walled steel lining could be adopted without pregrouting, installed under mud.

Drilling rates in the sedimentary rocks would tend to be considerably higher than in the harder igneous and metamorphic rocks, and this could offset the time and cost of pregrouting. It should also be pointed out the pregrouting might be necessary in igneous and metamorphic rocks if unfavorable fissure and joint patterns occur.

Shaft Raising

Shaft raising from an existing cavern underground shows considerable economy in comparison to conventional sinking or drilling shafts. In the CAPS project, only a relatively small shaft, approximately 6-ft finished diameter, would be required in addition to the main access shaft. Two basic methods exist for raising this size of shaft: "Alimak" raising and raise boring. Larger shafts are usually slashed down from ground level following a pilot raise of this size.

The Alimak raiser is a self-propelled access platform from which overhead drilling can be performed prior to blasting or hydraulic splitting. It can climb up an incline, or climb vertically on a rack-and-pinion mechanism using track fixed to the shaft walls. Motive power can be air, electric, or diesel. For a raised shaft 1,600-ft high, diesel power would probably be used, requiring ventilation. A shaft of any cross section can be excavated, but assuming a 6- to 8-ft square or circular shaft, a progress rate of 100 to 150 ft per week is quite feasible.

Raise boring is performed from the ground surface and, unlike the "Alimak" system, men are not required to be in the shaft during construction. This is an advantage in respect of safety and ventilation requirements. A pilot hole (typically about 14 in. in diameter) would be drilled from the ground surface to the existing cavern. A drilling body with roller cone bits would then be attached to the drill pipe, pulled upward, and rotated by the rig, thus raise boring the shaft to full size. The large-diameter shafts would be drilled out by two drilling bodies of different size, the smaller one leading and the larger one drilling to full size in one operation. The pilot hole could be drilled at an overall rate of about 100 ft per day, and a 6-ft shaft could be raised at about the same rate in sedimentary rocks. In the harder igneous and metamorphic rocks, drilling the pilot and raising the main shaft would take typically 40 percent longer (Refs. III-16 and III-18). Allowance must also be made for rig setting up time. The broken rock fragments or rock cuttings from both shaft raising techniques fall by gravity to the bottom of the shaft where they would be mucked away for removal through the access shaft.

In permeable water-bearing rock, a pregrouting program might be necessary for both methods of raising, and a grouted steel casing would be required through unfavorable overburden. A slip or jump-formed concrete lining could be constructed after completion of the raise.

Spoil Handling

Spoil handling in underground workings can conveniently be divided into two main operations. The first operation is the removal of blasted rock from the work face, and the second is the transportation of the rock to the ground surface. An intermediate operation which might be required would be crushing the blasted rock to a smaller size for efficient charging of muck skips or for pneumatic removal. At the ground surface the rock might be used for construction within the project, e. g., rock-fill dikes for surface reservoirs or trucked away and sold as aggregate. In both of these cases the rock would realize some value. In some areas there might be no need for dike rock fill and insufficient demand for aggregate, in which case a cost penalty would be incurred for disposal.

A discussion of the two main handling operations follows. ~~In summary, it is~~ concluded that mucking would be most efficient by means of diesel-powered, rubber-tired, front-end loaders with a bucket capacity of 3 to 5 cubic yards. These loaders would load, haul, and dump directly into the crusher. Crushed rock would probably be taken to ground level with a high-speed hoist.

Removal of Blasted Rock from Work Face

Systems for moving blasted rock from the work face (mucking) to the crusher or to haul trucks, include conveyor belt systems, rail track systems, and self-propelled diesel- or air-powered rubber-tired loading or load/haul units, in approximate order of increasing mobility. In the CAPS project, the main requirements are assumed to be a high rate of extraction of (in mining terms) a modest volume, with short-haul distances (up to 700 ft) in large cross section caverns. These requirements tend to favor the more mobile systems.

The conveyor belt and rail tracked systems are more suitable for either permanent or semipermanent installations over long-haul distances (e. g., mines) or for confined working in long tunnels or adits. Neither of these conditions will apply in the CAPS caverns.

Self-propelled rubber-tired loading units could be diesel or air powered and take the form of front-end loaders or overhead loaders, respectively. The air-powered overhead loader is usually the mobile part of a conveyor belt or rail-tracked hauling system and as such would probably not be used for the air storage cavern excavations.

The diesel-powered front-end loader could be used as a loader for a diesel haul truck or as a load-haul-dump unit from work face to crusher. Because of the short-haul distances involved and shaft access, it is probable that they would be used as load-haul-dump units. Typical front-end loaders have a bucket capacity of 3 to 5 cubic yards (broken rock) and an engine capacity of 170 to 240 bhp. They could be disassembled and taken down a 6-ft shaft, if necessary. It is estimated that three to four of these units working continuously for 10 hrs per day could load, haul, and dump into the crusher approximately 3,000 cubic yards (4,000 tons) of broken rock per day. This rate is sufficient to remove 250,000 cubic yards of cavern volume (400,000 cubic yards of broken rock) in approximately 7 months. This rate is quite realistic for multiface working when broken rock would always be available for mucking throughout a 10 hr shift. Alternatively, the same overall rate could be achieved by increasing the number of shifts and reducing the mucking rate per loader or number of loaders.

An alternative to the front-end loader is the diesel-powered auto loader. This machine combines the function of an overhead loader with body storage capacity of about 6 cubic yards. It is designed for load-haul-dump operations in relatively confined drifts and tunnels. Since the loading bucket must operate several times to fill the body, it is slower at loading than the front-end loader, which is also a more common and available item of construction plant. For these reasons the front-end loader (Fig. III-3) would probably be preferred to the auto loader.

Transportation of Rock to Ground Surface

Three alternative methods may be considered for transportation of rock to the ground surface: haulage by diesel truck up an inclined access tunnel, pneumatic hoisting, and high-speed mechanical hoisting. Although the use of diesel trucks and inclined tunnels is common for large cavern excavations such as hydroelectric powerhouses and oil storage caverns, the access tunnel rarely has to climb through a vertical distance of more than about 500 ft. The economic cutoff for this method is in the order of 700 ft (Ref. III-19), depending on the nature of the excavation and for the greater depth of an air storage cavern. This method is considered to be uneconomic. In sedimentary sequences of shales, coal, sandstones and evaporites, such as occur in the north central study area, a shaft would be considerably easier to sink and maintain in a stable condition than an inclined tunnel.

Pneumatic hoisting is a relatively recent technique which has been used to hoist material at approximately the rates required for the air storage cavern excavation (Refs. III-20 through III-22). Crushed rock is fed into a feeder unit through which is blown a jet of high-velocity, low-pressure air. The delivery pipe is fed from the feeder unit up a vertical shaft to ground surface, where the rock is discharged into a hopper or a stockpile. It has been estimated (Ref. III-21) that a 200 ton per hr system would require an 18-in. delivery pipe and a 2,000 hp blower unit. This power rating is comparable to mechanical hoisting. If this system were operational for 18 hr per day, it could hoist the equivalent of 250,000 cubic yards of intact rock in about 150 working days (7 months), matching the mucking capacity of the front-end loaders. All the components for a pneumatic hoisting system could be taken down a 6-ft diameter shaft with minor disassembly (Refs. III-21 and III-22). The air consumption would be on the order of 20,000 cfm, which could make a substantial contribution to the ventilation requirement.

High-speed mechanical hoisting is a well-developed technique in mining application. An extraction rate of 20,000 tons per day per hoist is feasible using a 16-ft shaft. In this size of shaft, the mucking hoists, man conveyance, water, compressed air, electrical and ventilation ducts could all be included. However, this extraction rate is far in excess of what would be required for a 250,000 cubic yard cavern and it might be more appropriate to consider small drilled or raised shafts and a reduced hoisting capacity. It is estimated that using a single 5-ft diameter skip, 15-ft deep (15-ton capacity), making twenty-four round trips per hr, an extraction rate of 3,000 cubic yards of crushed rock per 10-hr day could be achieved through a 6-ft diameter shaft 1,600-ft deep, using a maximum hoist speed of 1,600 ft per minute. This rate is well within the capabilities of modern capstan hoisting equipment, and is sufficient to match the mucking capacity of the front-end loaders of 400,000 cubic yards of broken rock in a 7-month excavation period.

Both the pneumatic and mechanical hoisting systems need the blasted rock to be passed through crushers to achieve optimum system efficiency. Manufacturers' data for crushers designed for underground use show that, with minor disassembly, a primary crusher with a capacity of 400 ton per hour could be taken down a 6-ft diameter shaft (Ref. III-23).

Ventilation

During the cavern construction there will be a demand for ventilation to remove fumes from diesel engines, blast gases, and duct from the blasting operations. Using a combination of statutory regulations and previous experience related to underground working, it is possible to estimate the ventilation demand, and the effect of this demand on the access/ventilation shafts.

The OSHA requirements with respect to tunnel and shaft construction are... "that the supply of fresh air shall not be less than 200 cfm for each employee underground". Also, "the linear velocity of air flow in the tunnel bore shall not be less than 30 ft per min in those tunnels where blasting or rock drilling is conducted, or where there are other conditions that are likely to produce dusts, fumes, vapors, or gases in harmful quantities". Basic operating mine ventilation requirements adopted by the Ministry of Natural Resources of Ontario, Canada are for 75 cfm/bhp of diesel-powered equipment and a minimum air velocity of 50 ft per min. This latter figure is generally applicable in tunnels or adits with a cross section up to 200 square feet. For air storage caverns which would have cross-sectional areas on the order of 5,000 square feet, a considerably lower velocity would probably be satisfactory. Threshold limits for concentrations of air contaminants are specified in OSHA and other codes. These limits should not be unduly restrictive with ventilation capacity designed to satisfy the above criteria.

Recent experience in the underground ventilation practice includes an allowance for approximately 100 cfm/bhp of diesel-powered plant at the Churchill Falls project, with allowance for personnel included. At a salt mine in Louisiana, a total of 240,000 cfm is supplied at the whole mine through 9-ft diameter drilled and lined shafts 800-ft deep, using 250 hp for the fans. In this mine there is approximately 1,000 bhp of diesel-powered equipment in operation, giving an equivalent of 250 cfm/bhp. At an underground propane storage cavern near Philadelphia, whose total storage volume is similar to that envisaged for the air storage caverns the ventilation capacity during construction was 53,000 cfm. Ventilation shafts consisted of one 60-in. and three 20-in. lined shafts 400-ft deep. The diesel-powered plant used in this operation had a maximum total engine capacity of 615 bhp giving an equivalent of 86 cfm/bhp.

From the above statutory regulations and practical experience, it is estimated that the ventilation requirement for the air storage cavern construction will be in the range of 50,000 to 100,000 cfm, assuming a total of 500 to 600 bhp of diesel-powered plant and some 20 persons working underground. Assuming a flow circulation of 50,000 cfm through two ventilation shafts 6 ft in diameter, with rough walls, 1,600-ft deep, and connected by large caverns, the air velocity in the shafts will be approximately 30 ft per sec and it is estimated that the fans will have to be of approximately 200 hp. This estimate makes allowance for one shaft partially obstructed with a hoist skip. Without the skip the power demand would be about 150 horsepower.

If the shafts are lined throughout (relatively smooth) the power requirement would be reduced to about 100 hp (75 hp without the hoist skip). With an air flow rate of 100,000 cfm and lined 6-ft diameter shafts, the air velocity would increase to about 60 ft per sec and the power demand to 700 hp (370 hp without the hoist skip). The use of larger access/ventilation shafts would reduce the power demand. A 50-percent increase in shaft diameter would cause a decrease in shaft friction power demand by a factor of about 5, given the same wall roughness and air flow rate.

From the above figures, it can be concluded that adequate ventilation could be achieved through 6-ft diameter or larger shafts without an excessive power demand. Lining the shafts would be most advantageous in reducing power demand and might well be a plant requirement for other reasons (e. g., prevention of groundwater inflow, prevention of air leakage and shaft safety). Ventilation shafts less than 6 ft in diameter should be lined, and might be suitable only for flow rates of up to 50,000 cfm.

Because of the cavern configuration of large, cross-sectional parallel tunnels, an increase in storage capacity by lengthening the tunnels would require little additional ventilation, provided that the same level of diesel equipment bhp is used during the excavation.

As an alternative to twin 6-ft diameter shafts, a single 6-ft shaft could act as ventilation supply and muck hoisting shaft, and two or more smaller ones could act as extraction shafts. It should be noted however, that at least two shafts must be equipped to provide personnel access/escape routes. A shaft smaller than about 4-ft diameter would be unsuitable for this requirement. However, at the very early stages of the cavern excavation it might be advantageous to have a smaller diameter ventilation shaft in addition to the 6-ft access/ventilation shaft. The latter would probably have insufficient cross section to allow the simultaneous use of an adequately sized hoist skip and a ventilation delivery pipe (assuming the ventilation return air goes back up the hoist skip portion of the shaft). For example, a 2-ft lined shaft could act as a ventilation extraction or delivery shaft of modest capacity, sufficient for air-operated excavation equipment or a reduced scale of diesel-powered equipment.

SPECIAL ASPECTS OF COMPRESSED AIR STORAGE

This section contains discussions of cavern air leakage, including methods of leakage control, and temperature effects on rock stability.

Air Leakage

The following discussions explore the implications of, and possible solutions to, air leakage from constant-pressure air storage caverns situated at a depth of 1,300 to 2,500 ft below ground level, at a pressure of 40 to 75 atmospheres, and balanced by a water column. The discussions focus on:

- Leakage from unlined caverns,
- Leakage from unlined caverns with a water curtain,
- Grouting, and
- Comparison of leakage prevention methods.

Most of the references cited are of Scandinavian origin, where considerable work has already been done in theory and in practice on the storage of fluids underground.

Previous experience, preliminary calculations, and recourse to references show that the rock permeability for air (k_a) at CAPS cavern depths is likely to be 10^{-4} cm/sec or less in competent rocks such as granites and massive limestones. It is emphasized that many exceptions to this generalization exist for certain rock types, such as sandstones, and for particular structural conditions, such as intensely faulted or jointed rock. Using probable storage cavern layouts, a rock mass permeability of this value implies a leakage rate of between 1 and 2 percent per day of the stored mass of air. This would be in addition to a conservatively estimated loss of 2 percent through absorption of air into water with the water-balanced, constant-pressure type of system. The total leakage rate of 3 to 4 percent is economically acceptable.

Present techniques for measuring in-situ rock permeabilities for water (k_w) are not capable of defining permeabilities lower than 10^{-6} cm/sec, except to indicate an "impermeable" rather than a permeable rock. This would, however, be adequate information for the selection and design of an air storage facility. It is sufficiently accurate to assume that the viscosity of water is a factor of 100 greater than the viscosity of air and that the permeability is inversely proportional to viscosity. This means that a rock fissure network having a permeability of 10^{-6} cm/sec for air (k_a).

The use of a water curtain, which is effective in the permeability range of 10^{-3} to 10^{-6} cm/sec (k_w), should be considered as a contingency item to be used if the rock mass were found to be substantially more permeable than the exploration data had initially indicated. Given a rock mass permeability (k_w) of 10^{-6} cm/sec or less, the only condition under which a water curtain might be used is if the ground surface conditions were particularly sensitive to air leakage.

Grouting would probably be restricted to treatment of localized permeable zones and for the lining of the air and water pressure shafts.

Leakage From Unlined Caverns

Analytical Models

For the purposes of obtaining order-of-magnitude values for the leakage rates, the use of relatively simple mathematical techniques can be justified. A general approach is required without the input of detailed geological and geotechnical parameters. Three relatively simple analytical methods have been used so far, and these are due to Barton (Ref. III-24), Janbu and Tokheim (Ref. III-25), and Berg and Noren (Ref. III-26). Barton's approach was evolved in conjunction with laboratory model studies of single-planar rock joints, and correlated with some field tests. The tests, by Di Biagio and Myrvoll (Ref. III-27) measured and compared flow rates and transport times of water and air injected into boreholes drilled in granite. The permeabilities measured were reduced to effective fissure widths and spacings according to the method developed by Snow (Ref. III-28). These widths and spacings were used in Barton's equations to predict the leakage rates and transport times, which compared quite well with Di Biagio and Myrvoll's field results.

Barton's expression for water flow from a circular tunnel or borehole at depth through a single vertical fissure to a horizontal planar ground surface is as follows:

$$Q_w = \frac{\pi b^3 g(P_r)}{6\mu_w \log_e (2D/r)} \qquad \text{(III-1)}$$

where
- Q_w = volumetric flow rate
- b = fissure width
- g = acceleration due to gravity
- P_r = excess pressure in opening
- μ_w = dynamic viscosity of water
- D, r = see Figure III-6A.

This expression can be used for air flow by altering the value of viscosity to that of air and interpreting the calculated flow rate as that occurring at the algebraic mean absolute pressure of the flow domain.

Janbu and Tokheim's approach is to use the mass permeability k (cm/sec) [or intrinsic permeability K (cm^2), in conjunction with the relevant viscosity values] for air leakage from underground openings of variable geometries to a planar or cylindrical ground surface or other permeable boundary. Their expression for water flow rate through the rock mass (or intersecting fissure network) to the ground surface is:

$$Q_w = \frac{2\pi K L P_r}{\mu_w G} \qquad \text{(III-2)}$$

and for air flow rate is:

$$Q_a = \frac{\pi K L P_o}{\mu_a G} \left[\left(\frac{P_s}{P_o}\right)^2 - \left(\frac{P_e}{P_o}\right)^2 \right] \qquad \text{(III-3)}$$

where
- K = intrinsic permeability
- L = length of opening
- G = shape factor
- μ_a = dynamic viscosity of air
- P_o = reference pressure
- P_s = air pressure in opening
- P_e = air pressure at ground surface.

For one-dimensional flow

$$G = 2\pi \frac{D}{B} \text{ (Figure III-6B).} \qquad \text{(III-4)}$$

For three-dimensional flow

$$G = \log_e \frac{(2D-r)\ (L+2r)}{[L+2(2D-r)]\ r} \text{ (Figure III-6C).} \qquad \text{(III-5)}$$

For infinitely long circular tunnels, Eqs. III-1 and III-3 give the same expression for air flow rate if the mass permeability of rock is related to the fissure width and spacing by means of the following expression due to Snow (Ref. III-28):

$$k_w = \frac{1}{12}\,\frac{b^3}{S}\,10^{-7}\ \text{m/sec} \tag{III-6}$$

where b = fissure width (m)
S = fissure spacing (m).

Note that:

$$K = \frac{k_w\,\gamma_w g}{\mu_w} \tag{III-7}$$

where

γ_w = unit weight of water .

Berg and Noren use flow nets to predict air leakage rates. This is an extension of conventional flow nets for water flow through porous media with an allowance for the compressibility of the air. Figure III-7 is based on data contained in the paper by Berg and Noren (Ref. III-26). It is not clear from their published work how the flow nets were derived. However, a computer program for producing gas flow nets should not be difficult to write, and this should be considered at a more advanced stage of the project.

Both Barton and Janbu give expressions for the leakage transport times for the steady-state and transient conditions. These are of limited interest to CAPS and are discussed in the section of leakage through initially saturated rock.

Leakage Through Dry Rock

Applying the three analytical methods discussed above to the example given by Berg and Noren (Ref. III-26) of an infinitely long tunnel (two-dimensional flow) of 2,200 sq ft cross section, at a depth of 530 ft, an air pressure of 16 atm and a rock mass permeability for air (k_a) of 10^{-4} cm/sec, gave the following approximate results for air leakage:

Method	Air Leakage Rate
Barton (Ref. III-24)	5.8 percent of volume per 24 hours
Janbu and Tokheim (Ref. III-25)	5.8 percent of volume per 24 hours
Berg and Noren (Ref. III-26)	5 percent of volume per 24 hours

The air leakage rate is the volume of air leaking in 24 hours from the storage volume, corrected to the storage pressure. For this problem there is little to choose between the methods. If the storage depth is increased and the storage pressure correspondingly increased, inspection of Eqs. III-3 and III-4 shows that the leakage rate will remain the same, since the storage pressure and leakage paths are increased by the same ratio, and all other factors remain constant.

If one compares, for example, the relative merit of storage of the same daytime generating capacity situated at either 1,300 or 2,000 ft depth (40 or 60 atm), the above argument shows that the leakage rate (as defined) is the same. The higher storage pressure at the deeper location implies a greater energy loss but, balancing this, the upper location needs a greater storage volume for the same mass of stored air (PV = constant at constant temperatures). It can be assumed that the geometries of caverns at 1,300 and 2,000 ft depths will be essentially the same, and, hence, the storage volume will be directly proportional to the storage plan area. The total amount of leakage would also be proportional to the plan area, so the mass of air lost for each cavern would be the same for the same overall permeability. Within the same rock type, the permeability will usually reduce (in some cases significantly) with depth, and this factor will tend to favor deep siting of the cavern to reduce leakage rates.

Leakage Through Initially Saturated Rock

For constant-pressure underground air storage balanced by a water column, the air pressure will usually be in excess of the groundwater pressure in the rock adjacent to the cavern roof. (An exception might be storage below a topographical dome with the surface reservoir located to one side of the dome.) Under these conditions, the air would advance (leak) upward through the rock fissures, displacing the water to the side. Noren et al (Ref. III-29) show that it takes only 2 weeks or so for the water to be displaced from the rock fissures for a storage depth of 1,300 ft and a rock mass permeability (k_w) of 10^{-6} cm/sec. For a rock permeability of 10^{-7} cm/sec, the displacement time under the same conditions would be less than half a year.

Thus, an air flow channel would be formed very rapidly between the storage cavern and ground surface, confined laterally by the displaced groundwater. Because of this confinement, the operating (stable) leakage rate would be less than for leakage through dry rock. For the same example used above, the air leakage rate would be about 2 percent for initially saturated rock compared with 5 percent for dry rock, based on the flow net examples of Berg and Noren (Ref. III-26).

Effect Of Cavern Geometry On Leakage

If a grid of tunnels were used instead of a single-tunnel, two-dimensional model, the air flow would tend toward one-dimensional flow from the plane of the tunnels to the ground surface. Berg and Noren (Ref. III-26) state that the leakage rate under these conditions, after the groundwater has been displaced, would be about 1.25 percent of the stored volume per day, compared with 2 percent for the single tunnel in saturated rock. The absolute values of the leakage rates in a one-dimensional flow situation would depend on the plan layout of the tunnels and the cavern height.

It can be assumed that any grid of tunnels would utilize the full plan area of rock within and directly above the perimeter of the grid as a flow channel. Thus, if the storage volume could be maximized within this perimeter, the leakage rate expressed as a percentage of the stored volume would be minimized. For example, parallel tunnels spaced at a center to center distance equal to three times their breadth (B) have an area ratio of 33 percent, the area ratio being defined herein as the plan area of the cavern openings divided by the total plan area within the grid.

If these tunnels were intersected at right angles by another set at spacings of 3B, the area ratio would increase to 55 percent. A room and pillar excavation using 20-ft square pillars and 30-ft clear openings would have an area ratio of 84 percent. For the same storage volume, leakage rates are inversely proportional to the area ratio, so the effect of the plan layout of the tunnels is evident.

Doubling the cavern height for the same storage volume (i. e., halving the total grid area) would halve the leakage rate. Berg and Noren's figure of 1.25 percent is for an intersecting tunnel grid of about 3B spacings with the tunnel height about 1B. Hence, for the examples above with heights of 1B and 2B, the leakage rates would be:

Leakage Rates, %/day

Tunnel Height	Parallel Tunnels at 3B Center to center	Intersecting Tunnels at 3B center to center	Room and Pillar AR = 84 percent
1B	2.0	1.25	0.81
2B	1.0	0.62	0.40

The leakage rates are expressed as a percentage of the total volume of contained air leaking out in 24 hours, at the cavern storage pressure.

Effect Of Geology On Cavern Geometry

It can be assumed for any CAPS project that a very competent rock mass would selectively be chosen for the air storage caverns. Such rocks as massive limestone, dolomites, granites, or granite gneisses would be particularly favorable. With such rocks it should be possible to maximize the area ratio and height (see previous section) without increasing the costs of roof and pillar (or wall) support excessively (probably only spot bolting would be required). However, there is obviously a trade-off here in terms of the benefits of leakage reduction against cost of support, and this should be investigated in more detail when the probable costs implied by certain levels of leakage rate are known. It may well be more economical to go for the more stable configuration of parallel tunnels with the minimum of support costs. Also, the use of a water curtain might be more economical than maximizing the area ratio and cavern height.

Effect Of Geology On Permeability Values

At present there is very little data available on the permeability values of relatively impermeable rocks, particularly at depths greater than about 500 ft. The importance of obtaining reliable permeability values is obvious, although it should not be overstated. Conventional interval packer tests are capable of identifying permeabilities (k_w) of less than about 10^{-5} cm/sec and possibly 10^{-6} cm/sec. Any permeabilities lower than 10^{-6} cm/sec appear to have no serious implications for the leakage rate, and the packer tests would at least identify the length over which no measurable amounts of water could be injected, i.e., k_w less than 10^{-6} cm/sec. Other methods for determining these low in-situ permeability values should be investigated. Permeability testing of intact core in the laboratory would give information of in-situ permeabilities of porous rocks only, since water or air-carrying fissures are generally at rather wide spacings. A borehole camera could identify only fissures implying the higher values of permeability (greater than 10^{-5} cm/sec) and might not be usable at 1,600-ft depth. Packer tests with air have potentially insuperable leakage problems at 1,600-ft depth and geophysical methods cannot give the permeability information required (Ref. III-30). Water curtains have an effective range from about 10^{-3} to 10^{-6} cm/sec (see subsequent discussion) and could be usefully held in reserve as a contingency in case rocks which were expected to have permeabilities of the order of 10^{-6} cm/sec or less proved to be more permeable during the excavation of the facility.

The discussion of two possible geological situations has some merit in illustrating the effect of geology on cavern geometry.

Case 1 - shale/limestone sequences

A massive, nonkarstic limestone will probably have a permeability (k_w) of less than 10^{-6} cm/sec. Clay shales which are commonly found interbedded with limestone and sandstone in north-central United States (e.g., Niagra Gorge) are unlikely to have permeabilities any greater than those of typical clays. The permeabilities (k_w) of clays usually range from the order of 10^{-7} to 10^{-9} cm/sec. A 1,300-ft thick depth of material with an average k_w of 10^{-6} cm/sec would have an air leakage rate on the order of 1.25 percent for one-dimensional flow from a cavern at 40-atm pressure. If there were a 70-ft thick layer of shale within the 1,300 ft, with a k_w of 10^{-7} cm/sec, the leakage rate would be reduced by a factor of about 0.7 to 0.88. If the 70-ft thick layer had a k_w of 10^{-8} cm/sec, the reduction factor would be about 0.17, giving a leakage rate of 0.21 percent. This is obviously a hypothetical example, but illustrates the benefit of a relatively impermeable natural layer (or layers) between the cavern and gound surface.

Case 2 - granite batholith

Berg and Noren (Ref. III-26) use a permeability value (k_a) for granite of 10^{-4} cm/sec for the calculation of air leakage rates from storage at 500-ft depth. Measurements of air leakage from boreholes at an 80-ft depth by Bernell and Lindbo (Ref. III-31) in a very tight granite imply a permeability value (k_w) of 2×10^{-7} cm/sec. Snow (Ref. III-28) provides data which show that typically, the fissure width in a wide variety of competent rocks at about 300-ft depth is 50 microns or less, and the effective fissure spacing below this depth is not less than 15 ft. Using these values in Eq. III-6 gives a permeability value (k_w) Of 0.23×10^{-5} cm/sec. Snow also shows that a marked decrease in fissure width and increase in spacing occurs in the interval from 0 to 300 ft. It should be noted that, although Snow's data are from specific sites, the conclusions when applied at new sites must be of a very general nature. If these data were extrapolated to 1,600-ft depth it might well be assumed that the permeability value (k_w) would reduce to the order of 10^{-7} cm/sec. A reduction in permeability of this magnitude (10^{-6} cm/sec near ground surface, 10^{-7} cm/sec at 1,600 ft) would reduce the leakage rate, in comparison with 10^{-6} cm/sec throughout, by a factor of about 0.2. Again, using the one-dimensional figure of 1.25 percent for a k_w of 10^{-6} cm/sec, this gives a leakage rate of $1.25 \times 0.2 = 0.25$ percent.

Effect Of Leakage On Surface Soils And Rocks

This is not a major item for consideration. There are measures which could be taken to reduce the harmful effects of uncontrolled air leakage through the surface soils (see Berg and Noren [Ref. III-26]). Ultimately, if the problems proved insuperable, a water curtain could be used. It is recognized that this factor is an environmental question which could arise, and further thought should be given to the question at an early stage of the design.

Leakage From Unlined Caverns With A Water Curtain

The patented water curtain system (Ref. III-32), in combination with grouting very permeable zones, is theoretically capable of preventing all air leakage from the storage cavern through the surrounding rock at the cost of a minor amount of water inflow. The curtain would enable rocks of a higher permeability to be considered for the siting of the storage cavern, and could be utilized to prevent the drainage (drying out) of the rock mass above the caverns, during construction.

Water Curtain Configuration

A typical water curtain configuration for use in gas or air storage is shown in Figs. III-8 and III-9, taken from the paper by Skanska and Hagconsult (Ref. III-33). Small cross-sectional tunnels would be driven above the main storage caverns. From these tunnels a series of horizontal parallel boreholes would be drilled at a fairly close spacing to provide a complete coverage in plan of the storage area. The spacing of the holes would be determined by the spacing, orientation, width and nature of the rock joints and fissures.

The tunnel system would then be sealed off with concrete plugs or bulkheads and filled with water. An overpressure would then be applied to the water in the tunnel/borehole system, sufficient to cause flow toward the storage cavern. The water supply/overpressure system could consist of a delivery pipe from a water tower above the surface reservoir level to the tunnel bulkhead.

Geological Conditions Under Which A Water Curtain Might Be Utilized

Water curtains would normally be used in the natural rock, untreated with grouts and without cavern linings. It is suggested by Noren, et al (Ref. III-29) that the economical range of rock permeabilities (k_w) suitable for accepting a water curtain system is from 10^{-3} to 10^{-6} cm/sec. For permeabilities lower than this range, the leakage rates would be quite low (less than 1 percent), and a water curtain would not give any significant savings through reduction of leakage rate.

For the curtain to be wholly effective, there must be good interconnection between fissure and joint systems in the region of the pressurized tunnels and the storage cavern. If the fissures are partly filled with secondary minerals, e. g., clay, calcite, and carbonate solutioning deposits, these fillings could create a "spaghetti"-like network of water conducting channels. The borehole network might easily thread through the channels without achieving an adequate number of intersections. In this way, air flow in the channels could leak past the water curtain. To overcome this problem it might be feasible to conduct a systematic hydraulic fracturing program to improve the joint and fissure interconnection, or to use the water curtain in connection with systematic grouting.

Flow Rates Of Water Required To Maintain A Water Curtain

The theoretical minimum potential gradient of water flow from the curtain to the cavern storage is about 0.4 for laminar flows, which are implied by the permeability range (for k_w) of 10^{-3} to 10^{-6} cm/sec.(Noren et al Ref. III-29). If turbulent conditions occur in more permeable zones, the minimum gradient would increase to about 0.8. These values are determined by the ability of the curtain to prevent air rising through the bedrock groundwater (within the fissures) in the form of bubbles.

A reasonable design value for the potential gradient of a water curtain system would be about unity. The potential gradient of the air flow from an unlined cavern through homogenous rock pressurized by a water column is also on the order of unity in the near vicinity of the cavern. Thus, the air flow rate from the cavern without a water curtain is related directly to the water flow rate to the cavern with a water curtain by the relative viscosities of the two fluids. (Air is approximately 100 times less viscous than water at ground temperatures.)

Using the leakage figure of 1.25 percent in rock of $k_w = 10^{-6}$ cm/sec leads to the following leakage comparison:

Rock Mass Permeability to Water, cm/sec	Air Leakage Rate, No Water Curtain, %	Water Inflow from Water Curtain, %
10^{-6}	1.25	0.0125
10^{-5}	12.5	0.125
10^{-4}	125	1.25
10^{-3}	1250	12.5

Leakage rates are expressed as a percentage of the total volume of air stored, leaking in 24 hours, at the cavern storage pressure. The potential for using the more permeable rocks when using a water curtain is obvious.

Grouting

Grouting would probably not be used for reduction of permeability of the entire periphery of the cavern. This would be uneconomical and would also imply unfavorable rock conditions for cavern and shaft stability during construction and service. However, grouting would be of use for treatment of locally permeable zones such as shear zones. Although the caverns would be sited in a geological situation chosen to avoid such zones, there is a high probability that some such zones would be encountered during construction. They could probably be dealt with effectively with cement or chemical grouts (or with a combination of the two) such that the permeability of these zones could be reduced to that of the surrounding rock mass.

Cement Grouts

Cement grouts could be used to effectively seal fissures greater than about 0.01 in. (0.25 mm) wide (Ref. III-34). For fissures spaced at 10 ft, this implies an initial rock mass permeability of the order of 10^{-3} cm/sec or more. This is in the permeability range for which a water curtain is uneconomical on the grounds of excessive water flow. Hence, air leakage through any local permeable zones of this nature which were encountered could not be controlled by installation of a water curtain, and these zones would first have to be sealed using cement and/or chemical grouts.

Chemical Grouts

Recent field work by Bergman et al (Ref. III-35) suggests that it is possible to successfully use chemicals such as Geoplast 45 and Stabilodur C75 in fissures as fine as 0.0004 in. (0.01 mm), which implies a rock mass permeability (k_w) of the order of 10^{-7} cm/sec, assuming a 10-ft fissure spacing. The rate of penetration of these chemical grouts into fissures of this size is on the order of 0.02 to 0.04 in. per sec (0.5 to 1.0 mm per sec). To form a 50-ft thick annulus around a tunnel would take about 4 to 8 hr at these rates. In fissures of 0.001 in. (0.03 mm) wide ($k_w = 10^{-6}$ cm/sec), the penetration rate increases to about 0.4 in. per sec (10 mm per sec) giving a penetration time of about 1/2 hr for a 50-ft annulus.

Chemical grouts might well have a use in the treatment of the air and water pressure shafts, but would probably be more expensive than a water curtain for preventing air leakage from the caverns.

Comparison Of Leakage Prevention Methods

The comparison of the applicable range and effectiveness of the different methods of preventing (or reducing) leakage is best shown in Table III-2. The ratio of the permeabilities of water and air in the rock mass is assumed to be 1:100 (inverse ratio of viscosities). The fissure width equivalent to a particular rock mass permeability has been calculated by the use of Eq. III-6, using a fissure spacing of 10 ft.

Temperature Effects

It can be assumed that the average temperature of the rock mass containing the air storage will be approximately 50 to 70 F. The rock walls will come into contact alternately with warmer air (during the charging mode) and colder water (during the generation mode), causing thermal expansion and contraction. However, the rock walls will be confined against tangential movement and stresses will develop.

Berg and Noren (Ref. III-26) have examined the effects of an air temperature of 122 F and a water temperature of 32 F applied to intact, fissured, and grouted igneous rock, at a temperature of 50 F. A summary of their tangential stress estimates appear in Table III-3. For the case of thermal expansion in both the intact and fissured rock, the 72 F temperature rise would create tangential compressive stresses on the order of 10 percent of the compressive strength. In the case of the grouted rock, the stresses would approach 50 to 60 percent of the strength of the grout. Tangential tensile stresses caused by the 18 F temperature fall could be of the same order of magnitude as the tensile strength of the rock in all these conditions. Radial stresses are a small percentage of the tangential stresses.

Tangential Stresses

The above stresses should be viewed in comparison with the compressive tangential stresses in the rock walls due to the excavation configuration itself. In general, the maximum compressive tangential stress for the geometry of opening envisaged for this project is on the order of 2 to 3 times the overburden pressure occurring close to the boundary of the cavern. At a depth of 1,600 ft this pressure is approximately 1,800 psi, giving a maximum tangential stress in the rock wall on the order of 3,600 to 5,400 psi. For this project a factor of safety of 2 to 3 against failing the rock walls and roof is appropriate, leading to a minimum required compressive strength of the order of 11,000 to 14,000 psi with no allowance for compressive temperature stresses.

It is considered unnecessary to apply a large factor of safety to the temperature stress increment - 1.5 would probably be adequate. Hence, the total minimum compressive strength required of the intact rock material would be on the order of 20,000 psi. This strength is common among good-quality igneous and metamorphic rocks (granites and gneisses but not some schists) and among high-quality, massive limestones and dolomites. It is, however, considerably higher than the strength of cement grouts, and thus grouted crushed or faulted zones would probably require systematic bolting and mesh support. Should the temperature rise be less, due to a higher natural rock temperature or lower air temperature, the thermal compressive stresses would be correspondingly lower.

It is readily apparent that a much higher temperature differential between air and rock could not be tolerated. For example, a 500 F air temperature would cause a sixfold increase in the thermal compressive stresses, which could only be tolerated in rocks with a compressive strength of about 40,000 psi - a very exacting criterion. This would also probably lead to the use of systematic bolting and mesh at many potential locations which would otherwise need only spot bolting.

The tensile stresses shown in Table III-3 imply cracking of intact rock or opening of existing fissures. However, in a properly designed cavern, the tangential compressive stresses should exceed these tensile stresses and no net tension should occur. The cyclic nature of the thermal regime would also lead to a fatigue effect. This might cause a reduction in the rock strengths on the order of 25 percent. This should be allowed for in the selection of factors of safety.

Radial Stresses

The maximum radial stresses in the rock wall would occur a short distance into the rock from the boundary and are estimated to be only a few percent of the tangential stress values. This should cause little difference to the radial compressive stress field, and thermal radial tensile stresses for 120 F rock wall temperature should also be contained within the radial compressive stress field.

With higher cyclic temperature changes, the tangential and radial tensile stresses might become large enough to cause near-surface fissures to open, and spalling and slabbing failures to occur in the walls and roof. These failure modes would probably precede compressive failure of the rock walls.

Air Storage Temperature

Although problems would be caused by thermal compressive stresses in the rock walls at high temperatures, the primary economic reason for minimizing the air storage temperature is that a lower temperature requires less storage volume. For example, at 120 F the volume required to store the same mass of air is about 60 percent of the volume required at 500 F. This benefit cannot be taken too far, however, since in addition to the increased cost of cooling the air, it is not possible to go too close to 32 F because of possible freezing of the storage water and ice formation upon expansion of the high-pressure air.

An upper figure of 500 F air storage temperature has previously been considered because this is the approximate boiling point of water at 50 atm storage pressure, and boiling the groundwater is considered to be undesirable. The behavior of rock at these temperatures, even prior to boiling the groundwater, is uncertain. There might well be problems with rock mineral decomposition and vapor pressure and partial pressure effects in rock joints leading to cavern wall instability. Environmental effects are unknown and problems could be severe. Because of these uncertainties it is considered prudent to adopt a conservative approach, and to limit the storage temperature to about 120 F. This temperature has been favored at the Brown Boveri installation at Huntorf (Ref. III-36).

Distinctions should be made between the temperature of the air entering the cavern, air storage temperature, and the temperature of the rock walls. Preliminary calculations have shown that there is insufficient heat stored in the air entering the cavern at 120 F to cause the temperature of the rock walls and storage water to rise by more than 20 F. This is a complex transient heat balance problem involving the thermal conductivity, density, and heat capacity of air, rock, and water. Given refinements in the analysis of this problem, it might be possible to show that air entering the cavern at a higher temperature would be compatible with satisfactory rock wall performance, also resulting in a sufficiently low storage temperature to allow an economic storage volume requirement.

CAVERN LAYOUT AND COSTS

This section contains discussions of the underground cavern layout, including orientation of the various tunnels and shafts which comprise the cavern complex, and parametric cavern cost estimates.

Underground Cavern Layout

The storage volume required for the preliminary reference design of a single-unit underground air storage complex amounted to 250,000 cu yds at 1600 ft below the surface. This is based on a 280-MW unit operating for 10 hours per day with an overall pressure ratio of 50. It was assumed that a suitable host rock could be identified in which the storage cavities would be located. Conventional methods were considered for excavation of the desired storage volume and for construction of the access shafts required.

In this section, alternative arrangements are discussed, and a layout for costing is presented. An estimate of cost for the proposed underground complex is given, together with parametric curves of cost for variations from the reference case.

Tunnels

It has been established in the section on leakage that the air leakage rate, expressed as a percentage of the stored volume, would be minimized if the storage volume were maximized within the area of the cavity boundaries. In other words (with respect to air leakage), an interconnected grid of high tunnels provides the best arrangement for storage volume. Such an arrangement is possible, and is indeed accepted practice, in the room- and-pillar mining technique. However, for reasons of structural integrity and ease of construction, a system of tunnels with the minimum of interconnections is preferred. This arrangement cuts down the number of intersections required and simplifies the control of ventilation air during the construction of the complex.

The span of the cavern opening will be dependent on the quality of rock encountered at the selected site, and might vary from 25 ft in some sedimentary limestones to 80 ft in good-quality igneous and metamorphic rocks. For estimating purposes, the span has been assumed to be 60 ft.

As stated above, it is advantageous as regards leakage to construct high cavities. This is also economic with respect to excavation costs, since a higher percentage of the rock can be removed by the lower cost benching method. The relative costs of excavation by the full heading and benching method are presented in

Table III-4. These costs are considered representative for the excavation of a 60 ft wide cavity in competent igneous rock in the northeast region of the study for which there is a fairly adequate data base. There have been numerous excavations in competent igneous and metamorphic rocks of the size envisaged for the air storage cavern. These have been predominatly underground hydroelectric machine hall excavations (Table III-1) in North America, and both machine hall and oil storage cavern excavations in Scandinavia.

In the case of competent sedimentary rocks, such as limestone and dolomite, there is little available cost data for excavations of this size, although there are some precedent excavations noted in Table III-1. The sedimentary rocks in general are softer than igneous and metamorphic rocks, but they are not necessarily easier to excavate because of this. Given the preselection of suitable rock masses, it is estimated that the excavation costs in competent sedimentary and igneous rocks would not be greatly different, neglecting such factors as labor cost availability.

It is obvious from Table III-4 that there are cost advantages to be gained by increasing the proportion of benching in any extensive tunnel excavation required. However, there might be penalties in the cost of other items of the plant for this cost advantage because of the range of pressures resulting from the varying water level in proceeding from the charged to empty condition of the storage chamber. Thus the economics of regulating this pressure must be weighed against any cost advantage. As a baseline case, an overall cavity height of 85 ft has been assumed. Excavation of this cavity would proceed using a full heading and two benches.

The volume of storage required for a single-unit complex operating for 10 hours at a pressure ratio of 50 has been established at 250,000 cu yds. To allow for some margin of safety during operation (since it would be undesirable to allow too high a concentration of dissolved air in the compensating water remaining in the cavern at any time), an allowance of 10 percent has been added to the excavated volume requirements for a total of 275,000 cu yds.

Shafts

It has been established that shafts are the most economical means of access to the levels being considered here. For safety reasons, two shafts would be required during construction of the storage cavities. The available methods of shaft construction and the size requirements indicate that, given suitable rock conditions, it would be possible to construct the complex using 6 ft diameter shafts. As a base case, however, it is considered that allowance be included in the estimates for a larger initial shaft to be constructed by the conventional sinking technique. A shaft with an excavated diameter of 14 ft and a 12 in. concrete lining is considered sufficiently representative of the requirements. The second shaft would be a smaller 6 ft diameter shaft constructed by drilling from the surface.

On completion of construction, the larger shaft would be used as the compensating water passage and would be connected to a surface reservoir. For operating purposes, the shaft must extend below the level of the underground storage area by an amount sufficient to create a "U-tube" capable of resisting a blowout during operation.* Euorpean experience has indicated that this amount should be in the order of 10 percent of the hydraulic head on the reservoir, and this figure has been adopted.

The 6 ft diameter shaft, which would be used for ventilation and as emergency access during construction of the underground cavities, will accommodate the compressed air supply line in the final installation. This will be a 3-ft diameter steel pipe supported over its length from the walls of the shaft.

The underground installation would be completed by the construction of the required concrete bulkheads. The proposed layout is presented in Fig. III-10.

Schedule

The proposed construction schedule for the underground complex is shown in Fig. III-11. Since events have to follow in sequence, the construction of the first vertical shaft is the key to the schedule. In order to provide a second means of access early in the schedule, it will be necessary to drill the second shaft from the surface in parallel with the conventional sinking of the main shaft. Past experience indicates that the overall sinking rate of 40 ft per week envisaged here can be attained comfortably. Discussions with shaft drilling contractors confirm that the second shaft could be drilled and lined within the time period allowed for sinking the main shaft.

Once access has been gained to the underground area, excavation of the main cavities can proceed. Allowing 2 months for development of the area and construction access, the total excavation could be completed in less than 1 year. This involves an average production of excavated rock of about 27,000 cu yds per month, which is well within the capabilities of the equipment proposed for the excavation and hoisting of rock spoil.

When excavation of the underground cavities and shafts is complete, it will be necessary to construct bulkheads, install the air line in one shaft, remove the headframe and hoist from the other shaft, and construct a water intake. The time for construction of the complete underground complex for the preliminary reference design is, therefore, estimated at just over 2 years.

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* This blowout, or champagne effect, is discussed in Part IV.

Some lead time will be required prior to the start of construction for ordering any special equipment required for hoisting. Currently, delivery time on the hoisting equipment necessary for removal of the material excavated from the cavities is around 2 yrs, which would mean that about 1-yr lead time must be added to the schedule presented above. On the other hand, it is highly probable that it would be possible to lease a mining hoist for the duration of the job, which would reduce the mobilization time to around 6 months.

Cost Estimates

The capital cost of the underground complex will depend on the site selected for construction of the installation. The local geology and labor rates will be instrumental in determining unit costs for the various activities. For the reference design, representative costs for a complex excavated in igneous rock in the northeast area of the study have been estimated. Labor rates in the area for the trades involved in the various construction activities have been reviewed, and an average figure used in compiling labor costs. Plant hire and purchase costs applicable to the area have been applied.

Basic Single-Unit Complex

The costs of the underground complex can be divided into two categories:

- shaft excavation and lining costs
- cavity excavation costs

Discussions were held with various contractors on the techniques and costs involved in constructing shafts to the depths required. The estimate presented for the water shaft is based on a 14 ft excavated diameter shaft with a 12 in. thick concrete lining, constructed using a multistage platform lowered from the surface. Total shaft depth to be excavated in this manner will be 1760 ft.

Normally, the second shaft would be raised when access was established underground, but in order to compress the overall schedule for the construction of the underground complex, it will be necessary to drill the second shaft from the surface while sinking the main shaft. Since this shaft will be used for emergency access during construction and could be used for permanent access when the installation is complete, it will be necessary to construct a concrete lining. The estimate allows for the cost of drilling an 8 ft diameter shaft with a 12 in. concrete lining.

The estimate of excavation costs for the cavities is based on the layout shown in Fig. III-10 with a tunnel cross section of 85-ft high by 60-ft wide. Drilling and blasting techniques using a heading and two benches have been assumed. Removal from the working face to the dump station at the shaft would be by rubber-tired front-end loaders, and the excavated rock would be crushed to minus 8 in. and hoisted to the surface using a mechanical hoisting system. Either a capstan or friction type hoist could be used in the size of shaft provided.

The total estimated cost of construction of the reference underground complex described above is presented in Table III-5.

Cost Variations

The cost estimates presented above have been developed for a specific assumption with regard to the geometry and depth of the storage cavities. However, variations in cavern geometry might be desirable in the overall optimization of the development. It is also possible that advantages in overall economy might result by adjusting the pressure ratio and, hence, the depth of the storage caverns from the surface. In order to determine the overall economics of such variations, some analyses of excavation costs for the various alternatives have been carried out. The details and results of the analyses are discussed below.

The cost estimate prepared was based on excavating a 60-ft wide cavity by the drilling and blasting technique, using a heading and two benches with a total height of 85 ft. The span of any cavity actually constructed will depend on the rock properties at the specific site selected, and might vary from 25 ft to 80 ft. For a specified tunnel cross section, the required storage volume could be varied by changing the total length of tunnel. Within reasonable limits, this will not affect the unit price of excavation. This is not the case with a change in cross section. The effect of a change in tunnel cross section by varying overall height of the cavity was examined for a 60-ft wide tunnel by costing the alternatives shown in Fig. III-12. The mean cost per cubic yard of excavation for each alternative was determined on the basis of the percentage of heading and benching contained in each alternative, and is presented below.

Cavern Height, ft	Mean Cost, $/cu yd
25	33.00
55	23.00
85	20.91
115	19.37

These figures are plotted in Fig. III-13, which presents the unit cost of excavating 275,000 cu yds of storage space in igneous rock in the northeast region by drilling and blasting techniques. Under similar conditions, any variation in the depth of the cavity below the surface within the range being studied would not significantly affect the unit prices presented.

For a given installation, operating cycle, and operating pressure, the total volume of cavity required will vary directly with the depth from the surface (see Fig. III-14). This variation in volume is relatively insignificant as far as the estimated unit costs of excavation are concerned, so the cavity construction cost will vary directly with the volume excavated. Development, mobilization, and completion costs will be independent of the depth at which the cavities are constructed, but shaft construction costs will vary with the depth of the installation.

Given the specific storage volume requirements and depth below the surface for the storage for a 280-MW installation, with a turbine inlet temperature of 2,000 F, the cost of underground storage facilities for 10 hours of operation can be estimated. This was done for a number of alternatives assuming a 60-ft wide cavity with heights of 55 ft, 85 ft and 115 ft. The estimated costs for the reference height of 85 ft for various depths below the surface are summarized in Table III-6. These costs, and the costs for the other cavity heights, are presented graphically in Fig. III-15.

REFERENCES FOR PART III

III-1 Atlas Copco: A Brief Guide to Mining Methods with Recommendations for Equipment, 1973.

III-2 Olson, J. J and T. C. Atchison: Research and Development - Key to Advances for Rapid Excavation in Hard Rock. Proceedings First North American Rapid Excavation and Tunneling Conference, Chapter 78, pp. 1393-1441, 1972.

III-3 Hakala, W: Program Director, Excavation Research Program, National Science Foundation, RANN. Private Communication, November 1975.

III-4 Harnback, P: An Inclined Gallery Through Hard Rock. Tunnels and Tunneling, May 1970.

III-5 Cook, N. G. W. and V. R. Harvey: An Appraisal of Rock Excavation by Mechanical, Hydraulic, Thermal and Electromagnetic Means. Proceedings Third ISRM Congress, Denver, Vol. 1, Part B, pp. 1599-1615, 1974.

III-6 Wang, F. D.: Professor of Mining Engineering, Colorado School of Mines. Private Communication, November 1975.

III-7 Morfeldt, C. O.: Storage of Oil in Unlined Caverns in Different Types of Rock. ISRM Fourteenth Symposium on Rock Mechanics, Pennsylvania, 1972.

III-8 Bruce, W. E.,and R. J. Morrell: Rapid Excavation in Hard Rock - A State-of-the-Art Report. Proceedings Deep Tunnels in Hard Rock - A Solution of Combined Sewer Overflow and Flooding Problems, pp. 187-219, 1970.

III-9 Maurer, W. C.: Novel Rock Disintegration Techniques. Fifteenth ASME Symposium on Resource Recovery, Albuquerque, New Mexico, March 1975.

III-10 Neudecker, J. W.: Design Description of Melting - Consolidating Prototype Subterrene Penetrators. Los Alamos Scientific Laboratory Report LA-5212-MS, 1973.

III-11 Gigo, R. G.: Description of Field Tests of Rock-Melting Penetration. Los Alamos Scientific Laboratory Report LA-5213-MS, 1973.

III-12 Hanold, R. J.: Large Subterrene Rock-Melting Tunnel Excavation Systems. Los Alamos Scientific Laboratory Report LA-5210-MS, 1973.

III-13 Westwood, A. R. C., and J. J. Mills: Application of Chemomechanical Effects to Fracture-Dependent Industrial Processes. Martin Marietta Laboratories Technical Report No. 75-39c, October 1975.

III-14 Clark, G.: University of Missouri at Rolla. Private Communication, November 1975.

III-15 McCarthy, D. F.: Underground Storage Facilities for Gaseous and Liquid Hydrocarbons. Pipeline and Gas Journal, March 1972.

III-16 McCarthy Engineering Construction Incorporated: Private Communications, 1975.

III-17 McCarthy, D. F.: Raise Boring Techniques. Coal Mining and Processing, March 1972.

III-18 Teton Exploration Drilling Company Incorporated. Private Communication, November 1975.

III-19 Acres American Incorporated. Muskingum Underground Pumped Storage Study, Phase II, Report to American Electric Power Service Corporation, 1972.

III-20 Whitworth, K.: United Kingdom Develops New Pneumatic Hoisting Technique. World Coal, pp. 24-25, April 1975.

III-21 Radmark Engineering Limited. Private Communication, November 1975.

III-22 Maschinenfabrik Karl Brieden and Company. Private Communication, December 1975.

III-23 Allis Chalmers (Canada) Limited. Private Communications, November 1975.

III-24 Barton, N. R.: A Model Study of Air Transport from Underground Openings Situated Below Ground Water Level. Paper T3-A, ISRM Symposium on Percolation Through Fissured Rock, Stuttgart, 1972.

III-25 Janub, N. and O. Tokheim: Noen Utledninger og Formler i Forbindelse med Vaeske-og Gass-strømning i Porøse Medier. Technical University of Norway, Trondheim, Internal Report, 1973.

III-26 Berg, N and D. Noren: Compressed Air Power Plants - Some Views on Underground Compressed Air Storage. Svenska Kraftverksforeningens, Publication 536B, 1969.

III-27 Di Biagio, E. and F. Myrvoll: In Situ Tests for Predicting the Air and Water Permeability of Rock Masses Adjacent to Underground Openings. Paper T1-B, ISRM Symposium on Percolation Through Fissured Rock, Stuttgart, 1972.

III-28 Snow, D. T.: Rock Fracture Spacings, Openings and Porosities. ASCE Proceedings, Vol. 94, No. SM1, pp. 72-91, 1968.

III-29 Noren, D., L. O. Emmelin and S. E. Paulsson: Compressed Air Power Plants "Airstore". Sealing of the Underground Compressed Air Storage, Swedish Consulting Group (SWECO), Stockholm, 1970.

III-30 Birdwell Division, Seismograph Service Corporation. Private Communication, November 1975.

III-31 Bernell, L. and T. Lindbo: Tests of Air Leakage in Rock for Underground Reactor Containment. Nuclear Safety, Vol. 6, No. 3, 1965.

III-32 Janelid, I.: Method of Preventing Leakage During Storage of a Gas or a Liquid in a Rock Chamber by Artificially Supplying a Gas or Liquid to the Rock Surrounding the Rock Chamber. U.S. Patent No. 3,670,503, application made June 20, 1972.

III-33 Skanska and Hagconsult: Unlined Rock Caverns for Compressed Air or Gas Storage. Project Gas Cavern Report, Stockholm, 1975.

III-34 Eighth International Conference on Large Dams. Edinburgh, Vol. 1, R40, Q28, p. 734, 1964.

III-35 Bergman, S. G. A., K. Kindman, L. Lundstrom, P. Soderman and S. Ullerud: Grouting of Tunnels to Prevent Small Scale Infiltration. Swedish Building Research Summaries, R25:1975.

III-36 Weber, O.: The Air Storage Gas Turbine Power Station at Huntorf. Brown Boveri Review, Vol. 62, pp. 332-337, July/August 1975.

TABLE III-1

LARGE CROSS-SECTIONAL UNDERGROUND EXCAVATIONS IN NORTH AMERICA

Locations	Function	Constr. Date	Dimensions LxWxH, ft	Rock Type	Roof Support
Bersimis, Quebec	Hydroelectric Machine Hall	1956	565 x 65 x 80	Paragneiss (metamorphic)	Concrete lining
Boundary, Colorado	Hydroelectric Machine Hall	1965	476 x 76 x 175	Limestone, dolomite (sedimentary)	Systematic bolting, wire mesh, gunite
Churchill Falls, Labrador	Hydroelectric Machine Hall	1970	1,000 x 81 x 145	Diorite, granite, gneiss (metamorphic)	Systematic bolting, wire mesh, gunite
Chute-des-Passes Quebec	Hydroelectric Machine Hall	1959	460 x 70 x 110	Paragneiss (metamorphic)	Concrete lining
Haas, California	Hydroelectric Machine Hall	1958	173 x 56 x 100	Granite (igneous)	Systematic bolting wire mesh, 8-in.gunite
Kitimat, B. C.	Hydroelectric Machine Hall	1955	1,140 x 82 x 139	Granite, Grandodiorite (igneous)	Concrete lining bolting
Loon Lake	Hydroelectric Machine Hall	-	110 x 75 x 110	Granite (igneous)	-
Morrow Point, Colorado	Hydroelectric Machine Hall	1963	206 x 57 x 134	Micaceous quartzite, mica, schist, and granite (metamorphic and igenous)	Systematic bolting and cables

TABLE III-1 (CONT'D)

Locations	Function	Constr. Date	Dimensions LxWxH, ft	Rock Type	Roof Support
Nevada Test Sites I and II	Test Facility	-	120 x 80 x 140	Tuff (volcanic ash)	Systematic bolting
Norad, Colarado	Defense Facility	-	600 x 45 x 60	Granite (igneous)	Systematic bolting
Northfield, Massachusetts	Pumped Storage Cavern	1970	- x 70 x 139	Granite/gneiss (igneous/metamorphic)	Drip lining only
Oroville, California	Hydroelectric Machine Hall	1961	550 x 70 x 120	Amphibolite (metamorphic)	Systematic bolting, wire mesh, 4-in. gunite
Outardes 3, Quebec	Hydroelectric Machine Hall	-	-	Diorite (igneous)	Systematic bolting
Portage Mountain, B.C.	Hydroelectric Machine Hall	1965	890 x 67 x 144	Sandstone, shale, coal (sedimenatry)	Systematic bolting, thin concrete arch
Raccoon Mountain Tennessee	Hydroelectric Machine Hall	1970	-	Limestone (sedimentary)	Systematic bolting
Rainer Mesa, Colorado	Test Facility	-	- x 80 x 140	Tuff (volcanic ash)	Systematic bolting gunite
Snoqualmie Falls, Washington	Hydroelectric Machine Hall	1899	200 x 40 x 30	-	-
Spaulding No. 1, California	Hydroelectric Machine Hall	1917	85 x 30 x 60	-	-

TABLE III-2

COMPARISON OF LEAKAGE PREVENTION METHODS

Permeability of air, m/sec and water, cm/sec	10^{-2}	10^{-3}	10^{-4}	10^{-5}	10^{-6}	10^{-7}	10^{-8}	10^{-9}
Fissure width for 10-ft (3-m) spacing, in. (mm)	0.03 (0.7)	0.01 (0.3)	0.006 (0.15)	0.003 (0.07)	0.001 (0.03)	0.0006 (0.015)	0.0003 (0.007)	0.0001 (0.003)
One-dimensional leakage rate (percent/day)	--		125	12.5	1.25	0.125		
Effective range of water curtain	Too permeable		Effective range			Not necessary		
Grouting ranges	Cement grout			Chemical grout			Not necessary	

TABLE III-3

ROCK PROPERTIES AND THERMAL TANGENTIAL STRESSES
High-quality igneous or metamorphic rock
(From Ref. III-26)

	Rock Condition		
	Intact	Fissured	Grouted
Young's modulus, psi	7×10^6	-	-
Modulus of deformation, psi	-	1.5×10^6	4.5×10^6
Compressive strength, psi	29,000	4,400	4,400
Tensile strength, psi	0-1,200	0-1,200	0-1,200
Thermal expansion, in./in./°F	0.55×10^{-5}	0.55×10^{-5}	0.55×10^{-5}
Temperature rise, °F	72	72	72
Temperature fall, °F	18	18	18
Heating, compressive stress, psi	2,900	600	1,800
Cooling, tensile stress, psi	750	150	450

TABLE III-4

COMPARATIVE COSTS OF
HEADING AND BENCHING EXCAVATION

	Heading, $/cu yd	Benching, $/cu yd
Labor	10.70	3.50
Materials	3.00	1.35
Construction Equipment	7.95	5.60
Direct Unit Cost	21.65	10.45
Indirect Cost and Profit	6.50	3.05
Unit Cost at Shaft Bottom	28.15	13.50
Roof Support* (nominal)	3.35	--
Crushing and Hoisting	0.90	0.90
Disposal at Surface	0.60	0.60
Total Unit Cost of Excavation	33.00	15.00

*Allows for pattern roof bolting over 100 percent of area, regarded as minimal requirement for safety reasons.

TABLE III-5

COST OF UNDERGROUND STORAGE FACILITIES
FOR PRELIMINARY REFERENCE DESIGN

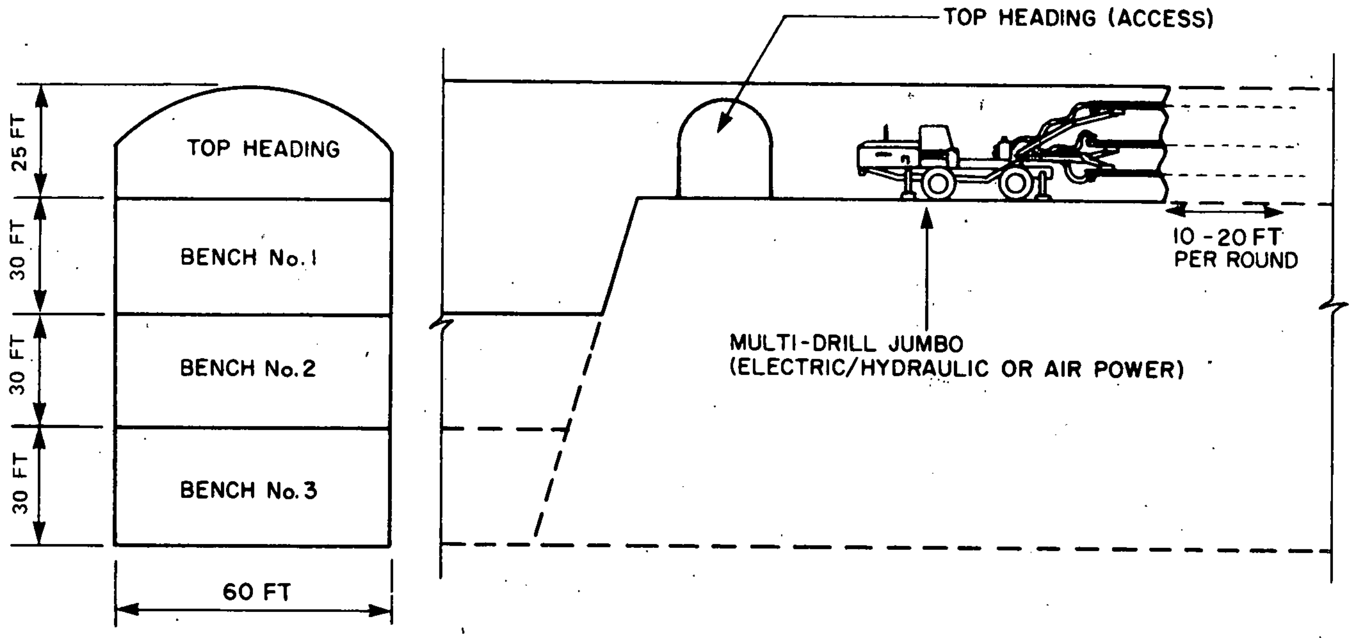
Item	Unit	Quantity	Unit Price, $	Cost 10^3\$	Total 10^3\$
Water Shaft (initial access)					
Mobilization	lump sum	--	--	350	
Sinking	lin ft	1,760	1,200	2,112	
Concrete lining (12 in.)	lin ft	1,760	300	528	
Furnishing and removal	lin ft	1,760	200	352	3,342
Development					
Shaft stations	cu yd	150	60	9	
Underground access	cu yd	4,000	40	160	
Mucking and Dumping	cu yd	500	200	100	
Headframe	lump sum	--	--	100	369
Cavity Excavation					
Heading	cu yd	75,000	33	2,475	
Benching	cu yd	200,000	15	3,000	
Rock bolting	lin ft	27,500	10	275	5,750
Air Shaft					
Drilling	lin ft	1,600	650	1,040	
Lining	lin ft	1,600	250	400	
Air pipe	lin ft	1,600	100	160	1,600
Finishing					
Concrete Bulkheads	cu yd	500	100	50	
Concrete Lining	cu yd	200	200	40	90
					11,151

TABLE III-6

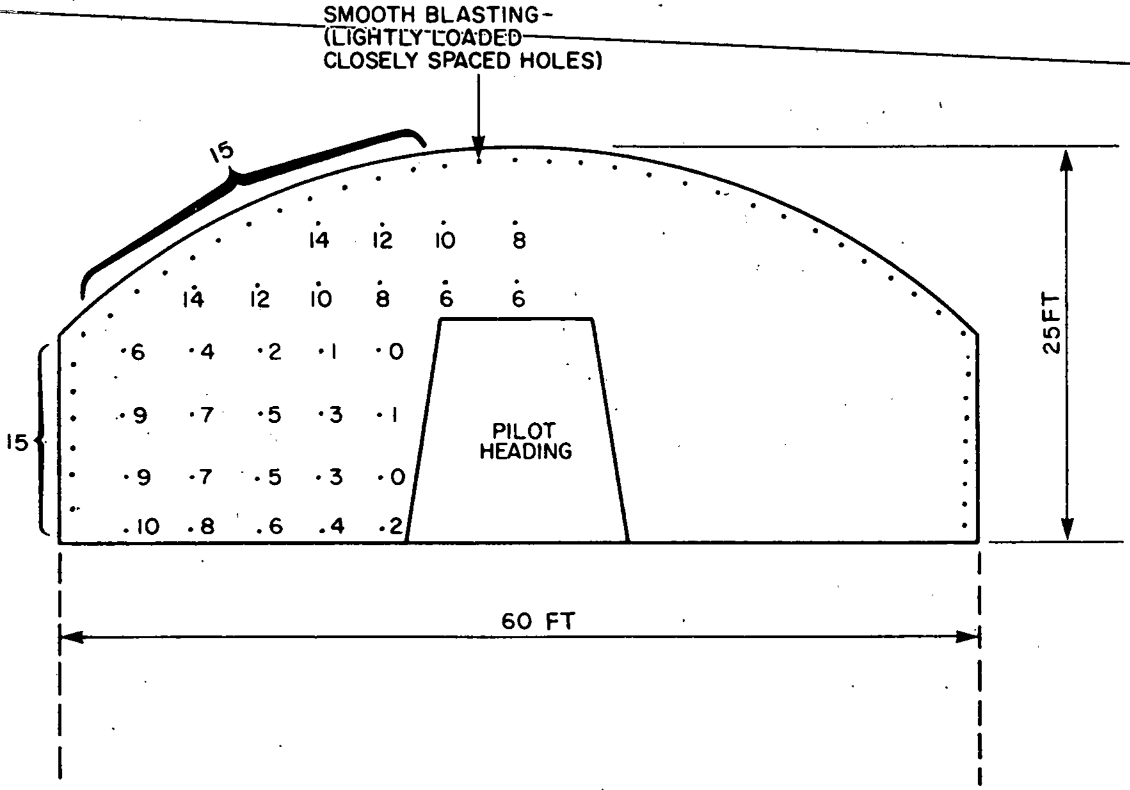
UNDERGROUND STORAGE COST PARAMETRICS

	Depth Below Surface, ft				
	1,200	1,400	1,600	1,800	2,000
Volume of Cavity, cu yd	375,000	315,000	275,000	240,000	210,000
	Costs in Thousand $				
Development and Mobilization	719	719	719	719	719
Water Shaft	2,244	2,618	2,992	3,365	3,740
Air Shaft	1,200	1,400	1,600	1,800	2,000
Cavity	7,838	6,587	5,750	5,018	4,391
Completion Cost	90	90	90	90	90
Total Cost	12,091	11,414	11,151	10,992	10,940
Unit Cost, $/cu yd	32.24	36.23	40.55	45.80	52.10
Specific Cost, $/kWhr	4.32	4.08	3.98	3.93	3.91

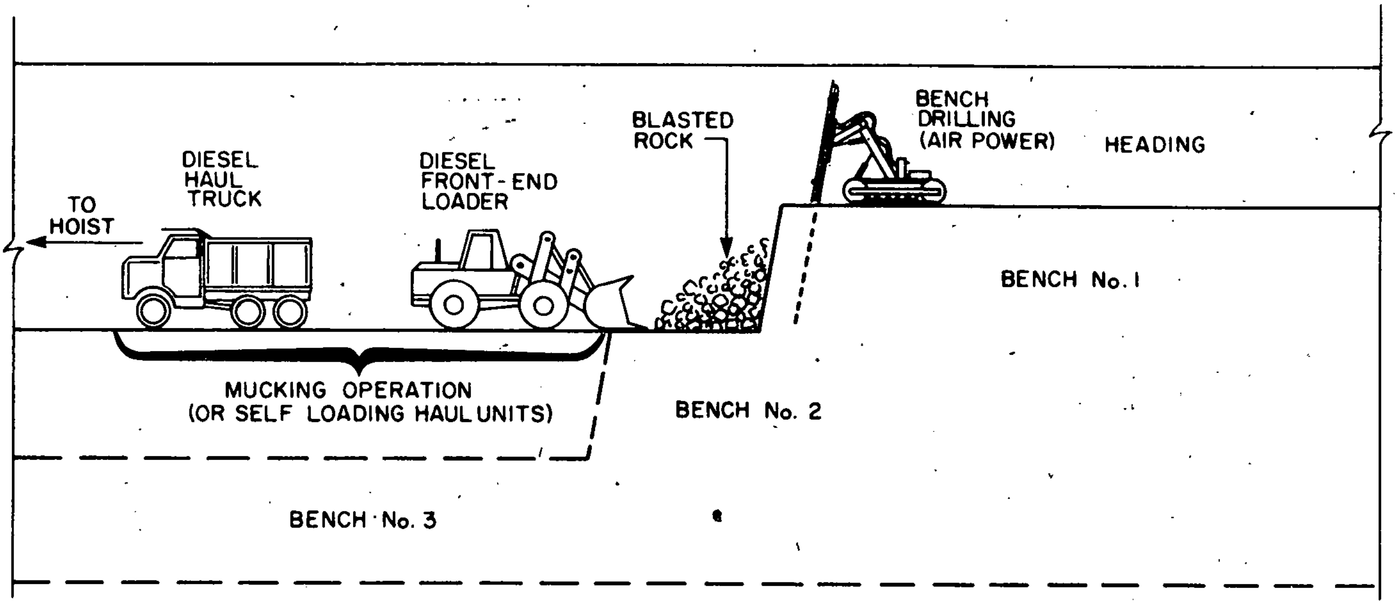
TYPICAL TOP HEADING DRILL-AND-BLAST PATTERN



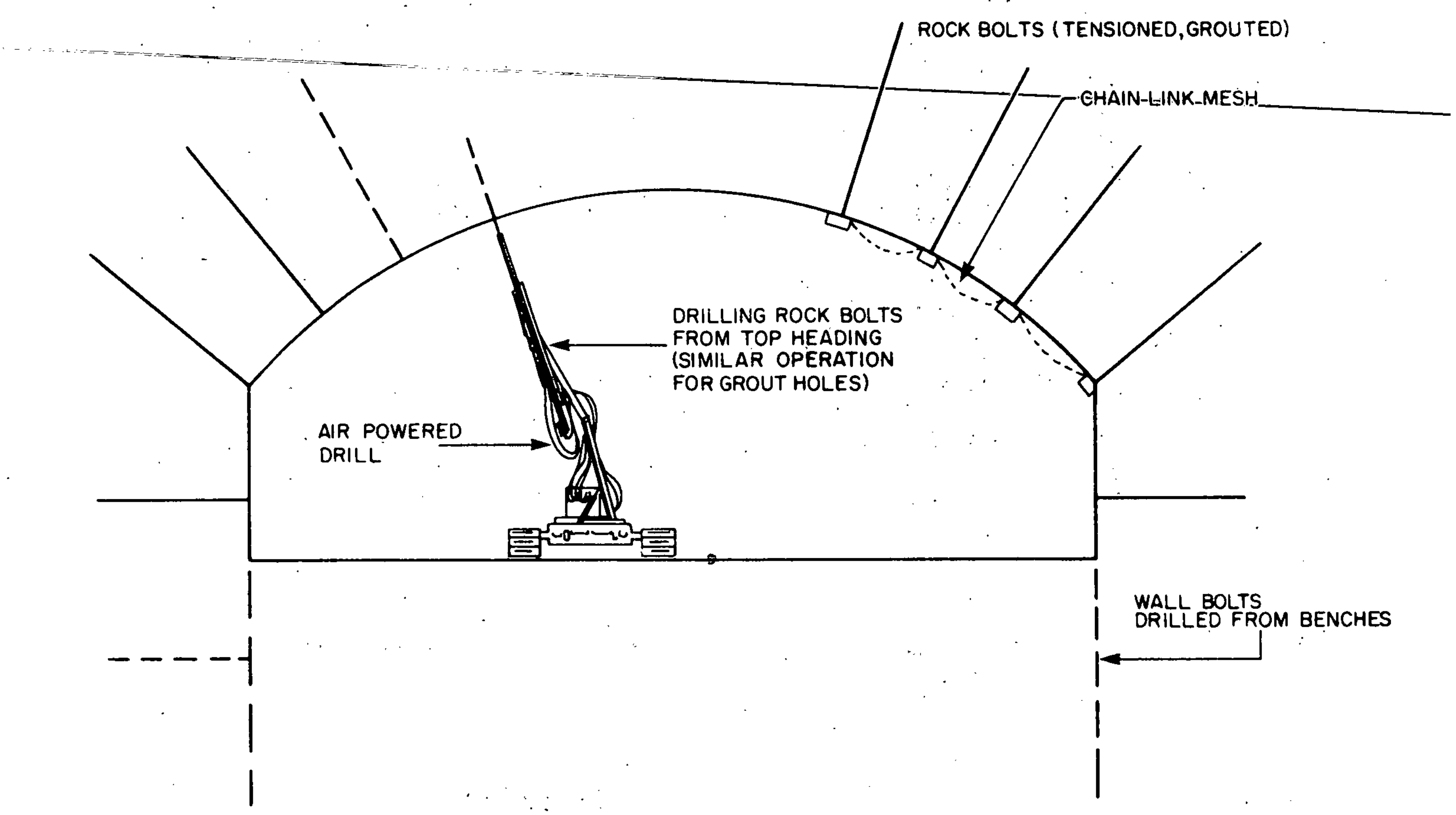
TOP HEADING AND BENCH CROSS SECTIONS



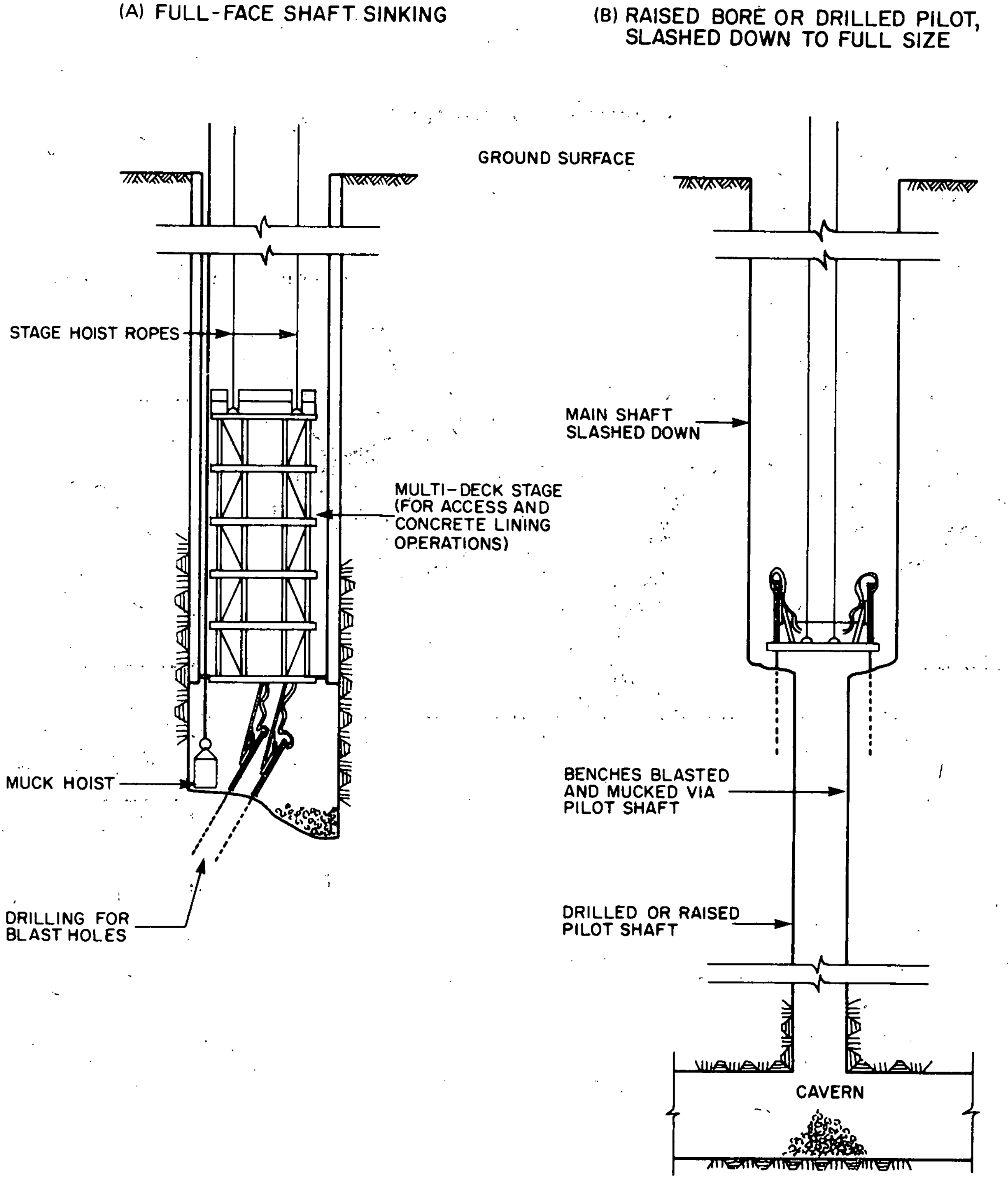
BENCH DRILLING AND MUCKING OPERATIONS



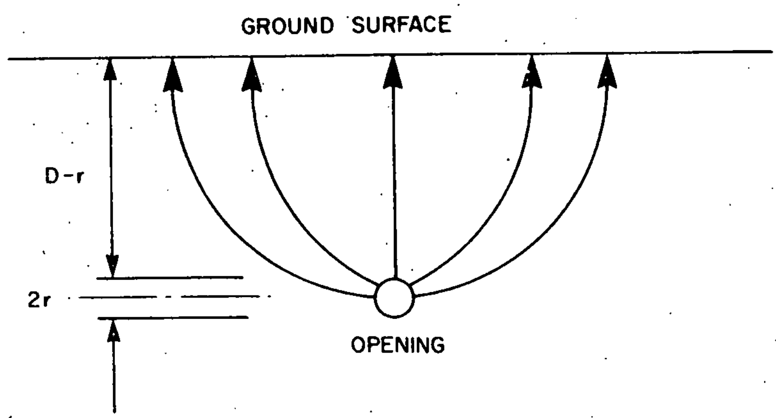
ROCK BOLT REINFORCEMENT



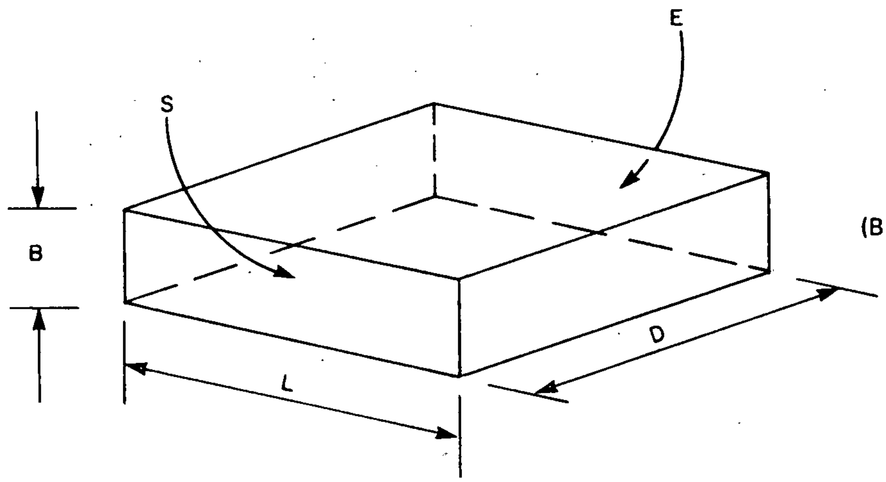
LARGE–DIAMETER SHAFT EXCAVATION METHODS



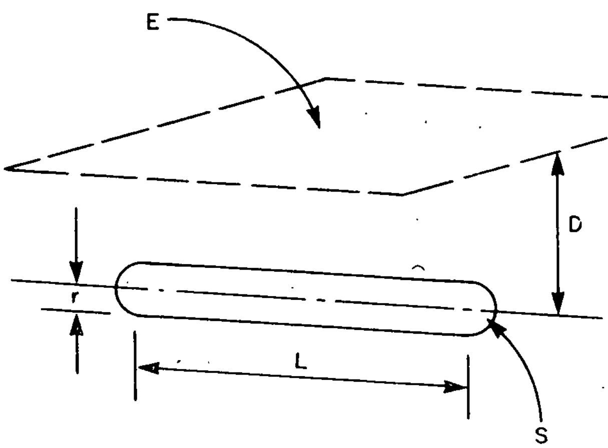
UNDERGROUND LEAKAGE MODELS



(A) FLOW FROM CIRCULAR OPENING TO HORIZONTAL GROUND SURFACE



(B) ONE–DIMENSIONAL FLOW

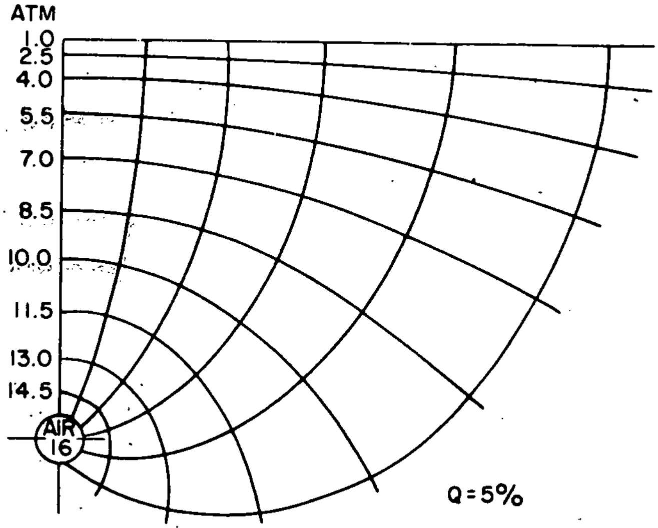


(C) THREE-DIMENSIONAL FLOW FROM TUNNEL TO GROUND SURFACE

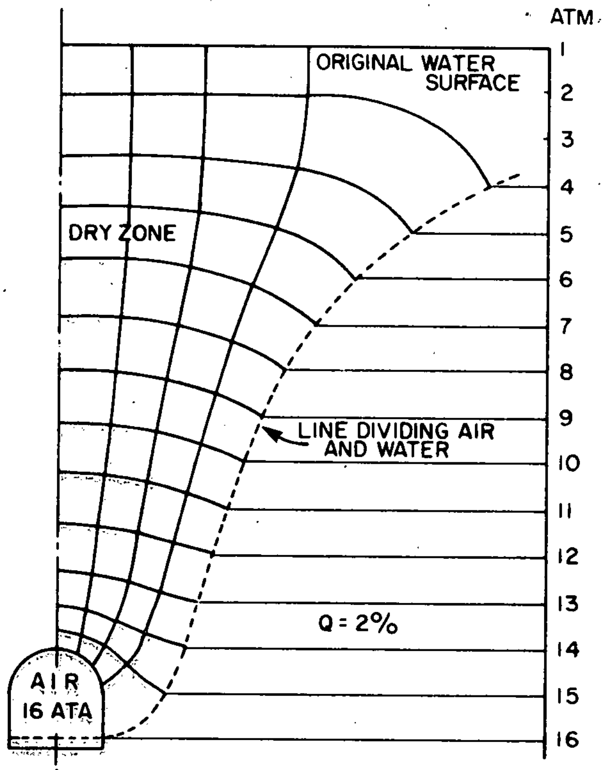
76–05–93–12

AIR LEAKAGE FLOW NETS

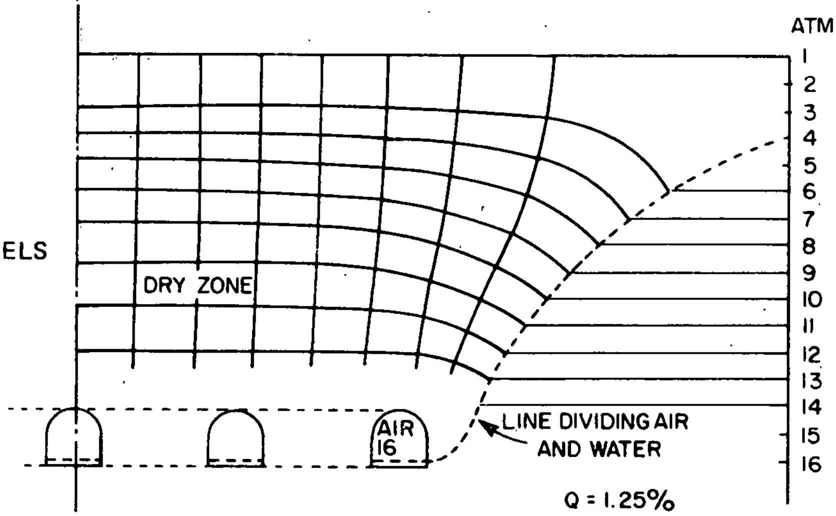
(A) DRY ROCK, SINGLE TUNNEL



(B) SATURATED ROCK, SINGLE TUNNEL



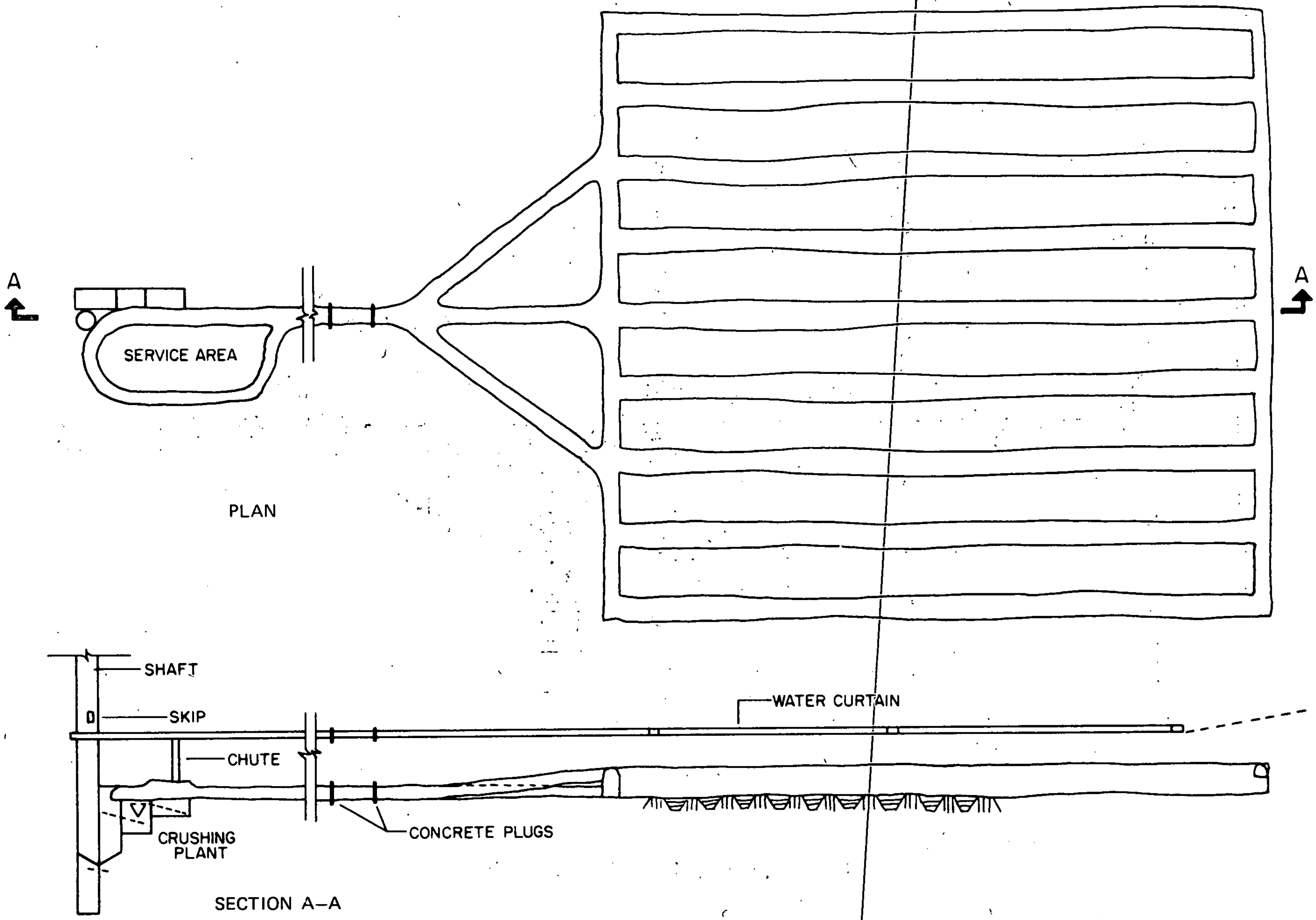
(C) SATURATED ROCK, INTERSECTING TUNNELS



76–05–93–13

R76-952161-5

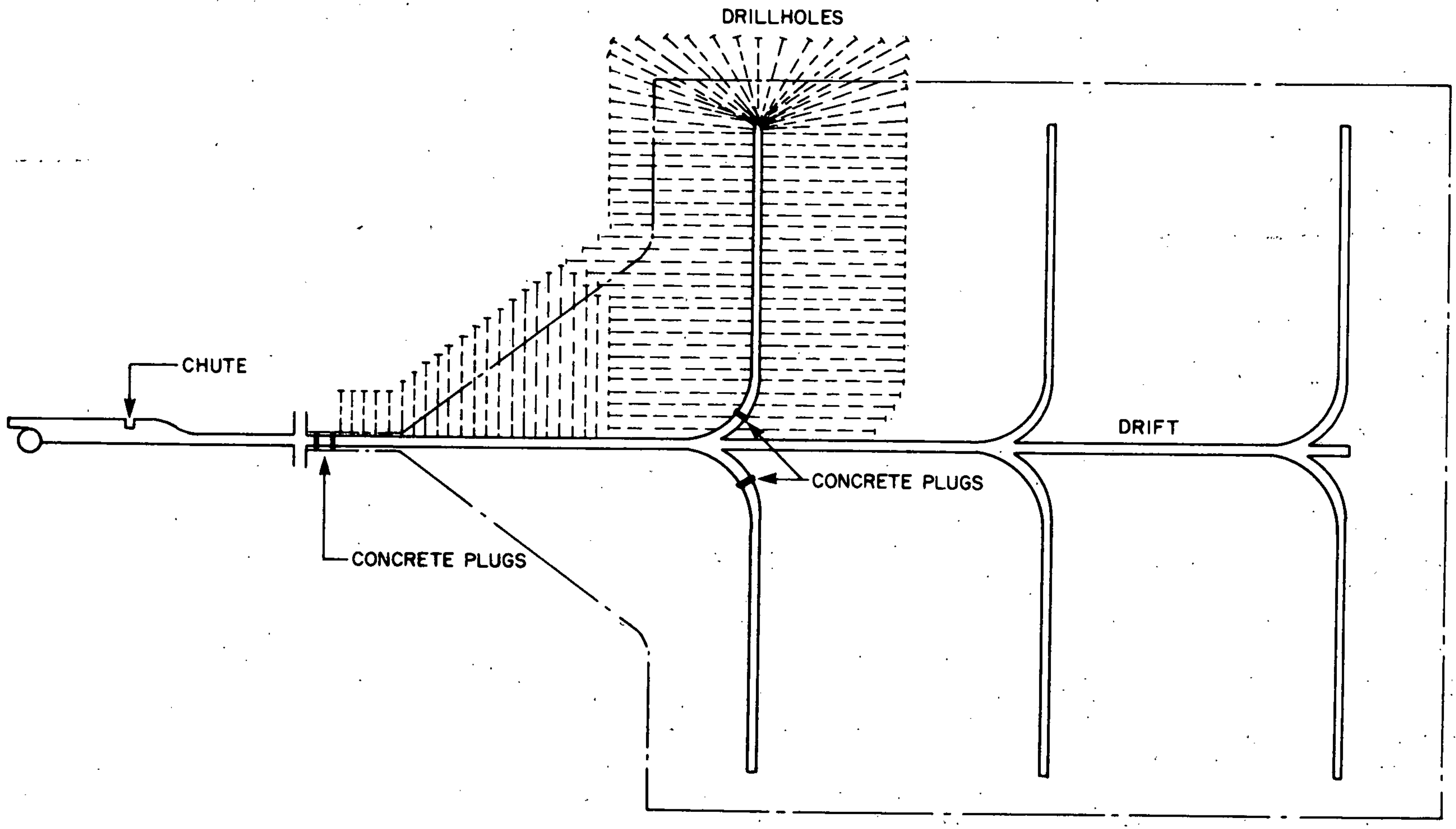
PRINCIPAL LAYOUT OF STORAGE CAVERNS



76-05-93-14

FIG. III-8

PLAN OF WATER CURTAIN SYSTEM

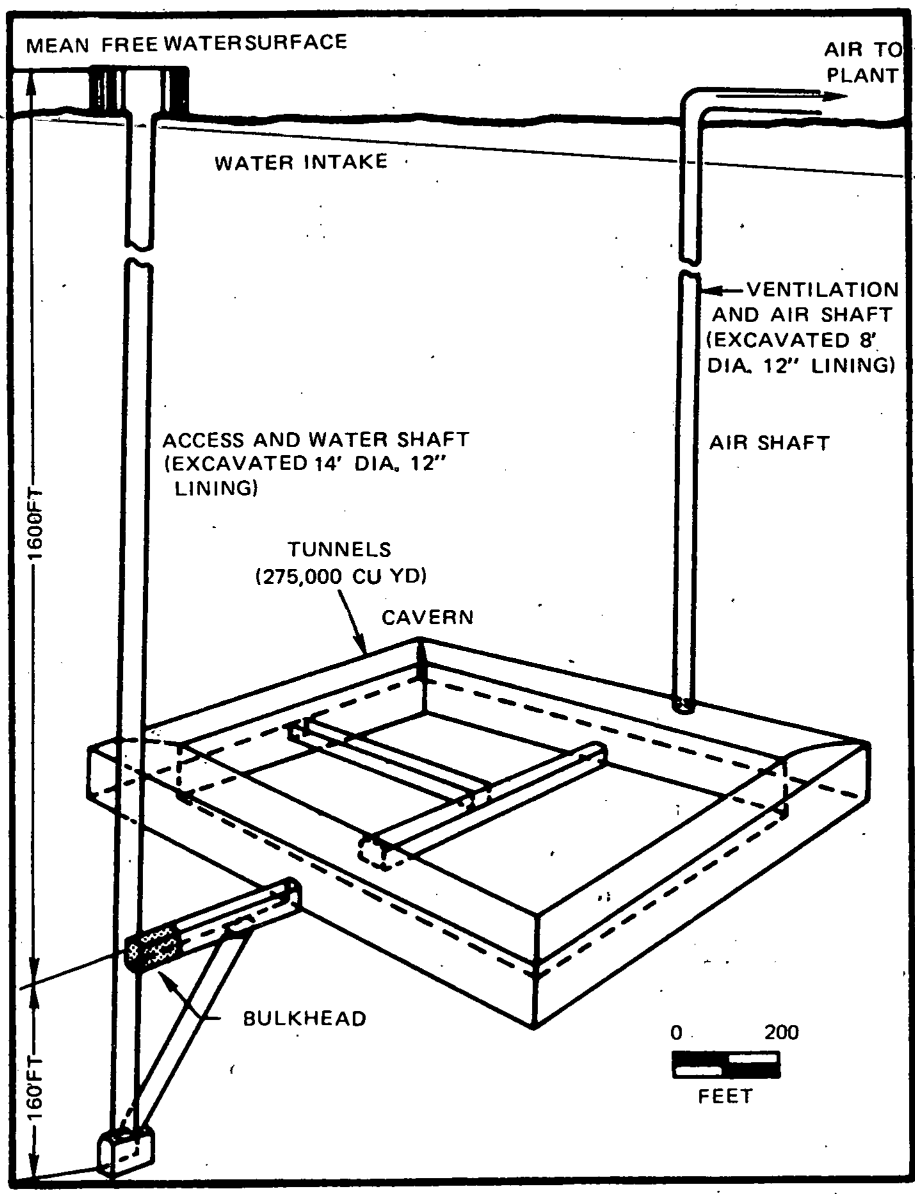


NOTE

DRILLHOLES SHOWN ON ONE BRANCH
OF SYSTEM ONLY

FIG. III-9

REFERENCE SINGLE–UNIT (280 MW) AIR STORAGE COMPLEX



CONSTRUCTION SCHEDULE FOR UNDERGROUND FACILITIES

FOR PRELIMINARY REFERENCE DESIGN

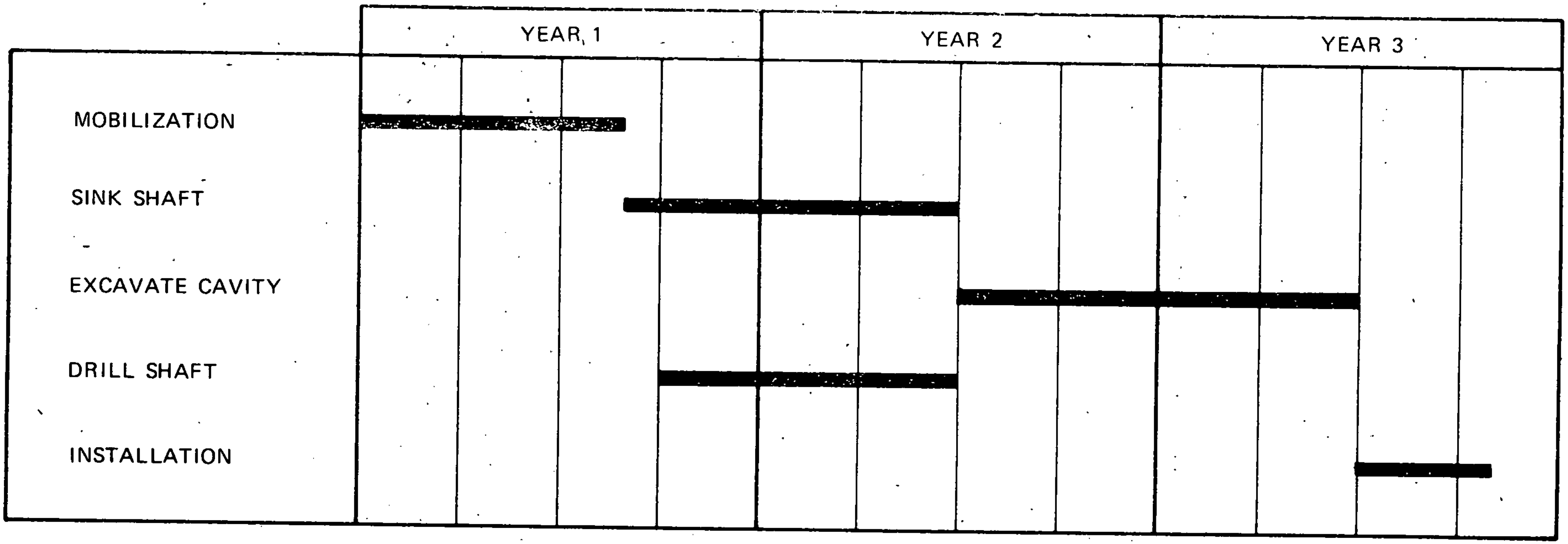
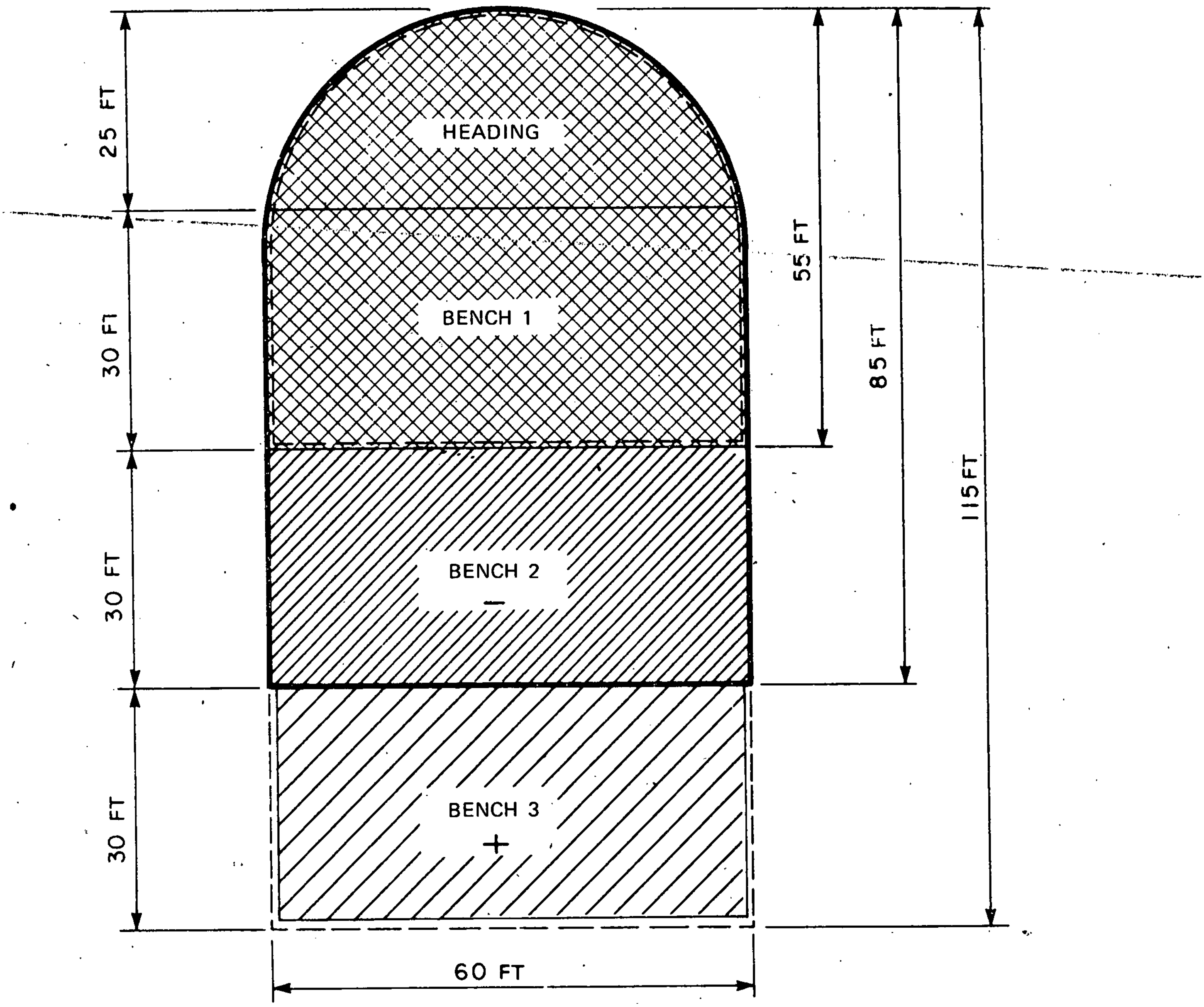


FIG. III–11

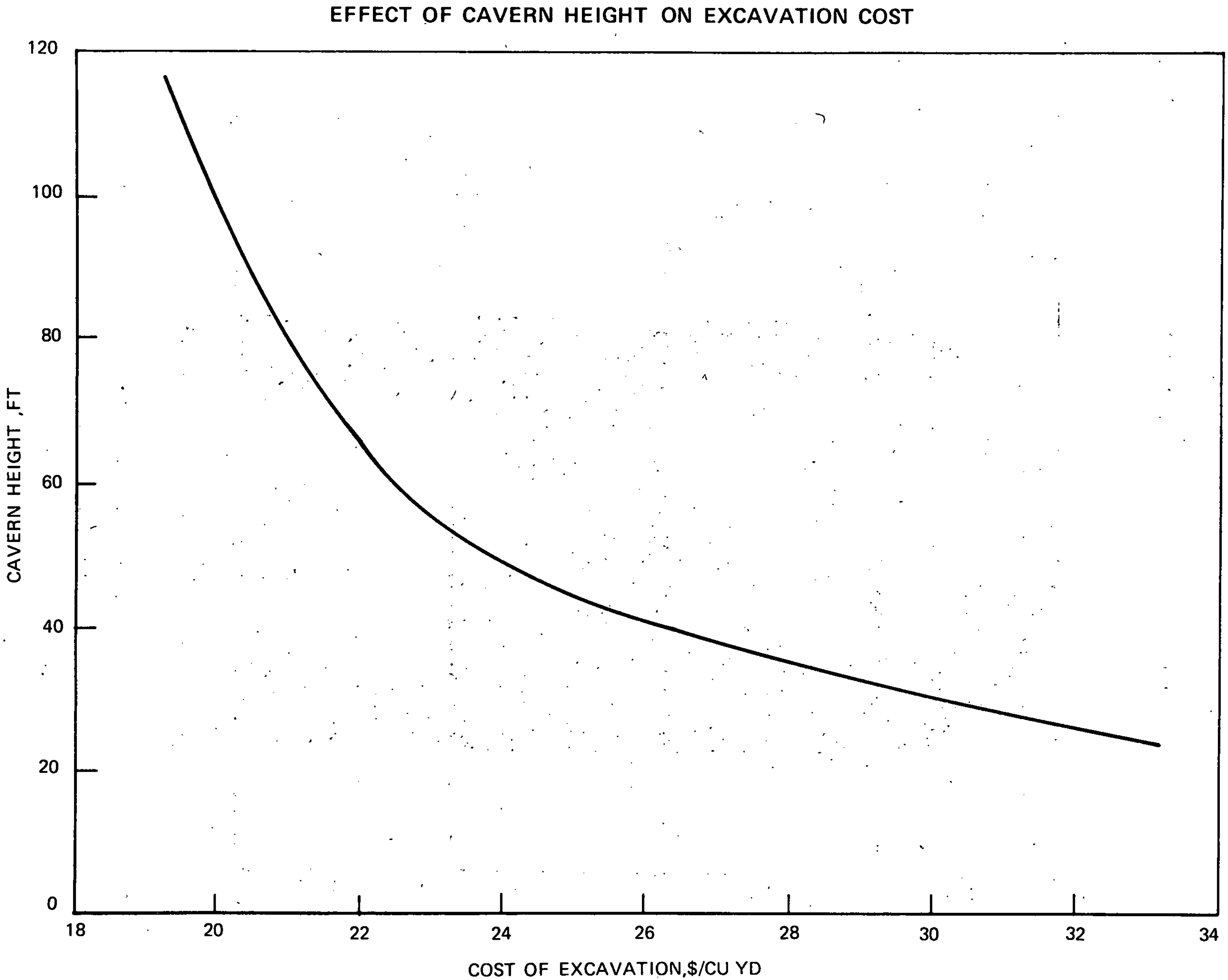
ALTERNATIVE CAVERN SHAPES



HEADING+2 BENCHES=STANDARD

HEADING+1 BENCH=ALTERNATIVE 1

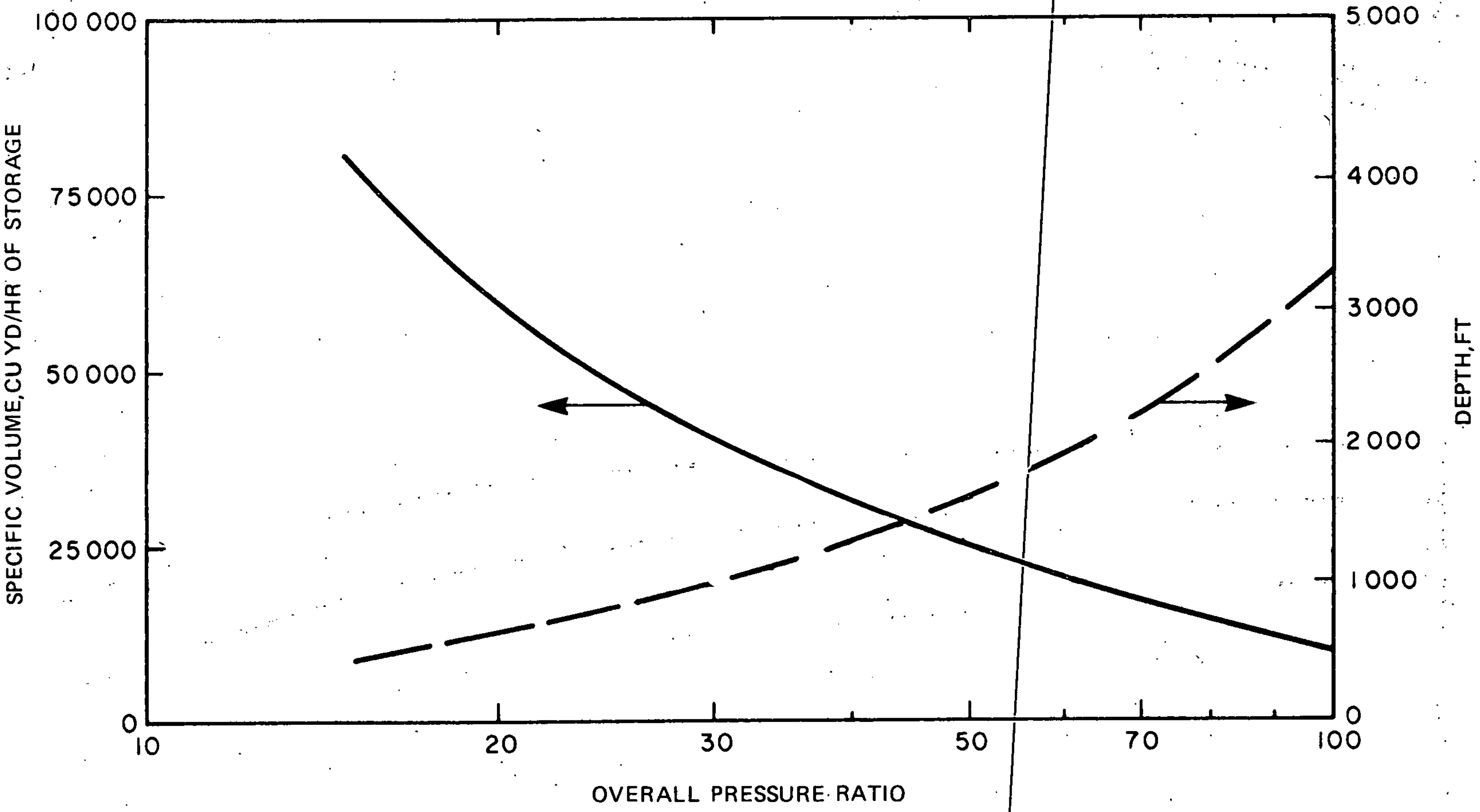
HEADING+3 BENCHES =ALTERNATIVE 2



R76–952161–5

FIG. III–13

CAVERN VOLUME REQUIREMENT

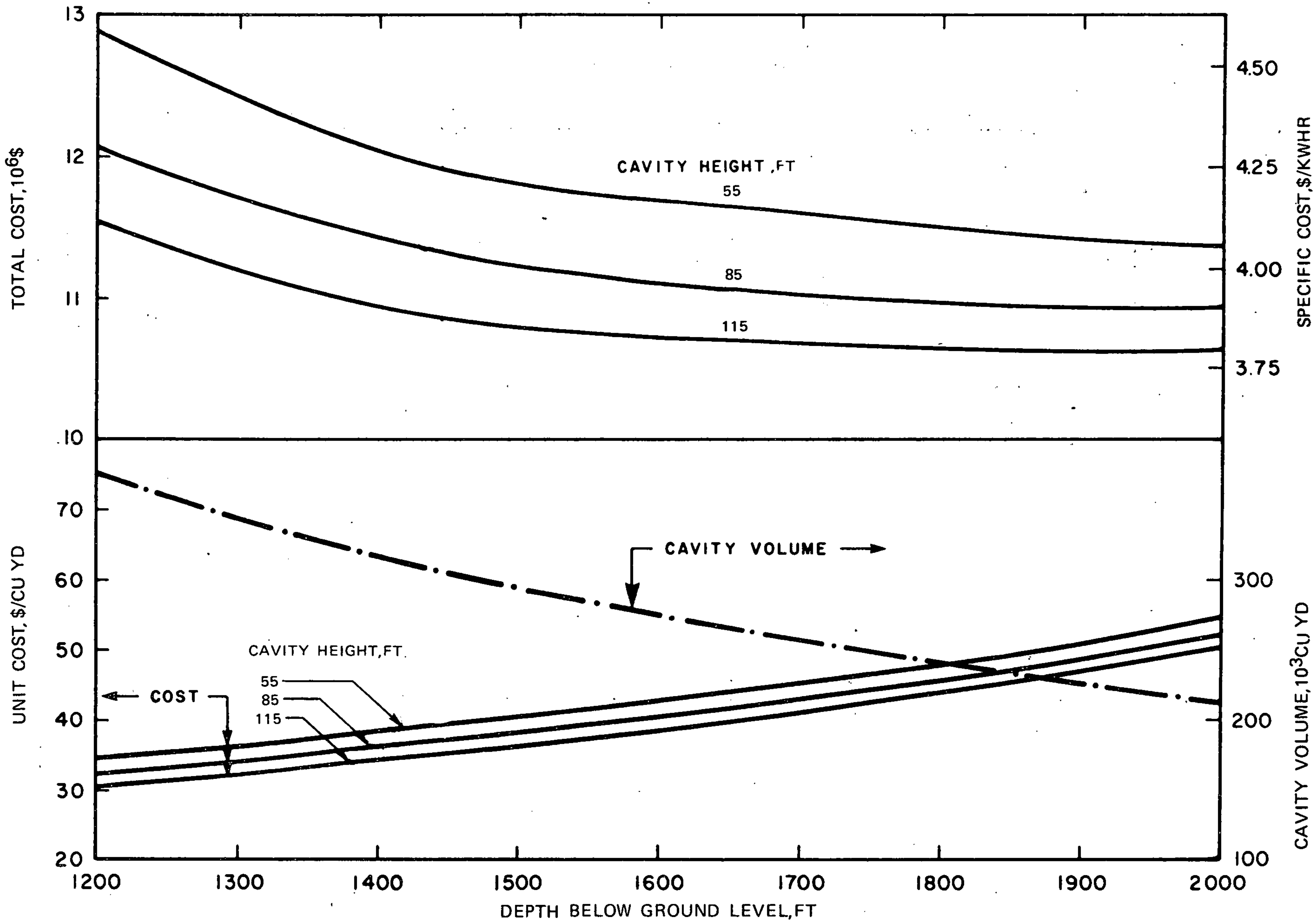


76–05–93–25

FIG. III–14

COST OF UNDERGROUND STORAGE FACILITIES

FOR REFERENCE DESIGN



R76-951261-5

76-05-93-23

FIG. III-15

PART IV

ABOVEGROUND FACILITIES EVALUATION

Ideally, the aboveground equipment for CAPS could be assembled using off-the-shelf components. Although the required level of technology is within the current state-of-the-art, off-the-shelf components specifically designed for CAPS applications do not exist. Consequently, existing equipment will need to be modified in order to reflect the unique design and operating conditions encountered in CAPS applications.

The following sections and associated appendices contain detailed descriptions of the main aboveground facilities. Included are commentaries on power generation equipment, auxiliary equipment, hydraulic facilities, and environmental considerations. Appendix B contains a detailed review of gas turbine component technology and Appendix C contains a detailed description of the advanced heavy-duty, open-cycle gas turbine (the 100-MW class FT50) which was used as the basis for designing the main low-pressure turbines and compressors for CAPS. Appendices D and E contain general descriptions of turbocompressor design and costing procedures, respectively, which could be used to estimate the characteristics of more efficient, less costly high-pressure components for future (i.e., second generation) CAPS applications. Appendix F contains a detailed description of the operation of synchronous clutches. Appendices G and H are concerned with the champagne effect, with the former presenting a simplified analysis for the diffusion of air through stagnant water and the latter presenting a two-phase fluid flow analysis.

POWER GENERATION EQUIPMENT

This section contains discussions of the general design philosophy, the low-pressure turbomachinery, the booster compressor, and the expansion turbine. Associated with this section are Appendices B, C, D, and E.

General Design Philosophy

In conducting the present program, a general design philosophy was adopted which called for maximum use of commercial or nearly developed components in order to avoid systems requiring extensive engineering or development expenditures. This approach led to the synthesizing of a first-generation system based entirely on available state-of-the-art components. As a result, some of the high-pressure equipment turned out to be relatively expensive suggesting the possibility of future cost reductions in CAPS plants based on the use of components optimized for future CAPS applications. The general design philosophy also called for a CAPS which could provide considerable flexibility in air storage pressure without altering the basic component configuration. This flexibility was deemed vital since any commercial CAPS must be capable of accommodating the specific overall pressure ratio corresponding to the air storage requirements dictated local geological conditions.

A reference CAPS design (see Fig. IV-1 and Table II-1) was established incorporating two stages of compressor intercooling, aftercooling, recuperation, and reheat. The low-pressure compressor and turbine sections utilized advanced state-of-the-art industrial design features of the type incorporated into the heavy-duty FT50 gas turbine developed by United Technologies Corporation (Refs. IV-1 and IV-2). The pressure ratio of these sections is approximately 16:1. A separate booster compressor would be used to bring the overall pressure ratio up to the desired level. Similarly, a high-pressure expansion turbine would be used to drop the pressure to 16 atm.

It should be noted that the reference design just mentioned is an initial judgment of what a commercially viable CAPS plant might be. This initial design served as a basis for preliminary equipment evaluation studies, parametric performance estimates, and system optimization work discussed in subsequent sections. As a result of these investigations, it was concluded that a slight modification in the system configuration (i.e., adding another stage of compressor intercooling) and corresponding changes in operating conditions (i.e., component efficiencies, temperatures, and pressures) would be desirable. These changes and the resulting conceptual design for prospective near-term (i.e., first-generation) CAPS applications are described in Part V in the section entitled System Optimization.

Low-Pressure Turbomachinery

The low-pressure turbomachinery comprises (refer to Fig. IV-1) the free, low and high turbines (FT, LT, and HT, respectively) plus the low and high compressors (LC and HC, respectively). These components would be based on, and utilize all the most advanced state-of-the-art (see Appendix B for a discussion of gas turbine technology) industrial design features incorporated in, the heavy-duty FT50 gas turbine (see Appendix C for a discussion of the FT50). The FT, LT, and HT FT50 turbine sections would be coupled together to comprise the CAPS low-pressure turbine section. The pressure ratio for each of these sections would be approximately 16:1.

Application of FT50 Hardware

The major portion of the FT50 engine has been retained, thereby avoiding extensive engineering and development expenditures. The FT50 design criteria have also been retained in all the low-pressure components for CAPS applications. Some areas will need redesign because of the component rearrangement, but these are regarded as low risk.

The P&WA Design Department prepared preliminary layout drawings for the rearranged FT50 components (Figs. IV-2 through IV-4). An overall view of the low-pressure turbomachinery is depicted in Fig. IV-2. Not shown are the ductwork, intercoolers, aftercooler, and high-pressure components. The CAPS arrangement has the FT50 industrial engine split into the following individually mounted modules connected by external shafting:

- low compressor and diffuser elbow duct (Fig. IV-3),
- high compressor, inlet and diffuser elbow ducts (Fig. IV-3) and
- burner section, inlet elbow duct, and four-stage turbine section (Fig. IV-4).

Presently, the FT50 free turbine rotates in the opposite direction from the high and low turbines. The arrangement of the modules in Fig. IV-2 was selected so that the rotational direction of the FT50 compressors and free turbine would be retained when installed in a new common-shaft system. The airfoils in the low and high turbines would have to be reversed to accommodate the opposite direction of rotation required for CAPS.

Intercooling would be provided between the compressors to reduce work and maintain the design value of corrected airflow through the high compressor. The high compressor would incorporate an extended power-output shaft for driving the booster compressor.

The compressor clutch would be mounted in the low-compressor inlet case while the turbine clutch would be mounted in the burner inlet section. These locations were tentatively selected, as opposed to the more maintainable motor-generator mounting, to provide in-house control on rotor and clutch assembly balance and to retain engine responsibility for the clutch oil supply system. Accommodation for axial growth and shaft angular misalignment would be built into the design of the clutches. Additional design and operating information on clutches is given in a subsequent section.

FT50 Modifications for CAPS

The CAPS arrangement basically necessitates modification of the FT50 components as follows:

Compressor section

1. Inlet case thrust bearing would be replaced by a 90,000-lb capacity bearing similar to that used in the FT50 free turbine.

2. Compressor intermediate case would be replaced by low-compressor diffuser and high-compressor inlet duct to accommodate compressor intercooling.

3. A new high-compressor diffuser case similar to that in the FT50 engine would be used, but with burner section deleted.

4. FT50 high and low compressors would be connected together to operate at 3600 rpm.

Burner-turbine section

1. Diffuser case would be modified in flowpath region since diffusion would already have occurred in high-compressor diffuser. Additionally, accommodations for mounting the clutch would have to be incorporated.

2. High-, low- and free-turbine rotors would be connected together to form a four-stage, 3600-rpm turbine.

3. New first- and second-stage turbine vanes and blades (i.e., new HT and LT turbines) would be required to accommodate the reversed direction of rotation.

4. The low turbine tangential on-board injection cooling air system would be simplified to reduce cost since its small contribution to performance would be less important in the CAPS system.

Secondary Cooling

Air enters the high compressor at 135 F. To avoid a disk rim-bore inverse thermal gradient, the FT50 low-compressor source cooling air would have to be deleted. Instead, 135-F air would be bled from the high-compressor inlet and circulated through the disk bores. This same air would then become the buffer supply to the inlet case and diffuser case bearing compartments.

Thrust Balance

Separation of the compressors and turbine into individually mounted modules necessitates thrust balancing of these rotors. Turbine thrust balance would be obtained by increasing the supply pressure from 50 to 295 psia to the seal located in the exit end of the free turbine. A new thrust-balance seal would be located in the exit end of the high compressor with the supply pressure being 250 psia at 325 F.

Duct Losses

Air would exit from the low compressor at a Mach number of 0.4 and discharge through a 2:1 area ratio diffuser. It would be difficult to hold this diffuser pressure loss to less than 3 percent, which is equivalent to losing a single dynamic head. This difficulty is associated with the high velocity flow from the low compressor and cannot be improved by lengthening the diffuser. Losses resulting from convergent flow into the high compressor would be in the range of 0.5 to 0.75 percent using a full-flow entry duct.

Low-Pressure Machinery Costs

A detailed description of the UTRC computer models capable of estimating the size and manufacturing costs of large, industrial-type gas turbine engines are given in Appendices D and E, respectively. Although several prototypes of the FT50 engine have been constructed and extensive cost information has been accumulated for this engine, sufficient modifications were made to its basic layout for the CAPS application to warrant use of the UTRC manufacturing cost model to reestimate its cost. Furthermore, since the results of the computer model correlated well with vendor-supplied cost estimates for the FT50 components, it is believed that the model cost estimates should provide a highly-acceptable basis for determining the selling price of the CAPS low-pressure turbomachinery.

The total estimated selling price for the CAPS low-pressure compressor (see Fig. IV-3) is estimated to be $2.810 million as shown in Table IV-1. This estimate includes all basic compressor components plus the casing, the inlet plenum and ducting preceding the first-stage of compression, the interstage exit and readmission ducting, and the exhaust ducting. In addition, the clutch housing and appropriate shafts are incorporated in this estimate. Not included are the special clutches and intercoolers. This price estimate, as well as that for the turbine and other components of the low-pressure section, is believed to be representative of the selling price for low-production-volume units of a single standard design.

The estimate of $4.545 million for the hot turbine section (see Fig. IV-4) includes an allowance for the complete burner section, the fuel control, all rotating equipment, the casing, bearings, shafts, inlet and discharge ducts, and in-shop assembly. The necessary allowances for the exotic material requirement of the rotating components and for the complexity of the burner component represent a large portion of this overall price estimate.

In addition to the basic, in-shop-assembled equipment, an allowance of $550,000 is made to cover the cost of the externally-mounted lube-oil cooler, the compressor barring motor (required to rotate the compressor during noncharging periods to prevent the compressor shaft from developing a permanent deflection), and all external piping and manifolds associated with the lubrication and fuel systems. Further, a $30,000 estimate is made for the price of the large castings supporting the low-pressure section on the concrete foundation.

Overall, the estimated price of the low-pressure turbomachinery alone is slightly less than $8 million as indicated in Table IV-1. This price is estimated FOB, the manufacturer's facilities and includes no allowance for installation charges.

Booster Compressor

There are many industrial compressor designs available. A partial listing of these designs includes centrifugal, reciprocating, axial-flow, helical-rotor, and sliding-vane compressors. Direct discussions with representatives from several major suppliers of industrial compressors, e.g., Allis-Chalmers and Ingersoll-Rand, resulted in almost identical recommendations for the booster compressor of the CAPS power plant. Each manufacturer recommended a vertically-split, multistage, centrifugal compressor with intercooling to maintain air temperature within tolerable material limits.

This type of compressor has been used since the turn of the century and is capable of delivering the highest discharge pressure levels required in CAPS.

Design Features

Except for those improvements made through manufacturers' research and development efforts directed at meeting new applications through innovations in design, metallurgy, and construction details, little in the way of major changes can be expected for the vertically-split design in forthcoming years. Basically, the unit is cylindrical in shape with a horizontal major axis and a single vertical parting line perpendicular to the major axis to accommodate the end plate (Fig. IV-5). It is from this single-seam design that the unit gains its high-strength capability. Units are designed with up to ten centrifugal impellers on a single shaft, although fewer stages are more common because of the high gas temperature problem. Diaphragms which are used to separate the adjacent stages contain the diffuser and turning passages. When assembled alternately with the impellers, the built-up diaphragm -impeller stack is inserted into the internal compressor cavity (barrel) as a single unit. The impeller and diaphragm materials generally are made from cast iron, whereas the case material might vary from cast iron to stainless steel depending on the specific application. Vertically-split designs, which also have found extensive use in low-density gas applications, are generally available in units with flow capabilities to 28,000 scfm, discharge pressures to 6000 psia, and input power requirements to 30,000 hp. To date, the major applications for this type of unit have been in petroleum refining, chemical processing, and refrigerant compression.

Most vertically-split centrifugal compressor designs can be coupled with various cooling processes to maintain working gas temperatures within tolerable material limits, to attain a higher pressure rise for a given unit frame size, or to minimize input power for a fixed pressure rise and flow rate. Three basic cooling processes are employed: (1) direct spray of water or other suitable liquid into the working gas on a selected stage-by-stage basis; (2) intercooling by directing the working gas away from the machine, through a heat exchanger, and then back into the next compression stage; or (3) water cooling the compressor case in and around the impeller and diaphragms. This latter technique allows compression to approach ideal, isothermal conditions and is applicable primarily to horizontally-split, centrifugal compressor designs which operate at reduced pressure levels due to the large parting line surface. For most applications, including CAPS, intercooling is preferred.

The vendors indicated that a single compressor with the specified CAPS flow rate and pressures has never been built. Although the operating requirements are within the realm of extrapolation from existing basic designs, a significant development program might be necessary. In order to operate within tolerable limits of standard materials (steels and cast iron), intercooling would be required. This approach would reduce the discharge temperatures to approximately 400 F, instead of 600 F which would occur in a straight through design without intercooling. It was calculated that reducing the inlet temperature to the boost compressor from the specified 200 F to a lower temperature (150 F) would have little meaningful effect on unit size or power requirement.

It was noted that the maximum size units of this compressor type which are currently in operation throughout the world are specially designed machines with ratings of approximately 30,000 hp at 5800 rpm. They operate on mixed refrigerant gas at inlet and discharge pressures of 130 psia and 475 psia, respectively. These units have been in operation since 1970 and have experienced impeller fatigue cracking, apparently due to high-pressure surging of the compressors under abnormal restart conditions. At high-pressure levels, these "hammer blow" surges excite the impeller disks causing fatigue cracking which, in turn, leads to subsequent impeller failure. At the elevated pressure levels anticipated in CAPS application, such factors might require extensive investigation to ensure all possible efforts have been expended to generate the most reliable design.

Booster Compressor Costs

As noted in the preceding section, the booster compressor represents an extension of the present state of the art of compressor design. However, representatives from both Allis Chalmer and Ingersoll-Rand suppliers of this equipment believe that a compressor engineered to CAPS specifications could be produced if an acceptable market is present (Refs. IV-3 and IV-4). One manufacturer believed such a unit could be supplied within thirty months after completion of the development program.

Both major manufacturers provided cost estimates, but tempered them with the comment that because of the difficulty in anticipating what the final system requirements and specifications would be, the only values they could supply are for estimating (budgetary or first-pass) purposes. One manufacturer estimated that a compressor to boost 750 lb per second of air from 15 atm to 60 atm (4:1 ratio) would cost about $975,000. The other estimated the price to be between $1.0 and $1.2 million. Both estimates assume that only the soleplate and lube oil console for the compressor would be supplied by the manufacturer. In neither case were intercoolers, aftercoolers, nor main driver units included.

Considering the severity of the application, the similarity of the price estimates adds confidence to the cost estimate used in this study. Variation of the cost of the basic compressor unit with the number of stages (to provide alternate pressure ratios) was requested from one manufacturer. The response was that, in general, a sum of approximately $100,000 would either be added to or removed from its basic six-stage machine for the respective addition or removal of one stage. This modest change is due to the nature of the vertically-split compressor design and to the use of a common frame size for similar units. However, it is cautioned that this change should not be generalized to other units or to greater-than-one-stage variations in the basic designs on which the quotes were based. Completely new estimates should be established in these latter cases. Except for potential increases in the costs of these machines with increased costs of the labor and/or material content (inflation), no radical change in unit design is expected that would substantially change the overall level of these vendor estimates.

As is the case with any new or advanced concept, the development funds required to meet a desired performance goal become an expense which ultimately must be paid from the cash flow of the program. Such funds requirements were neither investigated nor included in the overall selling price estimates provided by the vendors. Experience has shown that extensions of state-of-the-art machinery could, under severe design requirements, require a substantial funding effort. Manufacturers will make critical reviews of these expenses in view of their estimates of the market opportunities before they commit to the production of a unit such as the CAPS booster compressor. The only alternative in absence of a commercial manufacturer's commitment toward CAPS equipment would be for the Government to underwrite such development based on its belief that the overall benefit of the CAPS program to society as a whole would be greater than the extent of its funding commitment.

Axial Flow Booster Compressor Designs

In view of the fact that vertically-split, centrifugal compressor units represent old, mature technology, it was decided to investigate whether an axial-flow unit based on the use of a highly-efficient flow path might be an attractive alternative. The UTRC compressor sizing program described in Appendix D was exercised to define the physical parameters of such an axial-flow machine. From this program it was determined that in order to accommodate the flow of the system in a single machine, a rotational speed in excess of 3600 rpm would be necessary. (The 3600 rpm restriction is dictated because this is the maximum, ungeared speed at which an ac motor will operate.) Alternative design options such as use of direct current drive and power conditioning, the use of multiple compressors to reduce the mass flow per unit, or the use of a gear drive to increase compressor speed and thereby reduce its size were all considered unacceptable from economic and, in some cases, reliability standpoints. Such problems as bending moments and blade stress were not even considered in this cursory analysis, even though such problems would have to be surmounted before an axial-flow design could become practical. As a result of these preliminary analyses, it was concluded that pursuit of this alternative approach could not be justified on the basis of the potentially small amount of funds which might be saved and on the high degree of technical risk involved.

Expansion Turbines

Completely packaged hot gas expansion turbines are not available commercially in sizes much in excess of 15,000 hp. As a result, for high-power gas expansion applications purchasers must generally select single or multiple steam turbine units suitably modified to match the required hot gas operating conditions. In the strictest sense, modifying a steam turbine does not meet the "off-the-shelf" availability criterion as would the smaller packaged gas expansion turbines. In addition, steam turbines would be limited to inlet temperatures of 1000 F, and their operation with a working fluid other than steam would compromise their performance somewhat. Nonetheless, it is believed that the savings in specific cost ($/hp) relative to a multiple of packaged gas expansion turbines would most often be sufficient to justify selection of the modified steam turbines.

The approach followed in investigating gas expansion equipment, therefore, was to make inquiries of several independent manufacturers of steam turbines to identify steam units which could be esily modified to provide the 50-60 MW required from the CAPS expansion turbine. Most of the detailed performances, cost, and design work was based on data provided by De Laval for twin 27-MW turbines. Late in the program information was received from Stal Laval on a larger turbine capable of producing 60 MW at 3600 rpm.

Design Features

Most turbine manufacturers follow a philosophy similar to that identified for manufacturers of the booster compressor. Fig. IV-6 illustrates a typical expander design. That is, they use a high degree of interchangeability between various units in their respective product lines. The interchangeability between such components as disks, supports, journal and thrust bearings, cases, governors, and exhaust ends improves product reliability and leads to a lower manufacturing cost in products which then can be custom engineered for each customer application.

The nozzles of the expansion turbines are generally machined in diaphragm plates located between rotating stage members. In most instances, manufacturers of this type of equipment design their units to use low-reaction or impulse- (Curtis-) type airfoils, in contrast to the use of high-reaction stages commonly associated with gas turbines. Since all the pressure drop in each impulse stage occurs across the fixed nozzle, this type of design reduces significantly the thrust loads on the shaft of the machine. An impulse blade design also extracts a greater amount of work per stage and, for a given frame size (machine rating), leads to a unit with fewer stages than if reaction stages were used. In these units, the initial high-pressure stage blades are of a shrouded design. However, as the blade length increases, lashing wires are used to add strength and to prevent undue vibration. Most blades and nozzles contain high-chromium content stainless steel (400 series) or Monel, materials which are selected because of their characteristic strength, erosion/corrosion resistance, and excellent damping qualities.

Other features of the multistage (steam) expansion turbines included case designs whose walls are contoured in an axial direction so heat would be conducted rapidly away from the rotor, thereby minimizing thermally-induced stresses which could cause failure of the machine. The expander casing is usually split horizontally on the plane at the centerline. The turbine cases and exhaust cases would be partly forged and partly fabricated in an effort to reduce costs. Turbine lubrication oil systems are generally engineered to meet the specific requirements of a particular application, although a high degree of interchangeability exists among individual components in the lubricant systems of differently rated units. A turning gear arrangement, a subsystem similar to the barring motor concept used in large gas turbines, is generally provided with the expander units to rotate the turbine shaft slowly during shutdown, thereby preventing the shaft from bowing and taking on a permanent set.

Expansion Turbine Costs

Two steam turbine manufacturers Stal Laval (Ref. IV-5) and De Laval (Ref. IV-6), provided cost estimates for their units in a CAPS application. These cost estimates were accompanied with many of the stipulations expressed by the booster compressor manufacturers and should be viewed in that perspective. The estimated price for the De Laval unit is $2,550,000 for the turbine and generator, with $1,530,000 for the turbine alone. Two of these 27-MW units would be required. Stal Laval estimated the price to be $1,650,000 FOB Swedish port, exclusive of generator, for one of their units capable of generating 60 MW in CAPS. Part of the difference between the two estimates can be accounted for in the import fees and transportation costs which would have to be added to the Stal Laval estimate. However, during the installation there could be substantial savings accrued to the Stal Laval unit because of the reduced piping, foundation, etc. associated with installing a single unit versus two units. Unfortunately the Stal Laval estimate was received very late in the program. Consequently, the detailed power plant design and cost estimates in Part V, reflect the De Laval equipment and costs.

Hot Gas Expanders

It was indicated previously and in Ref. IV-7 that because of limited commercial applications, most turbine manufacturers do not make large gas expanders, but rather attempt to adapt steam turbine units to a particular use. Although this approach often compromises the performance of the turbine, it does make considerable economic sense since there is generally no large development program required for adapting the steam turbine, provided such criteria as long life expectancy, and low maintenance can be achieved. However, as part of this overall CAPS study, an attempt was made to design high-pressure, hot gas expanders specifically for the CAPS operating conditions. It is believed that because of unique design requirements, such as the high-efficiency flow path and metallurgy, this type of turbine could likely be produced only by one of today's large gas turbine manufacturers.

The computer model discussed in Appendix D was utilized for the expander turbine design exercise. It was initially assumed that a half-size expander would be desirable such that two units operating in parallel would handle the total flow. Based upon this assumption two separate expander turbine designs were undertaken corresponding to adiabatic efficiencies of 80 percent and 85 percent, respectively. The former is typical of present technology, whereas the latter is representative of second-generation technology. The characteristics of both turbines are presented in Table IV-2. Both incorporate a reaction value of 0.2 (a value of 0.0 is a "pure" impulse system and a value of 0.5 is a "true" reaction machine) and require seven individual stages. Estimated physical dimensions are also given in the table. Based on experience, the expander with the higher efficiency rating should have a lower cost per unit of output, although a detailed cost estimate at this time would be meaningless without an indication of the total potential unit sales and dates of production.

Should it be desirable to have only a single full-size expander, this could be accomplished with a larger turbine having only four stages. The estimated design characteristics for turbines with 80 and 85 percent efficiency are presented in Table IV-3. The prices of these machines were not calculated for reasons similar to those noted previously.

In all the expander unit designs, the material choice is within the present state-of-the-art, and blade and vane cooling is not required. The designs of these expander units should not be a major undertaking and would await only the economic justification in terms of unit sales for their development.

AUXILIARY EQUIPMENT

This section contains discussions of key auxiliary equipment (clutches and recuperator) required for CAPS applications.

Clutches

The operating sequence of a CAPS power plant requires that the turbine and compressor sections be alternately connected and disconnected from the motor/generator. This necessitates the inclusion of connecting devices between the motor/generator and turbine section, and between the motor/generator and compressor section. In addition, if it is desired to operate the low-pressure turbomachinery in a simple-cycle mode, an additional connecting device must be installed between the low-pressure compressor and the booster compressor.

In the study, two classes of connecting devices were considered: couplings and clutches. A coupling "makes a semipermanent connection between two shafts" (Ref. IV-8). A connection is semipermanent in that once the connection has been made by installation of the coupling, it cannot be disconnected until the coupling is at least partially disassembled. As a consequence, couplings, whether rigid, flexible, or fluid, can only be connected or disconnected to the two shafts when they are stationary and unloaded. The utilization of such a connecting device in a CAPS power plant is not desirable. The CAPS power plant must have the ability to switch as rapidly as possible from compression to generation and vice versa. This rapid switching cannot be accomplished if the motor/generator and turbomachinery must be brought to complete rest before connections and disconnections can be made.

Clutches, unlike couplings, "permit the disengagement of the connected shafts during rotation" (Ref IV-8). In this study, a review was made of clutch technology in order to identify any technological limitation which clutches might impose on CAPS applications. No such limitations were found. All major gas turbine manufacturers routinely use clutches in foreign synchronous condenser applications. In addition, Brown Boveri and Company (Ref. IV-9), and Stal-Laval Turbine AB, (Ref. IV-10) have both conducted extensive evaluations of clutches as part of their CAPS explorations.

Brown Boveri is a major supplier of equipment for the 290 MW-Huntorf CAPS power plant (Ref. IV-11). Brown Boveri chose a clutch designed by SSS Gear Works of London, England, which will be built under license by Renk of Augsburg, Germany. Stal-Laval was an early pioneer in defining the basic CAPS concept and at one point appeared on the verge of participating in the first installation (250 MW) of this type (Ref. IV-12). Stal-Laval chose to use a clutch designed and built by MAAG Gear-Wheel Company, Ltd., of Zurich, Switzerland.

The remainder of this section contains detailed descriptions of SSS and MAAG clutch designs and operation. These two clutches were used as a basis for discussion because they are representative of the latest clutch technology and information was readily available. Other manufacturers, such as Zurn Industries (Ref. IV-13), have the capability to make similar large clutches, but no attempt was made to survey all manufacturers.

SSS Clutches

The material for the following discussion on SSS clutch design and operation was primarily extracted from Refs. IV-14 through IV-18.

A clutch would be installed between the output shaft of the driving component (i.e., from the motor/generator during compression or from the turbine section during power generation) and the input shaft of the driven component (i.e., to the compressor during compression or to the motor/generator during power generation). Engagement of the two shafts would occur whenever the rotational speed of the output shaft overtakes the rotational speed of the input shaft. Once engaged, torque can be transmitted from the output shaft to the input shaft. Disengagement would occur whenever the output shaft rotational speed drops below that of the input shaft. Both engagement and disengagement proceed automatically once the appropriate commands have been signaled to the hydraulic locking systems.

Turbine Clutch Construction

The clutch can be considered to consist of four basic subassemblies. The input assembly, the main sliding assembly, the output assembly, and the relay helical sliding component (see Fig. IV-7). The input assembly is bolted to the gas turbine output shaft and forms a support for the main sliding assembly. The main helical splines impart axial motion to the main sliding assembly and, also, form part of the torque path when the clutch is engaged. The main sliding assembly has a set of internal helical splines which are in mesh with the helical splines of the input assembly. External spur gear teeth on the periphery of the main sliding assembly engage with the internal spur teeth on the output assembly. The main sliding assembly is also provided with a set of helical splines for activation of the relay clutch. The output assembly is bolted to the motor/generator input shaft. When the clutch is fully engaged, torque is transmitted from the main sliding assembly external gear teeth through the output assembly internal gear teeth.

The relay clutch ring forms part of the output assembly. It has internal spur gear teeth which engage the external teeth of the relay helical sliding component to initiate engagement of the main sliding assembly. The relay clutch ring also carries a set of four primary pawls. The purpose of the pawls is to align the relay clutch ring teeth precisely for interengagement at shaft synchronism, then initiate movement of the relay helical sliding component along its helical splines to engage the relay clutch ring teeth. The relay clutch ring teeth then initiate movement of the main sliding assembly to engage the main clutch teeth. The relay helical sliding component is in sliding engagement with the helical splines on the main sliding assembly and has external spur gear teeth. A set of four secondary pawls are also carried by this component.

The clutch also incorporates a powerful double acting dashpot, which is filled with oil from the clutch lubrication system (Fig. IV-8). This dashpot effectively cushions engagement and disengagement of the clutch and, when the hydraulic locking oil is shut off, ensures that the clutch will only disengage on the application of a sustained negative torque. During gas turbine deceleration, the clutch can be locked into engagement by a flow of hydraulic locking oil into the dashpot.

The primary pawls which are mounted on the relay clutch ring (Fig. IV-9) always rotate with the motor/generator. These pawls are lightly spring loaded into action and are nose heavy, so that when the motor/generator speed exceeds a predetermined speed, say 600 rpm, the centrifugal weight of the pawl nose keeps the pawls out of engagement with the external ratchet teeth of the relay helical sliding component. These pawls are therefore effective to engage the clutch at shaft synchronism when the motor/generator is at any speed between rest or 600 rpm.

The secondary pawls are mounted on the relay helical sliding component (Fig. IV-10) which is driven by the gas turbine. The pawls are nose heavy and are brought into action centrifugally when the gas turbine shaft rotates. They are lightly spring loaded to be out of contact with the ratchet teeth in the bore of the relay clutch ring when the turbine is below about 350 rpm. When the gas turbine is accelerating to engage the clutch with the motor/generator already rotating at full speed, these pawls skim on the rim of oil formed centrifugally around their ratchet teeth until, when approaching synchronism, full ratcheting action will commence in readiness to engage the clutch at synchronism. With the above pawl arrangement, both sets of pawls are inert when the motor/generator is at full speed and the turbine is at rest, but either one or the other set of pawls is effective to engage the clutch at synchronism at all operating speeds.

Turbine Clutch Operation

On starting up the complete system from rest, the clutch engages immediately as the gas turbine commences to rotate. When the motor/generator is rotating at full speed and electrically synchronized, the power from the gas turbine can be increased as required. When disengaging the clutch at speed, the gas turbine can be shut down and will come to rest while the motor/generator continues to rotate with the machine operating in the charging mode. During disengagement the clutch will disengage immediately as the power turbine slows down relative to the motor/generator. When changing from charging operation to generating, it is only necessary to accelerate the turbine up to full speed and, at the instant the turbine-driven components tend to overtake the motor/generator, the clutch will engage automatically.

A detailed description of the turbine clutch operation is presented in Appendix F.

Compressor Clutch Construction

The configuration of the main compressor clutch is basically similar to that of the turbine clutch, but with the following important differences:

a) The main sliding assembly of the compressor clutch is shifted initially by an external servo-mechanism; the movement being completed by oil pressure applied to the dashpot which, in this clutch, acts as a hydraulic cylinder.

b) The main sliding assembly is connected to the compressor mounted assembly by straight sliding splines.

c) A baulking mechanism is provided to prevent shifting the main clutch from the pawl-free position to the ratcheting position when the input (generator) speed exceeds the output (compressor) speed.

d) Since the clutch is only required to engage at speeds between zero and the maximum speed of the pony motor which drives the compressor (approximately 300 rpm), the primary and secondary pawls are not required.

In order to prevent compressor clutch engagement during the power generation mode, an externally controlled servo-mechanism is used to locate the clutch in a locked-out (pawl-free) position. In this position the pawls are free of the ratchet ring and the clutch will not engage. The compressor clutch servo-mechanism consists of a hydraulic cylinder connected by a spring link to a control fork. This control fork is fitted with two white-metalled thrust pads which act on the external flange of the clutch main sliding assembly. Lost motion is provided so that the servo-mechanism only shifts the clutch teeth part way into engagement, but is capable of shifting these teeth from the fully engaged to the fully disengaged position.

In the baulking mechanism arrangement, the clutch input member carries a grooved member with white-metal faces. Contained within this groove is a ring with teeth which engage with mating teeth in the clutch sliding component. When the clutch is overrunning in one direction, the slight friction between the white-metal bearing and the ring causes the ring to rotate through a very small angle, so the teeth contact one set of flanks of the teeth on the clutch sliding component. When the direction of rotation is in the opposite direction, the friction of the bearing causes the teeth to contact on the other flanks. When the clutch sliding component is shifted towards the ratcheting position, the meshing teeth described above are subjected to relative axial travel. The flank of one set of teeth has a step, so that if the direction of relative rotation is incorrect, this step will baulk the clutch from being shifted from the pawl free to the ratcheting condition.

The booster compressor clutch would be constructed similar to the main compressor clutch since the engagement/disengagement sequences associated with the booster compressor relative to the main compressor are analogous to the engagement/disengagement sequence associated with the main compressor relative to the motor/generator.

Compressor Clutch Operation

The clutch provided between the motor/generator and the compressor is required to engage whenever the gas turbine, driving through the motor/generator, tends to overtake the compressor driven by its low-speed pony motor. Once engaged, the clutch will transmit torque from the motor/generator to the compressor.

The booster compressor clutch is required to maintain engagement between the booster compressor and main compressor during normal CAPS operation and to disengage during simple-cycle operation. A detailed description of the main compressor clutch and booster compressor clutch operation is also presented in Appendix F.

Huntorf CAPS Application

In the Huntorf application, the compressor clutch must transmit 58 MW at 3000 rpm while the turbine clutch is required to transmit 290 MW at 3000 rpm. The torque transmitted by the compressor clutch is well within the present state of commercial technology. The high-speed, high-input requirement of the turbine clutch will represent an advancement since the largest 3000-rpm SSS clutch presently utilized by industrial gas turbines is rated at 80 MW. However, the expertise to manufacture high-torque clutches is available since a 340,000 hp clutch for use at 600 rpm has been built for a German pumped-hydro storage power plant (Fig. IV-11). The stresses that will be experienced in the Huntorf turbine clutch will be lower than those experienced in currently operational high-torque clutches. The physical size of the 290-MW turbine clutch is only 39-1/2 inches in diameter at its maximum point and 28 inches long between the two shaft flanges. The 58-MW compressor clutch is only 30-1/2 inches in diameter and 28-3/8 inches long.

Clutch Costs

Despite the complex sequence of operations and the high-torque transmission capabilities required of the clutch, its capital cost is relatively modest. Discussions with Waukesha Bearing, SSS's U.S. licenses (Ref. IV-18), revealed that the turbine clutch for the reference design in the study would cost in the neighborhood of $100,000. The main compressor and booster clutches would each cost around $70,000. The total capital cost of these three clutches is therefore $240,000 or approximately $1/kW. Waukesha would not manufacture the clutches in this country because some of the special metal alloys required for SSS clutches are available only in Europe. They could be manufactured in England and the assembled units shipped to the U.S. The cost estimates quoted above include the necessary import duty and transportation fees.

Clutch Maintenance and Lifetime

Since the Huntorf facility is not scheduled for operation until 1977 there has, obviously, been no operating experience associated with the SSS clutches to be installed there. However, information provided by SSS indicates that for a similar type of application (a hydro-electric pumped storage plant) over 1,000 engagements have been made on each of four 600-rpm, 100-MW clutches. All four clutches in this installation have given approximately six years of "trouble-free" service and have experienced no noticeable wear.

MAAG Clutches

The material for the following brief discussion of MAAG clutch designs and operation was extracted primarily from Refs. IV-19 and IV-20.

Turbine Clutch Construction

There are six major components to the MAAG clutch: the input coupling sleeve, spool piece, yoke, servomechanism, synchronizing mechanism, and output coupling sleeve (Figs. IV-12 and IV-13). The input coupling sleeve is bolted to the input shaft and forms a support for the spool piece. The yoke is attached to a servomechanism which hydraulically locks the yoke in the desired position. The servomechanism consists of a control piston and a main piston. The spool piece has external spur gears on both ends.

The torque is transmitted from the input shaft to the output shaft through the spool piece and its two sets of teeth when the clutch is engaged. The teeth of the spool piece and the teeth on the input coupling sleeve are always in contact, whether the clutch is engaged or disengaged. The teeth on the spool piece and those of the output shaft are only in contact during clutch engagement. The spool piece has a set of ratchets, mounted along the internal periphery which are used during the engagement/disengagement sequence.

The output coupling sleeve is bolted to the output shaft and also serves as a support for the synchronizing mechanism. There are two sets of teeth on the output coupling sleeve. The teeth along the outer periphery are used to transmit the torque during clutch engagement. The internal set of teeth are half helical and half spur and are in contact with the synchronizing mechanism while the clutch is engaging and during engagement. The synchronizing mechanism is placed internal to the spool piece. It has a set of pawls which contact the spool piece ratchets while the clutch is engaging and a set of external teeth which contact the internal teeth of the output coupling sleeve inner periphery during the initial steps of the engagement sequence. The pawls serve much the same purpose as they do in the SSS clutch in that they are actuated at input and output shaft synchronous speed. Otherwise, the pawls are ratcheting when the differential speed of the two shafts is low. At high differential speeds the pawls are separated from the ratchets by a thin film of oil.

For large capacity clutches that would be required in CAPS application, MAAG modified their basic design to increase torque-carrying capacity and reduce imbalance problems. They added conical surfaces on the external face of the spool piece and the internal face of the output coupling sleeve to provide alignment during engagement. In addition, mating flat surfaces perpendicular to the axis are provided on the spool piece and the output coupling sleeve. When under load, these mating surfaces are forced together and the torque is transmitted across them.

Turbine Clutch Operation

On starting up the complete system from rest, the clutch will automatically engage once the locking servomechanism has been programmed for engagement and the gas turbine begins to rotate. The gas turbine and motor/generator can then be accelerated to synchronous speed for power generation. For disengagement of the gas turbine to occur automatically during gas turbine slow down, the locking servomechanism must be programmed for disengagement. The gas turbine can then be brought to rest in preparation for the charging phase. Lastly, when changing from charging to generation, the gas turbine can be accelerated to full speed, and once the engagement command has been signaled to the locking servomechanism, engagement will occur automatically.

A detailed description of the turbine clutch operation is provided in Appendix F.

Compressor Clutch Construction

It is anticipated that the turbine and compressor clutches will be substantially different in design, but details for design of the MAAG compressor clutch are not available. However, since the compressor clutch will be designed to transmit a lower load than the turbine clutch (160 MW as compared to 250 MW), the compressor clutch design should merely be an extension of current technology.

MAAG CAPS Application

MAAG has not participated in an actual CAPS power plant installation. However, Stal Laval's interest in the concept led them to order a large clutch from MAAG. The clutch is presently being built and will be tested by Stal Laval in late 1976 (Ref. IV-21). It is rated at 250 MW at 3000 rpm. Physically, it is 36 inches long between shaft flanges and 38 inches in diameter.

Clutch Costs

Reliable cost estimates for MAAG clutches are not available, but they are expected to be about the same as for the SSS clutches (approximately $1/kW).

Recuperator

A recuperator is placed in a CAPS power plant in order to reduce the fossil fuel consumption and thereby reduce the electrical energy cost. This reduction in fossil fuel heat rate can be appreciable (i.e., from 5300 Btu(LHV)/kWhr with no recuperator to 4130 Btu(LHV)/kWhr with an 80-percent effective recuperator for a cycle pressure ratio of 66.3 and 2000-F high turbine temperature). For fuel costing $\$2.50/10^6$ Btu this saving is equivalent to an electrical energy cost reduction of $2.9 mills/kWhr. This energy cost reduction can be converted into a recuperator breakeven installed cost of $25.4kW (1560 hrs/yr plant utilization and 18-percent annual fixed charges). This, in turn, can be translated into an equipment cost of about $13.7/kW after reducing the installed cost by the appropriate factors for contingency, engineering and administration, and indirect costs. This cost appeared to be a reasonable goal to meet, thus an effort was expended to design a recuperator for the reference CAPS. The following paragraphs describe a recuperator design estimated to cost $7.7/kW which yields a net savings of about $6/kW.

Recuperator Design Conditions

For the design effort the conditions associated with an 80-percent effective recuperator which could be installed in a power plant with a cycle pressure ratio of 66.3 and a high turbine temperature of 2000 F was selected. The pertinent design conditions are displayed in Fig. IV-14.

These conditions differ somewhat from those presented in Ref. IV-7 in that they reflect CAPS operation with storage at 66.3 atmospheres as opposed to 50 atmospheres. Consequently, it reflects the results of the preliminary systems optimization which yielded an optimum overall pressure ratio of 66.3 (see section on System Optimization). It also reflects the final pressure drop refinement during which the actual pressure drop in the air shaft was calculated. This calculation yielded a value of 71 psia (7.3 percent of the cavern pressure) for a recuperator inlet pressure of 902.7 psia as opposed to the pressure drop arbitrarily assumed equal to 10 percent of the cavern pressure which was used in the previous recuperator design exercise.

There were two other changes in design philosophy as compared to that used in Ref. IV-7. The recuperator was split so that it could more conveniently be physically integrated into the power plant plot plan on either side of the inlet air filters. Secondly, upon consultation with Combustion Engineering (Ref. IV-22) large tubes (2 inches in diameter) were used in the design as opposed to the small tubes (1 inch in diameter) used in the original design. It was indicated that the larger tubes would be more compatible with industry practice for a recuperator of the size envisioned, and also the larger tubes would be of assistance in overcoming the relatively small pressure drops allowed for the large flows encountered.

From the outset four specific design considerations were recognized: (1) the high pressure in the recuperator cold side, (2) the low pressure drop permitted in the recuperator hot side, (3) the large mass flow coupled with near atmospheric pressure in the recuperator hot side, and (4) the possibility of corrosion near the hot side exhaust.

Recuperator Design Rationale

The high pressure ratio between the cold and hot sides of the recuperator precluded the use of a plate-fin heat exchanger since they are generally limited to a range of pressure ratios from approximately 16:1 to 20:1. Consequently, a shell-and-tube heat exchanger was selected with the hot, low-pressure gas flowing through the shell and the cold, high-pressure gas flowing through the tubes.

The combination of the large volume flow and the small permissible pressure drop in the hot side was the major design constraint. The volume flow could not be varied without compromising the performance of the recuperator. Any attempt at increasing the allowable pressure drop would decrease the volume flow because it would increase the turbine back pressure. Since the turbine back pressure forms the denominator in the pressure ratio expression, small changes in this value will result in large variations in the pressure ratio expression. Consequently, small increases in back pressure produce a significant decrease in pressure ratio resulting in reduced output power. Faced with this constraint, it can be readily understood why any design would have to have a large flow area between the tubes to handle the hot gas flow without producing a large pressure drop.

The large difference in gas density between the hot and cold sides means that the overall heat transfer coefficient would become very much a function of the hot-side convective heat transfer coefficient. Combining this with the low flow velocities required to maintain a low pressure drop on the hot side would mean that the hot side heat transfer coefficient would be low, which in turn, implies a large hot-side heat transfer surface. In addition, the temperature drop across the recuperator at any given cross section is inversely proportional to the convective heat transfer coefficient and the heat transfer surface area associated with that coefficient. If a thin-wall, bare tube design with its approximately equal inner and outer surface areas were used, corrosion problems would result at the hot-side exhaust as the exhaust gas temperature and hot-side metal temperatures are brought below the dew-point (Fig. IV-15). As a result, in order to reduce the physical size of the recuperator and the potential for corrosion, finned tubes were used in the design to increase the hot-side heat transfer surface and reduce the temperature drop on the hot side (Fig. IV-15).

The hot-side convective heat transfer of the finned surface selected was approximately equal to the cold side heat transfer, as a result the temperature of the metal was about midway between the hot and cold gas temperatures. At the cold-side inlet, 120-F air is being heated by 372-F exhaust gas. With the approximately symmetrical temperature drop, the metal temperature on the gas side is marginally equal to the levels which are normally considered acceptable to prevent condensation of sulfuric solutions from the exhaust (Refs. IV-23 and IV-24). Generally, one of several schemes are employed to overcome this problem. Increasing the hot side inlet temperature, recirculating part of the heated cold side gas into the cold side inlet thereby increasing the cold side inlet temperature, utilizing parallel flow heat exchangers rather than a counter flow configuration, or introducing corrosion-resisting materials are four possible approaches. The first three approaches were deemed impractical, the first, because of the reduced output accompanying increased turbine exhaust temperature, the second, because of the magnitude of the recirculated flow (about 15 percent of the cold-side flow) to achieve an acceptable metal temperature of 225 F and because it does not eliminate the problem during start-up; and the third, because of the increased heat transfer surface area required for a parallel flow heat exchanger relative to a counter flow heat exchanger.

Recuperator Design

With the aid of the heat exchanger design procedures in Refs. IV-25 through IV-29, the preliminary design of a recuperator was performed. A sketch of the design is presented in Fig. IV-16. This does not represent an optimum design either from the standpoint of minimum surface area, minimum volume, or minimum cost. However, it does represent a design which satisfies all aspects of the design requirements. Briefly, it is a shell-and-tube heat exchanger with three shells and two passes per shell. It contains approximately 260,000 sq ft of heat transfer surface. The fin-tube configuration and spacing is displayed in Fig. IV-17.

The hot-side metal temperature should be above 275 F throughout the first two shells and for about half of the third shell, consequently, conventional fin-tubes can be used there. These conventional materials would include a mild steel since the temperature levels are relatively low. Throughout the last half of the third shell a corrosion resistant coating might have to be used on the outside of the tubes to reduce the effect of corrosion.

Capital Costs

Discussions were held with individuals knowledgeable about heat exchanger costs (Ref. IV-22) in order to arrive at an estimate of the design recuperator capital cost. This estimate is $1.96 million for both recuperators or, alternatively, $7.7/kW.

Recuperator Effectiveness Trade-off

The choice of a recuperator with an effectiveness of 0.8 for the selected CAPS design was based primarily on its associated lowered heat rate as demonstrated in the parametrics. Once the design exercise had been completed and the resulting design characteristics such as surface area, heat transfer coefficients, and cost estimated, it was then possible to parametrically vary these characteristics as a function of effectiveness to evaluate whether the selection made was appropriate.

In Fig. IV-18 both recuperator cost and heat rate savings have been related to an equipment cost (cost exclusive of indirect cost factors) and a busbar power contribution cost (based on 1560 hours-per-year operation and 18-percent annual fixed charges) as a function of recuperator effectiveness. It can be seen that the heat rate savings is approximately linear while the recuperator cost curve displays a marked change in slope between 0.6 and 0.8 recuperator effectiveness. In Fig. IV-19 the difference between the heat rate savings and recuperator cost has been plotted. The net savings peaks at around an effectiveness of 0.7 with a value of $7/kW. The selected design at 0.8 effectiveness displays a net savings of about $6/kW. Thus the design selected is not optimum, however, it is close. Another iteration of the design procedure could have been attempted at 0.7 recuperator effectiveness, however, such an iteration was considered to be beyond the scope of this preliminary study.

HYDRAULIC FACILITIES

The CAPS concept investigated in this study is hydraulically compensated. This implies that surface water must be available to provide the compensation. In this section the surface reservoir and cooling system associated with the reservoir are discussed. In addition, an in-depth analysis of the champagne effort is presented.

Surface Reservoir and Cooling System

The hydraulic facilities required for a compressed air storage installation include a compensating water system and a cooling water system. It is possible that a dry-tower cooling system could be used, in which case the hydraulic facilities would be confined to the compensating water system. On the other hand, it is possible to utilize a cooling pond system as the compensating reservoir, thus combining the two systems. Both possibilities are considered and evaluated. Typical arrangements for the cooling systems are shown in Fig. IV-20.

Since site location will have a distinct bearing on both the type of cooling system adopted and the economics of the hydraulic facilities, several different geographic locations were considered in the analysis. No specific site was evaluated, and topographical variation was eliminated from the analysis by considering that the reservoirs and cooling ponds would be constructed on flat terrain and require complete sealing.

In the Boston area, the estimated cost of the system for the 50-atm case, including the compensating water reservoir, varies from \$5.2/kW for 90-F final temperature to \$49.4/kW for 85-F final temperature. Baltimore and Detroit area costs were consistently higher. Annual operating costs for the systems were also evaluated.

In order to estimate the economics of various pressure ratios, compilations were prepared for compensating water requirements due to the different volumes, cooling requirements due to the various related heat loads, and related costs. When a combined compensation reservoir and cooling pond system was considered, the overall system costs were found to be relatively insensitive to changes in volume or cooling requirements. However, in separate systems such as cooling towers with a separate compensating reservoir, variations with different operating pressure ratios were evident, and these results are presented. The more expensive cooling systems were not evaluated, but a similar trend could be expected.

The effect of heat transfer from the warm stored air was analyzed for both the rock temperature distribution and water heat gain. The results showed that the heat gain by the reservoir water would not be significant.

Final choice of the hydraulic system for a particular site will depend on the environmental restraints and availability of water. For the purposes of the present study, the combination of wet mechanical draft cooling towers and a compensating reservoir was selected because it represents a reasonable compromise between cost, siting flexibility, and other constraints.

Criteria

This section presents the basic data used in determining the cooling system requirements for the reference 280-MW plant depicted in Fig. V-1. The state points for summer operation of the air storage plant in the compression mode are defined and summarized in Table IV-4. All data are based on a standard ISO* cycle and 761.8 lb/sec inlet air flow rate.

The aftercooler exit for the 50-atm case corresponds to 120 F at 48.6 atmospheres. It should be noted that Points 3 and 5 at the exit of the first and second intercoolers have temperatures of 161 F and 172.5 F, respectively. The maximum cooling water temperature, therefore would be limited by the final cooling stage to less than 120 F. These figures give a basic cooling load of 5.942×10^8 Btu/hr during the compression phase for the 50-atm cycle. The addition of motor-generator cooling would give an increase of some 1 percent in the cooling load.

The following assumptions were made and used throughout the hydraulic studies:

(a) Only closed-cycle systems were considered. This avoids the pitfalls involved with once-through cooling systems, which are site specific in cost and not applicable to a general study area. This also eliminates saltwater cooling towers.

(b) Systems were compared in terms of equivalents. In the case of the cooling ponds, the surface reservoir for hydraulic compensation of the lower reservoir pressure could be combined with the cooling pond function. The cooling tower systems must, therefore, include the cost of a separate surface reservoir for the compensation function.

(c) Costs of the large-diameter system supply and return pipes were estimated separately and excluded from comparison.

(d) Cooling water temperature rise of 15 F was used. Since the available sizing information for cooling ponds was based on a 15 F rise and could not be readily converted to other values, all systems were evaluated using a 15 F ΔT, with the cold-water temperature as a variable.

*International Standards Ogranization, 59 F and 1.7 psia ambient.

The performance of a given cooling system varies with climatic conditions. Therefore, climatological data were obtained from the National Weather Service for localities within the study area. Three cities were chosen as typical locations: Boston, Massachusetts; Baltimore, Maryland; and Detroit, Michigan.

Large utility cooling systems and most industrial cooling systems are generally designed for summer daytime climatic conditions. However, the air storage plant cycle is such that the cooling system would operate primarily in the early hours of the day (12:00 midnight to 12:00 noon). The in-circuit systems (cooling towers and canals) would, therefore, be subjected to lower ambient temperature and higher relative humidity than are generally present during daytime operations. Average summer nighttime conditions were determined and used as the basis for cooling system sizing for the in-circuit systems. Cooling ponds would not be affected since the heat would be dissipated over the full 24-hour day.

The cold-water temperature from the cooling system was taken as a variable in the study, with a minimum temperature of 80 F and a maximum of 110 F. A high cold-water temperature would increase the design heat exchange surface area required or, in operation, would raise the compressor inlet temperatures and, therefore, the power requirements of the compressors. This would lead to additional deterioration of the net plant efficiency during summer operation.

No limits were placed on makeup water availability, although this could be an important consideration in the system selection for a given site.

Compensating Water Reservoir

For a hydraulically compensated air storage scheme, it would be necessary to have a volume of water at the surface equivalent to the air storage volume required at depth. This water could be contained in a reservoir isolated specifically for this purpose, or be withdrawn from an existing body of water such as a river, lake, or ocean.

The hydraulic facilities required at the surface will vary significantly from site to site, and the sensitivity of the cost of these facilities to changes in volume requirements for any one installation could be small compared to changes reflected by site conditions.

The structural requirements at the surface for hydraulic facilities will include an intake (at all locations), spillway structure (on a watercourse), and a dam or dikes (depending on geology and topography). There are too many variables here to be specific about the effect of changes in the volume of compensating water on the overall cost of the installation. Furthermore, if the cooling requirements for the plant were to be met by cooling ponds, the compensating water would come from the same source. Surface area plays such an important part in sizing the installation that the overall size of the reservoir would be insensitive to minor volume changes in compensating water.

It is necessary, however, to consider the cost of the compensating water facilities in evaluating the alternative cooling arrangements for the plant and, for this reason, an isolated reservoir capable of containing the volume of water necessary for the various operating pressures of the plant has been developed and costed. In order to remove the site specific aspect from the cost, it has been assumed that the reservoir would be constructed by building ring dikes on a flat plot of land and that complete sealing of the reservoir would be required. An intake structure to the water shaft has been included, but no provision has been made for a spillway structure.

Variations in volume requirements could be accommodated by increasing the height of the containing dikes or by enlarging the area enclosed. It probably would be desirable to minimize fluctuations in level of the water surface, since these variations would be cumulative with the water surface variations in the lower reservoir and would result in greater pressure variations on the stored air. Variations in water surface level were, therefore, confined to a maximum of 30 feet; however, 10-ft, 20-ft and 30-ft variations were evaluated. The associated lining and dike costs for the compensating reservoir are given in Table IV-5 in terms of the required storage volume for each level variation considered. These figures indicate that the 20-ft depth of reservoir is the most economical, and this has been taken as the base case in evaluating the overall hydraulic systems associated with an air storage complex.

Cooling Systems

Three basic closed-cycle cooling systems were investigated: cooling towers, cooling ponds, and floating spray cooling canals. Each of these is discussed below.

Cooling Towers

Cooling towers are divided into four general types: dry natural draft, dry mechanical draft, wet natural draft, and wet mechanical draft.

Dry natural draft towers have been built in Europe, but primarily as experimental towers and generally on a small scale. This type of tower was not considered in this study, despite its obvious operating economies, as it is expected to have a very high first cost and no manufacturers were identified in the United States.

The dry mechanical tower is commercially available in the United States, but it is not commonly applied to utility installations. Air cooling is far less efficient than evaporative cooling with the result that dry towers are significantly larger and more expensive than wet towers. The great advantage of the dry cooling tower is that it requires virtually no makeup water once it is operating. Therefore, it is particularly suited to areas where water is available only at high cost.

The wet natural draft tower is becoming more common--there are some thirty "Hyperbolics" presently in operation in the United States, including saltwater applications. These towers are, however, generally much greater in capacity than required by a 280-MW air storage plant.

Wet mechanical towers are the most common type, ranging from small commercial air-conditioning units to condenser cooling for 1,000-MW plus nuclear plants. Two types of construction are used--wood and concrete--in addition to various flow configurations.

The wet mechanical and wet natural draft towers release some 80 percent of their cooling load through evaporation. Thus, for large installations, significant makeup water quantities are required. Fogging and icing problems frequently occur downstream when the moisture in the heated air condenses. Some weather modification has been postulated from the operation of these towers (Ref. IV-30) and aircraft studies have shown plumes rising several thousand feed before forming clouds (Ref. IV-31). Drift loss and the resulting salt depositions on vegetation are additional problems.

Wet mechanical towers are troubled by plume recirculation effects which decrease capacity. These effects are reduced by locating the tower away from plant structures to allow the best possible wind conditions.

The wet/dry cooling tower system has been developed as an alternative to the wet tower to achieve lower yearly makeup requirements and fewer days of fog formation. The wet/dry tower operates by first drawing air through the wet section where it becomes moisture laden, and then through the dry section where the heat transferred to the moist air alters the air condition to some point below saturation. As the dry section is less efficient than the wet section, a tower for a given heat load would be larger than an equivalent wet tower. No information was available from suppliers regarding wet/dry towers for the air storage plant. However, the installed and operating costs for wet/dry towers are expected to be higher than for wet mechanical towers, although the installed cost is not expected to exceed the cost of a dry mechanical tower.

Cooling Ponds

The cooling pond relies completely on natural heat and mass transfer processes to provide cooling of process water. In terms of cooling capacity per unit land area, the cooling pond has a very low efficiency compared to mechanically assisted systems. However, the primary advantages of the cooling pond are lack of mechanical equipment, other than makeup and blowdown pumps, and corresponding low operating costs.

The large land areas involved with a cooling pond generally preclude their use in the densely populated northeast for power plant cooling. Most applications are in the south and west for chemical plants and utilities. The land requirements for the air storage plant, however, do not appear to be excessive for cooling water temperatures of 90 to 95 F.

The cooling ponds considered are too shallow to exhibit stratification, although a basin is assumed for the shaft and intake location. In the northern climate, ice formation could be a problem; therefore, additional depth was included to ensure that sufficient water would always be available.

A related heat dissipation system is cooling canals. Canals are essentially idealized ponds, as they eliminate a large portion of the mixing losses of the conventional pond. Canals are also in use in this country, with one of the largest installations at the Florida Power and Light Company's Turkey Point nuclear plant. An estimate for cooling canals for the Boston area is subsequently presented, but a more detailed study would be required to accurately estimate this system.

Fog formation from cooling canals and ponds is a potential problem. Unlike cooling towers, the plume is not discharged above ground level, but tends to travel along the ground for some distance. Fogging is a particular problem with cooling towers in the fall and winter months, and it is expected to be the same for cooling ponds. As such, fogging could be a serious hazard if the plant site were adjacent to public highways (Ref. IV-32).

Spray Cooling Canal

The spray cooling canal was included in the study as an alternative to cooling ponds and towers, as it combines some of the features of towers and ponds. The system investigated involves multiple nozzle floating spray units which cool heated discharge water as it progresses down the canal to the reservoir.

The spray canal requires a large amount of power for spray creation, but this is partially offset by lower circulation pumping costs and maintenance so that overall operating costs are generally lower than for cooling towers. The land area requirements are significantly lower than for a cooling pond, and the system cost is less affected by temperature because of the mechanical enhancement of the cooling process.

Costs of Hydraulic Facilities

The large geographic distribution of potential sites and the effect of the resulting variations in climatic conditions are reflected in the wide range of costs for the cooling pond systems. The mechanical systems show less variation in cost with location and are also less affected by the choice of cold water temperature. The estimated costs for hydraulic facilities are presented in Table IV-6. Figures IV-21 through IV-24 summarize the installed costs for the various systems investigated.

The cooling pond estimates were derived from sizing data given in a report by E. L. Thackston of Vanderbilt University (Ref. IV-33). This report is the presentation of results from a computer study of equilibrium temperatures for ponds of various sizes with a continuous heat input of 6 billion Btu/hr. Approximations were made to adjust Thackston's results to predict the cooling pond area required for the air storage plant. The results obtained are expected to be slightly conservative. However, should the ponds be undersized, the effect on costs would be minimal, as the effect on the higher temperature ponds would be negligible.

For several types of cooling towers, it was not possible to produce an envelope of costs due to the difficulty in obtaining the required generalized information from suppliers. The costs of these towers are presented as points in Figs. IV-21 and IV-23.

For the cooling canal system, costs have been determined for one location only, as a measure of the economics of this system compared with cooling ponds. No optimization of canal lengths, widths, retention time, or depths has been undertaken. Representative approximations have been made for these parameters, so the relative costs of the systems are considered to be valid. It should be noted that cooling canals are competitive with ponds at the lower cold water temperatures, but at these low temperatures the mechanical systems show a decided capital cost advantage.

To complete the evaluation of the various systems, it is necessary to estimate the operating costs associated with each. These costs will vary from a low of $5,052 per year for the cooling ponds to a high of $93,202 per year for the concrete wet cooling towers for the 50-atm cycle. Details are shown in Table IV-7 and Figs. IV-25 and IV-26. Operating costs are highly dependent on weather conditions, maintenance and water quality, and the figures presented are based on minimum requirements.

Conditions assumed for estimating operating costs are:

(a) For circulating water

- Piping head loss - 5 feet
- Plant head loss - 10 feet

(b) For makeup water

- Pumping head - 50 feet

The useful life of all systems was assumed to be 30 years. However, the wood structure wet tower has a claimed useful life of only 15 to 20 years, and this fact should be recognized in any evaluation of a system which incorporates this type of tower.

Champagne Effect

The CAPS power plant under consideration in this study incorporates hydraulic compensation to maintain essentially constant storage pressure in the cavern (Fig. IV-27). During cavern charging, air would be pumped into the underground cavern displacing water from the cavern into the vertical shaft and from there into the compensating reservoir. During power generation, air would be withdrawn from the cavern and water from the compensating reservoir would flow back into the cavern. Because of the high air pressure in the cavern some of the air will be forced into solution at the air-water interface. If the normal charging/ discharging cycle was interrupted for several weeks or more, the water would become saturated. Consequently, during subsequent cavern charging, saturated water would be forced up the water shaft and air would come out of solution, forming a two-phase, champagne-like bubble-water mixture. This bubble mixture could, under certain conditions, lead to unstable loss of head and blowout of the cavern.

Description of the Champagne Effect

The air dissolved in the water would remain in solution as long as the water remains at a pressure equal to or greater than the cavern pressure and the water temperature remains constant. However, during a charging cycle, water in the cavern would be pushed up the shaft where it will be exposed to reduced hydrostatic pressure. As the water reaches a level where the hydrostatic pressure is less than the saturation pressure the air would begin coming out of solution. If a given volume of water were saturated at cavern pressure, then an incremental amount of air would be released from solution as soon as the given volume of water rose above the cavern level. This process would continue until that particular volume of water reached the surface at which point virtually all of the dissolved air would have been released. (Henry's Law states that the amount of dissolved gas is directly proportional to pressure so that for a cavern pressure of 100 atmospheres only 1 percent of the dissolved air would remain in the water at atmospheric pressure.) As the particular volume of water rises, not only would the total mass of air released from solution increase with decreasing pressure, but the volume occupied by a unit mass of that air would correspondingly increase. For cavern pressures typical of those investigated in this study, the air released from a unit volume of water at the surface would occupy a volume equal to that of the water in which it had been dissolved.

While the bubbles that would be released would tend to rise faster than the water, the net effect would be a two-phase column having a lower average density than a water column resulting in a reduced hydrostatic pressure at the cavern level. Even if charging were to be stopped, there would still be an unbalanced bouyant force tending to accelerate the water in the water shaft. The water velocity would increase until frictional forces could counteract the difference between the cavern pressure and the hydrostatic head of the two-phase column. At the same time, water leaving the cavern would increase the air volume in the cavern, thereby reducing cavern pressure. If cavern pressure were reduced far enough, the water column would decelerate, eventually the velocity would reach zero, the bubbles would disengage and rise to the surface and the resultant increase in hydrostatic head would cause the cavern air to be recompressed by a flow of water into the cavern from the surface reservoir. Should the cavern be emptied of water before the water column is stopped, the air in the cavern would follow the water up the shaft, further accelerate the remaining water, destroy the water seal, and blow out through the water shaft.

The scenario described above is called the champagne effect. At a minimum, it would cause a geyser of water above the compensating reservoir. It could also enable the air in storage to escape. The momentum of the water could cause serious damage to the water intake and any other structures in the path of the geyser. Obviously, avoidance of the champagne effect would be desirable.

Early in the present study, it was learned that an analytical model had been proposed (Ref. IV-34) which appears to describe the champagne phenomenon. In discussions with the model's developer (see Ref. IV-35), he indicated that a U-shaped water seal (see Fig. IV-27) extending below the cavern would provide a negative hydrostatic head on the water shaft and prevent blowout. Air could not come out of solution in the water seal because the pressure there would always be greater than the cavern pressure at which the air went into solution. Therefore, that portion of the water seal beneath the water shaft would always be filled with water. If the air/water interface were to drop below the cavern into the water seal, the negative hydrostatic head would develop.

The water seal must be at least deep enough to overcome the loss in head in the water shaft. According to Refs. IV-34 and IV-35, a water seal extending below the cavern to a depth approximately 10 percent greater than the floor of the cavern should be suitable. Apparently, water seals with a 10-percent depth factor have been used successfully for years in the Swedish mining industry. Consequently, this factor was incorporated in all storage schemes considered herein (e.g., see Fig. III-10).

A mathematical model of the champagne phenomenon was made to understand the conditions leading to its occurrence and to identify appropriate design steps to control it. The analysis, presented in the following sections, is patterned after that in Ref. IV-34, with some modifications and additions. It seeks to quantify the relationships between cavern volume, depth, and degree of water saturation and relate these parameters to the depth of the water seal. Specifically, the degree of solubility of air in water was examined along with the rate at which the diffusion process might be expected to proceed. Assuming that the water does become saturated, the two-phase flow system was analyzed to determine the effect of the relative (slip) velocity of the bubbles with respect to the water. This slip velocity would tend to decrease the bouyant effect by allowing the bubbles to disengage from the column and thereby increase the relative density. To promote this disengagement effect, it has been suggested that the top part of the water shaft be enlarged to accommodate the increased volumetric flow of the two-phase mixture while maintaining a constant water velocity from bottom to top of the shaft. This too was investigated using an exact integration of the steady flow equations and considering the effect of the two-phase flow on frictional forces in the shaft. Finally, to provide a feeling for the water velocity that might be achieved and the impact on the design of the various options for control of the champagne effect, a simplified dynamic analysis was conducted. The analysis shows that under certain conditions the 10-percent depth factor might not be adequate. A desirable alternative to the water seal might be to enlarge the cavern to provide a cushion volume which would allow the cavern pressure to drop below the level that would cause blowout. Another alternative might be to continuously circulate fresh water between the cavern and reservoir to prevent cavern water from becoming saturated and triggering the champagne effect. The impacts of either alternative on costs, pumping requirements, and system performance have not been investigated. It is recommended that any future project involving the use of hydrostatic compensation include provision for a more complete dynamic analysis of the two-phase system.

Solubility of Air in Water

The solubility of air in water, at equilibrium, is given by Henry's Law (Eq. IV-1)

$$X_a = \frac{P_a}{H_e} \tag{IV-1}$$

where:

X_a = mole fraction of air in water at saturation

P_a = air pressure at which water becomes saturated, atm

H_e = Henry's Law Constant, atm

Henry's Law Constant, H_e, varies slightly with temperature and pressure (Fig. IV-28) but basically the amount of air dissolved in a mole of water increases proportionally to the pressure. The rates at which oxygen and nitrogen in the air dissolve in water are different, but this effect can be neglected for the present analysis.

Static Bubble Distribution at Equilibrium

As air released in the water shaft rises to the surface, it expands due to the decreasing pressure. The volume of air which must pass through the top of the shaft for a given volume of saturated water is given by Eq. IV-2.

$$\frac{\text{Volume of Air at Top}}{\text{Volume of Water}} = \left(\frac{P_a}{(H_e^1)_a} - \frac{P_o}{(H_e^1)_o} \right) \frac{\rho_w}{\rho_o} \qquad \text{(IV-2)}$$

where:

$(H_e^1)_a$ = $H_e \cdot MW_w/MW_a$ where H_e is evaluated at P_a

$(H_e^1)_o$ = $H_e \, MW_w/MW_a$ where H_e is evaluated at P_o

MW_w = molecular weight of water

MW_a = molecular weight of air

P_o = ambient pressure at top of shaft, atm

ρ_o = air density at P_o, lb/ft^3

ρ_w = water density, lb/ft^3

Equation IV-2 is plotted in Fig. IV-29 where the air/water volume ratio is seen to increase almost linearly with pressure. For a CAPS operating pressure in the 50-80 atm range, the volume of air released at the top of the shaft would be 2-3 times the corresponding volume flow of water. The effect on the vertical shaft can be seen more clearly in Fig. IV-30 where the static air volume per unit volume of water is plotted versus pressure for initial saturation (cavern) pressures of 5, 25, 50, and 75 atm. This volume ratio corresponds to the volume of air released from a bottle of unit volume of saturated water when opened at the pressures shown on the y-axis.

The presence of a large volume of air in the shaft would cause the flow to accelerate, producing a geyser. The behavior in a real system would differ from the static results discussed above because of dynamic effects, but the basic pattern would remain. The results in Fig. IV-30 show that the major volume expansion of bubbles from saturated water takes place in the last 5-10 atm (150-350 ft of static water head). This suggests that the geyser could be controlled by flaring the top of the vertical shaft to maintain low velocities. However, as is shown in a subsequent section, flaring the shaft affects the pressure drop which supresses bubble formation and growth, thus resulting in a worsening of the head loss.

Nonequilibrium Effects

The preceding discussion assumes that the water in the cavern was saturated. In a real CAPS operating through a suitable daily or weekly duty cycle, water in the cavern would frequently be replaced by fresh water from the compensating reservoir. The degree to which the water in the cavern would become saturated would depend on the diffusion rate of air through water and the turbulent mixing of water in the cavern.

A simplified analysis for the diffusion of air through stagnant water is presented in Appendix G. That analysis shows that diffusion is extremely slow. For the example treated in the appendix, the water would be only about 10 percent saturated after a period of one month. During the generation cycle, the flows of air from the cavern and of water into the cavern can be expected to produce turbulent motions which would expedite the diffusion process. In Ref. IV-34, an empirical formula based on air transfer in natural streams was used to deduce that the water would become 3-4 percent saturated during the "loading phase" (generation cycle) and that only after 15 days or more would the water approach saturation. Over the long term, the effects of natural convection, water heating due to the hot compressed air (about 120°F), water turbulence during inflow and outflow, and water stratification would all affect the amount of air dissolved in the water. Developing an adequate model for all these effects was beyond the scope of the present program.

For the purposes of the present study, it is sufficient to recognize that the saturation process might be quite slow. If the CAPS were used on a daily or weekly cycle, the amount of air dissolved in the water would be minimal and the champagne effect would not become severe. However, the CAPS design must allow for the contingency where the water becomes saturated, either through prolonged disuse or operation under highly stratified conditions. Consequently, the subsequent analyses of two-phase flow in the hydraulic system assume saturated water in the cavern.

General Description of Pipe Flow with Bubbles

Water flow with bubbles in a vertical pipe can exist with several quite different flow patterns, depending on the air/water proportions and the velocities involved. As the air/water flow ratio increases, the flow regime changes from simple water flow, to flow with bubbles, to slug flow, to froth, to annular, to mist, and in the extreme, air (see Fig. IV-31). The volume of air at a given point in the pipe is a function of two mechanisms, both pressure dependent. The first is from the air coming out of solution at a given local pressure (or equivalently, depth) according to Henry's Law; the second is air which came out of solution at a higher pressure (lower depth) and which has risen to the given height - expanding via the perfect gas law. The combination causes the exaggerated hyperbolic shape of Fig. IV-30. The problem is complicated by the fact that the restricted area of the pipe will cause the overall flow to speed up as the bubbles expand. These interacting effects make the problem too complex for a simple flow description.

Understanding the characteristics of two-phase flow is necessary for good design of the hydraulic system. A large volume of literature on two-phase flow exists because of chemical process, heat transfer, and oil industry problems. An extensive bibliography is given in Ref. IV-36. The brief discussion of two-phase flow in vertical pipes presented herein is based on Refs. IV-37 and IV-38.

Empirical correlations for two-phase flow have been developed through extensive laboratory experiments with short lengths (30 ft or less) of small-diameter pipe (from 1/2 to 2 1/2 in.). By plotting the "modified superficial gas velocity," XV_{SG}, versus the "modified superficial liquid velocity," YV_{SL}, as in Fig. IV-32, the appropriate flow regime can be identified. The following definitions apply to Fig. IV-32.

$$Y = \left(\frac{\rho_{wA} \quad \sigma_{wA}}{\rho_w \quad \sigma_w}\right)^{1/4} \qquad \text{(IV-3)}$$

$$X = \left(\frac{\rho_a}{\rho_{aA}}\right)^{1/3} Y \qquad \text{(IV-4)}$$

$$V_{SG} = \frac{Q_g}{A_s} \qquad \text{(IV-5)}$$

$$V_{SL} = \frac{Q_w}{A_s} \qquad \text{(IV-6)}$$

where:

ρ_w, ρ_a, σ_w = densities of water and air and surface tension of water/air at the flow conditions

ρ_{wA}, ρ_{aA}, σ_{wA} = densities of water and air and surface tension of water/air at 60 F and 1 atm

Q_g, Q_w = volume flows of gas and water

A_s = cross-sectional area of shaft.

For typical CAPS conditions (temperature = 60 to 120 F, pressure = 40 to 75 atm, Q_w = 150 to 250 ft^3/sec, Q_g/Q_w = 0 to 3 [from Fig. IV-29], and pipe diameter = 12 ft) the CAPS flow regime can be identified from Fig. IV-32. It is seen that the flow will be in the bubble and slug regimes. The slugs (bullet-shaped gas bubbles in Fig. IV-31) form and break down and then reform again, with several slugs existing at any one time and bubbles in between. Using the curves in Figs. IV-29, IV-30 and IV-32, plus a static expansion and mass balance, indicates that the transition from bubble to slug flow occurs near the top of the vertical shaft (upper 150-200 ft) where the pressure drops below about 5 atm. Thus, slug flow would appear to dominate the flow near the top where most of the bubble expansion occurs.

The transition from bubble to slug flow is caused by a differential velocity (or slip velocity, U_{slip}) with which bubbles rise relative to the surrounding water. The slip velocity varies according to the bubble size and is different for slugs than the bubbles. Various empirical relationships, together with estimated CAPS operating conditions are summarized in Table IV-8. As the air is released from solution, it progresses from small individual spherical bubbles to larger numbers of bubbles to coalesced bubbles which take the shape of spherical caps, or umbrellas of air. The slip velocity increases through this progression. The final slug flow stage is reached when the spherical caps are affected by the shaft wall, which occurs when the cap diameter is about 75 percent of the shaft diameter and the air/water volume ratio is about 0.56 $(0.75)^2$. The larger caps rise faster than the slugs. The caps catch up with and become part of the slugs, until some stable slug size is reached. A series of slugs, separated by bubbly water, then form in the pipe, as shown in Fig. IV-31. It should be repeated, however, that all of the flow regime correlations are based on observations of flow in small-diameter pipes. Fundamental questions exist regarding the stability of slugs as large as 8 to 10 ft in diameter. It is possible that a complete transition to slug flow would not occur in a CAPS installation.

The friction loss in the shaft is affected by the presence of bubbles. The Ros correlation for frictional pressure drop in two-phase flow (from Ref. IV-37) is given by:

$$\Delta P_f = 4f\left(\frac{L}{D}\right) \rho_w \left(\frac{V_{SL}^2}{2g}\right) \qquad \text{(IV-7)}$$

where

$$f = f_1\left(\frac{f_2}{f_3}\right)\left(1 + \frac{V_{SG}}{V_{SL}}\right) . \qquad \text{(IV-8)}$$

In the Reynolds number range of interest, the friction factor f_1 is identical to the conventional single-phase friction factor, f_w, for simple water flow. The factors f_2 and f_3 are corrections to account for holdup. The factor f_2 is given in Fig. IV-33 where the dimensionless diameter number, N_D, is:

$$N_D = D\left(\frac{\rho_w g}{\sigma_w}\right)^{\frac{1}{2}} . \qquad \text{(IV-9)}$$

The factor f_3 is a second-order correction given by:

$$f_3 = 1 + f_w\left(\frac{V_{SG}}{50\ V_{SL}}\right)^{\frac{1}{2}} . \qquad \text{(IV-10)}$$

Because the term V_{SL} used in Eq. IV-7 is the same as the bottom velocity, the ratio of frictional pressure drops with and without bubbles corresponds to the ratio of the two-phase friction factor to the friction factor for simple water flow which is given by:

$$\frac{f}{f_w} = \frac{f_2}{f_3}\left(1 + \frac{V_{SG}}{V_{SL}}\right) . \qquad \text{(IV-11)}$$

For the CAPS conditions given previously which could lead to slug flow at the top of the shaft, and using $f_w = 0.005$, Eqs. IV-9 through V-11 give a value of approximately 2.6 for the ratio of friction factors. That is, the friction factor at the top of the shaft where slug flow dominates could be 2.6 times the equivalent friction factor for all water flow. Consequently, a small but significant frictional pressure drop could occur in the top several hundred feet of shaft.

Steady-Flow Analysis of Vertical Shaft

The general equations discussed in the previous sections are applicable only to short sections of vertical pipe where velocities and other flow parameters do not change significantly. In order to analyze the two-phase effects over a long vertical shaft, it is necessary to express the governing relationships in differential form and integrate them over the length of the shaft. Such an analysis was developed and is described in Appendix H. The equations presented therein describe the combined flow of air and water, including bubbles coming out of solution and expanding as they rise. The scope of this study permitted only the steady flow versions of the equations to be solved in detail. Solving these steady flow equations for the range of conditions expected in CAPS provides a rough estimate of the U-bend depth necessary for the water seal to prevent cavern blowout. A dynamic analysis of the hydraulic system, including friction, would provide a less conservative, but more realistic, estimate for the required U-bend depth.

Two series of computer runs were made with the steady flow model in Appendix H to simulate constant diameter and flared vertical shafts. A range of anticipated operating conditions was considered. Two cavern pressures (50 and 66.3 atm) and corresponding hydrostatic depths were run with an average water velocity at the bottom of the shaft of 1.64 ft/sec. This velocity corresponds approximately to the flow required for a single-unit CAPS plant (187.5 cfs). In order to estimate the characteristics of the flow after it had accelerated, cases were also run for 16.4 ft/sec. The effect of differing slip velocities was estimated by running cases for 0 and 6.87 ft/sec slip velocity corresponding to very small bubbles and slug flow, respectively. The slip velocity was taken as constant because of the uncertainty as to local bubble size when the transition from one type of flow to another would occur and concern as to whether the empirical formulas based on experiments in small diameter pipes would directly apply to shafts up to 12 ft in diameter. The mean water temperature was assumed to be 125 F, corresponding to the air temperature in the cavern after charging.

Constant Diameter Shaft

The constant diameter case was based on an assumed shaft diameter of 12 feet. Results for this case are contained in Figs. IV-34 through IV-36 and Table IV-9. The ratio of the volume of air to the volume of water at a given cross section in the vertical shaft is shown in Fig. IV-34 for a cavern storage pressure of 66.3 atm. The variation of the air/water volume ratio with slip velocity is shown in the figure with the zero slip case at the right side of the shaded bands. Note that zero slip results in the worst case with the highest concentration of bubbles in the shaft because the bubbles cannot escape from the shaft. Note, also, that for zero slip the air/water volume ratio is almost independent of the water velocity at the bottom of the shaft. For high slip, there is a tendency for the bubbles to escape from the shaft. Note, also, that for zero slip the air/water volume ratio is almost independent of the water velocity at the bottom of the shaft. For high slip, there is a tendency for the bubbles to escape from the shaft and the air/water volume ratio is less. In this case, there is a strong dependency between the air/water volume ratio and the bottom velocity. The band in Fig. IV-34 for the high bottom velocity is considerably narrower than the band for low bottom velocity, indicating that the effect of slip is diminished as the water velocity is increased.

Curves for the velocity of the water versus depth are given in Fig. IV-35 for the previous example. The acceleration of water becomes most pronounced near the top of the shaft, corresponding to the increase in air/water volume ratio illustrated in Fig. IV-34. The ratio of the maximum water velocity at the top of the shaft to the water velocity at the bottom is summarized in Table IV-9 for several combinations of cavern pressure, bottom velocity, and slip velocity. For those cases where the bottom velocity does not accelerate above its design value (1.64 ft/sec) and the slip is high (6.87 ft/sec) the top/bottom velocity ratio is estimated to be only 1.5 to 1.7. However, for those cases where slip is small or where the bottom velocity accelerates to a high value (16.4 ft/sec), the velocity ratio could be as high as 2.6 to 3.4. Consideration must be given to the ability of the water grates and other intake structures to withstand these velocities.

Table IV-9 also gives the density ratio for the vertical shaft (ratio of average density of the two-phase mixture to the density of water). The reduction in density for the worst cases is on the order of 8 to 10 percent. Since the weight of the mixture in the vertical shaft is what pressurizes the cavern, a decrease in density must cause a corresponding decrease in cavern pressure. The variation of pressure in the shaft with depth is shown in Fig. IV-36 for the 66.3 atm cavern. The top curve is the hydrostatic pressure line for which the cavern is designed. (Note that the hydrostatic pressure curve crosses the cavern floor depth at 66.3 atm.) The pressure reduction due to the bubble-water flow is shown by the lower curves in the figure. The depth below the cavern flow at which the pressure would equal 66.3 atm ranges from approximately 60 ft to 270 ft. Since the water presumably became saturated at 66.3 atm pressure, the bubbles would come out of solution at all higher levels where the pressure is less than 66.3.

Flared Shaft

Since the velocity acceleration and increase in air/water ratio are concentrated in the uppermost part of the constant diameter shaft (Figs. IV-34 and IV-35), the suggestion has been made (see Ref. IV-35) that flaring the shaft at the top might assist in the dissolution of bubbles and avoid the velocity speedup. This problem was investigated as part of the two-phase flow analyses presented in Appendix H. The steady-flow equations derived in the appendix give the variation in shaft diameter which would result in constant water velocity from the cavern floor to the surface.

Table IV-10 lists the results of a series of computer runs for the flared-shaft model. Both the design velocity and slip velocity must be specified to determine the design. Two extremes in design velocity were selected to get a feel for whether the shaft should be designed for high or low initial velocity at the bottom. For the four combinations of design and slip, velocity selected, Table IV-10 lists the top/bottom area ratio and the volume ratio of the flared shaft relative to a straight shaft. The latter indicates the amount of additional excavation required for the flared design. For each shaft design, two values of bottom velocity (1.64 and 16.4 ft/sec) were selected. The resulting estimates for water velocity ratio and density ratio are also given in Table IV-10. It may be seen that the zero slip velocity design results in no velocity speedup for either bottom velocity. The excavated shaft volume, however, must be 20 percent greater than for a straight shaft due to the flare. Also, flaring the shaft leads to a greater density reduction (8-15 percent) and correspondingly greater loss of pressure in the cavern. Figures IV-37 through IV-39 illustrate the variations in shaft radius, air/water volume ratio, and pressure distribution, respectively, with depth. From Fig. IV-37 it may be seen that the flare is mainly in the top 500 ft of shaft. Also, the flare is relatively insensitive to design velocity for zero slip. This result was expected and is discussed in Appendix H.

Momentum Balance for U-Bend Depth

The preceding analyses do not give the U-bend depth necessary to prevent blowout. The reduction in pressure due to the flow in the vertical shaft (Figs. IV-36 and IV-39) leads to a depth below the cavern floor where the pressure would equal the cavern pressure. It is at this point that bubbles would first start to come out of solution and the flow would begin to accelerate. The U-bend must at least reach this depth. Otherwise, the pressure on the cavern-side of the U-bend would exceed the pressure on the vertical shaft side and the cavern would blow out. The minimum depth of the U-bend corresponding to the depth where bubbles would first appear is given in Table IV-11 and IV-12 for constant diameter and flared shafts, respectively.

In addition to the U-bend depth needed to equalize the bubble-caused loss of pressure, an additional depth is needed to absorb the inertia of the fast-moving water. If the water in the cavern side of the U-bend has descended below the cavern floor to the depth where the pressure in the vertical shaft part of the U-bend equals the cavern pressure (L_2 in Fig. IV-40a), the additional depth can be approximated by neglecting friction and equating the potential energy of the water in the additional U-bend depth (h in Fig. IV-40b) to the kinetic energy of the moving water in the shaft and U-bend. Mathematically,

$$\frac{1}{2} m_1 U^2 = \frac{1}{2} m_2 g h \qquad \text{(IV-12)}$$

where m_1 = mass of water and air in the vertical shaft and U-bend
m_2 = mass of water in the additional U-bend depth, h
U = water velocity at bottom of shaft (in U-bend).

The equation for the additional U-bend depth, h, is given by

$$h = \frac{U^2}{g}\left[1 + \sqrt{1 + \frac{g}{U^2}\left(\frac{\rho_{ave}}{\rho_w}\frac{V_s}{A_s} + L_3\right)}\right] \qquad \text{(IV-13)}$$

where $\frac{\rho_{ave}}{\rho_w}$ = ratio of average density of mixture to density of water
V_s = volume of shaft corresponding to L_1
A_s = cross sectional area of shaft at bottom
L_3 = horizontal connecting pipe length (taken at 50 ft).

For a straight shaft, the ratio V_s/A_s reduces to L_1, the total depth down to the point where bubbles start to form. By neglecting friction in the above analysis, a conservative estimate for h is obtained. A more complex model including friction should include the variation in water velocity with time. The analysis would be complicated by the fact that average density in the shaft depends strongly on velocity.

Estimates for the additional U-bend depth due to fluid inertia in straight shafts are summarized in Table IV-11 for two assumed fluid velocities. The total U-bend depth is the sum of the depth at which bubbles first appear and the additional depth due to the fluid inertia. The total U-bend depth, expressed in feet and as a percent of depth to the cavern flow, is also given in the table. For zero slip velocity, the total U-bend depth ranges from about 12 to 16 percent of cavern depth. For high slip, the U-bend depth would range from about 3 to 13 percent. The percent U-bend depth is essentially independent of cavern pressure (and depth). This results from Henry's Law, which states that the dissolved air is proportional to pressure, and the effects of air dissolution being concentrated in the uppermost part of the shaft.

Corresponding estimates for the U-bend depth for a flared shaft are summarized in Table IV-12. The required U-bend depth for zero slip is slightly higher than for the straight shaft and ranges from about 13 to 17 percent.

The U-bend depth estimates in Tables IV-11 and IV-12 are for two assumed values of the water velocity in the shaft. The next section presents a simplified method for estimating the water velocities that should be used to estimate U-bend depth.

Simplified Dynamic Analysis of Storage System

To obtain an understanding of the significance of the data previously presented it is necessary to estimate the magnitude of key physical parameters that might be attained during operation of a CAPS plant, especially the maximum water velocity at the bottom of the shaft. To do this within the scope of this preliminary feasibility evaluation, a simplified dynamic analysis was made of a fixed-configuration, hydraulically-compensated compressed air power system. This analysis is a quasi steady-state analysis to which dynamic effects were applied in a piecemeal manner. To avoid the complexities associated with this nonlinear problem and to simply obtain an order of magnitude appreciation for the system dynamics, an idealized solution was first obtained assuming instantaneous application of the bouyant force on an inertialess fluid in the water shaft. This idealized solution was then used as a basis to estimate the effects of shaft charging time (time to replace water in the shaft with water from the cavern) and fluid inertia on system response.

The results of this analysis suggest that peak water velocities several times larger than assumed in the previous section could be attained under certain circumstances. This implies that the required depth of the U-bend could be substantially deeper than previously indicated unless the cavern volume was enlarged to compensate for some of the bouyant head.

Discussion of Assumptions

Key assumptions made for the analysis include zero slip and fully-saturated water in the cavern. In addition, the flow in the vertical shaft was assumed to be like single-phase water. Specific values assumed for the system parameters are summarized in Table IV-13.

The assumption regarding full saturation of the water might seem overly pessimistic, but it represents a "worst-case" approximation. Some degree of saturation is bound to occur, and long idle periods are not completely unlikely. The assumption of zero slip, however, appears to be quite valid in light of the results presented in Table IV-9. For the case in Column 1 of Table IV-9, with low water velocity at the bottom, the maximum slip velocity is some four times greater than the water velocity. The bouyant head, which corresponds to the density loss, would be about .02 times the depth of the cavern or some 45 ft of water. This magnitude of head would be sufficient to accelerate the water column to a velocity of about 26 ft/sec, at which point the difference in bouyant head between the zero slip and maximum slip cases would be quite small (compare Columns 3 and 4 in Table IV-9). Velocities as high as 26 ft/sec could be attained, as is shown in the subsequent analysis.

Friction was approximated by assuming single-phase water flow throughout the column. As previously shown, the two-phase friction factor is related (see Eq. IV-11) to the volumetric ratio of air to water (V_{SG}/V_{SL}). With bubbles present, the friction factor at the surface could be as much as 2.6 times that for water flow only. However, as illustrated in Fig. IV-34, the volume ratio is significantly above unity only near the surface. A rough numerical integration over the shaft shows that the average effect of the air volume is approximately 10 percent of its effect at the surface. The resulting overall two-phase pressure drop would be approximately 16 percent greater than that calculated for water only. This would reduce the 26 ft/sec water velocity previously calculated to about 24 ft/sec, which is well within the desired accuracy of this analysis.

Idealized Analysis

The situation most susceptible to the champagne effect is when the cavern is almost full of air and additional air is pumped in to fill the cavern. As air is pumped into the cavern, water would be forced through the shaft to the surface reservoir. The frictional lead loss due to the water flow in the vertical shaft is given by Eq. IV-14.

$$H_f = \left(\frac{f}{2g}\frac{\ell}{D}\right) U^2 \qquad \text{(IV-14)}$$

where:

H_f = frictional head loss, ft

f = friction factor

ℓ = total shaft length (including U-bend), ft

D = shaft diameter, ft

U = water velocity, ft/sec.

Using the system parameters given in Table IV-13 the term within the parenthesis in Eq. IV-14 becomes equal to 0.068. For the idealized case with instantaneous application of the bouyant force to the water shaft the frictional head loss, H_f, would be equal to the bouyant head, H_b (224 ft), caused by the air bubbles. Also, since the fluid is assumed initially to be inertialess, the water in the shaft would instantaneously reach a velocity of 57 ft/sec as determined from Eq. IV-14.

However, as the water leaves the cavern the air volume increases (assuming no further air inflow) and the pressure drops, thereby reducing the effect of the bouyant head. Note that the product of cavern pressure times air volume in the cavern is constant, and the rate of change of air volume corresponds to the water flow rate from the cavern. Since the magnitude of the pressure and volume change will be on the order of 10 percent (which is sufficient to counteract the bouyant head), the relationship between pressure and volume can be linearized leading to the following equation for the rate of change of cavern pressure, or head, available to overcome friction:

$$\frac{dH}{dt} = -\left(\frac{\pi D^2 H_c}{4V}\right) U \qquad \text{(IV-15)}$$

where:

H = cavern pressure or head, ft

H_c = cavern depth, ft

V = cavern volume, ft^3.

Using the values from Table IV-13, the term within the parenthesis becomes 0.0188.

Equations IV-14 and IV-15 can be combined to yield:

$$\frac{dU}{dt} = -\left(\frac{\pi D^2 H_b}{V}\right) / 2\left(\frac{f\ell}{2gD}\right) = -0.138 \frac{ft}{sec^2} \qquad \text{(IV-16)}$$

Thus, following the initial increase in velocity to 57 ft/sec, the idealized system shows a linear decrease in velocity with the velocity falling to zero after 57/0.138 = 413 sec. At that time the cavern air volume will have increased by approximately 10 percent and the pressure will have decreased by a like amount. This is depicted graphically by the dashed line in Fig. IV-41.

Effects of Inertia and Shaft Charging

Two factors will affect the initial velocity achieved by the water column. These are: a) the reduction in pressure due to the amount of saturated water from the cavern needed to charge the shaft with bubbles and b) the inertia of the water which limits its acceleration. Both of these factors lead to a reduction in peak velocity. In the case of shaft charging, the full bouyant head could not be developed before the entire shaft water has been replaced with saturated water from the cavern. The shaft volume amounts to 11,900 yd^3 which is about 2.4 percent of the cavern volume. This increase in air volume would decrease cavern pressure by 2.4 percent or 53 ft of head. The remaining pressure imbalance (224-53 = 171 ft) would result in a velocity (from Eq. IV-14) of 50 ft/sec.

The pressure imbalance must, however, first accelerate the fluid. By applying Newton's Law, the initial acceleration would be given by

$$\frac{dU}{dt} = \frac{g}{\ell} H \quad . \tag{IV-17}$$

Since H is approximately equal to the cavern depth, H_c, the acceleration would be almost 2 ft/sec^2. For an average velocity of 25 ft/sec the acceleration would take about 25 sec and the reduction in cavern pressure (from Eq. IV-15) would be about 12 ft of head. The resulting net bouyant head would be 159 ft and the corresponding maximum velocity (from Eq. IV-14) would be about 48 ft/sec. Thus, even with a large 12-ft diameter shaft and the associated cavern pressure drop due to charging the shaft, the resultant maximum velocity is still within 20 percent (48 ft/sec vs 57 ft/sec) of that calculated neglecting charging and inertia effects.

Flow Transient

Once the peak velocity has been reached, it is necessary to estimate the ability of the fluid column to follow the ramp change in velocity caused by the decreasing cavern pressure. At any point, the linearized response of the velocity to changes in pressure has the form of a simple lag. The lag time constant is inversely proportional to inertia and the functional damping. Since inertia is essentially constant and friction varies with velocity squared, the frictional damping (dH/dU) varies linearly with velocity. It can be numerically shown that τ is equal to 651/U sec.

Recognizing that the response of a simple lag to a ramp input is another ramp having the same rate of change but delayed in time by an amount equal to the time constant τ, the actual velocity curve will diverge from the idealized curve as velocity decreases and τ increases. The resulting estimate is illustrated in Fig. IV-41 as a solid line.

Hydraulic Oscillations

At low velocities, the shaft/U-bend/cavern system becomes quite underdamped and oscillations could occur. The point at which the idealized velocity reaches zero corresponds to a balance between the bouyant head and the decrease in cavern pressure, e.g., there is no net accelerating or decelerative force acting on the water column, but since the column is still moving it has inertia; therefore, the velocity at that point (9.5 ft/sec) was taken as the initial velocity for a frictionless oscillation, the water column being the mass and the cavern pressure providing the spring effect. The resultant response provides a reasonably good estimate for the additional cavern volume (in excess of that required to simply balance the bouyant head) needed to bring the velocity to zero.

By combining Eqs. IV-15 and IV-17 the equation of motion becomes:

$$\frac{d^2U}{dt^2} = - \frac{g}{\ell} \frac{\pi D^2 Hc}{4V} U \quad . \qquad \text{(IV-18)}$$

After substituting the numerical values from Table IV-13 the term in parentheses becomes equal to 0.000212. For an initial velocity of 9.5 ft/sec and a deceleration rate of -0.08 ft/sec^2, the result is sinusoidal leading to zero velocity after another 75 sec (approximately 488 sec total elapsed time). Superimposing this sinusoidal variation on the ramp response produces an inflection as in Fig. IV-41. The actual response might go to zero without such an inflection.

The integral of the airflow over this 75-sec period shows a volume change of approximately 3000 yd^3 or 0.6 percent of the cavern volume. Thus, when velocity reaches zero, cavern pressure will have been reduced by a slightly larger amount than the initial bouyant head, and the net back pressure will tend to reverse the flow direction.

As the outflow rate approaches zero, the bubbles will tend to disengage thus raising the average density of the water column. Since cavern air pressure has been reduced by the magnitude of the bouyant head, an imbalance head approximately equal to the bouyant head but in opposite direction will exist. The flow into the cavern will be accelerated with any bubbles that remain in the shaft being swept back into the cavern. The resultant inflow transient should be quite similar to the outflow transient and has been sketched in Fig. IV-41 as the mirror image. As velocity again reaches zero, there will be another overshoot causing the flow to again reverse direction. However, since the water in the cavern will have been diluted with fresh water from the surface reservoir, the resulting bouyant head caused by further dissolution of air in the shaft will be much less than the initial bouyant head. The flow will continue to oscillate and decay through an underdamped sinusoid.

Significance of Dynamic Results

The preceding analysis indicates that an alternative to a deep U-bend would be to enlarge the cavern by about 10.6 percent. That is, if the amount of water remaining in the cavern upon completion of air injection were greater than or equal to 10.6 percent, the cavern could not blow out because the outflow velocity will have gone to zero and the cavern air could not reach the base of the shaft. If, however, the amount of water in the cavern were allowed to fall below 10.6 percent before stopping air injection, the outflow velocity would still be positive when the water level reaches the cavern floor. In this case a U-bend would be required to prevent blowout.

The analysis was extended to estimate the net bouyant head and water velocity entering the U-bend in terms of the water volume in the cavern at the start of the transient. The results are shown in Fig. IV-42. The minimum depth of the U-bend to prevent blowout would correspond to the net bouyant head.

An additional depth would be required, however, to overcome the momentum represented by the velocity curve in Fig. IV-42. A simple momentum balance similar to that previously used to estimate U-bend depth was used to estimate the maximum incremental depth required to overcome inertia. The result is shown in Fig. IV-43. The maximum required U-bend depth is the sum of the net bouyant head and the incremental head needed to overcome inertia. For low values of water remaining in the cavern (below about 3 percent) the velocity calculation is rather indefinite due to the effects of inertia and the time required to charge the shaft with saturated water. The velocities assumed in the previous two-phase steady flow calculation, 1.64 and 16.4 ft/sec, are seen to correspond approximately 9.6 to 10.6 percent respectively, of water remaining in the cavern at the start of the transient.

Operational and Design Considerations for the Champagne Effect

Several operational and design considerations have been suggested for helping to control the champagne effect. A U-bend seems to be desirable to prevent cavern blowout but the necessary depth of the U-bend could be excessive unless the cavern volume was oversized to take advantage of the pressure drop in the cavern due to a perfect gas law expansion. Further trade-off studies would be required to identify the optimum combination of U-bend depth and cavern oversizing.

As previously mentioned, a lengthy time (several weeks or more) is required for the water in the cavern to become saturated. Unless the water is saturated, the champagne effect will not be serious. This suggests that the problem could be avoided by intentionally replacing saturated water with fresh water from the surface This would occur naturally during normal daily operation of the system. But during extended outages, some supplemental pumping might be necessary.

In summary three methods have been identified which will help control the champagne effect: a U-bend, oversizing the cavern, and purging the saturated water. The specific method, or combination thereof, chosen for a particular installation will depend on the relative economics.

ENVIRONMENTAL CONSIDERATIONS

This section addresses the environmental interrelationships of a hydraulically-compensated compressed air power system. The commentary focuses on four topics: air emissions, noise, groundwater, and aesthetics.

A prudently designed CAPS facility should have little long-term adverse impact. It appears that the CAPS combustion systems can be expected to meet, or produce pollutant emissions less than, anticipated air emission statutory limits. If water-cooled intercoolers were to be adopted, impacts include potentially severe local fogging and the effects of thermal additions to a viewing body of water. If, as recommended, air-cooled intercoolers were to be adopted, impacts would be highly localized and associated mainly with the construction or mere presence of the facility. Once operational, the plant should not significantly affect ambient aesthetic, noise, water or air quality.

At the present time it is uncertain how state regulatory authorities would react to such a facility, however, it appears that licensing an air-cooled CAPS plant would be simpler than licensing a corresponding water-cooled facility. Quite possibly the discharge from the latter would be subject to either effluent standards, thermal standards, or both, whereas the discharge from the former would probably be considered insignificant.

Air Emissions

Air pollutants emitted from CAPS are basically the same as those from distillate-fired gas turbines. In general, CAPS will produce very low sulfur emissions due to the clean fuel burned. Particulates, while small in size, will also be minimal. The principal areas of concern will be the production of nitrogen oxides and carbon monoxide. Current burner designs produce little CO; however, many approaches to NO_x reduction may result in an increase in CO production. Also, it is generally recognized that increases in either combustion temperature or pressure will be accompanied by an increase in NO_x emissions. Therefore, the work which has been done has been directed primarily toward an evaluation of the CAPS nitrogen oxide formation characteristics. In other areas, sufficient work has been done to assure the ability to meet emission requirements. The combustion evaluations assume the use of FT50 hardware in a cycle as described in Fig. IV-1.

Environmental Regulations

Before an evaluation of the system can be made, it is necessary to review the regulations that are currently in force on both Federal and local levels as well as those suggested standards that have not as yet beem imposed. Some of these are summarized in Tables IV-14 through IV-16.

At this time, the only Federal EPA regulations on power system emissions are those promulgated for new stationary sources. These regulations (Table IV-14) apply to fossil-fueled steam generators of more than 250 million Btu/hr input (about 25 MW). The standards apply primarily to new plants, but they also apply to existing plants which are modified in such a way as to increase or alter the nature of their emissions. The implementation date was July 1, 1975.

Agencies other than the Federal EPA have set regulations which, in general, are significantly more severe. Perhaps the most restrictive are those of the San Diego County Air Pollution Control Board (see Table IV-15). These rules, which went into effect July 1, 1971, apply to all stationary fuel-burning equipment except units of less than 50 million Btu/hr burning natural gas, L-P gas, or a combination thereof. As can be seen in Table IV-15, the regulations are based upon emission in ppm (vol) in the effluent gas at 3 percent excess O_2, rather than mass emissions per unit of fuel input. The latter is a more convenient standard to work with, since the amount of excess air in the stack varies widely with advanced energy conversion systems and fuel form.

In November of 1973, the Federal EPA circulated for comment a preliminary draft of suggested regulations for gas turbine emissions, shown in Table IV-16. The suggested allowable SO_2 emissions are lower than the emissions from an equivalent fuel burning steam station, while the NO_x emissions are essentially the same. A standard for CO has been added since tests have indicated that NO_x control schemes might increase CO emissions. However, this might be unnecessary since engine manufacturers will try to minimize CO in order to improve performance.

Nitrogen Oxides

While there are at present no Federal standards regulating stationary gas turbine engines, much work has been done by engine manufacturers and fuel suppliers to achieve an understanding of the NO_x formation mechanism and to develop methods for its control. Oxides of nitrogen, commonly lumped together as NO_x, are receiving increasing attention as air pollutants. The various oxides are easily interconverted in the atmosphere, their ratio depending on the action of sunlight, oxygen, and other oxidizing or reducing agents present. The major contributors of these pollutants are the hot reaction zones of all air-breathing combustion engines. The nitrogen oxides are formed primarily as NO (nitric oxide), although small quantities of NO_2 (nitrogen dioxide) and N_2O (nitrous oxide) may also be formed.

Nitrogen Oxide Formation Mechanism

Two mechanisms are known to contribute to the formation of nitrogen oxides in combustion systems. The most important mechanism for gas turbines, and other systems which burn relatively clean fuels, is referred to as the thermal or hot air mechanism. In this mechanism, nitrogen and oxygen from the atmosphere react in the hot combustion zone to form nitric oxide. The second mechanism is important when relatively dirty fuels such as coal and residual fuel oil are burned. Most dirty fuels contain small but significant quantities of organic nitrogen compounds. Because nitrogen-carbon and nitrogen-hydrogen bond energies are so much lower than that for molecular nitrogen, much of the fuel nitrogen becomes oxidized during combustion. The formation rate of nitric oxide from fuel nitrogen is very rapid, occurring on a time scale comparable to that of the hydrocarbon combustion reactions. This mechanism is strictly fuel dependent and proceeds at lower temperatures than needed for the thermal mechanism. Fortunately, fuel nitrogen does not appear to be a problem when clean distillate fuel is used. Consequently, the remainder of this discussion is limited to the thermal mechanism.

Complex computer simulations have been developed by United Technologies, and others, that model the combustor internal flow-field, the combustion reactions, and the thermal NO_x kinetics (Refs. IV-39 through IV-44). The simplified NO_x kinetic predictive techniques are generally based on the fact that the NO formation rate is very slow relative to the hydrocarbon combustion reaction rate so that the two can be decoupled in predicting NO formation; i.e., the combustion reactions can be assumed to be at equilibrium in estimating NO formation rates. This is illustrated in Fig. IV-44 which is the result of a kinetic model considering both combustion and NO kinetics (Ref. IV-45). For that system the combustion reaction is essentially complete in 60 microseconds at which time the NO concentration is several orders of magnitude less than its final or equilibrium value.

The thermal mechanism for formation of NO from nitrogen and oxygen was originally proposed by Zeldovich (Ref. IV-46). It consists of the production of oxygen atoms followed by a chain of two reactions for the production of nitrogen oxide:

$$O_2 + M \rightleftarrows 2O + M \qquad \text{(IV-19)}$$

$$O + N_2 \rightleftarrows NO + N \qquad \text{(IV-20)}$$

$$N + O_2 \rightleftarrows NO + O \qquad \text{(IV-21)}$$

While there is evidence of the formation of "prompt" NO (Ref. IV-47), it is believed that the bulk of the NO is formed by the thermal mechanism and the current evaluation was based on that assumption. A fourth reaction,

$$OH + N \rightleftarrows NO + H \qquad \text{(IV-22)}$$

can become important when the oxygen concentration becomes low under fuel-rich conditions. Several equivalent solutions to the above equations are given in Refs. IV-48 through IV-50. They all assume that the nitrogen atoms are in equilibrium with the products of combustion and that the flame temperature is constant.

Because the NO formation rate is highly temperature dependent, virtually all NO is produced in the high temperature zone of the burner where conditions are generally close to stoichiometric combustion conditions. The NO_x standard in Table IV-16 corresponds to a NO_2 concentration of 0.35 lb/10^6 Btu (equivalent to a mole fraction of 0.00026) which is substantially below its equilibrium value. Accordingly, the reverse reactions in Eqs. IV-20 and IV-21 can be neglected. Also, since the forward rate constant of Eq. IV-21 is much higher than that of Eq. IV-20, the rate of NO production can be taken as twice that from Eq. IV-20 giving:

$$\frac{d(NO)}{dt} = 2\ k\ (O)\ (N_2) \qquad \text{(IV-23)}$$

where k is the forward rate constant for Eq. IV-20.

The aforementioned approximations lead to the following simplified solution (see Ref. IV-48):

$$\rho = \frac{\theta t}{2} \qquad \text{(IV-24)}$$

where

$$\rho = \frac{X_{NO}}{(X_{NO})_e} = \frac{\text{mole fraction NO}}{\text{mole fraction NO at equilibrium}}$$

t = apparent residence time, sec

and

$$\theta = \frac{4.24 \times 10^{15}\ P^{\frac{1}{2}}}{T} (X_{N_2})^{\frac{1}{2}} \exp\left(\frac{-114,572}{RT}\right) \qquad \text{(IV-25)}$$

where

θ = NO formation parameter, sec^{-1}

T = temperature, K

P = pressure, atm

X_{N_2} = mole fraction of N_2

R = gas constant, 1.987 $\frac{cal}{mole - K}$

The term θ, the linear rate of increase of NO concentration, is heavily dependent on temperature and, of interest in CAPS, increases with the square root of pressure. In a simple-cycle gas turbine, burner air inlet temperature increases with pressure ratio, producing higher flame temperatures. The increased temperature is generally the dominant factor in increased NO production rates. Fortunately for CAPS, increased storage pressure does not imply an increase in high-pressure burner inlet temperature because of the intercooling and aftercooling. This is shown in Table IV-17 which summarizes the factors of interest in NO formation for the parametric variation in storage pressure.

In the calculation of the parameter θ, another approximation can be made as suggested in Ref. IV-48. Because the mole fraction of N_2 in the stoichiometric fuel-air mixture is relatively constant (.71 to .73) over all conditions encountered, it is possible to construct the curve shown in Fig. IV-45, which is a function of temperature only. The function is a straight line on semilog paper because the value of the exponential term dominates.

CAPS Nitrogen Oxide Emissions

Since the FT50 engine is taken to be the basis of the CAPS turbomachinery, the CAPS combustors can be expected to have similar combustion and mixing characteristics. Thus, the principal factors affecting NO production rate can be expected to be the conditions in the flame zone that bear on NO formation rate, namely NO equilibrium concentration, pressure, temperature, and N_2 concentration. Since the FT50 is expected to meet the regulations, a figure of merit, β, for CAPS can be calculated as follows:

$$\beta = \frac{[\theta(X_{NO})_e]_{CAPS}}{[\theta(X_{NO})_e]_{FT50}} \qquad \text{(IV-26)}$$

where β is the ratio of NO produced per unit of fuel burned in the CAPS burner to that produced in the FT50 burner. Values of β less than unity will then meet NO_X emission standards. The basic assumption is that the two combustion zones are at the same equivalence ratio and that the apparent residence time, t, for NO formation remains unchanged.

Since CAPS has two burners it is necessary to determine the relative emission level, β, for each burner individually and then take a weighted average to arrive at the value of β for the overall system. It is assumed that combustion occurs at stoichiometric conditions and the values of θ and $(X_{NO})_e$ used in determining β are the result of an equilibrium calculation at those conditions. The variation in relative NO production as a function of system pressure ratio is shown in Fig. IV-46 for the high-pressure burner. At higher pressures the increased temperature causes the value of β to rise even faster than would have been expected from the square root of pressure term in Eq. IV-25. For pressures above 28 atm, NO production per unit of fuel burned would be greater than allowed, however, a relatively small portion of the fuel would be consumed in the high pressure burner so that a reduced level in the low-pressure burner could make up for the excess.

A combination of air and combustion products enters the low-pressure burner after having been expanded through the high-pressure turbine. Consequently, the low-pressure burner sees vitiated air, with the degree of vitiation depending on the overall fuel/air ratio in the high-pressure burner. Figure IV-47 shows the resultant low-pressure burner parameters as a function of fuel/air ratio in the high-pressure burner. In using the value of β in Fig. IV-45 for the low-pressure burner to estimate the contribution of the low-pressure burner to the total NO_x production, it is necessary to correct β to represent only the fuel burned in the low pressure burner. Direct application of Eq. IV-26 gives relative NO production based on the total amount of fuel burned in both high- and low-pressure burners. It is therefore necessary to multiply by the ratio of total fuel to fuel burned in the low pressure burner to obtain the corrected value. This correction is relatively small. The overall effect of the vitiated air is quite significant in reducing the amount of NO produced per unit of fuel burned.

In Fig. IV-48, the contribution of each burner is shown along with the net production per total pounds of fuel burned. Clearly, there is an optimum storage pressure from a pollution standpoint; but with the conditions assumed for this study, NO_x production does not appear to be a factor that will limit the application of CAPS. In fact, there is good reason to expect a reduced nitrogen oxide emission over simple cycle peaking engines.

<u>Carbon Monoxide</u>

While there is presently no carbon monoxide standard on industrial gas turbines, it is nevertheless important that the affect of the CAPS operating conditions on CO formation be evaluated to identify any potential problem areas. A review of the CO-to-CO_2 conversion mechanism shows that the increased operating pressure will result in a significant reduction in CO emissions while there should be little, if any, change due to the effect of combustion products in the air to the second or low pressure burner.

The evaluation of potential CO emission was done in a manner similar to that of NO_x formation. However, the mechanism differs in that the CO formed in the primary combustion region must be depleted. The required degree of depletion can only occur in the relatively cool zone downstream of the combustion zone. Large quantities of CO are formed in the primary combustion zone which are subsequently depleted by oxidation to CO_2. While the rate of depletion is rapid at combustion temperature, equilibrium values of CO generally exceed proposed requirements at all values of equivalence ratio greater than 0.7. As a result, the mixing zone downstream of the high temperature combustion zone is critical in achieving the desired CO emission levels.

Carbon Monoxide Combustion Mechanism

The carbon combustion mechanism is generally considered to be a two-step reaction with CO being produced and subsequently oxidized to CO_2 by the following reaction:

$$(CO) + (OH) \underset{k_b}{\overset{k_f}{\rightleftarrows}} (CO_2) + (H) \qquad \text{(IV-27)}$$

where k_f and k_b are the forward and backward rate constants. The parentheses denote molar concentrations of individual components.

The rate of CO depletion can be expressed as

$$\frac{d\,(CO)}{dt} = -k_f\,(CO)\,(OH) + k_b\,(H)\,(CO_2) \qquad \text{(IV-28)}$$

This equation can be integrated (Ref. IV-49) to yield

$$\ln\left\{\frac{(CO) - (CO)_e}{(CO_2)}\right\} = -k_f\,(OH)_e\left\{1 + \frac{(CO)_e}{(CO_2)_e}\right\} t \qquad \text{(IV-29)}$$

when it is assumed that H and OH are in equilibrium and that carbon conservation can be expressed as:

$$(CO) + (CO_2) = (CO)_e + (CO_2)_e \qquad \text{(IV-30)}$$

Recognizing the fact that $(CO)_e \lll (CO_2)_e$ at all lean equivalence ratios and substituting for the equilibrium value for (OH) in terms of (H_2O) and (O_2) gives:

$$\ln\left\{\frac{(CO)}{(CO_2)}\right\} = -\frac{k_f}{K_c^{\frac{1}{2}}}\,(H_2O)^{\frac{1}{2}}\,(O_2)^{\frac{1}{4}} \qquad \text{(IV-31)}$$

where K_c = equilibrium constant for OH formation given by:

$$K_c = \frac{(H_2O)\,(O_2)^{\frac{1}{2}}}{(OH)^2} \qquad \text{(IV-32)}$$

Using the relationship of Eq. IV-31 and evaluating the constants gives the relation between two different operating conditions, 1 and 2 (from Ref. IV-51):

$$\frac{\ln\left\{\dfrac{\left[\dfrac{(EI_{CO})}{28}\right]}{\left[\dfrac{(EI_{CO_2})}{44}\right]}\right\}_1}{\ln\left\{\dfrac{\left[\dfrac{EI_{CO}}{28}\right]}{\left[\dfrac{EI_{CO_2}}{44}\right]}\right\}_2} = \left[\frac{(EI_{H_2O})_1}{(EI_{H_2O})_2}\right]^{1/2}\left[\frac{(EI_{O_2})_1}{(EI_{O_2})_2}\right]^{1/4}\{e^{[-35,280/T_1][1-(T_1/T_2)]}\}\ x$$

$$\left[\frac{\left(\dfrac{P_1}{P_2}\right)}{\left(\dfrac{T_1}{T_2}\right)}\right]^{3/4}\left[\frac{\left(\dfrac{f}{a}\right)_1}{\left(\dfrac{f}{a}\right)_2}\right]^{3/4}\left[\frac{1+\left(\dfrac{f}{a}\right)_2}{1+\left(\dfrac{f}{a}\right)_1}\right]^{3/4}\left(\frac{t_1}{t_2}\right) \qquad \text{(IV-33)}$$

where:

EI = Emission Index, lb/1000 lb fuel

T = reaction temperature, R

P = pressure, atm

f/a = fuel/air ratio, lb/lb

t = apparent residence time, sec.

In the CO depletion zone the concentrations of dissociated species are negligible and the EI_{H_2O} and EI_{CO_2} are directly related to the fuel. Also, EI_{O_2} is a function of fuel/air ratio. These, therefore, can be specified in Eq. IV-33. The temperature T is really nonexistent, but is the characteristic temperature of the depletion zone. It is not really affected by oxidant temperature or combustion temperature, at least within reasonable limits. Therefore, the ratio T_1/T_2 is taken to be unity and Eq. IV-33 reduces to (from Ref. IV-51)):

$$\left(\frac{EI_{CO}}{2016}\right)_1 = \left(\frac{EI_{CO}}{2016}\right)_2^{(P_1/P_2)^{3/4}[(f/a)_1/(f/a)_2]^{3/4}\left\{\left[\frac{232}{(f/a)_1}-3385\right]/\left[\frac{232}{(f/a)_2}-3385\right]\right\}^{1/4} t_1/t_2} \qquad \text{(IV-34)}$$

In considering the effect of operation over a range of conditions for a given combustor, the analysis of Ref. IV-51 assumes a constant value of the flow parameter $W\sqrt{T}/P$ to provide a measure of the ratio t_1/t_2. This introduces the ratio $\sqrt{T_2/T_1}$ in place of t_1/t_2 in Eq. IV-34. However, in this application the assumption of the same or comparable residence time seems more apporpriate, so the ratio t_1/t_2 is taken as unity.

A further simplification of Eq. IV-34 is appropriate in considering emissions from a burner designed for a specific application. Here the fluid dynamics of the CO depletion zone can be assumed to be similar to that of the reference burner. As a result, the concentrations of CO, CO_2 and O_2 in the CO depletion zone will be similar and the fuel-air ratios in Eq. IV-34 can be taken as unity.

With the above simplifications, Eq. IV-34 reduces to:

$$\left(\frac{EI_{CO}}{2016}\right)_1 = \left(\frac{EI_{CO}}{2016}\right)_2 \left(\frac{P_1}{P_2}\right)^{3/4} \qquad \text{(IV-35)}$$

Carbon Monoxide Emission Estimates

For the CAPS system, the high pressure in the first burner gives (from Eq. IV-35) an EI_{CO} of approximately .0015 lb CO/1000 lb fuel based on an FT50 design value of 5 lb CO/1000 lb fuel at a pressure of 16 atm. While such a drastic reduction might not realistically materialize, it certainly does indicate that the likely trend is toward very low CO emission in the high pressure burner while the fuel burned in the second or low pressure burner would have an emission index comparable to that of the basic design for the FT50.

While the above extrapolation technique is believed to be based on a good representation of the mechanics of CO depletion, it is interesting to consider the available test data presented in Ref. IV-51. For various data points on a particular engine design, a plot of observed CO emission index as a function of fuel/air ratio shows a good deal of scatter. The use of a correction based on the above depletion mechanism reduced the scatter significantly. However, very good results were achieved using the simple linear empirical relationship:

$$\left(EI_{CO}\right)_{corr} = \left(EI_{CO}\right)_{observed} \left(\frac{P_{observed}}{P_{ref}}\right) \qquad \text{(IV-36)}$$

While this shows the same trend, clearly the degree of reduction of CO with pressure is much less than for Eq. IV-35. It must be emphasized, however, that Eq. IV-36 applies to data for a fixed design when run at varying pressures. Because of this, pressure increases are accompanied by an increase in air temperature. The primary effect of this temperature increase is a reduction in residence time which is proportional to the square root of temperature. For the CAPS system the estimate is based on a change in pressure only so that the actual CO reduction would be greater than that expressed by Eq. IV-36.

While not quantitative, the above discussion does show a definite trend toward lower CO emissions at high pressure. This, combined with a low pressure burner having characteristics similar to the equipment from which it is derived, provides sufficient confidence in the ability to achieve acceptable CO levels under CAPS operating conditions.

Particulates

Particulate emissions from a gas turbine power plant present a special problem in that the particle size is quite small (on the order of 0.1 micron) and they produce a degree of opacity in the stack gas that is out of proportion to their total mass. Because of their small size, these particles are more able to enter the respiratory system. Unlike diesel engine exhaust, however, gas turbine smoke is believed to contain negligible carcinogenic components (Ref. IV-52).

The visible nature of particulates in the gas turbine exhaust has resulted in the general use of an opacity measurement in defining the standards for particulate emissions. In some respects this is unfortunate since the degree of opacity is a function of total gas flow as well as the specific emission rate measured in pounds per million Btu. Current opacity levels for the FT4 gas turbine are on the order of 5 to 7 percent which corresponds to specific emission level of about .03 lb per million Btu. It is reasonable to expect that even lower specific emission levels will be achievable with FT50 hardware, and the resultant plume opacity will be primarily a function of exhaust flow rate. Since the degree of opacity is approximately a direct function of depth of the interfering media, for a constant specific emission level the opacity can be expected to increase by a factor of 1.7 over the FT4. While the resultant opacity is marginal, the continued development work being done in both combustion and fuel additives which agglomerate the small particles should produce significant advances, and particulate emission is not expected to be a problem in the CAPS concept.

Noise

Allowable noise levels are highly dependent on turbine location. In the case of CAPS, it is anticipated that the construction of the cavern and the compensating pond would result in a relatively rural location with little need for a high degree of sound attenuation. However, a standard FT50 installation would be able to meet the levels described in Table IV-18.

There are three important sources of noise associated with a gas turbine system; these are the inlet, exhaust stack, and casing radiated noise. In the CAPS application not all of these sources will be operative at any one time making the overall task of suppression somewhat easier than for a standard peaking system. Sound treatment is required on a gas turbine unit to attenuate noise generated from the gas turbine casing, generator casing, primary air inlet and gas turbine exhaust. In addition, cooling requirements for both the gas turbine and the generator necessitate cooling air inlets and exhausts which must also be silenced. A sound barrier, constructed of sound absorbing steel walls, is used to isolate the casing noise sources from the outside environment. Wherever airflow is required, whether for primary air or cooling, the basic means of sound attentuation in the ductwork is use of parallel sound absorbing baffles.

The gas turbine and generator units are housed in sound attenuating enclosures. The sound treatment is an integral part of the basic structure of the gas turbine and generator enclosure. The gas turbine enclosure is in turn housed in a weatherproof building. Enclosure wall and roof panels are constructed of a sandwich of mild steel sheets internally lined with fiberglass approximately four inches thick which is held in place by a perforated steel plate or screen. Sound absorbed in the fiberglass reduces the noise levels inside the enclosure while the steel wall reduces the noise levels transmitted to the outside outer building.

Storage Compensation Reservoir

The compensating reservoir could operate in two modes, as a storage pond for compensating water or as a combined storage/cooling pond. The environmental impact of each mode of operation is considered and recommendations made on which mode should be adopted in the plant design. Unless otherwise stated, all discussions refer to a single-unit plant located in the northeastern U.S., operating on a 20-hour cycle with a 20-foot reservoir drawdown.

Impact with Water Cooling

Reservoir water would be circulated through a series of heat exchangers (intercoolers and aftercoolers) to withdraw waste heat generated during the air compression cycle. The reservoir would serve as a heat sink, slowly releasing its thermal load to the atmosphere. The proposed design calls for a 45 F water temperature drop across the exchangers to reduce pumping requirements and reduce the pond volume. Spray coolers would operate periodically during hot weather to reduce the reservoir heat load. Even though, equilibrium water temperatures for facilities in the northeastern U.S. could exceed 95 F in the summer and 90 F in the winter.

Increased temperature would accelerate evaporation, so that in the summer over 0.6 cfs inflow would be required just to maintain reservoir volume. The initial dissolved and suspended solids content of this makeup would steadily accumulate in the reservoir, eventually fouling the heat exchangers. To limit solids concentration to roughly twice that of the inflow water would require about 0.6 cfs of blowdown; the total makeup water needed would be 1.2 cfs. These volumes are over twenty times those required by an air-to-air cooled plant of similar capacity. Moreover, even larger inflows might be necessary if intake water quality should be poor, or if environmental regulations require further dilution.

Serious fogging and icing problems could be expected near such a facility during the winter due to the warm open water. Local automobile traffic could be imperiled by poor visibility and slippery roads. Power and phone lines might be downed by heavy ice loading, thus inconveniencing and possibly endangering local inhabitants. Elevated reservoir temperatures would foster nuisance algal growths. Blue-green algae especially would flourish in the 90 F water (Ref. IV-53). These forms are particularly noxious; many grow rapidly and secrete large quantities of mucilage which can easily foul heat exchanger tubing. Use of algicides, such as copper sulfate or 2-3 dichloronapthoquinone, might adversely affect organisms exposed to the reservoir discharge (Ref. IV-54).

Elevated reservoir temperatures might also cause heavy mortalities among other organisms entrained in the reservoir intake flow or brought in when the reservoir was initially filled. Furthermore, organisms entrained in discharge water would be subjected to a sudden drop in temperature of up to 65 F in the winter: very few would survive.

The potential environmental hazards of such thermal additions would complicate permit applications for a water-cooled facility. Such a plant could be subject to Section 316A and B of the 1972 Federal Water Pollution Control Act which requires the applicant to demonstrate that the proposed facility will not significantly endanger the local ecology. This process would in itself increase project costs and might also increase construction costs if work schedules are interrupted.

In summary, use of the pressure-compensating reservoir as a source of cooling water could deprive CAPS systems of one of their major advantages -- a lack of serious adverse environmental impact.

Furthermore, such a facility would require much more water than a corresponding air-cooled plant, so that siting flexibility would be reduced.

Impact with Direct Air Cooling

Water Supply and Quality

The system as envisaged would require very little makeup water -- probably less than .05 cfs (25 gpm) -- even during the hot summer months. A plastic liner would prevent leakage from the reservoir. The low permeability (10^{-6} cm/sec) of rock selected for the cavern would limit seepage to about 10 gpm. Annual evaporation losses from the reservoir would average about 25 gpm. Precipitation would at least partially offset these minor losses.

The small makeup requirement could easily be provided by local surface or groundwater supplies on a continual or periodic basis. Approximately half of the makeup would be discharged as reservoir blowdown. Assuming blowdown equals net water loss, dissolved and suspended solids concentrations in the reservoir (and at the outfall) would be limited to roughly twice the inflow concentrations. In most cases, the chemical quality of a receiving body would not be significantly degraded by this small (about 2 gpm) system effluent. Alternatively, the effluent could be injected into a local aquifer provided that groundwater quality would not suffer.

Water Temperature

The thermal behavior of the storage reservoir would follow that of any similarly sized natural pond with one exception -- the reservoir could be less susceptible to freezing because of geothermal heating in the cavern. During the summer the reservoir temperatures would approach ambient levels.

Aquatic Biology

Plankton and small fish entrained in the makeup flow could be subjected to lethal or debilitating mechanical stresses. Based on recent experience (Ref. IV-53), mortality can be conservatively estimated at 10 percent. This is probably insignificant, considering the low flows involved. Larger organisms could be excluded at the reservoir intake.

The cycling of water from the cavern to the upper reservoir could also adversely affect entrained organisms. Sensitive species could be killed by the relatively rapid pressure changes. Mechanical abrasion would not be a problem -- the shaft diameter of 12 ft reduces velocites to only 1.6 ft/sec.

The pond would support algal growth if nutrient levels are sufficient. Most nutrients (nitrates and phosphates) would enter with the inflow; the balance would be contributed by airborne particulates and precipitation. No nutrients would be contributed by the cavern and lined reservoir. Therefore, the biological fertility of the reservoir would be chiefly a function of the initial nutrient content and inflow loading rate. Algal productivity should decline gradually if wells provide inflow, because groundwater is usually low in nutrients. Note that even if the pond becomes eutrophic, the receiving body would be little affected -- the discharge flow is an insignificant 12 gpm.

The reservoir might support insects such as mosquitoes and gnats whose larvae typically abound in small natural ponds. According to Ref. IV-55, mosquito infestation might be prevented by:

(i) Providing a windswept water surface to discourage insect landings.

(ii) Removing or preventing the growth of floating vegetation.

(iii) Clearing all overhanging vegetation, which would harbor adults.

Because high dikes surround the reservoir (i) is impractical. Option (ii) could be accomplished by screening the makeup water and periodically harvesting any floating plants. Option (iii) could be incorporated into site landscaping.

Fogging

Fogging problems at the plant during much of the year should be comparable to those at nearby water bodies of similar size. During the winter, however, the reservoir would be warmer than a natural pond due to geothermal heating and constant turnover. Even in northern climates, the reservoir is not expected to freeze over, and the resulting open water surface would generate fog. The encircling dikes should largely confine fog on site, although at times any roads adjacent to the plant might become fog bound and icy, endangering local motorists. Accordingly, the reservoir should be located where prevailing winds could sweep escaping fog away from roads and developed areas.

Discharge Permits

The reservoir discharge might be subject to State or Federal licensing. The responsible agency would issue a permit to discharge only if certain conditions are satisfied. The Federal Water Pollution Control Act (PL 92-500) authorizes three regulatory mechanisms that could be used to control reservoir discharges:

(a) States could require a National Pollutant Discharge Elimination System (NPDES) permit, which would incorporate a strict schedule for effluent quality improvement.

(b) States could require a Water Quality Certification, which would be issued only if the applicant demonstrated that inland receiving water (ambient) quality would not be degraded.

(c) States could require a similar certification covering coastal waters.

It is unclear how any given state would react to the discharge from an air storage compensating reservoir. However, even if strict controls were mandated, permit conditions should be relatively easy to meet -- effluent quality could easily be improved by increasing blowdown. For instance, if blowdown were increased to 1.0 cfs (450 gpm), the dissolved and suspended solids concentration in the reservoir would level off at 120 - 130 percent of the inflow concentration.

Excluding the Public

A security fence would prevent passers by from wandering onto the site. This fence could be placedeither at the perimeter of the plant property or atop the encircling dikes. Of these two arrangements, the former would better protect the site owner from liability suits because unauthorized personnel would be completely excluded from plant property. However, the latter would be much easier to patrol, and would allow greater flexibility in landscaping the outer walls (see subsequent discussion of aesthetics).

Groundwater

Plant operation might affect the local groundwater regime. Pressurized air leaking into the rock surrounding the cavern could disrupt the groundwater flow, reducing well yields "downstream" of the facility. Field studies prior to construction would be required to assess such potential problems.

Water infiltration would have negligible impact -- no water would escape from the lined reservoir and only about 10 gpm would escape from the cavern.

Aesthetics

The proposed plant and pond would be shielded from view by 30 - 40 foot dikes, 120 - 200 feet wide at the base. Any objectionable, unnatural geometric look could be masked by various landscaping practices.

Landscaping would serve three useful purposes: mask the unnaturally angular shape of the encircling dikes, intercept windblown matter which might otherwise settle in the reservoir, and block turbine intake and exhaust noise. Figures IV-49 and IV-50 illustrate configurations that might be adopted. For further information, the reader is referred to a recent review of landscape concepts for diked dredged material disposal areas (Ref. IV-56).

Impact of Plant Construction

Construction activities would temporarily disrupt a 50-acre area, and permanently destroy about 30 acres of wildlife habitat. Construction should be scheduled to avoid especially sensitive periods, such as spawning or nesting.

Local inhabitants, human or otherwise, would be subjected to noise and vibrations from machinery and blasting, vehicle exhaust fumes, and fugitive dust. Local waterways could be temporarily polluted by silt-laden runoff. Impacts such as these can often be controlled and reduced. Dust, for example, can be minimized by using blasting mats and spraying the construction area with water. Sediment run-off can be controlled via settling ponds during construction and, immediately following construction, by mulching or covering with biodegradable paper containing grass seeds.

Initial filling of the reservoir would require approximately 500,000 cubic yards of water. A large surface source of water would be required to fill the reservoir in a reasonably short period of time. For example, to fill the reservoir in 30 days would require over 5 cfs. Use of groundwater alone for filling could severely tax local aquifers; besides, it is unlikely that a sufficiently productive source would be located nearby. Use of municipal water would be much too costly, even if a willing supplier could be found.

Site construction will generate 3-400,000 cubic yards of rock in excess of fill requirements for the basic dikes. A small portion of this could be used on site in road bedding, building foundations, or drainage systems. Much or all of the excess could be used for landscaping. Any remainder must be removed for use or ultimate disposal.

Every effort should be made to identify local industries or individuals willing to accept -- or, better yet, buy -- the rock that cannot be used on site. Crushing or grading the rock to a potential user's specifications might prove worthwhile and even profitable. At the very least, users might be persuaded to periodically remove whatever rock they need from a centralized stockpile. Potential users include:

(a) Highway departments and construction firms (for fill, foundations, and drainage systems).

(b) Property owners desiring clean fill.

(c) Agencies or individuals involved in bank stabilization or erosion protection.

(d) Sand and gravel plants.

(e) Concrete suppliers.

REFERENCES

IV-1 Rice, N. C., F. L. Robson, A. J. Giramonti, and E. B. Smith: Open-Cycle Gas Turbine Topping Systems. 10th Intersociety Energy Conversion Engineering Conference, University of Delaware, August 1975.

IV-2 Dvorak, H. G. and W. H. Rogers, Jr.: An Advanced Industrial Gas Turbine for Utility Generation. American Power Conference, Chicago, April 1975.

IV-3 Bunch, R. J.: Ingersoll-Rand Correspondence with W. R. Davison of UTRC, November 18, 1975.

IV-4 Szczesny, F. F.: Allis Chalmers Correspondence with W. R. Davison, of UTRC, October 20, 1975.

IV-5 Jansson, S: Stal Laval Correspondence with A. J. Giramonti of UTRC, October 25, 1976.

IV-6 Redmond, K. E.: De Laval Correspondence with W. R. Davison of UTRC, August 11, 1976.

IV-7 Giramonti, A. J.: Preliminary Feasibility Evaluation of Compressed Air Storage Power Systems. UTRC Report R76-952161-4, May, 1976.

IV-8 Baumeister, T., ed. Mechanical Engineers's Handbook, McGraw-Hill Company, 1958.

IV-9 Giramonti, A. J.: Visits with European Experts on CAPS. UTRC Report R75-952171-1. September 1975.

IV-10 Giramonti, A. J.: Advanced Power System Discussions with European Organizations. UTRC Report N-151109-1, November 1974.

IV-11 Sys, Z. S.: Air Storage System Energy Transfer (ASSET). Presented at ERDA/EPRI Compressed Air Energy Storage Workshop, Airlie, Virgina, December 18, 1975.

IV-12 First Pumped-Air Storage Plant Nears Reality, Engineering News Record, February 17, 1972.

IV-13 Mechanical Power Transmission Handbook. Zurn Industries Manual, No. 564, 1970.

REFERENCES (CON'T)

IV-14 SSS Clutches for Pumped Air Storage Generating Plant. SSS Gear Note Reference 0674.

IV-15 Luthi, H. R.: Air Storage Power. Presented at CEA Spring Meeting, Vancouver, B. C., March 18, 1975.

IV-16 Mattick, W.: Huntorf-The World's First 290-MW Gas Turbine Air Storage Peaking Plant. Presented at the American Power Conference, Chicago, Ill. April 21-23, 1975.

IV-17 Automatic Synchro-Self-Shifting Clutch: Waukesha-SSS Catalog W-12.

IV-18 Discussions with Mr. J. Gruber of Waukesha Bearing, October 2, 1975.

IV-19 Synchronous Clutch Coupling: MAAG Gear Wheel Prospectus G40-E.

IV-20 Verkaufsingenieur, B. M.: Kurzbeschreibung der Kombination Anfahr-Synchronisirzahnkupplung. MAAG Gear Wheel.

IV-21 Telephone Conversation with Mr. S. Jansson of Stal Laval, February 4, 1976.

IV-22 Discussion with Combustion Engineering Personnel, May 17, 1976.

IV-23 Steam. Babcock and Wilcox Company, 1960.

IV-24 Shields, C. D.: Boilers. F. E. Dodge Corporation, 1961.

IV-25 Kays, W. M. and A. L. London: Compact Heat Exchangers. McGraw-Hill Company, NY, 1958.

IV-26 Kreith, F.: Principles of Heat Transfer. International Textbook Company. Scranton, Penn, 1964.

IV-27 Jakob, M. and G. A. Hawkins: Elements of Heat Transfer. John Wiley & Sons, NY, 1957.

IV-28 Fraas, A. P. and M. N. Oaisik: Heat Exhanger Design. John Wiley & Sons, NY, 1965.

REFERENCES (CONT'D)

IV-29 How to Select a Waste Heat Boiler. Waste Heat Energy Corporation.

IV-30 Huff, F. A. et al: Effect of Cooling Tower Effluents on Atmospheric Conditions in Northeastern Illinois. Illinois State Water Survey, Urbana, 1971.

IV-31 Cooling Towers and the Environment. American Electric Power Service Corporation, October 1974.

IV-32 Currier, E. L. et al: Cooling Pond Steam Fog. Journal of the Air Pollution Control Association, Vol. 24, No. 9, September 1974.

IV-33 Thackston, E. L.: Effect of Geographical Variation on Performance of Recirculating Cooling Ponds. National Environmental Research Center, Office of Research and Development, U. S. Environmental Protection Agency, Corvallis, OR 97330 - EPA-660/2-74-085.

IV-34 Larsen, I. and D. Noren: Hydraulics of Compressed Air Power Plants. Paper Presented at the Third Conference on Compressed Gas Economy. Budapest, Yugoslavia, April 1976.

IV-35 Milne, I. A., et al: Underground Storage-Summary of Visit to Sweden and Norway. August 24 to September 4, 1975. Acres American, Incorporated, October 1975.

IV-36 Gouse, S. W.: An Index to the Two-Phase Gas-Liquid Flow Literature. MIT Press, Cambridge, Mass., 1966.

IV-37 Grovier, G. W. and K. Aziz: The Flow of Complex Mixtures in Pipes. Van Nostraud, New York, 1972.

IV-38 Batchelor, G. K.: An Introduction to Fluid Mechanics. Cambridge U. Press, London, 1967.

IV-39 Roberts, R., L. D. Aceto, R. Kollrach, D. P. Teixeria, and J. M. Bonnell: An Analytical Model for Nitric Oxide Formation in a Gas Turbine Combustor. AIAA Journal, Vol. 10, No. 6, June 1972.

IV-40 Mador, R. J. and R. Roberts: A Pollutant Emission Prediction Model for Gas Turbine Combustor. Paper presented at 10th Annual AIAA/SAE Propulsion Conference, San Diego, October 21-23, 1974.

REFERENCES (CON'T)

IV-41 Moiser, S. A. and R. Roberts. Low-power Turbo Propulsion Combustor Exhaust Emissions. Vol. 1, Theoretical Formulation Design and Assessment. AFAPL-TR-73-36, June 1973.

IV-42 Moiser, S. A. and R. Roberts: Low-power Turbo Propulsion Combustor Exhaust Emissions, Vol. 2, Demonstration and Total Emission Analysis and Prediction. AFAPL-TR-73-36, April 1974.

IV-43 Moiser, S. A. and R. Roberts. Low-power Turbo Propulsion Combustor Exhaust Emissions, Vol. 3, Analysis. AFAPL-TR-73-36, July 1974.

IV-44 Moiser, S. A., R. Roberts and R. Henderson: Development and Verification of an Analytical Model for Predicting Emissions from Gas Turbine Engine Combustors During Low-power Operation, AGARD-CP-125, April 1973.

IV-45 .Marteney, P. J.: Analytical Study of the Kinetics of Formation of Nitrogen Oxide in Hydrocarbon-Air Combustion. Combustion Science and Technology, Vol. 1, pp 461-469, 1970.

IV-46 Zeldovich, J.: The Oxidation of Nitrogen in Combustion and Explosives. Acta Physiochmica, U.S.S.R., Vol. 21, 1946.

IV-47 Fenimore, C. P.: Formation of Nitrix Oxide in Premixed Hydrocarbon Flames. 13th International Symposium on Combustion, Salt Lake City, 1970.

IV-48 Shaw, H.: Fuel Modification for Abatement of Aircraft Turbine Engine Oxides of Nitrogen Emissions. NTIS Report No. AD752581, October 1972.

IV-49 Westenberg, A. A.: Kinetics of NO and CO in Lean Premixed Hydrocarbon-Air Flames. Combustion Science and Technology Vol. 4, 1971.

IV-50 Sarli, V. J.: Variation of NO Formation With Time and Humidity in the Combustion of JP-Fuels. United Aircraft Report No. UAR-L52, 1972.

IV-51 Sarli, V. J. et al.: Effects of Operating Variables on Gaseous Emissions. Air Pollution Control Association Specialty Conference on Air Pollution Measurement Accuracy as it Relates to Regulation Compliance, New Orleans, La., October 1975.

IV-52 Lindeer, L. H. and Heywood, J. B.: Smoke Emission from Jet Engines. Combustion Science and Technology, Vol. 2, 1971.

REFERENCES (CON'T)

IV-53 Glooschenko, W. A.: The Effects of Energy-Related Effluents on Productivity, Biomass, and Eutrophication in the Great Lakes. Proceedings of the Second Federal Conference on the Great Lakes, Public Information Office, Great Lakes Basin Commission, 1975.

IV-54 Gratteau, J.: Potential Algicides for the Control of Algae. Water and Sewage Works, 117(11): R-24 - ¢-61, 1970.

IV-55 Matheson, R. A.: A Handbook of Mosquitoes in North America. Charles C. Thomas, Publisher, 1929.

IV-56 Mann, R. W. H. Niering, R. Sabbatini, and P. Wells: Landscape Concept Development for Confined Dredged Material Sites U. S. Army Engineers Laboratory, Vicksburg, Missipi, 1975.

TABLE IV-1

ESTIMATED SELLING PRICE OF LOW-PRESSURE
TURBOMACHINERY COMPONENTS FOR CAPS SYSTEM

Item	Price
Low-Pressure Compression Section (Including Ducts)	$2,810,000
Burner and Low-Pressure Expansion Section (Including Ducts)	4,545,000
Lubricant Oil Cooler, Manifolds, Piping, Miscellaneous	555,000
Supports	30,000
Total Cost of Equipment in Low-Pressure Section	$7,938,000

TABLE IV-2

HOT GAS EXPANDERS FOR CAPS SYSTEM

(One Unit of Two)

Operating Conditions:

Inlet Temperature - 1000F

Inlet Pressure - 807.3 psia

Expansion Ratio - 3.45

Flow - 298.8 lb/sec

Characteristics	Design 1 (η_{adia} = 80%)	Design 2 (η_{adia} = 85%)
Nominal Reaction Value	0.2	0.2
Number of Stages	7	7
Hub-Tip Ratio, First Stage	0.918	0.925
Hub-Tip Ratio, Last Stage	0.804	0.824
Hub Diameter, First Stage	2.206 ft	2.287 ft
Hub Diameter, Last Stage	2.055 ft	2.15 ft
Blade Length, First Stage	1.18 in.	1.11 in.
Blade Length, Last Stage	3.01 in.	2.76 in.
Shaft Power	35,600 hp	37,800 hp

TABLE IV-3

HOT GAS EXPANDERS FOR CAPS SYSTEM

(Single Unit Required)

Operating Conditions:

Inlet Temperature - 1000F

Inlet Pressure - 807.3 psia

Expansion Ratio - 3.45

Flow - 597.7 lb/sec

Characteristics	Design 1 (η_{adia} = 80%)	Design 2 (η_{adia} = 85%)
Nominal Reaction Value	0.2	0.2
Number of Stages	4	4
Hub-Tip Ratio, First Stage	0.917	0.925
Hub-Tip Ratio, Last Stage	0.828	0.844
Hub Diameter, First Stage	2.920 ft	3.022 ft
Hub Diameter, Last Stage	2.763 ft	2.879 ft
Blade Length, First Stage	1.58 in.	1.47 in.
Blade Length, Last Stage	3.44 in.	3.19 in.
Shaft Power	71,100 hp	75,600 hp

TABLE IV-4

DESIGN STATE POINTS FOR COMPRESSOR OPERATION

50-Atmosphere Cycle (Reference Design)

Compressor inlet		
T_1 = 518 R	P_1 = 14.7 psia	H_1 = 123.8 Btu/lb
First intercooler entry		
T_2 = 805 R	P_2 = 60.2 psia	H_2 = 193.0 Btu/lb
First intercooler exit		
T_3 = 621 R	P_3 = 55.8 psia	H_3 = 148.5 Btu/lb
Second intercooler entry		
T_4 = 960 R	P_4 = 230.4 psia	H_4 = 231.0 Btu/lb
Second intercooler exit		
T_5 = 632.5 R	P_5 = 225.8 psia	H_5 = 151.3 Btu/lb
Aftercooler entry		
T_6 = 960 R	P_6 = 729.2 psia	H_6 = 231.0 Btu/lb
Aftercooler entry		
T_7 = 580 R	P_7 = 714.6 psia	H_7 = 138.7 Btu/lb

32-Atmosphere Cycle

T_1 = 518 R	P_1 = 14.7 psia	H_1 = 123.8 Btu/lb
T_2 = 805 R	P_2 = 60.2 psia	H_2 = 193.0 Btu/lb
T_3 = 621 R	P_3 = 55.8 psia	H_3 = 148.5 Btu/lb
T_4 = 960 R	P_4 = 230.4 psia	H_4 = 231.1 Btu/lb
T_5 = 751 R	P_5 = 225.7 psia	H_5 = 180.1 Btu/lb
T_6 = 960 R	P_6 = 451.5 psia	H_6 = 231.1 Btu/lb
T_7 = 580 R	P_7 = 442.5 psia	H_7 = 138.7 Btu/lb

62-Atmosphere Cycle

T_1 = 518 R	P_1 = 14.7 psia	H_1 = 123.8 Btu/lb
T_2 = 805 R	P_2 = 60.2 psia	H_2 = 193.0 Btu/lb
T_3 = 621 R	P_3 = 55.8 psia	H_3 = 148.5 Btu/lb
T_4 = 960 R	P_4 = 230.4 psia	H_4 = 231.1 Btu/lb
T_5 = 585 R	P_5 = 225.8 psia	H_5 = 139.9 Btu/lb
T_6 = 960 R	P_6 = 903.1 psia	H_6 = 231.1 Btu/lb
T_7 = 580 R	P_7 = 885.0 psia	H_7 = 138.7 Btu/lb

TABLE IV-5

COST OF SURFACE COMPENSATING RESERVOIR

Reservoir Volume, 10^3 Cu Yd	Cost, 10^3 \$ Level Variation 10 ft	20 ft	30 ft
150	---	616	727
190(1)	955	695	745
250(2)	1210	850	890
340(3)	1590	1080	1090
500	---	1516	1531
1000	---	2676	2494

(1)Corresponds to 62-atm cycle.
(2)Corresponds to 50-atm reference cycle.
(3)Corresponds to 32-atm cycle.

TABLE IV-6

COOLING SYSTEM INSTALLED COSTS
280-MW PLANT

	Boston			Baltimore			Detroit		
	80 F	85 F	90 F	85 F	90 F	95 F	85 F	90 F	95 F
Cooling Pond- Heavyweight Liner 50-atm Cycle									
Reservoir	12,547,000	4,824,000	2,669,000	13,781,000	5,768,000	3,435,000	11,032,000	5,204,000	2,925,923
Structures	50,000	50,000	50,000	50,000	50,000	50,000	50,000	50,000	50,000
Subtotal	12,597,000	4,874,000	2,719,000	13,831,000	5,818,000	3,485,000	11,082,000	5,254,000	2,975,000
$/kW	45.0	17.4	9.7	49.4	20.8	12.4	39.6	18.8	10.6
Pipe and pump*	500,000	500,000	500,000	500,000	500,000	500,000	500,000	500,000	500,000
Total $/kW	46.8	19.2	11.5	51.2	22.6	14.2	41.4	20.6	12.4
Cooling Pond- Lightweight Liner 50-atm Cycle									
Reservoir	9,055,000	3,975,000	2,299,000	9,894,000	4,563,000	2,868,000	8,213,000	4,157,000	2,490,000
Structures	50,000	50,000	50,000	50,000	50,000	50,000	50,000	50,000	50,000
Subtotal	9,105,000	4,025,000	2,349,000	9,944,000	4,613,000	2,918,000	8,263,000	4,207,000	2,540,000
$/kW	32.5	14.4	8.4	35.5	16.5	10.4	29.5	15.0	9.1
Pipe and pump*	500,000	500,000	500,000	500,000	500,000	500,000	500,000	500,000	500,000
Total $/kW	34.3	16.2	10.2	37.3	18.3	12.2	31.3	16.8	10.9
Cooling Canals- Heavyweight Liner 50-atm Cycle									
Reservoir	1,643,000	1,643,000	1,643,000	—	—	—	—	—	—
Canals	10,344,000	6,364,000	3,909,000	—	—	—	—	—	—
Structures	50,000	50,000	50,000	—	—	—	—	—	—
Subtotal	12,037,000	8,057,000	5,602,000	—	—	—	—	—	—
$/kW	43.0	28.8	20.0	—	—	—	—	—	—
Pipe and pump*	500,000	500,000	500,000	—	—	—	—	—	—
Total $/kW	44.8	30.6	21.8	—	—	—	—	—	—
Cooling Canals- Lightweight Liner 50-atm Cycle									
Reservoir	1,499,000	1,499,000	1,499,000	—	—	—	—	—	—
Canals	12,600,000	6,186,000	3,909,000	—	—	—	—	—	—
Structures	50,000	50,000	50,000	—	—	—	—	—	—
Subtotal	14,149,000	8,057,000	5,602,000						
$/kW	43.0	28.8	20.0	—	—	—	—	—	—
Pipe and pump*	500,000	500,000	500,000	—	—	—	—	—	—
Total $/kW	44.8	30.6	21.8	—	—	—	—	—	

TABLE IV-6 (Continued)

	Boston			Baltimore			Detroit		
	80 F	**90 F**	**100 F**	**80 F**	**90 F**	**100 F**	**80 F**	**90 F**	**100 F**
Spray Cooling Canal- 4-Nozzle Spray Units 32-atm Cycle									
Equipment	1,011,000	547,000	383,000	847,000	492,000	355,000	929,000	519,000	355,000
Canal	705,000	371,000	260,000	594,000	334,000	260,000	631,000	371,000	260,000
Reservoir	1,450,000	1,450,000	1,450,000	1,450,000	1,450,000	1,450,000	1,450,000	1,450,000	1,450,000
Structures	50,000	50,000	50,000	50,000	50,000	50,000	50,000	50,000	50,000
Pipe credit	−147,000	−147,000	−147,000	−147,000	−147,000	−147,000	−147,000	−147,000	−147,000
Subtotal	3,069,000	2,271,000	1,996,000	2,794,000	2,179,000	1,968,000	2,913,000	2,243,000	1,968,000
$/kW	11.0	8.1	7.1	10.0	7.8	7.0	10.4	8.0	7.0
Pipe and pump*	500,000	500,000	500,000	500,000	500,000	500,000	500,000	500,000	500,000
Total $/kW	12.8	9.9	8.9	11.8	9.6	8.8	12.2	9.8	8.8
Spray Cooling Canal- 12-Nozzle Spray Units 32-atm Cycle									
Equipment	1,038,000	556,000	371,000	815,000	519,000	334,000	890,000	556,000	334,000
Canal	779,000	445,000	278,000	612,000	390,000	278,000	668,000	445,000	278,000
Reservoir	1,450,000	1,450,000	1,450,000	1,450,000	1,450,000	1,450,000	1,450,000	1,450,000	1,450,000
Structures	50,000	50,000	50,000	50,000	50,000	50,000	50,000	50,000	50,000
Pipe credit	−147,000	−147,000	−147,000	−147,000	−147,000	−147,000	−147,000	−147,000	−147,000
Subtotal	3,170,000	2,354,000	2,002,000	2,780,000	2,262,000	1,965,000	2,911,000	2,354,000	1,965,000
$/kW	11.3	8.4	7.2	9.9	8.1	7.0	10.4	8.4	7.0
Pipe and pump*	500,000	500,000	500,000	500,000	500,000	500,000	500,000	500,000	500,000
Total $/kW	13.1	10.2	9.0	11.7	9.9	8.8	12.2	10.2	8.8
Spray Cooling Canal- 4-Nozzle Spray Units 50-atm Cycle									
Equipment	1,148,000	601,000	410,000	929,000	547,000	383,000	1,039,000	574,000	383,000
Canal	796,000	408,000	278,000	631,000	371,000	278,000	705,000	407,000	278,000
Reservoir	1,450,000	1,450,000	1,450,000	1,450,000	1,450,000	1,450,000	1,450,000	1,450,000	1,450,000
Structures	50,000	50,000	50,000	50,000	50,000	50,000	50,000	50,000	50,000
Pipe credit	−147,000	−147,000	−147,000	−147,000	−147,000	−147,000	−147,000	−147,000	−147,000
Subtotal	3,297,000	2,362,000	2,041,000	2,913,000	2,271,000	2,014,000	3,097,000	2,334,000	2,014,000
$/kW	11.8	8.4	7.3	10.4	8.1	7.2	11.1	8.3	7.2
Pipe and pump*	500,000	500,000	500,000	500,000	500,000	500,000	500,000	500,000	500,000
Total $/kW	13.6	10.2	9.1	12.2	9.9	9.0	12.9	10.1	9.0

TABLE IV-6 (Continued)

	Boston			Baltimore			Detroit		
	80 F	90 F	100 F	80 F	90 F	100 F	80 F	90 F	100 F
Spray Cooling Canal-12-Nozzle Spray Units 50-atm Cycle									
Equipment	1,149,000	593,000	408,000	890,000	556,000	371,000	1,001,000	593,000	371,000
Canal	863,000	445,000	306,000	668,000	445,000	278,000	779,000	445,000	278,000
Reservoir	1,450,000	1,450,000	1,450,000	1,450,000	1,450,000	1,450,000	1,450,000	1,450,000	1,450,000
Structures	50,000	50,000	50,000	50,000	50,000	50,000	50,000	50,000	50,000
Pipe credit	−147,000	−147,000	−147,000	−147,000	−147,000	−147,000	−147,000	−147,000	−147,000
Subtotal	3,365,000	2,391,000	2,067,000	2,911,000	2,354,000	2,002,000	3,133,000	2,391,000	2,002,000
$/kW	12.0	8.5	7.4	10.4	8.4	7.2	11.2	8.5	7.2
Pipe and pump*	500,000	500,000	500,000	500,000	500,000	500,000	500,000	500,000	500,000
Total $/kW	13.8	10.3	9.2	12.2	10.2	9.0	13.0	10.3	9.0
Spray Cooling Canal-4-Nozzle Spray Units 62-atm Cycle									
Equipment	1,257,000	656,000	465,000	1,011,000	601,000	437,000	1,148,000	629,000	437,000
Canal	854,000	445,000	334,000	705,000	408,000	297,000	779,000	445,000	297,000
Reservoir	1,450,000	1,450,000	1,450,000	1,450,000	1,450,000	1,450,000	1,450,000	1,450,000	1,450,000
Structures	50,000	50,000	50,000	50,000	50,000	50,000	50,000	50,000	50,000
Pipe credit	−147,000	−147,000	−147,000	−147,000	−147,000	−147,000	−147,000	−147,000	−147,000
Subtotal	3,464,000	2,454,000	2,152,000	3,069,000	2,362,000	2,087,000	3,280,000	2,427,000	2,087,000
$/kW	12.4	8.8	7.7	11.0	8.4	7.5	11.7	8.7	7.5
Pipe and pump*	500,000	500,000	500,000	500,000	500,000	500,000	500,000	500,000	500,000
Total $/kW	14.2	10.6	9.5	12.8	10.2	9.3	13.5	10.5	9.3
Spray Cooling Canal-12-Nozzle Spray Units 62-atm Cycle									
Equipment	1,260,000	667,000	445,000	964,000	630,000	408,000	1,112,000	667,000	408,000
Canal	947,000	501,000	334,000	724,000	501,000	334,000	835,000	501,000	334,000
Reservoir	1,450,000	1,450,000	1,450,000	1,450,000	1,450,000	1,450,000	1,450,000	1,450,000	1,450,000
Structures	50,000	50,000	50,000	50,000	50,000	50,000	50,000	50,000	50,000
Pipe credit	−147,000	−147,000	−147,000	−147,000	−147,000	−147,000	−147,000	−147,000	−147,000
Subtotal	3,560,000	2,521,000	2,132,000	3,041,000	2,484,000	2,095,000	3,308,000	2,521,000	2,095,000
$/kW	12.7	9.0	7.6	10.9	8.9	7.5	11.8	9.0	7.5
Pipe and pump*	500,000	500,000	500,000	500,000	500,000	500,000	500,000	500,000	500,000
Total $/kW	14.5	10.8	9.4	12.7	10.7	9.3	13.6	10.8	9.3

TABLE IV-6 (Continued)

	Boston 80 F	Boston 90 F	Boston 100 F	Baltimore 80 F	Baltimore 90 F	Baltimore 100 F	Detroit 80 F	Detroit 90 F	Detroit 100 F
Wet Mechanical Cooling Tower-62-atm Cycle									
Concrete Structure Research-Cottrell									
Tower	1,520,000	1,160,000	-	1,700,000	1,250,000	890,000	1,610,000	1,070,000	-
Reservoir	1,080,000	1,080,000	-	1,080,000	1,080,000	1,080,000	1,080,000	1,080,000	-
Subtotal	2,600,000	2,240,000		2,780,000	2,330,000	1,990,000	2,690,000	2,150,000	-
$/kW	9.3	8	-	9.9	8.3	7.0	9.6	7.7	-
Pipe and pump*	500,000	500,000		500,000	500,000	500,000	500,000	500,000	-
Total $/kW	11.1	9.8	-	11.7	10.1	8.8	11.4	9.5	-
Ecodyne									
Tower	-	-	-	-	-	1,036,000	-	-	-
Reservoir	-	-	-	-	-	1,080,000	-	-	-
Subtotal	-	-	-	-	-	2,116,000	-	-	-
$/kW	-	-	-	-	-	7.6	-	-	-
Pipe and pump*	-	-	-	-	-	500,000	-	-	-
Total $/kW	-	-	-	-	-	93	-	-	-
Wood Structure Marley Co.									
Tower**	-	543,000	-	-	589,000	-	-	589,000	-
Reservoir	-	1,080,000	-	-	1,080,000	-	-	1,080,000	-
Subtotal	-	1,623,000	-	-	1,669,000	-	-	1,669,000	-
$/kW	-	5.9	-	-	6.0	-	-	6.0	-
Pipe and pump*	-	500,000	-	-	500,000	-	-	500,000	-
Total $/kW	-	7.6	-	-	7.7	-	-	7.7	-

TABLE IV-6 (Continued)

	Boston			Baltimore			Detroit		
	80 F	90 F	100 F	80 F	90 F	100 F	80 F	90 F	100 F
Wet Mechanical Cooling Tower-50-atm Cycle									
Concrete Structure Research-Cottrell									
Tower	1,700,000	1,300,000	–	1,900,000	1,400,000	1,000,000	1,800,000	1,200,000	–
Reservoir	850,000	850,000	–	850,000	850,000	850,000	850,000	850,000	–
Subtotal	2,550,000	2,150,000	–	2,750,000	2,250,000	1,850,000	2,650,000	2,050,000	–
$/kW	9.1	7.7	–	9.8	8.0	6.6	9.5	7.5	–
Pipe and pump*	500,000	500,000	–	500,000	500,000	500,000	500,000	500,000	–
Total $/kW	10.9	9.5	–	10.6	9.8	8.4	11.3	9.3	–
Ecodyne									
Tower	–	–	–	–	–	1,160,000	–	–	–
Reservoir	–	–	–	–	–	850,000	–	–	–
Subtotal	–	–	–	–	–	2,010,000	–	–	–
$/kW	–	–	–	–	–	7.2	–	–	–
Pipe and pump*	–	–	–	–	–	500,000	–	–	–
Total $/kW	–	–	–	–	–	9.0	–	–	–
Wood Structure Marley Co.									
Tower**	–	609,000	–	–	660,000	–	–	660,000	–
Reservoir	–	850,000	–	–	850,000	–	–	850,000	–
Subtotal	–	1,459,000	–	–	1,510,000	–	–	1,510,000	–
$/kW	–	5.2	–	–	5.4	–	–	5.4	–
Pipe and pump*	–	500,000	–	–	500,000	–	–	500,000	–
Total $/kW	–	7.0	–	–	7.2	–	–	7.2	–

TABLE IV-6 (Continued)

	Boston			Baltimore			Detroit		
	80 F	90 F	100 F	80 F	90 F	100 F	80 F	90 F	100 F
Wet Mechanical Cooling Tower- 50-atm Cycle									
Concrete Structure Research-Cottrell									
Tower	1,840,000	1,410,000	-	2,060,000	1,520,000	1,080,000	1,950,000	1,300,000	-
Reservoir	695,000	695,000	-	695,000	695,000	695,000	695,000	695,000	-
Subtotal	2,535,000	2,105,000	-	2,755,000	2,215,000	1,725,000	2,645,000	1,995,000	-
$/kW	9.1	7.5	-	9.9	7.9	6.4	9.5	7.1	-
Pipe and pump*	500,000	500,000	-	500,000	500,000	500,000	500,000	500,000	-
Total $/kW	10.9	9.3	-	11.7	9.7	9.2	11.3	8.9	-
Ecodyne									
Tower	-	-	-	-	-	1,257,000	-	-	-
Reservoir	-	-	-	-	-	695,000	-	-	-
Subtotal	-	-	-	-	-	1,952,000	-	-	-
$/kW	-	-	-	-	-	7.0	-	-	-
Pipe and pump*	-	-	-	-	-	500,000	-	-	-
Total $/kW	-	-	-	-	-	8.8	-	-	-
Wood Structure Marley Co.									
Tower**	-	659,000	-	-	715,000	-	-	715,000	-
Reservoir	-	695,000	-	-	695,000	-	-	695,000	-
Subtotal	-	1,354,000	-	-	1,410,000	-	-	1,410,000	-
$/kW	-	4.9	-	-	5.1	-	-	5.1	-
Pipe and pump*	-	500,000	-	-	500,00	-	-	500,000	-
Total $/kW	-	6.7	-	-	6.9	-	-	6.9	-

TABLE IV-6 (Continued)

	Boston			Baltimore			Detroit		
	80 F	90 F	100 F	80 F	90 F	100 F	80 F	90 F	100 F
Wet Natural Draft Cooling Tower 50-atm Cycle									
Research - Cottrell									
Tower	–	–	–	–	–	4,500,000	–	–	–
Reservoir	–	–	–	–	–	850,000	–	–	–
Subtotal	–	–	–	–	–	5,350,000	–	–	–
$/kW-	–	–	–	–	–	19.1	–	–	–
Pipe and pump*	–	–	–	–	–	500,000	–	–	–
Total $/kW	–	–	–	–	–	20.9	–	–	–
Dry Mechanical Cooling Tower 50-atm Cycle			110°F						
Tower	–	–	8,000,000	–	–	–	–	–	–
Reservoir	–	–	850,000	–	–	–	–	–	–
Pipe credit	–	–	–150,000	–	–	–	–	–	–
Subtotal	–	–	8,700,000	–	–	–	–	–	–
$/kW	–	–	31.1	–	–	–	–	–	–
Pipe and pump*	–	–	500,000	–	–	–	–	–	–
Total $/kW	–	–	32.9	–	–	–	–	–	–

*Evaluated on constant cost basis

**Replacement required during life of plant

TABLE IV-7

COOLING SYSTEM YEARLY OPERATING COSTS AT FULL LOAD

BASIS: 5 Mills/kWhr

	Fan,$	Circulation Pumping,$	Makeup,$	Maintenance and Treatment,$	Total,$	Total,$/MWhr
Wet Towers						
Research — Cottrell						
Boston 80 F	11,058	14,232	266	61,159	86,715	0.0848
90 F	5,862	13,786	266	61,159	81,073	0.0793
Detroit 80 F	14,605	15,235	266	61,159	91,265	0.0893
90 F	6,749	13,003	266	61,159	81,177	0.0794
Baltimore 80 F	15,738	15,689	266	61,159	92,852	0.0909
90 F	7,639	13,229	266	61,159	82,293	0.0805
100 F	3,882	13,003	266	61,159	78,310	0.0766
Ecodyne						
Baltimore 100 F	9,644	19,408	266	61,159	90,477	0.0885
Marley						
Boston 90 F	9,257	16,169	266	67,510	93,202	0.0912
Detroit 90 F	7,200	16,169	266	67,510	91,145	0.0892
Baltimore 90 F	8,563	16,169	266	67,510	92,508	0.0905
Dry Tower						
Hudson Products						
110 F cold — 90 F air	54,458	29,952	0	4,000	88,410	0.0865

TABLE IV-7 (Continued)

	Spray Pump,$	Circulation Pumping,$	Makeup,$	Maintenance and Treatment,$	Total,$	Total,$/MWhr
Spray Cooling Canal						
Ashbrook Corporation						
Boston 80 F						
75–4 model	34,715	4,852	243	2,000	41,810	0.0409
75–12 model	24.506	4,852	243	2,000	31,601	0.0309
90 F						
75–4 model	20,422	4,852	243	2,000	27,517	0.0269
75–12 model	15,316	4,852	243	2,000	22,411	0.0219
100 F						
75–4 model	14,294	4,852	243	2,000	21,389	0.0209
75–12 model	10,211	4,852	243	2,000	17,306	0.0169
Detroit 80 F						
75–4 model	38,803	4,852	243	2,000	45,898	0.0449
75–12 model	27,570	4,852	243	2,000	34,665	0.0339
90 F						
75–4 model	21,442	4,852	243	2,000	28,537	0.0279
75–12 model	16,337	4,852	243	2,000	23,432	0.0229
100 F						
75–4 model	14,294	4,852	243	2,000	21,389	0.0209
75–12 model	10,211	4,852	243	2,000	17,306	0.0169
Baltimore 80 F						
75–4 model	42,885	4,852	243	2,000	49,980	0.0489
75–12 model	31,648	4,852	243	2,000	38,743	0.0379
90 F						
75–4 model	22,465	4,852	243	2,000	29,560	0.0289
75–12 model	16,336	4,852	243	2,000	23,431	0.0229
100 F						
75–4 model	15,316	4,852	243	2,000	22,411	0.0219
75–12 model	11,232	4,852	243	2,000	18,327	0.0179
Cooling Pond						
Boston 80 F	–	4,852	200	–	5,052	0.0049
85 F	–	4,852	200	–	5,052	0.0049
90 F	–	4,852	200	–	5,052	0.0049
Detroit 85 F	–	4,852	200	–	5,052	0.0049
90 F	–	4,852	200	–	5,052	0.0049
95 F	–	4,852	200	–	5,052	0.0049

TABLE IV-7 (Continued)

	Spray Pump,$	Circulation Pumping,$	Makeup,$	Maintenance and Treatment,$	Total,$	Total,$/MWhr
Cooling Pond (Cont'd)						
Baltimore 85 F	–	4,852	200	–	5,052	0.0049
90 F	–	4,852	200	–	5,052	0.0049
95 F	–	4,852	200	–	5,052	0.0049
Spray Cooling-32-atm Cycle						
Boston 80 F						
75–4 model	31,652	4,852	243	2,000	38,747	0.0379
75–12 model	22,464	4,852	243	2,000	29,559	0.0289
90 F						
75–4 model	18,380	4,852	243	2,000	25,475	0.0249
75–12 model	14,295	4,852	243	2,000	21,390	0.0209
100 F						
75–4 model	13,273	4,852	243	2,000	20,368	0.0199
75–12 model	9,190	4,852	243	2,000	16,285	0.0159
Detroit 80 F						
75–4 model	34,718	4,852	243	2,000	41,813	0.0409
75–12 model	24,507	4,852	243	2,000	31,602	0.0309
90 F						
75–4 model	19,400	4,852	243	2,000	26,495	0.0259
75–12 model	15,316	4,852	243	2,000	22,411	0.0219
100 F						
75–4 model	13,273	4,852	243	2,000	20,368	0.0199
75–12 model	9,190	4,852	243	2,000	16,285	0.0159
Baltimore 80 F						
75–4 model	37,780	4,852	243	2,000	44,875	0.0439
75–12 model	28,585	4,852	243	2,000	35,680	0.0349
90 F						
75–4 model	20,423	4,852	243	2,000	27,518	0.0269
75–12 model	15,315	4,852	243	2,000	22,410	0.0219
100 F						
75–4 model	14,295	4,852	243	2,000	21,390	0.0209
75–12 model	10,211	4,852	243	2,000	17,306	0.0169

TABLE IV-7 (Continued)

	Spray Pump,$	Circulation Pumping,$	Makeup,$	Maintenance and Treatment,$	Total,$	Total,$/MWhr
Spray Cooling-62-atm Cycle						
Boston 80 F						
75–4 model	37,748	4,852	243	2,000	44,843	0.0439
75–12 model	26,548	4,852	243	2,000	33,643	0.0329
90 F						
75–4 model	22,464	4,852	243	2,000	29,559	0.0289
75–12 model	17,358	4,852	243	2,000	24,453	0.0239
100 F						
75–4 model	16,336	4,852	243	2,000	23,431	0.0229
75–12 model	11,232	4,852	243	2,000	18,327	0.0179
Detroit 80 F						
75–4 model	42,887	4,852	243	2,000	49,982	0.0489
75–12 model	30,633	4,852	243	2,000	37,728	0.0369
90 F						
75–4 model	23,484	4,852	243	2,000	30,579	0.0299
75–12 model	18,379	4,852	243	2,000	25,474	0.0249
100 F						
75–4 model	16,336	4,852	243	2,000	23,431	0.0229
75–12 model	11,232	4,852	243	2,000	18,327	0.0179
Baltimore 80 F						
75–4 model	46,969	4,852	243	2,000	54,064	0.0529
75–12 model	34,711	4,852	243	2,000	41,806	0.0409
90 F						
75–4 model	24,507	4,852	243	2,000	31,602	0.0309
75–12 model	18,378	4,852	243	2,000	25,473	0.0249
100 F						
75–4 model	17,358	4,852	243	2,000	24,453	0.0239
75–12 model	13,478	4,852	243	2,000	20,573	0.0201

TABLE IV-8

SLIP VELOCITIES FOR BUBBLES AND SLUGS IN WATER
(From Ref. IV-37 and IV-38)

Description	Formula	Limits on Applicability	Application to CAPS (12-ft dia pipe)
Spherical bubble	$U_{slip} = \frac{ga^2}{9\nu \text{ water}}$	Bubbles are spherical only for $a \leq 6 \times 10^{-4}$m (approximately)	U_{slip} = 0 - 0.44 m/sec - applies to bubbles only when they first come out of the solution
Continuous bubble "swarm"	$U_{slip} = 1.41\left[\frac{\sigma g \Delta\rho}{\rho_L^2}\right]^{\frac{1}{4}}$	May be applicable to transition region from small spherical bubbles to spherical cap	U_{slip} = 0.23 m/sec for transition from spherical bubbles to spherical cap
Spherical cap bubble in infinite fluid	$U_{slip} = 2/3(gR)^{\frac{1}{2}}$	When the wall effects become noticeable, this formula is not applicable. Cap becomes slug at about 0.75D across nose	U_{slip} = 0 - 4.2 m/sec - does not include wall effects for large bubbles
Slug flow	$U_{slip} = 0.35\sqrt{gD}$	Applicable when bubbles form a slug - air volume equals $(.75)^2$ = .56 of water volume across shaft, or greater	U_{slip} = 2.095 m/sec - applies to top part of vertical shaft

(a = bubble radius, R = radius of curvature of bubble cap, D = shaft diameter)

TABLE IV-9

WATER VELOCITY AND DENSITY RATIOS IN CONSTANT DIAMETER SHAFT
STEADY STATE ANALYSIS

Shaft Diameter = 12 ft
U-Bend Depth = 328 ft below cavern floor

Cavern Pressure	66.3	66.3	66.3	66.3	50.0	50.0	50.0	50.0
Bottom Water Velocity, ft/sec	1.64	1.64	16.4	16.4	1.64	1.64	16.4	16.4
Slip Velocity, ft/sec	6.87	0	6.87	0	6.87	0	6.87	0
Water Velocity ratio - top/bottom	1.7	3.4	3.1	3.4	1.5	2.8	2.6	2.8
Density ratio - mixture average/water	0.978	0.898	0.931	0.910	0.981	0.910	0.938	0.918

TABLE IV-10

VELOCITY AND DENSITY RATIOS FOR FLARED SHAFT
STEADY STATE ANALYSIS

Shaft Diameter at Bottom = 12 ft
Cavern Pressure = 66.3 atm
U-Bend Depth = 328 ft below cavern floor

Design Water Velocity, ft/sec	1.64		1.64		16.4		16.4	
Slip Velocity, ft/sec	6.87		0		6.87		0	
Area Ratio - top/bottom	1.46		3.41		2.70		3.40	
Shaft Volume Ratio - flared/straight	1.03		1.20		1.12		1.20	
Bottom Water Velocity, ft/sec	1.64	16.4	1.64	16.4	16.4	1.64	16.4	1.64
Water Velocity ratio - top/bottom	1.00	2.05	1.00	1.00	1.00	0.46	1.00	1.00
Density Ratio - Mixture average/water	0.978	0.921	0.854	0.855	0.905	0.980	0.855	0.854

TABLE IV-11

U-BEND DEPTH FOR CONSTANT DIAMETER SHAFT WITH
ASSUMED INITIAL WATER VELOCITY

Shaft Diameter = 12 ft

Cavern Pressure, atm	66.3	66.3	66.3	66.3	50.0	50.0	50.0	50.0
Initial Bottom Velocity, ft/sec	1.64	1.64	16.4	16.4	1.64	1.64	16.4	16.4
Slip Velocity, ft/sec	6.87	0	6.87	0	6.87	0	6.87	0
Depth Where Bubbles First Appear, ft*	62	270	111	180	44	188	84	131
Additional Depth due to Fluid Inertia, ft	14	15	148	150	12	13	130	132
Total U-Bend Depth, ft*	76	285	259	330	56	201	214	263
U-Bend Depth, % of Depth to Cavern Floor	3.4	12.8	11.6	14.9	3.4	12.1	12.8	15.8

---

* Depth below cavern floor

TABLE IV-12

U-BEND DEPTH FOR FLARED SHAFT WITH ASSUMED INITIAL WATER VELOCITY

Shaft Diameter at Bottom = 12 ft
Cavern Pressure = 66.3 atm

Design Velocity, ft/sec	1.64	1.64	1.64	1.64	16.4	16.4	16.4	16.4
Bottom Velocity, ft/sec	1.64	1.64	16.4	16.4	1.64	1.64	16.4	16.4
Slip Velocity, ft/sec	6.87	0	6.87	0	6.87	0	6.87	16.4
Depth Where Bubbles First Appear, ft*	56	271	132	234	44	271	138	234
Additional Depth due to Fluid Inertia	14	14	148	150	14	14	148	150
Total U-Bend Depth, ft*	70	285	280	384	58	285	286	384
U-Bend Depth, % of Depth to Cavern Floor	3.2	12.8	12.6	17.3	2.6	12.8	12.9	17.3

* Depth below cavern floor

TABLE IV-13

ASSUMED SYSTEM PARAMETERS
FOR SIMPLIFIED DYNAMIC ANALYSIS

Cavern Pressure, P	66 atm
Cavern Depth, H_c	2240 ft
U-bend Depth, d	328 ft
Total Shaft Length, $\ell = H_c + 2d$	2896 ft
Shaft Diameter, D	12 ft
Cavern Volume, V	500,000 yd^3
Normal Charging Rate, Q	187.5 ft^3/sec
Bouyant Head with Zero Slip, H_b (10% of depth)	224 ft
Water Velocity at Bottom of Shaft, U_o	1.64 ft/sec
Friction Factor, $f = \dfrac{H_f}{\frac{\ell}{D} \cdot \frac{U^2}{2g}}$	.0180

where: H_f = ft of water

U = fluid velocity - ft/sec

g = acceleration due to gravity - ft/sec^2

TABLE IV-14

EPA AIR POLLUTION REGULATIONS FOR NEW OR MODIFIED EXISTING STEAM POWER PLANTS ABOVE 250 X 10^6 BTU/HR

July 1, 1975

Pollutant	Emission Level - lb/10^6 Btu		
	Coal	Oil	Gas
SO_2	1.2	0.8	-
NO_x	0.7	0.3	0.2
Particulates	0.1	0.1	0.1

TABLE IV-15

SUMMARY OF REGULATIONS FOR SAN DIEGO AIR POLLUTION CONTROL DISTRICT

Pollutant		Emission Level
Particulates		0.3 grain/scf @ 12% CO_2 Ringelmann No. 2
SO_2		500 ppm
NO_x (1)	Existing(2) Gas-fired	225 ppm
	Oil-or Coal-fired	325
	New (construction after 7/1/71) - Gas	125
	Oil- or Coal-fired	225

(1) The NO_x regulation is only for "nonmobile, fuel burning article machine, equipment or contrivence", having a maximum heat input rating of 50×10^6 Btu/hr. The NO_x concentration is based upon NO_2 at 3% excess O_2. This applies to gas turbine power generation equipment.

(2) Existing equipment must comply with the same regulations as new equipment after 12/31/73.

TABLE IV-16

SUGGESTED EPA AIR POLLUTION REGULATIONS FOR GAS TURBINE-BASED POWER SYSTEMS

Pollutant	Emission Level		
	Coal[1]	Oil[2]	Gas[2]
SO_2	1.2 lb/10^6 Btu	55 ppm (.3 lb/10^6 Btu)	55 ppm (.3 lb/10^6 Btu)
NO_x	0.7 lb/10^6 Btu	75 ppm	55 ppm
Particulates	0.1 lb/10^6 Btu	Ringelmann 0.5	Ringelmann 0.5
CO		90[3]-215[4] ppm (0.2-0.5 lb/10^6 Btu)	90[3]-215[4] ppm (0.2-0.5 lb/10^6 Btu)

(1) For coal fired plants it has been assumed that emission levels would be the same as for coal fired steam plants.

(2) Based upon 15 percent O_2 in exhausts.

(3) 50 x 10^6 Btu/hr or more.

(4) Less than 50 x 10^6 Btu/hr.

TABLE IV-17

FACTORS AFFECTING NO FORMATION FOR
VARIOUS SYSTEM STORAGE PRESSURES

		High Pressure Burner		Low Pressure Burner	
Storage Pressure (atm)	Air Temp (R)	F/A	Exit Temp (R)	Oxidizer Temp (R)	F/A At Exit
24.2	1099	.00255	1279	1214	.0224
30.1	1100	.00344	1342	1214	.0234
48.6	1102	.00554	1487	1214	.0256
60.2	1103	.00653	1554	1214	.0266
78.7	1105	.00782	1641	1214	.0280
98.3	1106	.00894	1714	1214	.0292

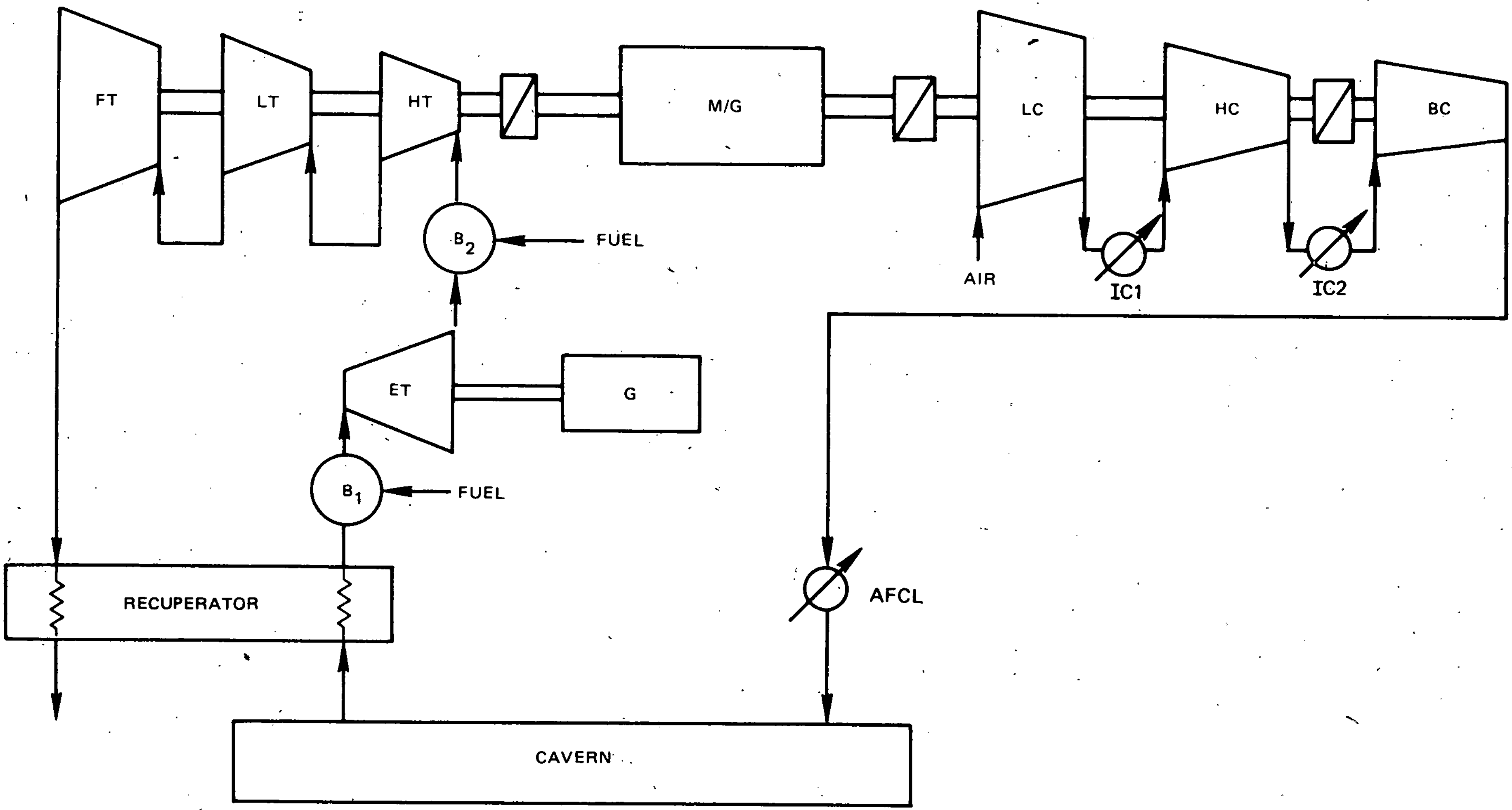
TABLE IV-18

SOUND PRESSURE LEVELS FOR A GAS TURBINE PLANT
DECIBELS

Sound Pressure Level *		Octave Band Center Frequency, Hz							
		63	125	250	500	1000	2000	4000	8000
Standard Industrial Sound Treatment									
	1 Unit	87	79	72	68	64	62	58	54
	2 Units	90	82	75	71	67	65	61	57
	3 Units	92	84	77	73	69	67	63	59
	4 Units	93	85	78	74	70	68	64	60
Standard Residential Sound Treatment									
	1 Unit	79	70	63	58	54	51	48	45
	2 Units	82	73	66	61	57	54	51	48
	3 Units	84	75	68	63	59	56	53	50
	4 Units	85	76	69	64	60	57	54	51
Standard Maximum S Treatment									
	1 Unit	71	60	53	48	44	41	38	36
	2 Units	74	63	56	51	47	44	41	40
	3 Units	76	65	58	53	49	46	43	42
	4 Units	77	66	59	54	50	47	44	43

* at 400'
reference 20 x 10^{-6} N/m^2

REFERENCE CAPS CONFIGURATION



76–12–18–6

FIG. IV–1

CAPS LOW–PRESSURE TURBOMACHINERY

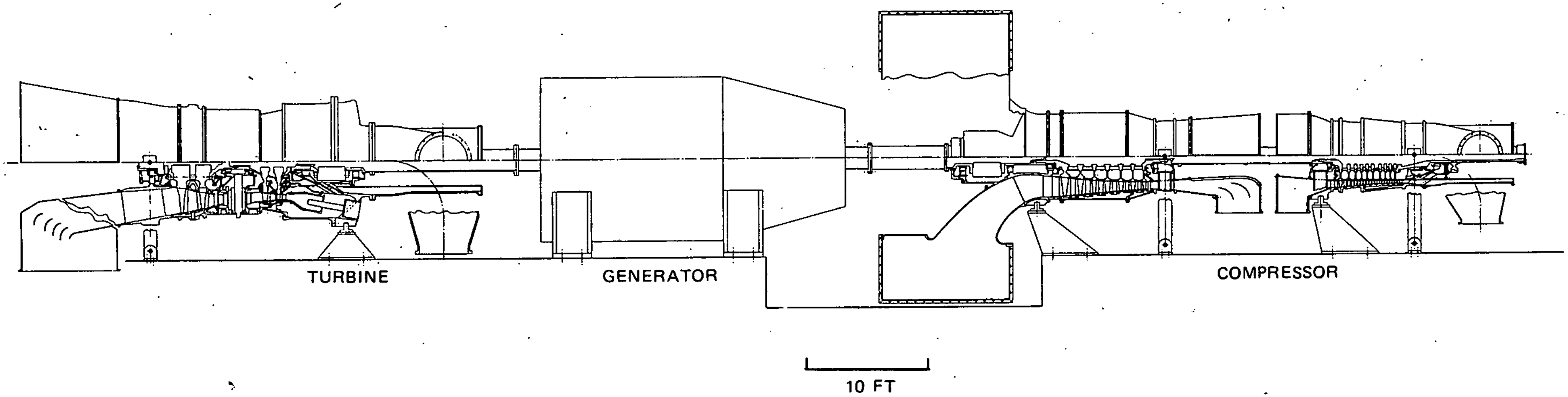


FIG. IV–2

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CAPS LOW-PRESSURE COMPRESSOR

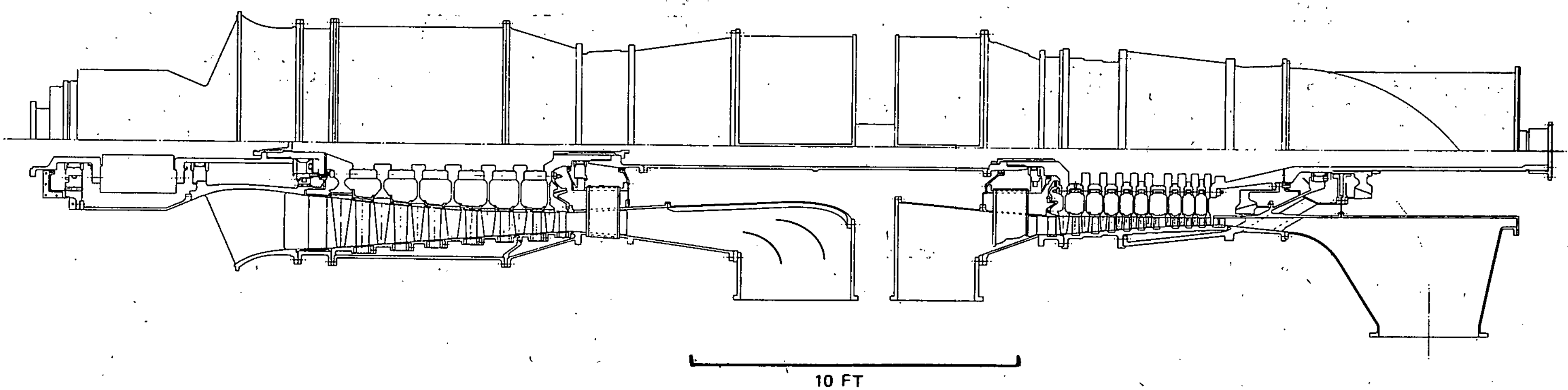


FIG. IV-3

76-04-106-19

CAPS LOW–PRESSURE TURBINE

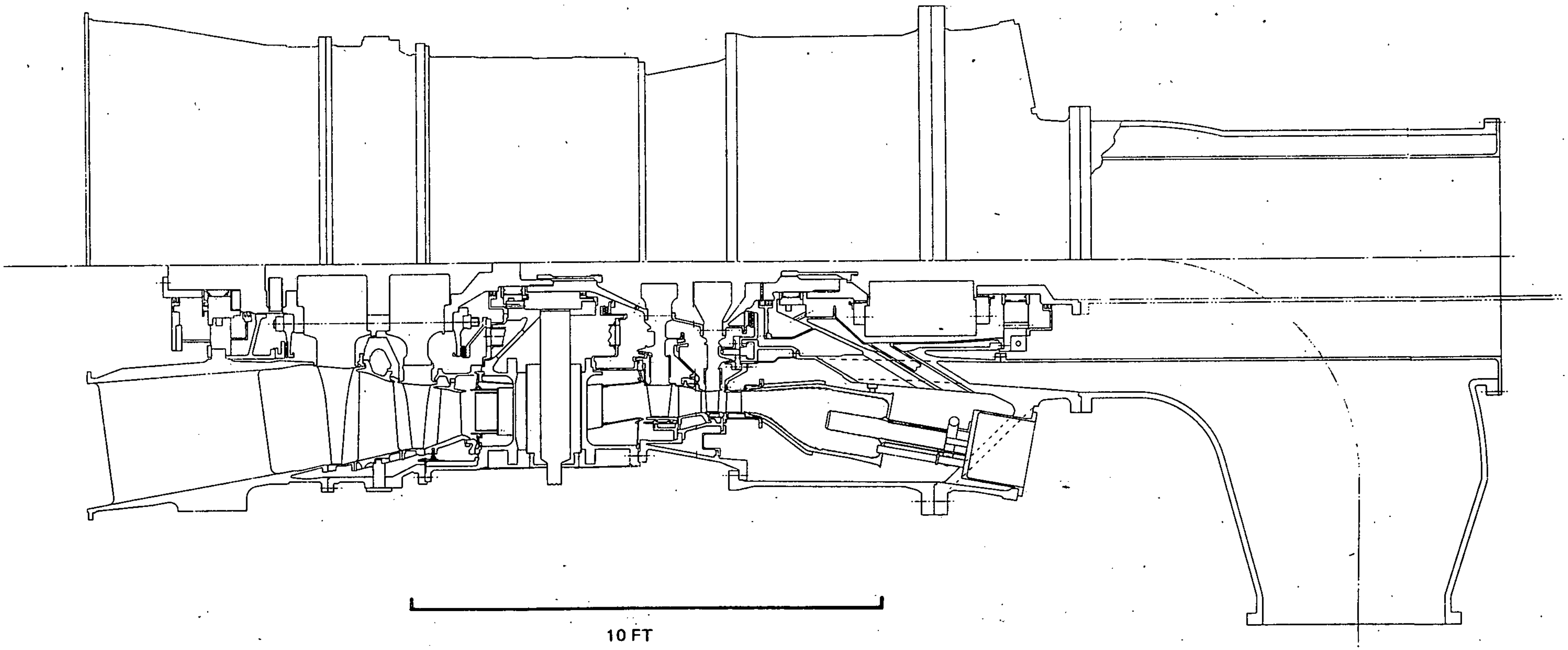
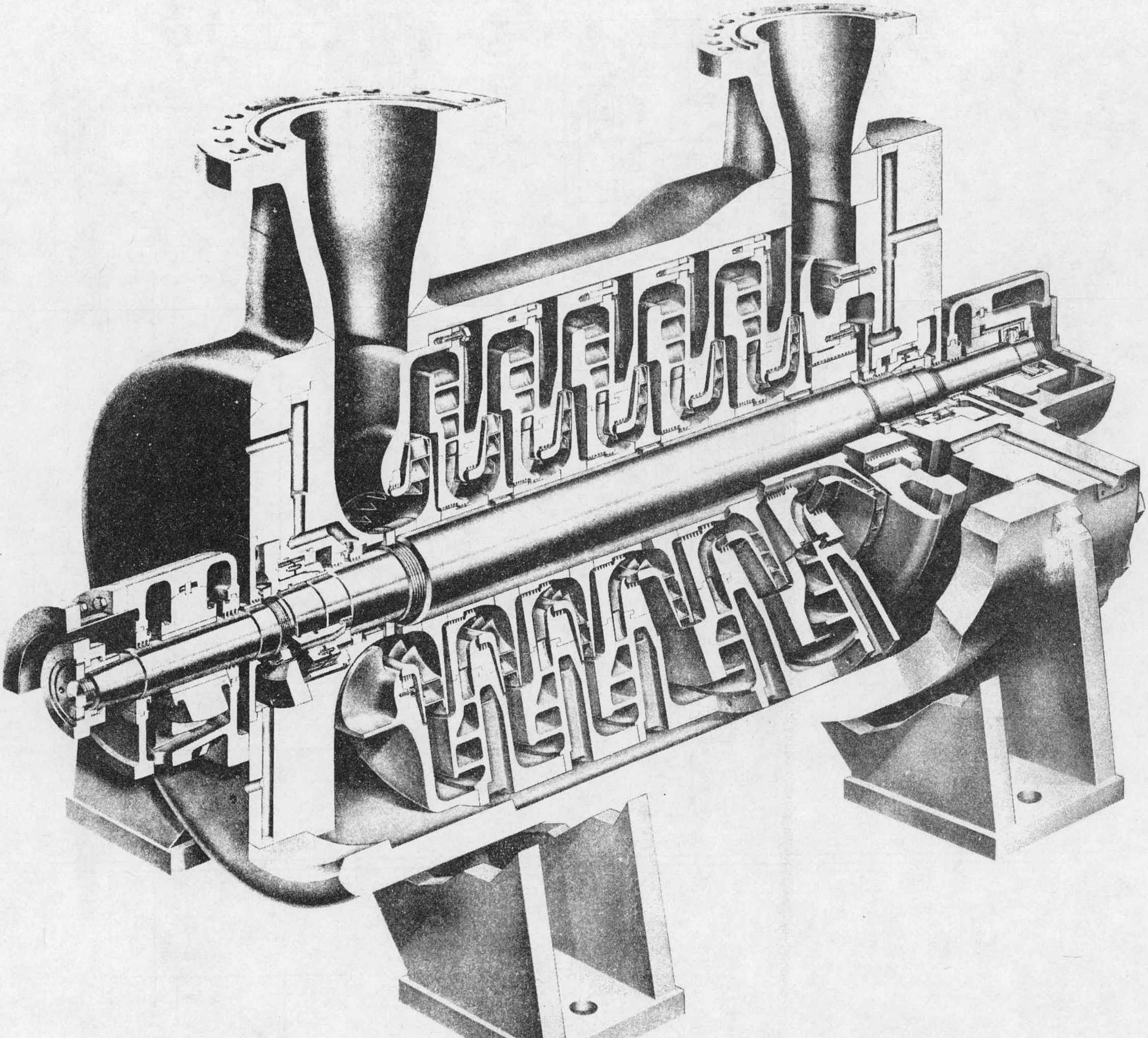


FIG. IV-4

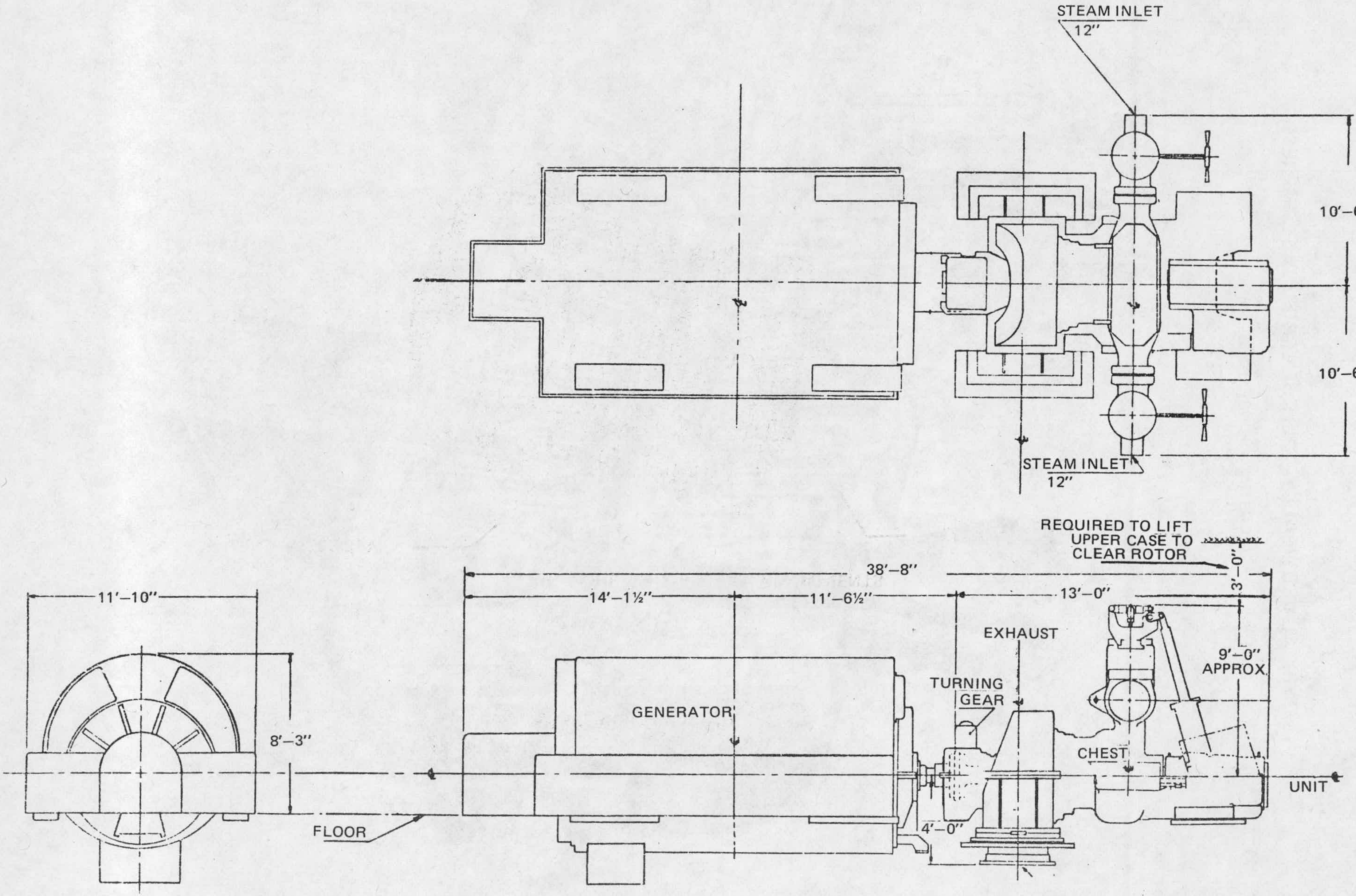
TYPICAL VERTICALLY–SPLIT CENTRIFUGAL COMPRESSOR

COURTESY OF INGERSOLL – RAND



A TYPICAL EXPANDER DESIGN

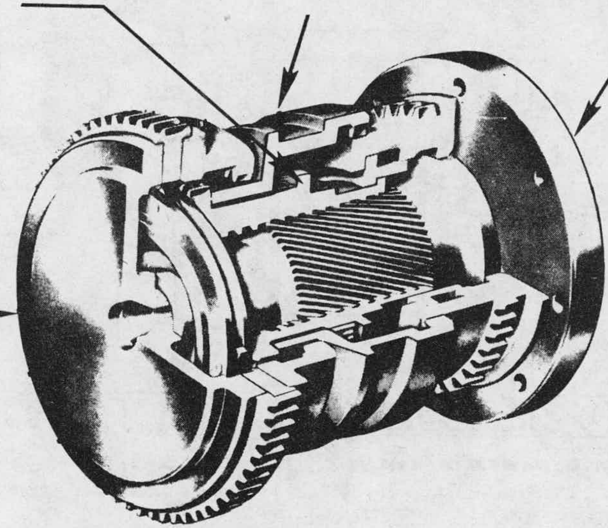
COURTESY OF DELAVAL



R76–952161–5

FIG. IV–6

SSS CAPS CLUTCH



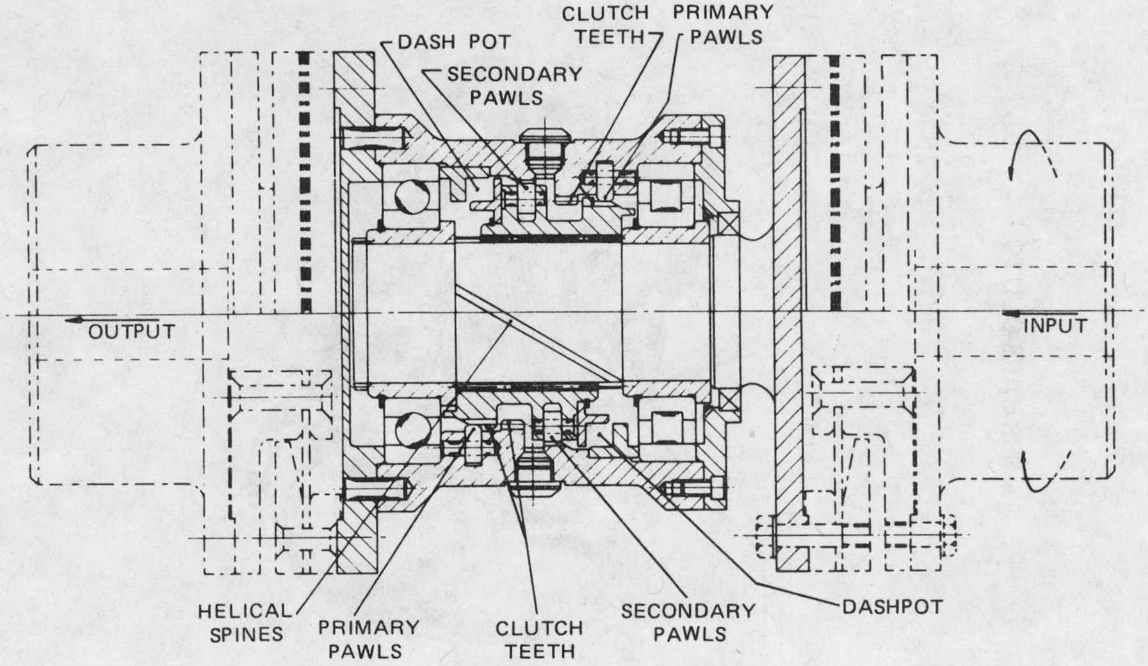
76–02–179–3

FIG. IV–7

BASIC ELEMENTS OF SSS CAPS CLUTCH

(FROM REF. IV–17)

TOP HALF ASSEMBLED FOR ANTI–CLOCKWISE ROTATION OF INPUT SHAFT

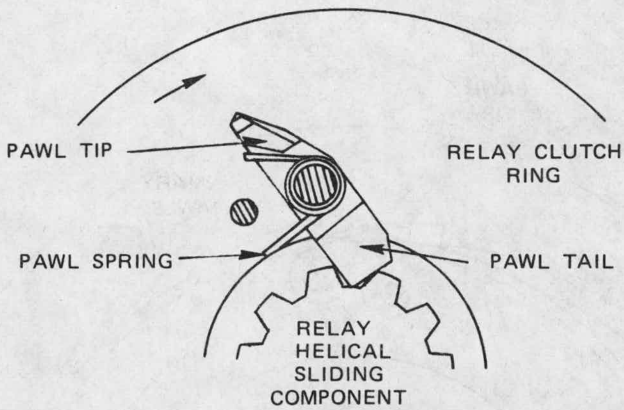


LOWER HALF ASSEMBLED FOR CLOCKWISE ROTATION OF INPUT SHAFT

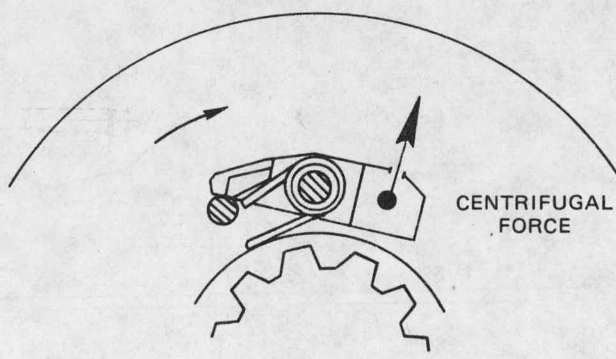
PRIMARY PAWL ACTION

(FROM REF. IV–17)

MOTOR/GENERATOR ROTATION < 600 RPM

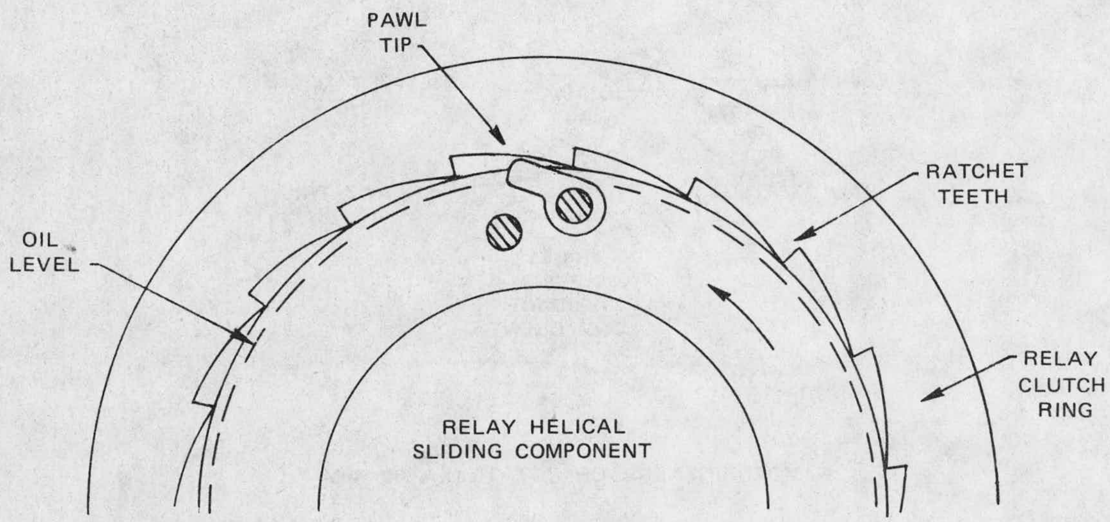


MOTOR/GENERATOR ROTATION > 600 RPM



SECONDARY PAWL ACTION

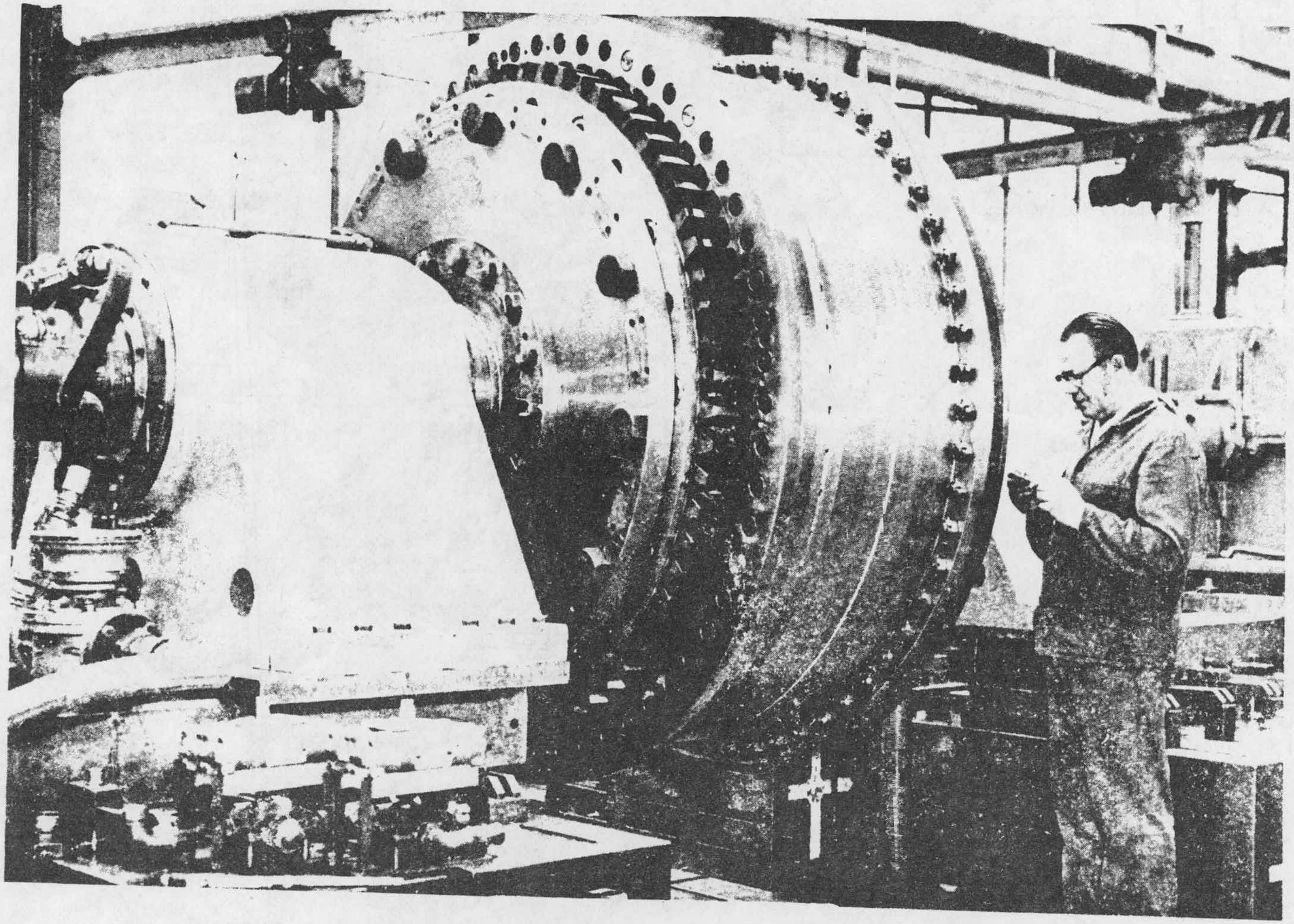
(FROM REF. IV–17)



SSS CLUTCH

340,000 HORSEPOWER

600 RPM



76–02–179–4

FIG. IV–11

MAAG CLUTCH

(FROM REF. IV–19)

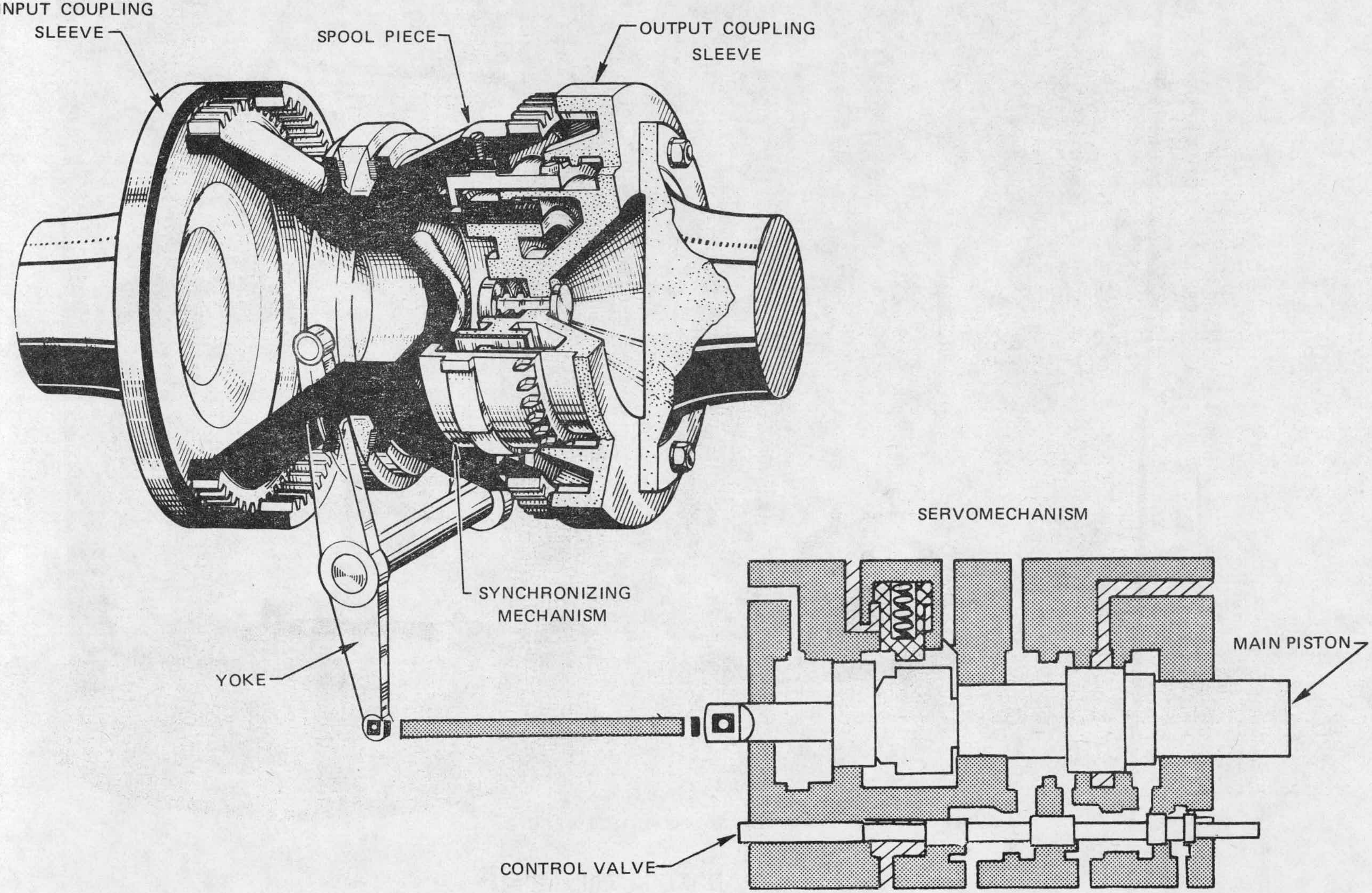
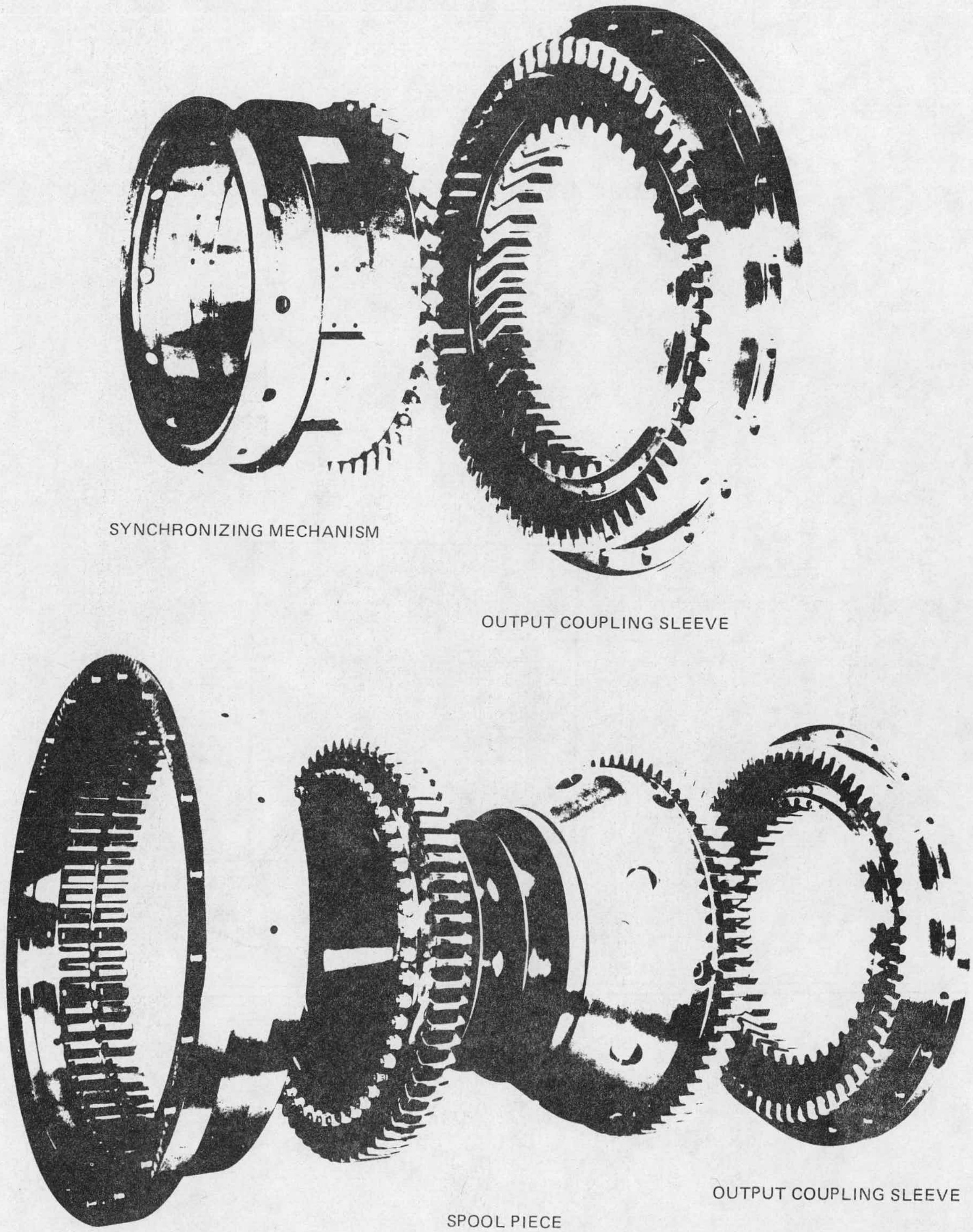


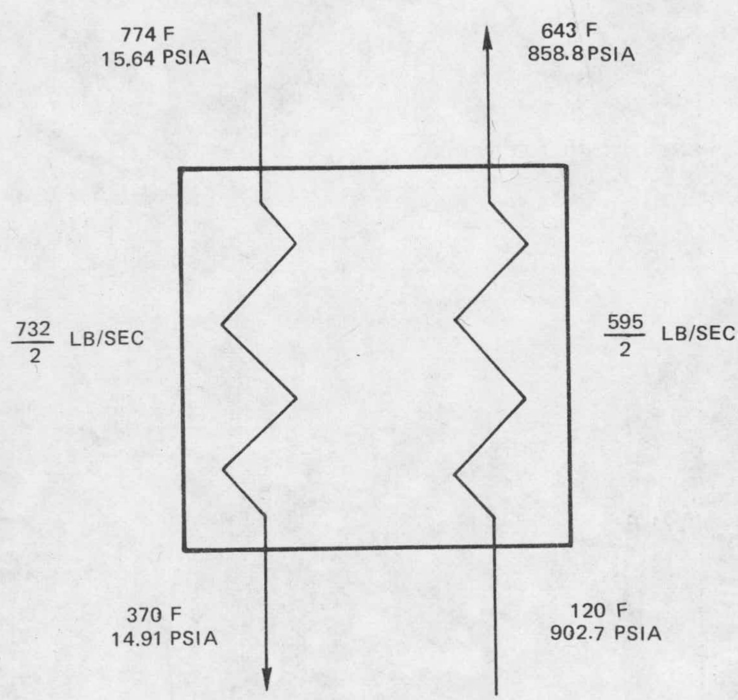
FIG. IV–12

BASIC COMPONENTS OF MAAG CLUTCH

(FROM REF. IV–19)

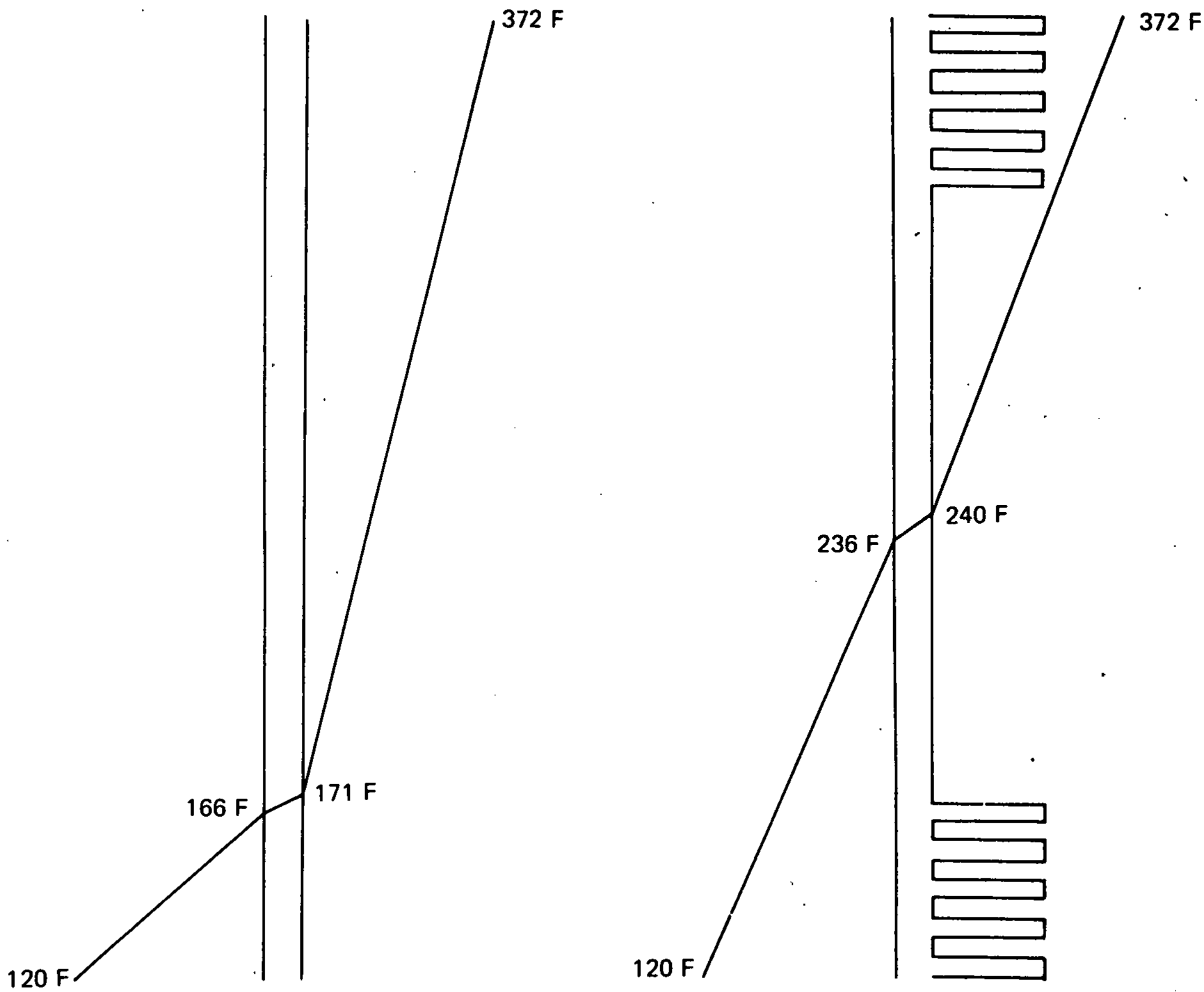


REFERENCE CAPS RECUPERATOR DESIGN CONDITIONS



TYPICAL TEMPERATURE PROFILES

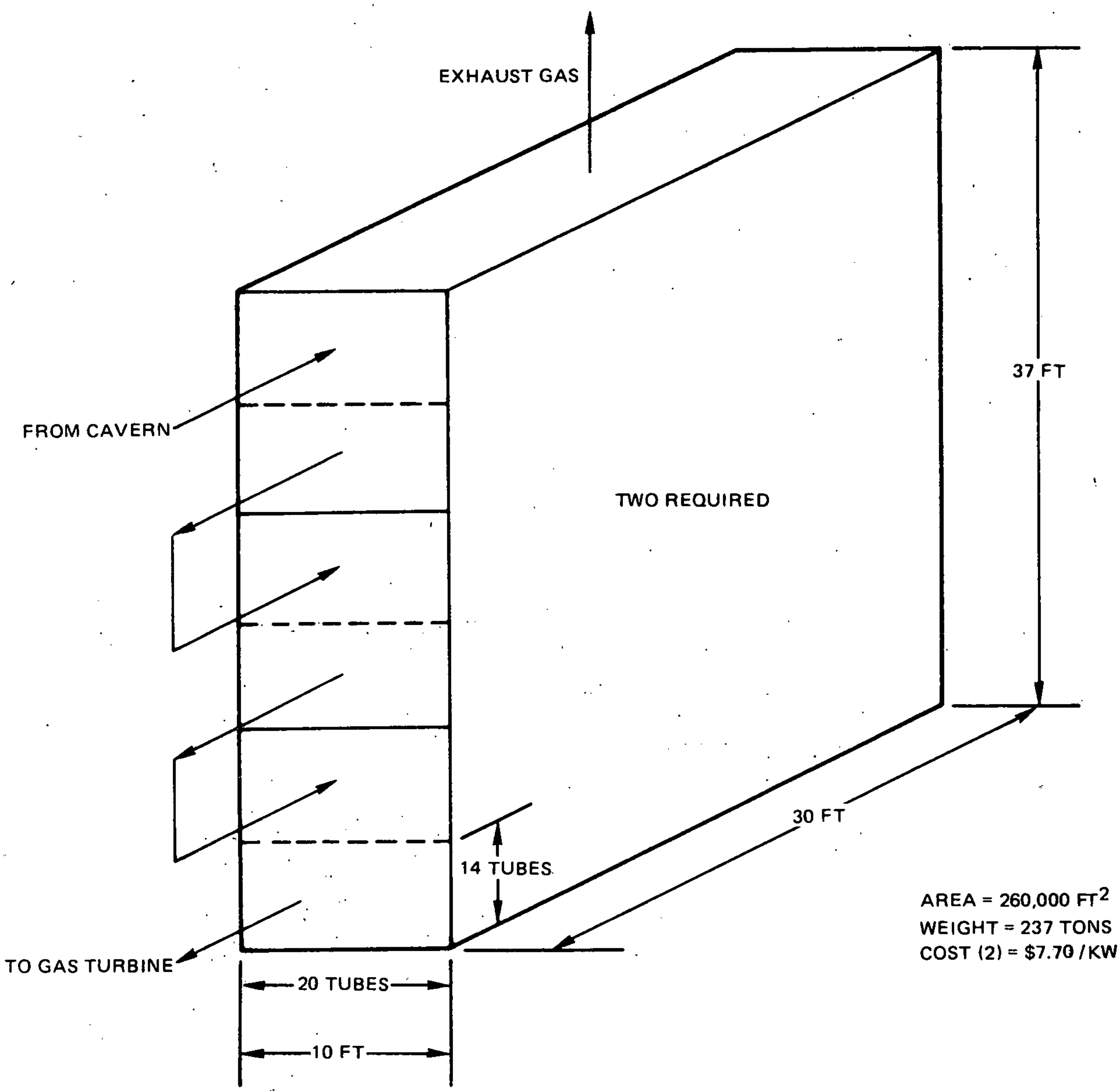
HOT SIDE EXHAUST



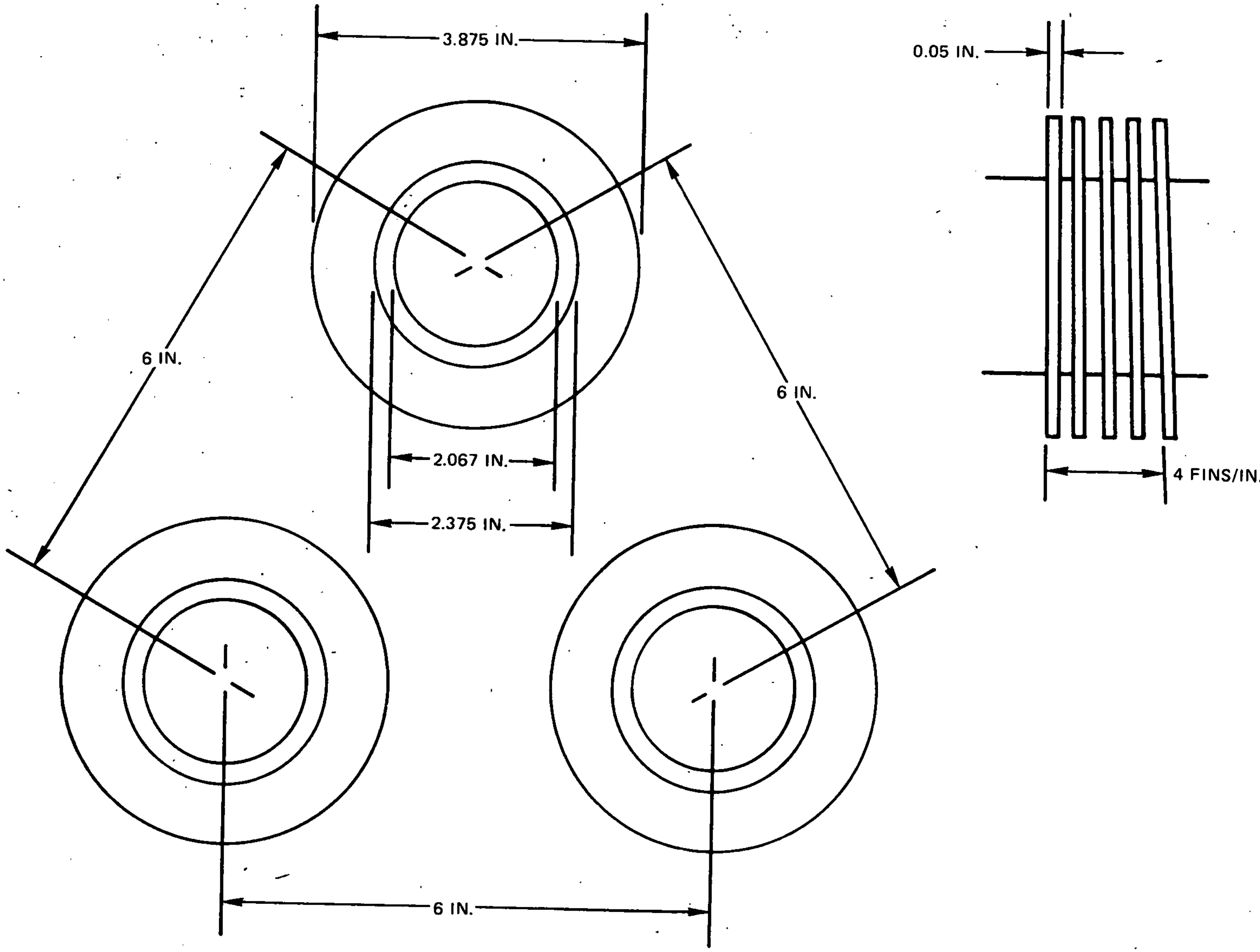
BARE TUBE

FINNED TUBE

RECUPERATOR CORE



RECUPERATOR CROSS-SECTION

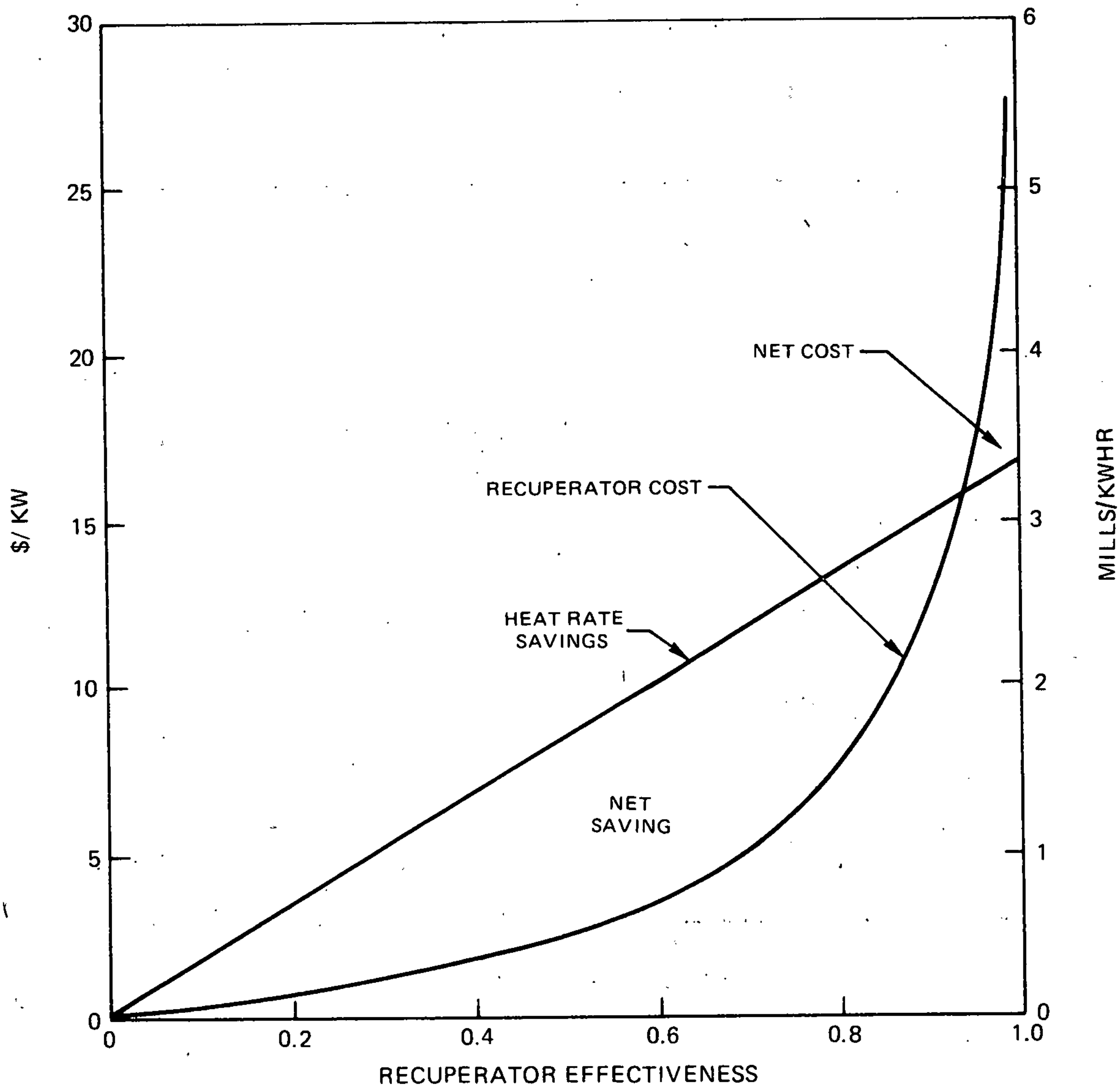


76-11-125-5

FIG. IV-17

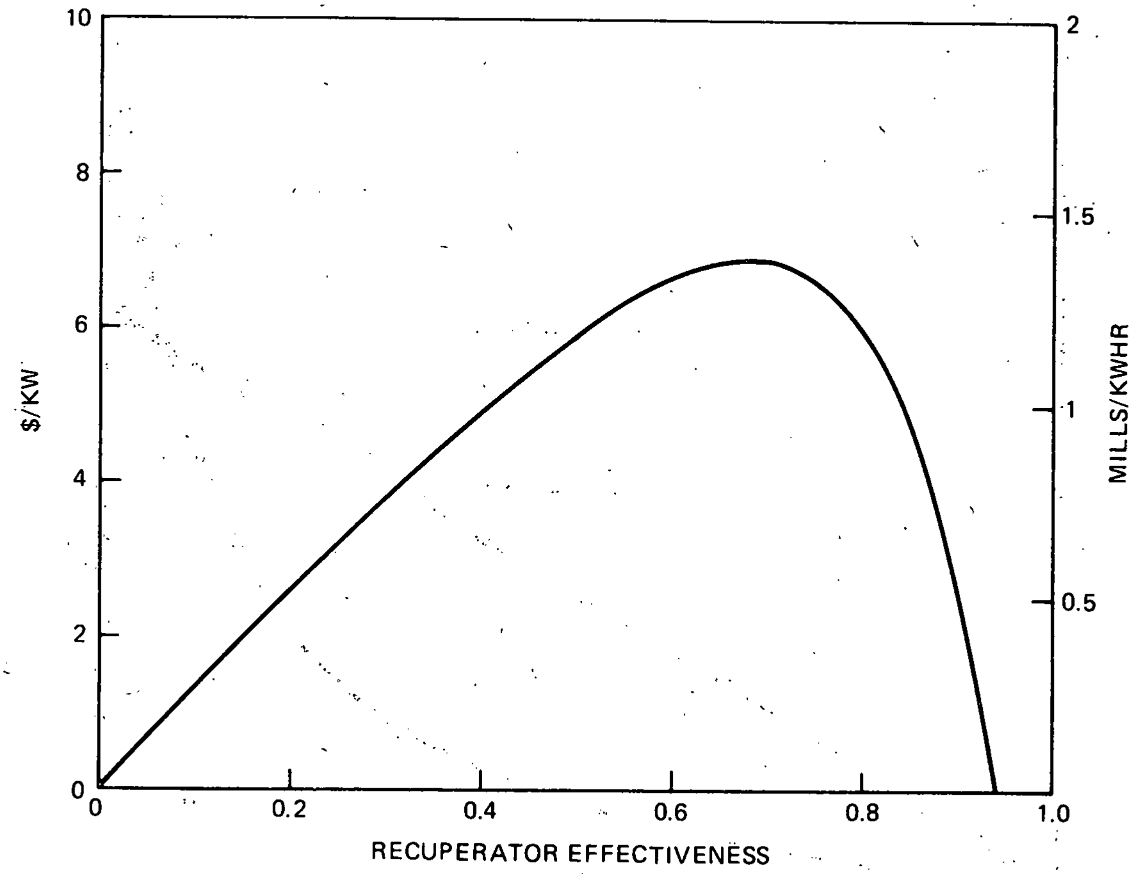
RECUPERATOR EFFECTIVENESS TRADE–OFF

AFC = 18%
CF = 1560/8760
ICF = 75%
FUEL COST = $2.50/10^6 BTU



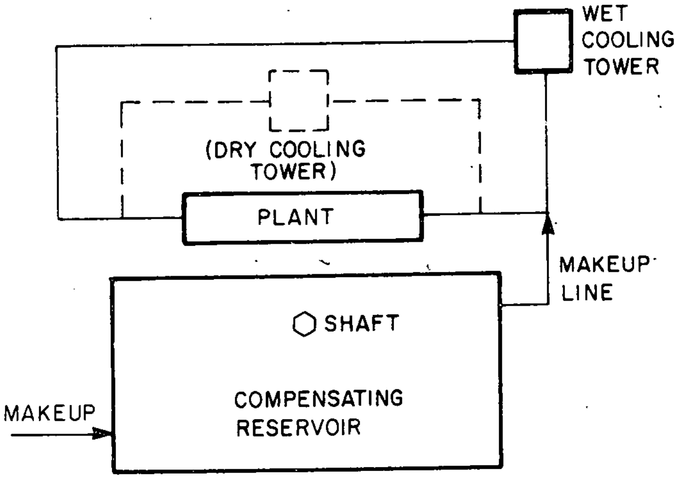
RECUPERATOR SAVINGS

AFC = 18%
CF = 1560/8760
ICF = 75%
FUEL COST = $2.50/10^6 BTU

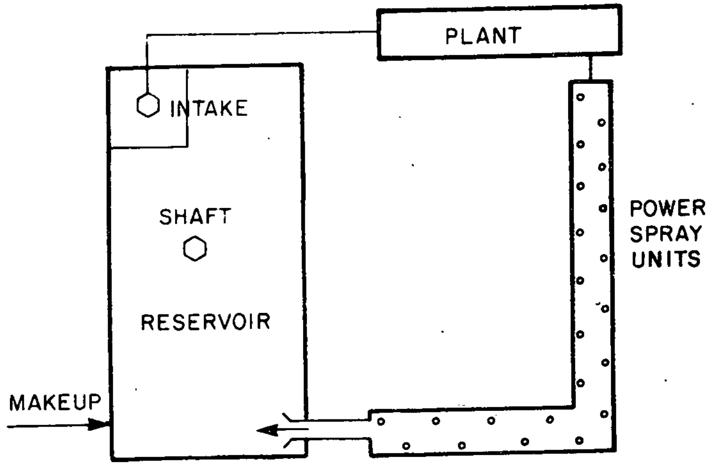


R76-952161-5

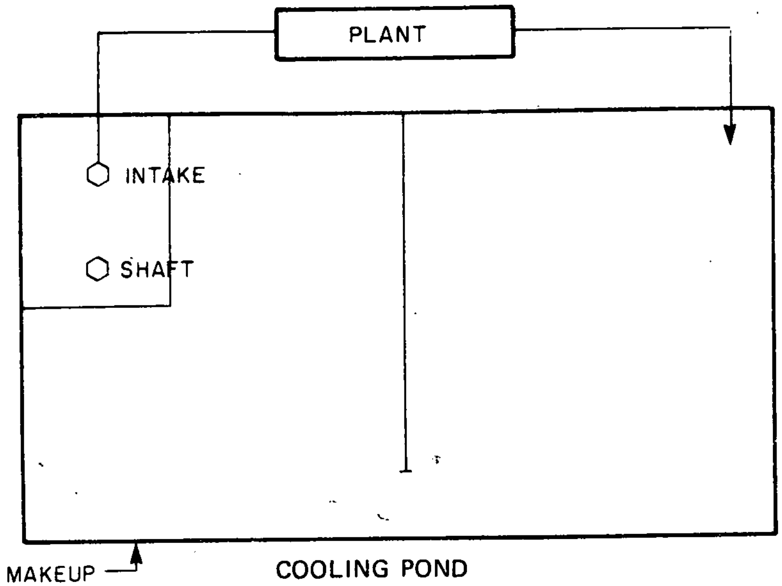
SCHEMATIC DIAGRAMS FOR ALTERNATIVE COOLING SYSTEMS



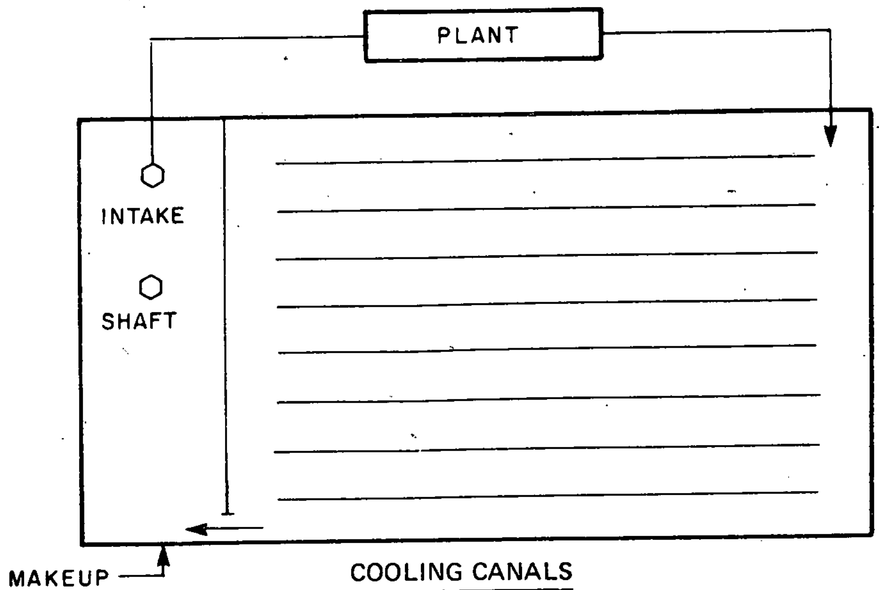
COOLING TOWER SYSTEM



SPRAY COOLING CANAL



COOLING POND



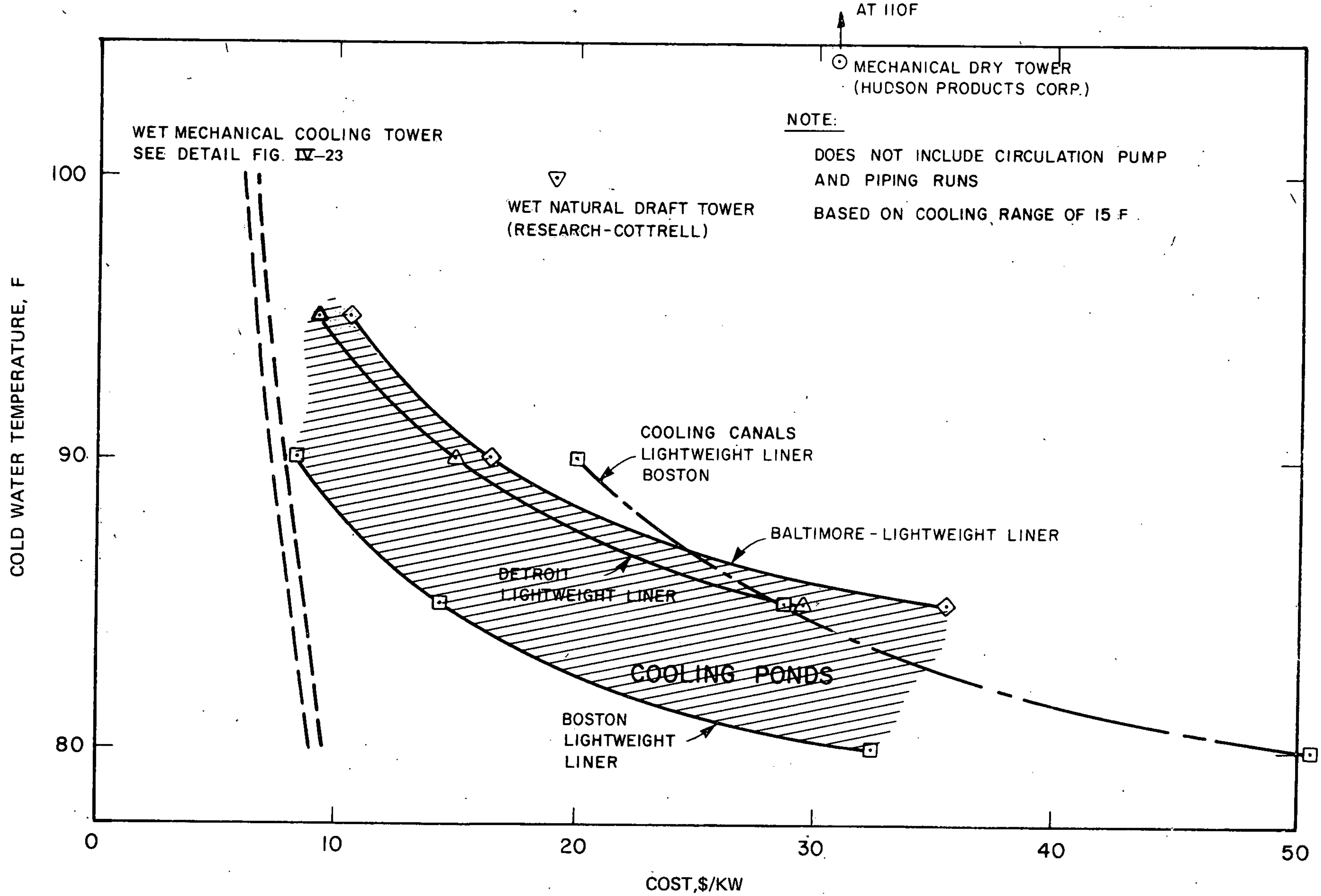
COOLING CANALS

76-05-93-24

FIG. IV-20

R76-952161-5

INSTALLED COST FOR COMBINED COMPENSATING SURFACE RESERVOIR AND COOLING SYSTEM (PART A)

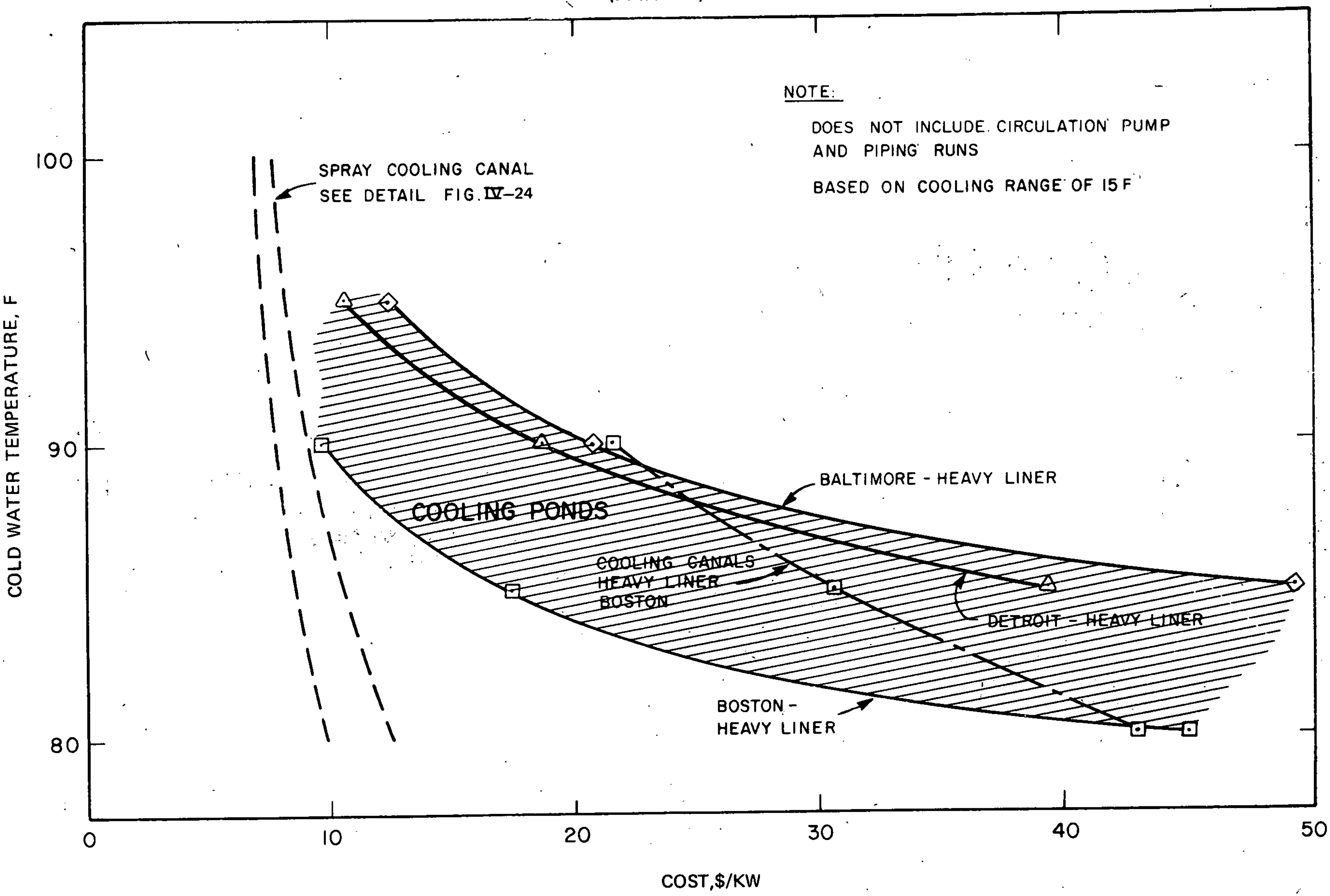


76-05-93-21

FIG. IV-21

R76-952161-5

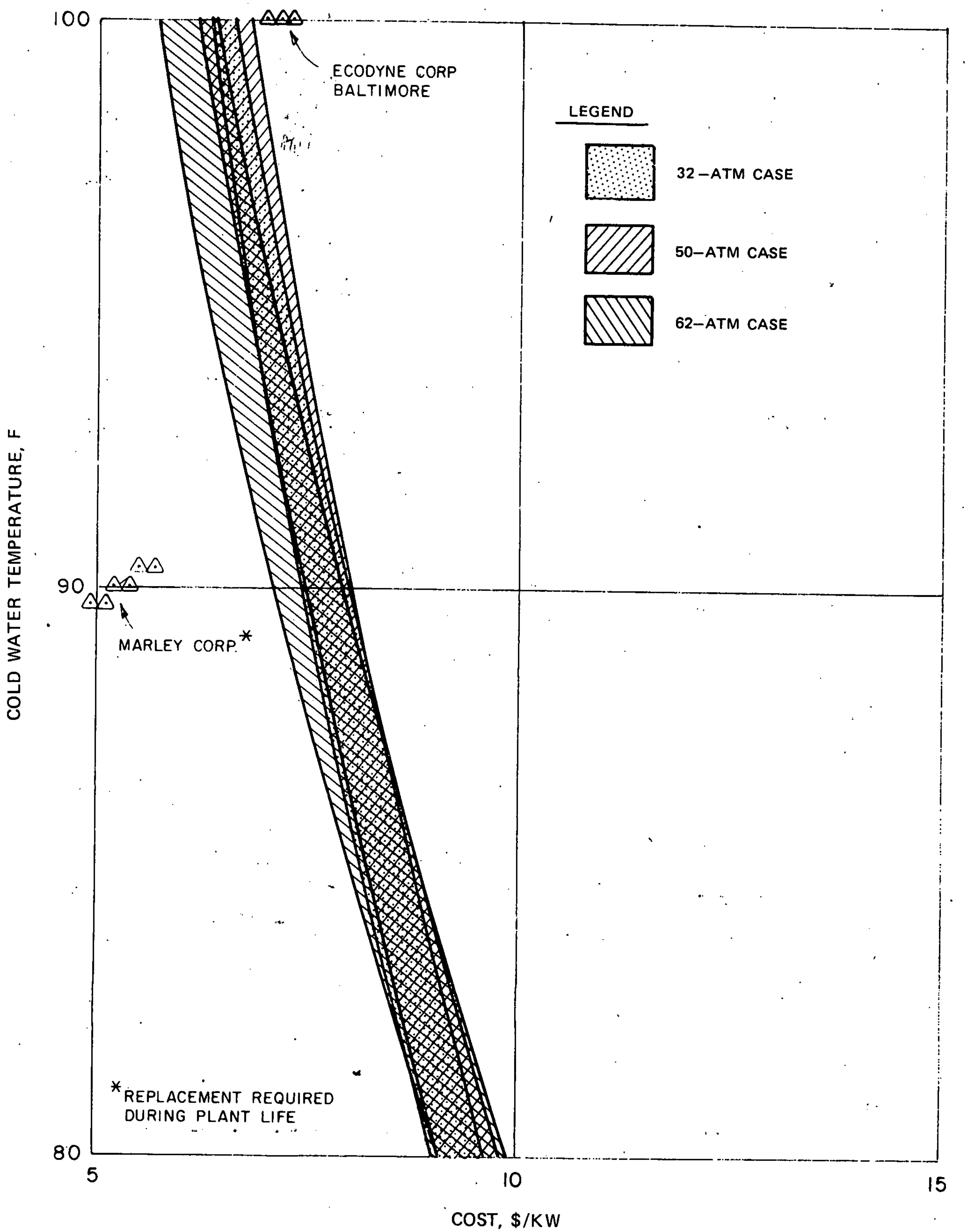
INSTALLED COST FOR COMBINED COMPENSATING SURFACE RESERVOIR AND COOLING SYSTEM (PART B)



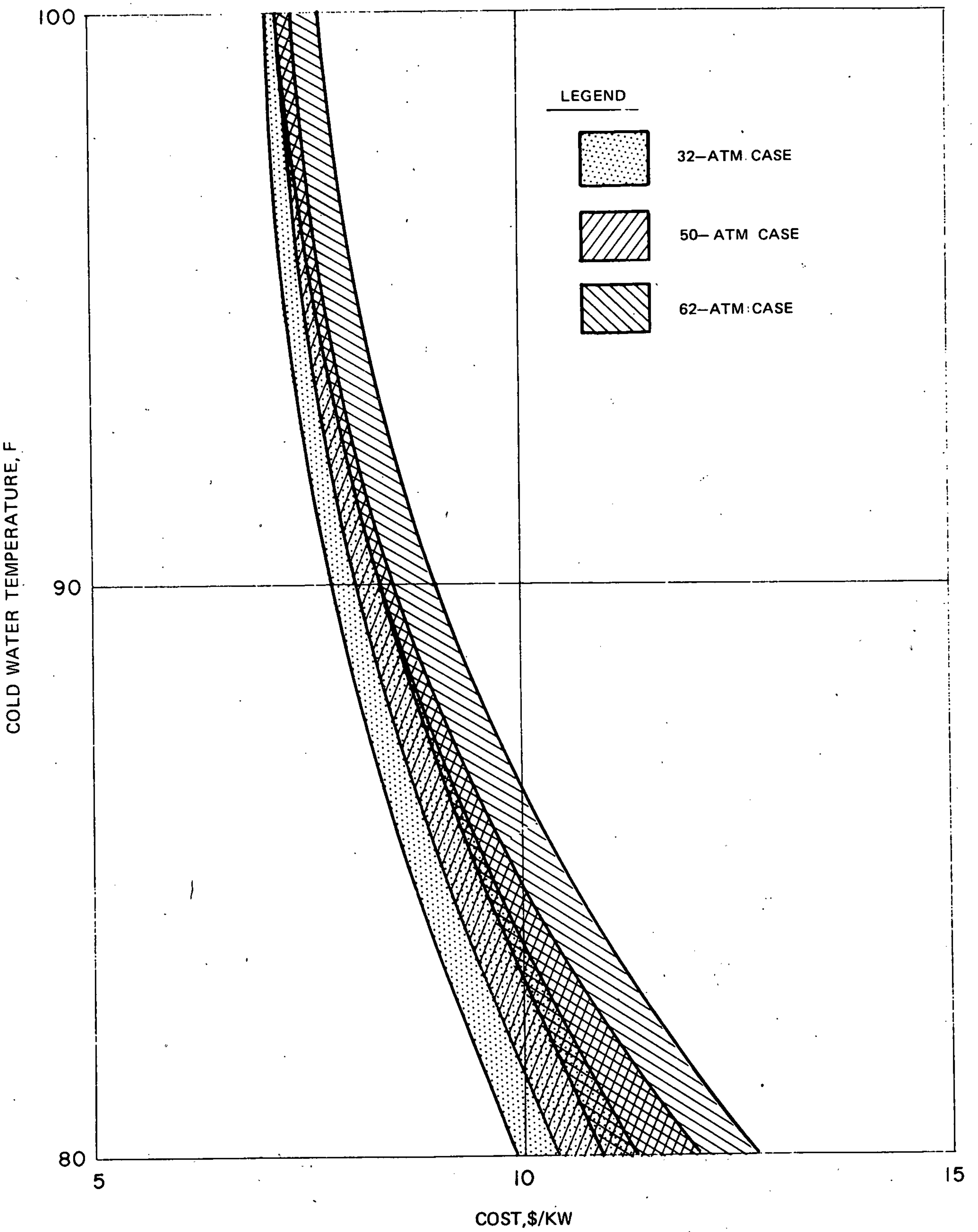
76-05-93-19

FIG. IV-22

INSTALLED COST FOR WET MECHANICAL COOLING TOWER AND SURFACE RESERVOIR



INSTALLED COST FOR SPRAY COOLING CANAL AND SURFACE RESERVOIR



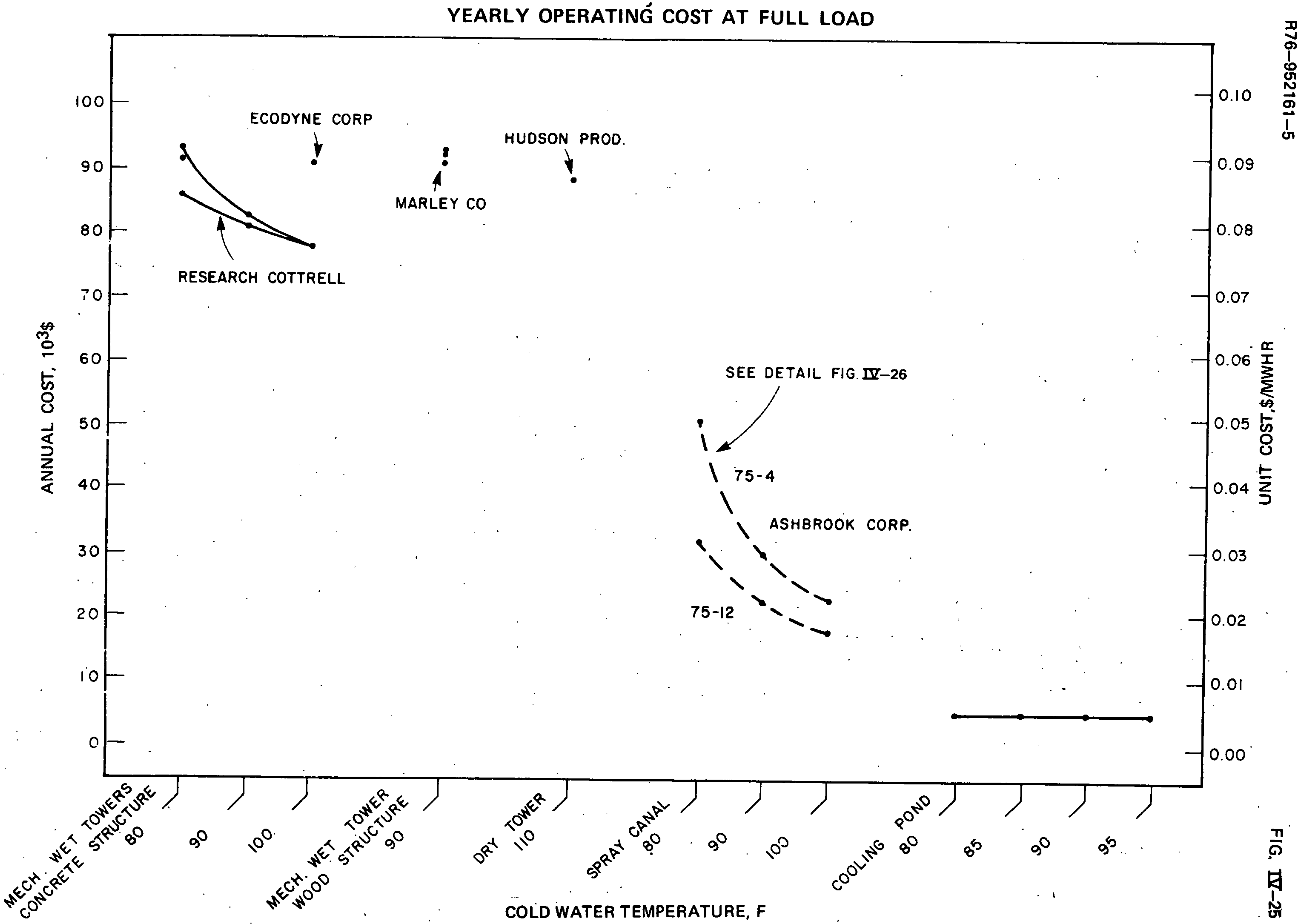
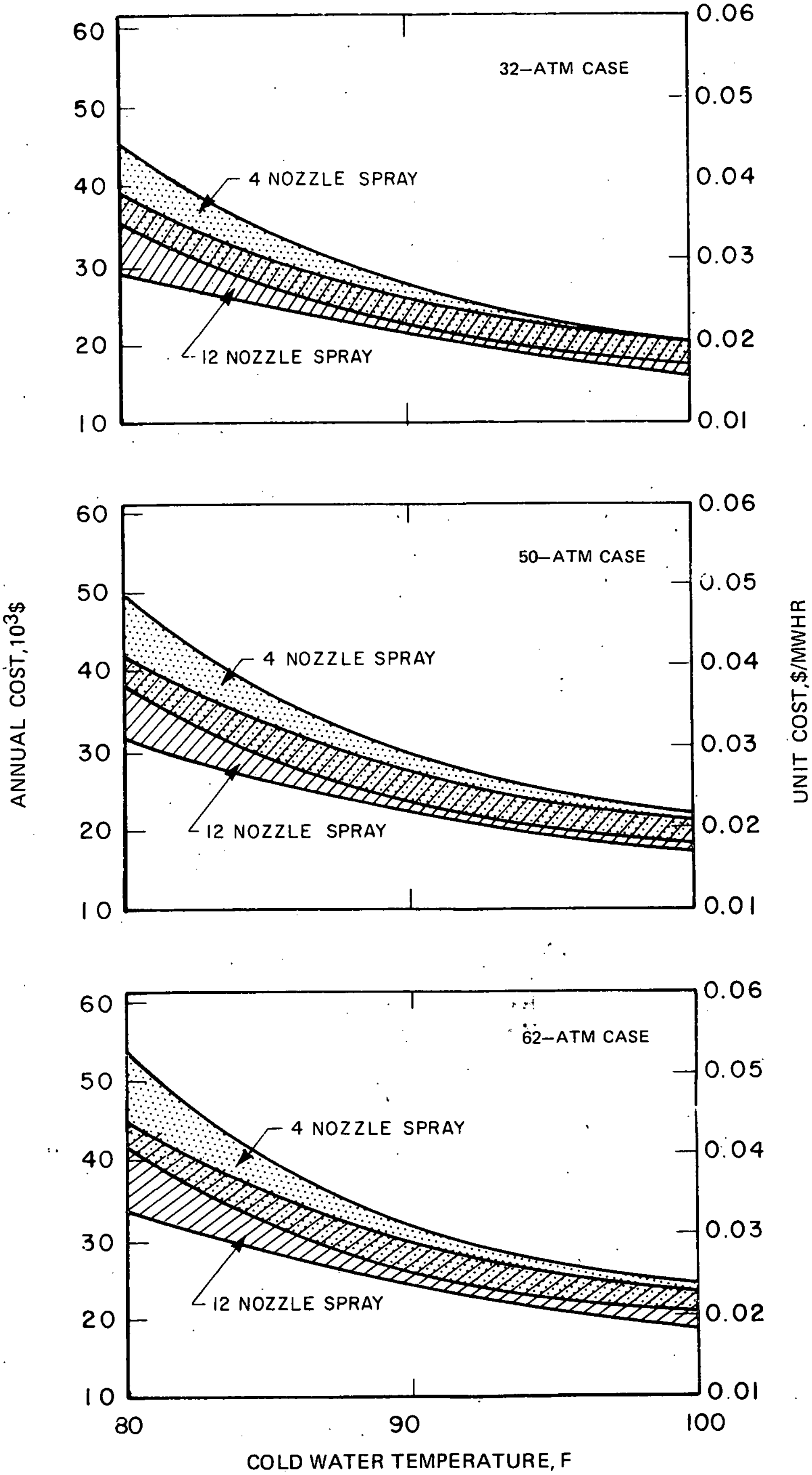

YEARLY OPERATING COST AT FULL LOAD
ANNUAL COST, 10^3\$
UNIT COST,\$/MWHR
ECODYNE CORP
HUDSON PROD.
MARLEY CO
RESEARCH COTTRELL
SEE DETAIL FIG. IV-26
75-4
ASHBROOK CORP.
75-12
MECH. WET TOWERS CONCRETE STRUCTURE
MECH. WET TOWER WOOD STRUCTURE
DRY TOWER
SPRAY CANAL
COOLING POND
COLD WATER TEMPERATURE, F
R76-952161-5
76-05-93-20

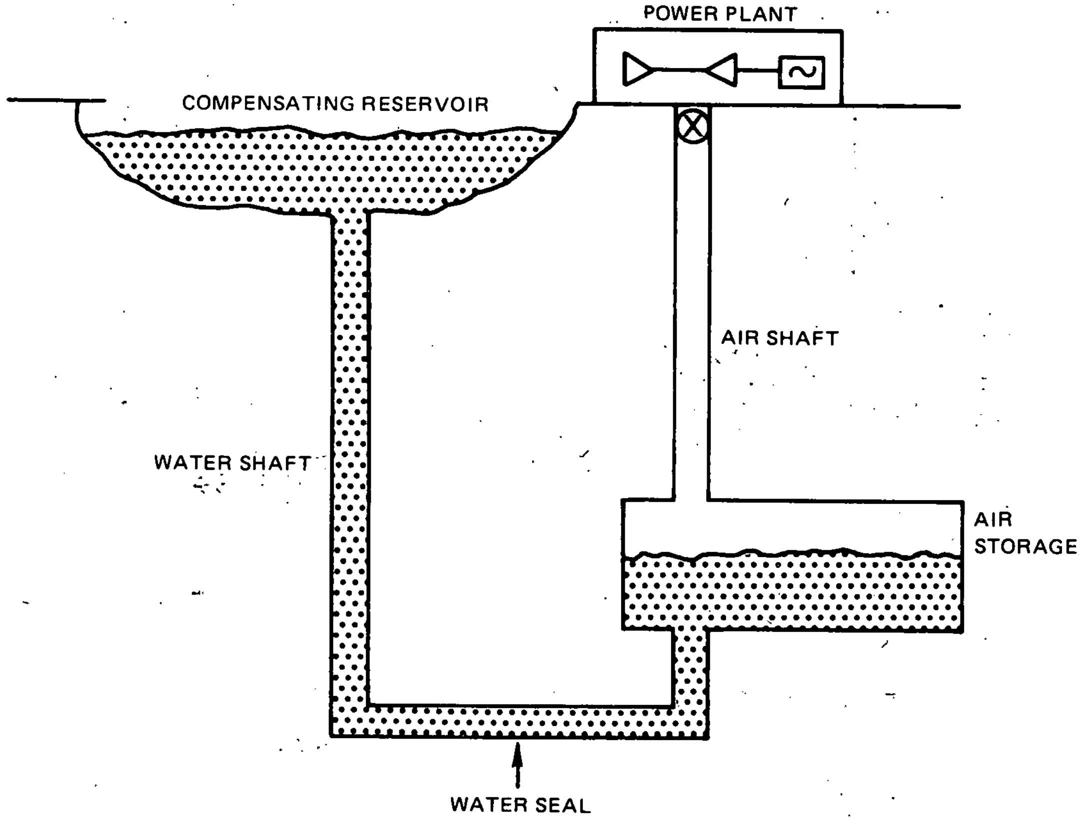
FIG. IV-25

OPERATING COSTS FOR SPRAY COOLING CANAL

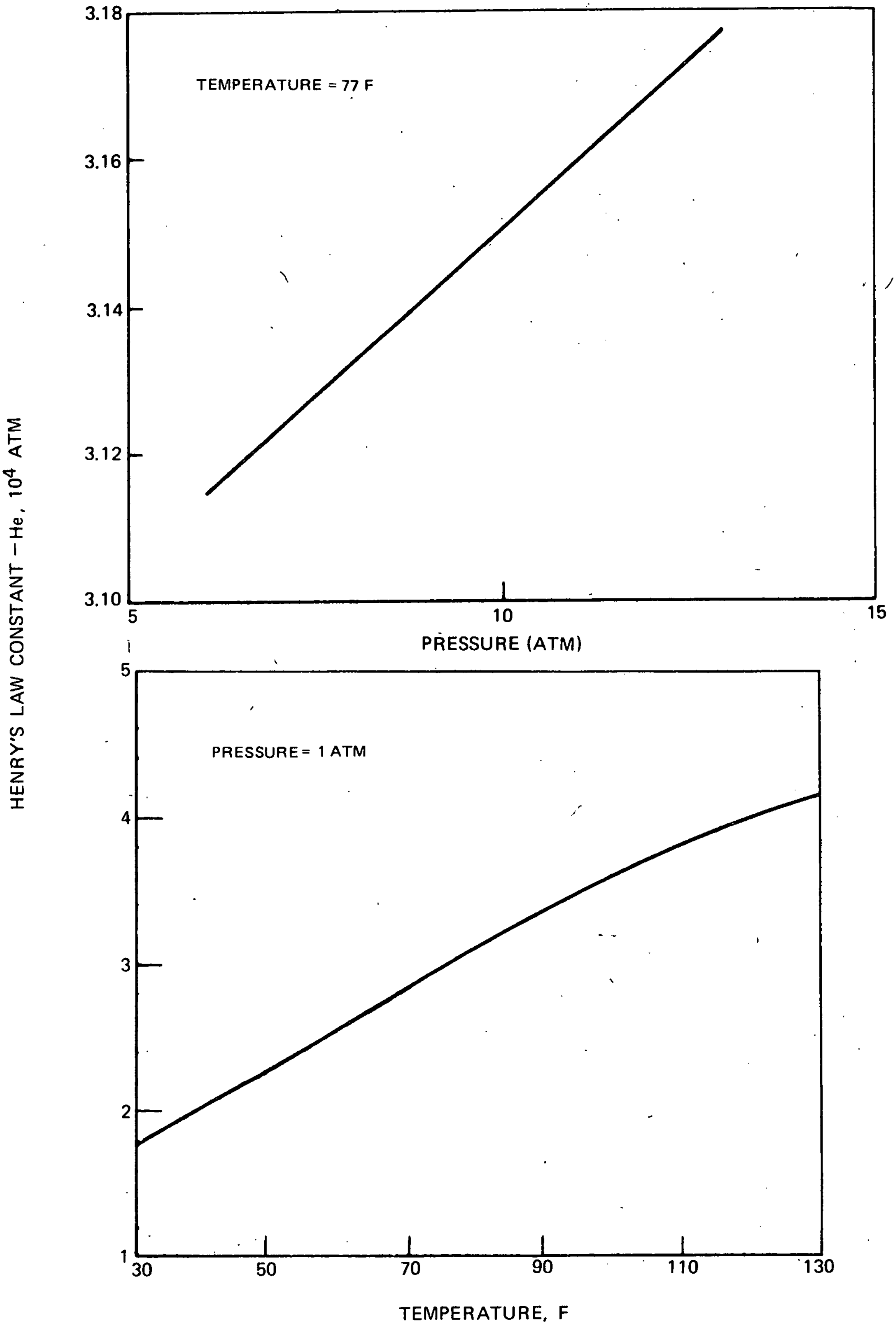


76–05–93–18

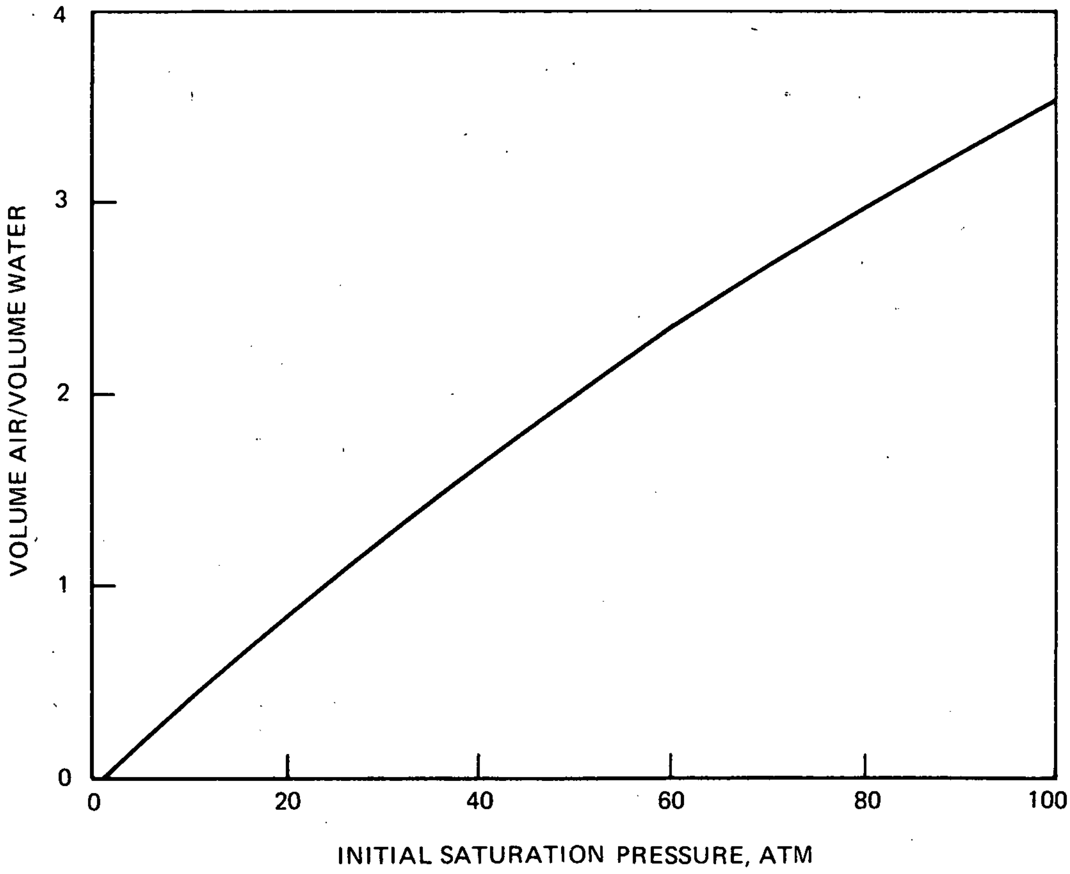
UNDER GROUND CROSS SECTION



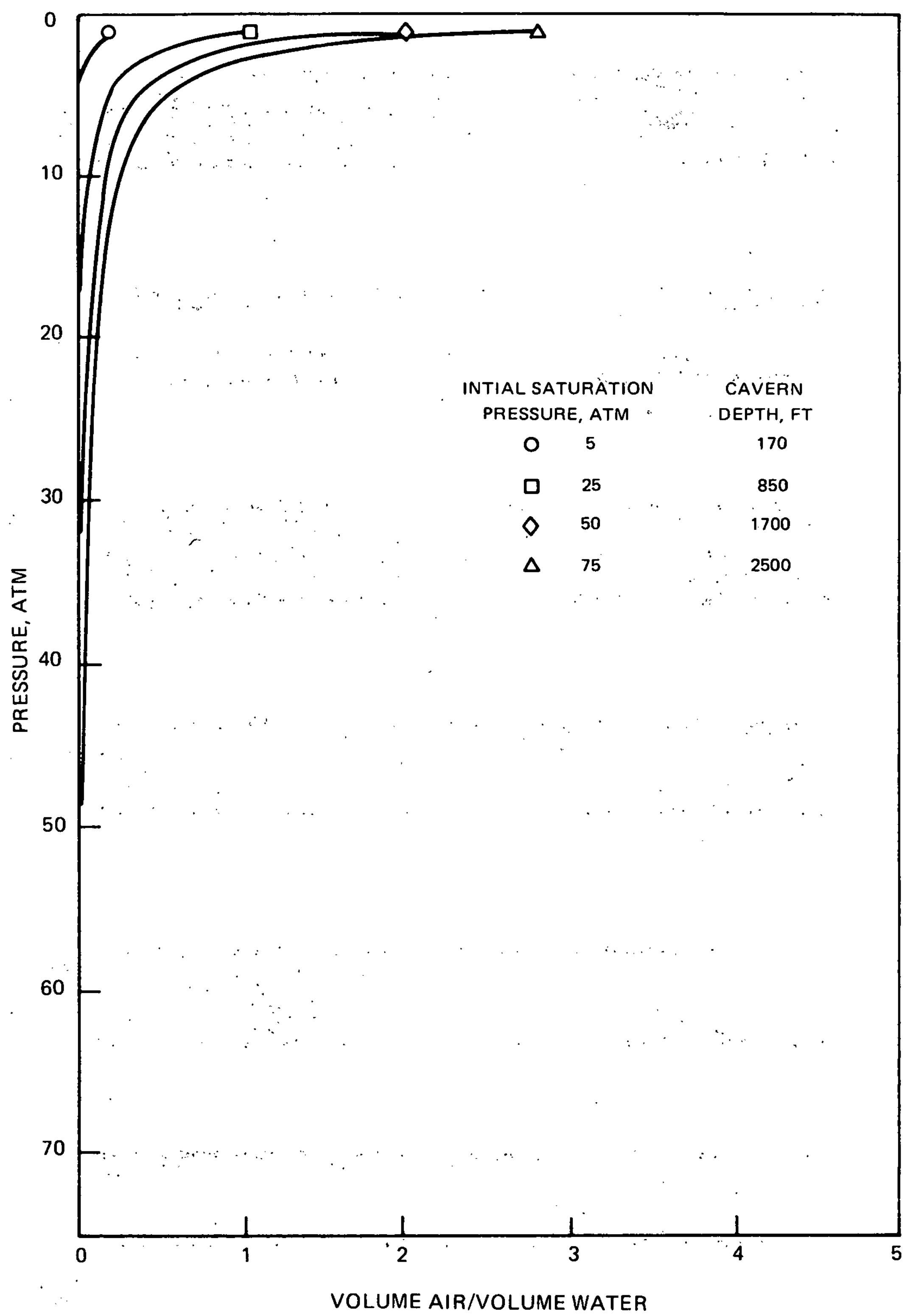
SOLUBILITY OF AIR IN WATER



VOLUME OF DISSOLVED AIR WHEN EXPANDED TO 1 ATM PER UNIT VOLUME OF WATER



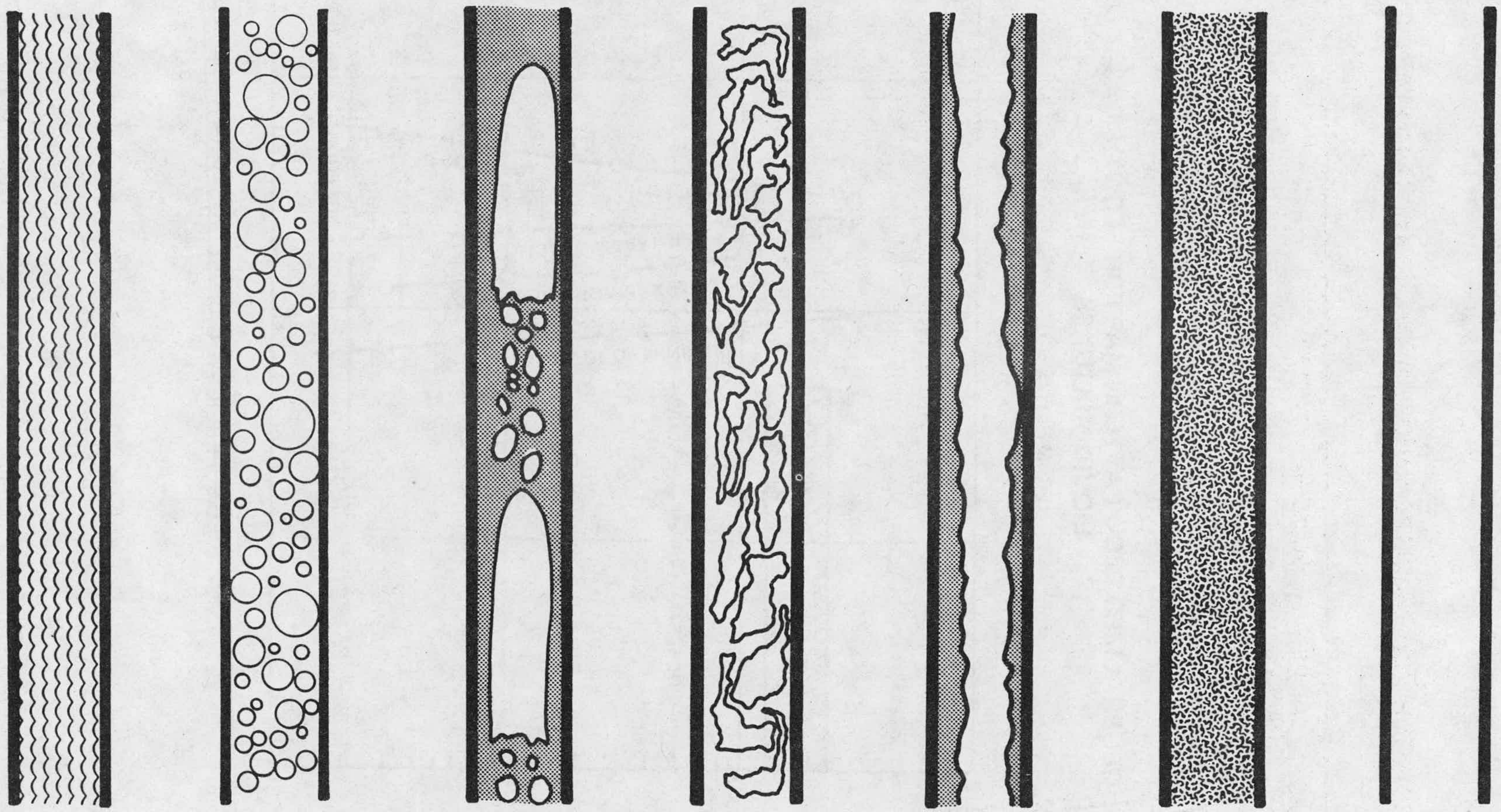
STATIC AIR/WATER VOLUME FOR VARIOUS INITIAL SATURATION PRESSURES



76-10-116-1

R76–952161–5

FLOW REGIMES FOR DIFFERENT AIR/WATER FLOW RATES



AIR/WATER FLOW RATIO (APPROX)	0	0 –0.5	1 –5	10	50	100	∞
REGIME	WATER	BUBBLES	SLUG	FROTH	ANNULAR FILM	MIST	AIR

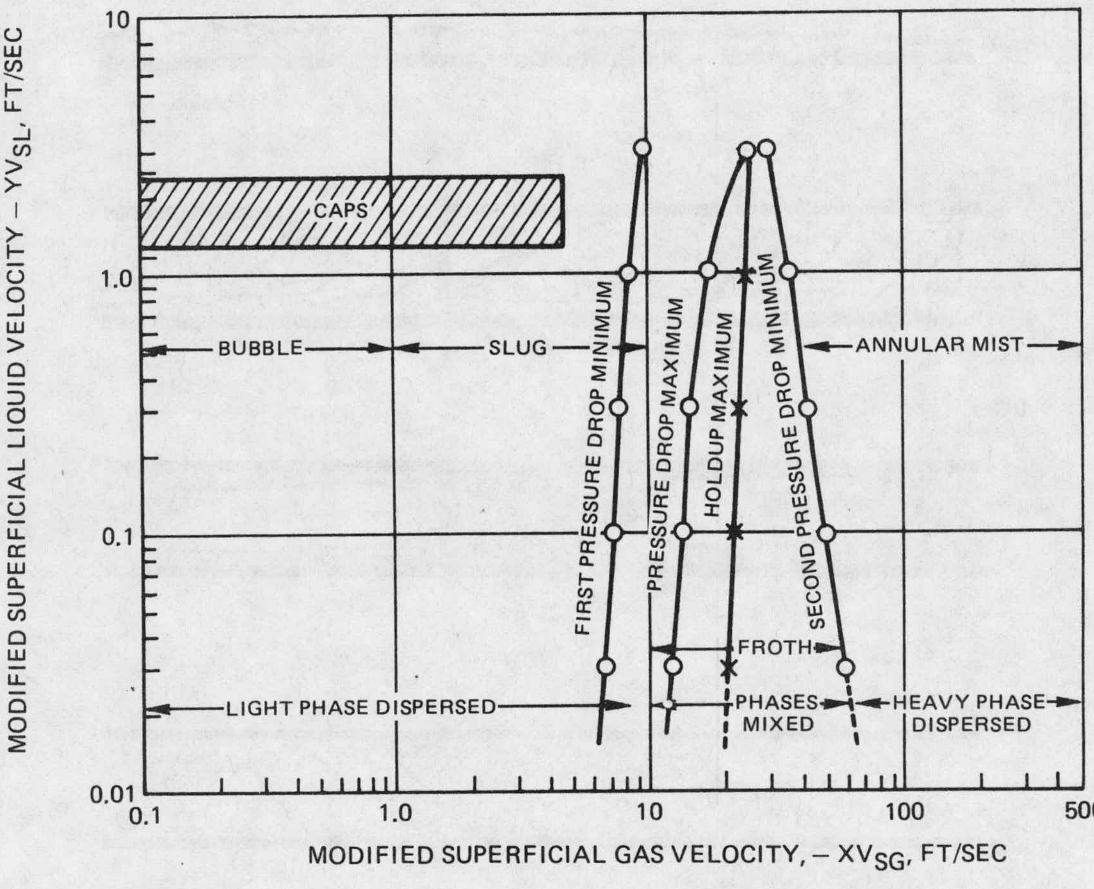
⟶ INCREASING AIR FLOW/WATER FLOW RATIO

76–10–81–9

FIG. IV–31

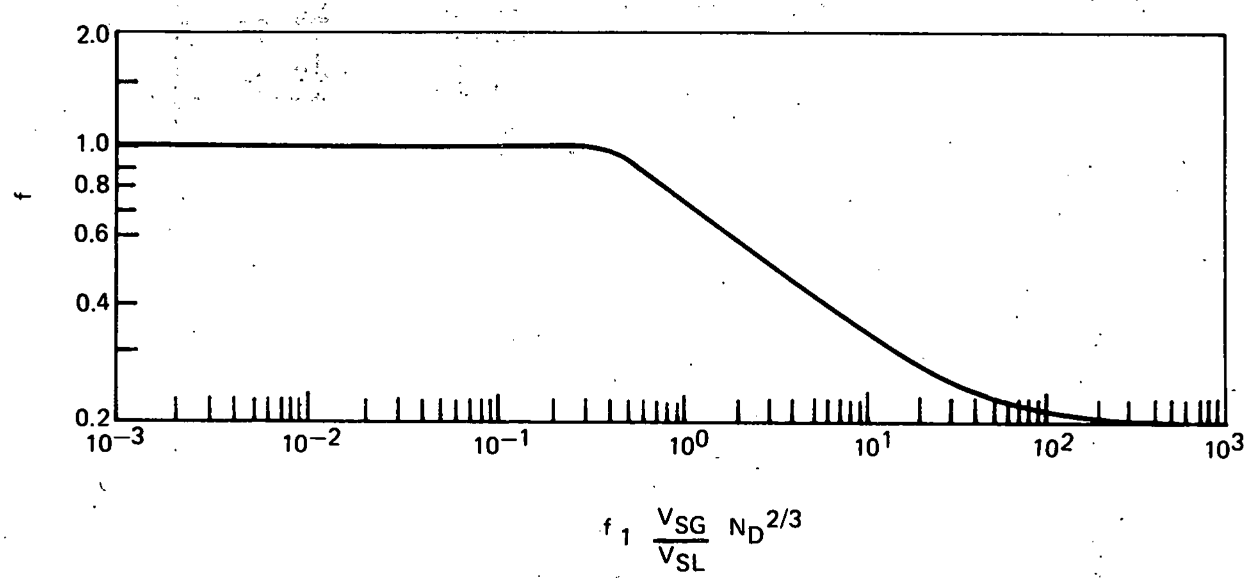
GENERALIZED FLOW PATTERN MAP FOR FLOW OF GAS–LIQUID MIXTURES

(FROM REF. IV–37)



ROS FRICTION FACTOR f_2

(FROM REF. IV–37)

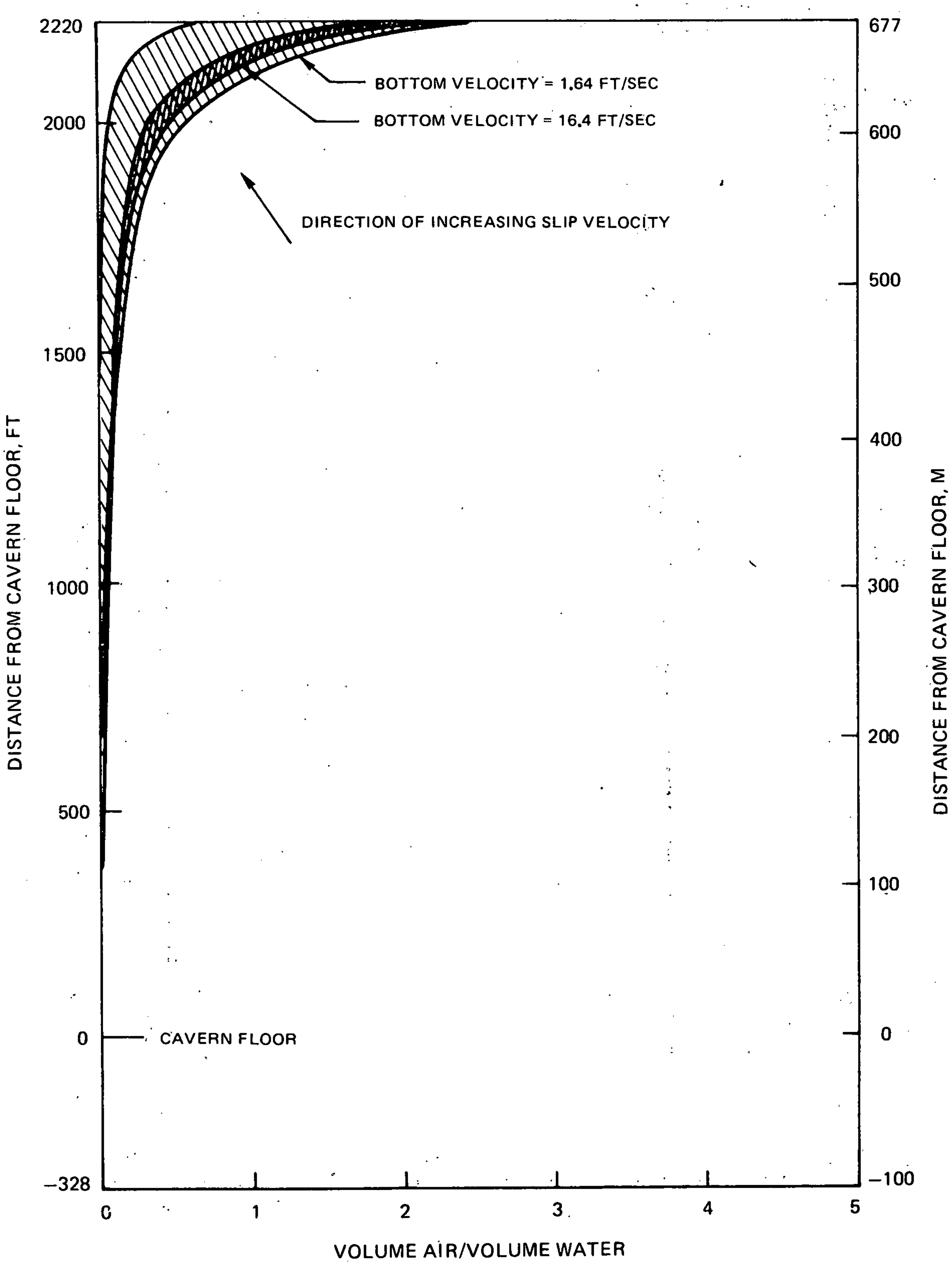


76–10–81–11

FIG. IV–33

AIR/WATER VOLUME RATIO CONSTANT DIAMETER SHAFT
STEADY STATE ANALYSIS

SHAFT DIAMETER = 12 FT
CAVERN PRESSURE = 66.3 ATM
SLIP VELOCITY = 0–6.87 FT/SEC



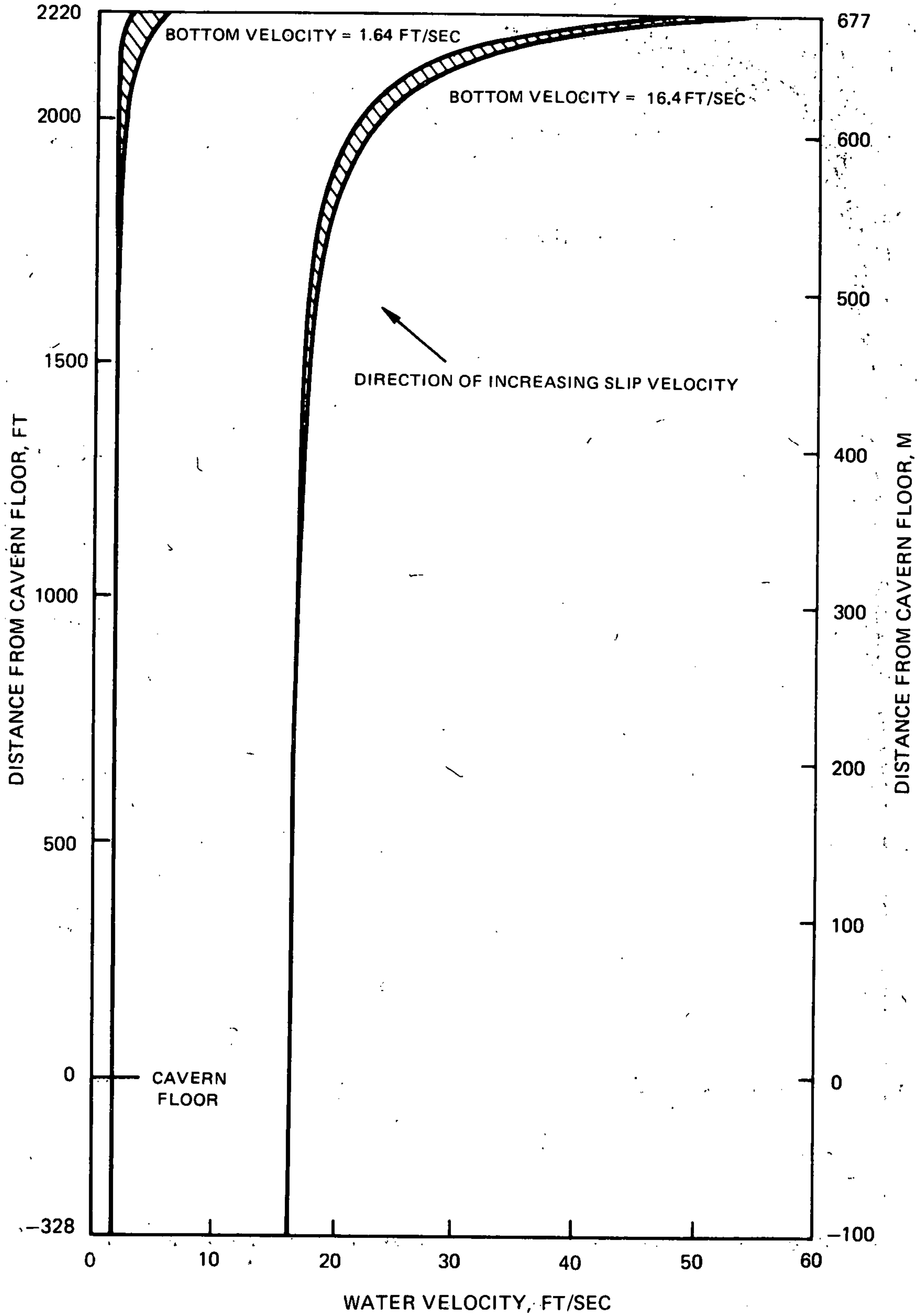
76–10–81–12

FIG. IV–35

WATER VELOCITY FOR CONSTANT DIAMETER SHAFT

STEADY STATE ANALYSIS

SHAFT DIAMETER = 12 FT
CAVERN PRESSURE = 66.3 ATM
SLIP VELOCITY = 0–6.87 FT/SEC



76–10–81–13

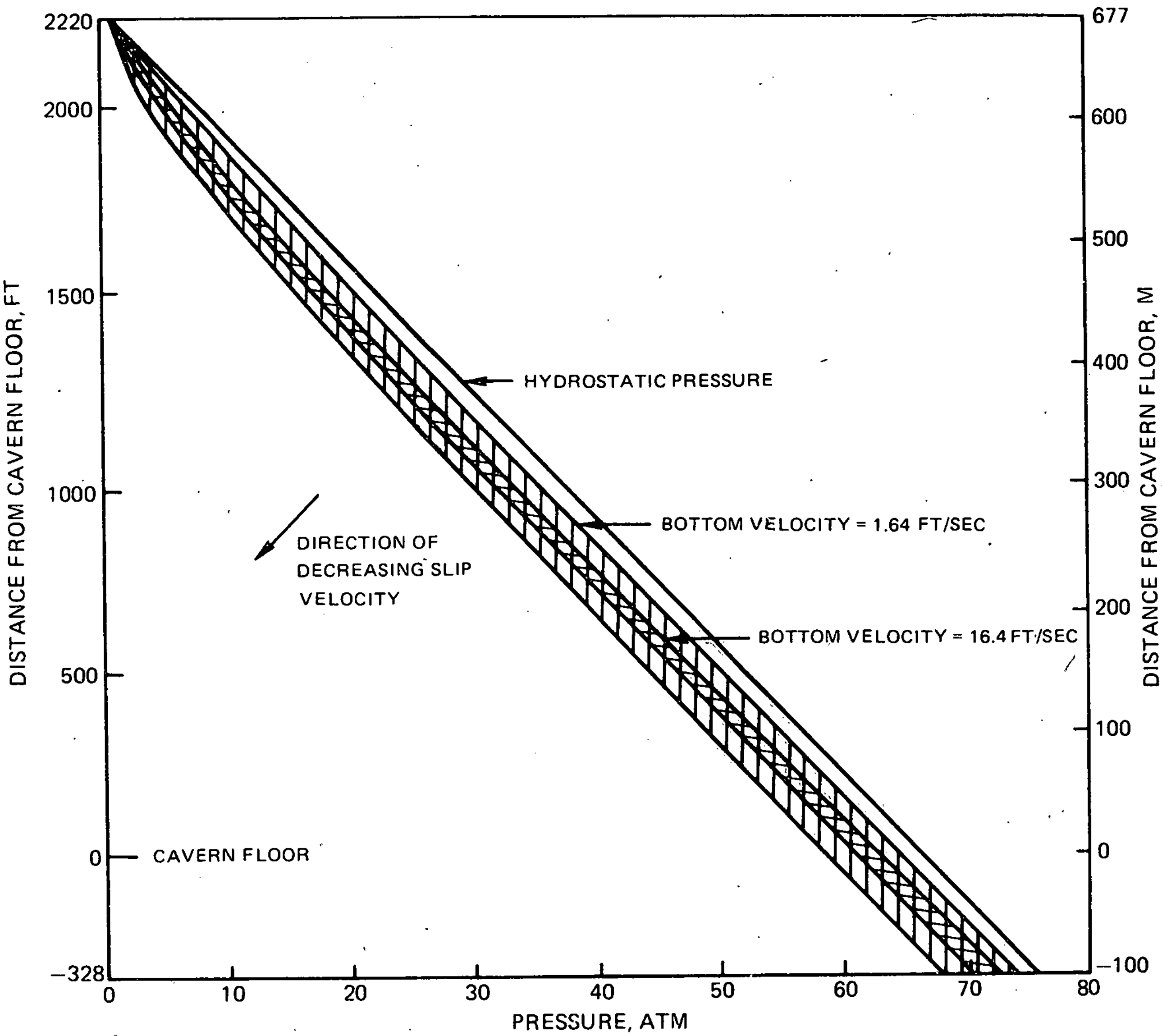
PRESSURE DISTRIBUTION FOR CONSTANT DIAMETER SHAFT

STEADY STATE ANALYSIS

SHAFT DIAMETER = 12 FT

CAVERN PRESSURE = 66.3 ATM

SLIP VELOCITY = 0–6.87 FT/SEC

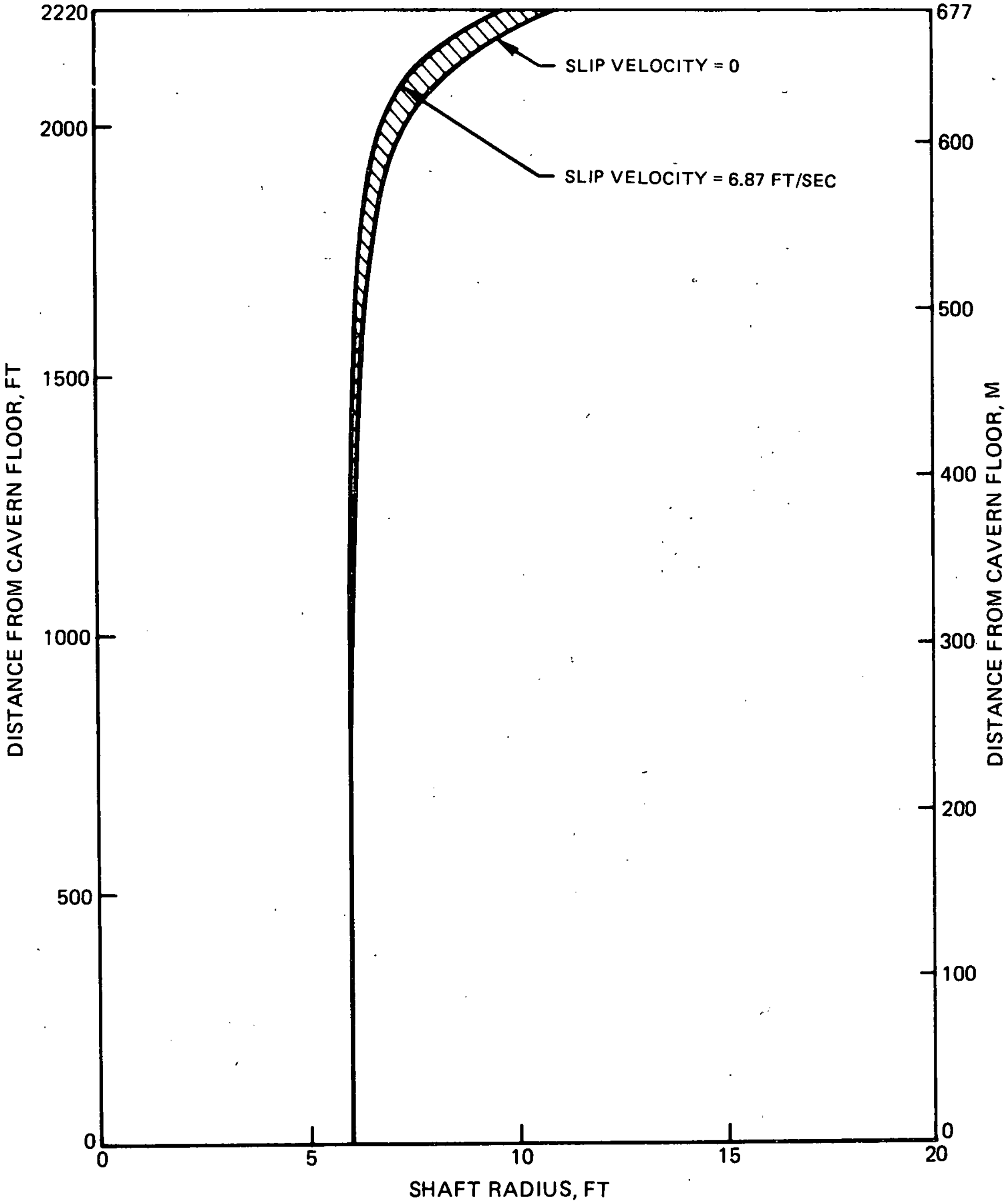


SHAPE OF FLARED SHAFT

STEADY STATE ANALYSIS

CAVERN PRESSURE = 66.3 ATM

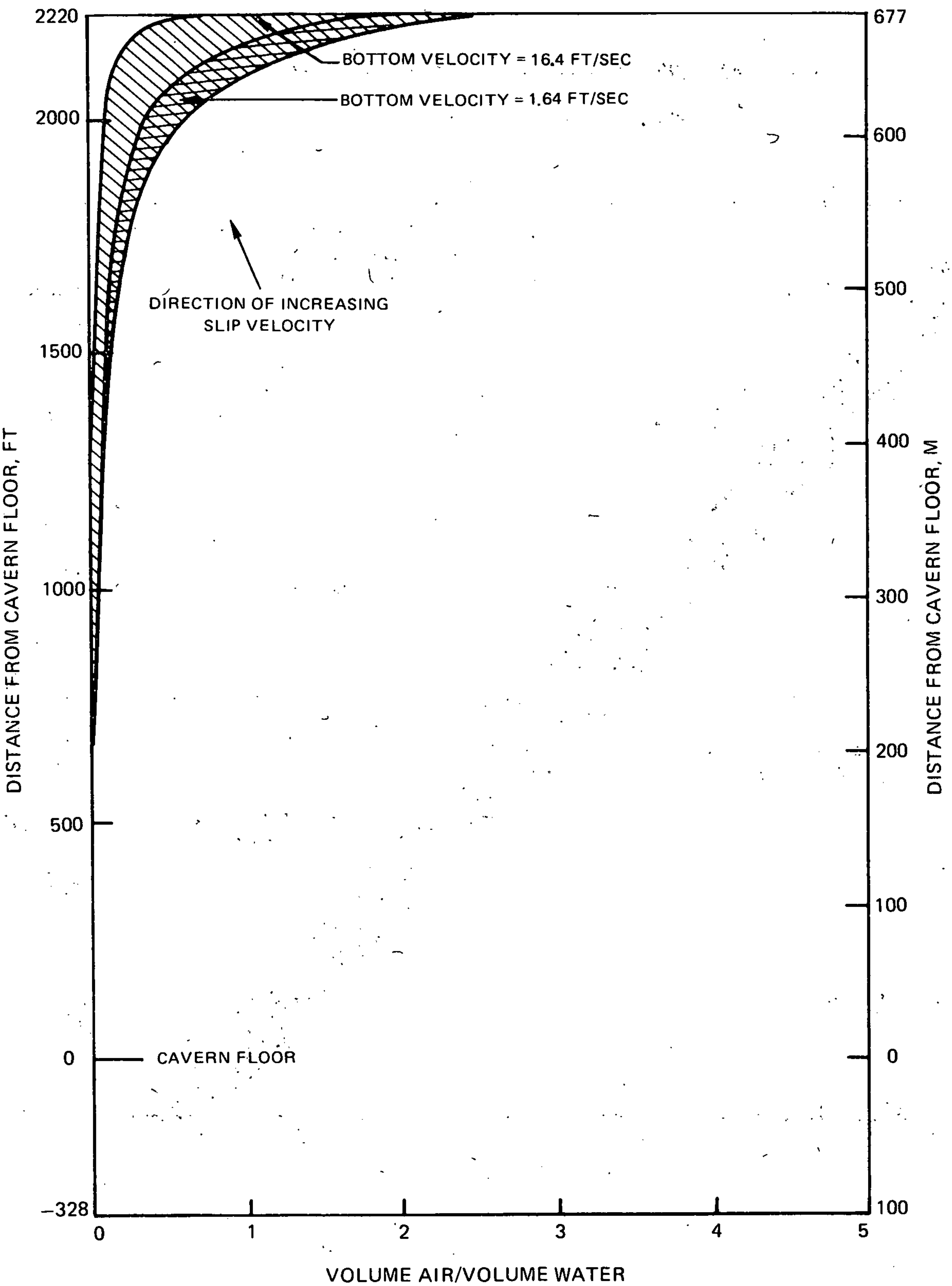
DESIGN BOTTOM VELOCITY = 1.64 FT/SEC



76–10–81–14

AIR/WATER VOLUME RATIO FOR FLARED SHAFT

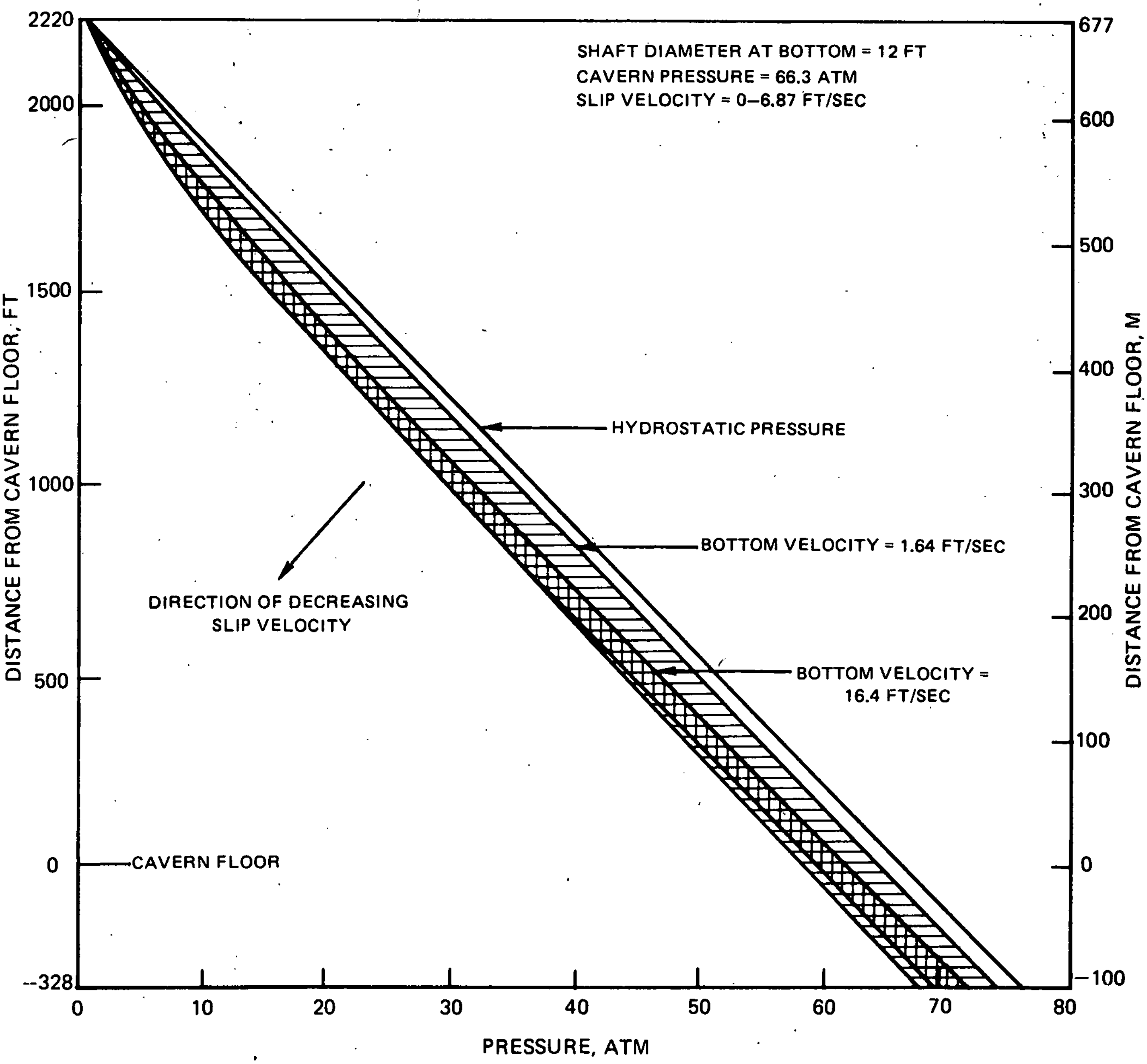
STEADY STATE ANALYSIS



76–10–116–4

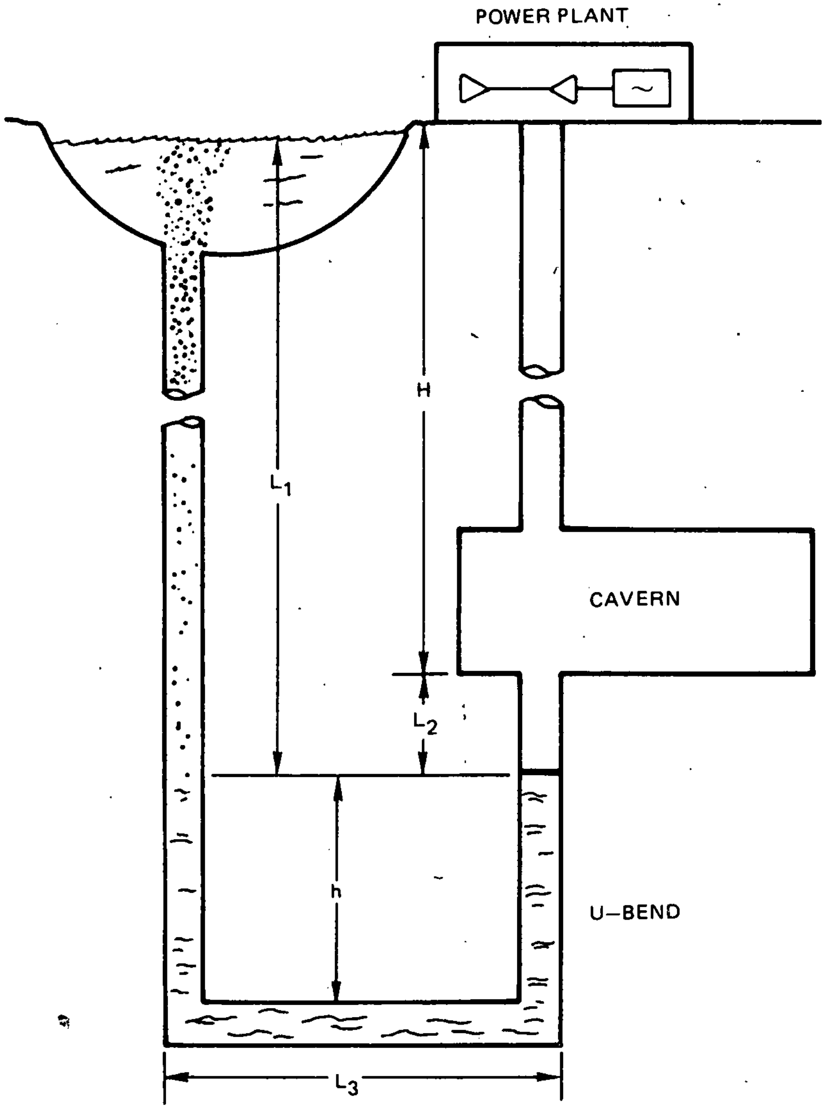
PRESSURE DISTRIBUTION FOR FLARED SHAFT

STEADY STATE ANALYSIS

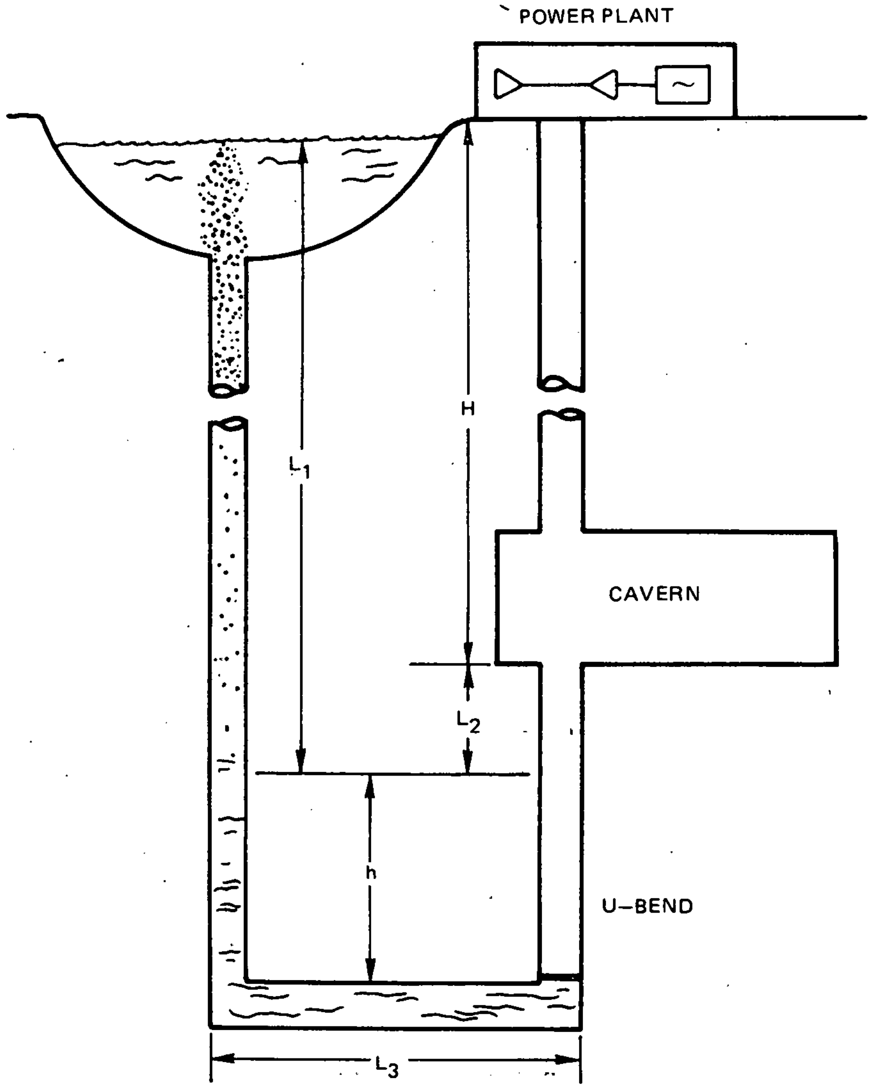


76–10–116–3

WATER SEAL CONFIGURATION



a) DEPTH TO BALANCE CAVERN PRESSURE



b) DEPTH TO BALANCE PRESSURE PLUS INERTIA

76–10–81–2

FIG. IV–40

VELOCITY TRANSIENT HISTORY – CHAMPAGNE EFFECT

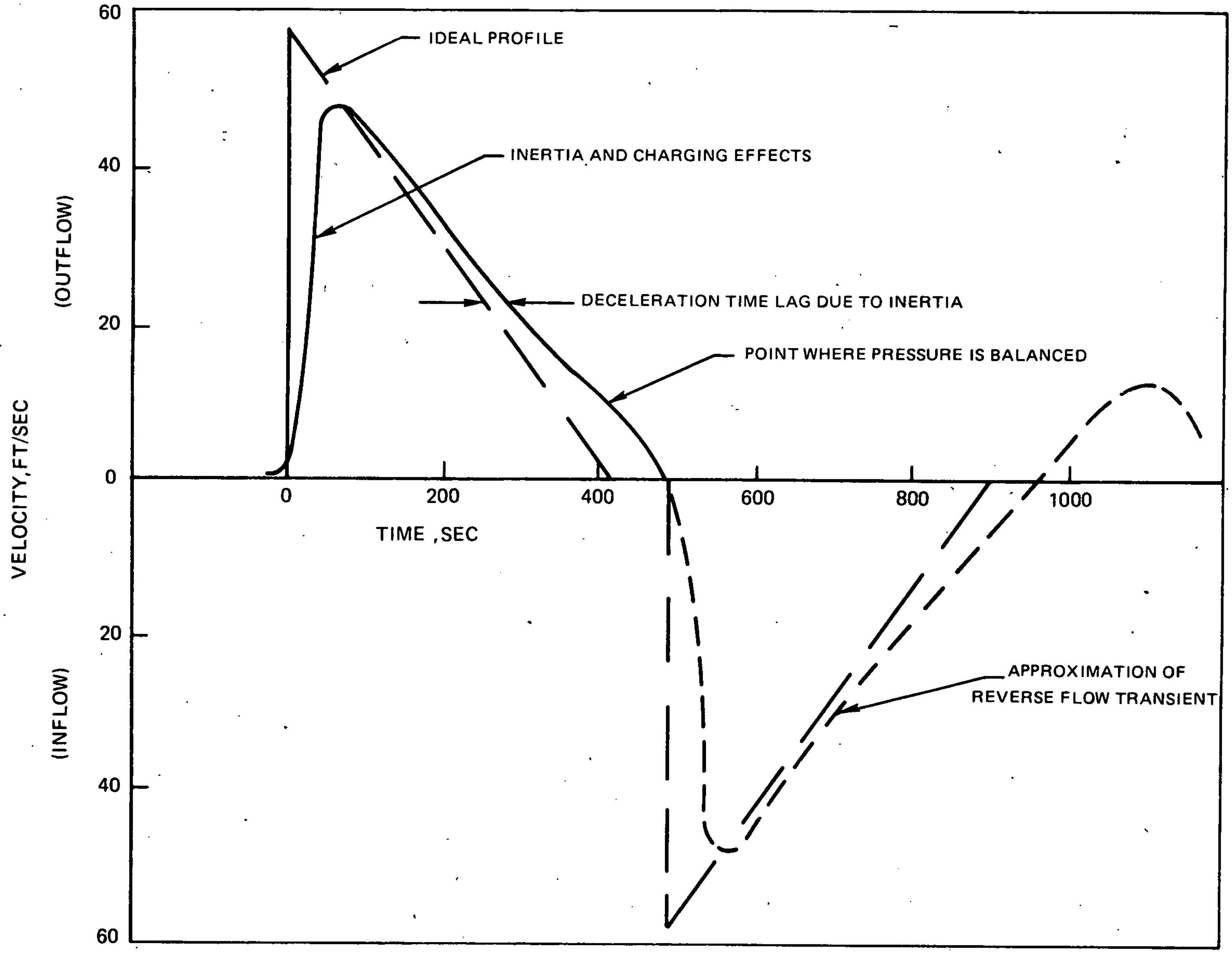
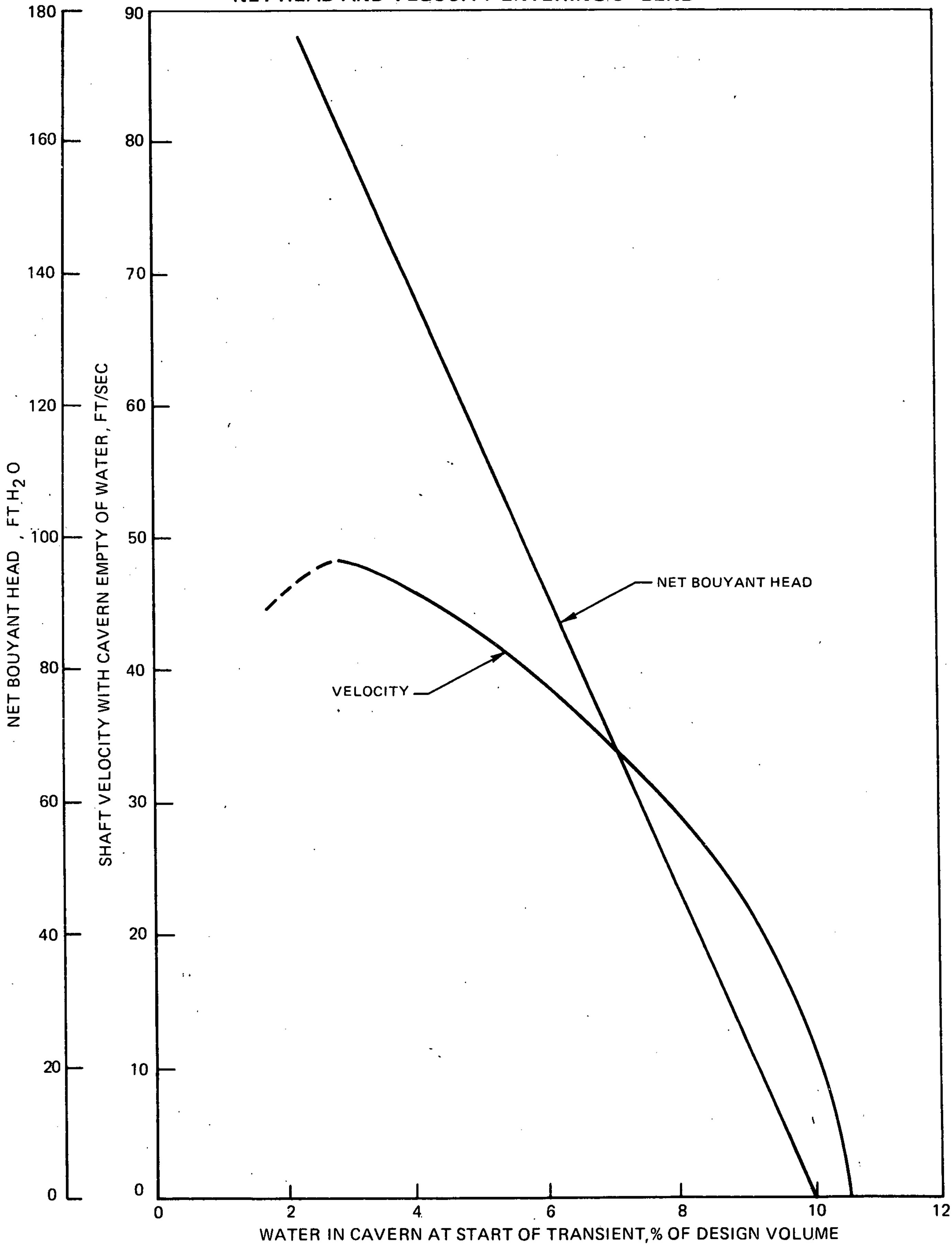


FIG. IV–41

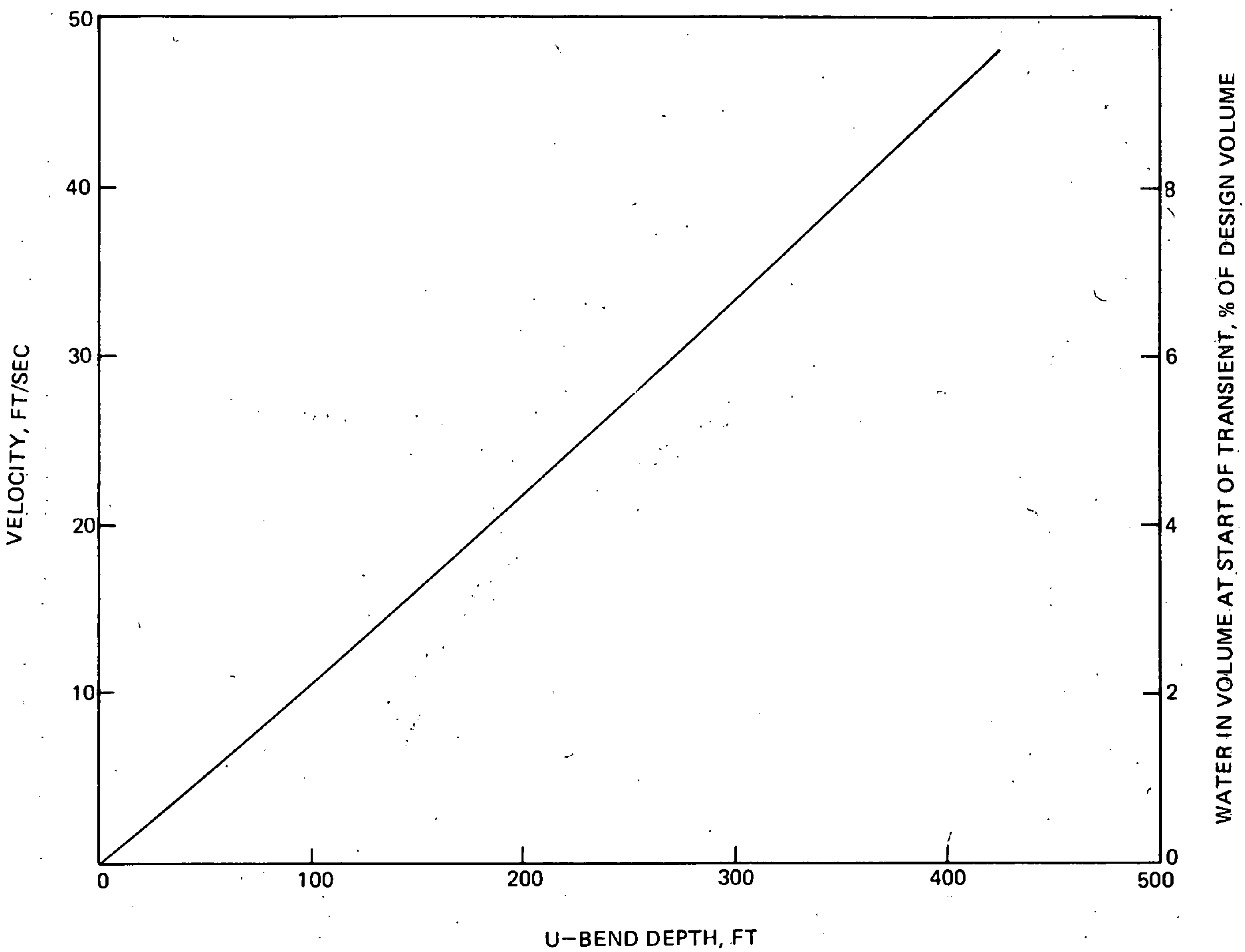
76–12–67–1

NET HEAD AND VELOCITY ENTERING U–BEND



76–12–67–2

MAXIMUM INCREMENTAL U–BEND DEPTH TO OVERCOME INERTIA



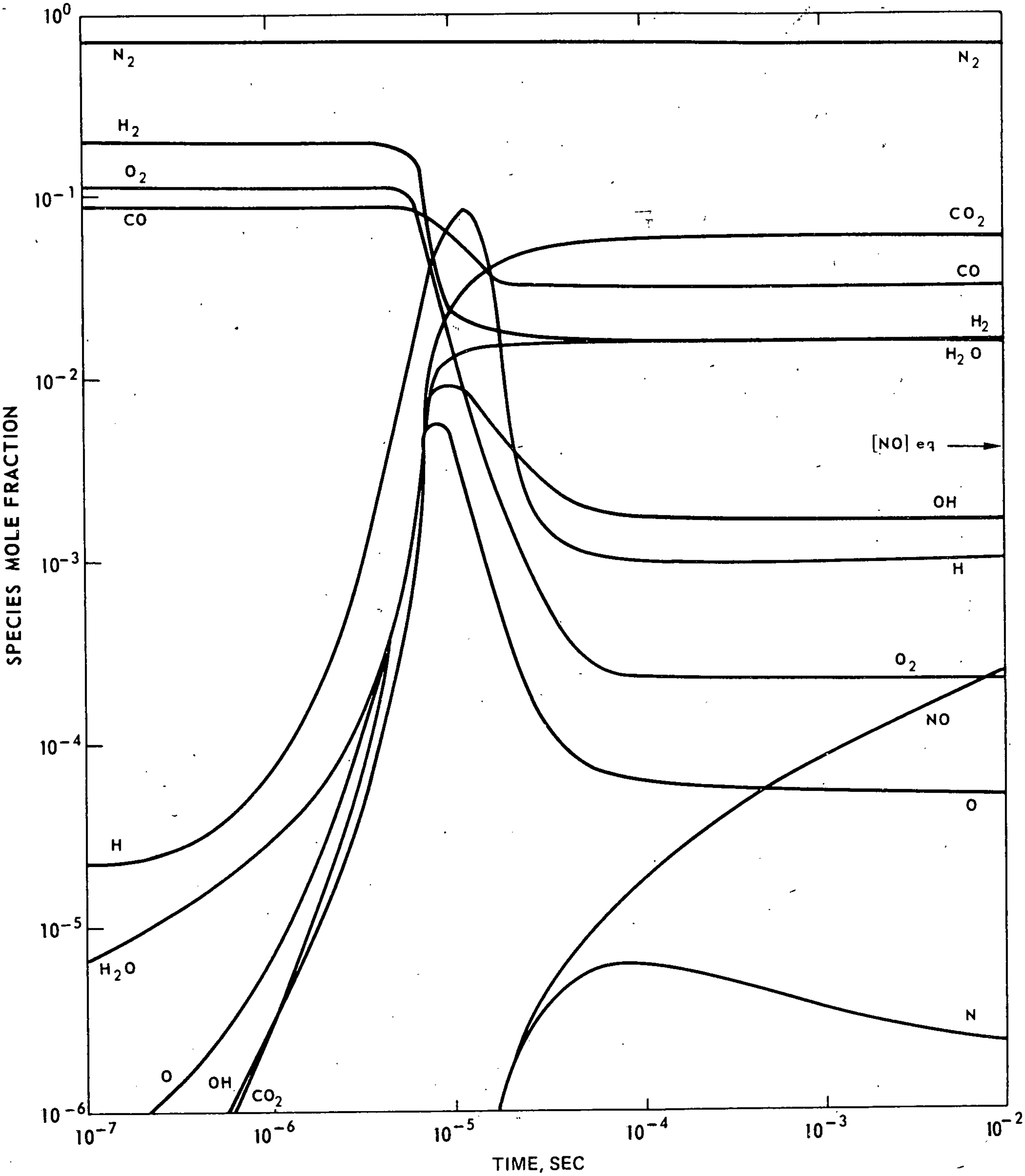
CONCENTRATION–TIME PROFILES FOR PREMIXED H_2–CO–AIR MIXTURE

INLET TEMPERATURE = 1800 R

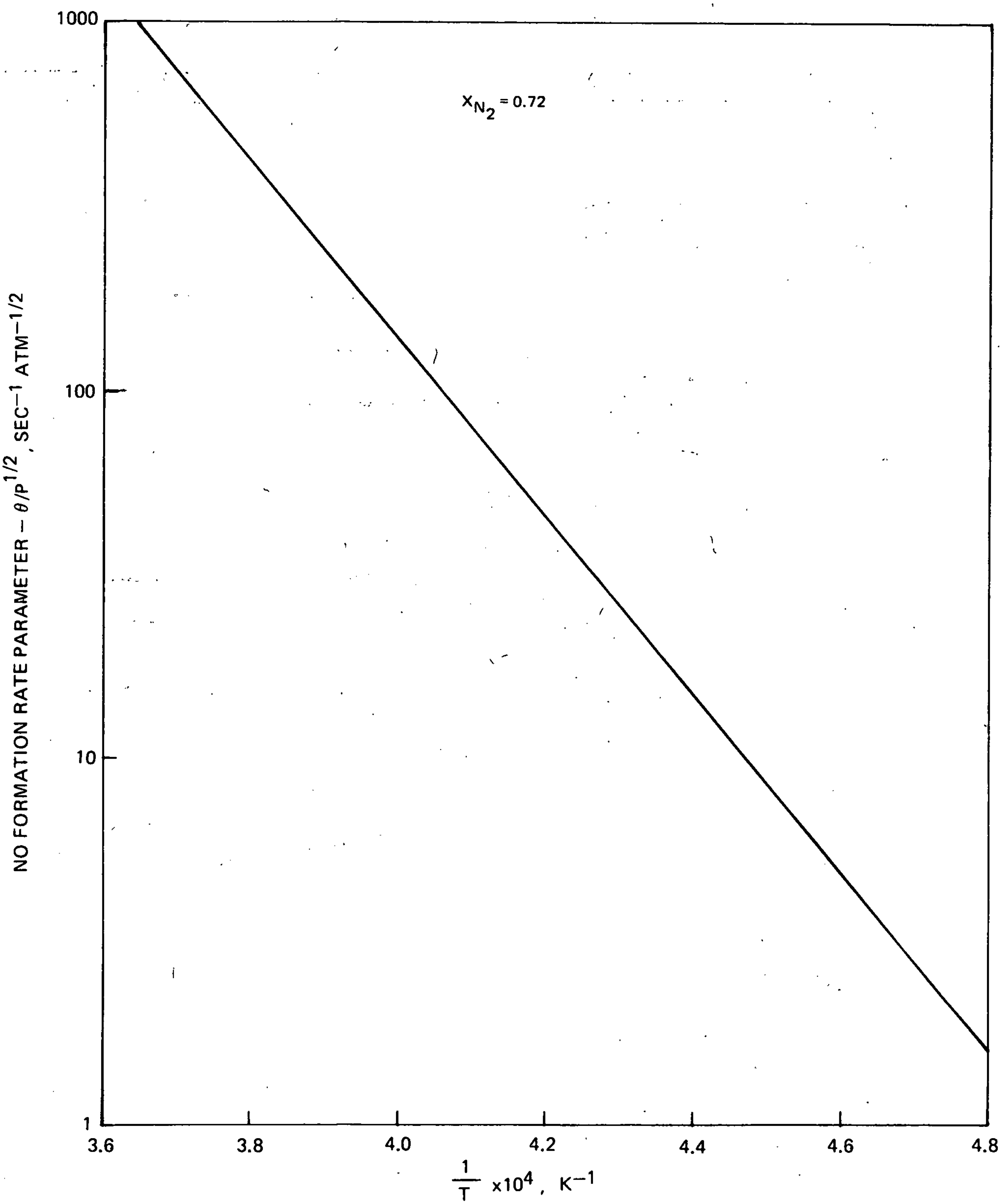
PRESSURE = 10 ATM

COMBUSTION TEMPERATURE = 4460 R

EQUIVALENCE RATIO = 1.0

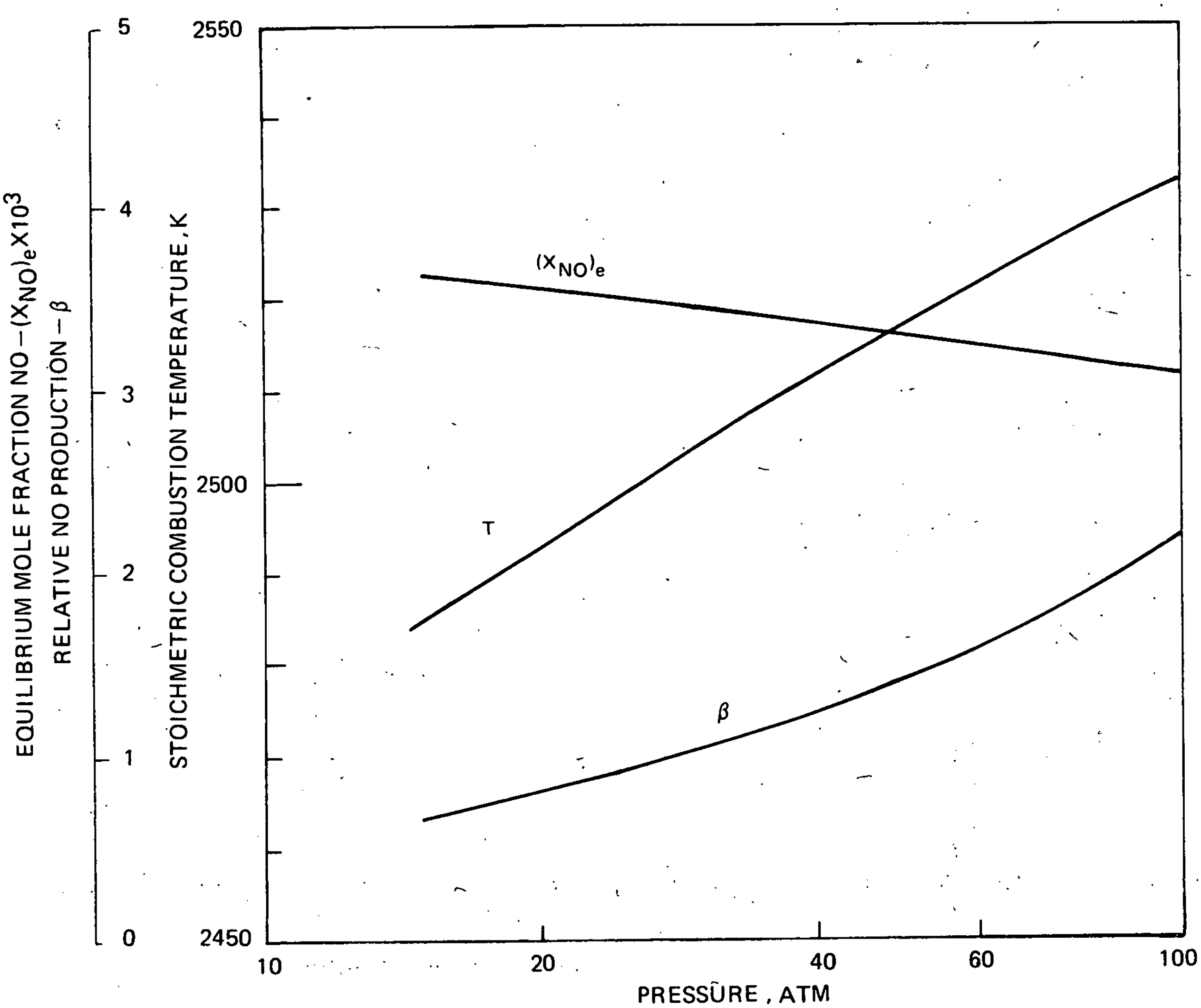


NO RATE PARAMETER AS A FUNCTION OF TEMPERATURE



76–05–223–3

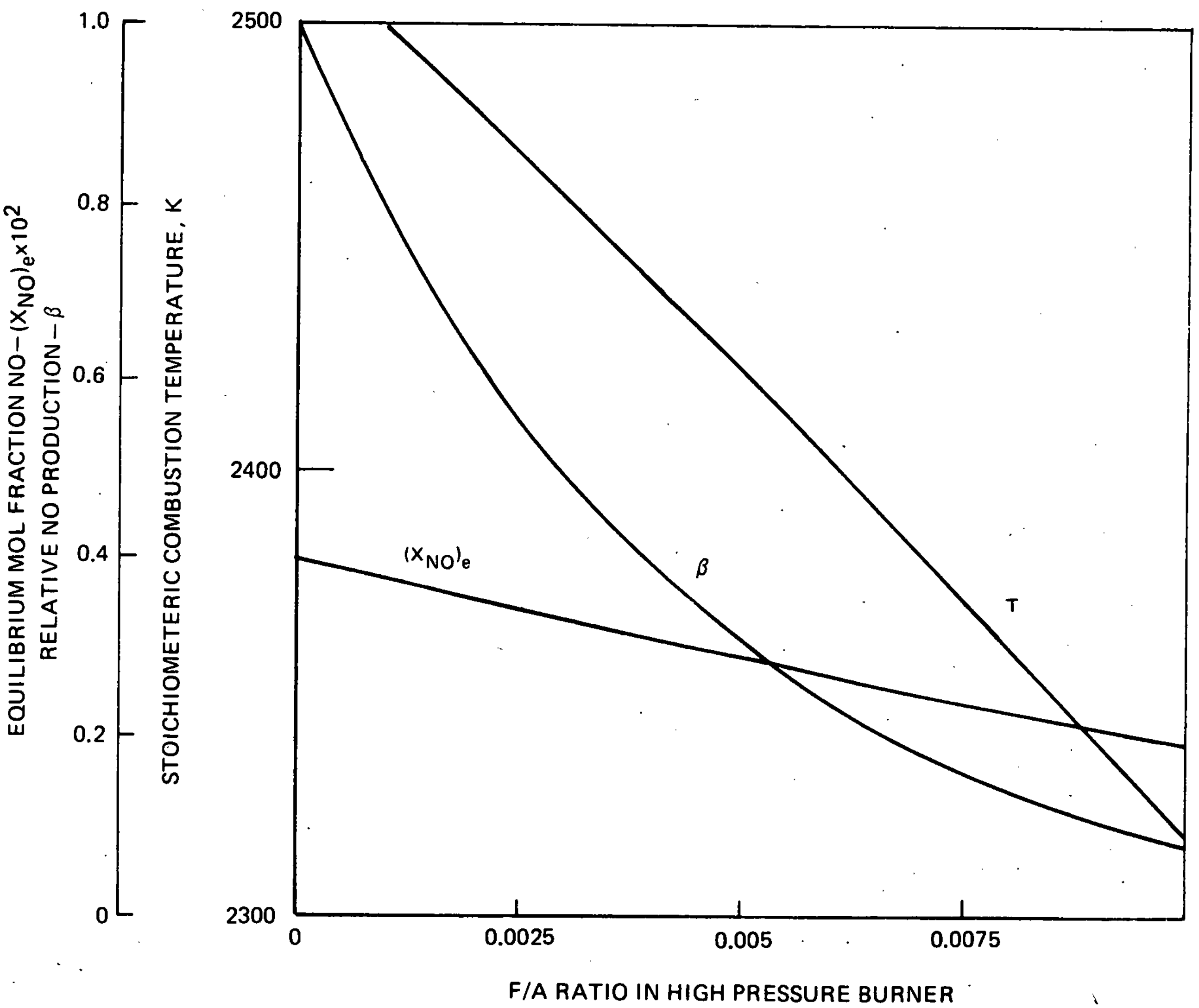
NO PRODUCTION IN HIGH PRESSURE BURNER



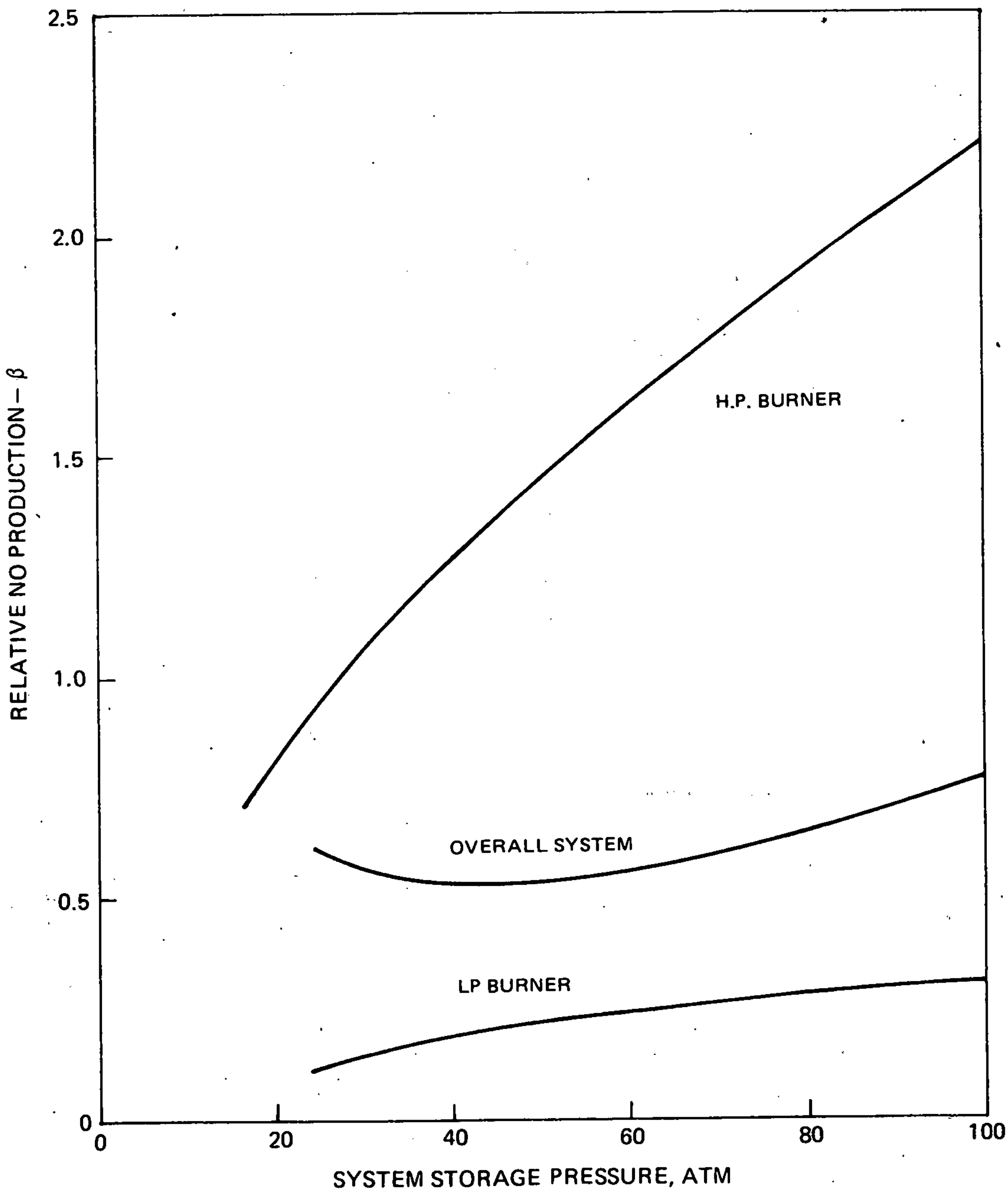
EFFECT OF VITIATED AIR ON LOW PRESSURE BURNER NO PRODUCTION

OXIDIZER TEMPERATURE = 674K

PRESSURE = 16 ATM

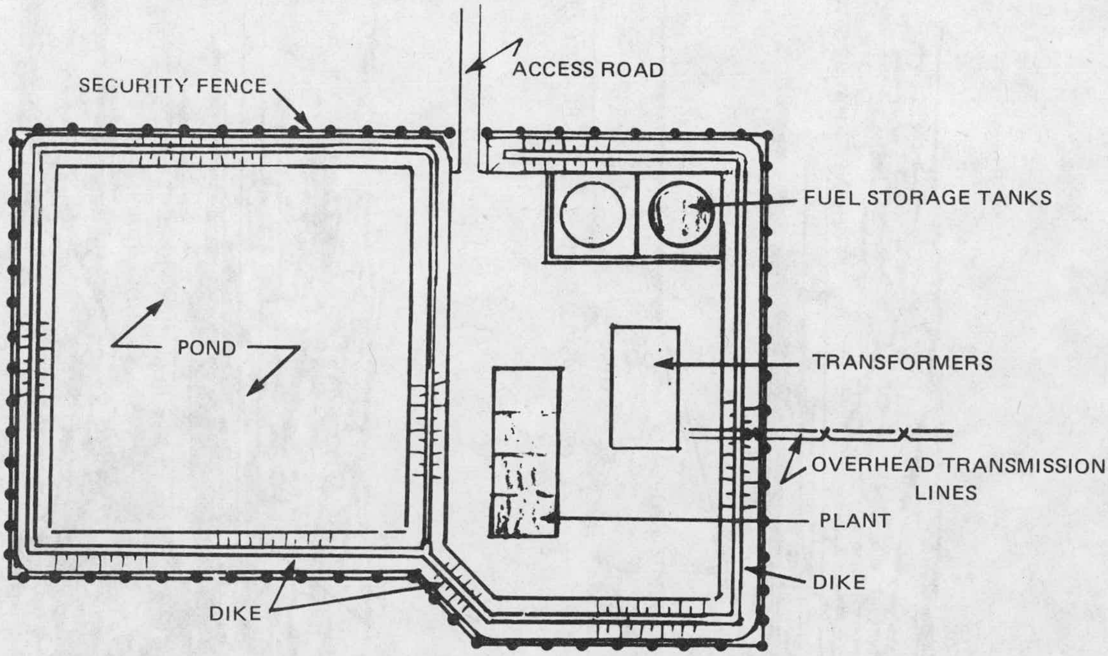


OVERALL NO PRODUCTION

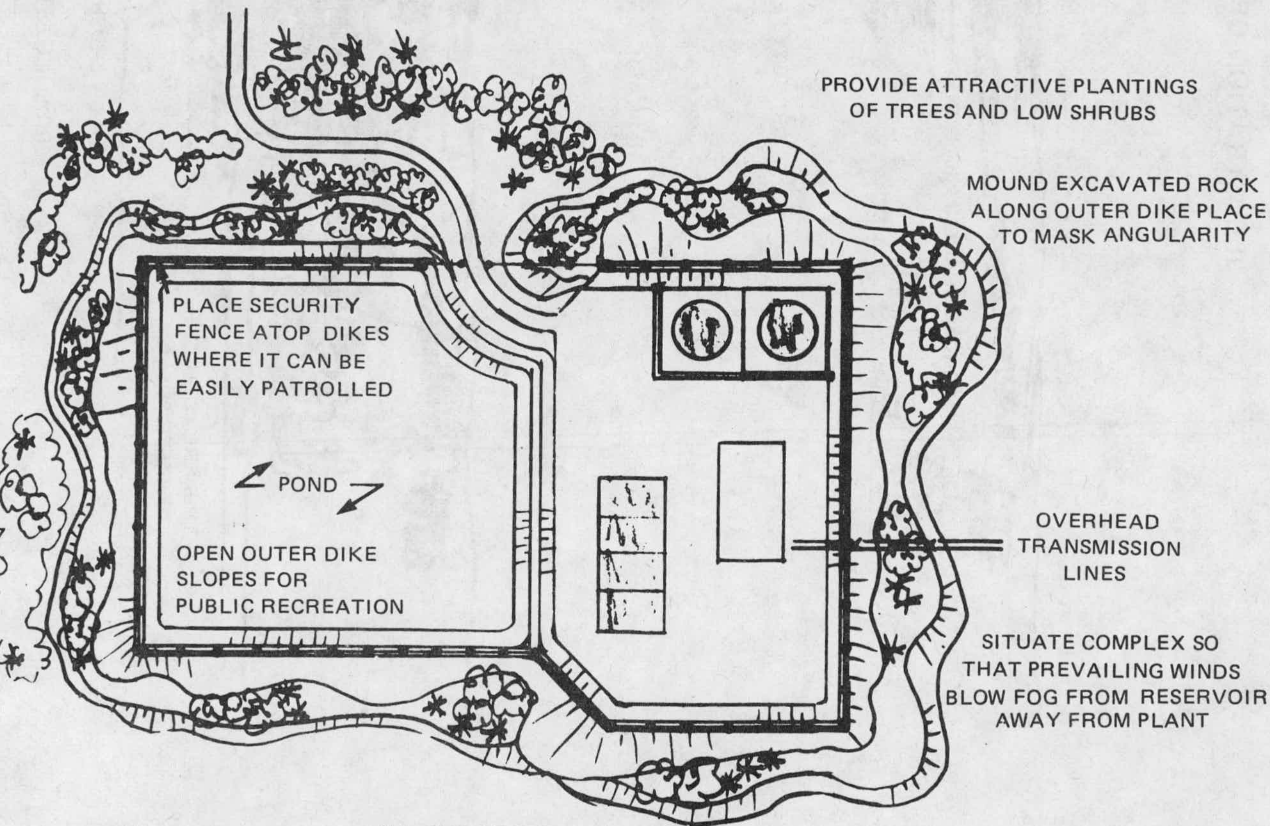


76–05–223–5

PLAN VIEW OF AESTHETIC TREATMENT

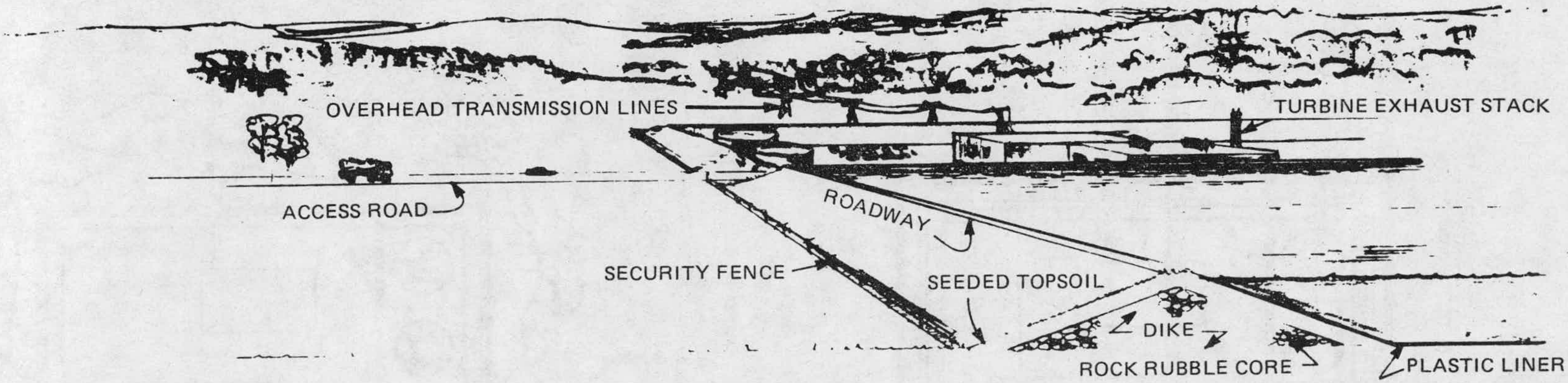


(a) BASIC PLAN

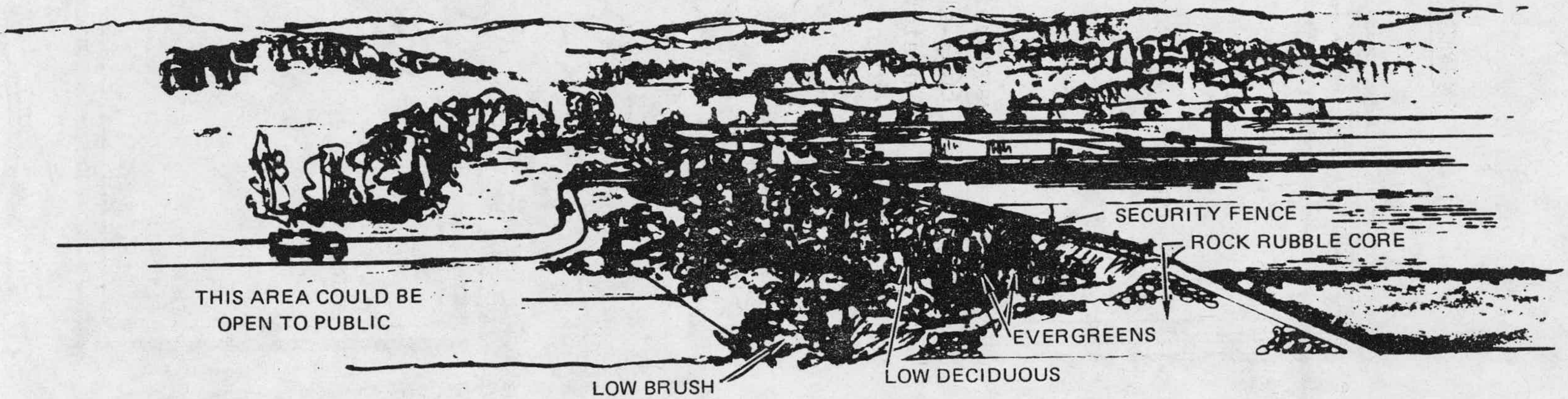


(b) SUGGESTED TREATMENT

ILLUSTRATION OF AESTHETIC TREATMENT



(a) BASIC ARRANGEMENT



(b) SUGGESTED TREATMENT

PART V

COMPRESSED AIR POWER SYSTEM DESIGN

The major components associated with the below-ground and aboveground facilities have been covered in Parts III and IV, respectively, of this report. The combining of these two major segments into an integrated power system is the topic of this part of the report.

Both the performance and cost parametric analyses, described in detail in this part, indicate that storage of air should be at pressures greater than 50 atmospheres. The constraint that CAPS incorporate modified off-the-shelf components eventually led to the selection of a 66.3-atm storage pressure for a reference system design. Additional parametric performance based upon the 66.3-atm CAPS revealed that, unlike the simple-cycle gas turbine, off-design operation is possible with little performance penalty. The 66.3-atm CAPS was also used as the basis for one-, two-, and three-unit power plant designs each incorporating three different levels of storage capacity. The direct capital costs of these nine plants varied from $225/kW of installed capacity for a three-unit (757.5 MW) plant with 6-hr storage capacity to $380/kW for a single-unit (252.5 MW) plant with 40-hr storage capacity.

The following sections and related appendices contain detailed descriptions of the system optimization, the plant design and the cost estimates. Appendix I contains an overview of the analytical model utilized to generate the parametric performance data, while Appendix J presents some comments about real gas effects on CAPS performance estimates.

SYSTEM OPTIMIZATION

This section contains discussions of the CAPS parametric performance analyses which identified the significance of key design parameters; the economic optimization considering trade-offs among fuel, compression energy, storage, and equipment costs which lead to the selection of the final system configuration; and off-design operation of the selected configuration.

Parametric Performance Analysis

The thermodynamic performance of CAPS is important because it determines, among other things, the energy requirements of the system, component sizes, and ultimately the costs of the above-ground and below-ground facilities. Using appropriate values for component efficiences, pressure losses, and other engineering design parameters, the state-of-the-art performance program (see Appendices I and J) was used to conduct preliminary parametric performance analyses for the configuration in Fig. V-1 in order to estimate the required characteristics of the turbomachinery and air storage facilities in terms of the cycle pressure ratio at storage (P_9/P_0), turbine inlet temperatures (T_{13} and T_{16}), and other pertinent cycle variables (refer to state points in Fig. V-1).

The primary power plant independent parameters investigated are listed in Table V-1. The range of low turbine inlet temperatures are representative of current and advanced technology gas turbines while the range of cycle pressure ratios bracket reasonable upper and lower storage pressure limits. These limits are set at the lower end by the low-pressure turbomachinery and at the upper end by reasonable cavern depths. The variation in recuperator effectiveness allows for performance trade-offs between heat exchanger size, cost and fuel savings. The cavern flow leakage was varied over a range assumed adequate to cover any reasonable value which would be acceptable to a CAPS installation. The discharge temperature reflects compressor material technology while the compressor airflow value indicates the appropriate physical size of the turbomachinery. These design variables and constraints were used to determine general power plant performance and sizing parameters without regard to limitations imposed by current commercial equipment.

Three separate sets of parametric calculations were made. These sets were defined by the conditions associated with the expansion turbine. Initially, the expansion turbine inlet temperature was fixed at 1500 F and the exhaust temperature (T_{14}) at 1000 F. The second set of calculations was made with the expansion turbine inlet temperature fixed at 1500 F and the exhaust pressure (P_{14}) at 16 atmospheres. In the third set, the expansion turbine exit conditions corresponding to the FT50 burner inlet were selected (T_{14} = 750 F, P_{14} = 16 atm) so that the FT50 could be considered as the basis for the low-pressure turbomachinery design.

Results from the parametric calculations are discussed in the remainder of this section. Detailed results are presented for the first and third sets of calculations dealing with the fixed expansion turbine temperatures and fixed expansion turbine exit conditions corresponding to the FT50 design, respectively. Only summary results are presented for the second set of calculations for comparison purposes.

Parametrics for Fixed Expansion Turbine Temperatures

Component Sizing Parameters

The physical sizes of the turbomachinery and cavern determine, to a great extent, the costs of the above ground and below-ground facilities. Specific turbine flow rate (lb of air/kWhr) provides an indication of turbomachinery size and cost. In general, the lower the specific turbine flow rate the smaller in diameter and less expensive will be the turbomachinery. This parameter is plotted on Fig. V-2 in terms of cycle pressure ratio and high turbine inlet temperature. The remaining cycle variables (recuperator effectiveness and cavern leakage flow rates) have little effect on specific turbine flow. It can be seen from this figure that there is significant inverse variation of specific turbine flow with high turbine inlet temperature and cycle pressure ratio. This would indicate that high pressure ratio and high turbine temperatures are desirable for minimizing the size and cost of the turbomachinery.

A measure of the size of the storage cavern is the specific storage volume (ft^3 of cavern volume required/kWhr). Estimates of specific storage volume are presented in Fig. V-3 in terms of cycle pressure ratio and high-turbine inlet temperature. Again, there is an inverse variation, this time of specific storage volume, with cycle pressure ratio and high-turbine inlet temperature. The significance of the variation with cycle pressure ratio is dramatically evident from the figure. Perhaps not so dramatically indicated is the significant effect of high-turbine inlet temperature, especially at higher cycle pressure ratios. For example, an increase in high-turbine inlet temperature from 1600 F to 2400 F at a cycle pressure ratio of 50 could reduce the cavern volume requirement by approximately one-quarter.

Energy Requirements

The operation of CAPS requires energy input during two parts of the cycle. Electrical energy would be consumed during compression of the air prior to storage, and chemical energy (fuel) would be consumed during generation to raise the temperature of the air prior to expansion.

Compression mode

The specific compressor energy requirement (kWhr/lb of air compressed) indicate the electrical energy demand during compression. This requirement is depicted in Fig. V-4 in terms of cycle pressure ratio. The specific compressor power is independent of the remaining cycle variables.

In the compression mode, the compressor power is supplied by the electrical grid at off-peak power rates. The gross energy for pumping is a function of the energy required during compression, the available energy during expansion, the efficiency of the system supplying the energy for compression, and the cavern leakage. Figure V-5 indicates the energy ratio, expressed as equivalent compression heat rate, for pumping during the compression cycle prior to storage. Factored into the pumping energy are the off-peak power plant heat rate (10,000 Btu/kWhr [HHV]), the electrical and mechanical losses (4 percent) and the cavern air leakage (4 percent).

Generation mode

The fuel system heat rate (Btu of fuel burned/kWhr [LHV]) indicates the chemical energy demand during the generation mode. This requirement is depicted in Fig. V-6 in terms of cycle pressure ratio and high turbine inlet temperature. Three observations can be drawn here: (1) increasing the cycle pressure ratio has the effect of reducing fuel system heat rate, (2) the fuel heat rate is one-third to one-half that experienced in the operation of conventional fossil plants, and (3) decreasing the high-turbine inlet temperature decreases the fuel system heat rate. This latter observation is contrary to what is commonly observed in conventional gas turbine performance. However, it should not be unexpected. In the limit, if no fuel were added to the cavern air prior to expansion, power would still be generated due to a decrease in the available energy of the pressurized air. In such a situation, a fuel heat rate of zero would result at some reduced turbine inlet temperature. Obviously, the power generated in such a limiting case would be considerably reduced as compared to that generated by the addition of fuel.

Figure V-6 is presented with the recuperator effectiveness set at 80 percent. Figures V-7 through V-9 depict the effect of this variable on performance for high-turbine inlet temperatures of 1600, 2000, and 2400 F, respectively. Obviously, as recuperator effectiveness increases the fuel heat rate decreases. But, a high effectiveness indicates a larger and more costly recuperator. These heat exchangers have been discussed in a prior section of this report entitled Recuperator.

Round-trip heat rate

With the aid of Figs. V-5 and V-6, it is possible to estimate the round-trip or overall heat rate of CAPS. The round-trip heat rate (HHV) presented in Fig. V-10 includes the pumping energy plus the chemical energy added to the burners. For a high turbine inlet temperature of 2000 F and a cycle pressure ratio of 65, the round-trip heat rate reaches a minimum and has a value of about 12,000 Btu/kWhr (HHV). This is higher than the 10,000 Btu/kWhr of advanced state-of-the-art gas turbines, but better than the 13,000 to 15,000 Btu/kWhr for pumped hydro storage.

Output Power

The CAPS output power during generation mode is presented in Fig. V-11 as a function of cycle pressure ratio. As illustrated in the figure, power output increases with increasing cycle pressure ratio and increasing turbine inlet temperature. In a practical sense, since increasing the turbine inlet temperature corresponds to an advancement in the state-of-the-art (see Appendix B), the output at design conditions corresponding to existing turbomachinery could be increased by increasing the depth of storage within the limitation imposed by the local geology. The power estimates in Fig. V-11 are based on the gross electricity generated and have not been debited for house loads or transformer losses.

Parametrics Based on FT50

Ideally, equipment for CAPS should be assembled using off-the-shelf components. It should be noted, however, that although the required level of technology is within the current state-of-the-art, specific off-the-shelf equipment designed for CAPS does not exist at present in the U.S. Consequently, the design of existing components must be modified to reflect the unique requirements of CAPS. With this fact in mind, the preliminary parametric results presented in the previous section were repeated using the FT50 engine as the basis for the low-pressure turbomachinery. Specific characteristics for the FT50 are given in Appendix C. The most significant impact that use of this engine would have on CAPS would be to limit the temperature and pressure of the gas entering the low-pressure turbomachinery burner (station 14, Fig. V-1) to those conditions encountered at the inlet to the FT50 burner (approximately 750 F and 16 atm, respectively). Lower gas temperatures might be acceptable, but not higher temperatures.

With the FT50 turbomachinery fixing the low-pressure portion of the cycle, a series of calculations were made to match the remaining components of the system. The results of these calculations are shown in Figs. V-12 through V-15. For each of these figures, the gas temperature into the expansion turbine was selected to provide the proper turbine exit conditions to match the FT50. The expansion turbine inlet temperature increased from about 800 F for a cycle pressure ratio of 24:1 to about 1250 F for a cycle pressure ratio of 100:1.

Parametric Comparison

As mentioned previously, the performance characteristics derived from a consideration of the three sets of expander operating conditions display similar trends. Two of these characteristics, fuel system heat rate and output power, are compared in Figs. V-16 and V-17 respectively.

Looking at the fuel system heat rate comparison first (Fig. V-16), it is noticed that in all three instances the heat rate decreases with increasing cycle pressure ratio. For the first set of data (labelled I), the curve is steeper. This results from the fixed expansion turbine inlet and exhaust temperatures because the fixed expansion turbine pressure ratio forces most of the pressure drop at low pressure to occur in the less efficient expansion turbine ($\eta_{ad} \cong .8$) instead of the more efficient low-pressure turbomachinery ($\eta_{ad} \cong .9$). As the cycle pressure ratio increases, the role of the low-pressure turbomachinery in generating power increases, yielding increased efficiency. For the last two parametric sets the low-pressure turbomachinery pressure ratio was held constant at 16:1. The second set of data (labelled II) exhibits slightly better efficiency than the third set (labelled III) because the expansion turbine exhaust temperature (T_{14}), associated with the second set over the range of cycle pressure ratios, is higher than the fixed 750 F of the third set, resulting in a reduced fuel requirement to reheat the gas up to 2000 F.

In viewing the output power for all three sets of data (Fig. V-17) it can be seen that power increases with cycle pressure ratio. The data from the first and second sets tend to merge at high pressure ratio as the variable expansion turbine exhaust temperature in the second set approaches the 1000 F value of the first set. The output power data from the third set is lowest because of the decreased contribution by the expansion turbine resulting from its lower inlet temperature.

Despite these differences, it can be stated that adoption of design characteristics which simulate state-of-the-art equipment (the third set with a fixed low-pressure turbomachinery pressure ratio reflecting FT50 components and an expansion turbine pressure ratio reflecting vendor equipment availability and local geology) does not appreciably affect system performance.

Economic Optimization

The conceptual design, parametric performance, and equipment cost estimates previously described provide the basis for overall system economic optimization.

The object of the optimization study was to identify the CAPS configuration and operating conditions which would lead to lowest peak power cost for future commercial applications. This study involved trade-offs between above ground equipment and below ground reservoir sizes and costs.

Peak-Load Duty Cycle

In order to provide a meaningful basis for subsequent cost analyses, it was necessary to try to "second guess" the outcome of the program and postulate an appropriate duty cycle for CAPS. The duty cycle is important because it determines the required cavern storage capacity and the amount of off-peak compression that could be done on weekends using lowest-cost energy.

As noted in Ref. V-1, the extent to which energy storage capacity could be utilized in an electric utility system depends on such factors as: the utility system load characteristics, the system generation mix, the characteristics of the energy storage system (in this case, CAPS) and the economics of energy storage compared to alternative types of peaking and intermediate generating capacity. It is also noted in Ref. V-1 that approximately 50 percent of the off-peak energy for an average utility is available on weekends, necessitating consideration of a weekly cycle, i.e., charging to full capacity on the weekend and then discharging partially each weekday with some additional charging during the weeknight off-peak periods until the amount in storage reduces to zero by the next weekend. Examination of the weekly load variations in New England (Ref. V-2) tends to confirm this result.

In order to cover the range of operating cycles likely to be encountered (based on an analysis of the data in Ref. V-2) three nominal cavern storage capacities were considered: 6-hr, 20-hr and 40-hr (in Part VI a 10-hr capacity is also considered). The 6-hr capacity corresponds to the minimum storage capacity envisioned for CAPS operating on a daily charge/discharge cycle with no weekend charging (Fig. V-18). The 40-hr capacity corresponds to the maximum storage capacity envisioned for a CAPS operating on a weekly charge/discharge cycle and making full use of weekend pumping capacity. The 20-hr capacity is representative of a more practical case lying somewhere between the two extremes.

The nominal storage capacity labelling utilized in describing each of these operating cycles merely indicates the maximum magnitude of stored air. This occurs after charging over the weekend for the 20-hr and 40-hr storage capacities and after daily charging for the 6-hr storage capacity. It is expressed in hours of continuous generation time potential at design conditions after accounting for cavern air losses during charging and discharging. For example, in the 40-hr storage case, if no recharging were to occur during the week, there would be sufficient air in the cavern to provide 40 hours of generation time at design conditions after accounting for cavern leakage and absorption of air into the water. A schedule of the compression time and generation time are presented for the three operating cycles in Table V-2. The data in Table V-2 and Fig. V-18 is for a hypothetical utility which could only compress for about 6 hours each weeknight. In Part VI consideration is given to extending the weeknight comparison period to 10 hours.

Parametric Capital Cost Estimates

For each of the duty cycles described above, a series of parametric cost estimates was made for systems using the FT50 as the basis for the low-pressure turbomachinery (the third set of data in the previous discussion). Cycle pressure ratio (equivalent to cavern storage pressure, in atmospheres) was used as the primary parameter. A summary of the pertinent cycle performance parameters is given in Table V-3, and the characteristics of the storage reservoir are given in Table V-4. Included in the reservoir characteristics are total volume, including 10-percent capacity margin, and estimated cavern construction time including allowances for development and mobilization, shaft sinking, cavern examination, and completion.

The selection of the number of benches utilized (see Table V-4 and Fig. III-1) was based upon incorporating the greatest cavity height possible in order to minimize cavern excavation costs (see Fig. III-15) within the constraint of a 5-percent pressure variation in the cavern to minimize turbomachinery performance losses. In Fig. V-19 the cavern heights over which a 5-percent pressure variation will occur are indicated. Also indicated are the cavern heights which result from excavating the heading plus zero to three benches. By maximizing the number of benches within the 5-percent constraint, the least costly cavern results with minimum performance loss.

The estimates of capital cost for 6-hr, 20-hr and 40-hr storage capacity are shown in Fig. V-20. These capital costs are based upon a mid-1976 start date. The completion data varies depending on the depth of the cavern (cycle pressure ratio) and the storage capacity. This range of construction times is from 32 months to 65 months. The range has been accounted for in both the interest during construction and escalation. The characteristic feature of all three curves in Fig. V-20 is the decrease in capital cost with increasing overall pressure ratio (except for the 6-hr storage capacity which reaches a minimum around 60:1). The reasons for this decrease are displayed in the next three illustrations (Fig. V-21 through V-23). In these figures storage refers to the underground excavation costs; equipment to the aboveground facilities; balance of plant to the civil and electrical (excluding generator) components of the aboveground facilities; miscellaneous to the contingency, engineering and administration; and indirect to the escalation and interest during construction. It can be observed by viewing these illustrations that the reduction in storage costs is the main reason that the total installed cost decreases as overall pressure increases.

It can also be observed that the costs associated with the equipment and balance of plant are not strong functions of overall pressure ratio since they remain relatively constant across the range of storage pressures considered.

Three major items enter into the cost of storage: cavern excavation which is dependent on the volume excavated, shaft excavation which is dependent on the depth, and mobilization and installation which are fixed. In the 40-hr and 20-hr storage capacity, the cavern excavation costs are overriding because of the size of the cavern. Net cost savings still occur up to 100 atmospheres, despite the increased depth, because of the reduced cavern size and resulting reduced costs that the higher pressure permits. For the 6-hr storage capacity with its smaller storage cavity this effect occurs only up to 60 atmospheres whereafter any increase in depth produces a net cost increase because shaft excavation cost increases outweight cavern excavation cost reductions. The obvious conclusion from Figs. V-20 through V-23 is that high overall pressure ratios (above 50:1) must be used to achieve acceptable CAPS capital costs.

Parametric Power Cost Estimates

Parametric power cost estimates were made for each of the duty cycles. The annual fixed charge was varied between 15 percent and 20 percent, the fossil fuel cost between $2 and $5 per million Btu and the pumping energy cost between 5 mills/kWhr and 40 mills/kWhr. The range selected for the annual fixed charge reflected the summation of the reasonable variances which can be anticipated in the factors which comprise the annual fixed charge (i.e., return on investment, depreciation, taxes, insurance and administration). The fossil fuel cost range covers those values which are prevalent today on up to the values which could be anticipated in 1985 with an energy escalation rate of approximately 8 percent.

Both the fixed charges and fossil fuel rates may vary from utility to utility, however, within any given utility their values tend to remain relatively fixed (i.e., less than 20 percent variation) over an extended period of time. However, the value associated with the pumping energy is more mercurial, since it reflects the marginal costs associated with the power generation equipment supplying the pumping energy. The variability of this cost is illustrated in Fig. V-24 obtained from Ref. V-3. Displayed in the figure is a summary of the Pennsylvania, New Jersey and Maryland Power Pool (PJM) cost experience for the recent 52 week period ending with May 1976. The averaging of 250 (5 x 52) cost rates for each hour of the average weekday and of 52 rates for each hour of the weekend masks some important weekly, daily, and seasonal variations, particularly during on-peak periods. However, the off-peak period power cost are affected less by this averaging process, and the indicated durations and costs of the potential compression periods are believed to be fairly representative. Significant information about these potential compression periods is contained in this illustration. For example it can be estimated that 40 hours of off-peak compression energy could be supplied on a daily basis (8 hours on Monday through Friday) for an average cost of 17.6 mills/kWhr. The same amount of compression energy could also be supplied on a weekly basis (6 hours on Tuesday through Friday and 16 hours on the weekend) for an average cost of 14.7 mills/kWhr. The range of pumping power costs selected for the parametric analysis thus reflects this variation in generating mix and also the possible increase in marginal cost as both base-load nuclear and coal prices increase with time.

The power cost distributions for all three duty cycles are displayed in Figs. V-25 through V-27. In all three cases, the annual fixed charges account for the major proportion (at a minimum 45 percent) of the power cost. Since the pumping costs, fuel costs, and operating and maintenance costs remained relatively constant over the range of cycle pressure ratios considered, variations in the capital costs (from Fig. V-20) are the prime reason for the pronounced variation in power cost with increasing pressure ratio. A summary of the resulting busbar power costs is given in Fig. V-28.

Optimum Storage Pressure

Based upon the results of Figs. V-20 and V-28 for 6-hr storage capacity, it appears that a system with an overall pressure ratio of about 60 would lead to lowest capital and power costs. For the 20-hr and 40-hr capacity systems, these illustrations indicate that costs would continue to decline at least up to a pressure ratio of 100. Consequently it appeared that the range of optimum cycle pressure ratio considered in this study is 60-100 (corresponding to 2000-3400 ft depth). In any case, the optimum appears to be relatively flat, suggesting that almost any appropriate depth could be selected depending on local geological conditions.

The final selection of the overall pressure ratio depended on consideration of the capabilities of existing booster compressors and expansion turbines. In this consideration the principal concern centered on the expansion turbine, since several commercial models of centrifugal booster compressors are identifiable which could deliver up to 75-atm air at the required flow rate (see previous section in Part IV entitled Power Generation Equipment) at a cost of about $1 million. This narrowed the range of interest to 60-75 atmosphere. Upon further consultation with the expansion turbine manufacturers it was learned that one of the manufacturers, De Laval, has a steam turbine which could be modified to operate in CAPS with an overall pressure ratio of around 65, if two units were used in parallel to handle the flow (Ref. V-4). Armed with this knowledge, a standard industrial booster compressor was then selected which would produce an overall pressure ratio near 65. The resulting pressure ratio, 66.3, subsequently became the CAPS reference design operating point for the purposes of this study.

Selected CAPS Operating Characteristics

The preceding parametric performance and economic optimization studies provided the rationale for selecting a CAPS reference design point based on the use of FT50 components for the low pressure turbomachinery and compressed air storage at 66.3 atm. This section summarizes the design and off-design performance of CAPS. The specific system design characteristics are described in the next section.

Design Point Performance

A detailed heat and mass balance for the selected CAPS design conditions is given in Table V-5. Station numbers noted in the table correspond to locations identified in Fig. V-1. The estimated gross output from a single-unit system is 254.3 , and the fuel heat rate is 4129 Btu/kWhr (LHV). Key performance parameters for this system are summarized in Table V-6.

Also shown in Table V-6 is the performance of the selected system operating without air storage as a conventional simple cycle gas turbine. In this operating mode, only the low pressure ratio turbomachinery (cycle pressure ratio = 15.7:1) and recuperator would be operative; the booster compressor, aftercooler, cavern storage, and expander turbines would be uncoupled from the system. The remaining turbomachinery is essentially the FT50 open-cycle gas turbine. Since the turbine components must provide the power for air compression, the net output would be only 77 MW.

Off-Design Operation

The selected CAPS configuration is composed of existing state-of-the-art components which have specific design characteristics. During the operation of CAPS, however, it might be desirable to operate the system at other than design conditions to meet a particular utility operational characteristics. Perhaps the most obvious design excursion effecting system performance is a variation from the 1:1 ratio of charge time to discharge time. The charge time corresponding to a large axial-flow, fixed-geometry, constant-speed compressor could not be altered significantly because these machines can operate only over a narrow range of airflow. The use of variable geometry inlet guide vanes would extend this range slightly. Turbines, however can operate over a wide range of airflow by throttling the inlets. This fact could be used to extend the generation time, at reduced output, and thereby reduce the charge/discharge time ratios. An alternative approach would be to add additional compressors in parallel to reduce compression time or add additional turbines in parallel to reduce generation time. This approach would be relatively expensive because of the extra equipment. Furthermore, the extra equipment could not be used in the simple-cycle mode to generate emergency power. It is anticipated, therefore, that off-design performance could best be achieved by retaining the basic CAPS configuration selected and simply varying the output of the turbine to meet the different charge/discharge ratios which might be required in the utility application. The following paragraphs describe the off-design performance characteristics of CAPS utilizing this approach.

Off-Design Constraints

There are two principal means for varying the output of CAPS: vary the turbine airflow rate and vary the turbine inlet temperature. These variations can only be accomplished within limited design constraints. The turbine airflow rate can be reduced by throttling, but because of the choked flow condition in the turbine vanes the airflow can not be increased beyond the physical limitation imposed by the design of the associated piping and control system. The selected CAPS design point sited in Table V-5 is based on an airflow rate which is approximately 96 percent of the original FT50 design value. (Ninety-six percent was used to nominally adjust for cavern leakage yet still maintain a 1:1 charge/discharge time ratio.) The FT50 has the capability of handling an airflow rate approximately 10 percent above its original design point at maximum peak loading; consequently the selected CAPS design could handle an airflow ratio almost 15 percent above the CAPS design value.

The recuperator places a lower limit on the degree to which the airflow rate could be decreased. When operating in a part load condition, less energy is extracted from the gas flow by the turbine, consequently, the temperature exiting the turbine, which is the temperature entering the recuperator (T_{17}) increases. The recuperator was designed with low-carbon steel which has a maximum metal temperature of about 800 F. The convective and conductive heat transfer coefficients are such that the maximum metal temperature will be exceeded in the hot-side recuperator inlet, if T_{17} is greater than about 850 F. The effect of these two constraints on the allowable relative airflow rate operating regime (CAPS design airflow at 731 lb/sec equals unity) is presented in Fig. V-29. Also shown in this illustration is the effect of turbine inlet temperature (T_{16}) on the operating regime. As would be expected, a lower T_{16} would normally produce lower turbine exhaust temperature; therefore, a greater range of airflow rate is permissible before the maximum exhaust temperature is exceeded.

Off-Design Performance

Airflow discharge rate can be controlled by use of a throttle valve. At constant T_{16} the output power decreases approximately linearly with airflow as is shown in Fig. V-30. At constant airflow the power decrease is approximately proportional to the temperature drop across the turbine.

Off-Design Fuel System Heat Rate

The effect of discharge airflow rate of fuel system heat rate is nominal over a large range of relative airflow as is shown in Fig. V-31. At constant T_{16}, decreasing the airflow increases the heat rate due to a moderate decrease in turbine component efficiencies at the reduced flow. The heat rate decreases slightly at constant airflow with decreasing temperature as was observed in the parametric studies.

It is possible to crossplot the data presented in Figs. V-30 and V-31 to yield the results of Fig. V-32. It should be noted that at constant T_{16} it is possible to vary the output power over a wide range, yet still produce little effect on the heat rate. It would also appear subject to the airflow constraint, that it is possible to maintain the same output power and reduce heat rate by merely reducing T_{16}. However, by adding the two lines of constant airflow, it can be seen that in order to achieve such an effect it would be necessary to increase airflow. Obviously, this would decrease the available generation time.

The FT50 simple cycle gas turbine performance characteristics have also been added to this crossplot. It can be seen that a decrease in output power is accompanied by a dramatic increase in heat rate. This occurs in simple cycle operation, even though the turbine output drops off approximately linearly with airflow rate, because the compressor power remains relatively insensitive to airflow rate resulting in a reduced net output. In comparing the simple cycle gas turbine and CAPS it can be concluded that CAPS has far better off-design performance characteristics.

Charge/Discharge Time Variations

The major objective in discussing the off-design performance of the selected CAPS design has been to provide an indication of the performance of the selected system in a utility environment where the ratio of charge time to discharge time is different from unity. The trends displayed in the previous illustrations as a function of relative airflow can be related to the charge/discharge ratio as shown in Fig. V-33. It can be observed from this figure that the output power reduction is approximately linear; therefore, about the same number of kWhrs of electrical energy will be produced. In addition, there is little change in heat rate with charge/discharge ratio.

From the proceeding discussion it can be concluded then that there is little or no penalty in running the selected CAPS in the off-design mode subject to the turbomachinery and recuperator constraints. It can also be concluded that off-design characteristic of CAPS are superior to the off-design characteristic of the open cycle gas turbine upon which the system is based. Lastly it can be concluded that the selected CAPS could be introduced into different utilities with wide variations in operating characteristics without suffering performance penalties.

COMPRESSED AIR POWER SYSTEM PLANT DESIGN

This section deals with the conceptual design, layout, and costs estimates for compressed air power system (CAPS) plants in hard rock using hydrostatic compensation to maintain the full storage pressure. All design conditions, including the storage pressure of 66.3 atm are based on the results of the system optimization discussed in the previous section. Storage capacity of the plant was varied between 6 and 40 hours at rated load for single- and multiple-unit plants to determine the effect on plant cost.

Plant Design

A base plant design capable of operating on a weekly cycle was chosen so that low-cost power available during the weekend could be utilized for charging up the storage cavern. A single unit with 20-hr storage capacity was selected for this base design so that parametric variations of lesser and greater storage could be evaluated. The parametric variations examined include:

Base Case	1 unit with 20-hr storage capacity
Storage Alternates	6 hours (daily cycle) and 40 hours (full weekly cycle)
Unit Alternates	2 units and 3 units

The design aspects of each major component of these plant are briefly discussed in the following subsections.

Primary Power Equipment

The primary power equipment and turbomachinery related components are identified in Fig. V-1. A listing of the major components, together with supplier and model designation, are presented in Table V-7. Discussions have already been presented for the low-pressure machinery (Part IV and Appendix C), booster compressor (Part IV), expansion turbines (Part IV) clutches (Part IV and Appendix F) and recuperators (Part IV). Consequently, only a few summary comments about the actual devices utilized in the plant design are included below.

Low Pressure Turbomachinery

The selected low-pressure turbomachinery is based on the United Technologies FT50 industrial gas turbine. (See Fig. IV-2.) Its two compressor sections (Fig. IV-3) with 7 stages and 10 stages, respectively, and one stage of intercooling will compress 761.8 pounds of air per second from ambient up to 15.7 atmospheres. The compressor sections will require 123.4 MW of shaft power to drive them. They will be mounted on a common shaft with the booster compressor, motor/generator, low-pressure turbine sections and three clutches. The three turbine sections contain 1-stage, 1-stage, and 2-stages (Fig. IV-4) respectively and will generate 200.3 MW of electrical power in expanding a total of 735.4 lbs/sec of combustion gases from 15 atmospheres to a little over ambient pressure.

Booster Compressor

The booster compressor is a six-stage Ingersoll-Rand, 687RE vertically-split, multistage, centrifugal compressor (see Fig. IV-5) with a single stage of inter-cooling located after the third stage. It would be preceded by an intercooler and followed by an aftercooler. The booster compressor will consume 76.5 MW of shaft power to compress 761.8 pounds of air per second from 15.4 atmosphere to 62.9 atmospheres (the pressure head of air in the vertical air shaft will raise the air to the net storage pressure of 66.3 atmosphere). It will require 76.5 MW of electrical power.

Expansion Turbine

The two parallel-flow expansion turbines are modified 13-stage DeLavel YJ steam turbines. Each will handle 298.8 lbs of air per second and generate 27.0 MW of power over a pressure range from 58.4 atm to 15.9 atm.

Clutches

SSS or MAAG clutches could be used to transmit power either from or to the motor/generator. The SSS clutch designation is 280T while the MAAG clutch designation is MS-85. The booster compressor clutch could also be a SSS 280T with a single set of clutch teeth as compared to the two sets in the larger clutch.

Recuperator

Two recuperators would be used, each with 260,000 sq ft. They would be of shell-and-tube design with three shells and two passes per shell. The high-pressure air will be inside the tubes. The first two shells and about half of the third shell would be constructed of conventional finned tubes. A protective corrosion-resistant coating would be used on the outside of the tubes in the last half of the third shell.

Motor/Generator and Generator

The selection of a motor/generator supplier was limited by the disinterest of U.S. manufacturers to supply the required equipment. As a result, overseas manufacturers were contacted, and the equipment offered by Brown Boveri Corporation was selected as the basis for the conceptual design. The nominal 200-MW motor/generator offered is rated at 225 MVA, 17 kV, with hydrogen cooling and static excitation. The 54-MW generator for the expansion turbines is rated at 60 MVA, 13.8 kV, with water cooling and shaft excitation.

Supporting Equipment and Facilities

Intake Filters and Silencers

The selection of intake filters for the conceptual design was based upon the filtering specification for the standard FT50 gas turbine package (Ref. V-5). This specification requires a filtering performance of 99.7 percent particulate removal at 5 microns and 95 percent particulate removal at 2 microns, for a total flow rate of approximately 600,000 cubic feet per minute. Proposals for a two-stage pleated fabric filter cartridge system including moisture eliminators were obtained. The silencing system was based on a modular arrangement of cells designed to attenuate the sound pressure level emitted from the compressor intake. The system was divided to provide a convenient double intake arrangement.

Exhaust Silencer and Stack

The design concept for the exhaust silencer and stack was in effect to have the silencer perform both functions. The height of the recuperator exit is 37 feet above grade and the exhaust silencers are 40 feet in height, with a transition duct length of approximately 15 feet, for a total elevation at the silencer exit of 92 feet. The silencer units are supplied by Burgess Environmental Systems Division and fabricated of stainless steel with a Corten steel outer skin. The expected life of these units is 15 to 20 years.

Fuel Storage Tanks and Transfer Facilities

The main fuel storage tanks were sized on the basis of 40-days storage for the plant when operating at a nominal fuel heat raee of 5000 Btu/kW hr. The tank design selected is the steel, floating roof type, provided with rock and earth dikes ten feet in height. Two tanks were specified to allow operation of the plant in the event of tank maintenance.

Fuel from the transfer pumps discharge into an underground fiberglass wall tank sized to provide 40 minutes fuel supply at plant load. Fuel is pumped from this tank by DeLaval constant displacement (Imo) pumps (two operating, one standby) at 60 psig into the primary burner loop, and is extracted from the loop by an Imo pump delivering 35 gallons per minute at 1200 psig for the high pressure burner circuit. Fuel not consumed by the burner spills into the primary supply loop return through a pressure reducing valve. The low pressure burner pumps also operate from the primary loop, and are supplied with the basic FT50 turbine package.

Compressor and Oil Cooling System

The initial compressor cooling system concept utilized water as the cooling medium with heat dissipation from either the compensation reservoir, cooling towers, or sprays. Water cooling was discarded for the following reasons:

(a) Cooling towers occupy additional space and consume considerable quantities of makeup water.

(b) The blowdown water also presents a treatment problem.

(c) Utilizing the compensation pond for cooling raised problems of algae production and blowdown to maintain concentration levels of solids in the water.

(d) The use of spray units only increased the evaporation losses and blowdown loads projected for a cooling reservoir.

The conceptual plant design presented herein is, therefore, based on the use of direct air-to-air heat exchangers for intercooling and aftercooling during the compression phase of the plant cycle (Fig. V-34). To simplify the design of piping, relief valves and other pressure vessels, the low pressure and intermediate pressure intercoolers are both designed for an internal pressure of 250 psia. This additional pressure incurs no cost penalty as the added cost of construction for higher pressure more than offsets the additional costs of providing pressure relief and silencing systems in the low pressure intercooler circuit. The Ecodyne MRM air-to-air mechanical draft finned tube intercoolers were located on the machine house roof to reduce piping run lengths and provide an adequate air flow for cooling (Fig. V-35). The main motor/generator and expander turbine generator both require cooling water to dissipate thermal losses. The use of either a cooling tower or a circulating water loop from the compensation pond was examined, with the latter selected. The system supplies the estimated requirement of 4300 gpm using two operating pumps (one standby) fed by a pipe with a flow control valve used to compensate for the variable head conditions. Cooling water is supplied to both generators, and to the lube oil cooler. This last item is air cooled on the standard FT50 package, but with the increase in plant machinery, air/water heat exchangers were assumed. Main piping to this equipment from the pumphouse is 16 inches in diameter, rubber-lined schedule-40 steel.

Piping

The main piping between compressors (refer to Fig. V-34) was sized to comply with the allowable pressure drop between sections using the Weymouth formula. This resulted in piping diameters of 52 and 50 inches between the low and high compressors, 42 and 40 inches between the high and first booster section compressors, 32 and 30 inches between the first and second booster sections, and 24 inches to the aftercooler. No isolation valves were provided in these pipe sections, but relief valves were included for protection from overpressure. Relief valves were selected from Anderson-Greenwood, and vent silencers from Burgess-Manning. Allowances were made for piping thermal expansion but detail studies were not performed.

Water condensation during the compression phase will approach 45 gallons per minute. Moisture separators of the type used in the gas industry have been incorporated in the piping between each intercooler and compressor and downstream of the aftercooler. These are required to reduce erosion damage of compressor blading and piping. The units incorporated in the conceptual design are manufactured by Perry Equipment Company.

The surface piping to the cavern from the aftercooler discharge was sized for the allowable pressure drop at 28 inch outside diameter, 1.125 inch wall thickness. The turbine supply pipe from the cavern shut-off valves was sized at 32 inches, 1.25 inch wall.

The vertical shaft pipe to the cavern is supported by a lower anchor and expansion is upwards through guides in the air pipe shaft. To prevent overstressing of the pipe at the shaft collar due to thermal expansion in the shaft, 300 feet of horizontal pipe was allowed between the shaft collar and anchored shut-off valves. Alternative methods considered for absorbing the thermal expansion were:

- Sliding packed expansion joints
- Ball and socket hinge joints

These alternates were discarded due to the requirements of maintenance shutdowns and discontinuity of the pipe wall.

The cavern shutoff valves selected were two parallel 16-inch Fisher "hi-ball" control valves. Check valves were included for reverse flow prevention. The bypass line, for simple cycle operation, between the high compressor and recuperators was sized at 42-inch diameter. The bypass lines between the recuperators and the high turbine was sized at 20 inches. Allowances were made for gate valves in these lines.

Instrumentation and Control

There are a number of major control systems external to the flow and speed controls for the main machinery:

(1) Cavern/reservoir level control

(2) Piping overpressure/failure relief system

(3) Cooler fan sequencing.

The cavern/reservoir level control system has two functions. The first involves an instrumentation system which integrates the water levels in the upper and lower reservoirs to determine the requirement for makeup water. The second is water level indication and plant shutdown capability, in the event that either the cavern or reservoir water levels are dangerously low. This system must be highly reliable, as unscheduled repairs to the storage cavern instrumentation are an exceedingly expensive proposition.

Apart from conventional temperature, pressure and flow instrumentation and remote operating valves, the auto/manual relief valve located in the turbine air supply line deserves mention. This valve will normally relieve above the full cavern operating pressure in the event of a valve failure downstream, but must also be operable in the simple cycle plant mode. In this latter case, the design pressure is approximately one-fourth of normal operating conditions, leading to the use of either a controller for valve operation, or a switching valve system with relief valves set at different pressures.

The conceptual study did not investigate the economic tradeoffs of continuous or temperature controlled cooling system for operation. If temperature control is desirable, an additional system for sequential startup/shutdown of cooler fans will be required.

Instrumentation and control of the main turbomachinery is significantly more involved than a simple cycle gas turbine. Control systems for this equipment must handle the differing flow characteristics with speed of axial and centrifugal compressors and expanders and fired turbines; exit temperatures from the expander turbines and the low turbines; and inlet pressure under all conditions of startup, part-load, full-load and simple cycle operation. Approximate allowances were made in the cost estimates for such a control system, but no attempt was made to design the system as this was clearly beyond the scope of this study.

Electrical Equipment - Single Line Diagram

General Arrangement

The proposed single-line layout arrangements of electrical equipment, i.e., generators, step-up power transformers, high voltage switchyard and equipment to supply auxiliary power, are shown in Figs. V-36 and V-37. Figure V-36 illustrates the arrangement for a single-unit installation, and Fig. V-37 shows the arrangement for a three-unit installation. The arrangement for a two-unit installation would be similar to that in Fig. V-37. The basic difference in the two arrangements is in the electrical layout of the high voltage equipment as explained in the next section. In both arrangements, the important considerations are security and flexibility of operation of the system.

Each 200-MW motor/generator set and 54-MW generator is connected to a step-up transformer on a "unit system" basis. The 200-MW motor/generator set will be switched on the high voltage side. The 54-MW generator will have, in addition to the high-voltage switching, its own indoor-type generator circuit breaker to facilitate the supply of auxiliary power. In both cases, the connections between each generator and the generator transformer, located outside in the yard, will be by means of naturally cooled 14-and 18-kV isolated-phase bus duct.

High Voltage Switchyard

Main power switching will be performed in an outdoor switching station adjacent to the power house. A 230-kV high-voltage system has been considered based on an assumed 230-kV transmission network existing nearby into which the power output of the plant under study will be fed.

Two types of 230-kV circuit breakers can be considered:

(i) oil circuit breaker, or
(ii) SF6 gas outdoor type circuit breaker.

Generally speaking, the oil circuit breaker is less expensive and is field-proven. On the other hand, the more modern SF6 gas type of circuit breaker has a faster operating time (typically 2 cycles against 3 cycles for an oil breaker) and is considered to require relatively reduced maintenance. At this time, no attempt has been made to make a final selection and recommend the type of circuit breaker. For the purpose of this study and reasons of economy, the cost estimates and the layouts are based on the use of oil circuit breakers. Eventual selection will, no doubt, depend on the preferences of the individual utilities.

Referring to Figs. V-36 and V-37 for parametric layout, it can be seen that the switchyard for the single unit installation is in the form of a "ring-bus", and for two- and three-units the arrangement is "breaker-and-one-half". The number of major items is summarized in Table V-8.

Auxiliary Power Supply

Power to pony starting motors, cooling fans, station service and auxiliaries will be supplied from the low voltage side of the 54-MW generator-transformer and, where available, from a local feeder. Reliability and economy are the main considerations. The pony starting motors will be supplied from a 4,160-volt bus. The 13.8-kV disconnects and structure and the step-down transformers will be located outside adjacent to the power house.

Plant Structures and Auxiliaries

Plant Buildings

The main powerplant building is of the full-enclosure type, with sectional steel siding and complete facilities for the operating staff within. The main building dimensions for each unit, exclusive of the auxiliary equipment and expansion turbine rooms, are 500 feet by 335 feet, with a height to the top of the roof mounted intercoolers of approximately 78 feet. The building houses the plant auxiliary equipment in rooms contained in small building wings, and most of the large diameter plant piping runs are below the operating floor level in the basement area. The turbomachinery is serviced by an overhead travelling crane with a capacity of 50 tons. An enclosed maintenance area 65 feet in length has also been allowed for at the expansion turbine end of the building for each unit. The control room is located in a building wing near the midpoint of the turbomachinery at the operating floor level. For the two- and three-unit plants, control of all units has been consolidated into a single room near the midpoint of the plant building.

The main turbomachinery foundation pedestals are carried on a continuous slab to minimize the potential for relative movement (Fig. V-38). This slab is separated from the building footings. The expansion turbines are mounted on a separate foundation as they are not subject to alignment with the main shaft.

Pumphouses

There are three pumphouses on or near the site - one each for the makeup water pumps, generator/lube oil cooling system pumps, and the fuel transfer pumps. All are minimum structure steel siding type buildings with a minimum of services.

Miscellaneous Site Buildings and Structures

Miscellaneous buildings on the site include the four-bay vehicle garage and the guardhouse. The latter is a brick structure with complete facilities for the plant guards, and includes the site security system monitoring equipment. Other structures on the site are the 50,000-gallon fire tank, foam tank, and a 20-meter meteorological tower.

Auxiliaries

The plant building fire extinguishing system is comprised of a perimeter hydrant loop and a high pressure carbon dioxide system for the main machinery, control room, and auxiliary equipment. A separate CO_2 system has been allowed for each unit in the plant.

The turbomachinery auxiliaries included in the design are the high pressure lubrication oil system (expanded from the base system supplied with the FT50 turbine package), the turbine wash system, and the compressor heating system.

Noise Control Facilities

Requirements for sound control were determined from the tentative OSHA standards of 85 dBA continuous in the working environment and the EPA standards of 45 dBA at the property line. Most of the plant noise would emanate from the turbomachinery. Estimated sound pressure levels from the main low-pressure turbomachinery are presented in Table V-9.

The turbomachinery will undoubtedly require sound absorbing enclosures in order to comply with the above OSHA standard. The intake plenum, as the plant is laid out, may require a noise absorbing outer skin in the vicinity of the operating floor to maintain a sound pressure level of 85 dBA at a distance of three feet from the ducting. Silencers are installed in both the intakes and discharge of the turbomachinery, as described previously.

External to the plant building, the plant layout incorporates a perimeter dike extending from the plant side of the reservoir, which completely encloses the plant, switchyard, and fuel storage area. This dike is tentatively equal in height with the surface reservoir dikes and has a threefold purpose. First, the dike is primarily an aid in the control of general site noise, particularly from the plant air intakes and exhaust. It will deflect noise upwards and provide greater distances for the dissipation of sound pressure before intersecting the off-site ground reception level. This "free-field" effect attenuates noise by 6 dBA for each doubling of the distance from the source to the receiver. Second, the dike will provide aesthetic treatment of the plant; and third, it will aid in the disposal of rock spoil from the underground excavation.

Site Layout

The plant has been laid out in a rectangular plan, surrounded by a continuous dike except at the entrance (See Fig. V-39 for the single-unit plant and Fig. V-40 for the three-unit plant). The main items of the site are the reservoir, equipment area, water and air shafts, oil storage tanks, switchyard and transformers and the main building.

Surface Plant Layout

A number of building layouts were possible, ranging from a full outdoor plant to a totally encapsulating masonry structure. The former did not seem appropriate for a plant site in the North Central or Atlantic Coastal regions. The latter appeared excessive. The concept of the plant layout was that the building form should be adequate to completely enclose the main turbomachinery. Initially, one expander turbine was expected for the design (Ref. V-6). When it was realized that two would be required, two wings were added to the equipment building in order to house the expanders. The long building (Fig. V-41) is a result of the basic length of the turbomachinery and shafting, which could not be positioned any other way (Fig. V-38). This results in a long machine hall with two generators which, unlike large conventional steam turbine sets, are positioned at both the end and the middle of the machinery.

Layouts of the two- and three-unit plants would have resulted in a more conventional floor plan had the machinery been set parallel. However, this presented problems with routing of large piping and bus ducts, and did not appear to be efficient in land use, given the substantial reservoir size, the switchyard requirements, and the fuel storage area. Therefore, the units were positioned end to end, forming a continuous machine hall and sharing a single high-capacity, low-usage crane. This made pipe routing problems relatively simple, and facilitated layout of the transformers and switchyard. In addition, the intercoolers and vent silencers could be mounted on the roof, and the quantity of large diameter piping required was significantly reduced.

Underground Facilities

The underground storage facilities are typically vaulted roof chambers, with height, width, and arch radius determined by rock properties (Fig. V-42). For general siting purposes, these have been assumed at 85-foot floor-to-top of arch height, 60-foot width, with a 30-foot radius for the arch. As discussed in Part III of this report, the storage cavern is excavated by a heading and benching method, as opposed to the typical room and pillar mining method used in ore seams. Multiple chambers are required if the storage volume is expected to fit within the site boundaries on the surface. This is accomplished using a finger-like arrangement of the storage caverns (See Figs. V-39 and V-40). The design of such caverns requires a geological survey using core drills to determine rock properties. Analysis of stresses is then performed in the design phase to determine optimum cavern shape and to minimize rock bolting.

Cavern access for construction is by the water shaft, through a heading driven across to the cavern at a point above the bottom of the shaft. Initial development is performed through this heading, while the muck chute and hoist loading station are prepared. Rock crushing and screening equipment is located in a room near hoist skip loading point. When construction is finished, the muck chute and machinery room become part of the U-bend for control of the champagne effect.

There are two shafts connecting the underground storage cavern to the surface facilities. The compensating water shaft is sunk with a diameter of 12 feet, and for all variations is 2500-feet in depth. The shaft is also concrete lined for the full depth.

The air pipe shaft is first drilled to the 2250-foot cavern depth and then raise bored from the cavern to the surface to the required diameter. Final inside diameter is 6 feet for the single unit or 8 feet for multiple unit plants, after the lining has been placed.

At the base of the air pipe a keyed concrete plug is poured to anchor and support the base of the pipe and seal the air in the cavern from escape to the open air shaft. The bottom of the plug is flared to gradually reduce the velocity of the air entering the cavern.

Compensation Reservoir

The surface reservoir is assumed to be built on an essentially flat area of land utilizing crushed rock from the underground excavations to form the ring dikes. Earth removed during site preparation is spread over the rock surface on the inner face of the dikes, and an impermeable liner is installed to prevent any leakage from the reservoir. The liner is tentatively an EPDM synthetic material, but this should be optimized in a final design.

The design variation in water level is 20 feet, and with five-feet minimum depth and five feet of freeboard, this results in a dike height of 30 feet. The variable capacity of the reservoir is a minimum of 10 percent greater than the cavern volume to ensure that the shaft will remain completely filled at all times.

Cost Estimates

Capital Costs

The capital cost estimates were prepared using quoted and estimated prices from manufacturers, prices for similar equipment used elsewhere, cost estimation using similar equipment of different size or capacity, and in some cases, engineering judgment. The plant estimates were compiled using a code of accounts based on the Federal Power Commission Code. All costs are in June 1976 dollars.

The capital costs for one- two-, and three-unit plants are summarized in Tables V-10, V-11, and V-12 for 6-hr, 20-hr and 40-hr storage capacities, respectively. The engineering and construction management figures shown in the tables are rough allowances rounded to the nearest million. Estimated variations of these costs due to construction schedule did not appear to have any significant effect.

The specific capital costs per unit of net power output are presented in Table V-13. It will be noticed that the single unit output does not agree exactly with the value presented in Table V-6. This deviation merely reflects the house loads on the system. These loads are itemized in Table V-14. For multiple units the house loads were considered to be proportionate to the number of units and the single unit estimate was simply multiplied by the number of units.

Operating and Maintenance Costs

The operating and maintenance costs of small power plants are significantly affected by company and union policies, local ordinances, and types of fuels used. The estimate of O & M costs made during this study did not include any parametric variations due to these effects. Presentation of any estimated operating costs must necessarily be qualified by listing the assumptions made. Therefore, the entire estimate is reproduced in Tables V-15 and V-16. Variations in cost were attributed to the number of units, and not to storage capacity. The summary of O & M costs is given in Table V-17. Total annual O & M costs range from $7.18/kW yr for a single-unit plant to $4.29/kW yr for a three-unit plant.

REFERENCES

1. Sulzberger, V. T., and J. Zemkoski: The Potential for Application of Energy Storage Capacity on Electric Utility Systems in the U.S. - Part I. IEEE Winter Power Meeting, January 1976.

2. Smith, J. R.: New England Power Planning (NEPLAN) Correspondence with E. B. Smith of UTRC. March 16, 1976.

3. Loane, E. S.: GPU Service Corporation Correspondence with A. J. Giramonti. July 14, 1976.

4. Redmond, K. E.: Delaval Correspondence with W. R. Davison of UTRC. August 11, 1976.

5. Dvorak, H. G., and W. H. Rogers, Jr.: An Advanced Industrial Gas Turbine for Utility Generation. American Power Conference, Chicago, April 1975.

6. Giramonti, A. J.: Preliminary Feasibility Evaluation of Compressed Air Storage Power Systems. UTRC Report R76-952161-4. May 1976.

TABLE V-1

CAPS PARAMETRIC PERFORMANCE ANALYSIS PARAMETER VARIATIONS

Turbine Inlet Temperature, T_{16}, F	1600, 2000, 2400
Cycle Pressure Ratio (at storage), P_9/P_0	15-100
Recuperator Effectiveness, %	0-80
Cavern Leakage, %	0-10
Maximum Compressor Discharge Temperature, T_2, T_4, T_6, T_8, F	500
Storage Temperature, T_{10}, F	120
Compressor Airflow, lb/sec	761.8

TABLE V-2

CAPS OPERATING CYCLES

Nominal Storage Capacity, hr	6	20	40
Compression Time			
hr/weeknight	6.2	6.2	6.2
hr/weekend	6.5	21.5	43.0
Generation Time			
hr/weekday	6.2	8.8	12.5
hr/week	31.2	43.8	62.5
hr/year (1)	1560	2190	3125
Annual Load Factor, %	17.8	25.0	35.5

(1) 50 weeks/yr

TABLE V-3

CAPS PARAMETRIC CHARACTERISTICS

Low-Pressure Turbomachinery Based on FT50 Design

Cycle Pressure Ratio	24.2	30.1	48.6	60.2	78.7	98.3
Power (1), MW	209	220	244	256	271	284
Specific Cavern Volume, ft^3/kWhr	7.90	6.06	3.38	2.60	1.88	1.44
Heat Rate (LHV), Btu/kWhr	4245	4211	4147	4121	4089	4065
Pumping Energy (HHV) Btu/kWhr	7611	7824	8033	8044	8315	8426

(1) Based on 4-percent leakage and 80-percent recuperation effectiveness.

TABLE V-4

CAPS STORAGE RESERVOIR CHARACTERISTICS

Cycle Pressure Ratio	24.2	30.1	48.6	60.2	78.7	98.3
Depth, ft	820	1020	1648	2040	2668	3332
Cavern Height, ft						
5 percent of depth[(1)]	41	51	82	102	133	166
Value used	55	55	85	115	115	115
Number of Headings	1	1	1	1	1	1
Number of Benches	1	1	2	3	3	3
Cavern Volume[(2)], 10^3 cu yd						
6-hr Storage	404	326	202	163	125	100
20-hr Storage	1348	1085	672	543	415	332
40-hr Storage	2696	2170	1344	1085	831	665
Construction Time[(3)], months						
6-hr Storage	32	32	32	34	35	38
20-hr Storage	46	45	42	42	43	45
40-hr Storage	56	55	51	51	50	52

(1) Corresponds approximately to 5 percent pressure variation in cavern

(2) Includes 10 percent capacity margin.

(3) Includes allowances for development and mobilization, shafts, cavern excavation and completion.

TABLE V-5

HEAT & MASS BALANCE FOR SELECTED CAPS DESIGN CONDITIONS

Station No.[1]	1	2	3	4	5	6	7	8	9	Cavern
W, lb/sec	761.8	761.8	761.8	761.8	761.8	761.8	761.8	761.8	761.8	761.8
T, F	59.0	345.0	161.0	500.0	200.0	377.6	200.0	409.4	120.0	120.0
P, psia	14.7	60.2	55.8	230.4	225.8	440.3	430.1	925.2	906.7	973.7
h, Btu/lb	124.0	193.0	148.5	231.1	157.9	201.0	157.9	208.8	138.7	138.7

Station No.	10	11	12	13	14	15	16	17	18
W, lb/sec	594.6[2]	594.6	3.05	597.6	597.6	12.72	610.3	735.4	735.4
T, F	120.0	643.3	-----	1000.0	661.5	-----	2000.0	774.0	371.5
P, psia	902.7	858.8	-----	807.3	234.0	-----	220.0	15.6	14.9
h, Btu/lb	138.7	266.6	-----	360.4	272.5	-----	655.4	305.7	202.3

Compressor Power, Mw

LC	56.2
HC	67.2
BC	76.5
Total	199.9

Turbine Power, MW

ET	54.0
FT50 Turbines	200.3
Total, gross	254.3

Heat Exchanger Heat Transfer, Btu/Sec

IC1	33901.
IC2	55727.
IC3	32819.
Aftercooler	53398
Recuperator	76023
(Effectiveness - 80%)	

(1) Station numbers refer to locations identified in Fig. V-1

(2) Plus 136.7 lb/sec of cooling air for total of 731.3 lb/sec withdrawal rate. This is 4 percent less than compressor flow rate to adjust for cavern leakage.

TABLE V-6

PERFORMANCE SUMMARY FOR SELECTED CAPS POWER PLANT

	Normal CAPS Operation	Emergency Simple-Cycle Operation
Cycle Pressure Ratio	66.3	15.7
High Turbine Temperature, F	2000	2000
Expansion Turbine Temperature, F	1000	--
Recuperator Effectiveness, %	80	80
Cavern Leakage, %/day	4	--
Specific Turbine Flow, lb/kWhr	8.46	10.84
Specific Storage Volume, ft^3/kWhr	1.857	--
Specific Compressor Energy, kWhr/lb	.0729	.0450
Pumping Power, MW	200.0	--
Pumping Energy Rate, Btu/kWhr (HHV)	8189	--
Fuel System Heat Rate, Btu/kWhr (LHV)	4129	10081
Round-Trip Heat Rate, Btu/kWhr (HHV)	12565	10685
Gross Power Output, MW	254.3	77.1

TABLE V-7

CAPS COMPONENTS

Component	Supplier	Designation
Low-Pressure Turbomachinery	United Technologies	FT50
Booster Compressor	Ingersoll-Rand	687 RE
Expansion Turbine	Delavel	YJ
Clutches	SSS MAAG	280T MS-85
Motor/Generator	Brown Boveri	---

TABLE V-8

MAIN HIGH-VOLTAGE EQUIPMENT/CIRCUITS

Item No.	Type of Equipment/ Circuit	Equipment/Circuit - Quantity One-Unit	Two-Units	Three-Units
1	230-kV Oil Circuit Breakers	4	12	15
2	230-kV Disconnect Switches	12	32	40
3	Step-Up Power Transformers	2	4	6
4	Number of Overhead Lines	2	4	4
5	Total Number of 230-kV Circuit Entries	4	8	10

TABLE V-9

ESTIMATED SOUND PRESSURE LEVELS FROM CAPS LOW-PRESSURE TURBOMACHINERY

Sound Pressure Level, dBre 10^{-12} Watts

Octave Band Center Frequency, Hz

Inlet	104/109	107/112	113/118	116/121	119/124	144/149	142/149	138/143	136/141
Case	97	108	122	124	115	125	121	126	122
Exhaust	147	146	146	145	150	151	149	152	133

TABLE V-10

CAPITAL COST SUMMARY FOR 6-HOUR STORAGE

FPC Account	Description	Capital Cost, 10^3 \$* 1-Unit	2-Unit	3-Unit
1.0	Land & Land Rights	344	414	549
1.1	Structures & Improvements	3,724	5,969	8,144
1.2	Plant Equipment	12,546	18,403	23,763
1.4	Turbomachinery & Related Items	30,250	60,341	90,429
1.5	Electrical	2,894	6,535	8,949
1.6	Miscellaneous	341	425	496
	Direct Cost	50,099	92,087	132,330
	Contingency - 15%	7,515	13,813	19,850
		57,614	105,900	152,330
	Engineering & Construction Management	14,000	16,000	18,000
	Direct Capital Cost	71,614	121,900	170,180

* June 1976 dollars

TABLE V-11

CAPITAL COST SUMMARY FOR 20-HOUR STORAGE

FPC Account	Description	Capital Cost, 10^3 $* 1 Unit	2 Unit	3 Unit
1.0	Land & Land Rights	458	562	716
1.1	Structures & Improvements	4,310	7,005	9,575
1.2	Plant Equipment	20,689	34,544	48,041
1.4	Turbomachinery & Related Items	30,250	60,341	90,429
1.5	Electrical	2,894	6,535	8,949
1.6	Miscellaneous	341	425	496
	Direct Cost	58,942	109,412	158,206
	Contingency - 15%	8,841	16,412	23,731
		67,783	125,824	181,937
	Engineering & Construction Management	14,000	16,000	18,000
	Direct Capital Cost	81,783	141,824	199,937

*June 1976 dollars

TABLE V-12

CAPITAL COST SUMMARY FOR 40-HOUR STORAGE

FPC Account	Description	Capital Cost, 10^3 $ * 1 Unit	2 Unit	3 Unit
1.0	Land & Land Rights	516	883	1,149
1.1	Structures & Improvements	5,030	8,333	11,520
1.2	Plant Equipment	32,138	57,791	82,776
1.4	Turbomachinery & Related Items	30,250	60,341	90,429
1.5	Electrical	2,894	6,535	8,949
1.6	Miscellaneous	341	425	496
	Direct Cost	71,169	134,308	195,319
	Contingency - 15%	10,675	20,146	29,298
		81,844	154,454	224,617
	Engineering & Construction Management	14,000	16,000	18,000
	Direct Capital Cost	95,844	170,454	242,617

*June 1976 dollars

TABLE V-13

SPECIFIC CAPITAL COSTS

June 1976 Dollars

	1 Unit	2 Units	3 Units
Net Output, MW	252.5	505.0	757.5
		Specific Cost, $/kW	
6-Hour Storage	283.6	241.4	224.7
20-Hour Storage	323.9	280.8	263.9
40-Hour Storage	379.6	337.5	320.3

TABLE V-14

CAPS HOUSE LOADS

Steady State Conditions

MODE	MW	
Compression		
Parametric Estimate	199.9	
Motor Loss (η = .984)	3.2	
Transformer Loss (η = .995)	1.0	
Pumps, Fans, Ventilation	1.8	
Lights (Night and Outside)	0.2	
Gross Power Consumption	206.1	Single Unit
	402.2	Two Units
	618.3	Three Units
Generation		
Parametric Estimate (Gross) (Includes Generator Inefficiencies)	254.3	
Transformer Loss (η = .995)	(1.3)	
Pumps, Motors, Misc.	(0.4)	
Lights (Day Only)	(0.1)	
Net Power Consumption	252.5	Single Unit
	505.0	Two Units
	757.5	Three Units

TABLE V-15

ANNUAL PLANT STAFF LABOR COSTS

June 1976 Dollars

		Number of Units		
		1	2	3
Title, Salary & Overhead Costs				
Plant Supervisor	@ $45,000	4	4	4
Operator	@ $37,500	4	4	4
Operator's Helper	@ $30,000	4	4	4
Electrician	@ $37,500	4	7	10
Electrician's Helper	@ $30,000	4	7	10
Mechanic	@ $37,500	4	7	10
Mechanic's Helper	@ $30,000	4	7	10
Secretary/Cleric	@ $25,000	2	2	2
Guards	@ $25,000	8	8	8
Janitorial	@ $25,000	4	8	12
Groundskeeping	@ $25,000	4	7	10
Total Plant Personnel		46	65	84
Plant Personnel Cost		$1,440,000	$2,020,000	$2,600,000
Unallocated Labor				
Engineering		$200,000	$250,000	$300,000
Utility Staff*		$ 50,000	$ 50,000	$ 50,000
Total Labor Cost		$1,690,000	$2,320,000	$2,950,000

*i.e., Extra men for overhaul work.

TABLE V-16

ANNUAL MATERIALS & EXPENSES

June 1976 Dollars

	1 Unit	2 Unit	3 Unit
Fuel System Maintenance			
Fuel Analyses	5,500	7,500	9,500
Sampling Supplies	600	1,200	1,500
Fuel System Supplies	1,000	1,200	1,300
Compressor Maintenance			
Chemicals and Solvents	1,700	3,400	4,800
Lubricants	300	600	900
Overhaul Parts	20,000	40,000	60,000
(Inspection & Certification Fees)	200	400	600
Turbine/Generator Maintenance			
Lubricants	1,200	2,400	3,600
Hydrogen	100	200	300
Solvents	300	600	800
Turbine Overhaul	30,000	60,000	90,000
General Maintenance			
General Supplies	50,000	75,000	100,000
First Aid	100	200	200
Staff Service Supplies	2,000	3,000	3,500
Building Service Supplies	10,000	16,000	20,000
Communication Line	1,000	1,000	1,000
Total Materials & Expenses	$124,000	$211,500	$297,000

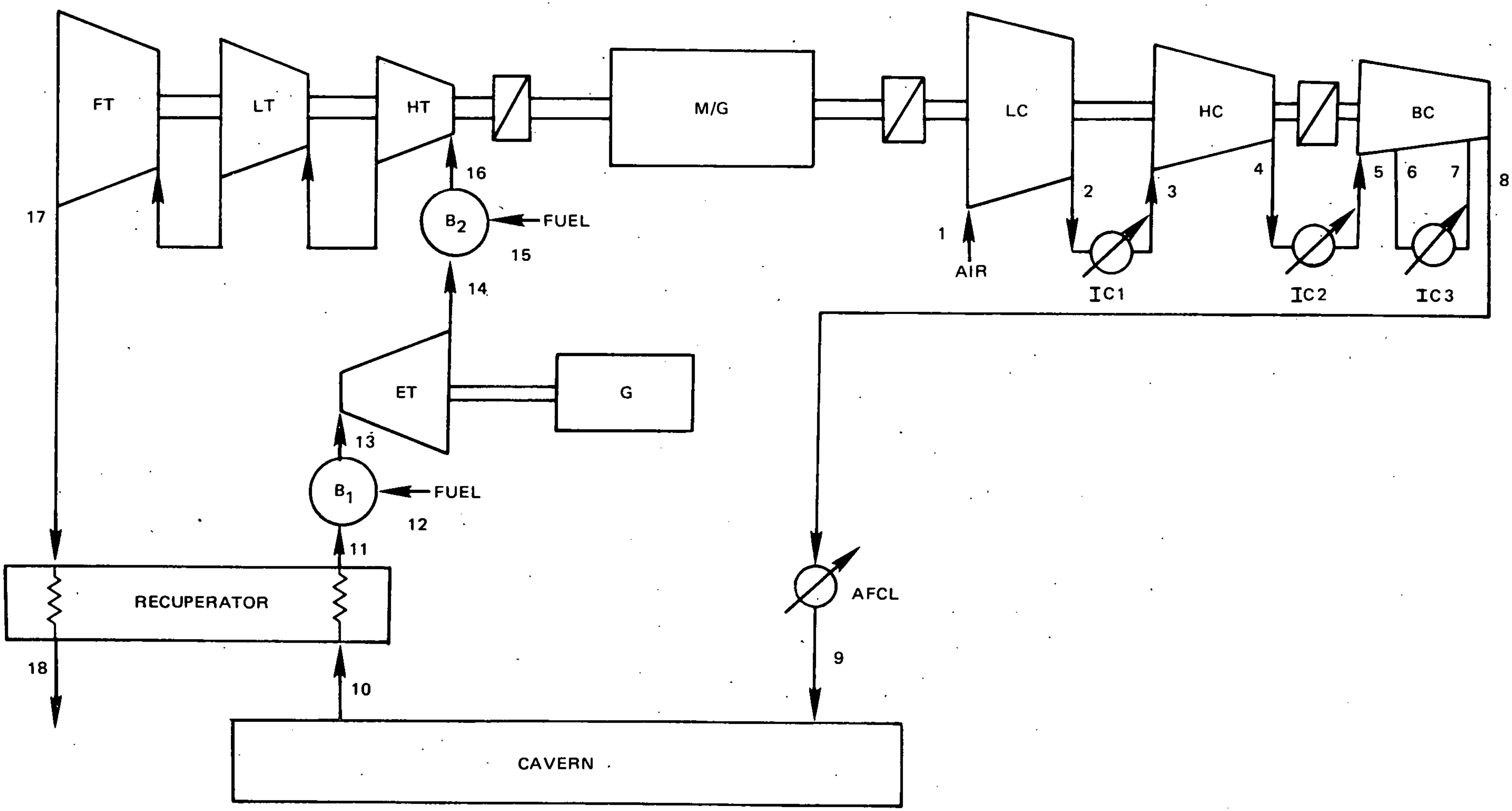
TABLE V-17

OPERATING AND MAINTENANCE COST SUMMARY

June 1976 Dollars

	Number of Units		
	1	2	3
Annual Costs, 10^3 \$/yr			
Total Labor Cost	1,690	2,320	2,950
Total Materials & Expenses	124	212	297
Total O&M Cost	1,814	2,532	3,247
Specific O&M Cost, \$/kWyr	7.18	5.01	4.29
Specific O&M Cost at Design Capacity, mills/kWhr			
6-Hour Storage (1560 hr/yr)	4.60	3.21	2.75
20-Hour Storage (2190 hr/yr)	3.28	2.29	1.96
40-Hour Storage (3125 hr/yr)	2.30	1.60	1.37
Specific O&M Cost at 85% of Design Capacity, mills/kWhr			
6-Hour Storage (1326 hr/yr)	5.41	3.78	3.24
20-Hour Storage (1862 hr/yr)	3.86	2.69	2.30
40-Hour Storage (2656 hr/yr)	2.70	1.89	1.62

CAPS CONFIGURATION



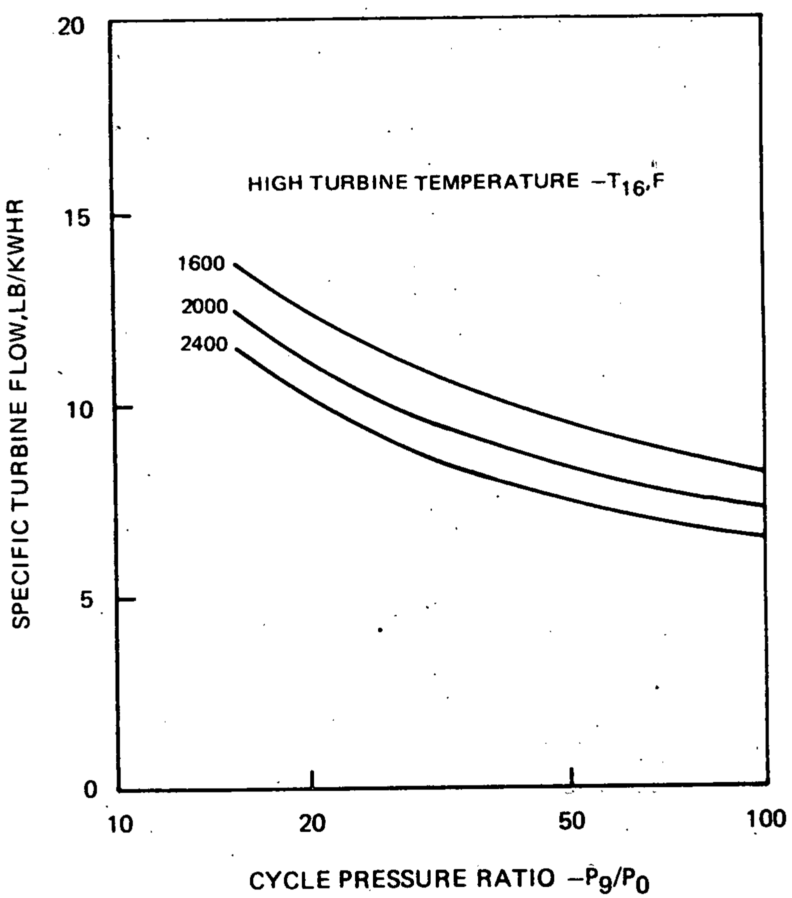
R76-952161-5

76-05-172-4

FIG. V-1

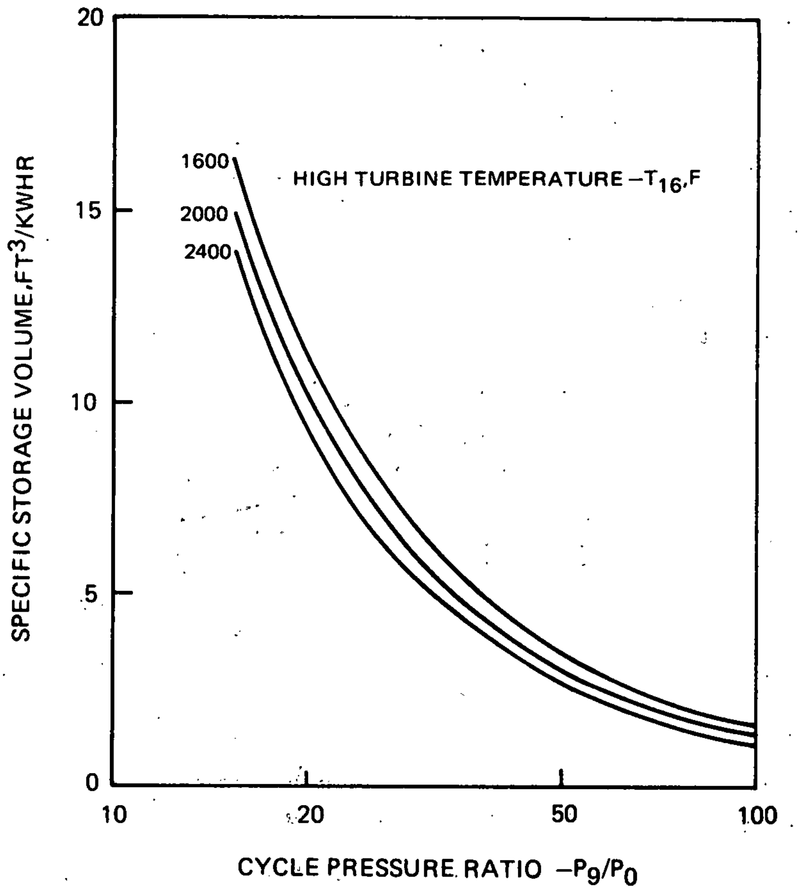
SPECIFIC TURBINE FLOW

EXPANSION TURBINE INLET TEMPERATURE, T_{13} = 1500F
INDEPENDENT OF RECUPERATOR EFFECTIVENESS
INDEPENDENT OF CAVERN LEAKAGE



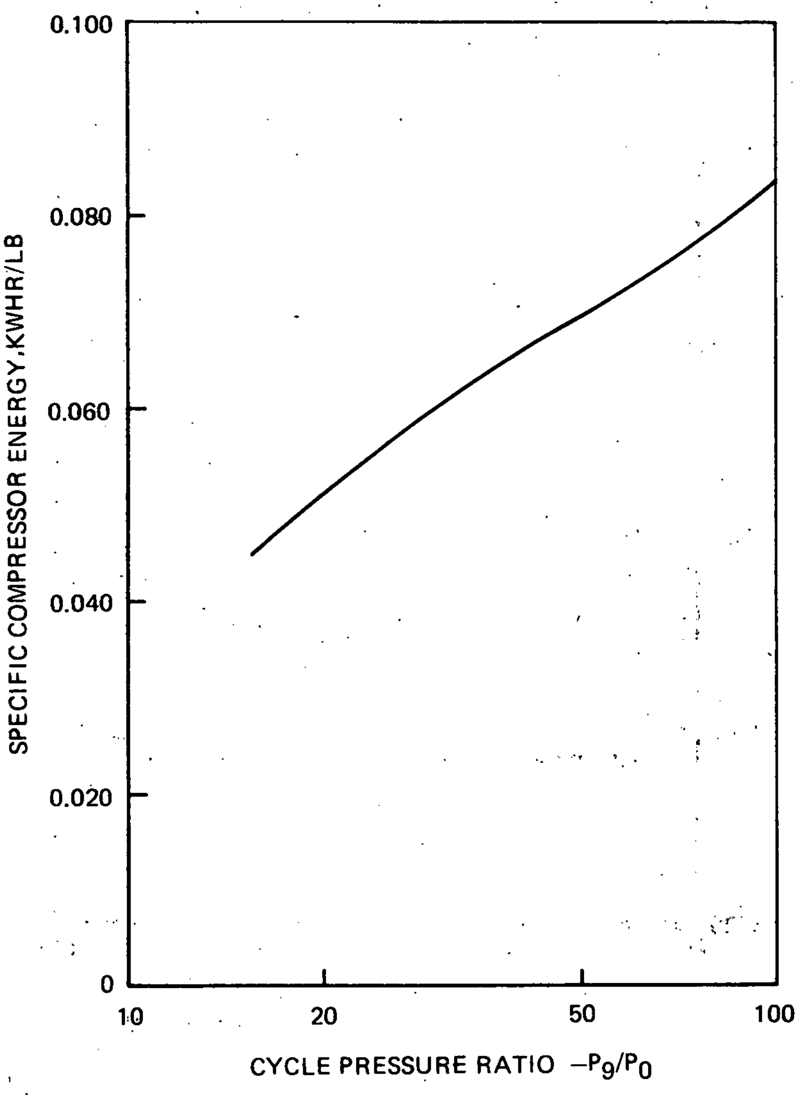
SPECIFIC STORAGE VOLUME

EXPANSION TURBINE INLET TEMPERATURE, T_{13} = 1500F
INDEPENDENT OF RECUPERATOR EFFECTIVENESS
CAVERN LEAKAGE = 4%



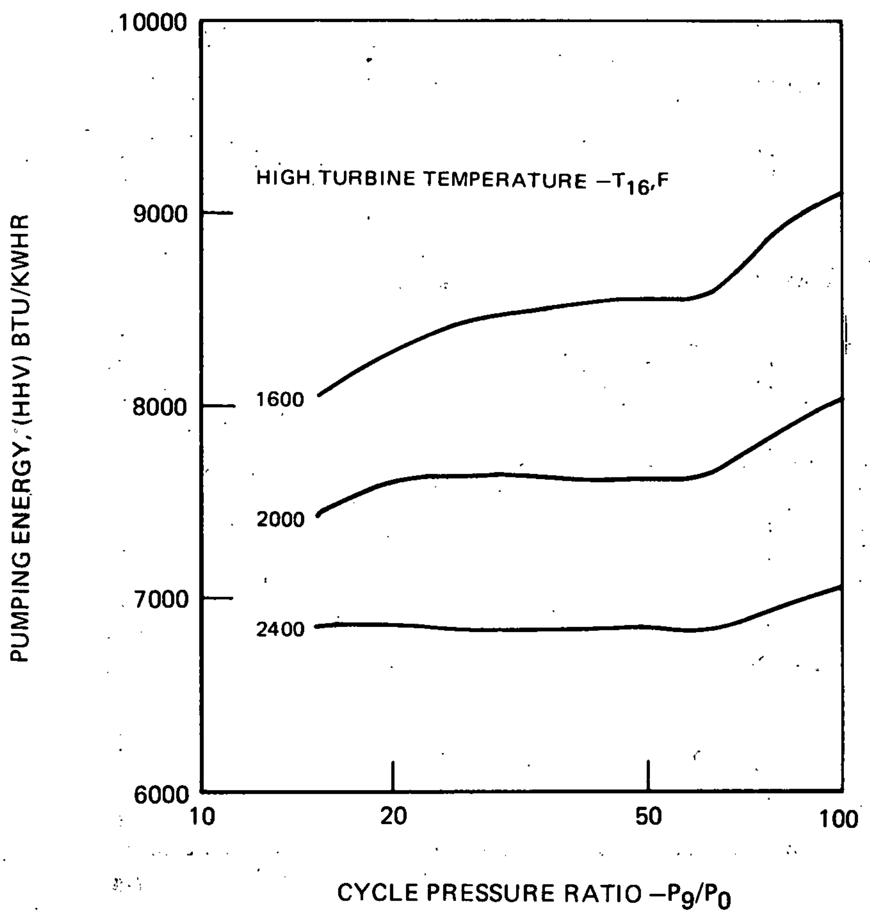
SPECIFIC COMPRESSOR ENERGY

EXPANSION TURBINE INLET TEMPERATURE, T_{13} = 1500F
INDEPENDENT OF RECUPERATOR EFFECTIVENESS
INDEPENDENT OF CAVERN LEAKAGE



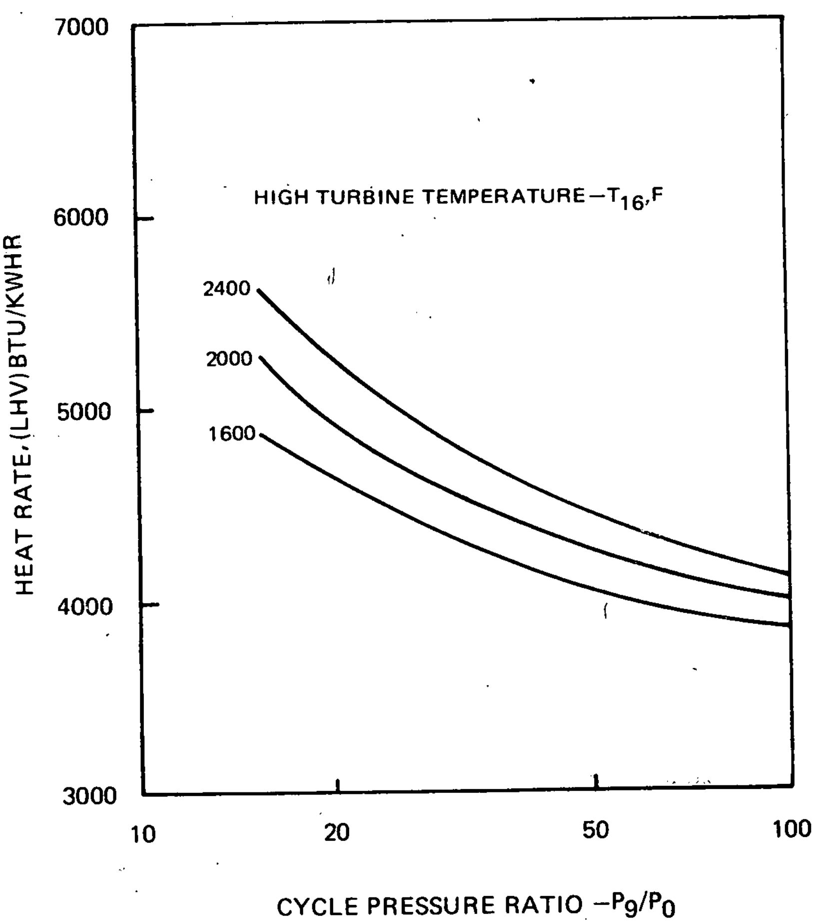
COMPRESSOR PUMPING ENERGY

EXPANSION TURBINE INLET TEMPERATURE, T_{13} = 1500 F
INDEPENDENT OF RECUPERATOR EFFECTIVENESS
CAVERN LEAKAGE = 4%



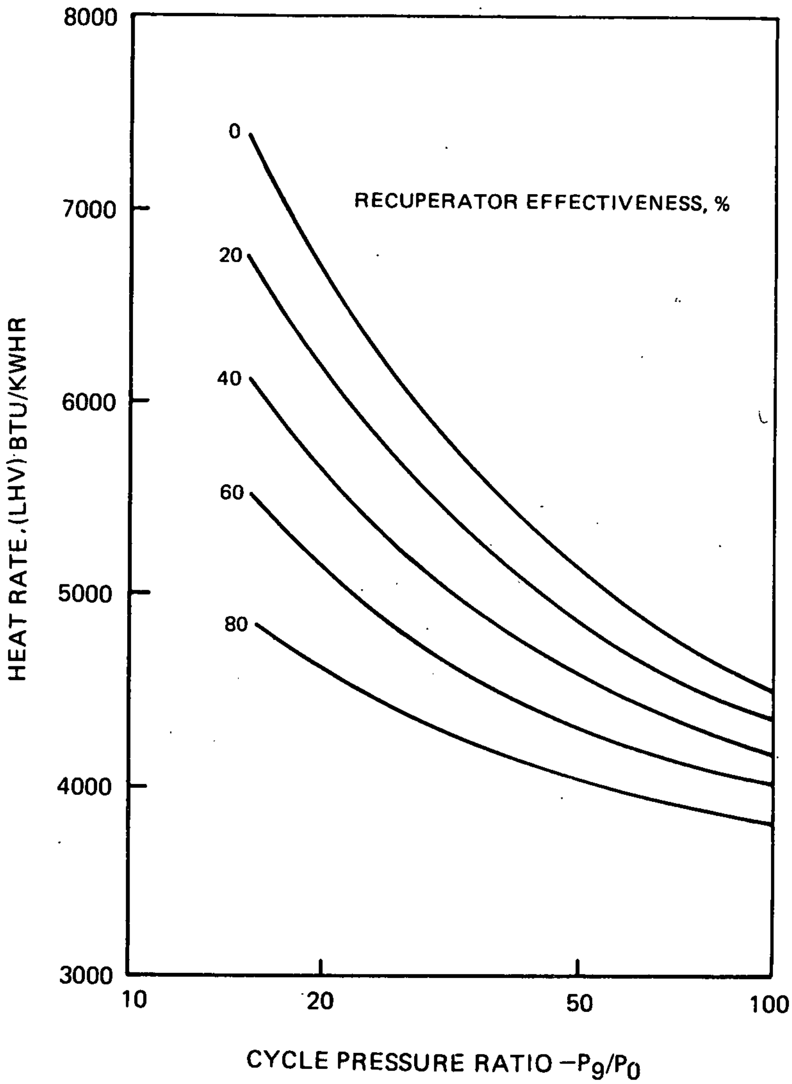
FUEL SYSTEM HEAT RATE

EXPANSION TURBINE INLET TEMPERATURE, T_{13} = 1500F
RECUPERATOR EFFECTIVENESS = 80%
INDEPENDENT OF CAVERN LEAKAGE



PERFORMANCE VARIATION WITH RECUPERATOR EFFECTIVENESS

HIGH TURBINE INLET TEMPERATURE, T_{16} = 1600F
EXPANSION TURBINE INLET TEMPERATURE, T_{13} = 1500F
INDEPENDENT OF CAVERN LEAKAGE



PERFORMANCE VARIATION WITH RECUPERATOR EFFECTIVENESS

HIGH TURBINE INLET TEMPERATURE T_{16} = 2000F
EXPANSION TURBINE INLET TEMPERATURE T_{13} = 1500F
INDENPENDENT OF CAVERN LEAKAGE

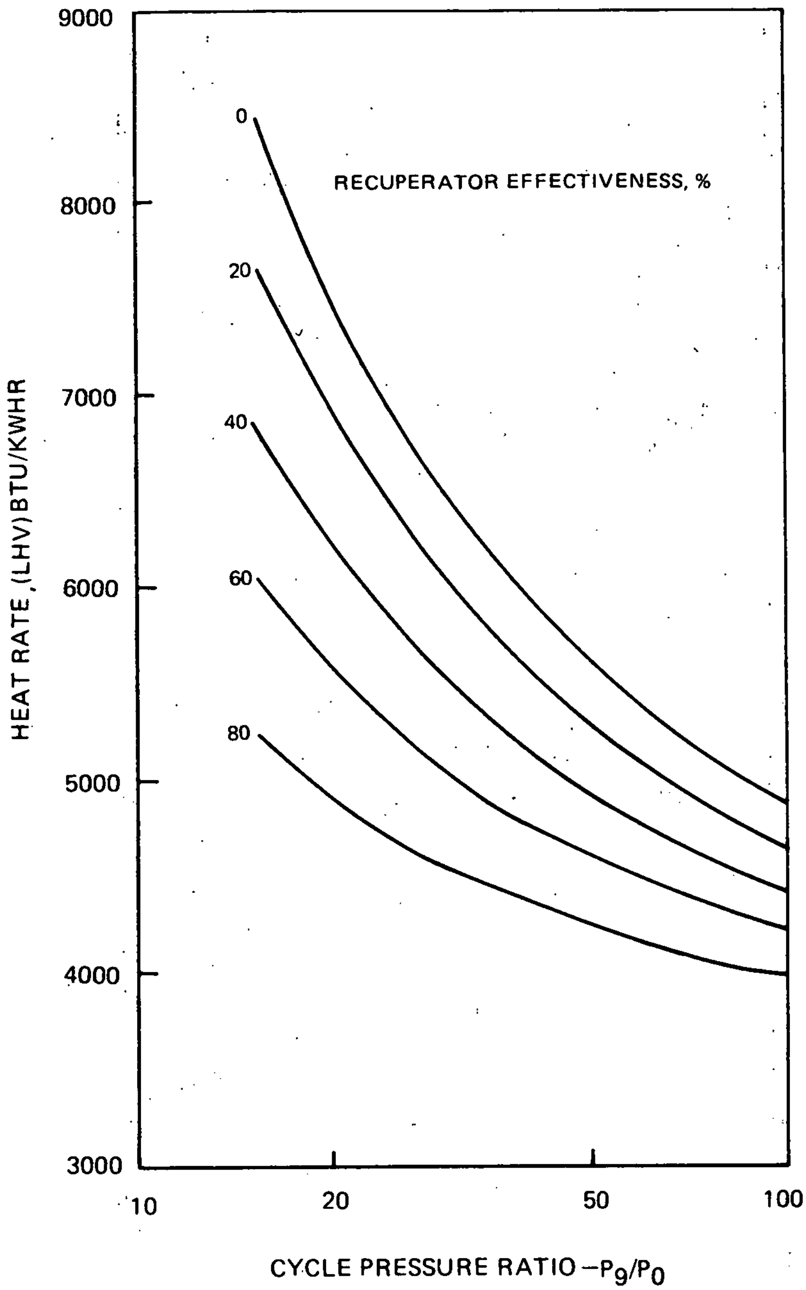
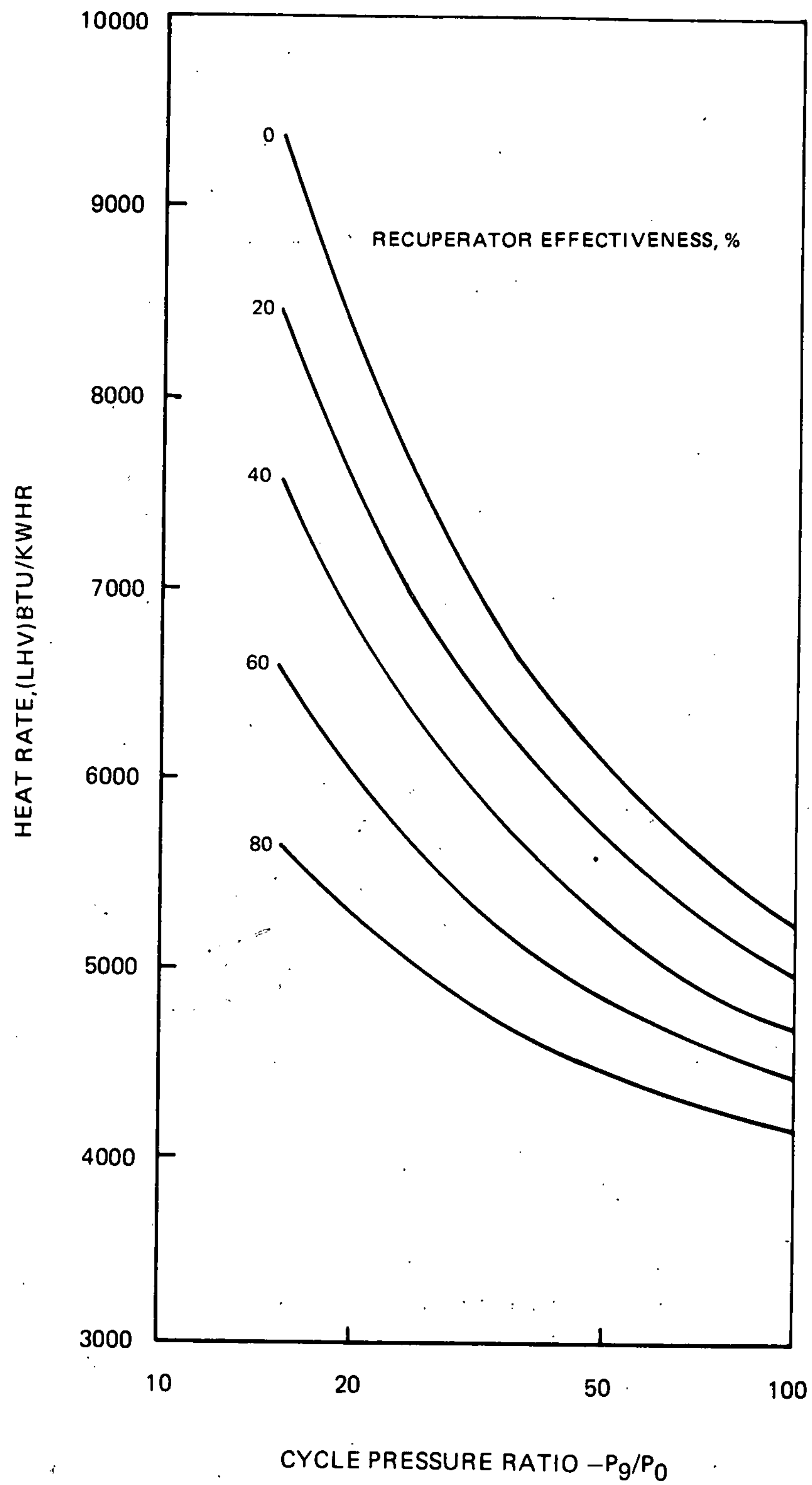


FIG. V–9

PERFORMANCE VARIATION WITH RECUPERATOR EFFECTIVENESS

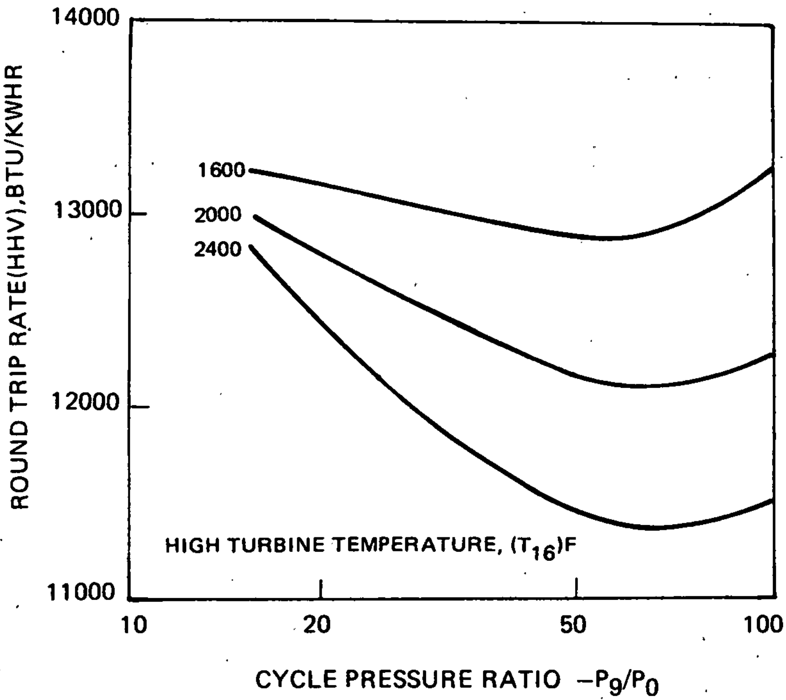
HIGH TURBINE INLET TEMPERATURE, T_{16} =2400F
EXPANSION TURBINE INLET TEMPERATURE, T_{13} =1500F
INDEPENDENT OF CAVERN LEAKAGE



76–05–201–8

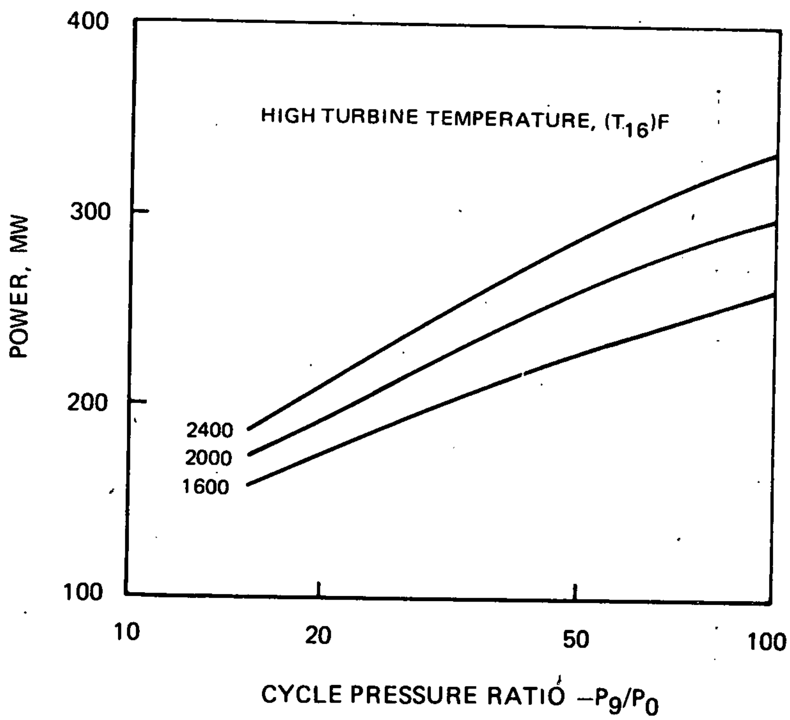
ROUND – TRIP HEAT RATE

EXPANSION TURBINE INLET TEMPERATURE, T_{13} = 1500F
RECUPERATOR EFFECTIVENESS = 80%
CAVERN LEAKAGE = 4%



POWER OUTPUT

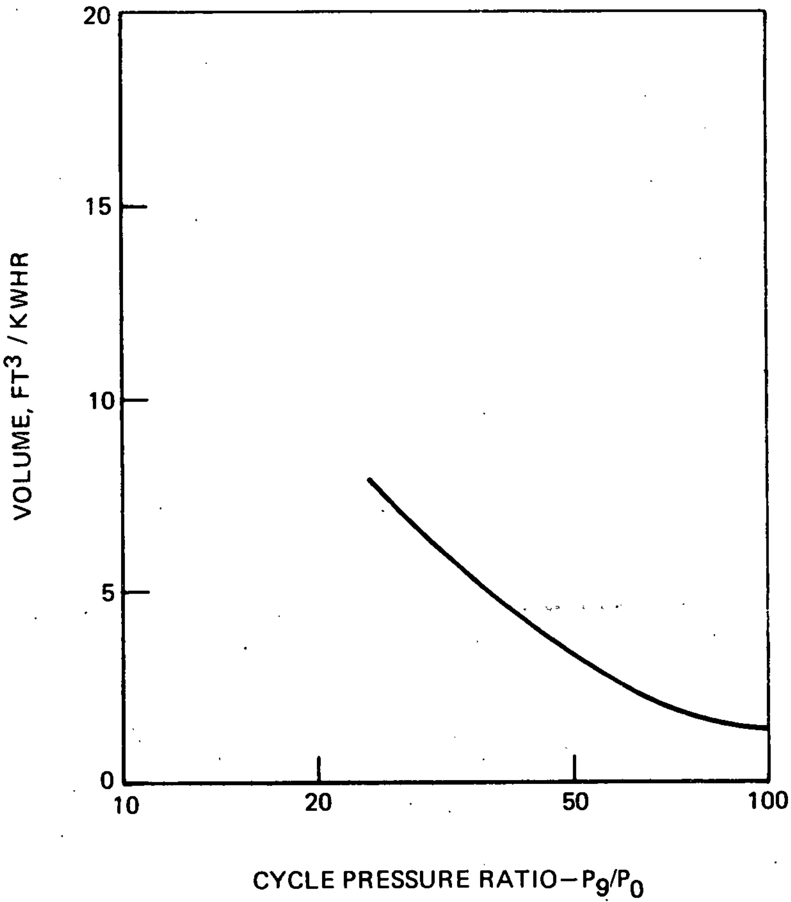
EXPANSION TURBINE INLET TEMPERATURE, T_{13} = 1500F
INDEPENDENT OF RECUPERATOR EFFECTIVENESS
CAVERN LEAKAGE = 4%
761.8 LB/SEC



SPECIFIC STORAGE VOLUME

LOW PRESSURE TURBOMACHINERY BASED ON FT50 DESIGN

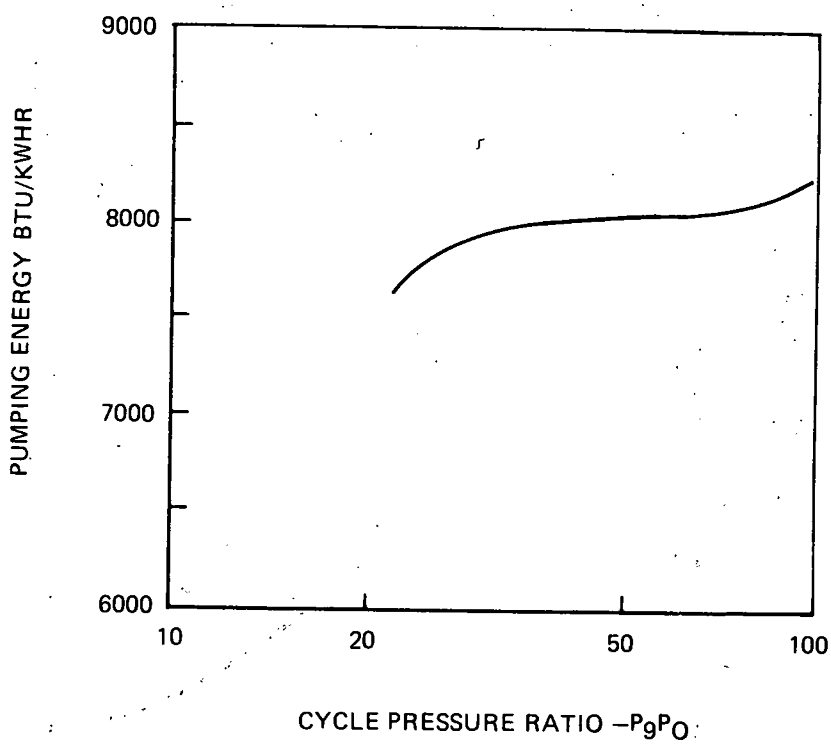
HIGH TURBINE TEMPERATURE, T_{16} = 2000F
INDEPENDENT OF RECUPERATOR EFFECTIVENESS
CAVERN LEAKAGE = 4%



COMPRESSOR PUMPING ENERGY

LOW PRESSURE TURBOMACHINERY BASED ON FT50 DESIGN

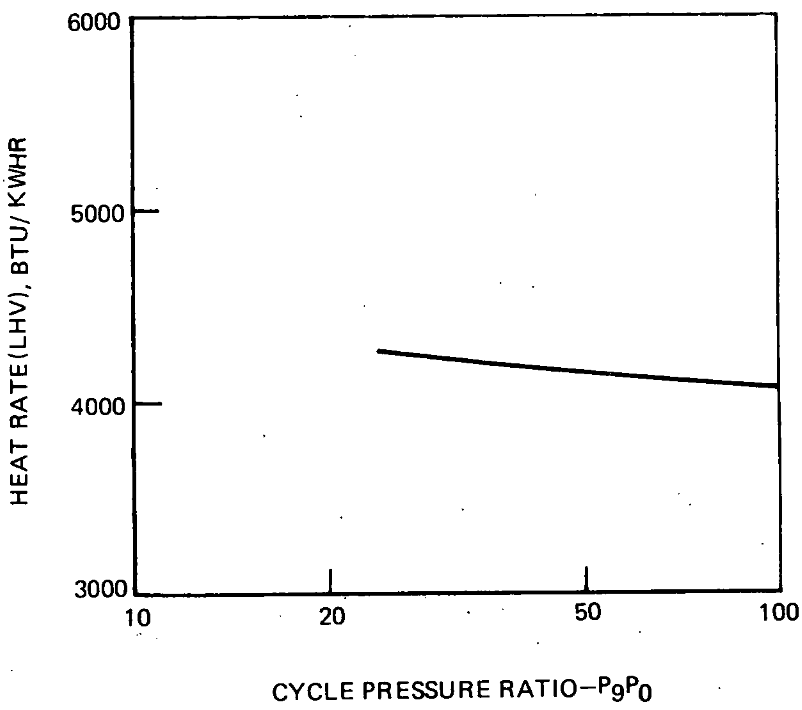
HIGH TURBINE INLET TEMPERATURE, T_{16} = 2000
INDEPENDENT OF RECUPERATOR EFFECTIVENESS
CAVERN LEAKAGE = 4%



FUEL SYSTEM HEAT RATE

LOW PRESSURE TURBOMACHINERY BASED ON FT50 DESIGN

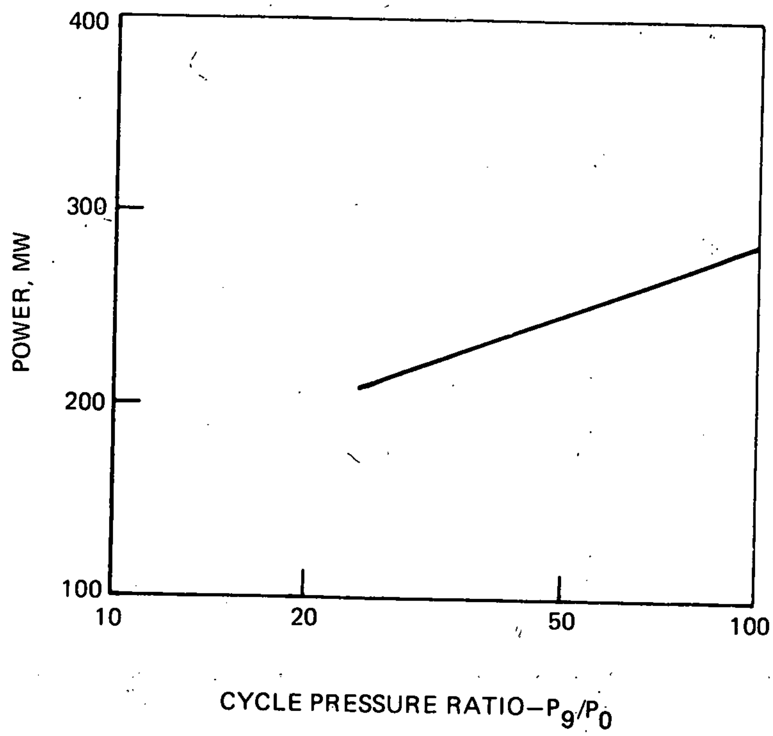
HIGH TURBINE INLET TEMPERATURE, T_{16} = 2000F
RECUPERATOR EFFECTIVENESS = 80%
INDEPENDENT OF CAVERN LEAKAGE



POWER OUTPUT

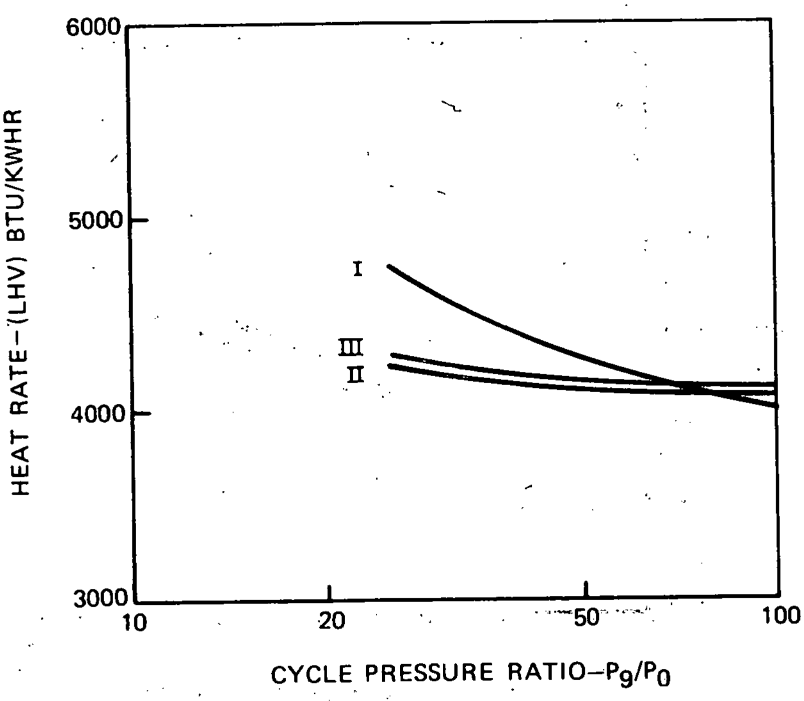
LOW PRESSURE TURBOMACHINARY BASED ON FT50 DESIGN

HIGH TURBINE INLET TEMPERATURE, T_{16} = 2000 F
INDEPENDENT OF RECUPERATOR EFFECTIVENESS
CAVERN LEAKAGE = 4%



FUEL SYSTEM HEAT RATE COMPARISON

HIGH TURBINE INLET TEMPERATURE, T_{16} = 2000F
RECUPERATION EFFECTIVENESS = 80%
INDEPENDENT OF CAVERN LEAKAGE



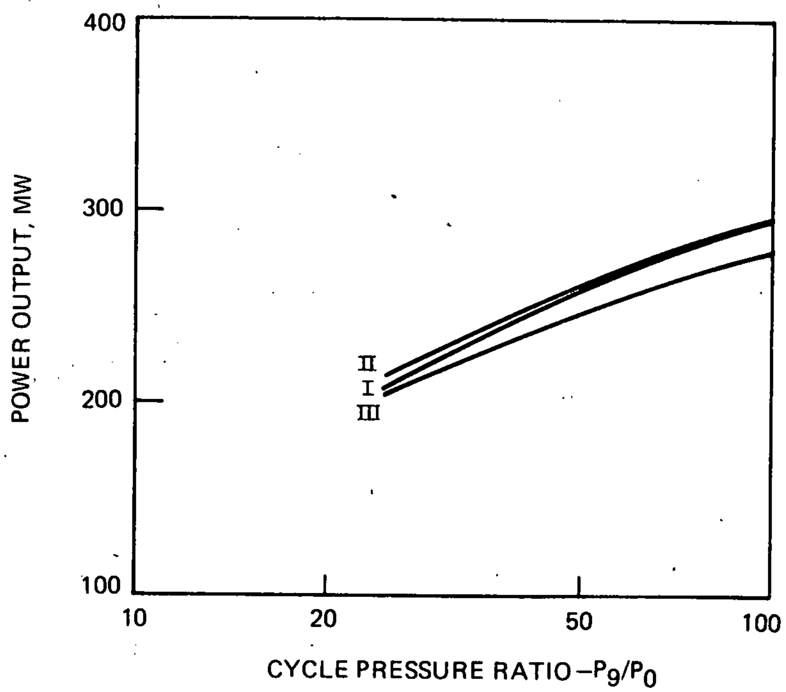
I EXPANSION TURBINE INLET TEMPERATURE, T_{13} = 1500F

II EXPANSION TURBINE EXIT TEMPERATURE, T_{14} = 750F

III LOW-PRESSURE TURBOMACHINERY BASED ON FT50 DESIGN

OUTPUT POWER COMPARISON

HIGH TURBINE INLET TEMPERATURE, T_{16} = 2000F
RECUPERATOR EFFECTIVENESS = 80%
INDEPENDENT OF CAVERN LEAKAGE



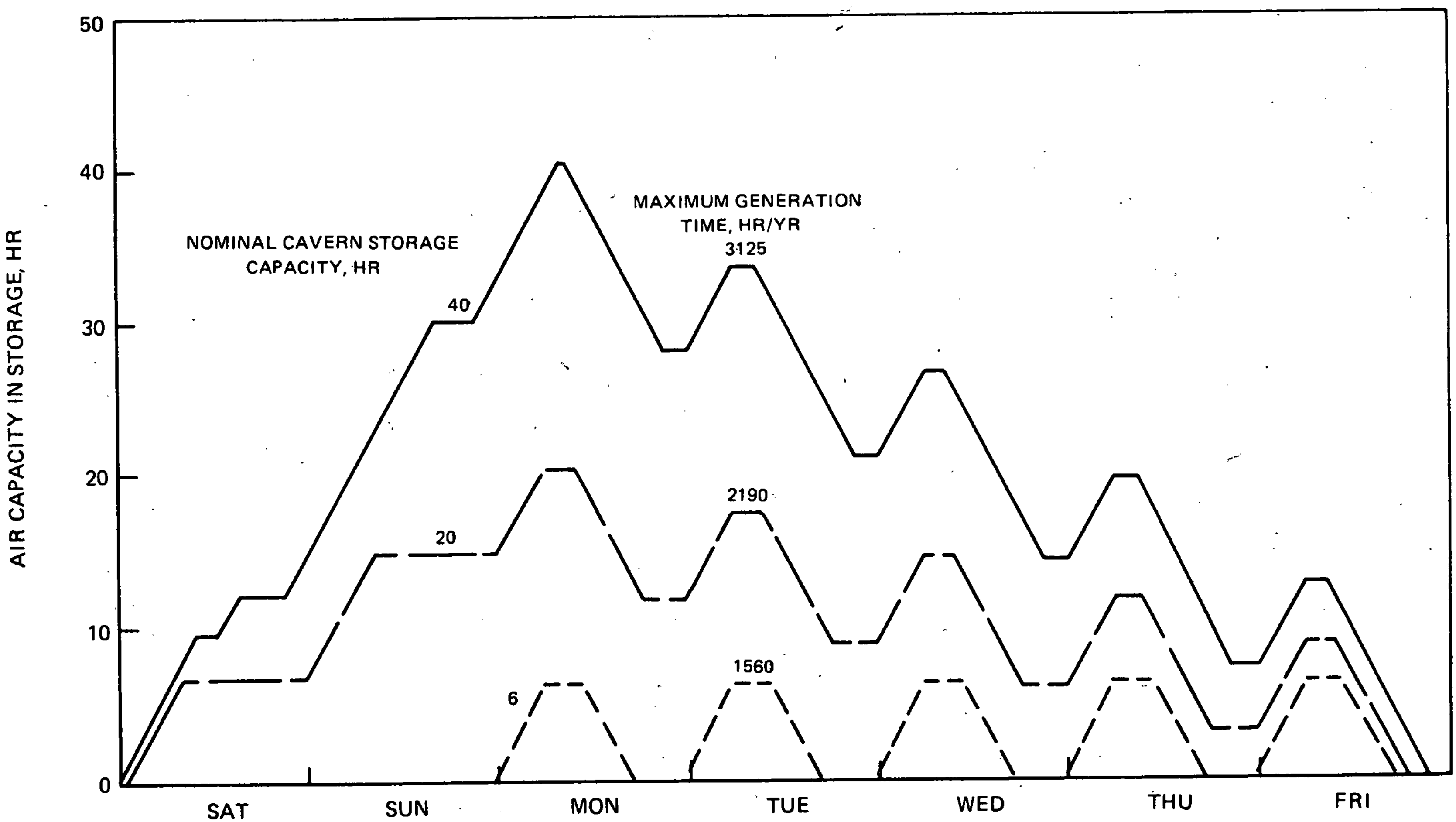
I EXPANSION TURBINE INLET TEMPERATURE, T_{13} = 1500F

II EXPANSION TURBINE EXIT TEMPERATURE, T_{14} = 750F

III LOW-PRESSURE TURBOMACHINERY BASED ON FT50 DESIGN

R76-952161-5

CAPS OPERATING CYCLES



76-12-18-2

FIG. V-18

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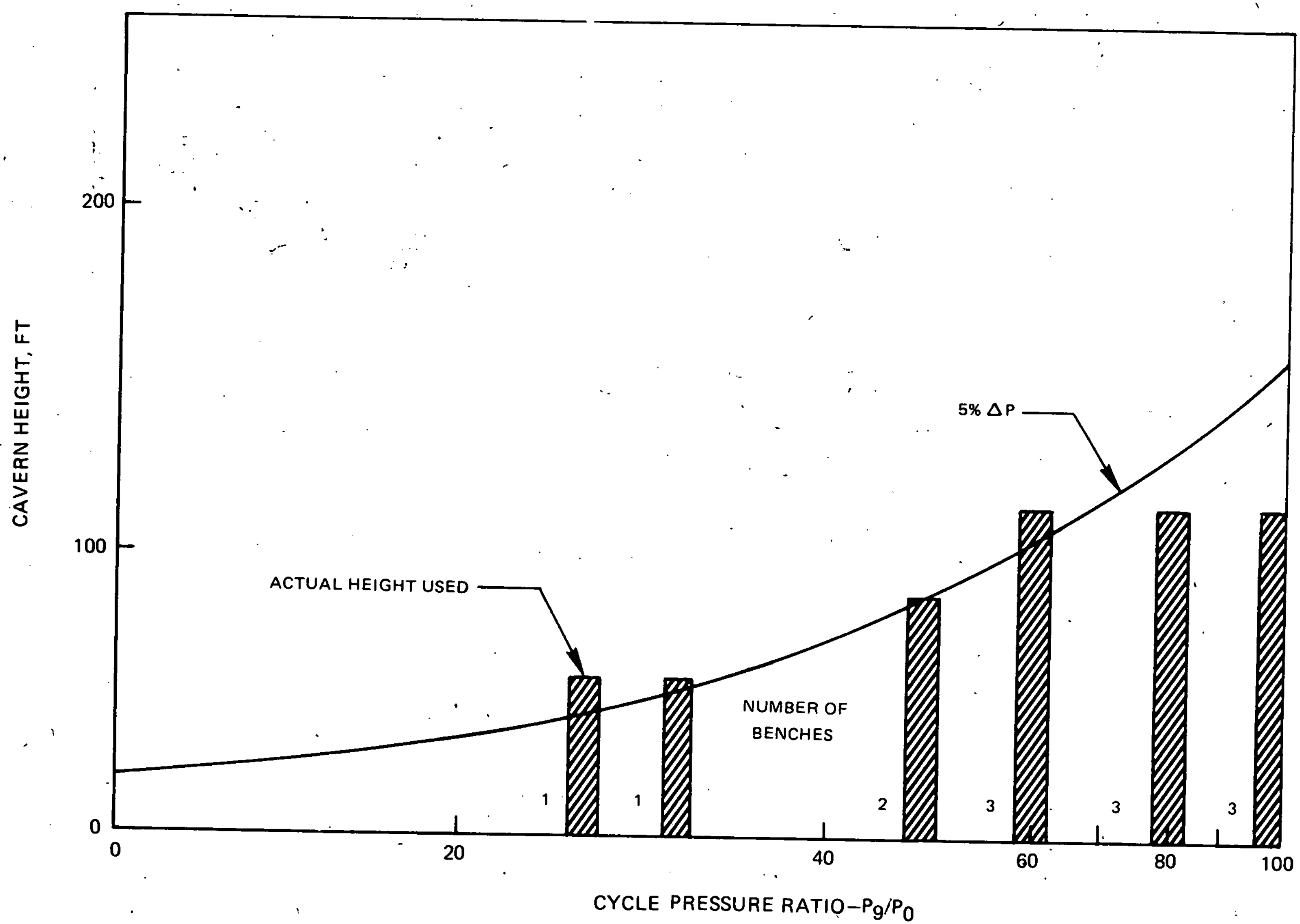
CAVERN HEIGTH VARIATION



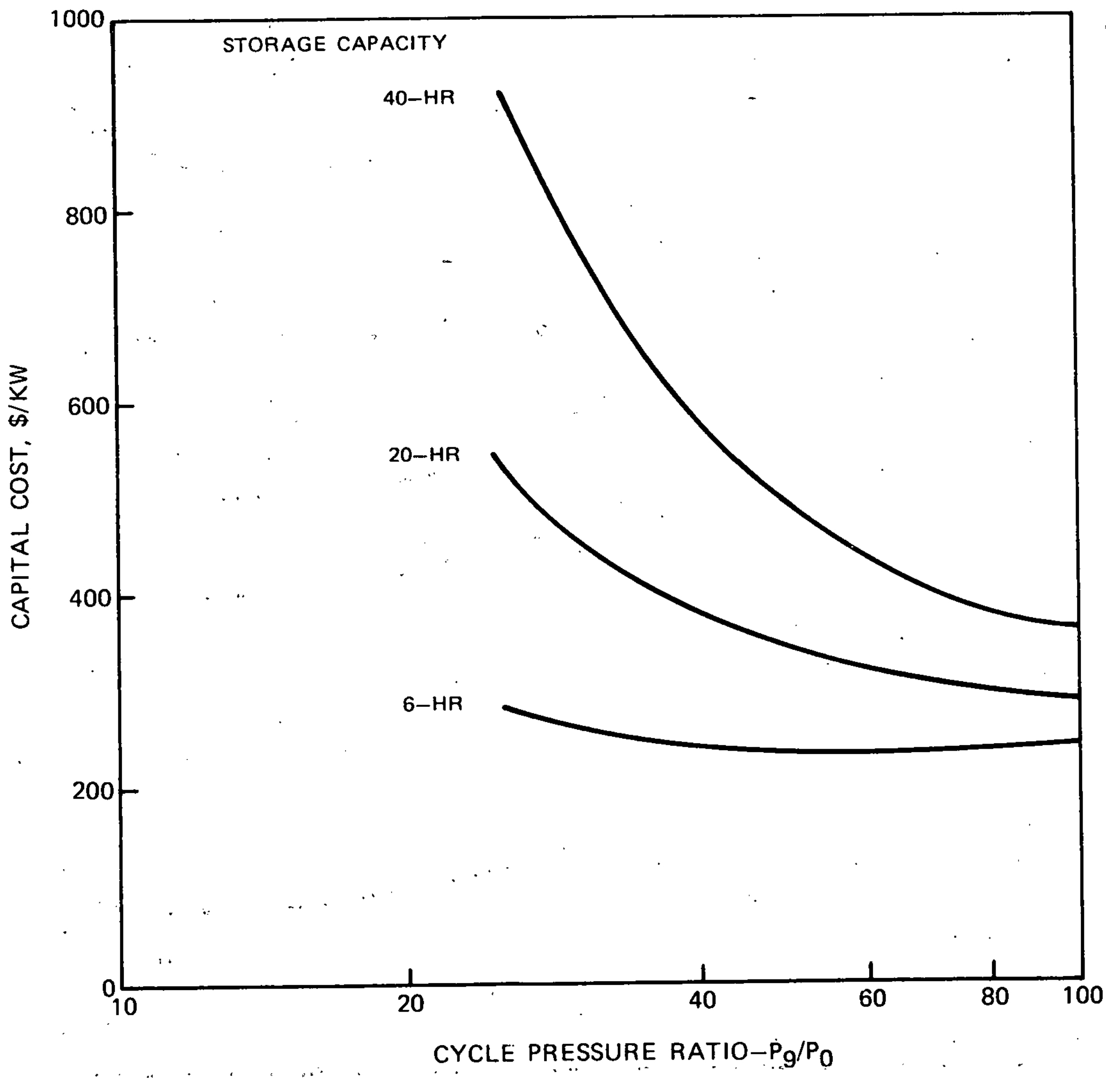
76-12-18-3

FIG. V-19

CAPS INSTALLED COSTS

MID–1976 START DATE

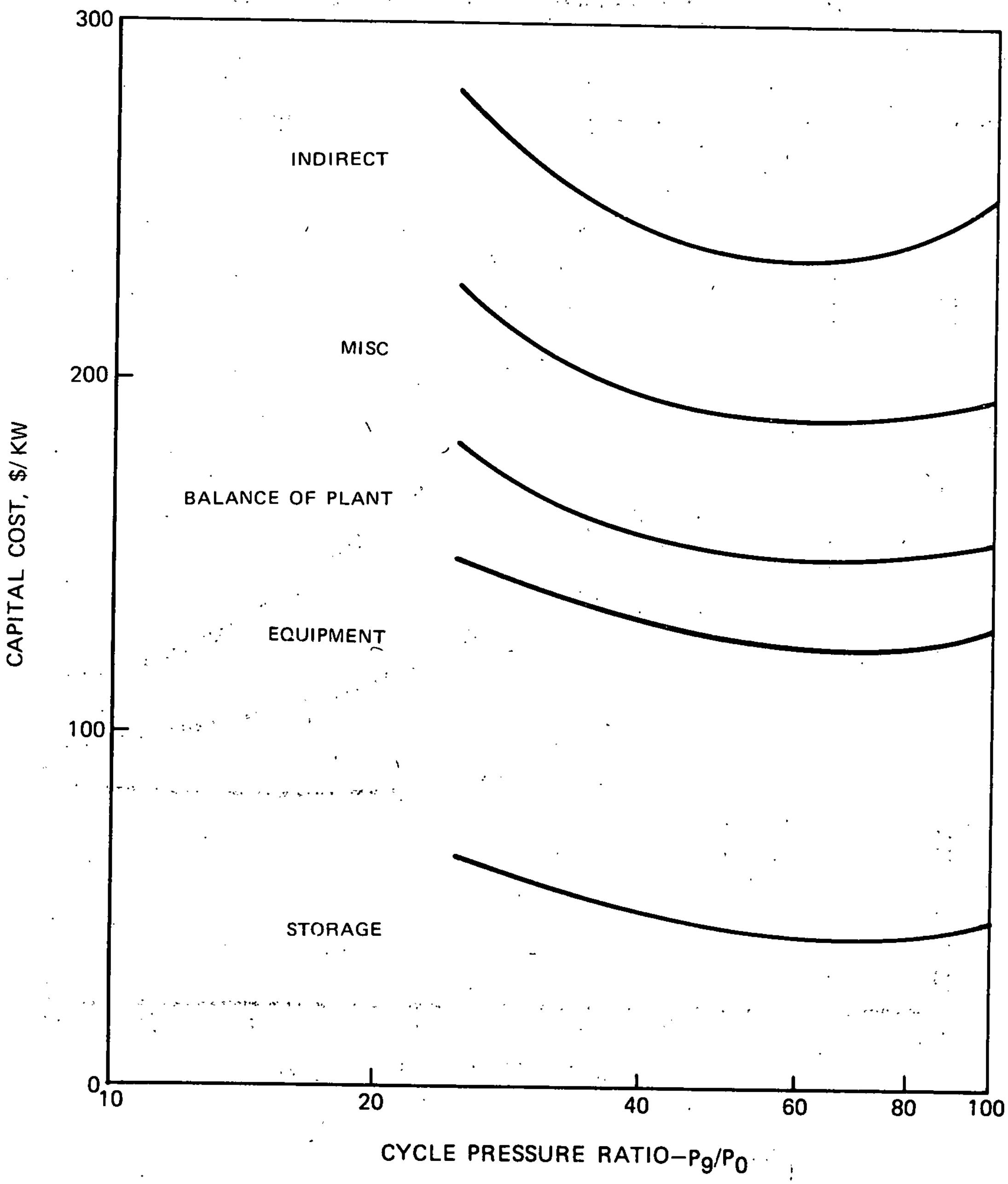
LOW–PRESSURE TURBOMACHINERY BASED ON FT50 DESIGN



CAPS COST DISTRUBUTION FOR 6–HR STORAGE CAPACITY

MID–1976 START DATE

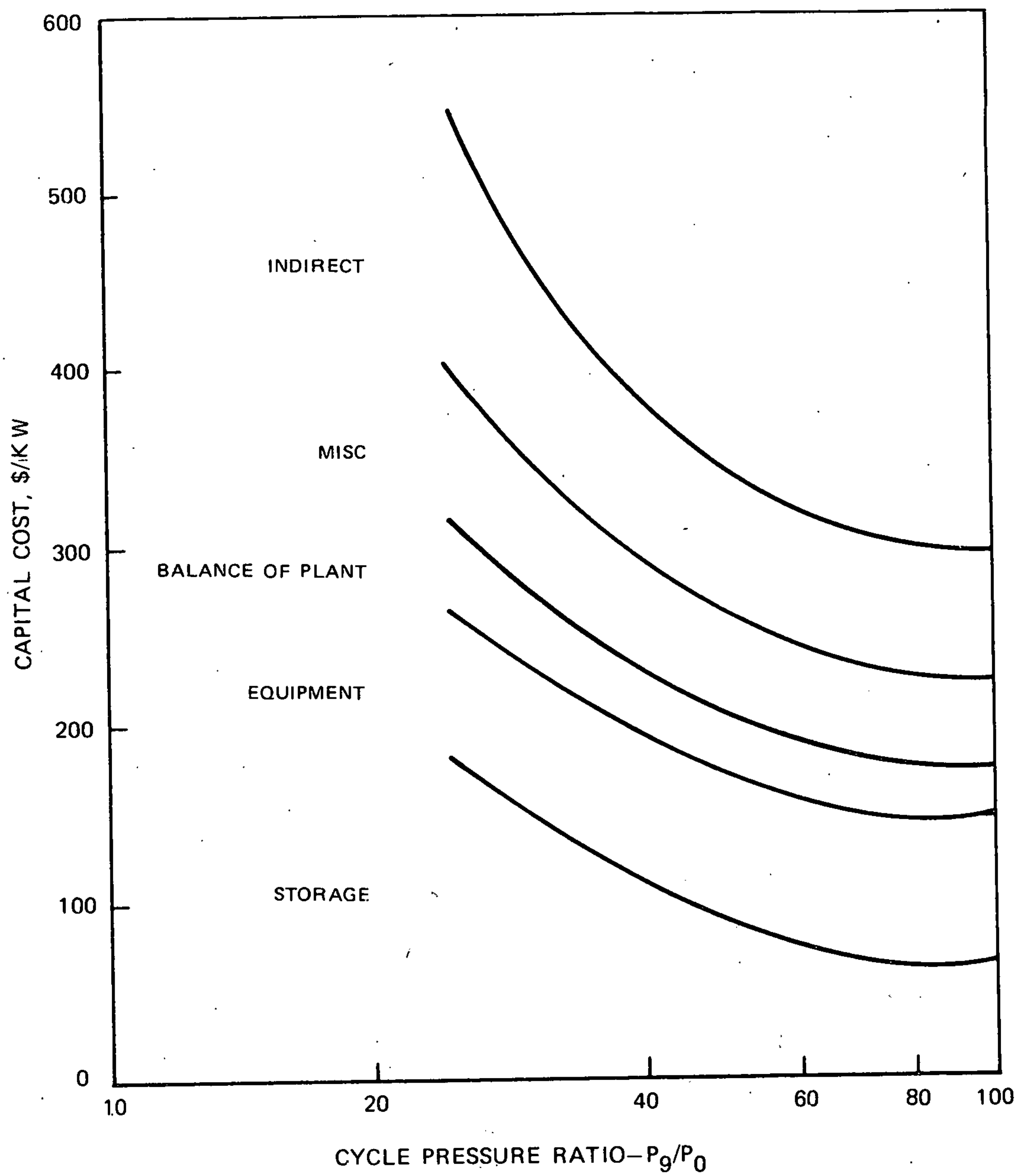
LOW–PRESSURE TURBOMACHINERY BASED ON FT50 DESIGN



CAPS COST DISTRIBUTION FOR 20–HR STORAGE CAPACITY

MID–1976 START DATE

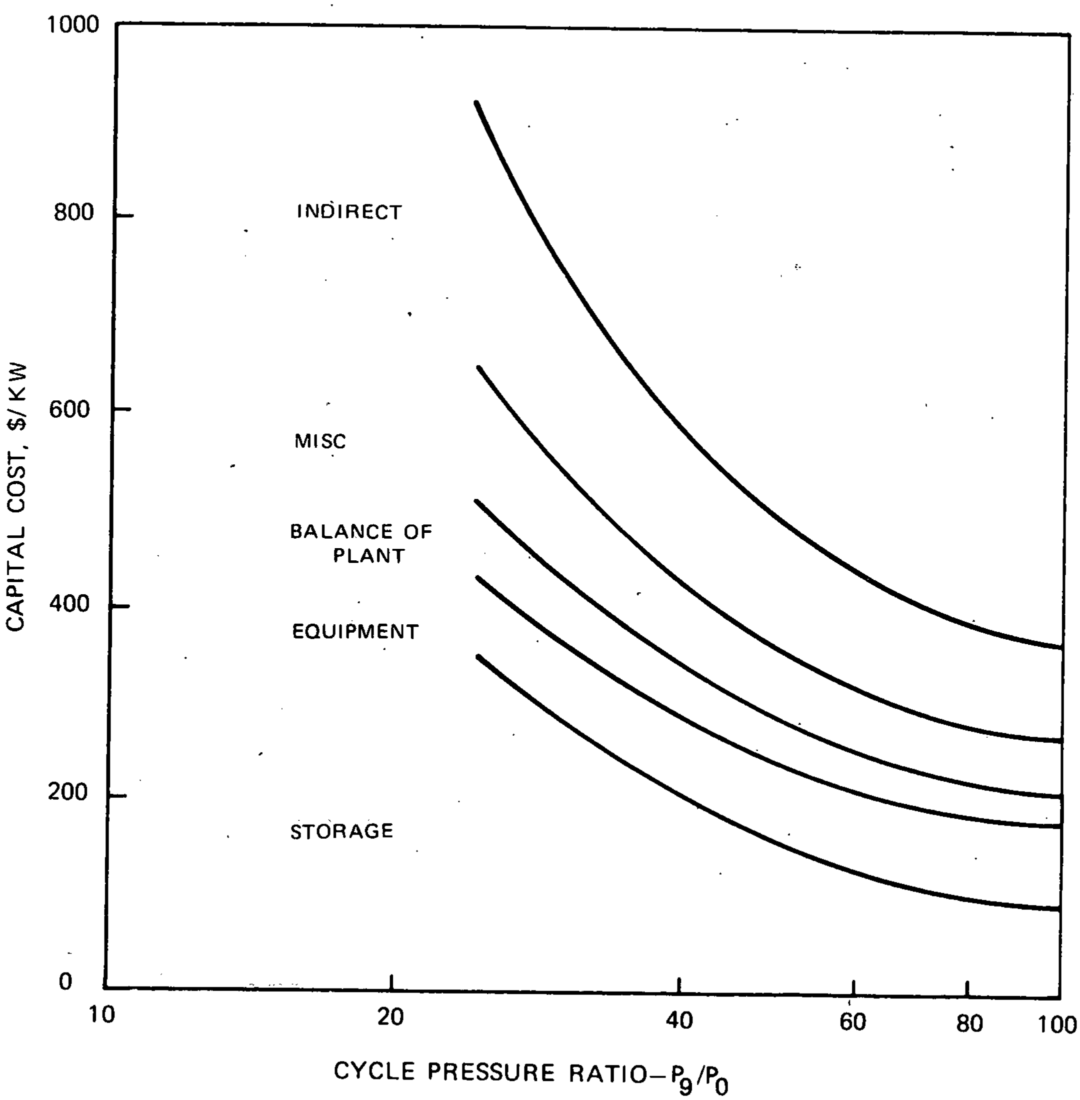
LOW–PRESSURE TURBOMACHINERY BASED ON FT50 DESIGN



CAPS COST DISTRIBUTION FOR 40–HR STORAGE CAPACITY

MID–1976 START DATE

LOW–PRESSURE TURBOMACHINERY BASED ON FT50 DESIGN



R76–952161–5

ANNUAL AVERAGE MARGINAL GENERATION COST

(FROM REF. V – 3)

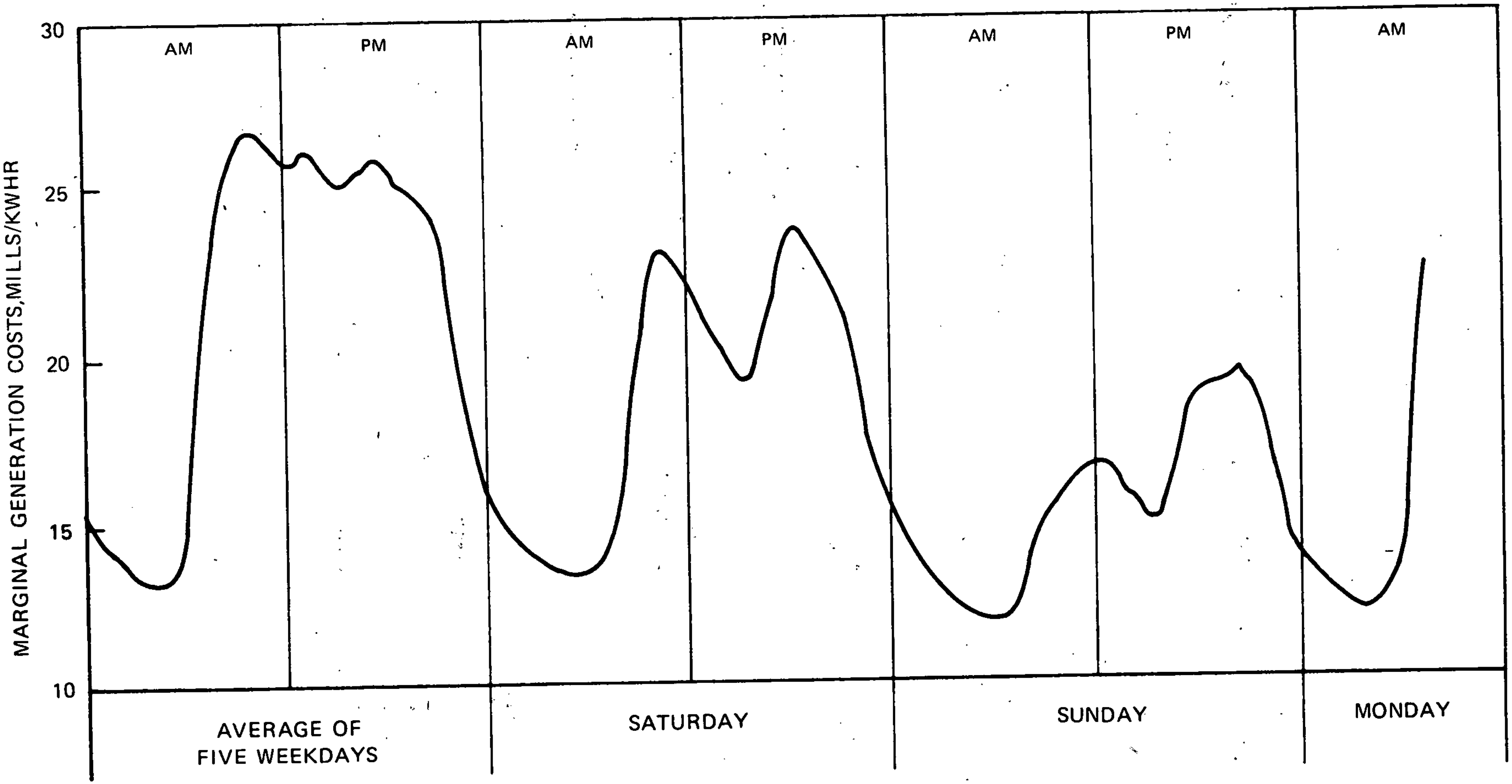


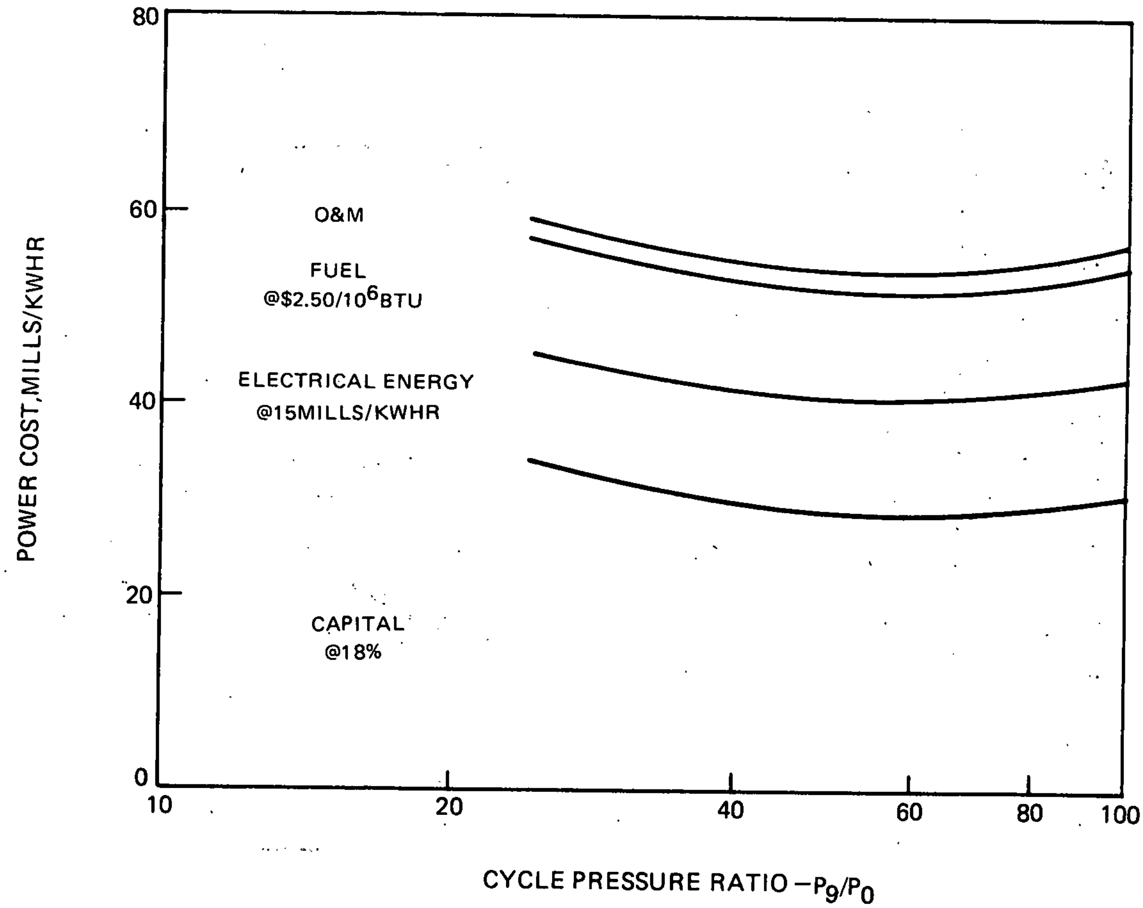
FIG. V–24

76–12–156–11

PARAMETRIC CAPS POWER COST DISTRIBUTION FOR 6–HR. STORAGE CAPACITY

MID–1976 START DATE

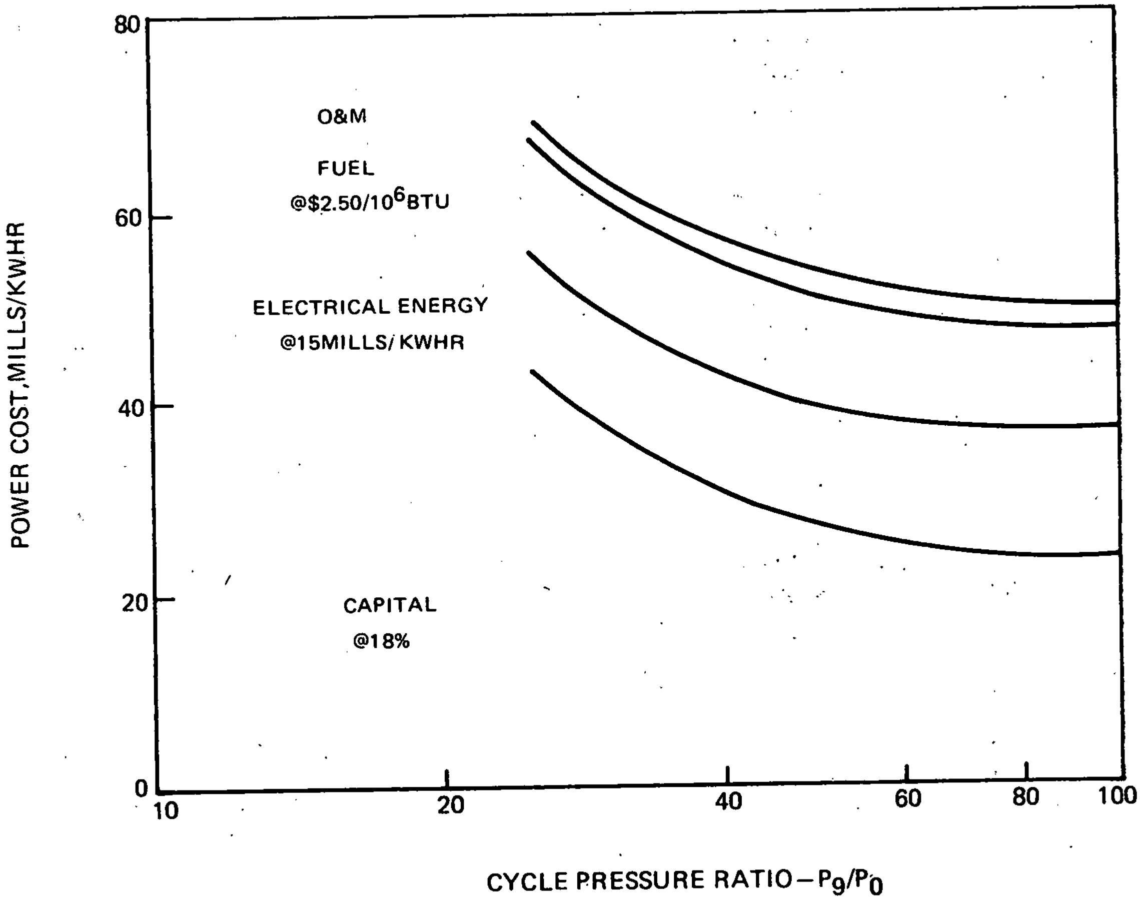
LOW–PRESSURE TURBOMACHINERY BASED ON FT50 DESIGN



PARAMETRIC CAPS POWER COST DISTRIBUTION FOR 20–HR. STORAGE

MID–1976 START DATE

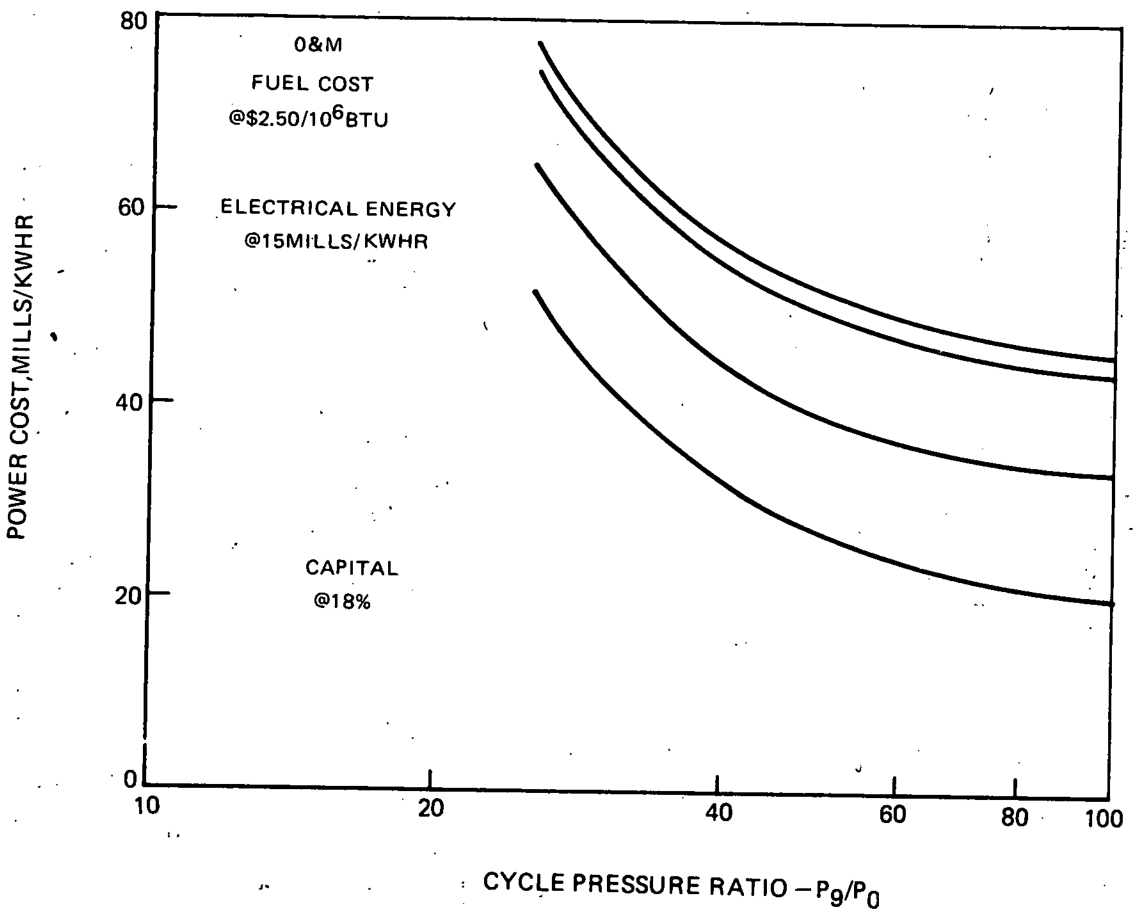
LOW–PRESSURE TURBOMACHINERY BASED ON FT50 DESIGN



PARAMETRIC CAPS POWER COST CONTRIBUTION FOR 40–HR STORAGE

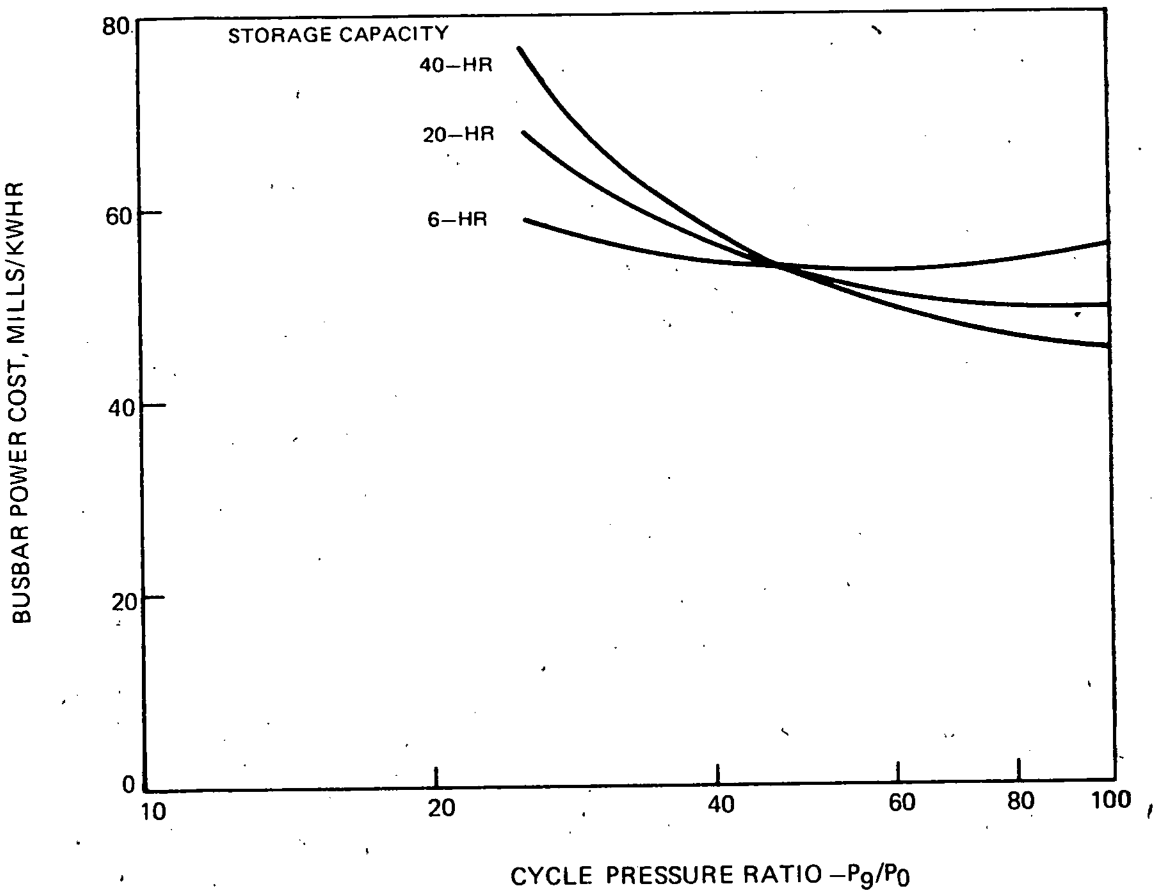
MID–1976 START DATE

LOW–PRESSURE TURBOMACHINERY BASED ON FT50 DESIGN

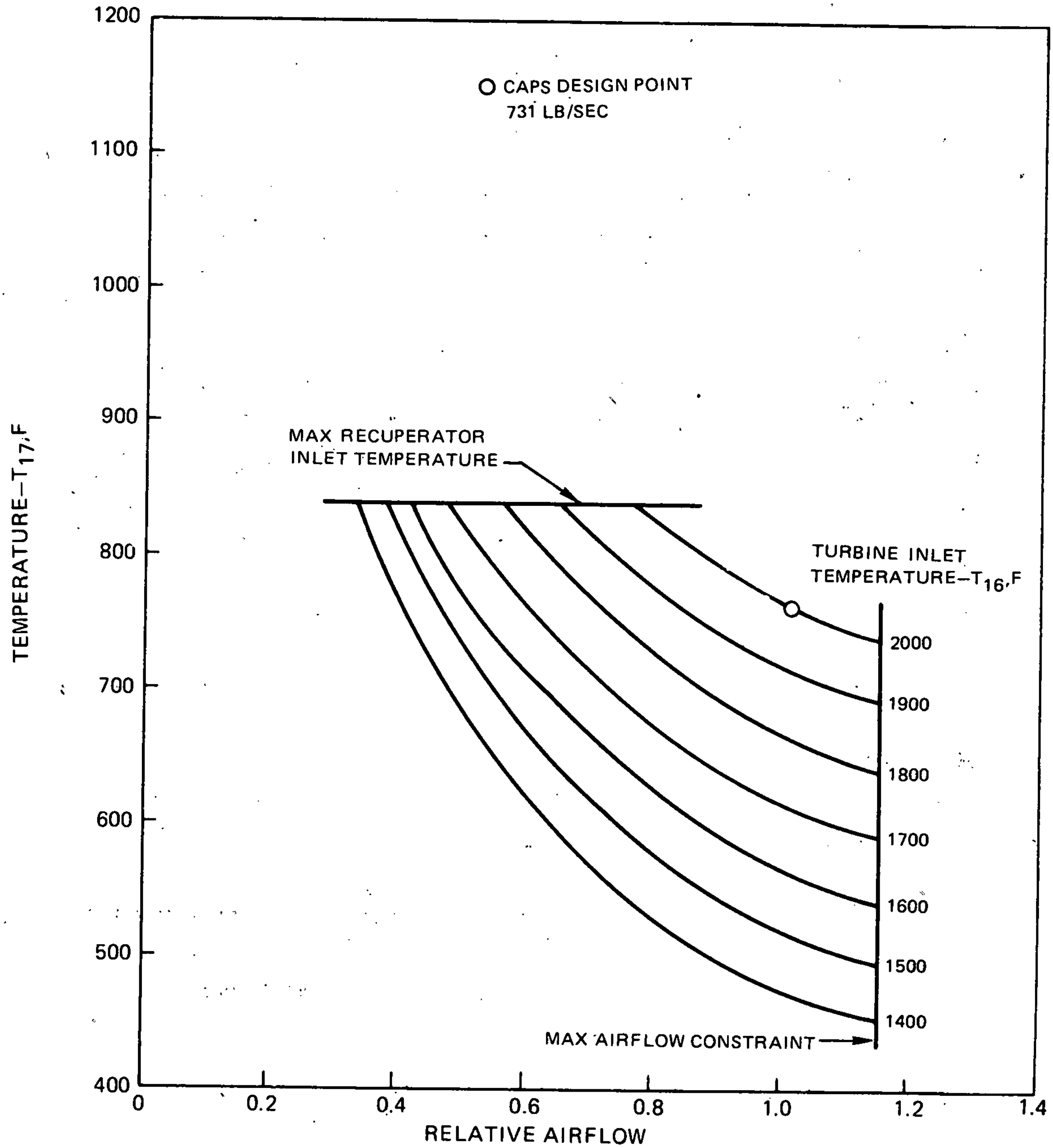


PARAMETRIC CAPS POWER COST

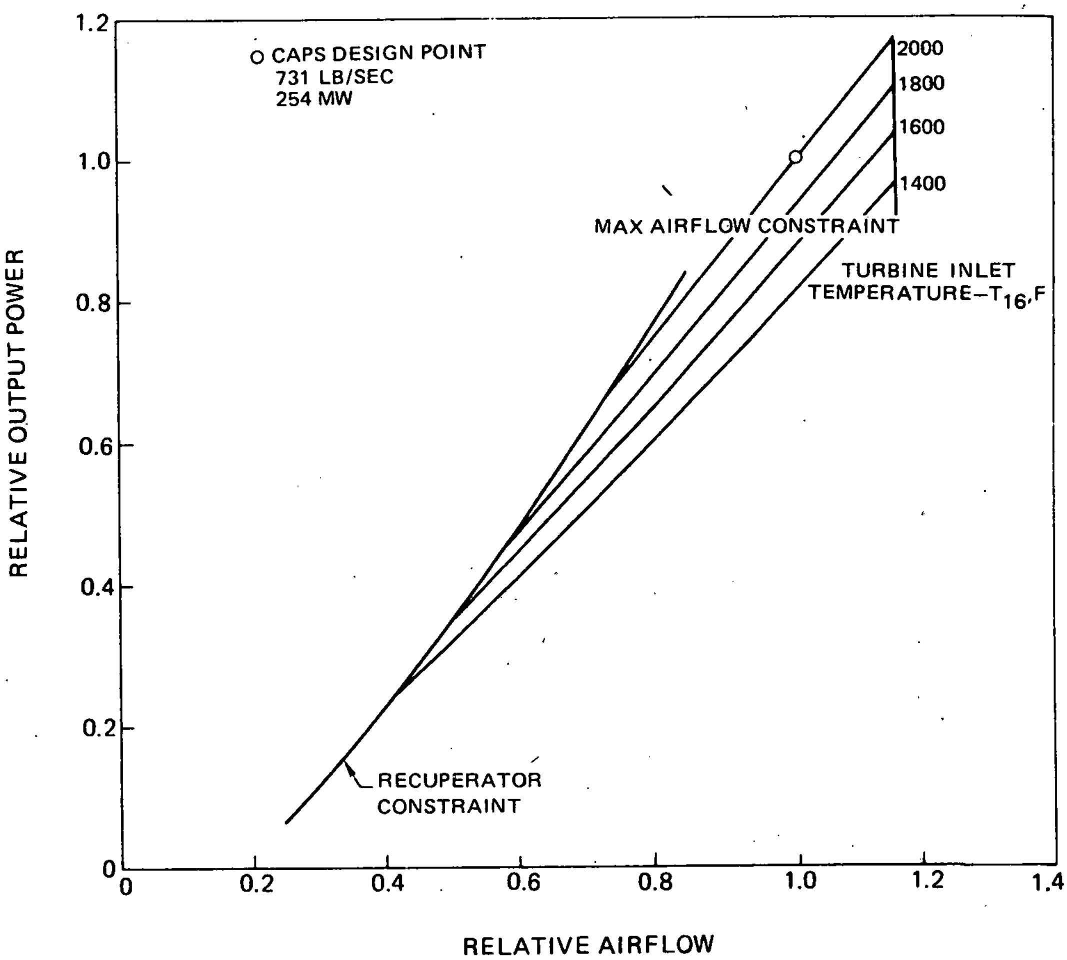
MID–1976 START
CAPITAL CHARGES=18%
ELECTRIC ENERGY COSTS=15 MILLS/ KWHR
FUEL COSTS=$2.50/10^6 BTU
O&M CHARGE=2 MILLS/KWHR
LOW–PRESSURE TURBOMACHINERY BASED ON FT50 DESIGN



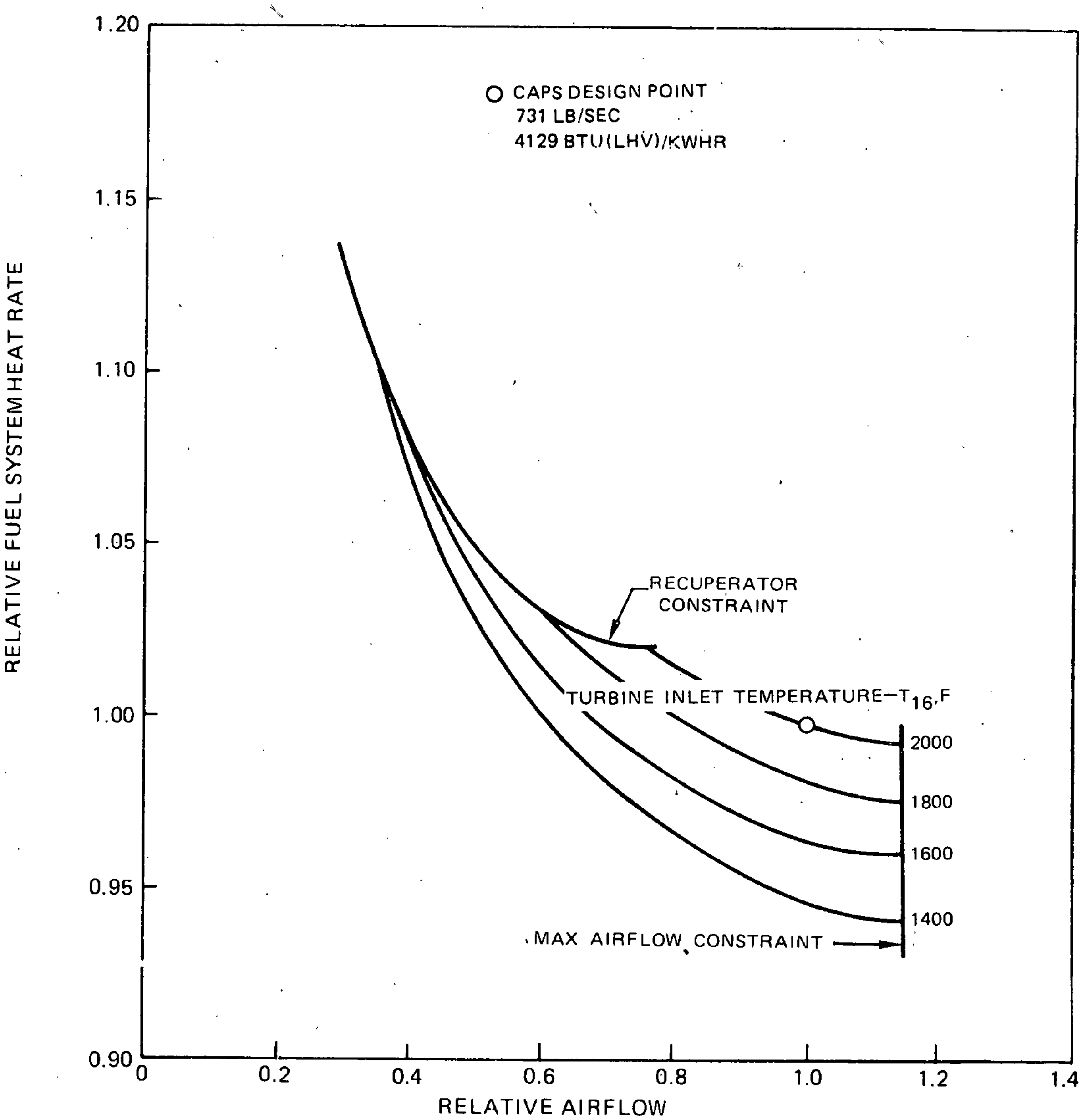
OFF–DESIGN AIRFLOW CONSTRAINTS



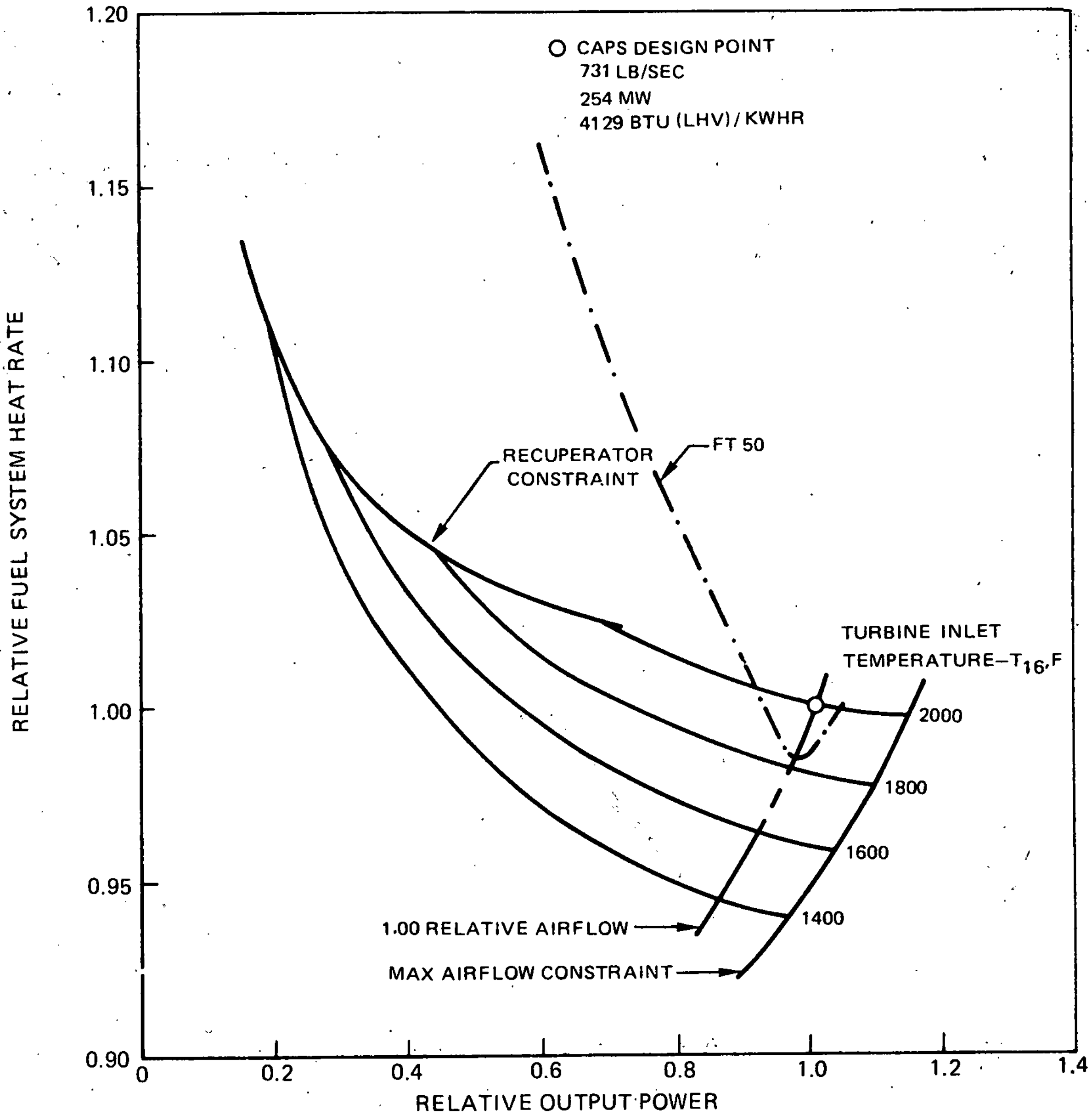
OFF–DESIGN OUTPUT POWER



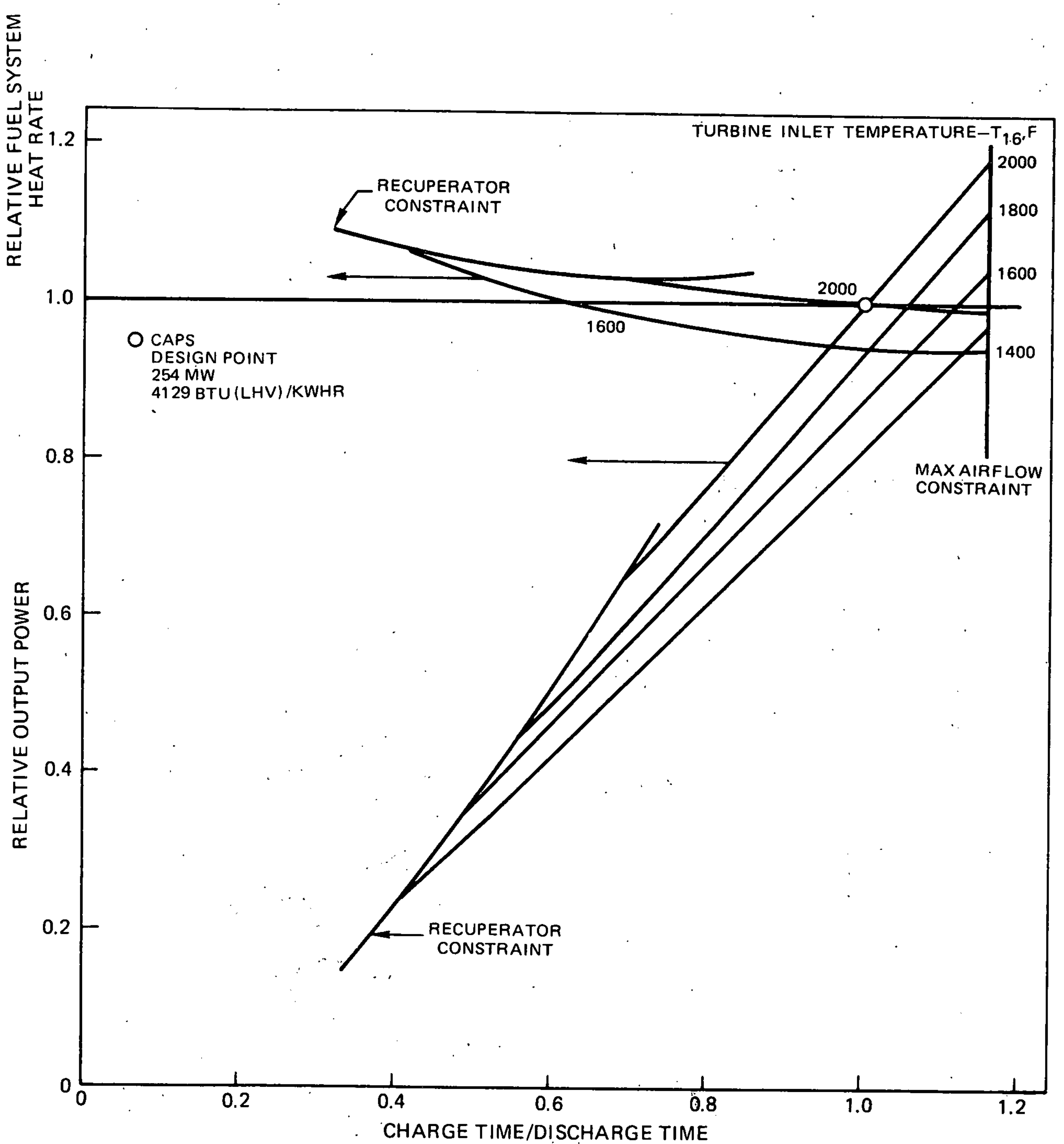
OFF–DESIGN FUEL SYSTEM HEAT RATE



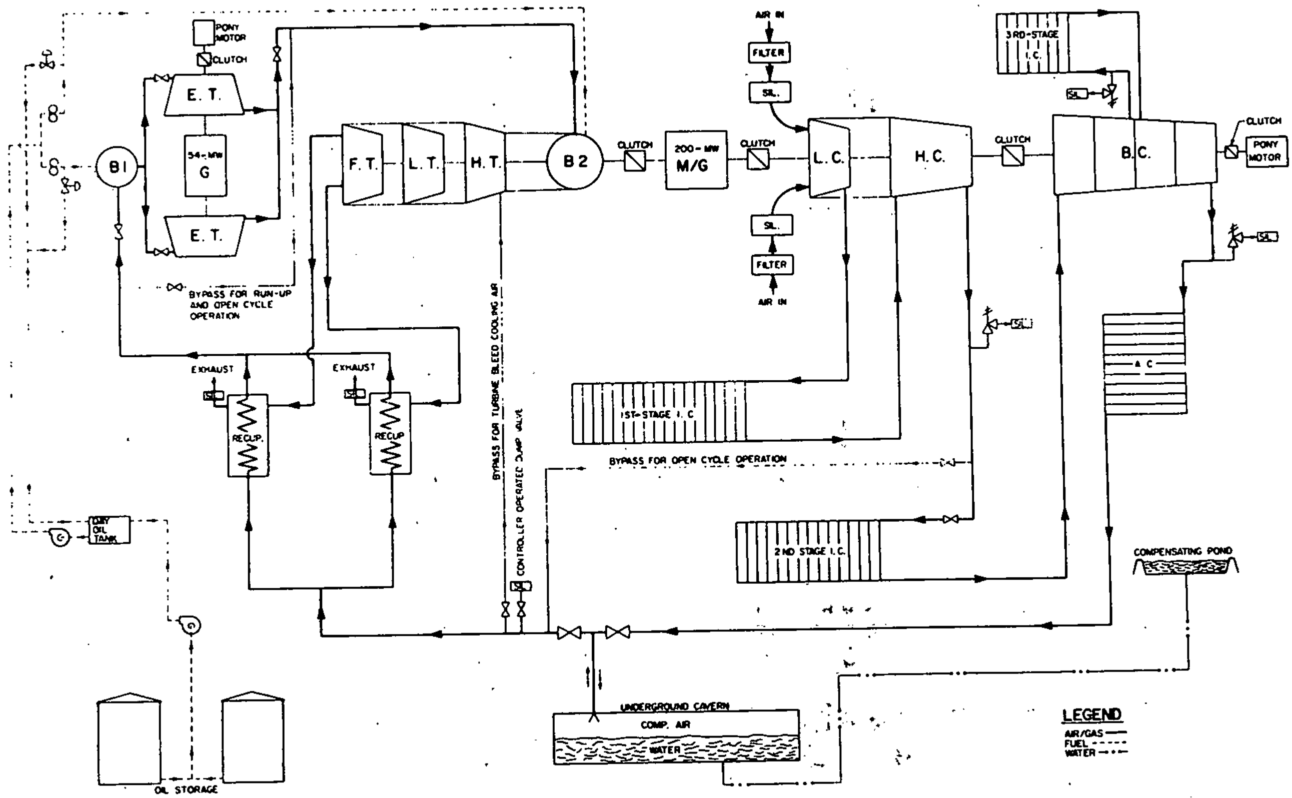
OFF–DESIGN PERFORMANCE



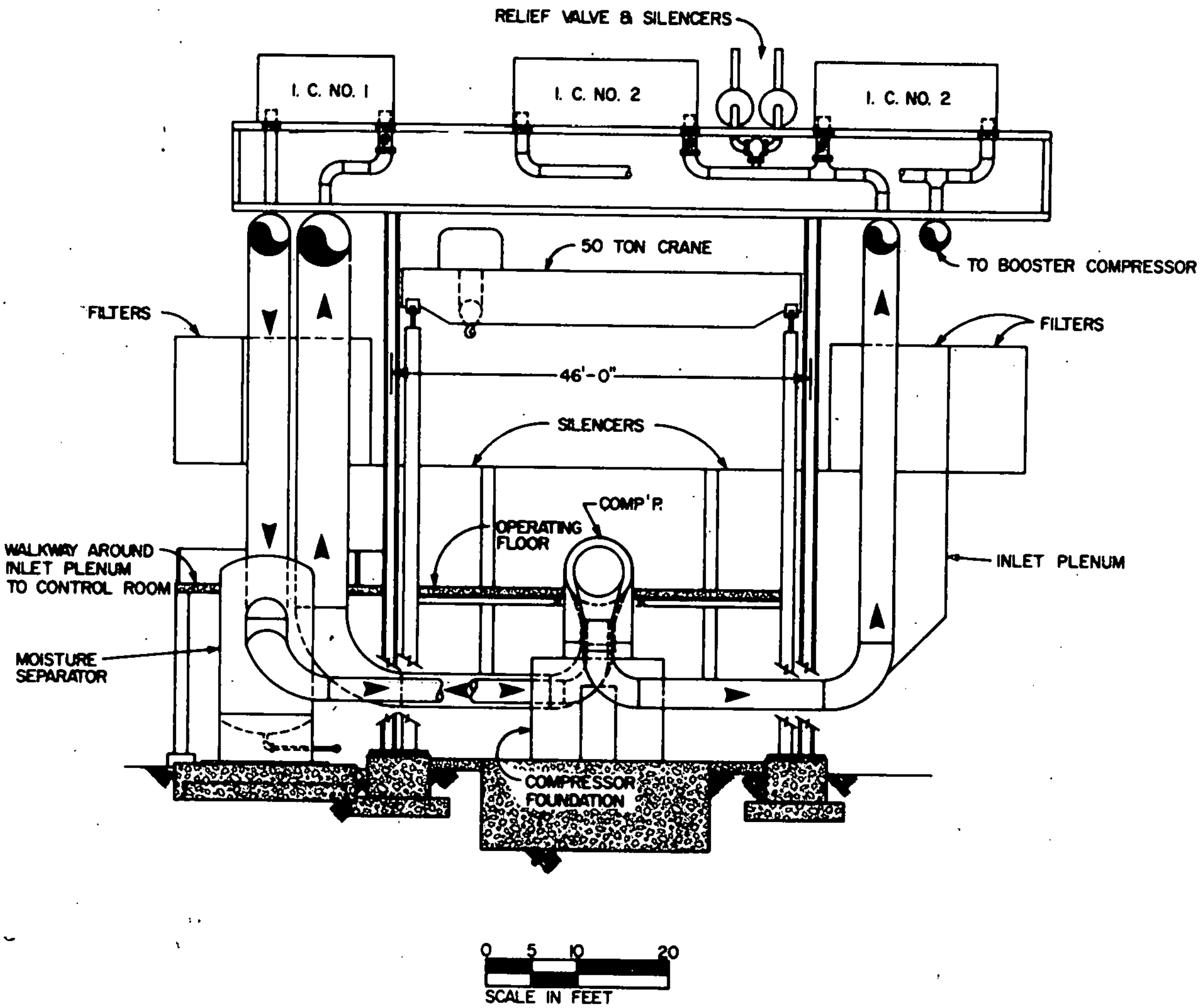
CHARGE/DISCHARGE TIME RATIO



COOLING SYSTEM AND PIPING SCHEMATIC DIAGRAM



PIPING ELEVATION FOR AIR – TO – AIR INTERCOOLERS



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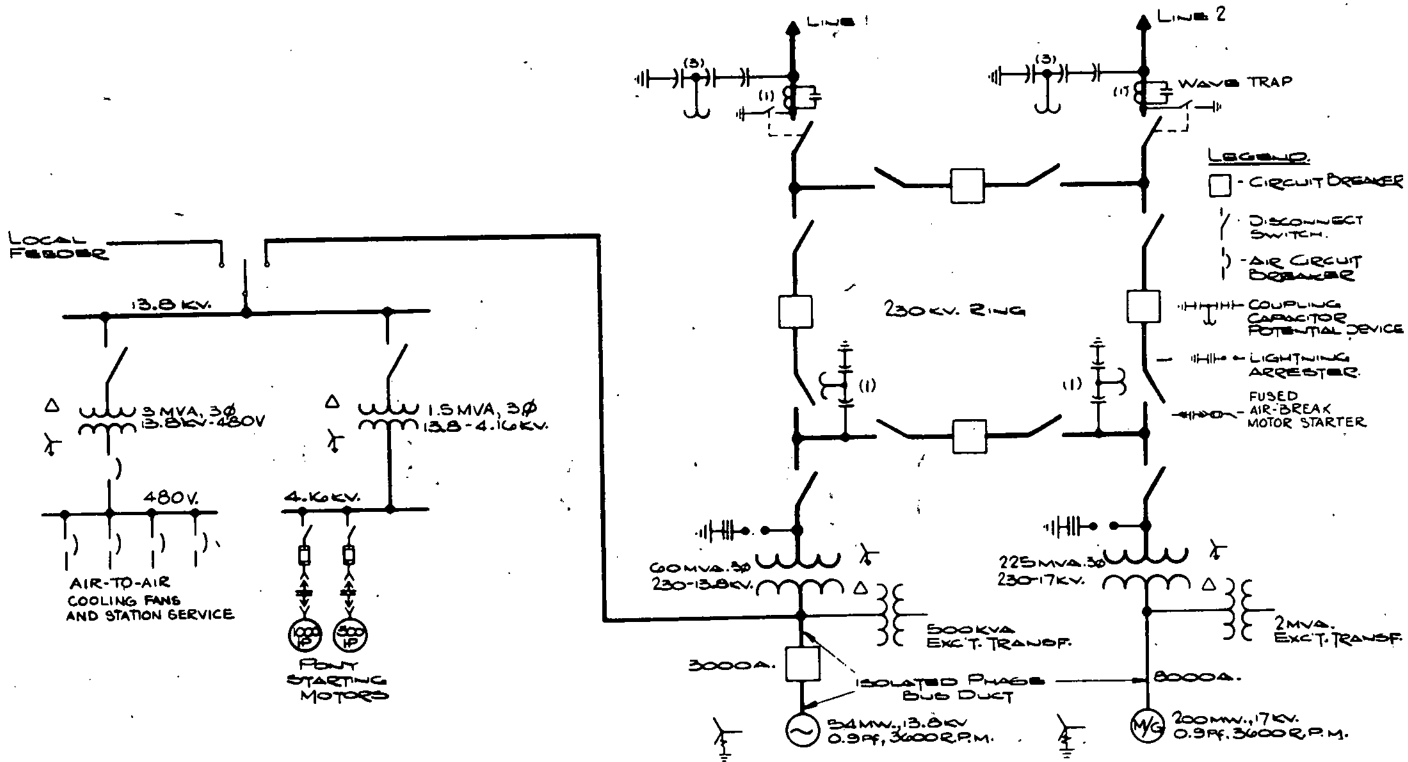
ELECTRICAL SINGLE LINE DIAGRAM FOR SINGLE UNIT



76-12-156-8

FIG. V-36

ELECTRICAL SINGLE LINE DIAGRAM FOR THREE UNITS

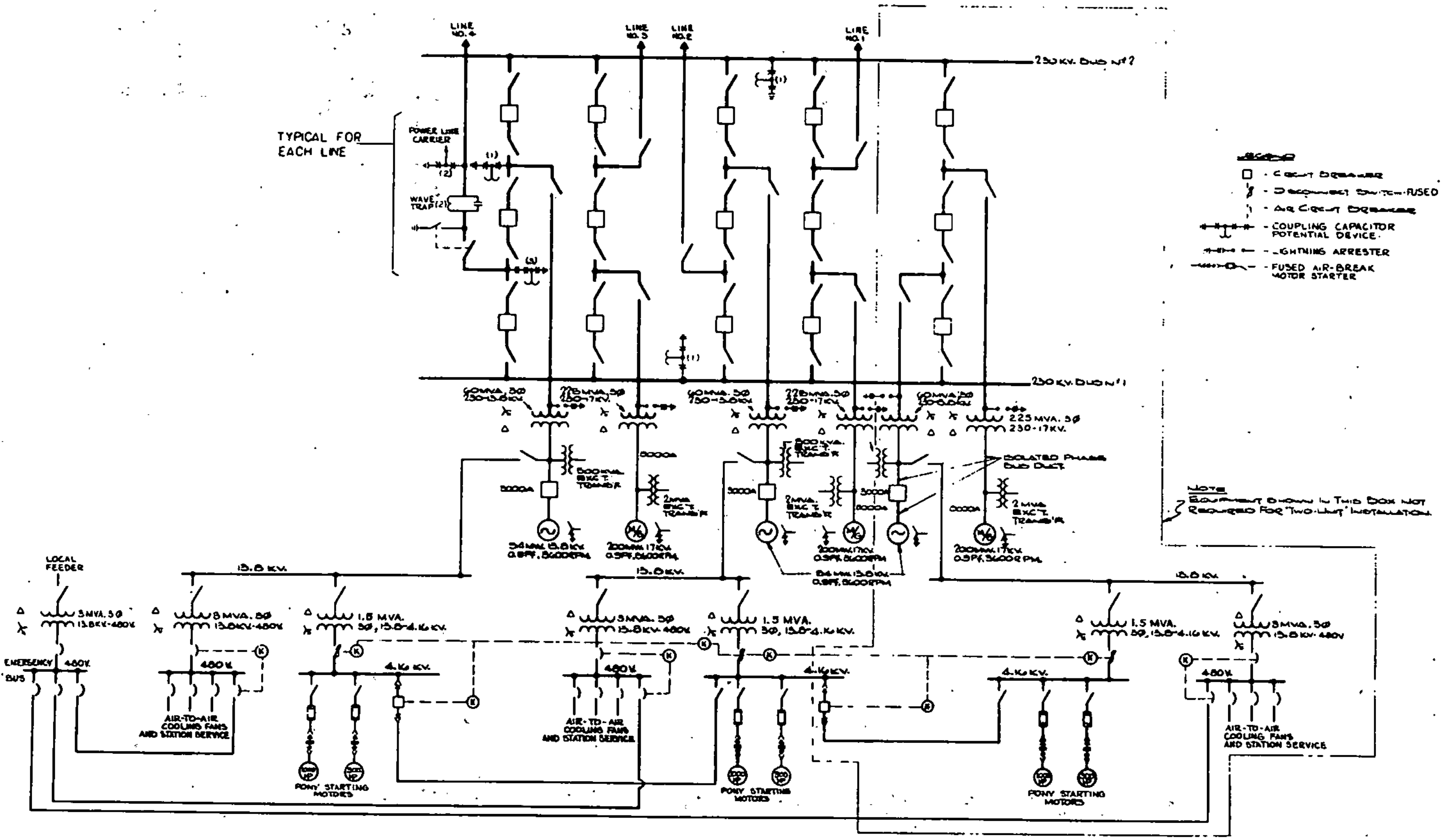
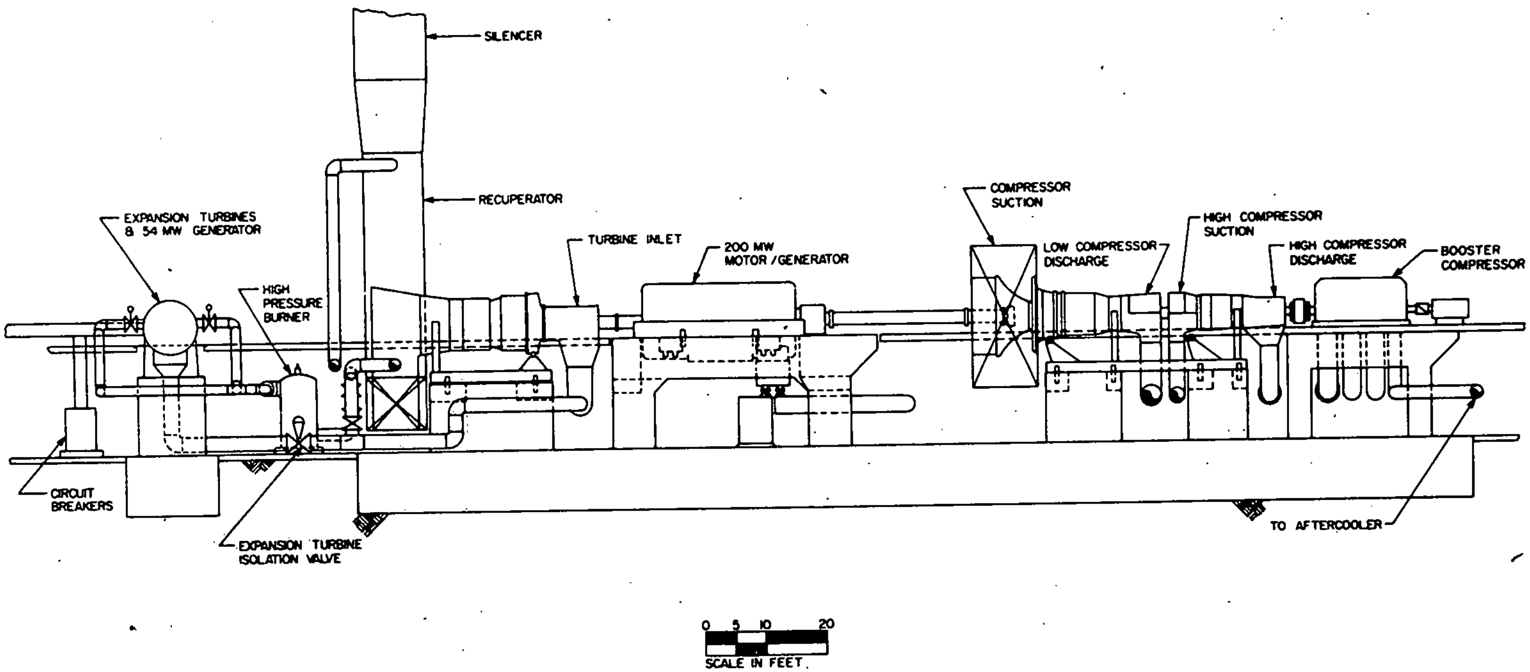
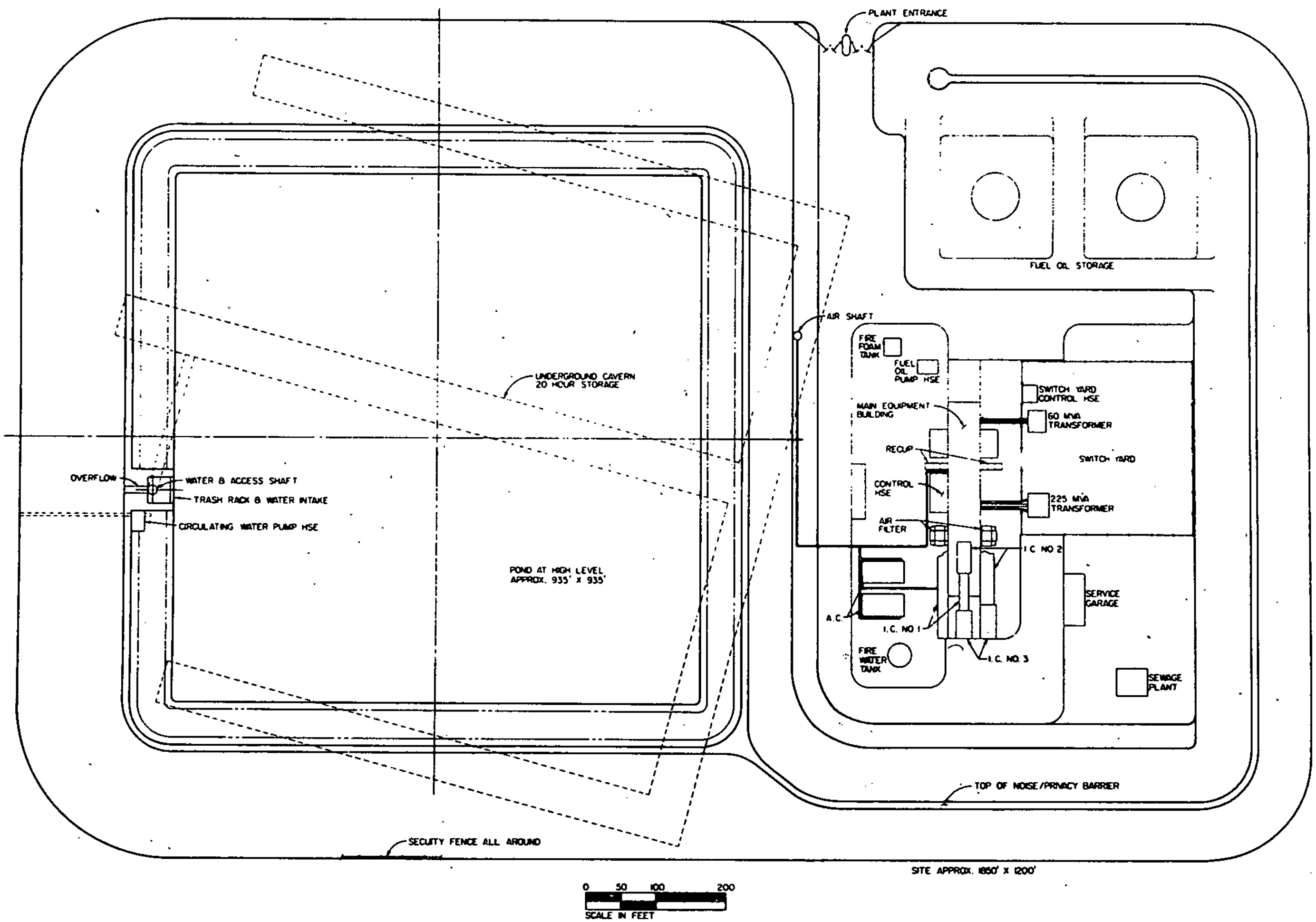


FIG. V-37

TURBOMACHINERY ELEVATION

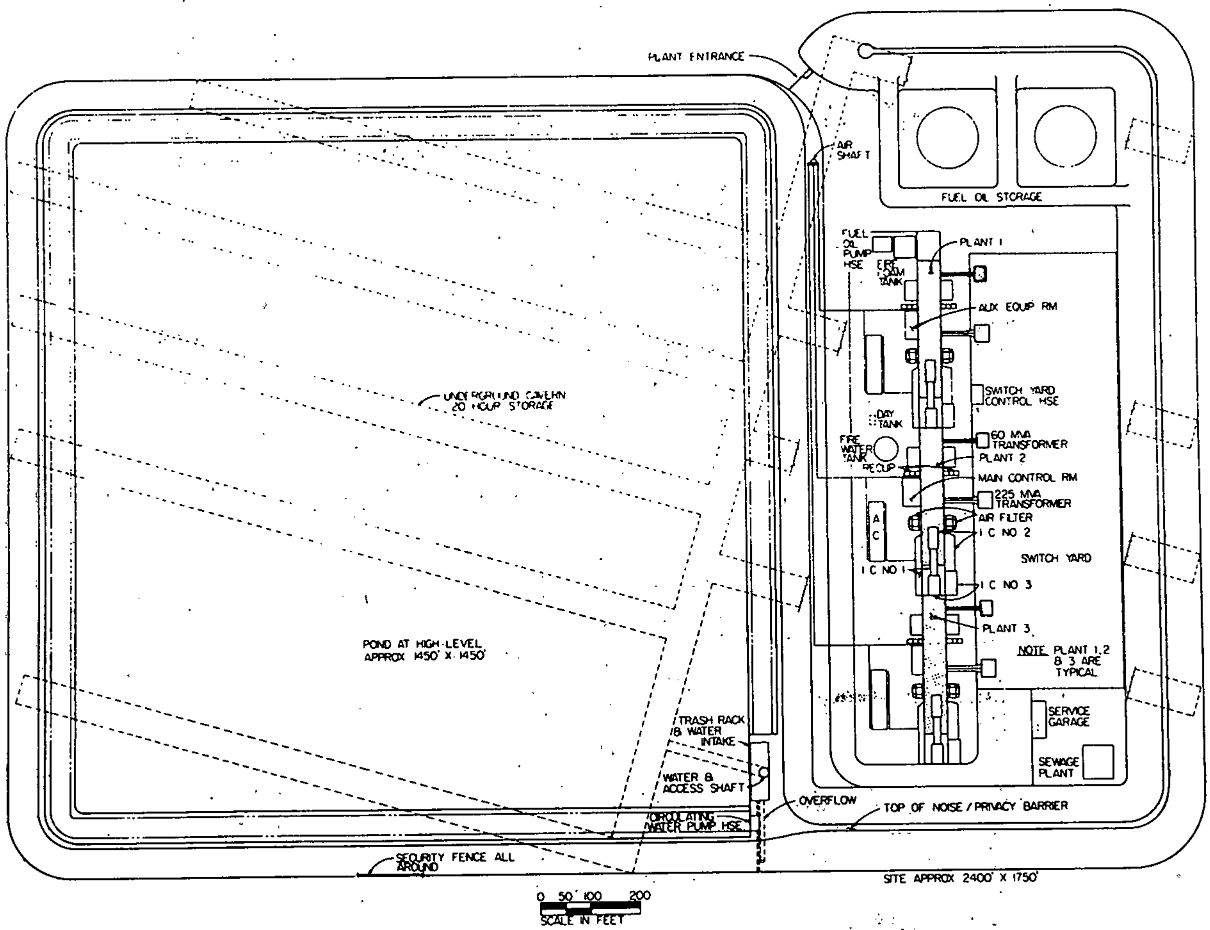


CAPS PLOT PLAN FOR SINGLE UNIT

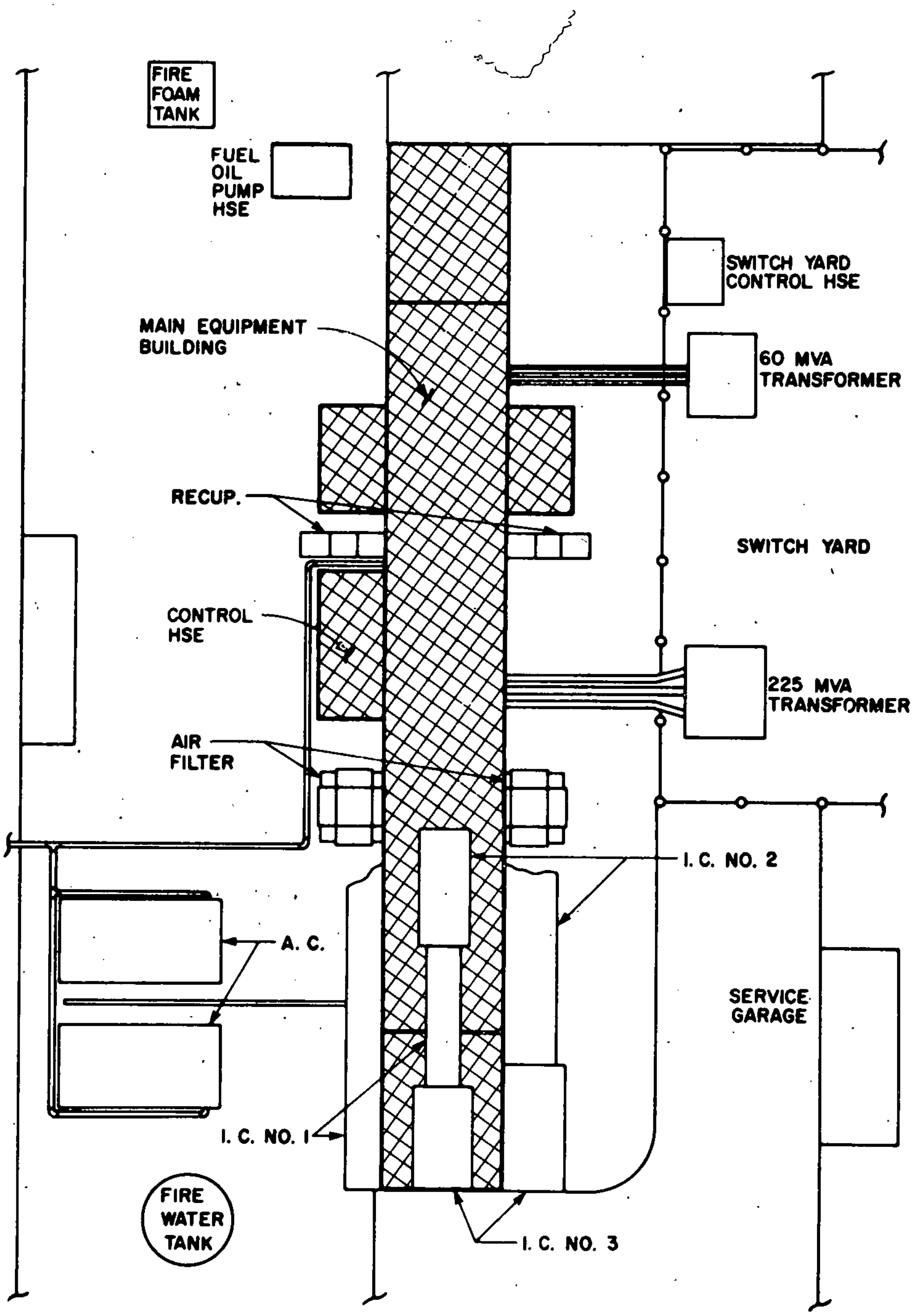


76-12-156-7

CAPS PLOT PLAN FOR THREE UNITS



MAIN EQUIPMENT AREA



76–12–156–1

R76–952161–5

CAPS PLANT ELEVATION FOR SINGLE UNIT

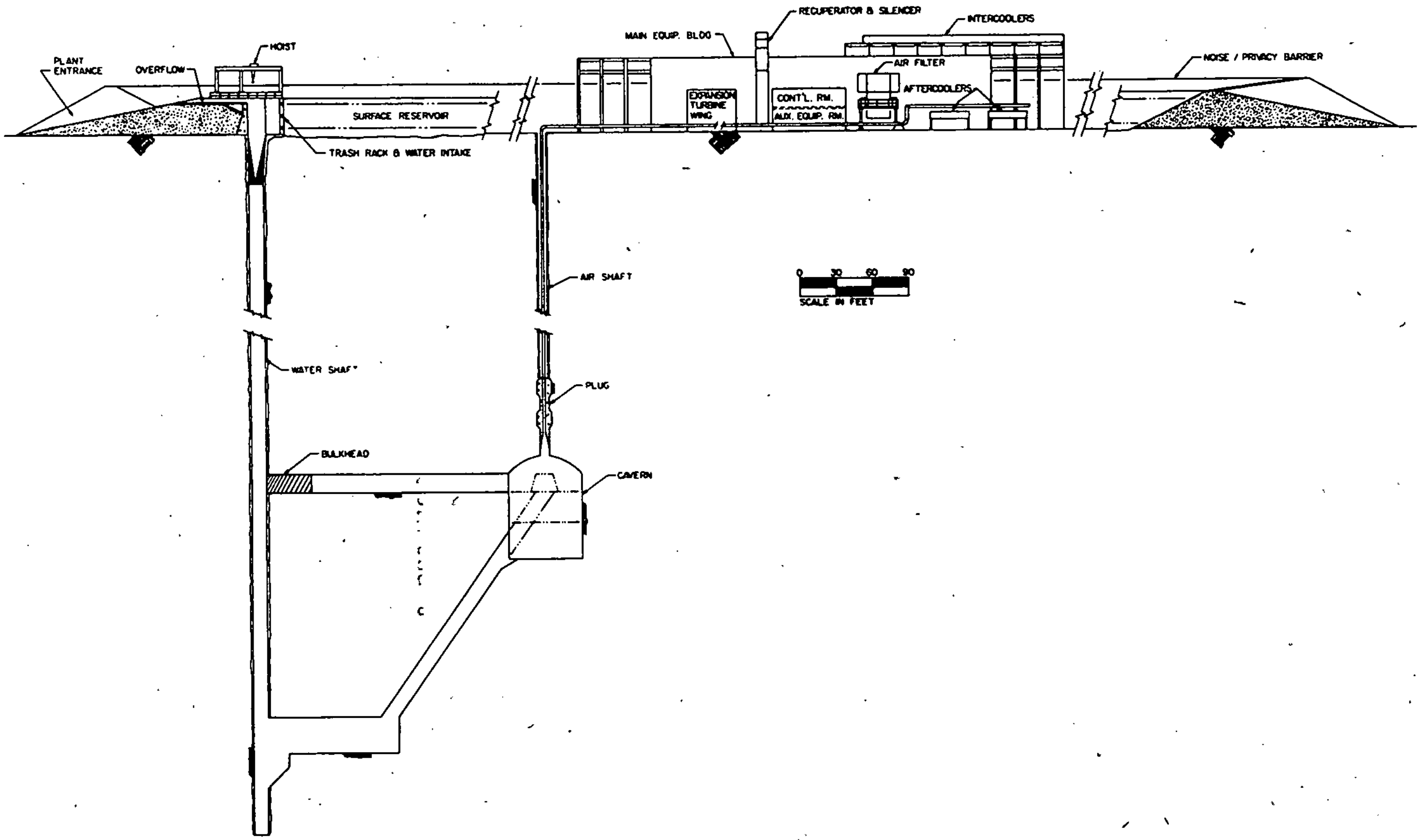


FIG. V–42

76–12–156–4

PART VI

ELECTRIC UTILITY APPLICATION STUDIES

During the final phase of this program, the economics of applying compressed air power systems to electric utility systems were evaluated. Results from this evaluation are reported in this part of the report. The first section describes the alternative types of generating equipment which electric utilities can use to meet their load requirements. The second section discusses general economic considerations such as load demands and competitive characteristics. The third section contains an economic evaluation of CAPS based on two specific case studies and general considerations of siting, comparative economics, and fuel usage. Finally, a brief discussion is given of the future prognosis for CAPS.

GENERATION ALTERNATIVES

Each electric utility in the United States is legally obligated to meet whatever demands for electric power might occur in the service area in which it has a franchise. The demands for electricity which must be met over the course of a year are often represented by a load duration curve in which each hourly load is shown as a function of the accumulated number of hours per year it persists. An example of a representative load duration curve is presented in Fig. VI-1. Three broad categories of power demand can be identified. Base load is essentially constant and generally results from heavy industrial power consumption, nighttime heating and cooling loads, and daytime office, light industry, and residential usage. Intermediate load generally results from steady office, light industry, and residential usage which is not balanced by nighttime loads. Peaking load is typically a short term load, often resulting from the use of air conditioning equipment in residences and offices for summer peak utilities or early evening lighting and appliance loads for winter peak utilities.

Equipment Mix

In selecting a mix of generating equipment to meet its load demands, an electric utility can choose from several types of power plants which range in sophistication from internal combustion engines to nuclear power plants of over 1000 MW. The various types of generating equipment differ in their capital and operating costs and, therefore, differ as to the type of load demand which they can best supply.

Base load demands on most utility systems are met with plants ranging in capacity from 300 to 1300 MW and comprising the most modern, high-efficiency, large-capacity units. With high capital costs but low operating costs, these units are designed specifically for use at high capacity factors. These plants are generally Rankine-cycle steam plants, including pressurized water and boiling water nuclear reactors and coal-, oil-, or natural gas-fired steam generators.

Intermediate or mid-range loads are usually met with older fossil-fueled steam plants originally installed to provide generating capacity for base loads. As newer, more reliable units which can supply base load power at lower cost are added to the system, older units are displaced to the intermediate range to provide power that the base load unit could not efficiently provide. To minimize the forced outages which would result from using the older units in a manner for which they are not completely suited, system temperatures are frequently maintained between generation periods by keeping boilers fired at minimum load. Although thermal shocks associated with rapid load rise are thereby avoided, fuel useage is substantially increased.

Alternative sources of mid-range power in recent years have been steam plants especially designed for cycling duty and combined-cycle plants. With combined cycle power plants, the exhaust gases from a large gas turbine are fed to a heat recovery boiler designed for cycling service and which, in turn, supplies a steam turbine. Conversion to combined-cycle operation, termed repowering, can also allow the useful life of a small existing steam plant to be extended. For these conversions, the steam turbine generator of the existing plant is retained, a gas turbine and heat recovery boiler are added, and the original boiler is scrapped.

Peak loads are generally met with units which are specifically designed for loads of short duration. Simple or regenerative cycle gas turbines or nonreheat steam plants are typically used for this purpose. Such peaking units are typically designed to avoid complication and, thereby, to ease maintenance. Thermal capacitance is kept low to minimize forced outages. Generation costs (mills/kWhr) with peaking plants vary from utility to utility but generally are approximately double the costs with base-load plants. It is true that fixed costs of peaking units are low because capital costs are lower than for base load units and because relatively uncomplicated designs allow lighter staffing of peaking plants. However, because of the low capacity factors for these plants the annual owning costs per unit of energy generated is very high. Also, the operating costs of peaking plants are high when compared with those of other types of plants. One reason is that heat rates are higher with peaking plants, especially at part load. A second reason is that the distillate oil usually burned in gas turbines is more expensive than other types of fuel.

Some types of electric generating plants, such as run of river hydroelectric plants, can be used to meet base loads, mid-range loads or peaking loads as required. Hydroelectric power can typically be generated at lower cost than is possible with other generation alternatives; there are no fuel costs, and fixed charges are low because the economic life of a hydroelectric plant is long. (By law, a life of fifty years is allowed for tax purposes.) On the other hand, rainfall upstream could limit hydroelectric capacity during critical peak summer periods. Moreover, nearly all practical hydroelectric power sites in the northeastern U. S. have been utilized. Thus, hydroelectric power is generally a factor only as an element of the existing generating mix of a utility.

The Role of Energy Storage

Energy storage has been used by some electric utilities as an alternative to the direct addition of generating equipment. Conventional pumped storage hydro, the only commercially proven method of storage, has become popular in the north where the terrain suits such plants, but concern about the environment and basic economics have recently limited the number of sites which remain to be developed. However, other methods of storage, including compressed air power systems, underground pumped storage hydro, oil thermal storage, hot water storage, flywheels, batteries, and super-conducting magnets, are under development. Of these, only underground pumped storage hydro, compressed air, and oil thermal storage appear to have near-term promise (Ref. VI-1).

The role of energy storage on electric systems has usually been defined by the need for peaking capacity. An electric utility must have generating equipment sufficient to meet the maximum or peak demand on its sytem whenever it occurs. Because peak demands on a system occur for only a small accumulated fraction of the year, the generating equipment which is used to meet the peak demands necessarily operates at low capacity factors. A demand thus arises for a means of providing peaking capacity which can be economically operated at low capacity factors.

One basis for this demand is economic; capacity specifically designed for peaking purposes can reduce the long term costs of a utility system. This is particularly true when compared to the practice of some utilities of meeting peak loads by using older generating units which have been displaced from base load operation by newer, more efficient units. This procedure has been possible because the units added to many fossil fired plants during the 1940's and 1950's ranged in size from 100 to 300 MW and could, to some extent, be cycled fairly rapidly. As units are displaced from the base load, however, their annual capacity factor necessarily drops leading to a lower lifetime capacity factor. When generating equipment designed to operate at low capacity factors is added specifically for peaking capacity, the annual capacity factors of base loaded units can be maintained at a high value throughout their life resulting in a higher lifetime capacity factor. With this higher capacity factor, the large fixed costs of a base load unit can be spread over a larger output, and the unit cost imputed for electricity generated by the plant can be reduced. A higher capacity factor for a base load unit can also provide savings in fuel costs. With the expected lifetime capacity factor affecting the heat rate chosen for a new base load unit, the higher the capacity factor expected the lower the heat rate which it is economical to choose.

Recently a new demand for generating equipment specifically designed for peaking capacity has arisen. The units of 500 MW or more which have been installed on many utilities cannot be displaced to peaking capacity because they are not suited to rapid cycling and operation at low capacity factors. That is, because of the very large metal and refractory surfaces in these units, between 6 and 24 hours are required to rise to full load from either cold start or low load. Rapid cycling merely invites failures in piping, tubes, or refractory; this is particularly true with nuclear power plants. Since such units cannot operate at the low capacity factors inherent for peaking capacity, an explicit need arises for generating equipment which can be economically operated at low capacity factors.

A major function of energy storage on an electric system is to provide this peaking capacity. In the past, however, energy storage could be considered for peaking capacity only by electric utilities located in areas where geologic conditions allowed the use of conventional pumped hydro systems. Most utilities, therefore, installed gas turbines to provide peaking capacity. Because the methods of storage now being developed will be less restricted by geologic conditions, energy storage on electric systems as an alternative to gas turbines might be considered by more utilities. Addition of any energy storage to the generating capacity of an electric system will require, however, an economic evaluation of the storage system relative to alternatives.

GENERAL ECONOMIC CONSIDERATIONS

The factors which must be considered in the economic evaluation of additions to generating capacity include the load demands on the utility and the costs, primarily capital and fuel, of the alternative power plants which could be used to meet these loads. The evaluation of a storage system must also be based on consideration of the cost of off-peak electricity used for pumping purposes.

Utility Load Demands

An evaluation of a storage system must take into account not only the magnitude and duration of the peak loads which the system will help to meet, but also the off-peak loads which will determine how much energy is available for pumping the system. Summary data for 1971 provided in a recent study by Public Service Electric and Gas Company (N. J.) show significant variation in load characteristics among utilities (Ref. VI-1). The load characteristics of utilities might be different in the future, however, because since 1971 utilities have increased their efforts to improve system load factors and to control the peak load demands on their systems. To the extent that these efforts succeed, the requirements which any method of storage will have to meet in coming years to be considered for inclusion in a utility's generating expansion might differ from present requirements.

System Load Factors

Figure VI-2 shows the cumulative frequency distribution of system load factors in 1971 (from Ref. VI-1). For most utilities, these load factors were lower than in the early 1960's. Underlying these declines in load factors, which contributed significantly to utility needs for peaking capacity, was growth in air conditioning loads. The spreading use of air conditioning in the residential and commercial sectors during the 1960's shifted many utilities from a winter peak to a summer peak and caused summer peak loads to increase more from year to year than winter peaks. Even with the air conditioning market likely to grow more slowly in coming years, the growth which is expected in the use of electricity for home heating should contribute to increasing system load factors for many utilities. Thus, electric reliability councils in regions where electric heating has the most potential are projecting increases in load factors. However, declines in load factors are projected to continue in the southern and western states where heating loads will not balance summer air conditioning loads (see Ref. VI-2).

Capacity factors for utilities will improve because of increases in system load factors and also because of diversity interchanges among utilities. Transmission inter-ties among utilities permit the peak load on one utility to be met by he other utilities in the pool. In effect, the surplus capacity of one utility provides the standby and spinning reserve required by another. Existing generating capacity is thereby utilized more fully, and required additions to generating capacity are lower.

However, system load factors and capacity factors calculated from annual data can create a false impression of the practical possibilities of storing off-peak energy. Although annual load factors for individual utility systems may range from about 35 percent to almost 80 percent, daily load factors typically range from about 75 percent to 90 percent. Annual load factors are lower than daily load factors because many utilities have significant seasonal and weekly differences in their load demands. To utilize completely the percentage of off-peak energy which appears available from inspection of an annual load duration curve would, therefore, require a means of seasonal storage. The capital costs for seasonal storage facilities, however, are excessive. Apparently more viable are the daily storage systems using nighttime off-peak energy and weekly storage systems using nighttime energy plus energy stored from weekend generation.

If system load factors and capacity factors improve, the design requirements of a storage system also become more stringent. Improved load factors reduce the amount of off-peak energy which can be stored to supply peak needs. Figure VI-3, based on 1971 data, shows this inverse relationship between system load factors and the amount of peak energy which can be transferred from off-peak through a storage system. As system load factors increase, only storage systems of high efficiency designed for weekly cycles remain viable. With a high system load factor limiting the charging time available from base load capacity, a high ratio of discharge time to charge time becomes more valuable.

Load Management

In coming years, utilities might give more consideration to introducing changes in rate schedules in an effort to control peak loads. Under a system of peak load pricing, for example, electricity consumed during periods of peak demand would be priced higher than electricity consumed during off-peak periods. Although a change to peak load pricing could, theoretically, dampen the growth in the peak demand for electricity, empirical evidence with which to measure the possible magnitude of this effect is scanty because such rate schedules have had limited application in the US. However, limited experiences in other countries (e.g., France, Germany and the United Kingdom) in structuring electricity rates which vary by time of day have induced experimental programs in the US to determine whether alternative rate structures would alter the pattern of electricity demand. Adoption of different rate schedules by utilities across the country will be conditional in large part on the success of these experimental programs.

Utilities will also try more direct methods of controlling the load demands on their systems. These methods include time-controlled water heating which can be turned off during peak demand periods, systems which substitute heat from storage for electric heat during peak demand periods, and load shedding devices which turn off selected equipment for short periods when the maximum system load is being approached. Although such load management techniques would probably have their most significant impact in the residential sector, their application could be limited if utilities incur large costs by investing in control devices.

To the extent that these approaches have any effect, the daily load shapes for many utilities might be flatter in coming years than at present. Thus, a distribution of peak load durations in the future analogous to that in Fig. VI-4 for 1971 should show durations of 1 to 4 hours occurring less frequently, and durations of 5 or more hours occurring more frequently. Any system of energy storage would therefore be required to deliver power more hours a day than is presently required of peaking capacity. This trend might tend to make large capacity storage systems like CAPS and undergound pumped hydro more attractive in the future, relative to systems like gas turbines and batteries, because their pronounced economies of scale make them more economical at higher capacity factors.

Competitive Cost Considerations

The economic evaluation of additions to generating capacity should involve calculations for a complete utility system and compare the total costs of operating the system with alternative mixes of generating capacity. System costs are important because installation of a particular type of capacity could affect production costs for other capacity on the system. In the past, for example, when addition of a new base load plant could be considered as one alternative for peaking capacity, the displacement of older units to loads of shorter duration would necessarily affect their operating costs as their capacity factors dropped. Similarly, addition of a compressed air power system to capacity could impact the costs of operation differently than addition of, say, gas turbines. The energy for compression must be supplied by plants at or near the base load, thus affecting their operating costs.

Selected economic evaluations for complete utility systems are contained in the next section. This section contains simple order-of-magnitude comparisons between expected costs for CAPS plants and the costs which are known for existing types of generation capacity performing comparable service. Specifically, trade off comparisons are made for the costs of compression electricity, fuel, and capital for CAPS relative to the comparable costs for gas turbines, conventional pumped hydro, and combined cycle systems.

Energy Costs

The annual fuel costs of operating gas turbines and combined-cycle plants depend on the price of the distillate oil which must be burned, the respective heat rates, and the total hours of operation. Pumped hydro systems burn no fuel directly while generating electricity, but an indirect fuel cost must be charged against the plant because of the off-peak energy required for pumping. A CAPS plant is a hybrid system having both of these characteristics; it uses stored energy for most of its power but also burns some fuel.

Energy costs for a gas turbine and for a compressed air power system were compared for different annual hours of operation, for different prices of distillate oil, and for different costs of electricity used for compression. The heat rate assumed for a gas turbine was 12220 Btu(HHV)/kWhr and for a compressed air storage system was 4377 Btu(HHV)/kWhr. The compressed air storage system was assumed to require 0.82 kWhr of off-peak electricity to compress air for one kWhr of peak load generation. As would be expected, energy costs for CAPS are lower than for a gas turbine over a wide variation of the parameters. The savings in energy costs can be regarded as a credit to CAPS vis-a-vis a gas turbine and are plotted in $/kWyr in Fig. VI-5 for 1560 hours of operation and in Fig. VI-6 for 2190 hours of operation. Other things being equal, savings in energy costs with CAPS are higher for lower costs of electricity used in compression, for higher prices of distillate oil, and for longer hours of operation.

Energy costs were also compared between CAPS and pumped hydro (either conventional or underground), where the energy cost for both systems included the cost of off-peak electricity used for compression/pumping. A pumped hydro installation was assumed to require 1.4 kWhr of electricity for pumping for each kWhr of electricity generated. In this case, energy costs for CAPS were higher than for pumped hydro over a wide variation of the parameters, and this difference can be regarded as a cost penalty against CAPS vis-a-vis pumped hydro. These penalies are plotted in Fig. VI-7 for 1560 hours of operation and in Fig. VI-8 for 2190 hours of operation. Other things being equal, the energy cost penalty charged against CAPS vis-a-vis pumped hydro is lower for higher costs of compression/pumping electricity, for lower prices of distillate oil, and for lower hours of annual operation.

Finally, the energy costs of CAPS were compared to those of a combined cycle system. The heat rate for the combined cycle system was assumed to be 8550 Btu/kWhr. In this case, the credit for energy cost savings which could be given to CAPS was especially sensitive to variations in the cost of electricity for compression and in the price of distillate oil. For high prices of distillate oil and low costs of compression electricity, energy costs for CAPS are lower than for a combined cycle and can be regarded as a credit for CAPS. The credits/penalties are plotted in Fig. VI-9 for 2190 hours and in Fig. VI-10 for 3125 hours.

Capital Costs

Addition of a CAPS plant to a utility's generating capacity would entail annual carrying charges on the capital invested in the plant. The magnitude of these carrying charges depends on the construction cost of the plant and on the fixed charge rate applied against the construction cost.

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The construction cost, in $/kW, for nine combinations of CAPS plant size (number of units) and cavern storage capacity were estimated in Part V and summarized in Table V-13 in June 1976 dollars. Obviously, any future plant would incur escalation and interest charges until construction is completed. The magnitude of these additional charges will vary with construction time and depend on the specific escalation and interest rates estimated by each utility. For construction times from 3 to 5 years, escalation at 6 to 9 percent per year, and interest at 9 to 12 percent per year, these additional charges could increase the net construction cost by 30 to 60 percent.

The fixed charge rate to be applied against the estimates for construction cost depends on the cost of capital to the utility, dividends to stockholders, tax rates of federal, state, and local governments, the rate of depreciation charged against the plant (related to plant life), administration, insurance rates and permitted allowances for plants under construction. Given this complex of factors, the fixed charge rate would necessarily vary from plant to plant with values ranging from 15 to 30 percent depending on plant life and utility factors. However, the fixed charge rate for CAPS should be somewhere in between those for gas turbines and pumped hydro because the essential features of CAPS are common to gas turbines and pumped hydro systems.

As a first approximation to the competitiveness of a CAPS plant relative to alternatives, the differences in energy costs between CAPS and alternatives plotted in Figs. VI-5 through VI-10 can be compared to the differences in the annual fixed charges on the capital which would be invested in each type of generating capacity. That is, the energy credit (or penalty) of CAPS can be subtracted from (added to) the annual fixed charges ($/kWyr) of the CAPS plant, and the resulting value then compared to the annual fixed charges of the alternative. Where this value falls below the carrying charge of the alternative, investment in CAPS appears attractive.

Such simple comparisons of differences in fuel and capital costs do not take into account costs for operation and maintenance and for transmission which might affect the competitive balance between a CAPS plant and an alternative. Moreover, even these simple comparisons require assumptions about the costs of distillate oil, about the costs of electricity used for compression, and about the first costs and fixed charge rates for alternative generating equipment. Because these various costs are likely to vary from utility to utility, economic evaluation of CAPS relative to alternatives on specific utility case studies is of considerable interest.

Miscellaneous Factors

Other factors might also influence the choice between CAPS and alternative generating equipment. In particular, reliability and operating flexibility are important considerations for generating equipment used to meet intermediate and peak loads. While gas turbines and combined cycle plants might appear potentially attractive in these respects, fuel availability might limit the extent to which they could be used. Thus, combining a means for energy storage with base load power plants might be advantageous for reasons other than economics. A CAPS plant, for example, could be substantially more flexible than conventional alternatives, and its dependence on the availability of fuel would be less.

ECONOMIC EVALUATION OF CAPS

Method of Analysis

Among the numerical computer models and other analytical tools used for planning future generation in the electric utility industry, four generic types of models are quite widely used. These are:

1. Screening curves
2. Production cost/dispatch models
3. Reliability/capacity margin models
4. Optimum expansion models

Screening curves are graphs of the busbar power costs ($/kWyr or the equivalent) versus hours per year of use for competing types of new power generation equipment. The annual costs for each power plant type are calculated as the sum of the annualized capital cost (the capital cost times an annual cost recovery factor), the operating and maintenance cost, and the fuel cost (the heat rate times the price of fuel). The result is a series of comparative curves with the plant producing lowest cost power (at the busbar) on the bottom. The bottom line plant will vary according to the number of hours the plant is run per year. A low capital cost plant (e.g., gas turbines) will produce lowest cost power for a small number of hours per year while a low fuel cost plant (e.g., coal or nuclear) will produce lowest cost power for a large number of hours per year. Generally speaking, a plant must be at or near the bottom line in order to be bought by a utility. (Additional factors such as environmental impact, flexibility, fuel supply, and siting modify a strict bottom line decision.) The screening curves should be calculated using operating and maintenance costs and heat rate curves which vary according to the number of hours per year of use. A plant which is used for summer or winter peaking only for a few hours a day will have a higher heat rate than the same plant run at base load. Certain maintenance functions will have to be performed whether or not a plant is being run extensively while others are proportional to plant use. In summary, a screening curve provides a preliminary estimate of whether a new power plant is competitive with alternative types of generation. Example screening curves are given in the next section.

Production cost, or dispatch, computer models are used to simulate the actual hour-by-hour decision process as to which power plants should be turned on or off to meet the demand for electricity. These models are typically run to simulate a multi-year time period for an assumed mix of existing and new power plants. Then another expansion pattern of new power plants can be assumed and the program run again. By comparing the results of numerous successive runs, a satisfactory mix of new equipment can be determined. Production cost models enjoy a high level of acceptance in the electric utility industry. They require a large amount of utility data, much of which is usually proprietary. For example, part-load heat rates for each unit on a system, unit cost data, and demand projections for up to thirty years (or plant lifetime) are required. Storage plants provide a problem for the hourly dispatch of power, whether real or modeled, in that an ad hoc decision procedure must be used to specify how much energy should be put into storage each night and how much of the stored energy should be discharged during the next day. One type of dispatch model solves this problem by dispatching power to a projected weekly load duration curve. By this method the load leveling effects of using nighttime and weekend power to pump up a storage plant are clear.

Reliability, or capacity margin, computer models are used in conjunction with the production cost models to estimate the capacity margin needed to meet a given reliability requirement. Typically, the reliability requirement is that the demand must be met every day except at most one day in ten years. EFOR (Equivalent Forced Outage Rate) and planned maintenance data are required for these probabalistic models. By running a capacity margin model for the same assumed expansion mix of plants and time period as the production cost model, the cost of an expansion mix can be obtained while ensuring a given level of reliability.

Optimum expansion models provide in one computer run the specific mix of power plants that will produce lowest cost power for the multi-year period under study. In this respect they provide information that the production cost model does not. Computer size and time limitations, however, cause the demand to be represented by a load duration curve, typically annual, and the various power plants to be grouped, typically, into no more than fifteen different generic types. For example, a large number of steam plants might be grouped into four or five categories, each with representative cost and operating data. The load duration curve used is divided horizontally into several peak-to-base load demand categories and the production costs, as well as the fixed costs, of satisfying these demands are summed to give a total owning and operating cost for the system over the period studied. The optimum solution calculated is for minimum total owning and operating cost. Optimization models are useful for evaluating generic types of equipment on a broad strategy basis. The general guidelines as to capacity mix and timing of new capacity can then be "fine tuned" by use of a production cost model. One such optimum expansion model, the DYNamic Optimization of the Generating Equipment Mix (DYNOGEM) is described in Appendix K. This model was developed by United Technologies and used for internal market studies of current and future gas turbine products.

Data were obtained from one electric utility and one utility power pool for use in investigating the application of CAPS in realistic utility expansion scenarios. These data were used in the next section, without modification, to prepare screening curves and execute the DYNOGEM optimum expansion model for two case studies corresponding to the utility data. The screening curves permitted an initial analysis of whether CAPS would be competitive with alternate types of new power generation equipment. The optimum expansion program was then used to decide which kind, how many, and when the future additions should be made while considering the plants already owned, or committed to, by the utility under study. If the cost of electricty from two alternate new power plants is close (within a few percent), the program will choose the one which is mathematically cheaper. Clearly, in these cases other factors such as environmental, conservation, siting, and uncertainties in the various cost and technical projections should be considered before making the final equipment selection. A general discussion of these considerations follows the case studies to put the numerical results into proper perspective and identify uncertainties in the input data and assumptions which could affect the competitive position of CAPS.

Case Studies

Two case studies, one for a single utility and one for a utility pool, were undertaken to evaluate CAPS within the context of particular electric utility demand growth, load shape, existing equipment and projected costs. In both case studies, screening curves and optimization expansion models are used to study future expansion plans and how CAPS might fit into them. CAPS data from Tables V-6, V-13, and V-17 has been used to generate operating cost data that is consistent with the escalation and interest rates used in each case study. In particular, a six hour weeknight pumping limit was assumed.

NEPOOL Case Study

The New England Power Pool (NEPOOL) coordinates the bulk electric power generation and transmission of 27 member utilities representing over 98 percent of the power requirements of New England. Northeast Utilities, serving one million customers in Connecticut and Western Massachusetts, and New England Electric System, serving another million customers in Massachusetts and Rhode Island with small service areas in Vermont and New Hampshire, are both members of NEPOOL and participated in the present CAPS evaluations. These two utilities suggested that NEPOOL would be the proper level to plan large storage plants, such as CAPS, and supported the request for data from NEPOOL. The New England Power Exchange (NEPEX) is the operational arm of NEPOOL, dispatching power to New England on a daily basis, and New England Power Planning (NEPLAN) is the planning arm of NEPOOL. NEPLAN provided expert advice regarding the use of the computer models described above and provided data from NEPOOL for the screening curves and optimum expansion model.

The NEPOOL planning data (from Ref. VI-3) is summarized in Tables VI-1 and VI-2. The peak load demand for NEPOOL is projected to grow from 19,433 MW in 1981 to 66,851 in 2001, an average growth rate of 6.4 percent, while the annual load factor stays essentially constant at 66 to 67 percent. Very few utilities have load factors this high (see Fig. VI-2). A typical annual load duration curve, for 1990, is shown in Fig. VI-11. The data given in Tables VI-1 and VI-2 has been used to develop screening curves for the NEPOOL system. Figure VI-12 illustrates the projected busbar power costs for new plants in 1980 dollars. Since NEPLAN assumes a uniform 6 percent escalation of costs, the relative position of these curves will be the same for each year. The curves show that gas turbines and conventional pumped storage plants produce the lowest cost power for operation up to about 2600 hours per year. Oil steam plants briefly take the lowest cost electricity position, with nuclear power being very competitive and dominating above about 3100 hours per year. Coal steam is not competitive, but combined (gas and steam) cycle power plants are close for intermediate (2000-3500 hours per year) duty. CAPS plants are marginally competitive against gas turbines for 1000 to 3000 hours per year of use, but are not competitive with low capital cost ($300/kW) conventional pumped storage plants.

The screening curves in Fig. VI-12 do not take into account either planned maintenance or forced outages (including forced part loads). One way to incorporate these factors into screening curves is to add the percent of the year needed for maintenance to the EFOR to yield a yearly availability percent. Sufficient capacity must then be bought so that the available capacity would be sufficient to meet the peak demand. Since different plants have different availabilities, the curves in Fig. VI-12 would be shifted relative to each other when availabilities are taken into account. Figure VI-13 shows the modified (for availability) busbar power costs versus hours per year for new plants using the NEPOOL data. There is a small shift in the relative position of the curves but the general trends and conclusions are the same as for Fig. VI-12. The modified busbar power curves are the type used by the DYNOGEM optimum expansion program. If the busbar power costs alone, assuming 100 percent availability, are used in this model unrealistic results would be obtained.

Insight into the competition between new and existing power plants can be gained by examining Fig. VI-14 which shows the operating costs (capital charges excluded) for existing plants. A comparison of Figs. VI-13 and VI-14 shows that the owning and operating costs of new nuclear plants is competitive with the operating costs alone for some existing plants. This explains why the recent trend in New England has been to install a large nuclear capacity and push existing fossil plants into reserve, resulting in a high reserve capacity.

Because the relative costs of electricity from CAPS plants are higher (in some cases, only slightly higher) as shown in the screening curves, extensive runs of the optimization model were not made for the NEPOOL data. A sample run, using the costs contained in Figs. VI-13 and VI-14, is shown in Fig. VI-15. Only nuclear plants and gas turbines would be added through 1995, at which time small amounts of conventional pumped storage would be added. Off-peak pumping energy for the storage plants would be provided by nuclear plants.

PEPCO Case Study

The Potomac Electric Power Company (PEPCO) serves customers in the Potomac River basin including Washington, D. C. and parts of Maryland and Virginia. PEPCO is a summer peaking utility reflecting the heavy air conditioning load of its predominantly residential, commercial and governmental customers. PEPCO is a member of the Pennsylvania, New Jersey, and Maryland Interconnection (PJM).

The PEPCO planning data (from Ref. VI-4) is summarized in Tables VI-3 and VI-4. In view of the uncertainty in future oil supply, oil-fired steam and combined cycle plants are not included in the list of future equipment options. The peak PEPCO load demand was assumed to grow from 3773 MW in 1976 to 5456 MW in 1985, corresponding to an average growth rate of 4.2 percent. After 1985, the growth rate was taken to be 3.5 percent. For the purposes of this study, the annual load duration curve was assumed constant in shape. Figure VI-16 shows the PEPCO annual load duration curve which yields an annual load factor of 47 percent. Very few utilities have load factors this low (see Fig. VI-2).

Screening curves for the busbar power costs of new equipment are shown in Figs. VI-17 and VI-18; Fig. VI-17 using new coal plant pumping costs and Fig. VI-18 using new nuclear plant pumping costs. For both figures a constant availability of 85 percent was assumed for all plants. A lower overall availability might have been more appropriate, and this might have tended to favor storage plants. With nuclear pumping costs the screening curve indicates that from about 600 to 3125 hours per year storage plants produce the least cost power. Underground pumped storage hydro plants produce lower cost power than CAPS, due to their lower estimated capital costs, over most of the intermediate load range; however, CAPS would be more economical than gas turbines. The pumping costs from coal used in Fig. VI-17 are much higher than for nuclear pumping, but even with these higher costs the storage plants appear capable of producing lower cost power than competing plants in the intermediate range from about 820 to 2200 hours.

The DYNOGEM optimum expansion program was run from 1981 through 2005 to estimate the expansion pattern which would produce lowest cost power while taking into account the existing plants and plants already committed to the PEPCO system. The results, presented in Fig. VI-19, show that after the addition of committed oil, coal, and nuclear plants no new capacity would be needed until 1995. From that date, gas turbines would be added for peaking duty and underground pumped storage would be added for intermediate duty. The larger capacity of existing and committed fossil steam units would be used to meet the remaining intermediate-to-base load range. The storage plants would be pumped by coal-fired units with all of the nuclear capacity used to meet the base-load demand. Additional base-load nuclear capacity would be added after the year 2000.

Discussion of Results

The results given in the two preceding case studies show that the long term trend is toward gas turbines for peaking, energy storage for cycling, and nuclear for base load duty. Gas turbines are desirable for peaking duty, despite their low efficiency and high fuel cost, because of their relatively low capital cost. Nuclear plants are desirable for base load duty, despite their very high capital cost, because of their low fuel cost. Moderately priced coal could also be competitive for some base load duty. Energy storage plants are desirable for cycling or intermediate service because they take advantage of low cost off-peak energy from nuclear and coal plants. In so doing, storage plants displace some gas turbines resulting in less oil consumption. Depending on the circumstances, both CAPS and pumped storage hydro (conventional aboveground and underground) can be attractive for intermediate service. The following discussion deals with the relative competitiveness of these alternatives with respect to siting potential, economics, fuel availability, and utility load patterns.

Siting Considerations

The siting potential for conventional pumped storage hydro is usually limited by surface topography whereas CAPS (and underground pumped storage) has no such limitation. For example, the hilly New England countryside provides many locations for conventional pumped storage hydro facilities, and this is a major factor in their projected low capital cost ($300/kW). However, the hills in New England are located mainly in the western part of the six states (e.g., 1600 MW of pumped storage hydro plants are currently located in the Berkshires in the western half of Massachusetts), whereas many of the load centers (notably Boston and Providence) are located on the eastern seaboard. The separation of suitable sites and load centers is even more pronounced in other parts of the country. In contrast, it appears that CAPS could be located near any load center in the north central and northeast study regions because of the distribution of favorable hard rock geology. CAPS plants do require sizeable plots of land (approximately 0.01 acres/MWhr of storage capacity), but since the volume of the surface reservoir is several times less than the comparable reservoir volume for conventional or underground pumped storage hydro the land requirements are considerably less.

The costs of transmission lines from conventional pumped storage sites to the load center, such as from the Berkshires to Boston, could well equal the difference in capital cost between the conventional pumped storage and CAPS plants. Typical transmission line costs in the northeast range from $150,000/mile (wood poles) to $500,000/ mile (steel poles), and the right of way could increase these amounts substantially. Such costs were not included in this study since they are highly site specific.

In addition, the lower environmental impact due to the savings on transmission lines, right of way, siting away from scenic hilltops, and use of less land because of smaller surface reservoir provide additional, although somewhat intangible, advantages for CAPS. There has been organized opposition to conventional pumped storage hydro facilities at Canaan Mountain in western Connecticut, Indian Point in nearby New York state, and other potential sites. This opposition has been ameliorated somewhat by the opening of recreational facilities at some sites, such as the hiking and cross country skiing trails at the Northfield Mountain pumped storage facility in western Massachusetts. The less stringent site restrictions and reduced environmental impact of CAPS over conventional pumped storage hydro were not reflected in the costs used in the previous case studies.

Economic Comparisons

The bases for the previous economic comparisons were selected to be representative of realistic, although not necessarily typical, electric utility operating conditions. Considerable effort was placed on the estimated cost and performance characteristics of CAPS, but only limited attention was given to how they would be operated, the corresponding characteristics of alternative plants, and site related costs. As a result, some cost inequities are present in the two case studies. One example is the cost difference associated with the siting considerations discussed above.

Another example concerns the comparative costs for underground pumped storage hydro plants. The costs for the underground pumped storage facility used in the PEPCO case study were for a four-unit, 2000 MW plant whereas the CAPS costs were for a two-unit, 500 MW plant. The economies of scale for both types of plant are significant (see Table V-13 for CAPS trend). A 2000 MW CAPS plant could encompass considerable savings through lower cost bulk excavation and shared facilities, such as the use of the main access shaft for multiple units. Furthermore, it is not clear that the cost estimates for CAPS and underground pumped storage were prepared on a consistent basis. The uncertainties in costs for both types of plants, because of their undeveloped nature, are substantially greater than for developed plants. Consequently, the propriety of comparing these two plants and implying superiority of one over the other at this time is highly questionable.

A significant advantage of CAPS which was not factored into the costs used in the case studies is the option of running CAPS in a simple-cycle mode. The gas turbine could be run during the underground excavation period, thus permitting some power to be generated, earning partial return on investment, and reducing the escalation and interest charges during the construction period. The gas turbine could also be operated in this mode whenever the cavern is shut down for maintenance or depleted of air. The value of this flexibility and emergency capability could be quite high for some utilities.

In both case studies, a single number was used for the heat rate of a given plant type regardless of the hours per year of use and corresponding duty cycle. This heat rate should be varied since additional fuel consumed during startup, shutdown, and part-load operation could be substantial. The net effect would be to cause the screening curves to be curved rather than straight, with the result that base-load coal and nuclear plants would be somewhat less competitive in the intermediate load range (2000 to 5000 hr/yr) relative to storage systems. Recall from Part V that CAPS plants have exceptional part-load performance.

The assumption that only six hours of weeknight pumping would be available for storage plants is an overly conservative one which almost precludes CAPS from any major market. If this assumption were relaxed to permit more hours of weeknight pumping, then storage plants would be able to generate power for more hours per year without any further investment. As a result, they would become more competitive with gas turbines and other cycling plants. A comparison of the maximum annual generation times for 6 hr and 10 hr weeknight pumping periods is given in Table VI-5 in terms of the CAPS nominal cavern storage capacity. This change can be represented on the screening curves by extending the curves out to more hours per year corresponding to the estimated additional generation time.

Such an extension of generation time was made in Figs. VI-20 and VI-21. These figures represent generic screening curves which should apply to a broad spectrum of electric utilities. They do not correspond specifically to either of the two previous case studies, although data from both cases (summarized in Table VI-6) were used to prepare them. Conventional pumped storage was omitted from the comparison because suitable sites are unavailable in many parts of the country. Also, underground pumped storage was omitted in order to limit CAPS comparisons to established commercial alternatives. Other assumptions implicit in Figs. VI-20 and VI-21 are that the relative differences in future fuel prices will be unchanged and all fuel types will be available when needed.

Figure VI-20 depicts a hypothetical comparison indicative of the near term competitive economics with off-peak compression energy derived from coal-fired steam plants. From this comparison it is seen that lowest cost power would be produced by gas turbines from 0 to about 3000 hr/yr, by coal plants from 3000 to about 5000 hr/yr, and by nuclear plants above 5000 hr/yr. The costs for CAPS plants with 10 and 20 hour storage capacities approach the competitive range between 2000 and

3000 hr/yr, but the CAPS competitive position is only marginal unless the use of gas turbines should be limited because of their high fuel consumption.

Figure VI-21 is based on the use of nuclear energy for off-peak compression and represents a hypothetical future comparison that could be made after substantial amounts of nuclear capacity are added for base-load duty. Since the trend toward a nuclear expansion is firmly established and almost all generation expansion analyses predict a strong continuation of this trend, the type of comparison in Fig. VI-21 with low pumping costs seems quite appropriate. This figure shows CAPS to be very competitive over a broad range from about 1500 to 4000 hr/yr.

If the generic screening curves in Fig. VI-21 are superimposed on representative load duration curves (such as in Figs. VI-1 and VI-6), it is possible to estimate the percent capacity that would be required for each type of power plant at some very distant time in the future when all existing capacity will have been retired. This type of hypothetical analysis assumes that all relative costs remain the same and technology is stagnant, but it illustrates the "trend" in future additions that will tend to minimize costs. By performing such an analysis it can be shown that about 10 percent of the system generating capacity should be CAPS, almost independently of system load factor.

Based solely on economic considerations, the preceeding comparisons show that a necessary condition for the CAPS plant designed herein to be competitive is to have excess nuclear capacity for pumping. The analyses are necessarily preliminary, and additional study is desirable to better define the interrelationships between storage and pumping plants. Seasonal and weekly load-duration curves would provide insights not possible by using annual load-duration curves. Additional plant data on part-load heat rates and the use of production cost models would be useful in evaluating trade-offs during daily load variations. Ideally, a detailed modeling effort would involve iterations using both production cost and optimization programs to determine annual heat rates (including start up, part load, and spinning reserve) and other operating data.

Fuel Considerations

The recent energy crisis left deep scars and bitter memories in the minds of many utility planning and operating personnel charged with the responsibility of satisfying the simultaneous demands of customers, investors, and environmentalists. The most ideal fuels, clean petroleum fractions and natural gas, are in short domestic supply. The Nation has generally turned to oil imports to meet growing demands. Oil imports to the US (see Ref. VI-5) accounted for about 24 percent (3.4 million barrels per day) of oil consumption in 1970. By 1973 imports increased to about 36 percent (5.9 million barrels per day). Increased imports projected for 1985 are alarming (about 47 percent or 10.3 million barrels per day).

Although most of this oil is used in the transportation, commercial, and residential sectors, oil consumption by utilities in gas turbines and cycling steam plants is substantial. Utilities are in an almost unique position (relative to the rest of the economy) where they can dramatically reduce oil consumption by shifting to coal and nuclear plants, supplemented by storage systems. The shift away from gas turbines might not be the most economical choice, as evidenced by the previous economic analyses, but at least the shift is feasible. Some utilities have indicated a willingness to move in that direction (as suggested by the good response to the ERDA/EPRI energy storage demonstration program). Of course, the spector of oil allocations and legal restrictions on oil usage for power generation is noteworthy. In any case, the oil conservation ethic is very strong among utilities.

Since the oil consumption by CAPS, as reflected by its heat rate, would be substantially below that for gas turbines (4377 vs 12220 Btu/kWhr), CAPS could make a valuable contribution toward energy independence either by reducing total consumption or by more efficiently utilizing existing supplies. For example, if CAPS were to replace gas turbines in a hypothetical system having 1000 MW of gas turbine capacity operating with an annual load factor of 5 percent, the potential savings in oil would be over 500,000 barrels per year. The replacement nighttime pumping energy would, of course, be derived from relatively low cost and abundant indigenous coal and nuclear resources.

FUTURE PROGNOSIS

CAPS provides a method to utilize more fully the energy from high capital cost nuclear power plants and thereby generate electricity for intermediate loads at costs competitive with alternative fossil-fueled plants. Also, the abundance of sites, relative to conventional pumped storage, allows the choice of locations to be near major load centers. The transmission and environmental impact costs associated with remote siting would tend to make CAPS more competitive with conventional pumped storage. Although the cost comparisons are dependent on utility specific factors, the general competitive situation looks promising. The situation should improve in the future with the trend toward nuclear generation, a strong conservation ethic, and the prospects for future improvements in CAPS.

Prospective Improvements

The general design philosophy adopted for this study was to make maximum use of commercial or nearly developed components and technology in order to avoid extensive engineering or development expenditures. The resulting near term, or first generation, design involved several compromises and assumptions which lead to higher costs. It is apparent, however, that CAPS possesses unique opportunities for future improvements through design advancements, fuel substitution, and storage options. With the prospect of these improvements the growth potential for CAPS seems very bright.

Since the low-pressure portion of the system already incorporates the most advanced technology currently available, most of the potential design improvements relate to the high-pressure components, especially the expansion turbine. Future designs should consider use of large single-unit expansion turbines instead of multiple, parallel flow units. In addition to the economies of scale for the turbine, associated savings in foundations, buildings, piping, and controls could be substantial. Putting the expansion turbine on the main shaft might also result in lower costs, although at the expense of operating flexibility. Still farther into the future, the possibility exists of incorporating advanced materials, cooling, and aerodynamics into the expansion turbine to raise its operating temperature, increase power output, and reduce specific cost.

Currently, gas turbines are limited to using relatively clean petroleum-derived fuels. Considerable effort, however, is being expended to develop advanced concepts which will permit gas turbines to operate on coal or coal-derived clean fuels. Most of this work is being funded by the Energy Research and Development Administration in an attempt to develop advanced power systems for base-load power generation. Some of these advanced concepts could be considered for CAPS applications. One such concept is continuous coal gasification to produce a clean, low-Btu fuel gas which would be continuously burned in a gas turbine. In a CAPS application, the continuously operating gas turbines would drive the compressors during off-peak periods and excess air not needed for the turbines would be put into storage (see Ref. VI-6). During the peak load period, the turbines would drive the electric generator with air for the turbines and gasifier taken from storage. Another promising concept involves the use of coal fired, pressurized fluid bed air heaters in place of the CAPS oil burners (Ref. VI-7). Yet another (Ref. VI-8) would replace the oil burners by steam heated air heaters where the steam would be raised in a nuclear reactor. Thermal storage could also be considered to reduce or eliminate oil consumption. All of those concepts would involve long term development, but the important point is that CAPS could be adapted to almost any prospective future energy source.

Another attractive feature of CAPS is the flexibility in storage options. This study focused on storage in mined hard work caverns with the result that CAPS could be located within 50 miles of any major load center in the two study regions. Much of the remainder of the US is underlain by similar geological formations which should also contain numerous siting opportunities. Other storage methods which also look promising are aquifers and solution-mined salt caverns. Since one or more of these storage alternatives exists almost everywhere in the US, the siting aspect should not in any way limit the future growth potential of CAPS.

Concluding Remarks

Compressed air power systems are technically feasible and potentially attractive for future peaking and intermediate-load power generation. The competitive position is highly dependent on utility-specific factors. It is only marginal in near-term applications where fossil energy (specifically coal) must be used for off-peak compression, but it improves substantially for future applications as nuclear capacity becomes available for compression. Prospective improvements in design, cost reductions, and siting flexibility suggest that CAPS should become a viable alternative for future power generation. To ensure success for this worthwhile concept, an ambitious demonstration and technology support program should be pursued.

REFERENCES

VI-1. Public Service Electric and Gas Company, An Assessment of Energy Storage Systems Suitable for Use by Electric Utilities, EM-264, EPRI Project 225, ERDA E(11-1)-2501, Final Report, Volume II, July 1976.

VI-2. Federal Power Commission, FPC News, Volume 9, Number 29, p. 68, July 16, 1976.

VI-3. Letter with attached data tables from James R. Smith, Director, New England Power Planning, March 16, 1976.

VI-4. Letter with attached data tables from Anthony J. Como, Potomac Electric Power Company, August 13, 1976.

VI-5. U. S. Imports to Reach 10 Million B/D. The Oil and Gas Journal, September 6, 1976.

VI-6. Haydock, J. L.: Some Technical Developments and Their Potential Influence on Fuel Supply and Demand, Fuels Planning Conference, Toronto, October 15-18, 1972.

VI-7. Harboe, H.: Importance of Coal for Heat and Power Generation. Stal Laval Report 354 E 03.73, 1973.

VI-8. Private communication with J. L. Haydock, Acres Consulting Services, 1975.

TABLE VI-1

NEPOOL ESTIMATED OPERATING COSTS, PERFORMANCE AND CAPACITY DATA

Plant Type	Capital Cost (1) $/kW	Fixed Charge Rate %/yr	O & M Cost (2) $/MWyr	Heat Rate (7) Btu(HHV)/kWhr	EFOR	Planned Maintenance Weeks/year	1980 Installed Capacity (3) MW
Existing and Committed Plants							
Hydroelectric	-	-	ng (4)	-	0.0	0.00	1512
Conventional Pumped Storage Hydro	-	-	ng	-	2.0	0.78	-
Nuclear	-	-	10403	10631	12.0	7.00	4297
Base Load Fossil	-	-	11183	9131	10.3	3.68	2717
Intermediate Fossil #1 (higher efficiency)	-	-	8312	9536	6.5	3.32	5489
Intermediate Fossil #2 (lower efficiency)	-	-	13453	11944	4.4	4.08	4334
Peaking Fossil	-	-	33650	15922	2.0	1.76	716
Gas Turbines & Diesels	-	-	ng	13503	15.4	1.75	1718
New Plants							
Nuclear	710	18.97	8888	10510	12.0	7.00	-
Base Load Coal	570	19.02	36614	9713	17.0	6.00	-
Base Load Oil/Coal	420	19.02	8798	9000	10.4	6.00	-
Intermediate Fossil Coal	780	19.02	41122	10266	14.2	4.00	-
Intermediate Fossil Oil/Coal	540	19.02	11137	9520	7.5	4.00	-
Combined Cycle	375	19.35	14250	8550	11.3	2.00	-
Peak Oil	560	19.02	14990	10615	5.9	4.00	-
Gas Turbine	170	19.66	3325	12220	10.0	2.00	-
Conventional Pumped Storage Hydro	300	19.67	1350	- (5)	2.0	1.75	-
CAPS Plants - two units (by UTRC)							
CAPS-6 hour storage	316	19.66	3790	4377 (6)	10.0	2.00	-
CAPS-20 hour storage	374	19.66	3790	4377 (6)	10.0	2.00	-
CAPS-40 hour storage	468	19.66	3790	4377 (6)	10.0	2.00	-

(1) All costs are in 1980 dollars. NEPOOL assumes 6%/year escalation of all costs. Interest during construction included.
(2) O&M is operating and maintenance. The O&M are for hours/year of generating time typical of the plant type (peak, etc.).
(3) The following are committed additions to the NEPOOL system:
Nuclear 1981 + 1150 MW, 1982 + 2330 MW, 1983 + 1150 MW, 1984 + 1150 MW, 1986 + 2300 MW
Intermediate Fossil #1, 1981 + 270 MW
Gas Turbines 1982 + 120 MW
(4) ng means none given
(5) New and Existing Pumped Storage Hydroelectric plants are assumed to have an electricity out/electricity in efficiency of 75%.
(6) CAPS plants are assumed to have an electricity out/electricity in efficiency of 123.4%.
(7) Heat rates are estimated annual Btu in/kWhr out.

TABLE VI-2

NEPOOL ESTIMATED FUEL COSTS

Fuel	Cost,[1] $\$/10^6$ Btu
Coal (high sulfur)	1.80
No. 6 Oil (low sulfur)	2.50
No. 2 Oil (low sulfur)	3.40
Nuclear	.45[2]

---

(1) All costs are in 1980 dollars. NEPOOL assumes 6%/yr escalation of all costs.
(2) The nuclear fuel cost assumption is under review.

TABLE VI-3

PEPCO ESTIMATED OPERATING COSTS, PERFORMANCE AND CAPACITY DATA

Plant Type	Capital Cost[1] $/kW	Fixed Charge Rate %/yr	O & M Cost[2] Fixed, $/MWyr	O & M Cost[2] Variable, $/MWhr	Heat Rate[6] Btu (HHV)/kWhr	1980 Installed Capacity[3], MW
Existing Plants						
Coal Steam-Base (> 10,000 Heat Rate)	-	-	5138	2.61	10700	660
Coal Steam-Base (< 10,000 Heat Rate)	-	-	4257	0.99	9600	2130
Coal Steam-Cycling	-	-	8661	1.69	12500	152
Oil Steam-Base	-	-	7120	2.12	15200	82
Oil Steam-Cycling	-	-	6606	2.68	12600	1425
Committed Plants						
Oil Steam-Cycling	300	29	2275	2.82	11700	600
Coal Steam-Base	540	23	3670	1.44	9500	0
Gas Turbines	-	20	147	4.80	16300	561
Nuclear	562	30	2863	0.49	10500	0
New Plants						
Nuclear	562	30	4991	0.56	10500	-
Coal Steam-Base	540	23	3670	1.44	9500	-
Underground Pumped Storage Hydro	270	26	440	0.17	(4)	-
Gas Turbines	192	20	147	4.26	16000	-
CAPS Plants - two units (By UTRC)						
CAPS - 6 Hour Storage	297	24	2823	2.71	4377[5]	-
CAPS - 20 Hour Storage	345	24	2823	1.93	4377[5]	-
CAPS - 40 Hour Storage	415	24	2823	1.36	4377[5]	-

(1) All costs are in 1980 dollars. A 9%/year escalation through 1982, 7%/year through 1985 and 6%/year after 1985 was assumed. Interest during construction not included.

(2) O & M is operating and maintenance.

(3) The following are committed additions to the PEPCO system: Coal Steam - Base (committed) - 1982 + 800 MW; Nuclear - 1985 + 1178 MW, - 1987 + 1178 MW

(4) Pumped storage hydroelectric plants are assumed to have an electricity out/electricity in efficiency of 72%

(5) CAPS plants are assumed to have an electricity out/electricity in efficiency of 123.4%

(6) Heat rates are estimated annual Btu in/kWhr out.

TABLE VI-4

PEPCO ESTIMATED FUEL COSTS

Fuel	Cost, \$/10^6 Btu 1977	1980	1985(1)
Coal	1.20	1.55	2.10
No. 6 oil	1.85	2.80	3.60
No. 2	2.35	3.55	4.60
Nuclear	----	----	0.60

(1) For projecting fuel prices beyond 1985 the following escalation factors were assumed:

Coal - 5 percent per year - 1986 through 2005
#6 Oil - 8 percent per year - 1986 through 2005
#2 Oil - 8 percent per year - 1986 through 2005
Nuclear - 6 percent per year - 1986 through 2005

TABLE VI-5

COMPARISON OF CAPS ANNUAL GENERATION TIME

Nominal Cavern Storage Capacity, hr	Maximum Annual Generation Time[1], hr/yr Weeknight Pumping Period 6 hr	10 hr
6	1560	1560
10	1740	2500
20	2190	3010
40	3125	3940

(1) Based on 5-day week and 50-week year

TABLE VI-6

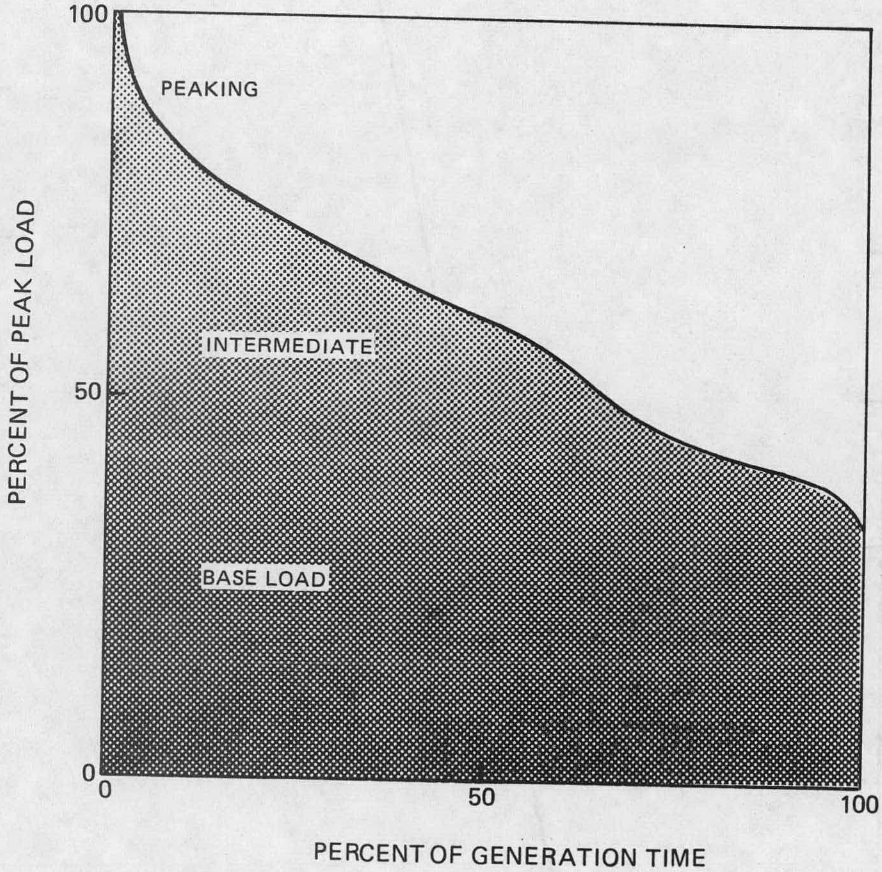
DATA SUMMARY FOR GENERIC SCREENING CURVES

1980 dollars

Plant Type	Capital Cost [(1)] $/kW	Fixed Charge Rate, %	O&M Costs Fixed $/MWyr	O&M Costs Variable $/MWhr	Heat Rate Btu(HHV)/kWhr
Nuclear steam[(2)]	562	30	4,991	0.56	10,500
Coal steam[(3)]	540	23	3,670	1.44	9,500
Gas Turbines[(4)]	192	20	147	4.26	12,220
CAPS-10[(4)]	311[(5)]	24	2,823	2.32	4,377
CAPS-20[(4)]	345[(5)]	24	2,823	1.93	4,377
CAPS-40[(4)]	415[(5)]	24	2,823	1.36	4,377

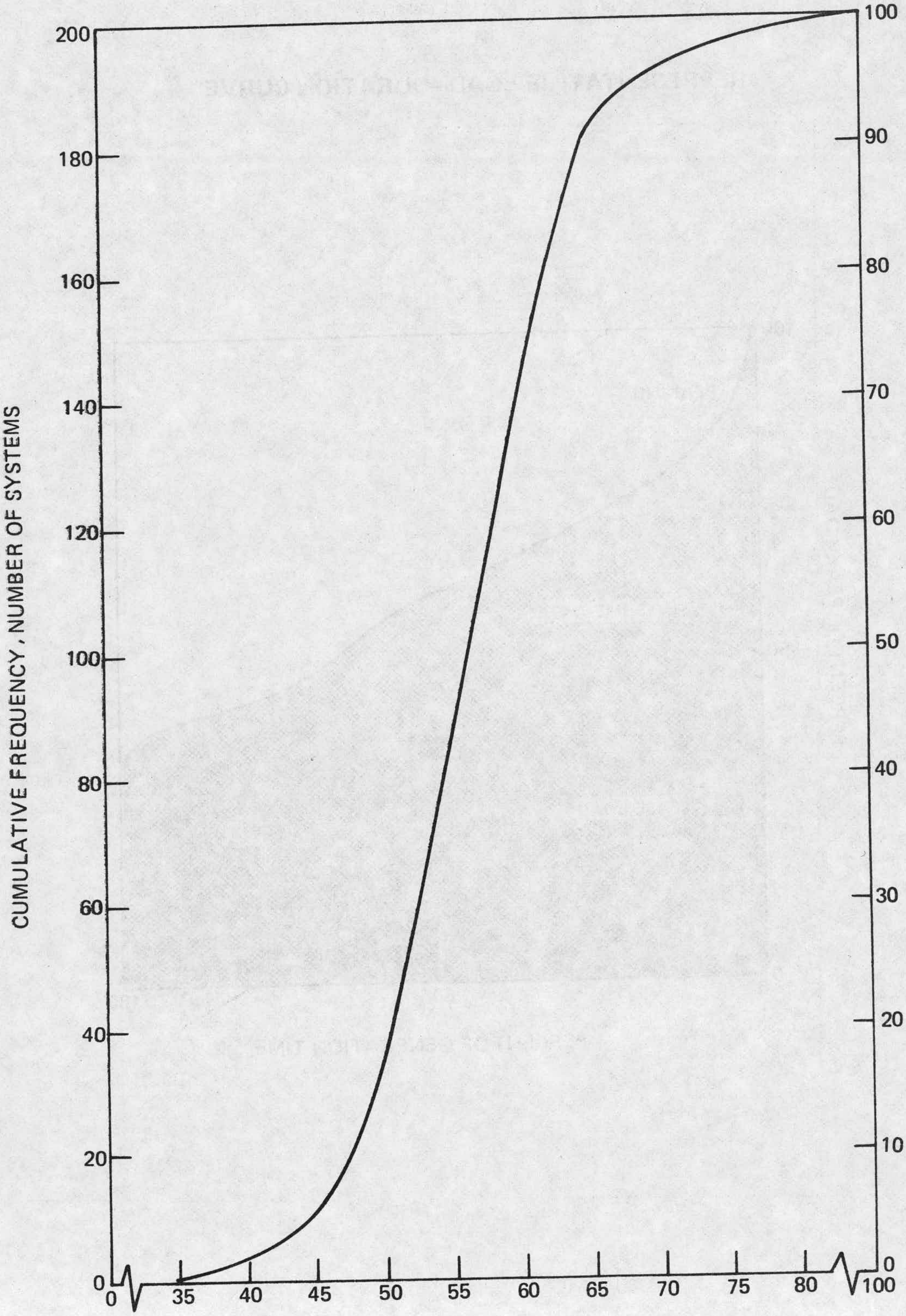
(1) Interest during construction not included.
(2) Fuel @ $0.60/10^6 Btu
(3) Fuel @ $1.55/10^6 Btu
(4) Fuel @ $3.55/10^6 Btu
(5) Two-unit CAPS plants, 505 MW

REPRESENTATIVE LOAD–DURATION CURVE



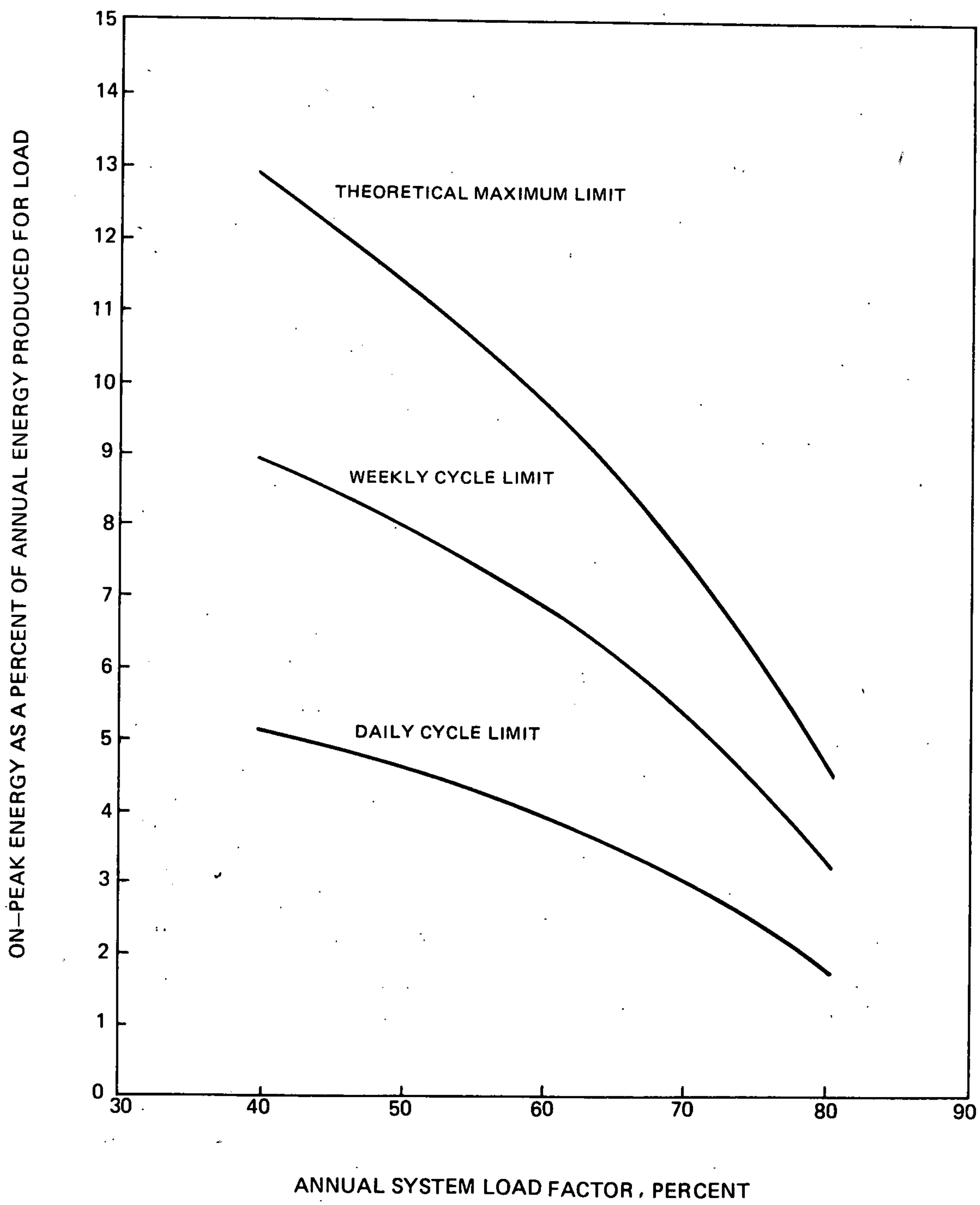
DISTRIBUTION OF ELECTRIC UTILITY 1971 ANNUAL SYSTEM LOAD FACTORS

(FROM REF. VI-1)



PRACTICAL LIMITS OF ON-PEAK ENERGY CAPABLE OF BEING SUPPLIED BY THE OFF-PEAK ENERGY AVAILABLE ON U.S. ELECTRIC UTILITIES

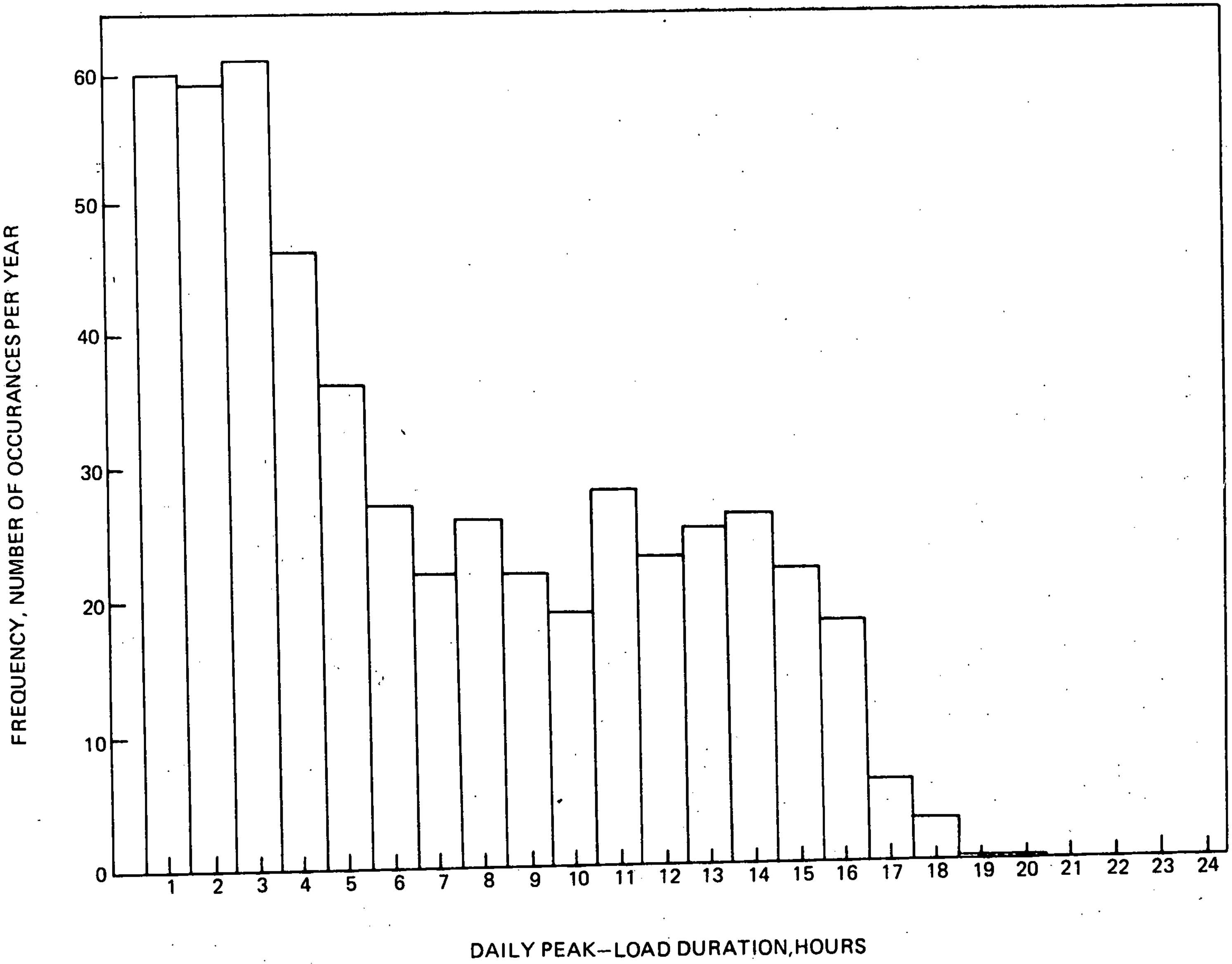
(BASED ON REF. VI-1)



R76–952161–5

PEAKING LOAD DURATION FREQUENCY OF U.S. ELECTRIC UTILITY SYSTEMS

(FROM REF. VI–1)

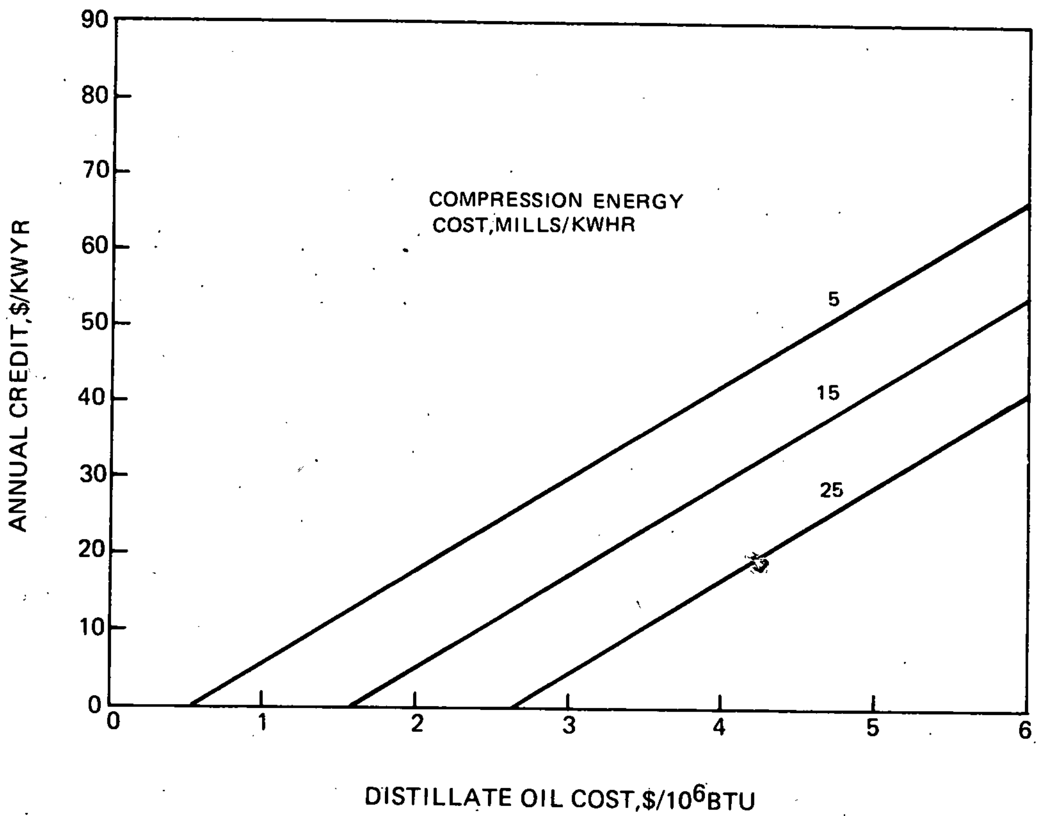


76–12–168–8

FIG. VI–4

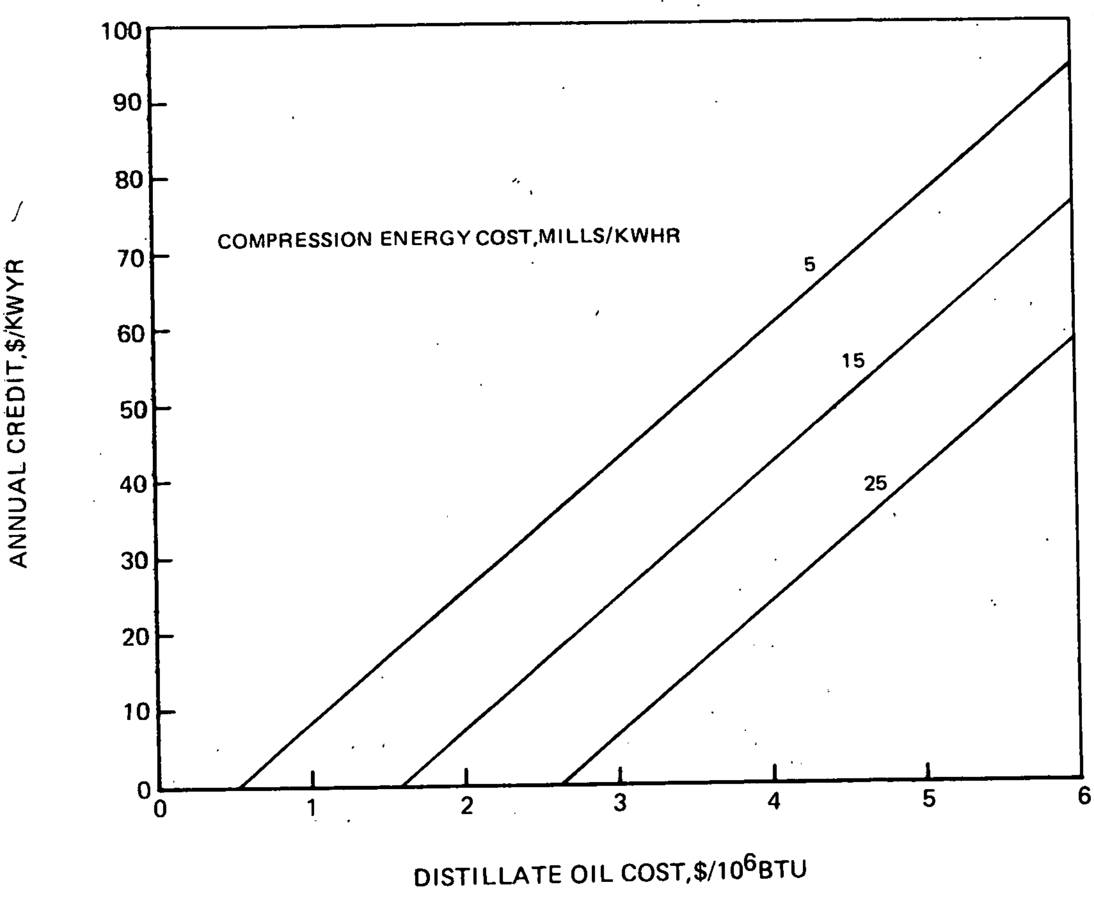
CAPS ENERGY COST CREDIT RELATIVE TO GAS TURBINES

1560 HR/YR



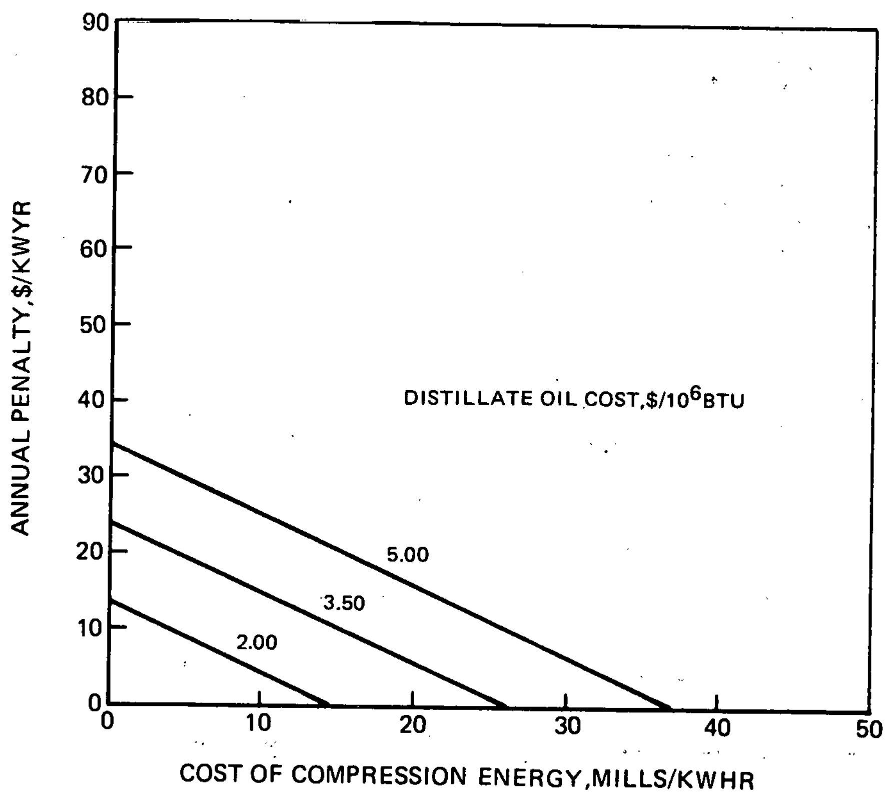
CAPS ENERGY COST CREDIT RELATIVE TO GAS TURBINES

2190 HR/YR



CAPS ENERGY COST PENALTY RELATIVE TO PUMPED HYDRO

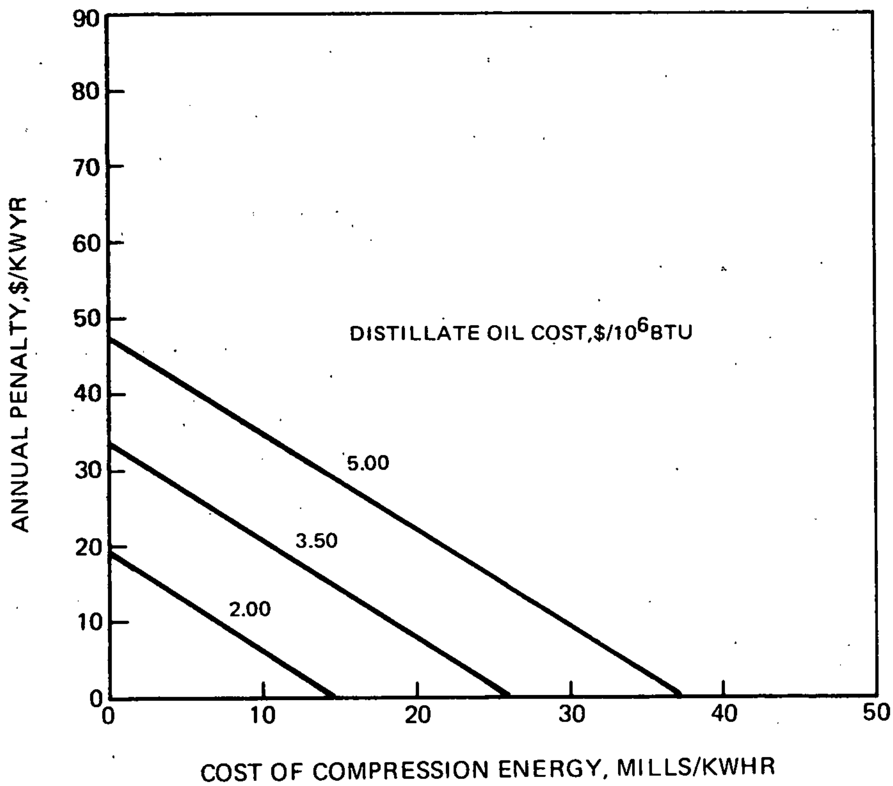
1560 HR/YR



76–12–168–3

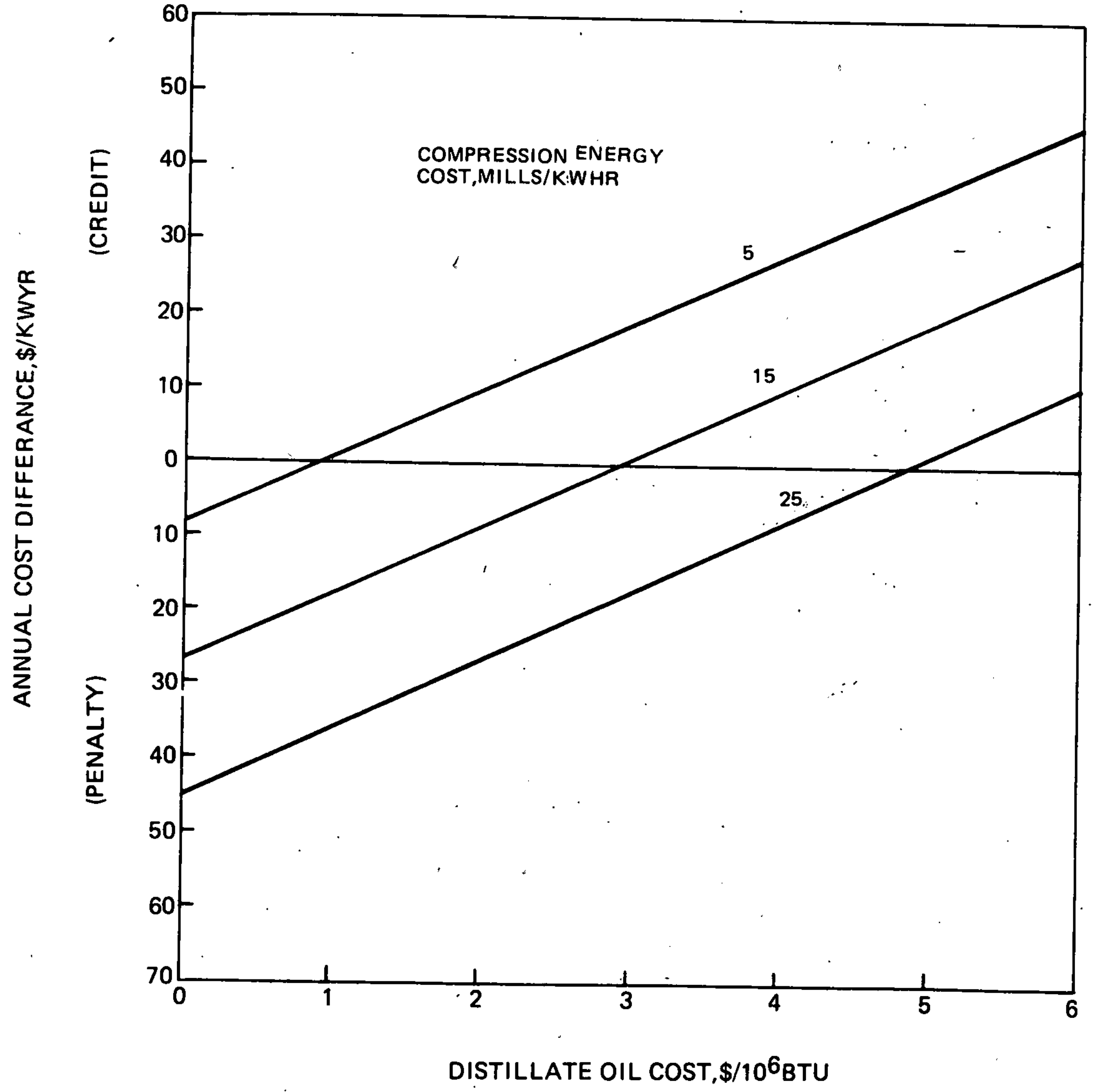
CAPS ENERGY COST PENALTY RELATIVE TO PUMPED HYDRO

2190 HR/YR



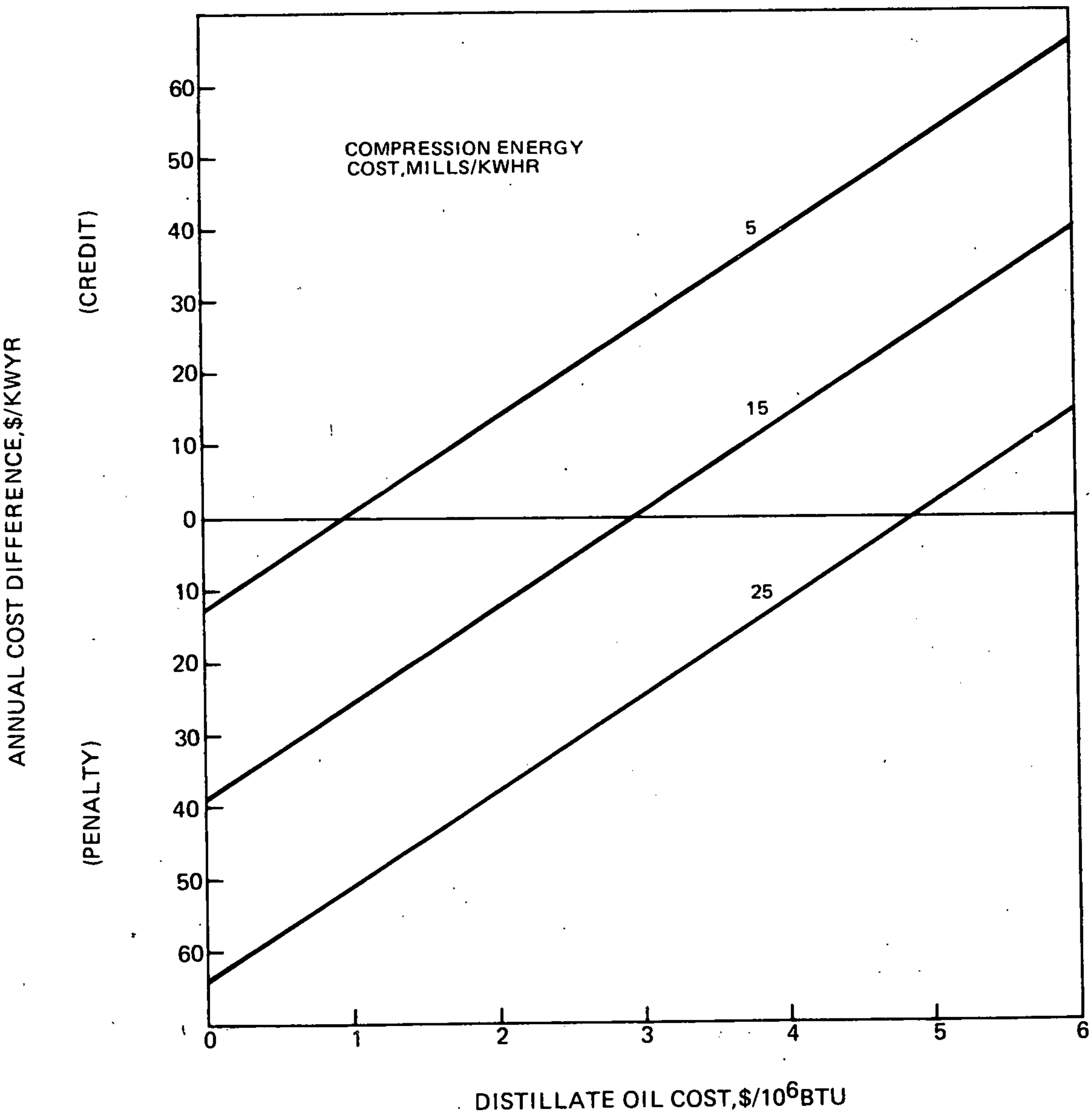
CAPS ENERGY COST RELATIVE TO COMBINED CYCLE

2190 HR/YR



CAPS ENERGY COST RELATIVE TO COMBINED CYCLE

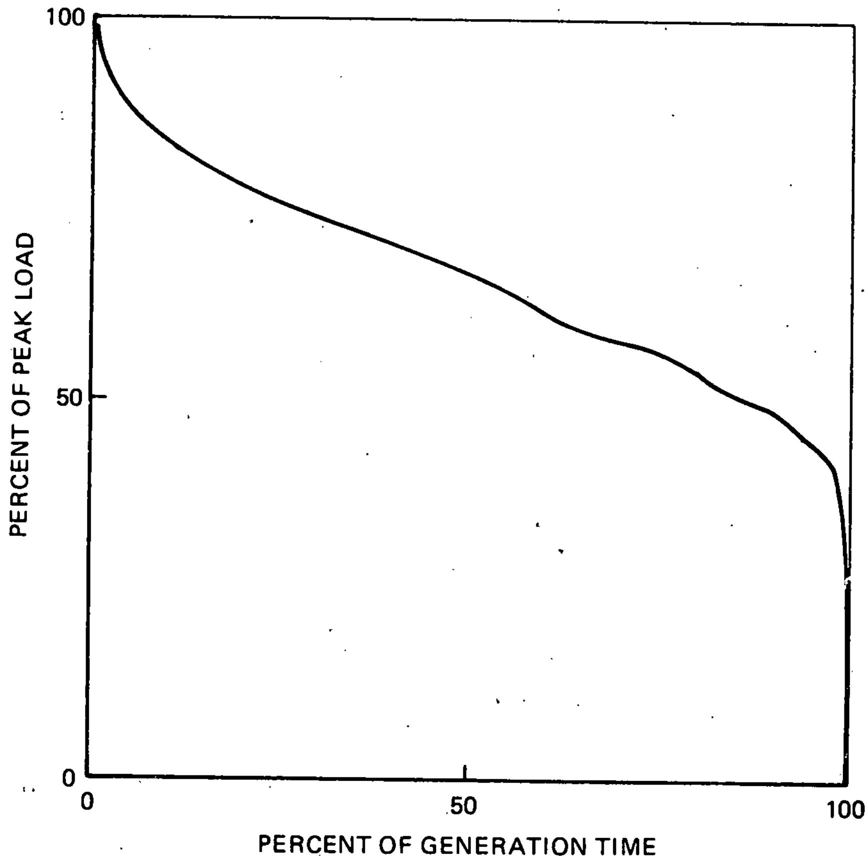
3125 HR/YR



76–12–168–10

NEPOOL ANNUAL LOAD DURATION CURVE–1990

LOAD FACTOR = 67%

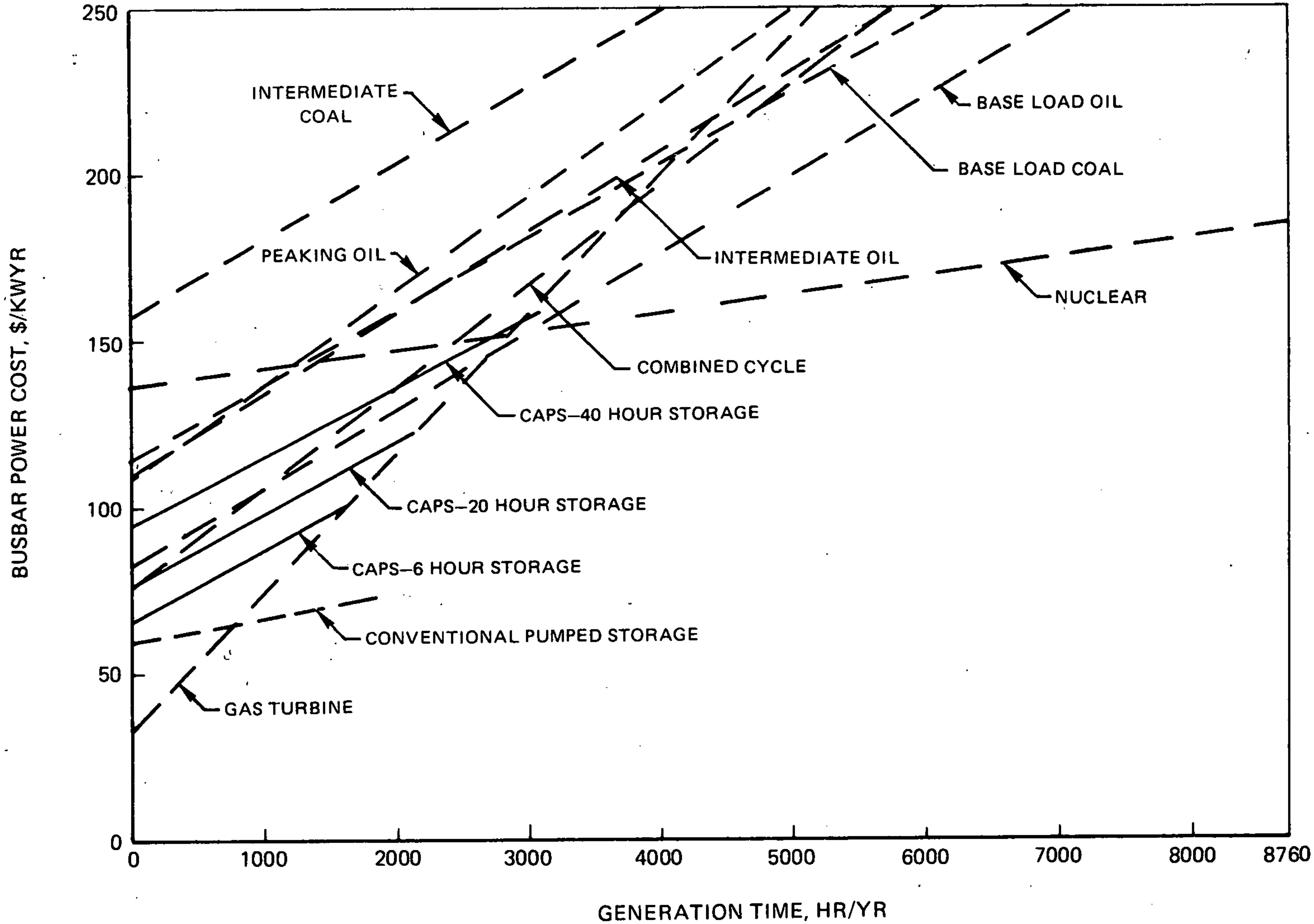


R76-952161-5

NEPOOL SCREENING CURVES

NUCLEAR PUMPING FOR STORAGE

1980 DOLLARS



76–12–66–7

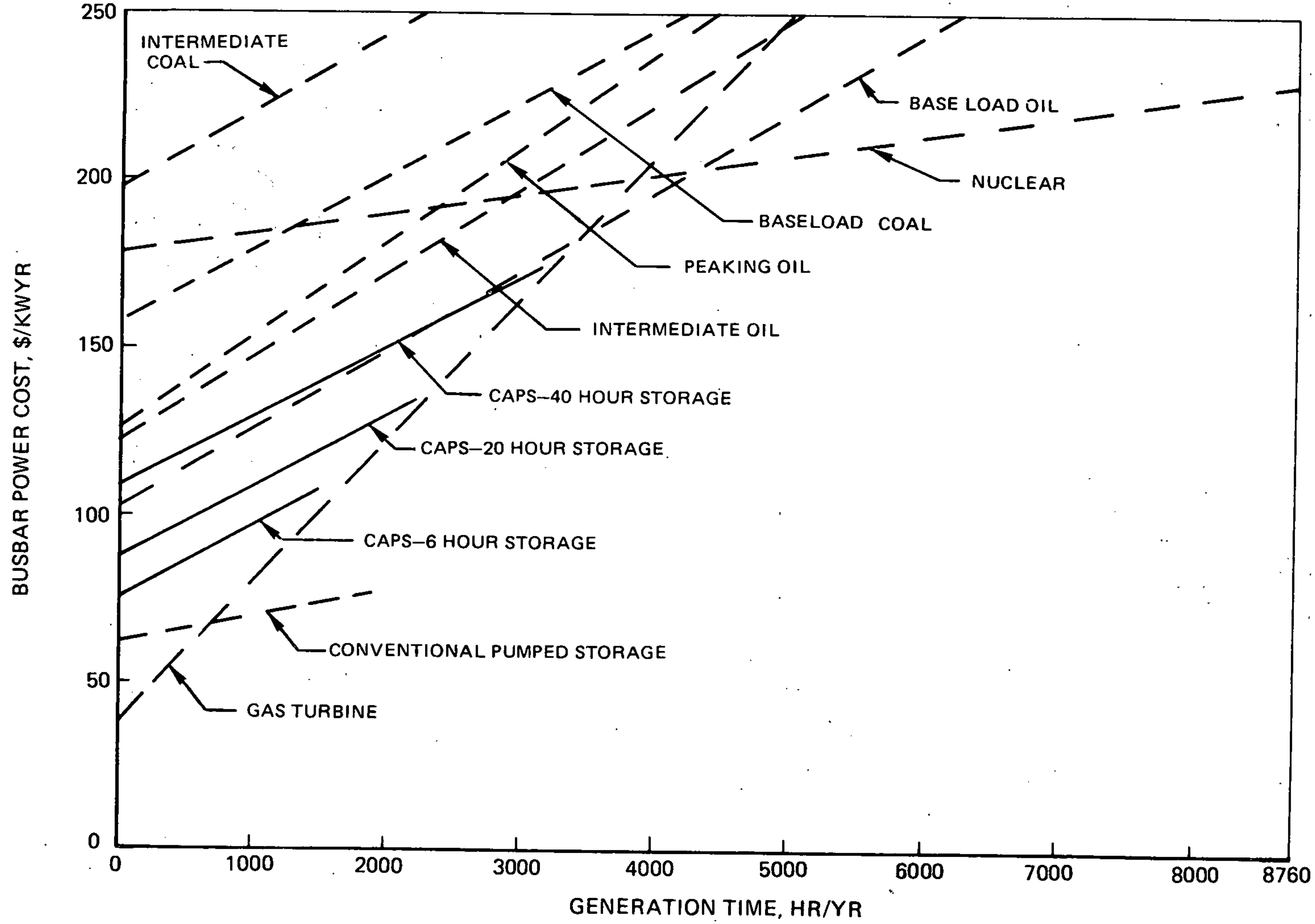
FIG. VII–12

R76-952161-5

NEPOOL SCREENING CURVES – COSTS MODIFIED FOR UNAVAILIBILITY

NUCLEAR PUMPING FOR STORAGE

1980 DOLLARS



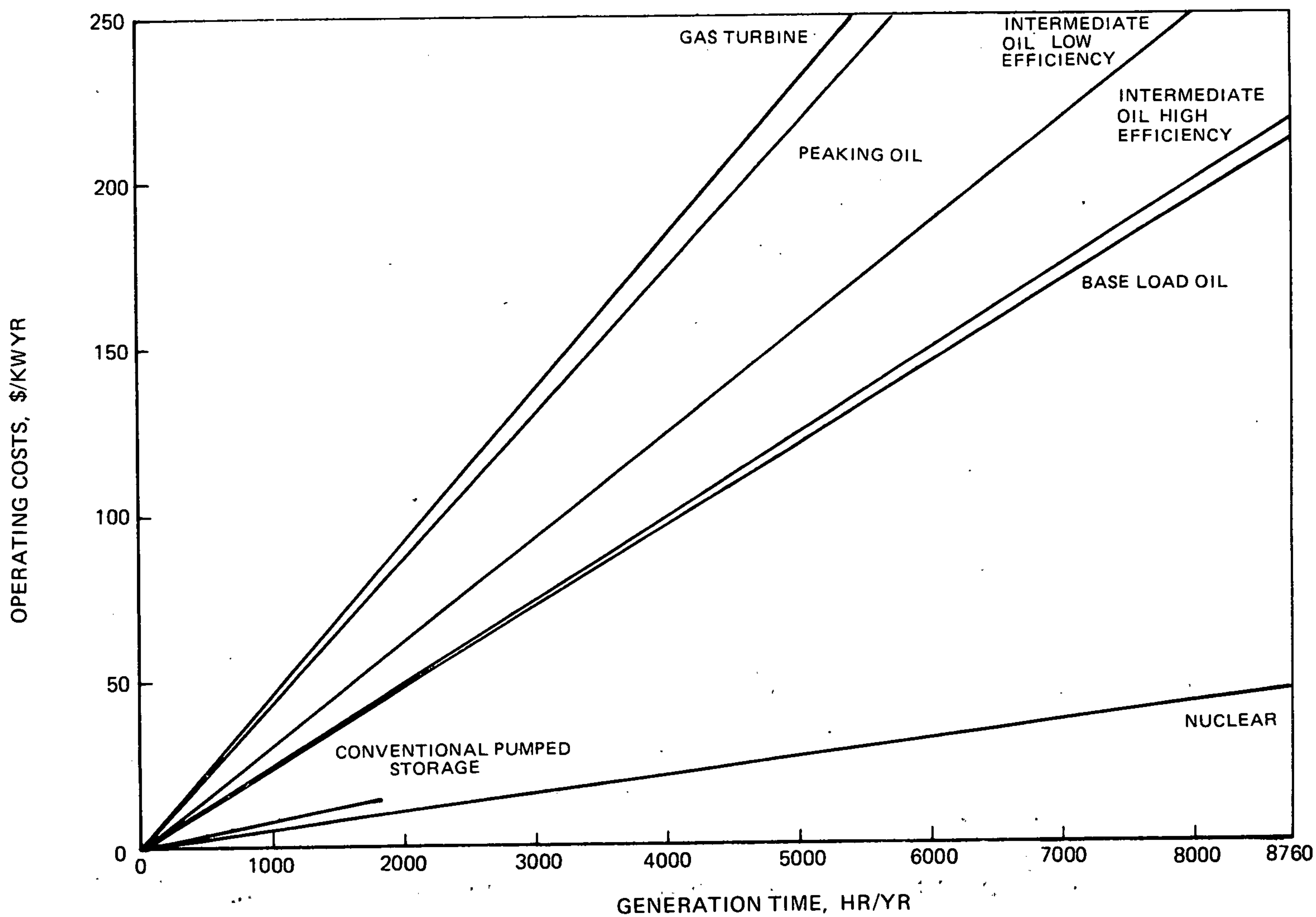
76–12–66–8

FIG. VI–13

NEPOOL OPERATING COSTS FOR EXISTING EQUIPMENT

NUCLEAR PUMPING FOR STORAGE

1980 DOLLARS



76–12–66–9

FIG. VI–14

NEPOOL OPTIMUM GENERATION EXPANSION

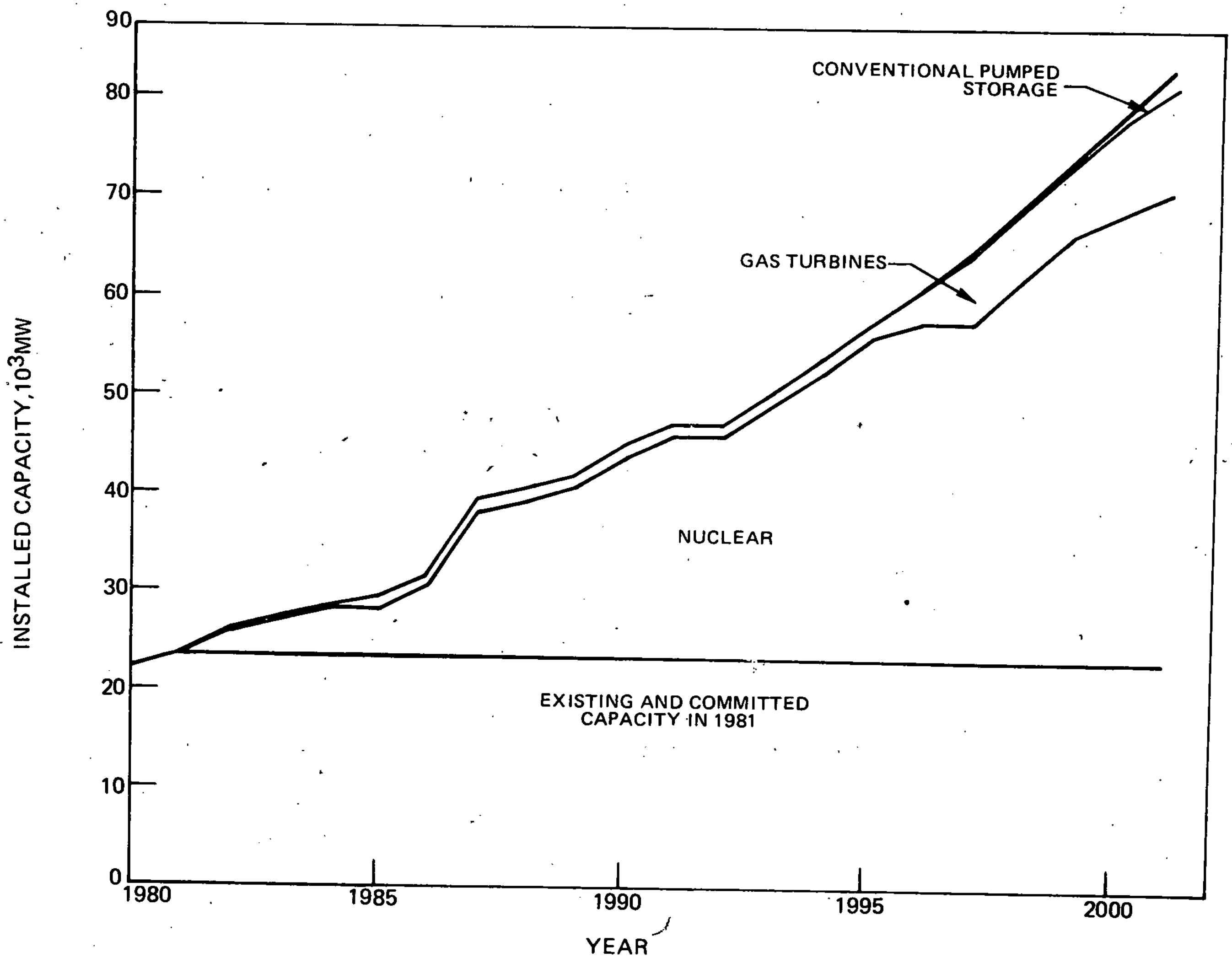
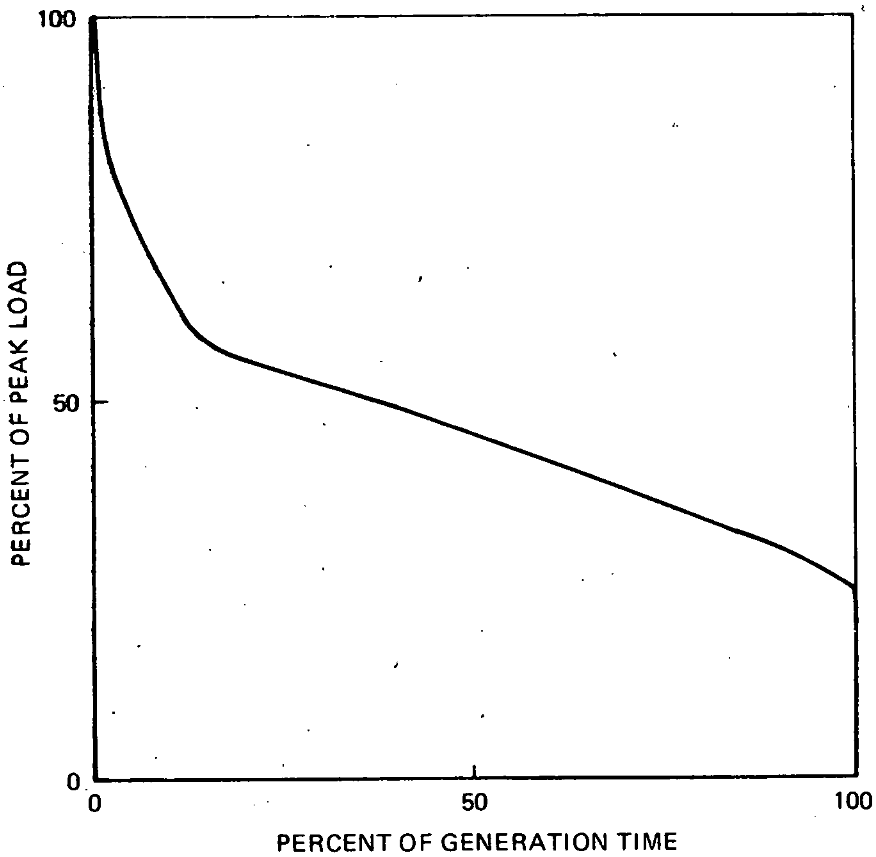


FIG. VI–15

PEPCO ANNUAL LOAD DURATION CURVE

LOAD FACTOR = 47%



PEPCO SCREENING CURVES

COAL PUMPING COSTS
1980 DOLLARS

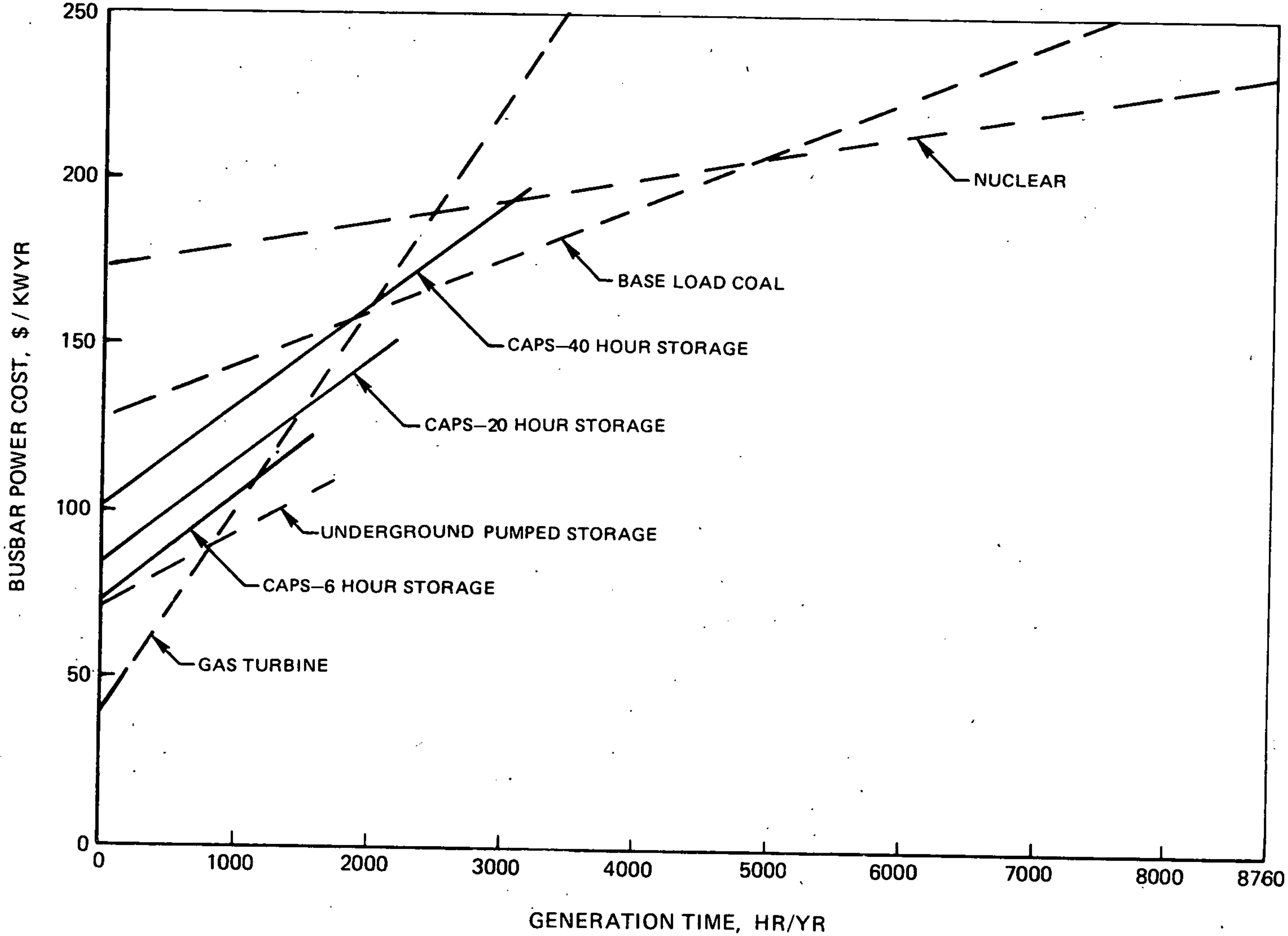


FIG. VI-17

R76–952161–5

PEPCO SCREENING CURVES

NUCLEAR PUMPING COSTS

1980 DOLLARS

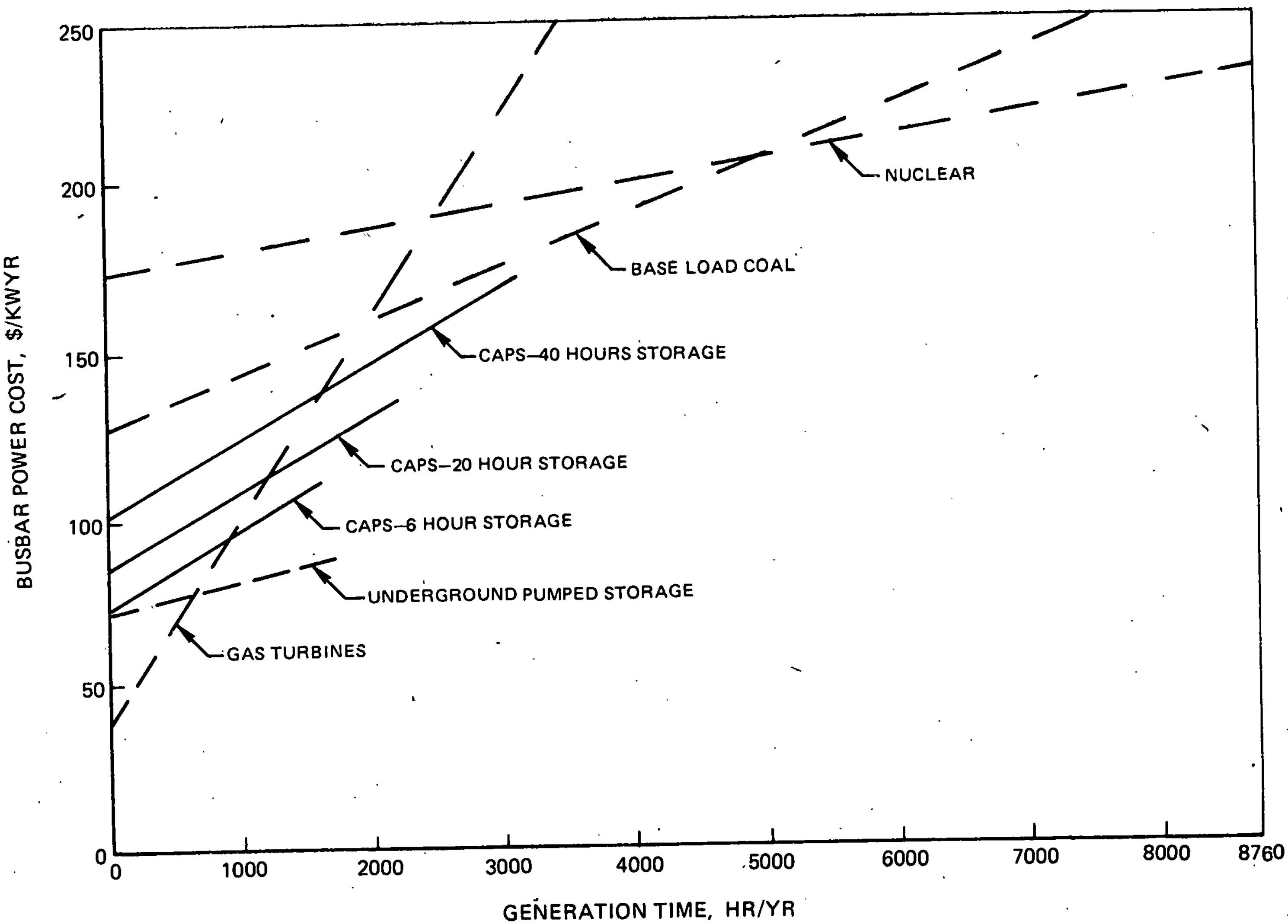


FIG. VI–18

76–12–66–2

PEPCO OPTIMUM GENERATION EXPANSION

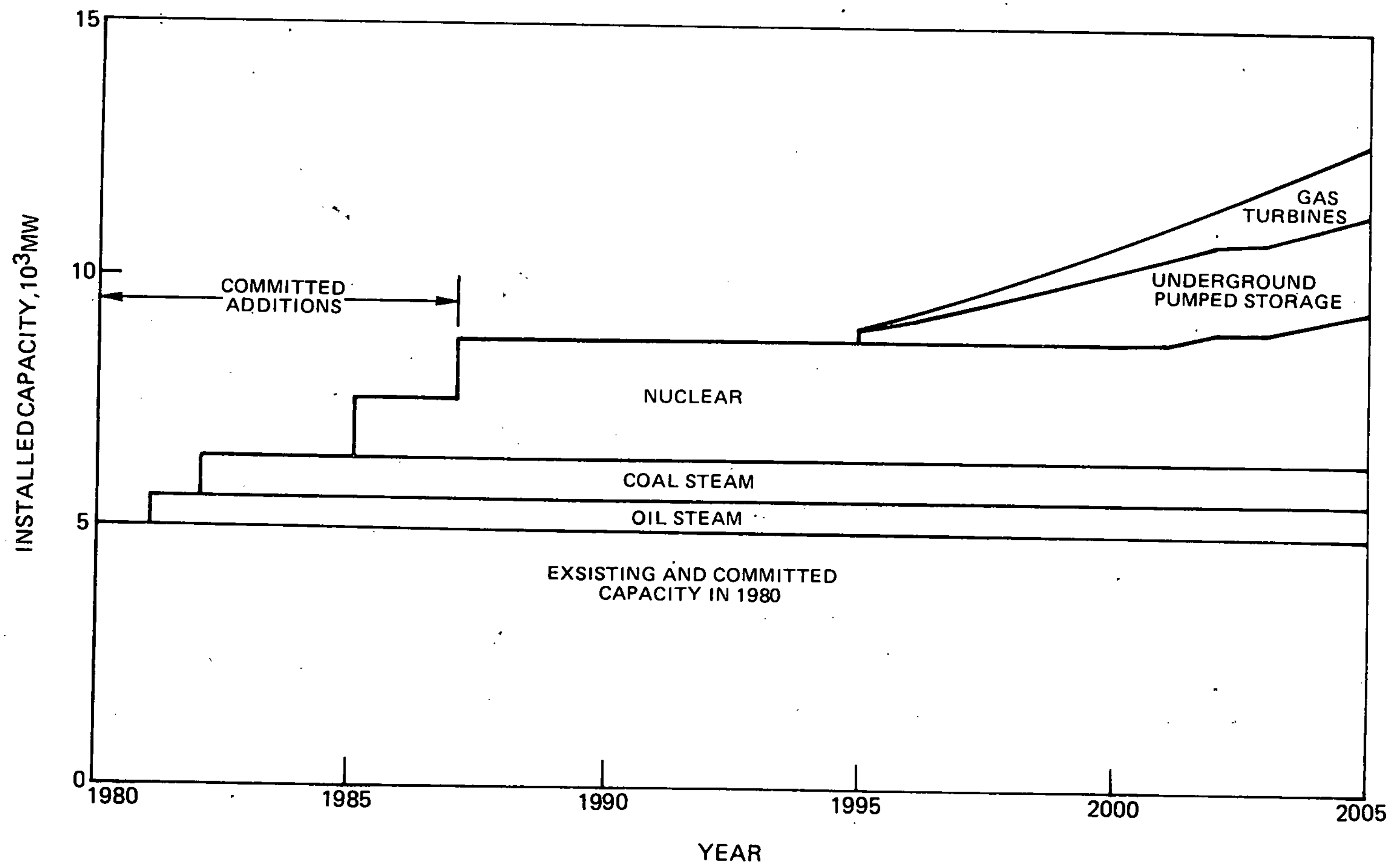


FIG. VI–19

POWER GENERATION COST COMPARISON

COAL PUMPING AT 16 MILLS/KWHR

1980 DOLLARS

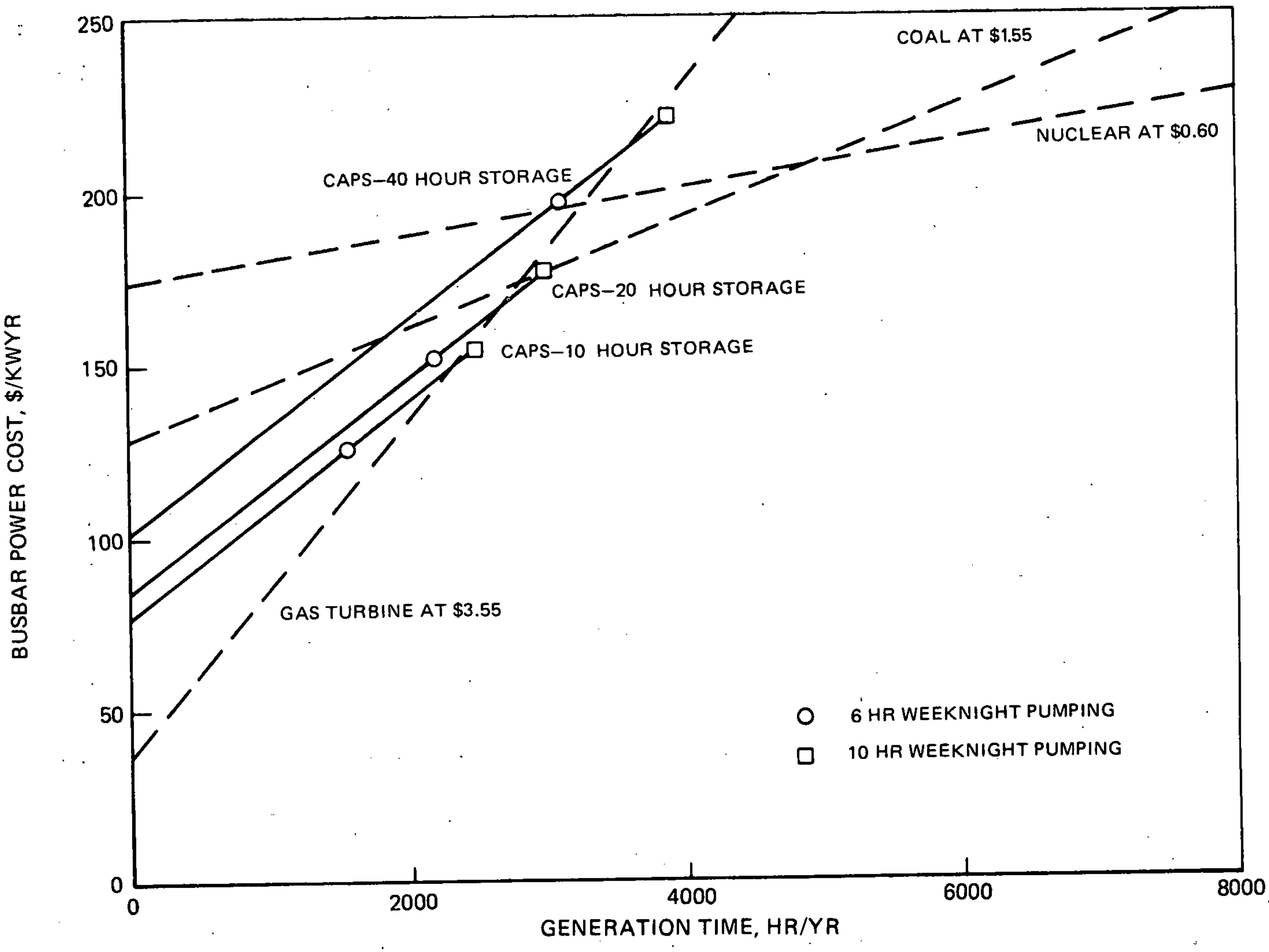


FIG. VI-20

R76–952161–5

POWER GENERATION COST COMPARISON

NUCLEAR PUMPING AT 7 MILLS/KWHR

1980 DOLLARS

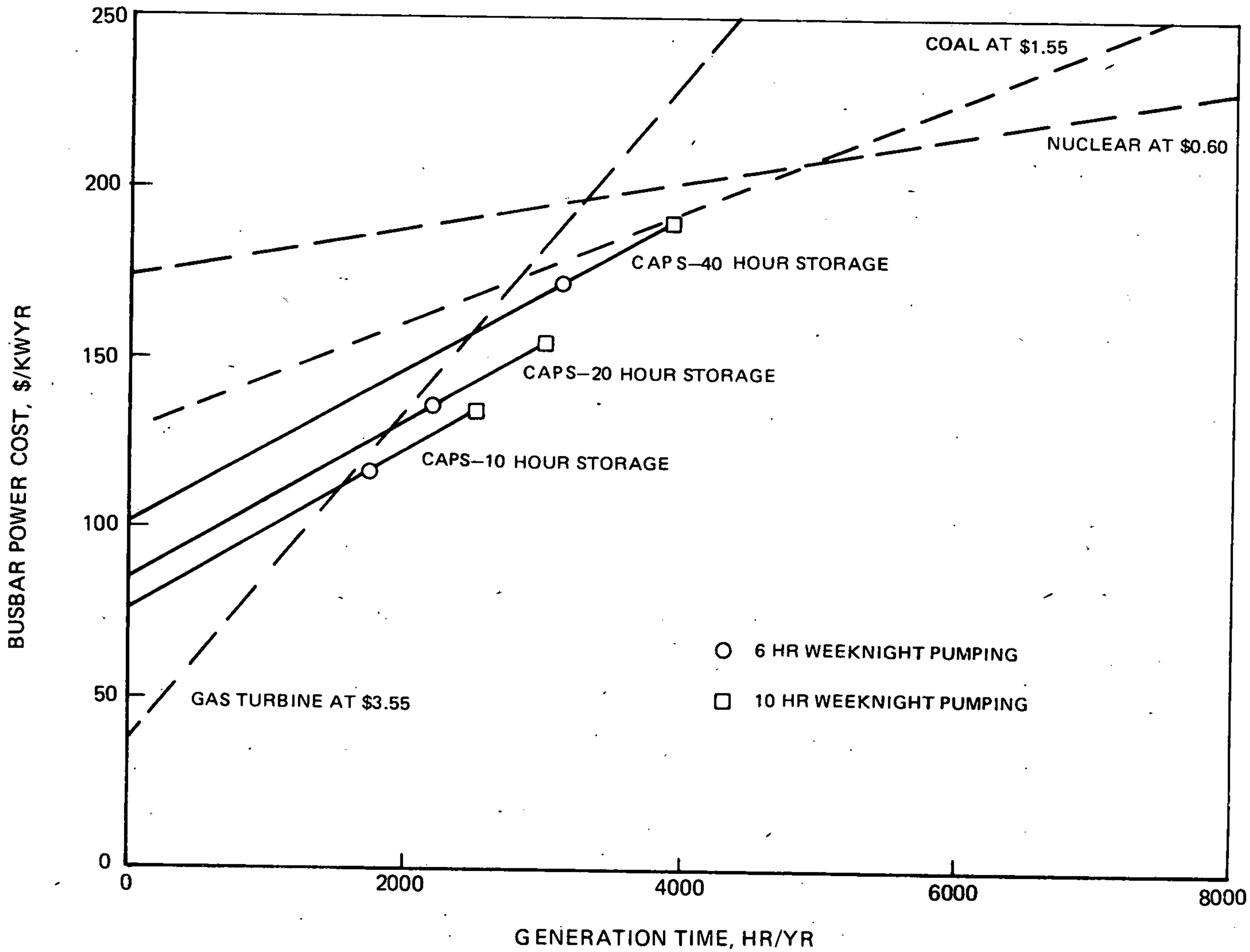


FIG. VI–21

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76–12–66–1