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DEVELOPMENT OF A SOLAR DESICCANT DEHUMIDIFIER

Second Technical Progress Report

By
M. E. Gunderson
K. C. Hwang
S. M. Railing
J. Rousseau

November 10, 1978

Work Performed Under Contract No. EG-77-C-03-1591

AiResearch Manufacturing Company of California
A Division of the Garrett Corporation
Torrance, California

U.S. Department of Energy

Solar Energy

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78-15444

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Prepared by

M.E. Gunderson
K.C. Hwang
S.M. Railing
J. Rousseau

Approved by

J. Rousseau

MASTER

Prepared for
U.S. Department of Energy
Contract EG-77-C-03-1591

GARRETT

AIRESEARCH MANUFACTURING COMPANY
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CONTENTS

SECTION 1

INTRODUCTION AND PROGRAM STATUS

PROGRAM OBJECTIVE

This is the second technical progress report prepared by AiResearch Manufacturing Company of California for the United States Department of Energy under Contract No. EG-77-C-03-1591. The report covers activities from April 1, 1978 to November 1, 1978. The most important items of work done during this period were the system optimization studies and preliminary design. Previous documentation includes the first technical progress report (AiResearch Report No. 78-14957, Volumes 1 and 2) which covered the work done in the first six months of the contract.

This program is aimed at the development of a solar desiccant dehumidifier featuring a rotary bed of granular silica gel and a rotary regenerator. This dehumidifier can be used for air conditioning through adiabatic saturation of the process airstream. Figure 1-1 illustrates a complete air conditioning system based on this principle.

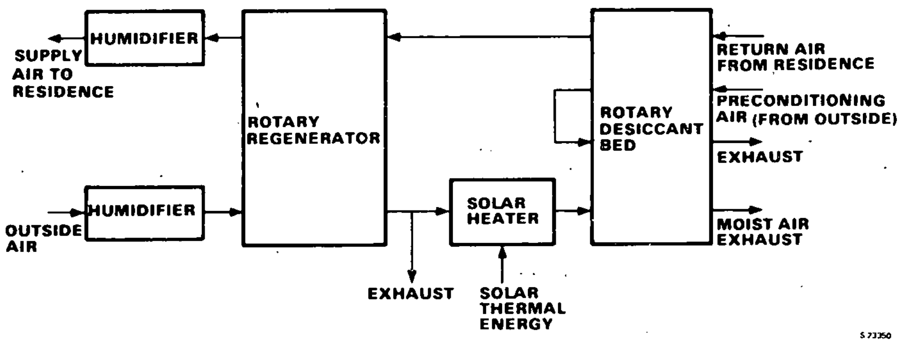

Figure 1-1. Air Conditioner Schematic

The desiccant dehumidifier equipment arrangement is shown in Figure 1-2. The regenerator matrix is a fine screen of galvanized steel wire. The solar heater is located between the desiccant bed and the regenerator. Operation is as described below.

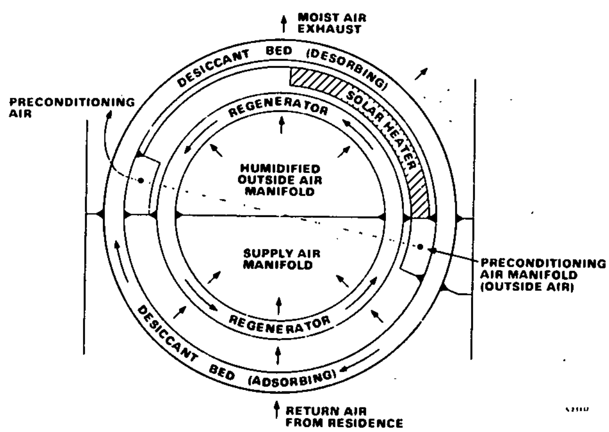

Figure 1-2. Dehumidifier Equipment Arrangement

Warm humid return air from the residence is directed to the adsorbing side of the desiccant bed. Water is adsorbed from the airstream, the temperature of which is increased in the process. The air is then cooled by the rotary regenerator. The specific humidity of this airstream is low enough that its dry-bulb temperature can be lowered by adiabatic humidification to levels adequate for sensible cooling, while retaining reasonable latent cooling capacity.

Ambient outside air is used to regenerate the sorbent bed and cool the rotary regenerator. This stream is humidified adiabatically and circulated through the rotary regenerator. About one-half of this air is then exhausted from the package without flowing through the dryer. The remainder is heated in a fixed-boundary heat exchanger by solar thermal energy and used to desorb the desiccant.

A minor amount of air directly from outside is circulated through the hot portion of the sorbent bed as it rotates from the desorbing zone to the adsorbing zone. In this manner, the bed is cooled to a temperature where it can adsorb moisture when exposed to the return air from the residence. This preconditioning airflow is then used to preheat the bed prior to desorption, thus reducing the solar thermal energy necessary for this process.

SUMMARY

The results of the system optimization studies are presented in Section 2. The studies involved an extensive investigation of the energy saving potential and economic viability of the solar desiccant dehumidifier in different locations in the United States. Conventional electric vapor compression, and solar absorption and Rankine systems also were investigated for comparison.

In general, it was found that the solar desiccant equipment, either by itself or in a hybrid system with an electric vapor compression air conditioner, is economically viable for all three locations considered. Substantial energy savings can be effected as well.

Section 3 documents the seal tests done at AiResearch to develop practical dynamic air seals. Leakage and friction tests were performed on a variety of material combinations and configurations. Dacron felt and silicone rubber were found to give an acceptable combination of leakage, friction, and cost characteristics.

As part of the commercialization studies for the desiccant equipment, a questionnaire was sent to residential air conditioning equipment distributors. The results of the questionnaire are presented in Section 4.

The specifications and drawings for the 1.5-ton prototype are contained in Section 5.

PROGRAM STATUS

Figure 1-3 shows the program schedule, with current status indicated. Phase II (Conceptual Design) has been substantially completed, with the exception

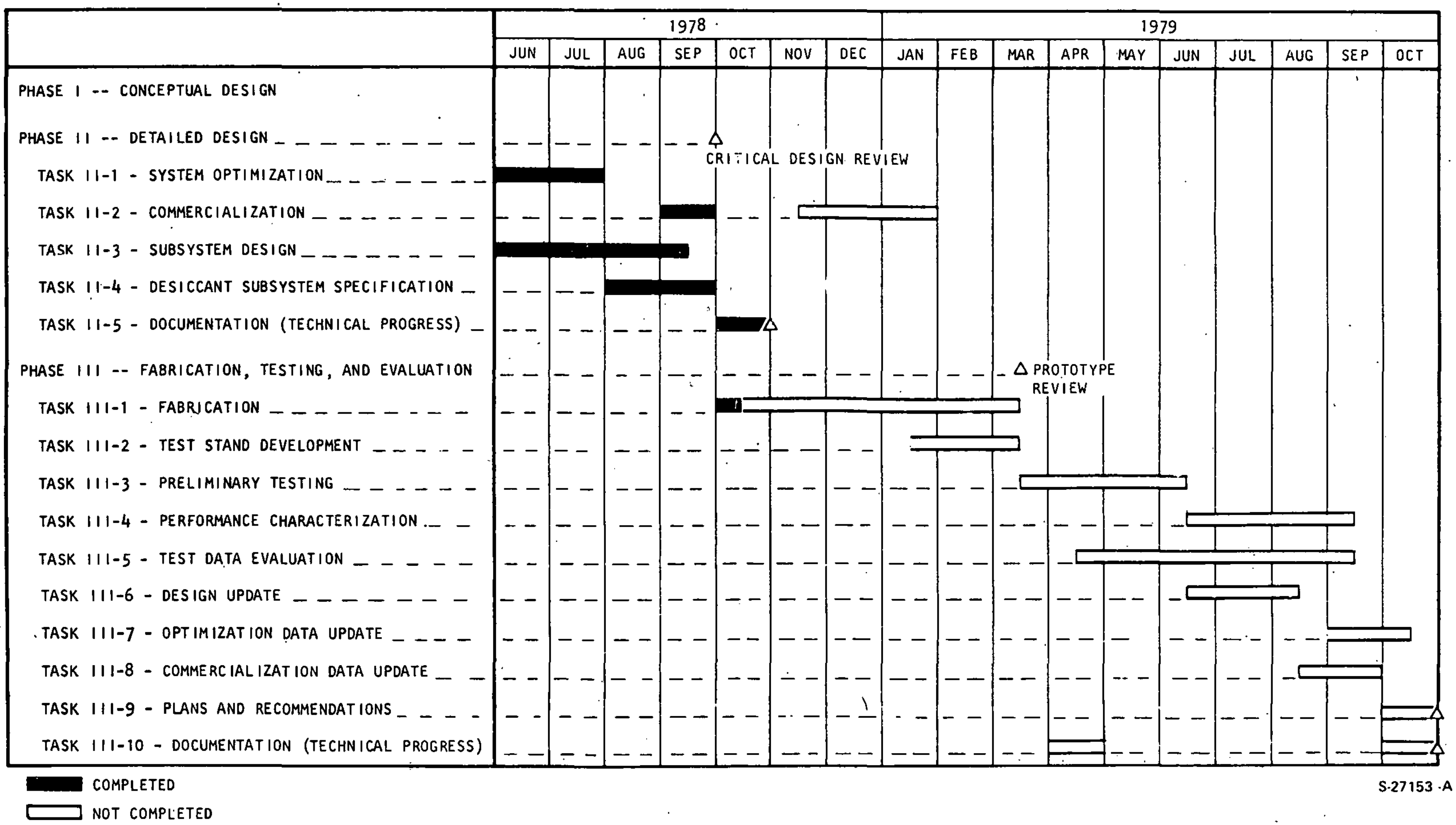

Figure 1-3. Program Schedule

of Task II-2 (Commercialization). The commercialization work will be completed after coordination with Dunham-Bush.

Phase III (Fabrication, Testing, and Evaluation) has been started. Some long leadtime components have been ordered. A detailed fabrication schedule is due to be released the second week in November.

SECTION 2

SYSTEM TRADE STUDIES

SUMMARY

This section includes the final documentation of Task II-1 (System Optimization). The information in this section was presented in part as AiResearch Preliminary Technical Report No. 78-15349 (August 25, 1978), which was distributed to DOE and SERI personnel for review at a meeting at SERI in Golden, Colorado on August 31, 1978. Following this meeting, the analysis was extended to the detailed investigation and comparison of absorption and Rankine systems.

In order to adequately study the feasibility of the solar desiccant system in terms of primary energy savings and economic viability, year-round performance calculations have been performed for various system configurations and different locations. This report details the performance models and economic assumptions used in these studies and presents the results.

Three locations were chosen as being representative of major applications in the United States: Phoenix, Arizona; Apalachicola, Florida; and New York, New York. A baseline system was assumed for each location that would provide 60 percent of space and hot water heating from solar energy, and space cooling by a conventional electric air conditioner. The incremental costs and energy savings for various desiccant systems were then computed with reference to the baseline. Additionally, the economic sensitivity to energy cost escalation rates, initial energy cost, and collector costs were studied.

In general, it was found that the solar desiccant equipment, either by itself or in a hybrid system with an electric air conditioner, is economically viable for all three locations considered. Substantial energy savings can be obtained as well.

Table 2-1 shows the major characteristics of recommended systems found to be reasonable for the 1775-sq ft house model used in this study. The systems shown provide good year-round comfort levels, as well as primary energy savings. There are no significant lifetime costs to the user, referenced to the baseline system.

TABLE 2-1

RECOMMENDED SOLAR DESICCANT COOLING SYSTEMS

Location	Desiccant Cooling Capacity, tons	Vapor Compression Cooling Capacity, tons	Collector Area, sq ft
Phoenix, Arizona	4.0	0.0	860
Apalachicola, Florida	1.5	1.5	425
New York, New York	1.0	0.0	1100*

*Sized by space and domestic water heating requirements.

SYSTEM CONFIGURATION AND PERFORMANCE

Air Conditioning System Computer Model

Figure 2-1 shows the complete air conditioning system as characterized by the computer model used for this study. The model is constructed to allow a wide range of system configurations to be investigated. For example, a vapor compression air chiller is shown in series with the desiccant dehumidifier. This arrangement enables different combinations of vapor compression and desiccant cooling capacities to be tried by changing two input data values. Additionally, either the desiccant or vapor compression capacity can be set equal to zero, so the performance of a purely desiccant or vapor compression system can be evaluated. Different control schemes, auxiliary energy addition methods, and equipment characteristics also can be evaluated.

The performance of the desiccant dehumidifier subsystem is also discussed in this section. The characteristics of the other major equipment elements are as follows.

1. Humidifiers

The humidification efficiency (i.e., the ratio of the actual dry bulb temperature drop to the maximum possible adiabatic dry bulb temperature drop) is assumed to be 90 percent. This performance is achievable with a corrugated cellulose saturator core. Air side pressure drop is calculated as a function of air velocity using a correlation supplied for commercially available saturator core material.

Under some conditions, it is desirable to bypass some of the airflow around the humidifier in the supply circuit. This occurs in locations where the conditioned space humidity tends to become too high, accompanied by too low dry-bulb temperatures. It has been found that bypassing 20 percent of the supply airflow eliminates this problem.

2. Fans

The overall electric power-to-air efficiency assumed for the supply and exhaust fans is 50 percent. This figure is representative of good quality commercial fans. The fans are two-speed, so that the mass flow rate of air can be reduced to 50 percent of full flow in order to reduce system cooling capacity.

3. Solar Collectors

Efficiency and pressure drop correlations for commercially available double-glazed flat plate collectors are used. The particular collectors selected have been used in other AiResearch solar projects, and therefore quality and performance are known.

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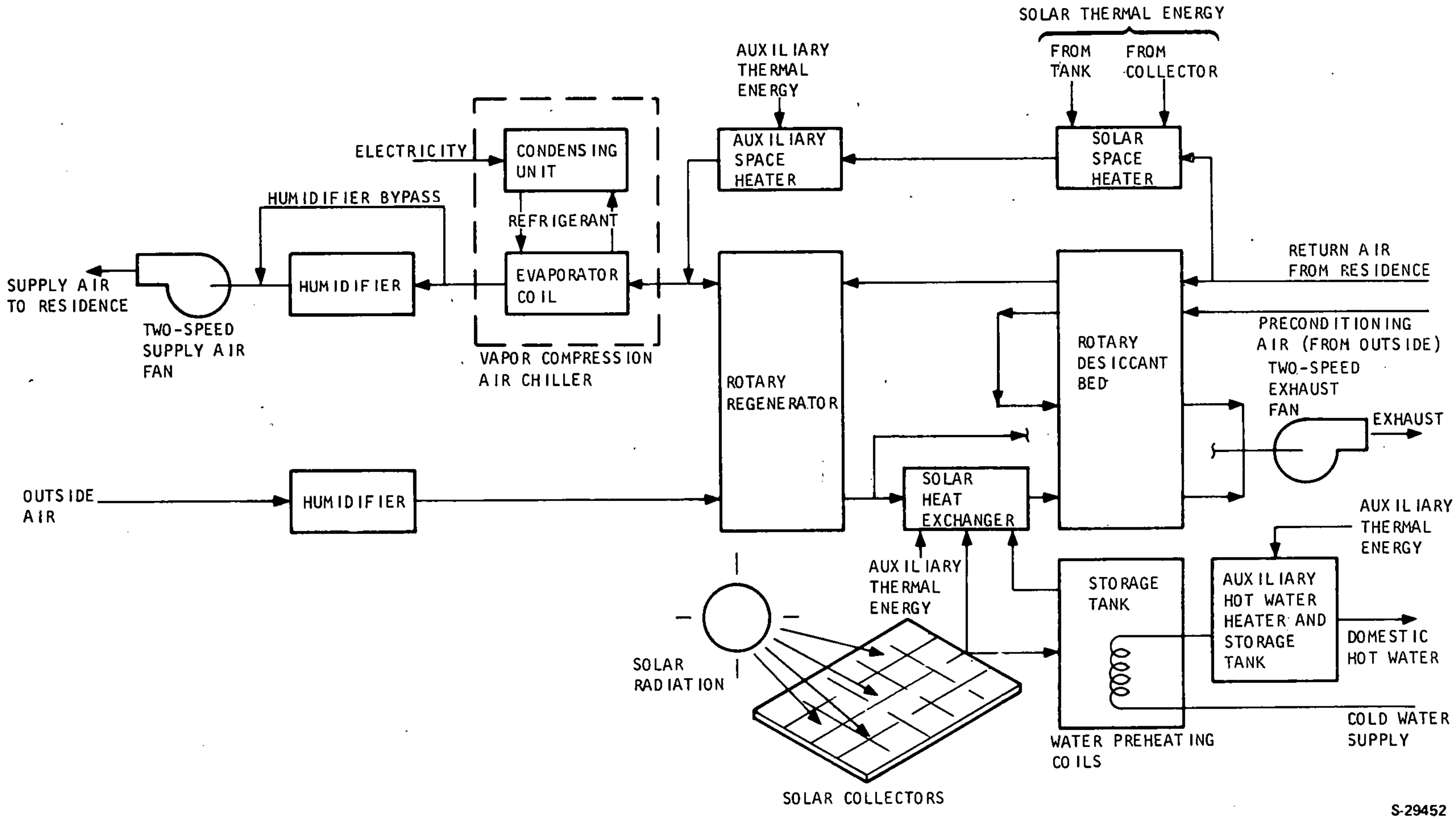


Figure 2-1. Complete Air Conditioning System as Characterized by Computer Model

4. Storage Tank

The solar thermal energy is stored in an insulated tank full of water. The storage temperature is limited to 200°F to eliminate the need for pressure vessel construction. Previous AiResearch studies have shown that about 10 lb of water per square foot of collector area provides an economical storage system, so the size of the tank is varied in proportion to the size of the collector field in the computer model. Convective energy losses are calculated from the sides of the tank and piping to the ambient. The losses are not included in the cooling load or heating input to the conditioned space.

5. Space Heaters

The solar and space heaters are assumed to be of adequate size to handle the instantaneous load. If adequate solar thermal energy is not available from the storage tank and collectors, the auxiliary heater is switched on. This simplified treatment of the heating process is equivalent to assuming that properly sized heaters have been selected and installed.

6. Domestic Water Heaters

Domestic hot water preheating coils are installed in the storage tank. The normal practice in such installations is to use a very large coil yielding a very high effectiveness; the preheated water temperature therefore closely approaches the storage tank temperature. Like the auxiliary space heater, the auxiliary hot water heater is assumed to be of adequate size to meet any instantaneous load.

7. Desiccant Subsystem Solar Heat Exchanger

The heat exchanger, which is a part of the desiccant subsystem insofar as performance goes, is supplied with solar thermal energy from either the collectors or the storage tank, whichever can supply the higher temperature. In the case when adequate solar thermal energy is not available, and the particular system configuration includes auxiliary desiccant cooling, auxiliary thermal energy is provided to the solar heat exchanger at a specified temperature. The auxiliary energy is treated as a closed loop; the water leaving the heat exchanger returns directly to the auxiliary heater, and not to the storage tank. Therefore, no auxiliary energy is expended to heat the storage tank.

Desiccant Subsystem Performance Model

The desiccant subsystem configuration was the subject of an extensive optimization study, the results of which led to the present design. The optimization work has been previously documented (Refs. 1 and 2). After review of the program by DOE, the capacity of the basic prototype was changed from 3 to 1.5 tons. The performance predictions for the smaller unit are shown in Figures 2-2, 2-3, and 2-4. In this study, the cooling capacity relationships were scaled as needed to correspond to the desired design-point capacity.

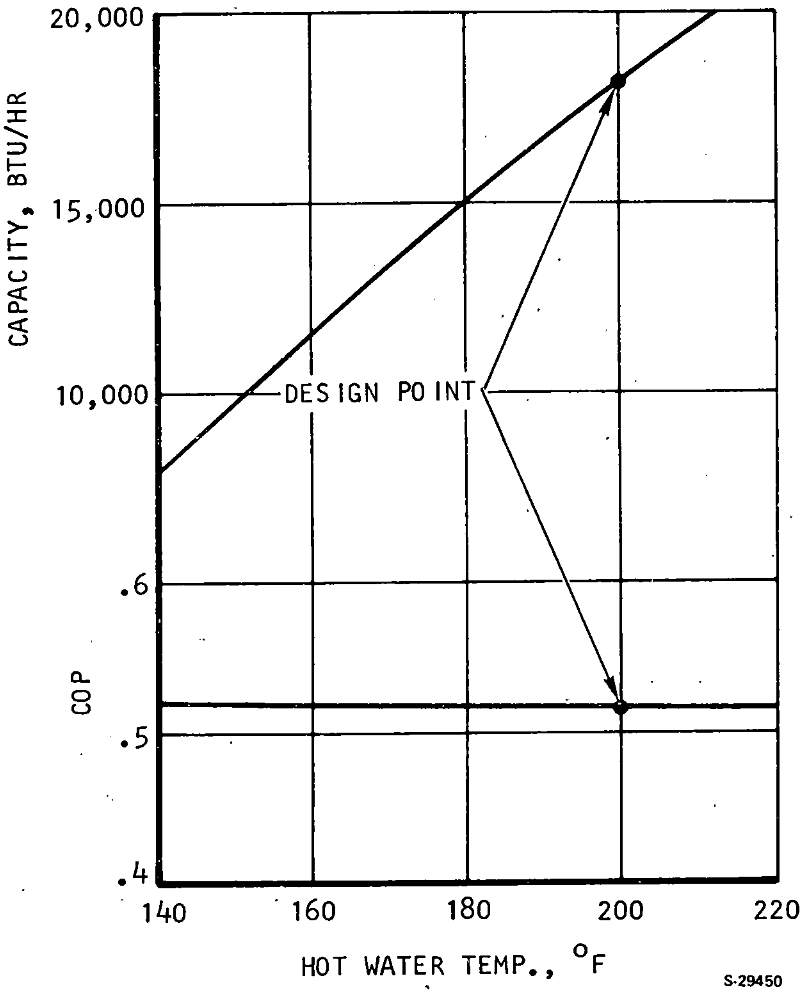


Figure 2-2. Variations in Capacity and COP as Functions of Hot Water Inlet Temperatures

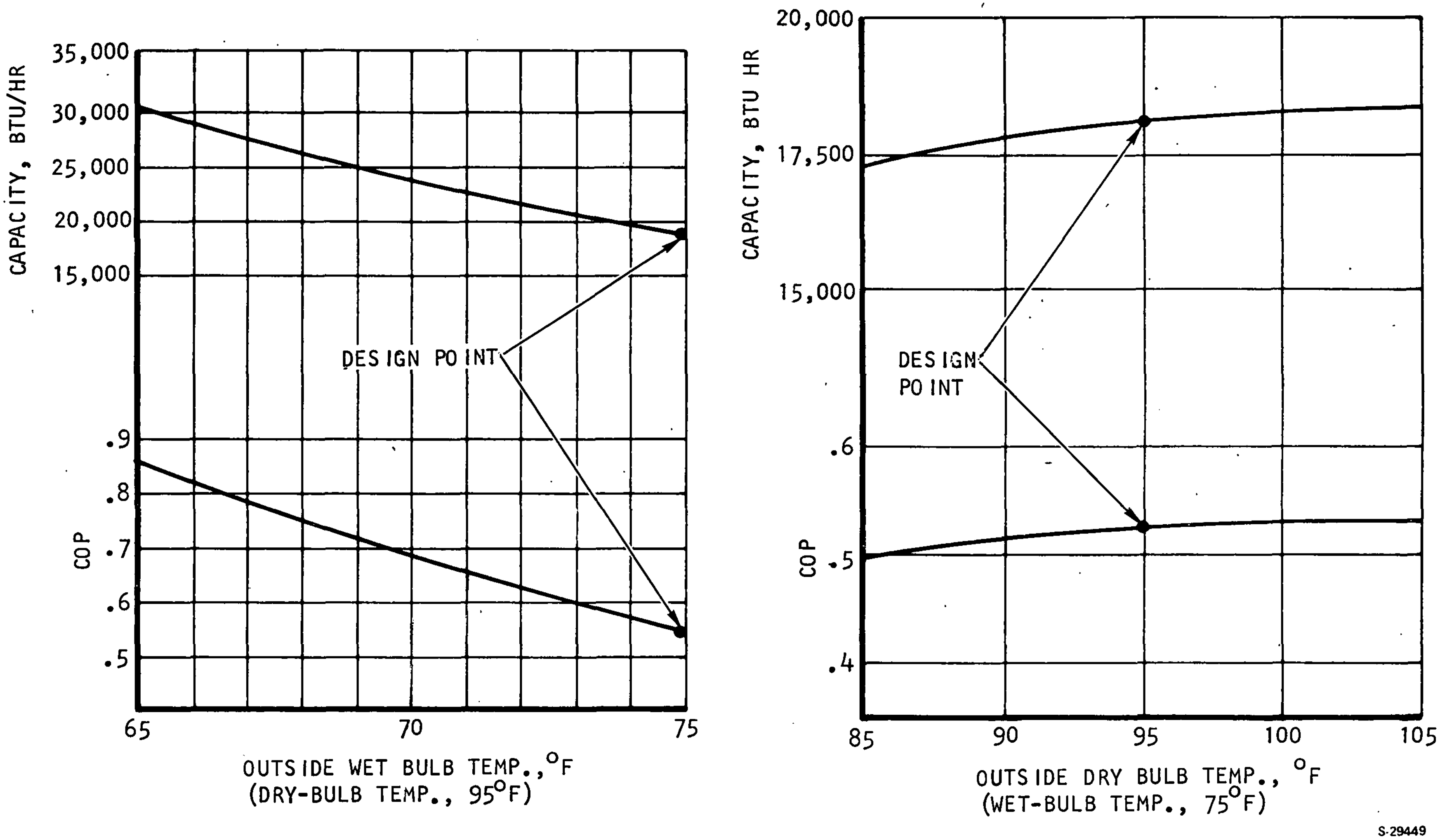


Figure 2-3. Capacity and COP vs Outside Wet-Bulb and Dry-Bulb Temperatures

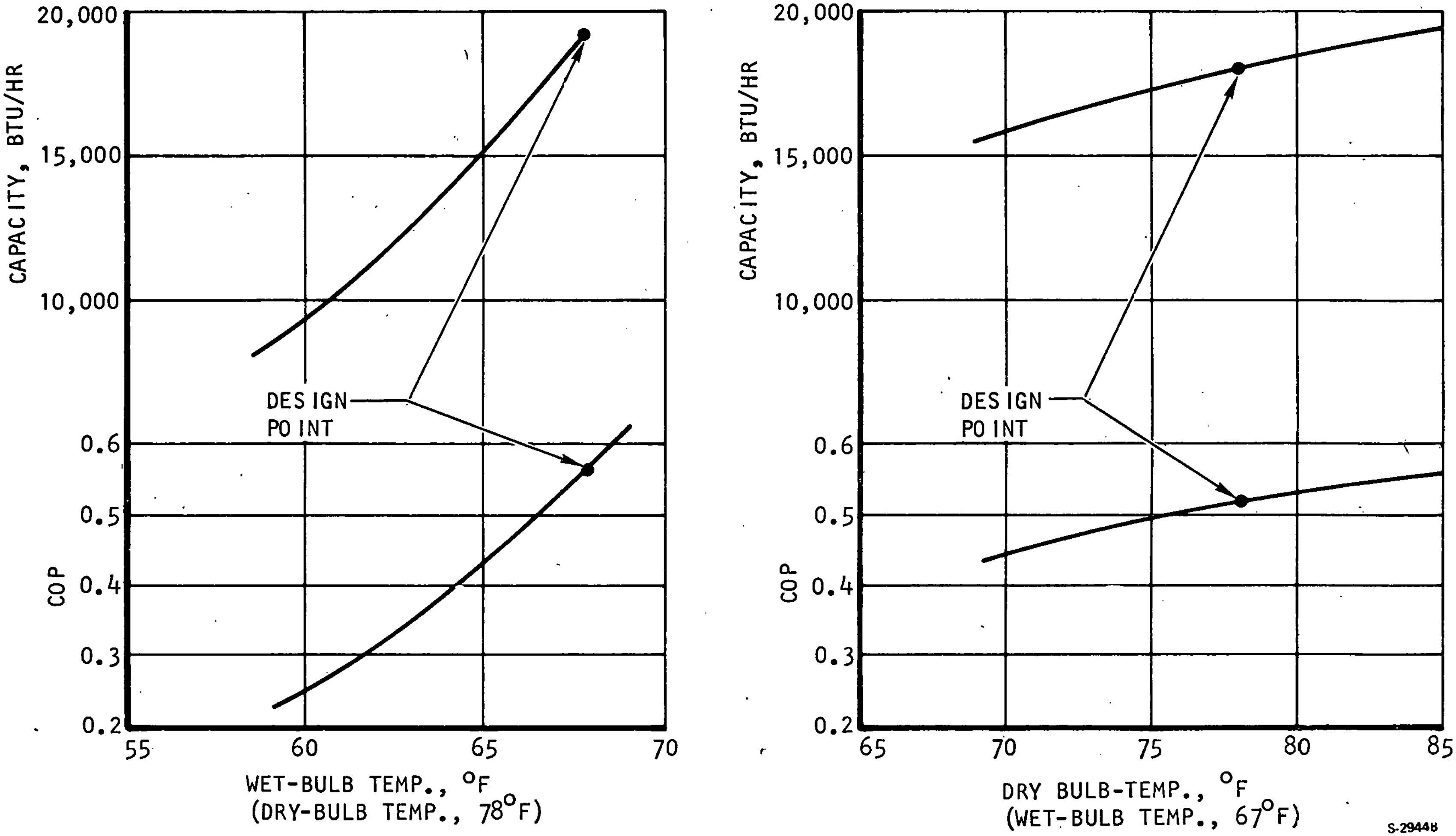


Figure 2-4. Increased Capacity and COP as a Result of Increased Inside Wet- or Dry-Bulb Temperature

Figure 2-3 shows that outside wet-bulb temperature increases will decrease capacity and COP. Outside dry-bulb temperature changes have a lesser effect.

Figure 2-4 shows that increases in inside wet-bulb or dry-bulb temperature will increase capacity and COP. The effect of wet-bulb temperature is particularly pronounced, suggesting that the desiccant system would be most suitable for high latent load application.

This type of off-design performance prediction must be used cautiously because a change in only one parameter is likely never to occur in actual operation. For example, an increase in outside wet-bulb temperature will most likely be accompanied by an increase in indoor wet-bulb temperature due to increased latent loads from infiltration. The net result may be no change in either capacity or COP. The computer model of the integrated load-system behavior accounts for the coupling among the parameters.

Control Schemes

The purpose of any air conditioning system is to maintain the conditioned space in a comfortable condition. While this appears to be a reasonably simple goal, the expression of comfort in measurable parameters is quite elusive. Several correlations are available that give the percentage of persons who would feel comfortable at particular conditions, but the disparities among these correlations indicate very inexact measurements. Furthermore, comfort is generally given as a function of dry-bulb temperature and one other variable that indicates humidity, such as wet-bulb temperature or relative humidity. Both wet-bulb temperature and relative humidity are difficult to sense accurately and repeatably with simple instruments. It is, therefore, not practical to use humidity as a control parameter for a residential air conditioner.

Figure 2-5 shows the comfort zone used in this study. This zone was constructed from several of the published correlations. In the computer model, the fraction of the time that the conditioned space is outside the zone gives an output of "percent of time not comfortable". This can be an indication of the effectiveness of a particular system. The comfort zone is not used to control the system.

Figure 2-6 illustrates the control scheme that is included in the computer model. Four cooling mode control temperatures, T_{vc}, T_{hi}, T_{lo}, and T_{off}, are specified for a particular system as inputs. The system is controlled as shown based on the controlled space dry-bulb temperature. It has been found that significant changes in the percent of time not comfortable can often be made by shifting these four temperatures up or down by 1°F. The best set of temperatures is the highest that would give acceptable comfort levels, since this would result in the lowest energy consumption.

This control scheme has been selected for two primary reasons. First, experimentation has shown that good comfort levels can be maintained without sensing controlled space humidity. Depending upon the location and the system configuration, a set of four control temperatures can be selected that will match latent and sensible cooling capacity closely to the latent and sensible loads.

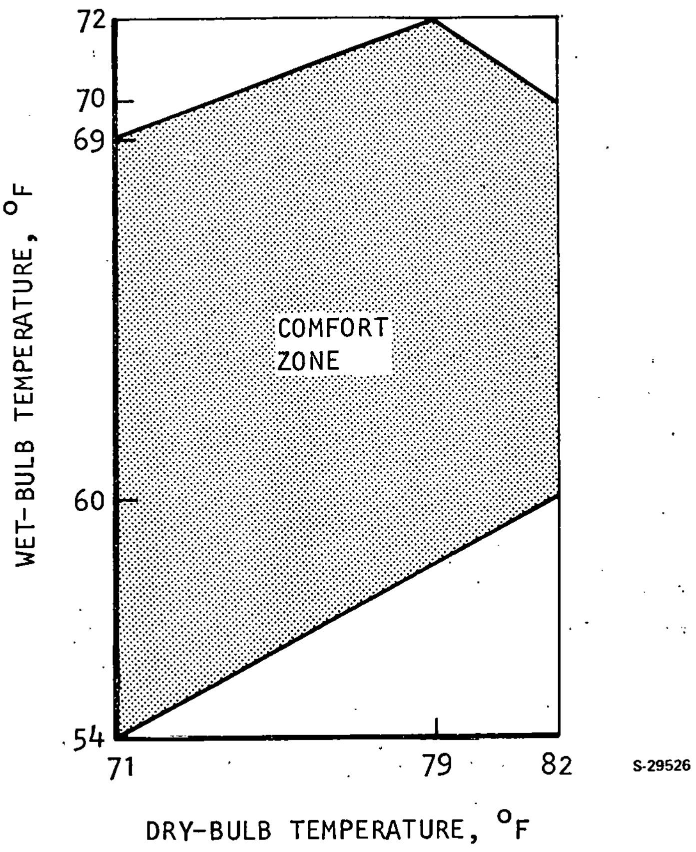


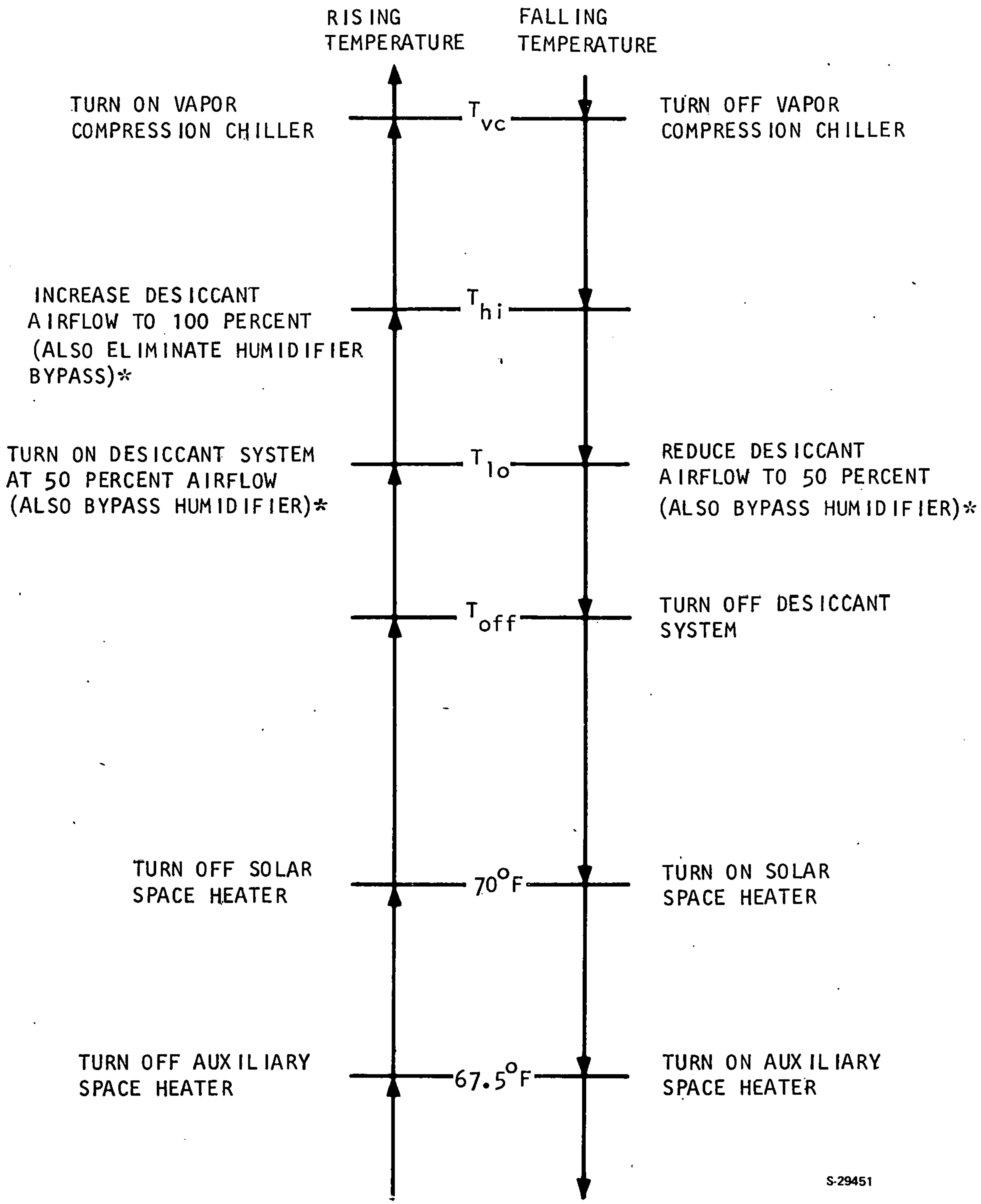
Figure 2-5. Comfort Zone

The second consideration involves a fundamental characteristic of the desiccant subsystem. Upon initial startup, several rotational cycles of the desiccant bed are required before steady-state operation is achieved. This would be on the order of a half-hour or so. Therefore, it is not practical to cycle the system on and off to match average capacity to the load as is the practice with conventional vapor compression air conditioners. This problem is overcome by reducing the speed of the fans, and thus the airflow and capacity, when the temperature of the controlled space drops below a value that would require full capacity. By this method, the desiccant bed is kept in operation and the transient startup problem is avoided.

In some locations, it is also desirable to bypass some supply air around the humidifier when the fans are turned down to low speed. twenty percent of the supply air is normally bypassed, which is sufficient to keep the conditioned space humidity from becoming too high.

Other Solar Cooling Systems

For the purposes of comparison, solar powered absorption and Rankine cooling systems can be substituted into the computer model in place of the desiccant subsystem. The absorption system used in this study is a conventional single effect LiBr/H_2O chiller. It has a performance typical of that for commercially



*HUMIDIFIER BYPASS OPTIONAL FOR SOME LOCATIONS

Figure 2-6. Computer Model Control Scheme available equipment. Currently, new absorption equipment is undergoing development by other DOE contractors. When this new equipment is on the market, improved performance should be possible, especially at low water supply temperatures.

The Rankine cooling system is quite unconventional. The equipment, which is now under development at AiResearch under NASA sponsorship, is a combination cooling and heating system. Only the cooling function has been used in this study. An important feature of this equipment is the integral permanent magnet motor, the armature of which is integral with the turbocompressor rotor. A schematic of this system is shown in Figure 2-7.

Detailed descriptions and performance information for the absorption and Rankine cooling equipment are given in Reference 1.

Example of Computer Model Output

Figure 2-8 is an example of the output of the AiResearch computer model used in this study. Data are tabulated for each month of the year, as well as an annual total. This example shows a 1.5 ton desiccant/2.5-ton vapor compression hybrid system, using 500 sq ft of Daystar collectors, with no auxiliary energy addition for the desiccant system. This output was a result of a study to optimize the cooling control temperatures.

ECONOMIC MODEL AND ENERGY CONSUMPTION ASSUMPTIONS

Baseline System Datum

The economic analysis conducted for this project is intended to show whether or not the solar desiccant system is a viable method of solar-powered air conditioning. In most locations, air conditioning is considered a nonessential addition to a residence, whereas space and domestic hot water heating are essential. A basic solar energy system for a house would therefore most likely be a heating system with either conventional or solar air conditioning added as an option. To be proven viable, the desiccant system must be shown to be an acceptable alternative to a conventional electric air conditioner when added to a basic solar heating system.

The baseline systems for this study are therefore defined as solar space and water heating systems that provide 60 percent of the combined space and water heating load for the residence, plus conventional electric air conditioners of sufficient size to maintain the residences in the comfort zone at least 95 percent of the time during air-conditioning season. The cost and energy consumption of the baseline system for a particular location are taken as the datum for the comparison of other systems at that location.

The baseline systems for the three locations are shown in Table 2-2.

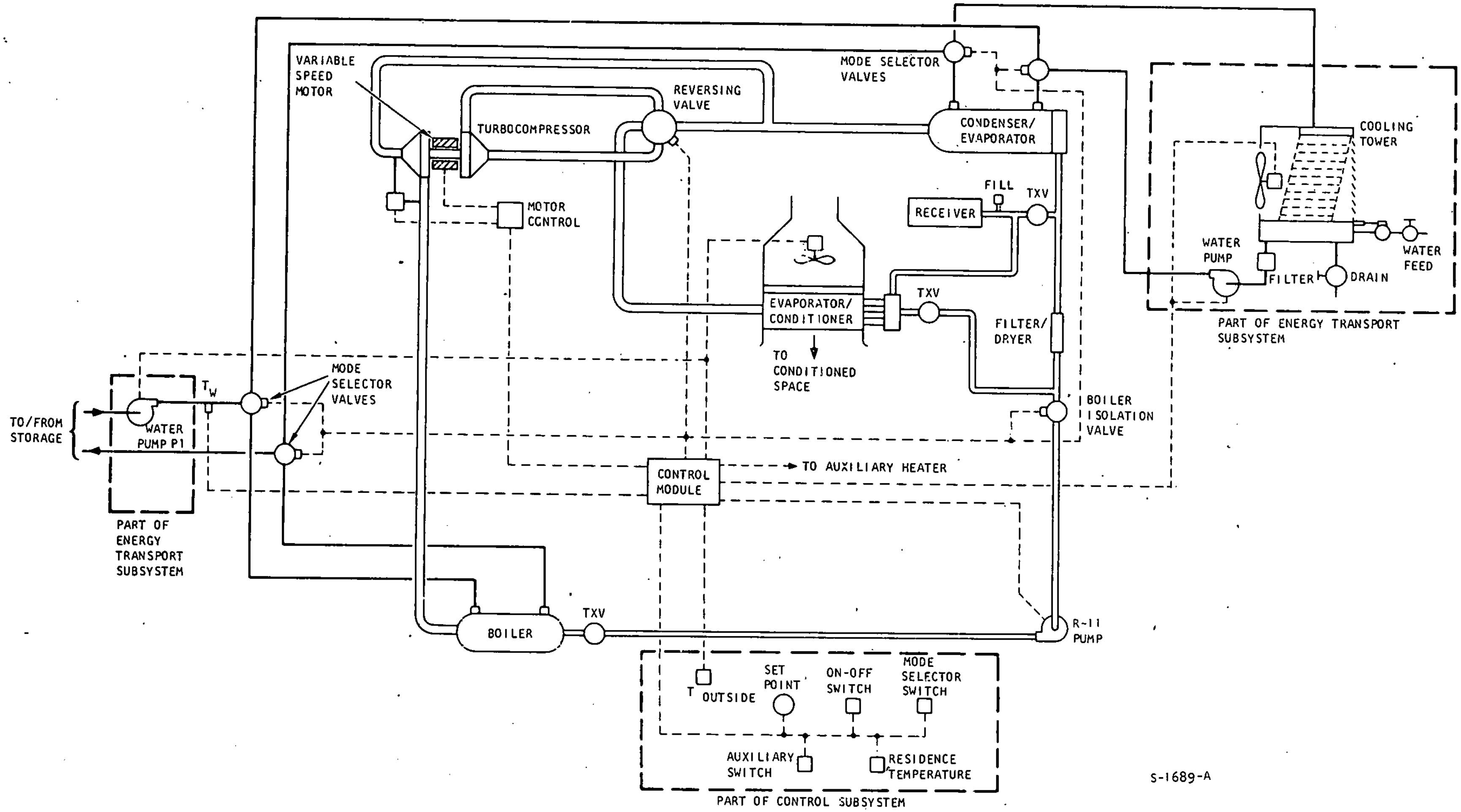


Figure 2-7. Rankine Cooling System

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APALACHICOLA 1.5T DSAC + 2.5T VCAC PARALLEL DAYSTAR COLLECTOR   RUN ON 27 JUN 78 AT 15:43:44   PAGE 3

COLLECTOR AREA(SQ FT)= 500.   TANK SIZE(LB)= 5000.   % TIME NOT COMFORTABLE= 15.03

MONTH	1	2	3	4	5	6	7	8	9	10	11	12	YEAR
COLLECTOR													
QINC,BTU	.1560+08	.1452+08	.2016+08	.2517+08	.3053+08	.2435+08	.2621+08	.2329+08	.2434+08	.1803+08	.1961+08	.1782+08	.2596+09
QCOL,BTU	.8539+07	.5883+07	.7806+07	.6234+07	.1293+08	.1076+08	.1170+08	.1060+08	.1158+08	.7444+07	.8140+07	.9624+07	.1112+09
% QCOL	54.74	40.51	38.73	24.77	42.37	44.19	44.64	45.50	47.59	41.29	41.50	54.00	42.85
C PWR,KWH	32.86	24.49	25.57	19.68	36.42	34.10	36.27	32.01	31.93	24.95	24.95	32.55	355.8
HEATING													
QHT,BTU	.1145+08	.5632+07	.6093+07	.1871+07	.0000	.0000	.0000	.0000	.0000	.3410+06	.5963+07	.9066+07	.4042+08
QHAUX,BTU	.4440+07	.1440+07	.9600+06	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.8400+06	.1080+07	.8760+07
QHSOL,BTU	.7010+07	.4192+07	.5133+07	.1871+07	.0000	.0000	.0000	.0000	.0000	.3410+06	.5123+07	.7986+07	.3166+08
% SOL HTG	61.22	74.43	84.24	100.0	.0000	.0000	.0000	.0000	.0000	100.0	85.91	88.09	78.33
H PWR,KWH	145.8	76.47	66.67	8.575	.0000	.0000	.0000	.0000	.0000	1.575	51.97	108.8	459.9
COOLING													
QCS,BTU	.0000	.2422+05	.0000	.3723+06	.4123+07	.6009+07	.6845+07	.6820+07	.5501+07	.2592+07	.2458+06	.0000	.3253+08
QCL,BTU	.0000	.1963+05	.0000	.4902+06	.2250+07	.3169+07	.4070+07	.4351+07	.2899+07	.1768+07	.2905+06	.0000	.1931+08
QCT,BTU	.0000	.4385+05	.0000	.8625+06	.6373+07	.9177+07	.1092+08	.1117+08	.8401+07	.4360+07	.5363+06	.0000	.5184+08
SHF	.0000	.5524	.0000	.4317	.6469	.6547	.6271	.6105	.6549	.5945	.4583	.0000	.6276
QCAUX,BTU	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
QCSOL,BTU	.0000	.6920+05	.0000	.1620+07	.1032+08	.8300+07	.9114+07	.8118+07	.9088+07	.4733+07	.9967+06	.0000	.5236+08
% SOL CLG	.0000	100.00	.0000	100.00	85.76	42.45	37.66	31.74	52.83	52.66	100.00	.0000	48.60
% VC CLG	.0000	.0000	.0000	.0000	14.24	57.55	62.34	68.26	47.17	47.34	.0000	.0000	51.40
% AUX CLG	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000	.0000
VC PWR,KWH	.0000	.0000	.0000	.0000	95.42	576.6	743.6	840.3	439.4	221.3	.0000	.0000	2917.
DS PWR,KWH	.0000	.9000	.0000	19.05	195.4	199.9	222.6	200.4	199.6	100.1	15.06	.0000	1153.
DOMESTIC HOT WATER													
QTOT,BTU	.1551+07	.1401+07	.1551+07	.1501+07	.1551+07	.1501+07	.1551+07	.1551+07	.1501+07	.1551+07	.1501+07	.1551+07	.1826+08
QAUX,BTU	.7168+06	.3874+06	.2915+06	6480.	1902.	.1585+05	502.5	.1722+05	.2147+05	.6664+05	.2004+06	.5247+06	.2251+07
QSOL,BTU	.8345+06	.1014+07	.1260+07	.1495+07	.1549+07	.1485+07	.1551+07	.1534+07	.1480+07	.1485+07	.1301+07	.1026+07	.1601+08
% SOLAR	53.79	72.35	81.21	99.57	99.88	98.94	99.97	98.89	98.57	95.70	86.65	66.17	87.68

*FIN

Figure 2-8. Example of Computer Model Output

TABLE 2-2

BASELINE SYSTEM CHARACTERISTICS

Location	Collector Area, sq ft	Fuel Oil Equivalent (FOE) Consumption, 10^6 Btu/yr	Vapor Compression Air Conditioner, tons
Phoenix, Arizona	175	183.1	4.0
Apalachicola, Florida	250	146.0	3.0
New York, New York	1100	138.7	1.0

Fuel Oil Equivalence (FOE)

Electrical and auxiliary thermal energy requirements for a particular system are taken from the computer model output. These values do not reflect conversion efficiencies. To convert to standard units of FOE, a 65 percent efficiency is assumed for the auxiliary space and water heaters, and for the heater supplying auxiliary thermal energy to the desiccant system. Electrical energy is converted to FOE by applying a 30 percent overall conversion and transmission efficiency.

Energy Costs and Escalation Rates

Two significant factors in the economic viability of energy conservation systems are (1) the starting costs of energy and (2) the rate at which the costs can be expected to increase. Present costs of energy can be readily found for a particular location, but the escalation rate is, unfortunately, a matter of guesswork.

The energy escalation rate is defined as the compound interest rate at which the cost of energy increases, exclusive of the effects of inflation. That is,

$$(1 + f) = (1 + e)(1 + i)$$

where f = rate at which energy rates increase, decimal

e = energy escalation rate, decimal

i = inflation rate, decimal

Therefore,

$$e = \frac{1 + f}{1 + i} - 1$$

Figure 2-9 shows the range of energy starting costs and escalation rates investigated in this study. For the baseline systems, nominal costs of \$0.05/kwhr and \$3.00/10^6 Btu were assumed for electricity and fossil fuel, respectively. The nominal escalation rate was taken as five percent.

Equipment Costs

1. Solar Collector Costs

The cost of solar collectors and associated plumbing, pumps, storage tank, and installation was taken as \$20 per sq ft for the baseline systems. Values of \$15 and \$10 per sq ft were also used in the study.

2. Vapor Compression and Desiccant Subsystem Costs

The installed costs of the vapor compression chiller and desiccant systems are shown in Figure 2-10 as functions of cooling capacity. The vapor compression prices were assembled from manufacturer's quotations and construction estimating data. The desiccant subsystem cost estimates were made by AiResearch with consultation from Dunham-Bush.

Life Cycle Costs

The assumed lifetime of all the systems considered is 20 years. This was selected as an attainable goal in terms of equipment longevity. Although it would be possible to design equipment for longer lifetimes, to do so would undoubtedly make the equipment too expensive for residential installations. Also, it is not realistic to try and project parameters such as fuel costs longer than 20 years in the future.

All money amounts referenced in this study are in 1978 dollars. Energy cost escalation rates, as well as the real interest rate, are exclusive of the effects of inflation. In terms of the rate charged by banks for residential loans, the real interest rate is the compound interest rate that would be charged for a loan if the inflation rate was zero. As inflation differs from zero, the bank rate charged must include both inflation and the real interest rate, as follows:

$$(1 + b) = (1 + i)(1 + r)$$

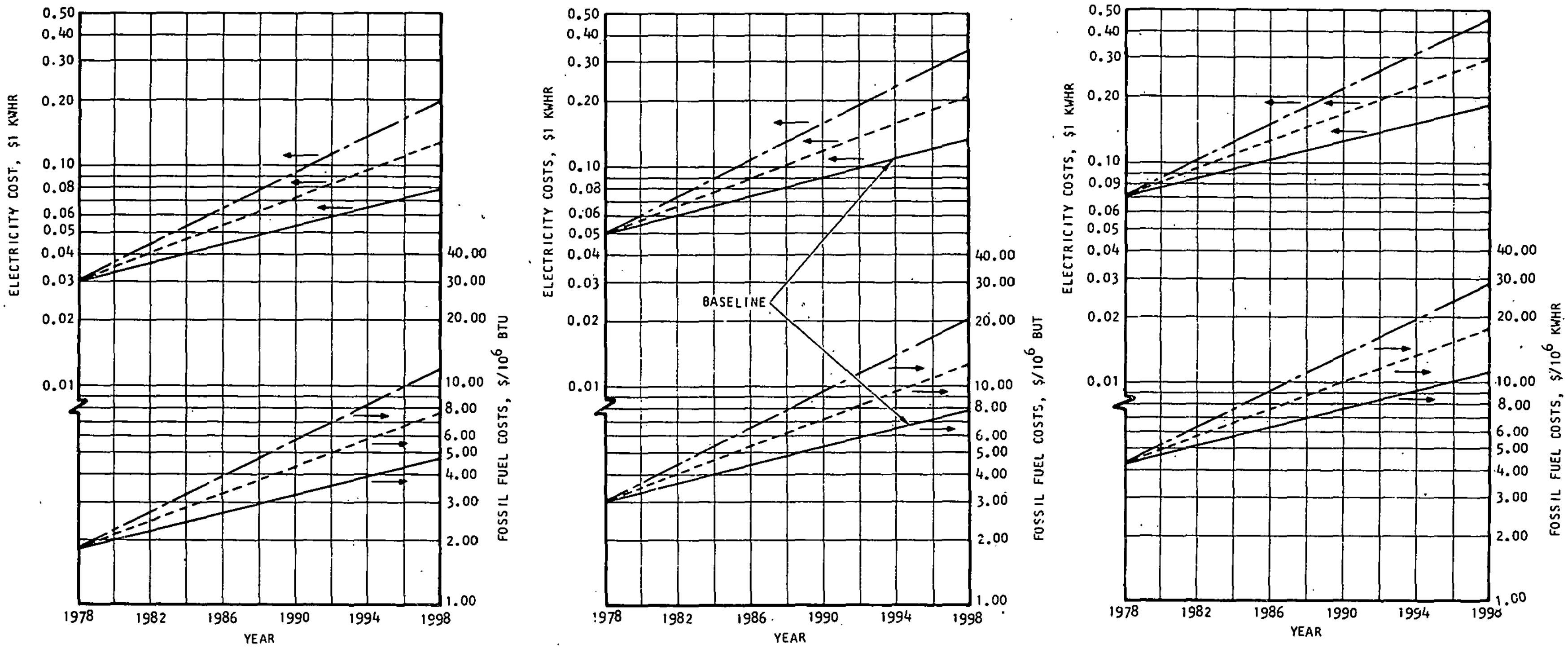
where b = total bank interest rate, decimal

i = inflation rate, decimal

r = real interest rate, decimal

or,

$$r = \frac{1 + b}{1 + i} - 1$$



LOW STARTING COSTS

NOMINAL STARTING COSTS

HIGH STARTING COSTS

ENERGY COST ESCALATION RATE
- - — 10 PERCENT
- - - - - 7.5 PERCENT
——— 5 PERCENT

Figure 2-9. Energy Costs

S-29480A

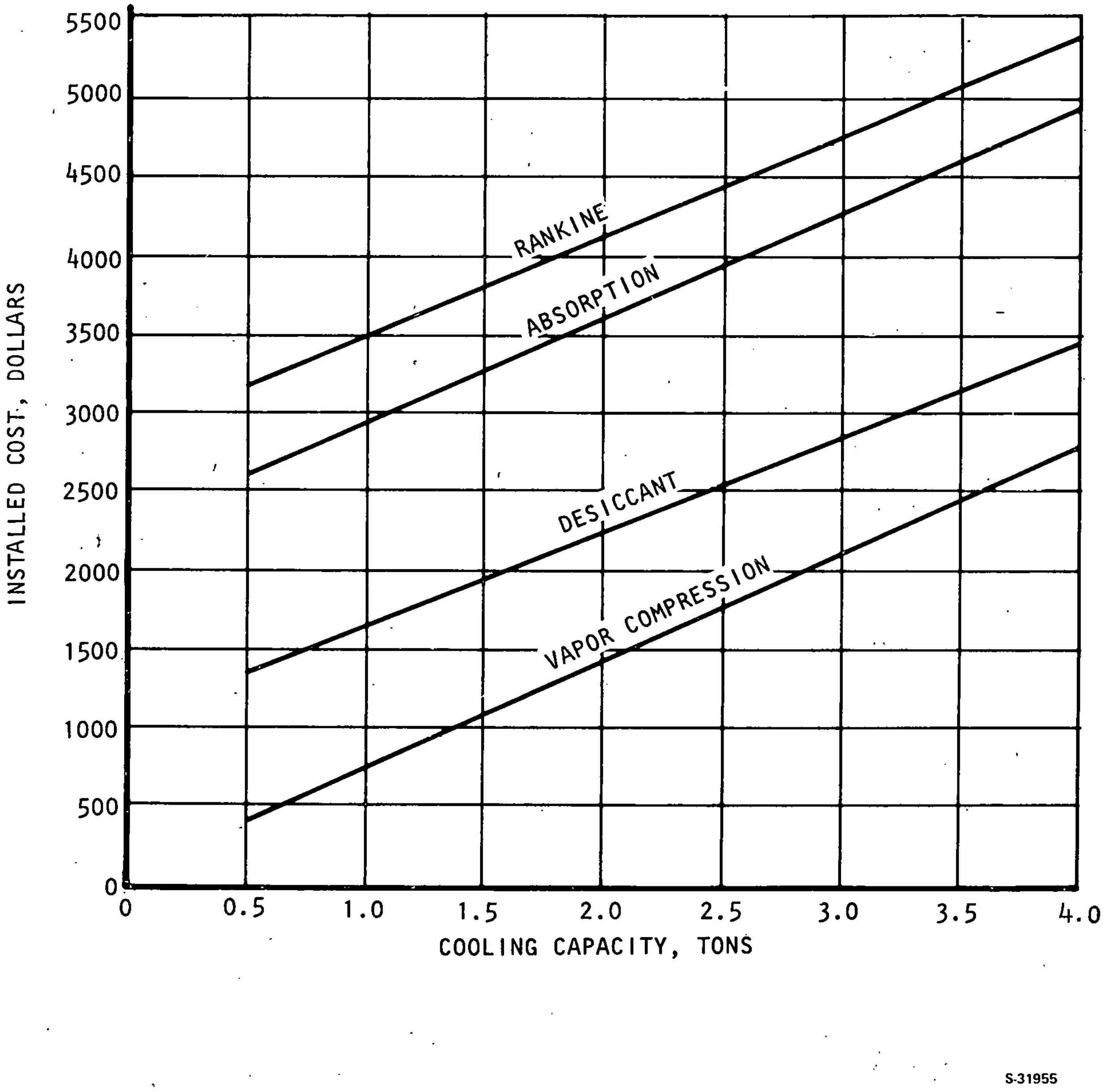


Figure 2-10. Installed Costs of Desiccant, Vapor Compression, Absorption, and Rankine Equipment

In this study, the bank interest rate was nine and three-quarters percent ($b = 0.0975$), and inflation was six and one-half percent ($i = 0.065$). The resulting real interest rate is 3.1 percent ($r = 0.031$).

RESULTS

For each of the three locations, several system configurations are presented here. For each configuration, the size of the collector field is varied over a reasonable range, for different values of energy cost and escalation rate, and collector cost. The results show the amount of energy (in FOE terms) that is expended or saved each year, and the 20-year lifetime cost or benefit, referenced to the baseline datum.

In each city, the first system is a solar space and water heating system with an electric air conditioner. The collector area that yields 60 percent of the space and water heating by solar energy is, by definition, the baseline system. Collector areas larger than the baseline provide additional heating, whereas smaller areas rely more upon auxiliary fuel for heating. Of course, the zero collector area case is a conventional nonsolar heating and cooling system.

The next system for each city is a solar space and water heating system, with a desiccant cooling system incorporating auxiliary thermal energy input. When insufficient solar energy is available, auxiliary thermal energy is supplied. The zero-collector case would be a fuel-fired desiccant system.

Solar desiccant, absorption, and Rankine systems are shown for each city for purposes of comparison. These systems do not have auxiliary thermal augmentation, but the Rankine system is motor augmented.

Depending upon the requirements of the particular location, some desiccant and vapor compression hybrids are reasonable configurations. Where applicable, these are presented.

The cooling control temperatures, capacities, and other specific details are listed for the particular systems in the sections that follow.

Phoenix, Arizona

Table 2-3 shows the characteristics of the systems presented for Phoenix, Arizona.

Figure 2-11 shows the percent of time in the comfort zone, and Figure 2-12 shows FOE savings and expenditures with respect to the baseline. Present value benefit or cost of the systems at nominal collector and energy costs are shown in Figure 2-13. Figures 2-14, 1-15, and 2-16 show present value benefit or cost sensitivity to energy escalation rates, initial energy costs, and collector costs, respectively, for several of the systems.

TABLE 2-3

PHOENIX, ARIZONA AIR CONDITIONING SYSTEMS

Desiccant Capacity tons	Vapor Compression Capacity, tons	Absorption Capacity, tons	Rankine Capacity, tons	Cooling Control Temperatures T_{vc}	T_{hi}	T_{lo}	T_{off}	Auxiliary Cooling Thermal Energy	Humidifier Bypass	Minimum Collector Area sq ft*
0.0	4.0	0.0	0.0	75	--	--	--	--	--	0
2.5	1.5	0.0	0.0	78	77	75	74	Yes	No	0
2.5	1.5	0.0	0.0	78	77	75	74	No	No	710
1.5	2.5	0.0	0.0	78	77	75	74	No	No	570
4.0	0.0	0.0	0.0	--	77	75	74	No	No	860
0.0	0.0	4.0	0.0	74**	--	--	--	No	No	1000
0.0	0.0	0.0	4.0	74**	--	--	--	No	No	0

*Based on minimum acceptable comfort level.
**Cooling system switched on and off at T_{vc}.

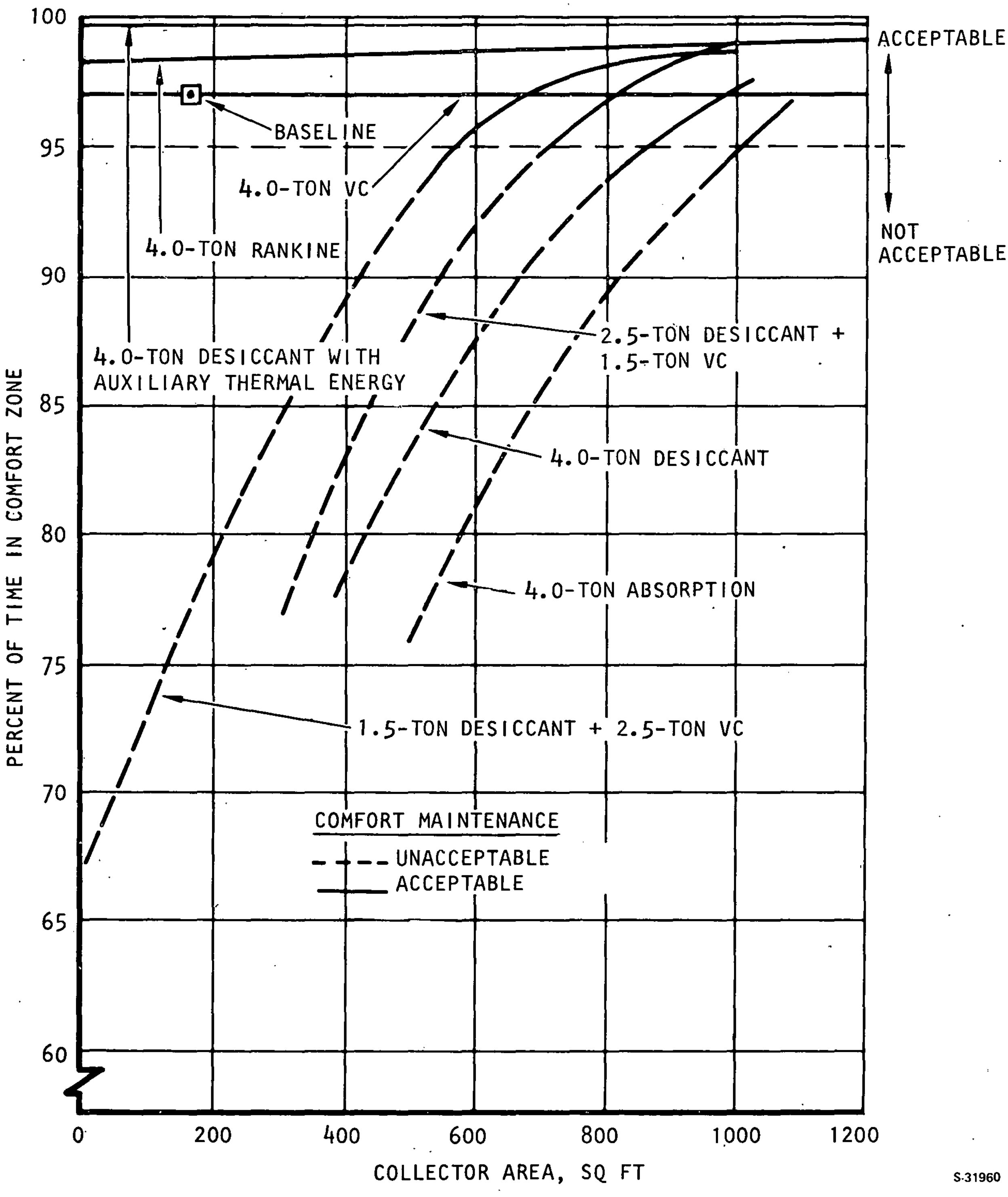


Figure 2-11. Percent Comfort for Phoenix Systems

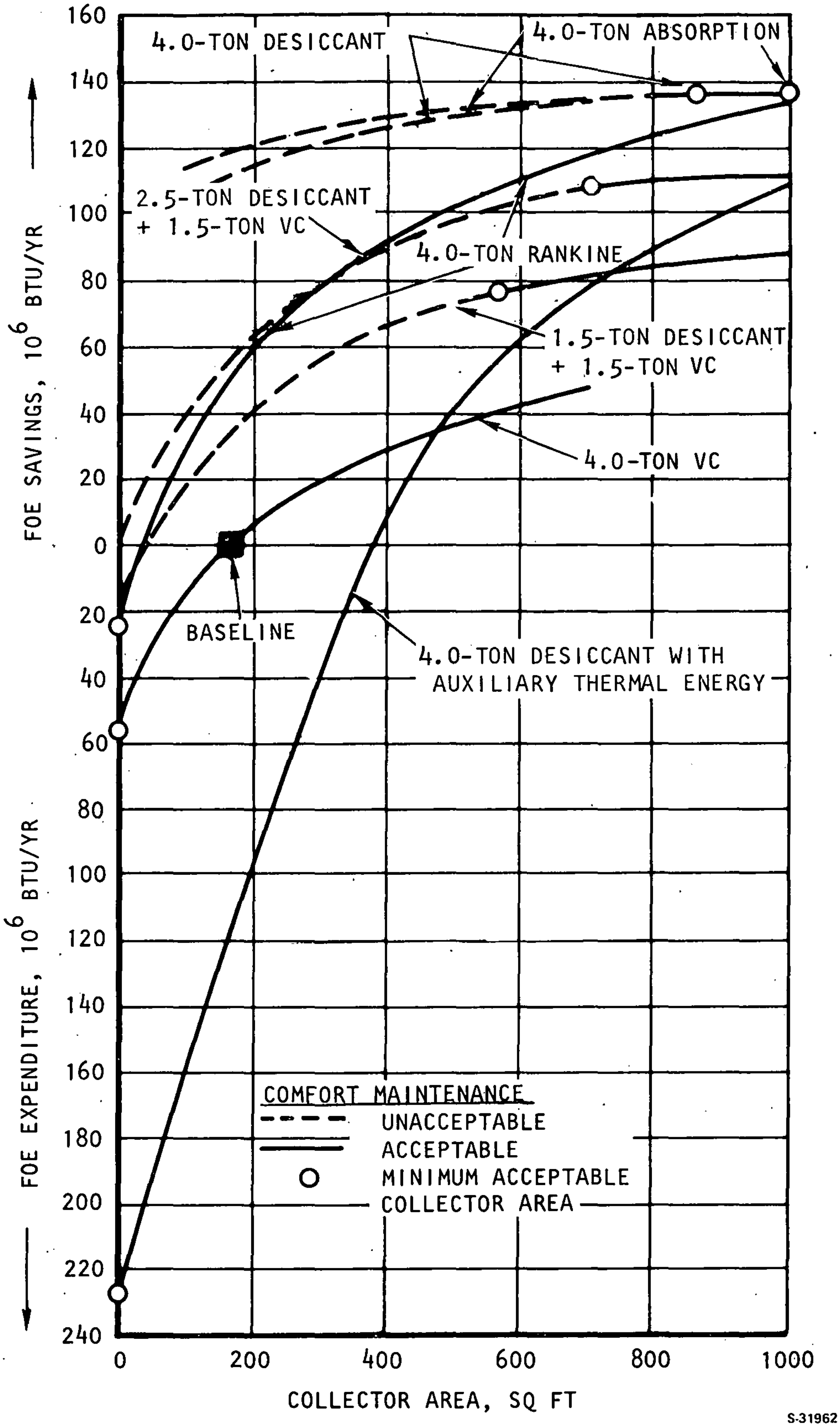
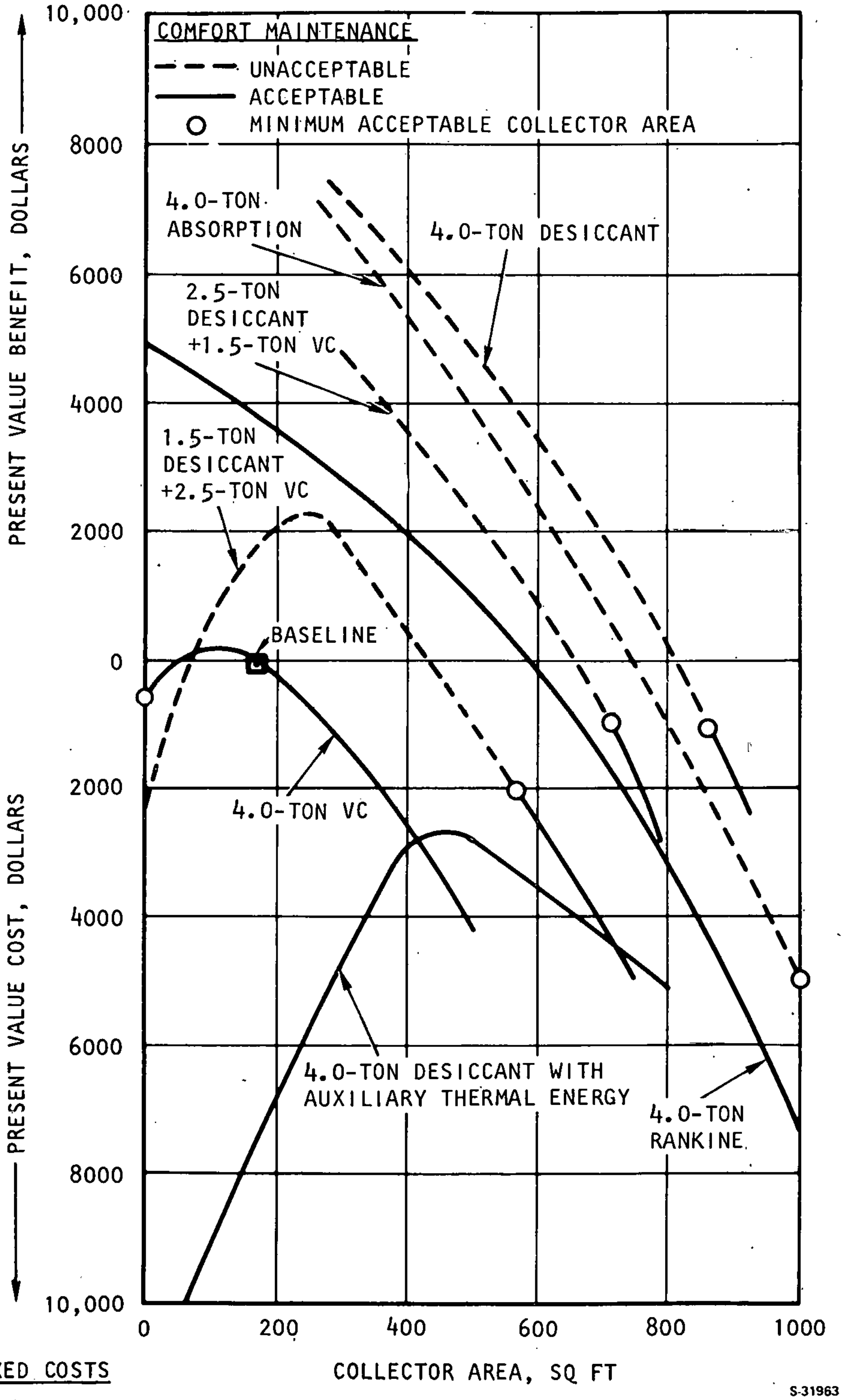


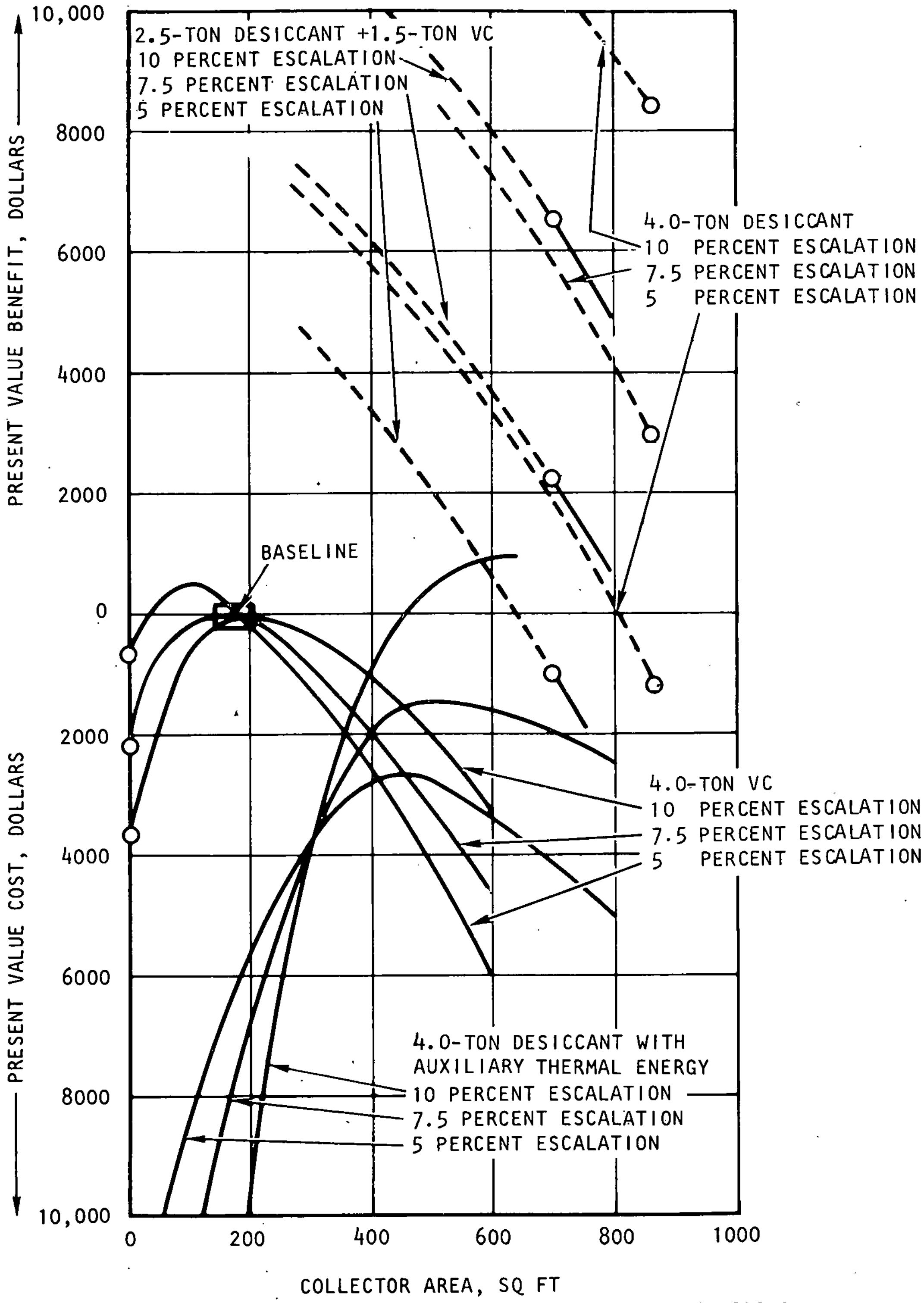
Figure 2-12. FOE Consumption of Phoenix Systems



FIXED COSTS

$20/SQ FT
$0.05/KWHR
$3.00/10^6 BTU
5 PERCENT ESCALATION

Figure 2-13. Present Value of Phoenix Systems at Nominal Energy and Collector Costs



COMFORT MAINTENANCE

- - - UNACCEPTABLE

—— ACCEPTABLE

O MINIMUM ACCEPTABLE COLLECTOR AREA

FIXED COSTS

$20/SQ FT

$0.05/KWHR

$3.00/10^6 BTU

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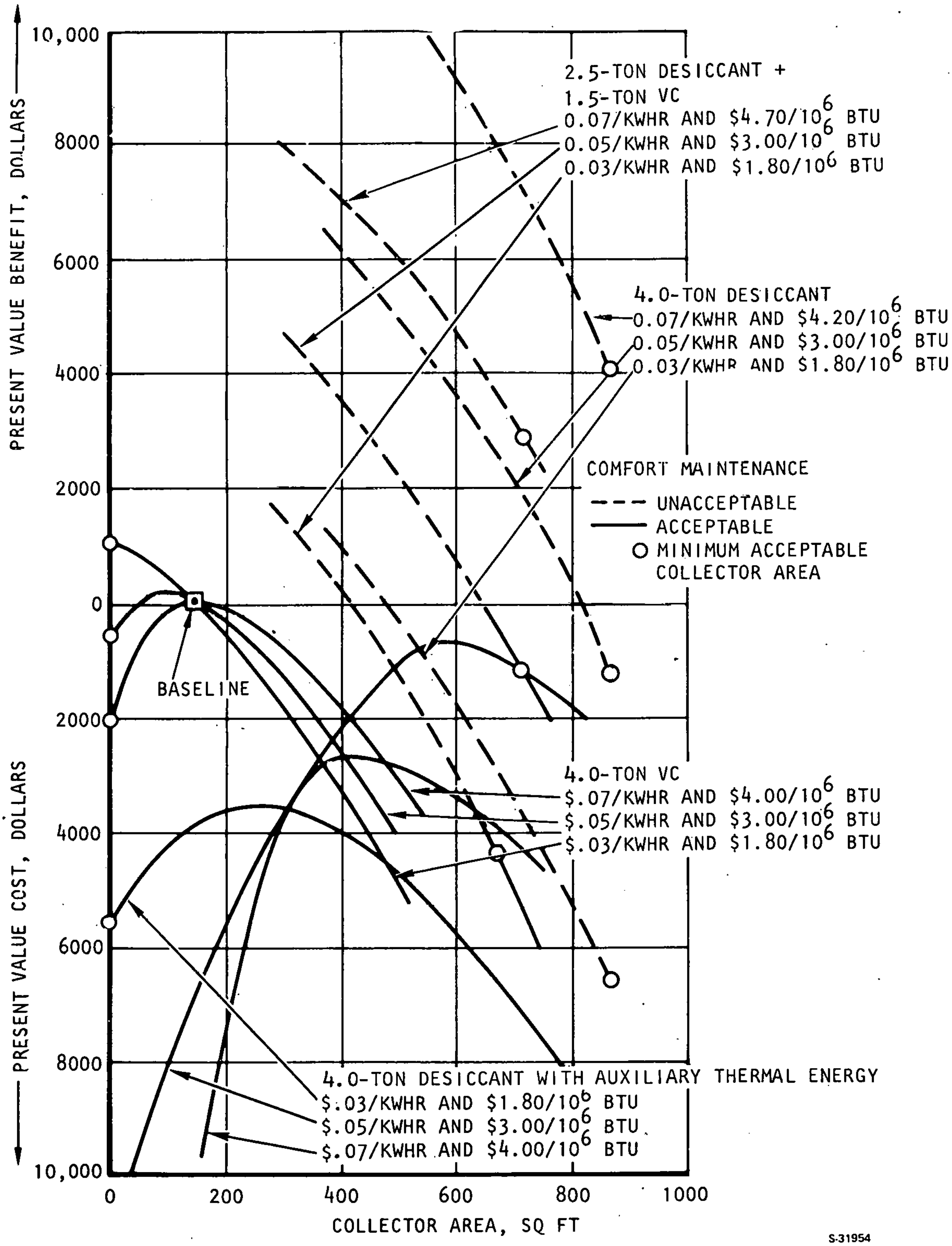
Figure 2-14. Present Value Sensitivity to Energy Escalation Rates for Phoenix Systems

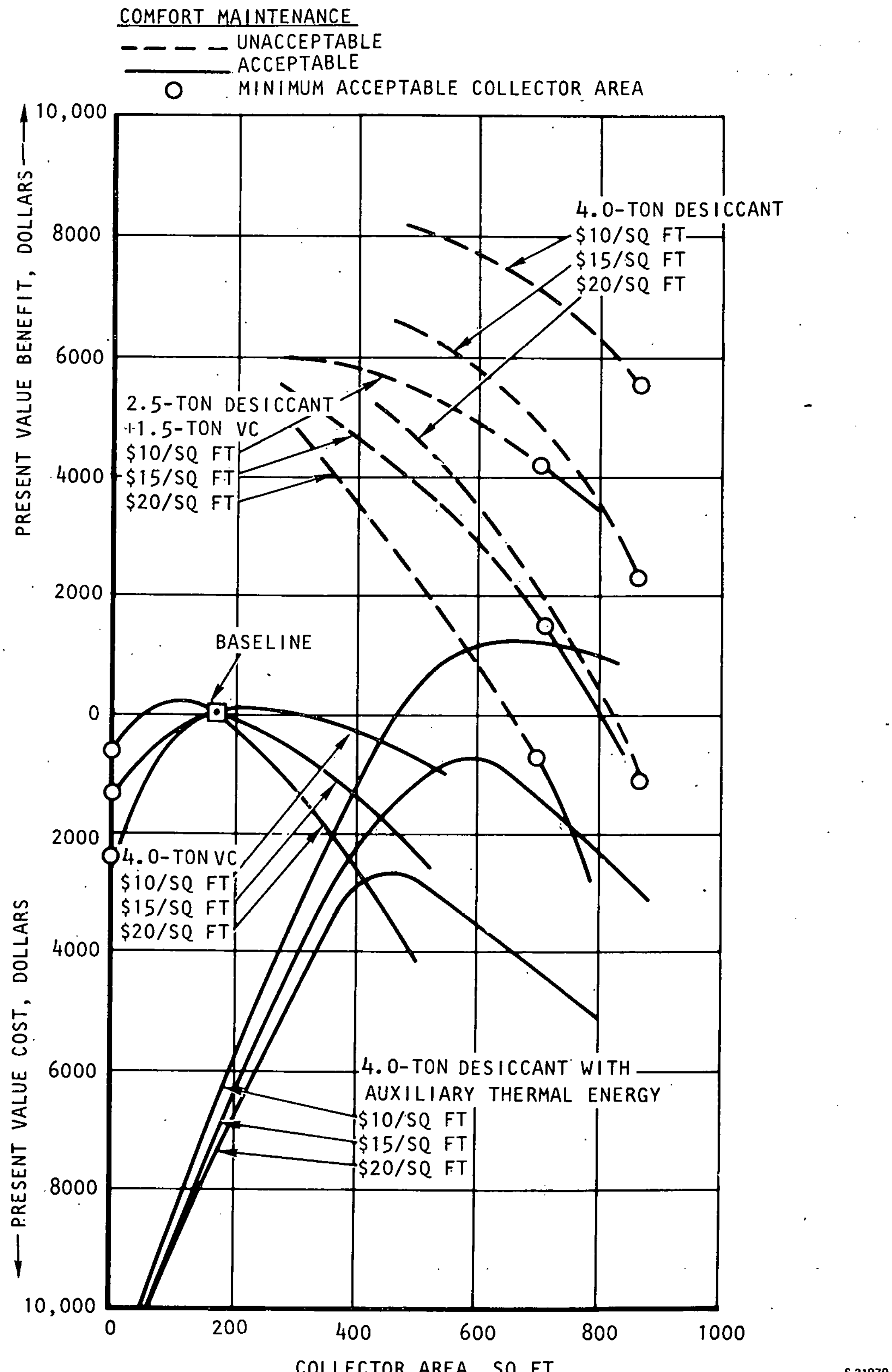


COMFORT MAINTENANCE
------ UNACCEPTABLE
——— ACCEPTABLE
O MINIMUM ACCEPTABLE COLLECTOR AREA

FIXED COSTS

$20/SQ FT
5 PERCENT ESCALATION

Figure 2-15. Present Value Sensitivity to Initial Energy Costs for Phoenix



FIXED COSTS

5 PERCENT ESCALATION
$0.05/KWHR
$3.00/10^6 BTU

Figure 2-16. Present Value Sensitivity to Collector Cost for Phoenix Systems

An important relationship in the case of Phoenix is the high sensible cooling load requirement coupled with the relatively small (175 sq ft) baseline collector area. Although there are moderately high winter heating requirements, there is adequate solar radiation in the winter to meet the heating load with a small collector. Therefore, the baseline collector is much too small to support a solar cooling system. The lowest collector area required to maintain comfort (with the 1.5-ton desiccant plus, 2.5-ton vapor compression system) needs an additional 395 sq ft of collector over the baseline requirements; the other solar systems require even more. The capital costs of these additions must be charged against the cooling system. The two nonhybrid nonaugmented systems, 4.0-ton desiccant and 4.0-ton absorption, require 860 and 1000 sq ft of collector, respectively, which are considerable increases (685 sq ft and 825 sq ft) over the baseline.

Fortunately, it is possible to save a considerable amount of FOE by installing these larger collectors. Figure 2-12, for example, shows that either the 4.0-ton desiccant or 4.0-ton absorption system can save about 135 x 10^6 Btu/yr FOE at their respective minimum collector areas. The magnitude of the savings is due to the large amount of electricity that is replaced by solar thermal energy. Neither the desiccant nor absorption systems benefit from an FOE savings standpoint by increasing their collector areas over the minimum comfort sizes, although comfort continues to improve somewhat. A desirable system may therefore have a somewhat larger collector, at the option (and expense) of the purchaser.

A significant advantage of the desiccant system over the absorption system is shown in Figures 2-11 and 2-12. In terms of FOE savings on an annual basis, the two systems are nearly identical, with the absorption system showing a slight advantage with small collector fields. However, in order to maintain the same level of comfort in the conditioned space, the absorption system requires about 16 percent more collector area than a desiccant system of the same capacity.

The two hybrid systems (2.5-ton desiccant plus 1.5-ton vapor compression, and 1.5-ton desiccant plus 2.5-ton vapor compression) are methods of obtaining acceptable comfort levels with smaller collector fields than required for either the purely desiccant or absorption solar systems. This reduces the fraction of the load carried by solar energy, and the FOE savings are reduced accordingly.

The 4.0-ton desiccant system with the addition of auxiliary thermal energy will, as shown in Figure 2-11, keep the conditioned space comfortable nearly all of the time, regardless of collector area. This is done at a large FOE expense, however, as shown in Figure 4-2. An absorption system with thermal augmentation was also investigated, with results that were quite similar to the augmented desiccant system.

The 4.0-ton Rankine system is another approach that will achieve comfortable conditions nearly all the time with any collector area, due to the electric motor augmentation in the cooling mode, as shown in Figure 2-11. FOE consumption at low collector areas is significantly better than a conventional 4.0-ton vapor compression air conditioner, as shown in Figure 2-12, but both absorption and desiccant systems are superior until the collector area is increased to over 1000 sq ft.

The economic picture shows some definite advantages for solar cooling, but also significant sensitivity to assumed energy and collector cost parameters. As shown in Figure 2-13, the major advantage of the desiccant system's smaller collector over that of the absorption system results in a difference in present value. About $4000 is saved using the desiccant system over the absorption system, primarily due to lowered collector expense.

As previously noted, the nominal escalation rate of five percent, electricity cost of $.05/kwhr, and fuel cost of $\$3.00/10^6$ Btu appear to be reasonable estimates. Unless energy becomes more expensive, it is difficult to show an economic advantage for the systems that also show an energy conservation advantage. Figures 2-14 and 2-15 show the behavior of the cost of several systems at different energy escalation rates and starting costs. Higher energy costs and escalation rates will result in significant cost improvements.

Less significant improvements in the cost picture are made by lowering of collector costs. As shown in Figure 2-16, a reduction in installed collector costs from $20/sq ft for $15/sq ft would improve the 4.0-ton desiccant system from about $1000 cost to a $2100 benefit. While this change is less dramatic than those effected by energy cost increases, this magnitude of change in collector costs is likely to occur as collectors move into mass production, and at least this improves the economic problem from a loss to a gain situation.

Figures 2-13, 2-14, 2-15, and 2-16 do not include the effects of any tax incentive programs. Arizona has enacted a variety of tax credit programs that allow exemptions from property tax and sales tax, and certain deductions from an individual's income taxes. The benefit of these programs would depend upon the taxpayer's income tax situation, and therefore cannot be generalized sufficiently to include in a study such as this. However, an income tax credit of $1000 is provided that is applicable to almost all installers of solar energy equipment. Since this represents a cash benefit in the first or second year of the life of the project, this credit would have the effect of shifting the cost axis down by $1000 on Figures 2-13, 2-14, 2-15, and 2-16. While this is not particularly significant in terms of the overall cost of the project, if combined with other applicable tax credits it could be important, especially since it would tend to reduce the capital expenditure required at the outset of the project.

Apalachicola, Florida

Table 2-4 shows the characteristics of the seven systems presented for Apalachicola, Florida. Figure 2-17 shows the percent of time in the comfort zone, and Figure 2-18 shows FOE savings and expenditures with respect to the baseline. Figure 2-19 shows present value benefit or cost of the systems at starting point collector and energy costs. Figures 2-20, 2-21, and 2-22 show present value benefit or cost sensitivity to energy escalation rates, initial energy costs, and collector costs, respectively, for several of the systems.

AIRESEARCH MANUFACTURING COMPANY OF CALIFORNIA

TABLE 2-4

APALACHICOLA, FLORIDA AIR CONDITIONING SYSTEMS

Desiccant Capacity, tons	Vapor Compression Capacity, tons	Absorption Capacity, tons	Rankine Capacity, tons	Cooling Control Temperatures T_{vc}	T_{hi}	T_{lo}	T_{off}	Auxiliary Cooling Thermal Energy	Humidifier Bypass	Minimum Collector Area, sq ft*
0.0	3.0	0.0	0.0	74	--	--	--	--	--	0
3.0	0.0	0.0	0.0	--	76	75	73	Yes	Yes	0
2.5	1.5	0.0	0.0	78	76	75	73	No	Yes	400
1.5	1.5	0.0	0.0	78	76	75	73	No	Yes	390
3.0	0.0	0.0	0.0	--	76	75	73	No	Yes	2450
0.0	0.0	3.0	0.0	73**	--	--	--	No	--	4460
0.0	0.0	0.0	3.0	73**	--	--	--	No	--	0

*Based on minimum acceptable comfort level.
**Cooling system switched on and off at T_{VC}.

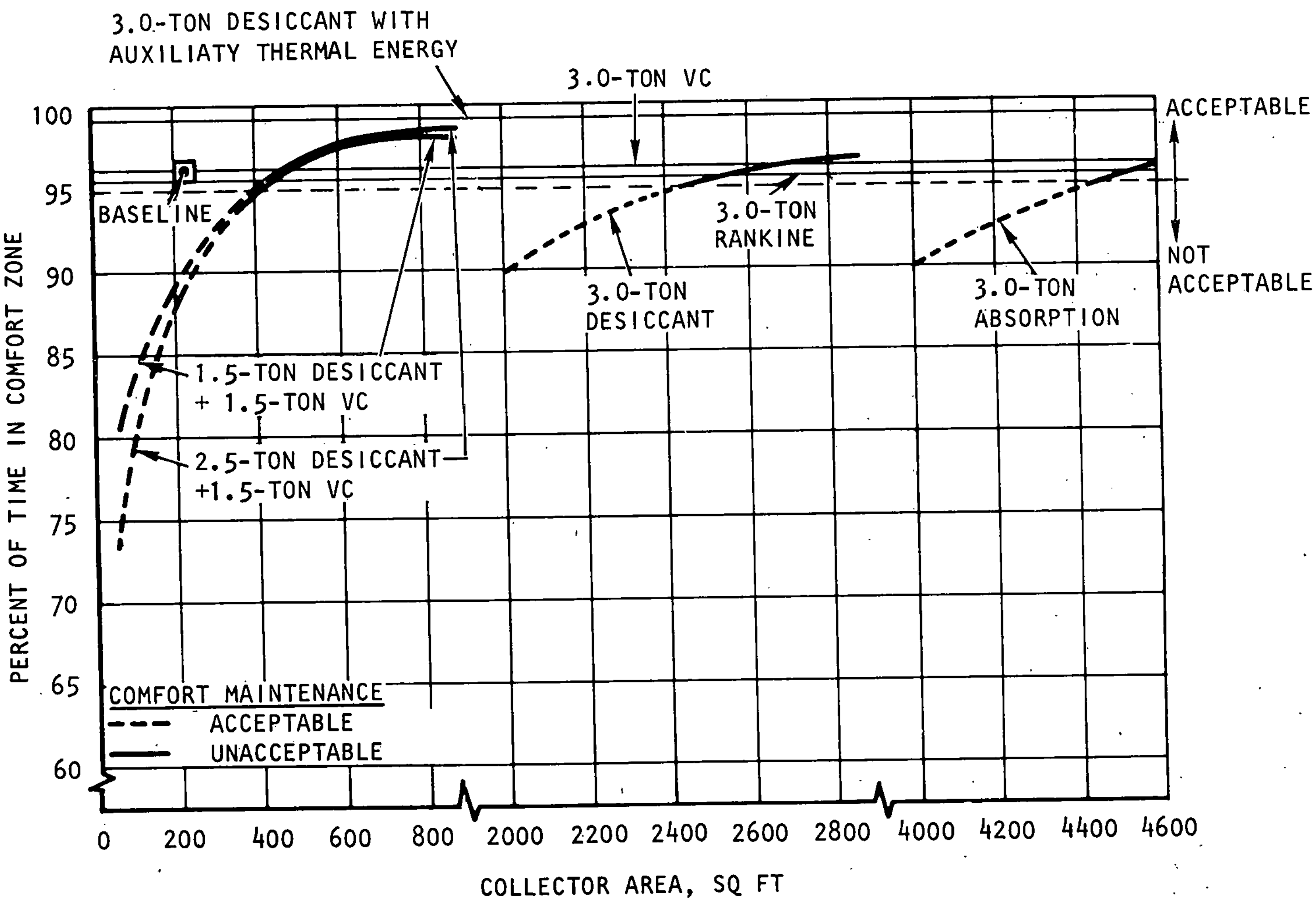


Figure 2-17. Percent Comfort for Apalachicola Systems

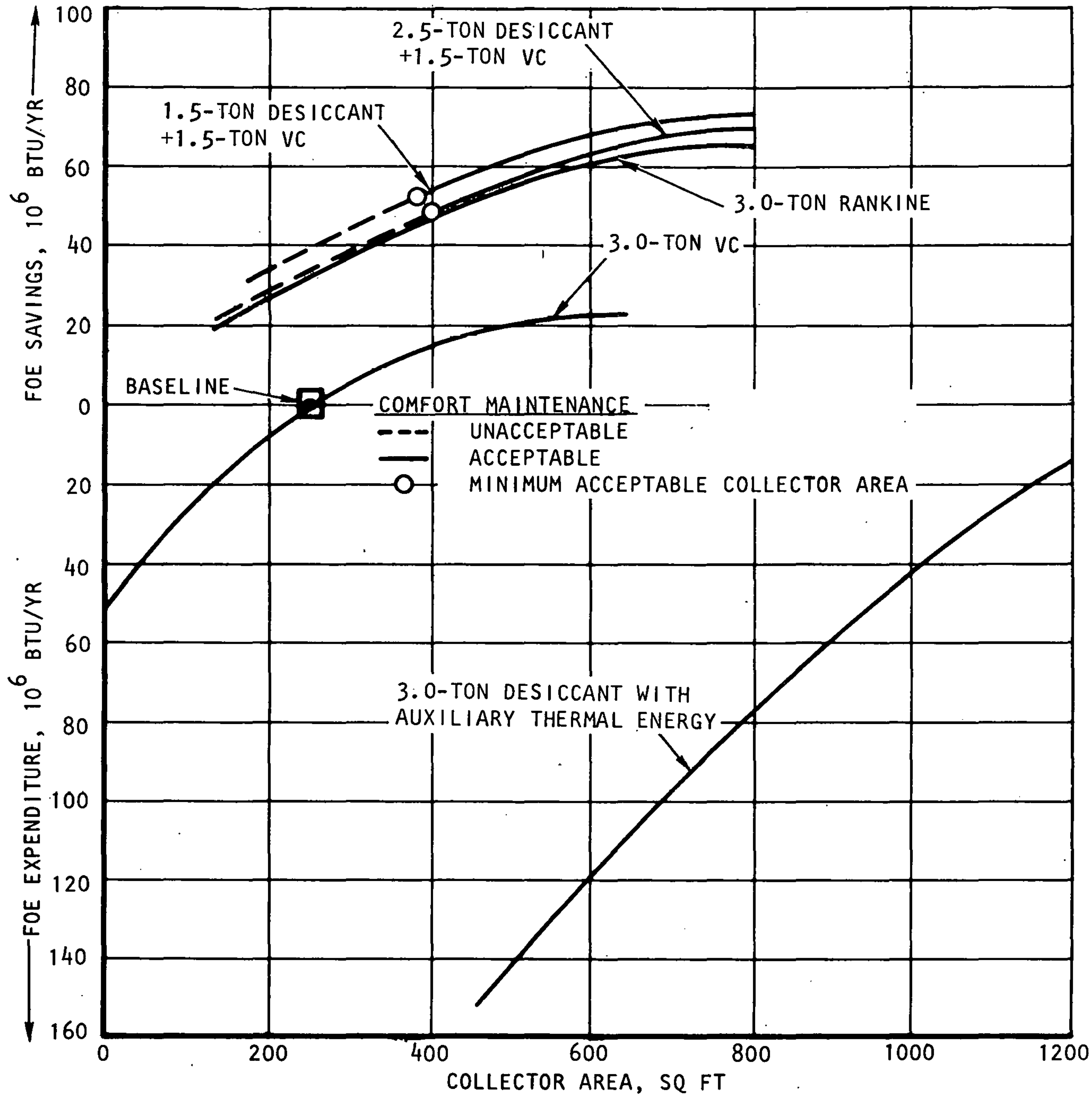
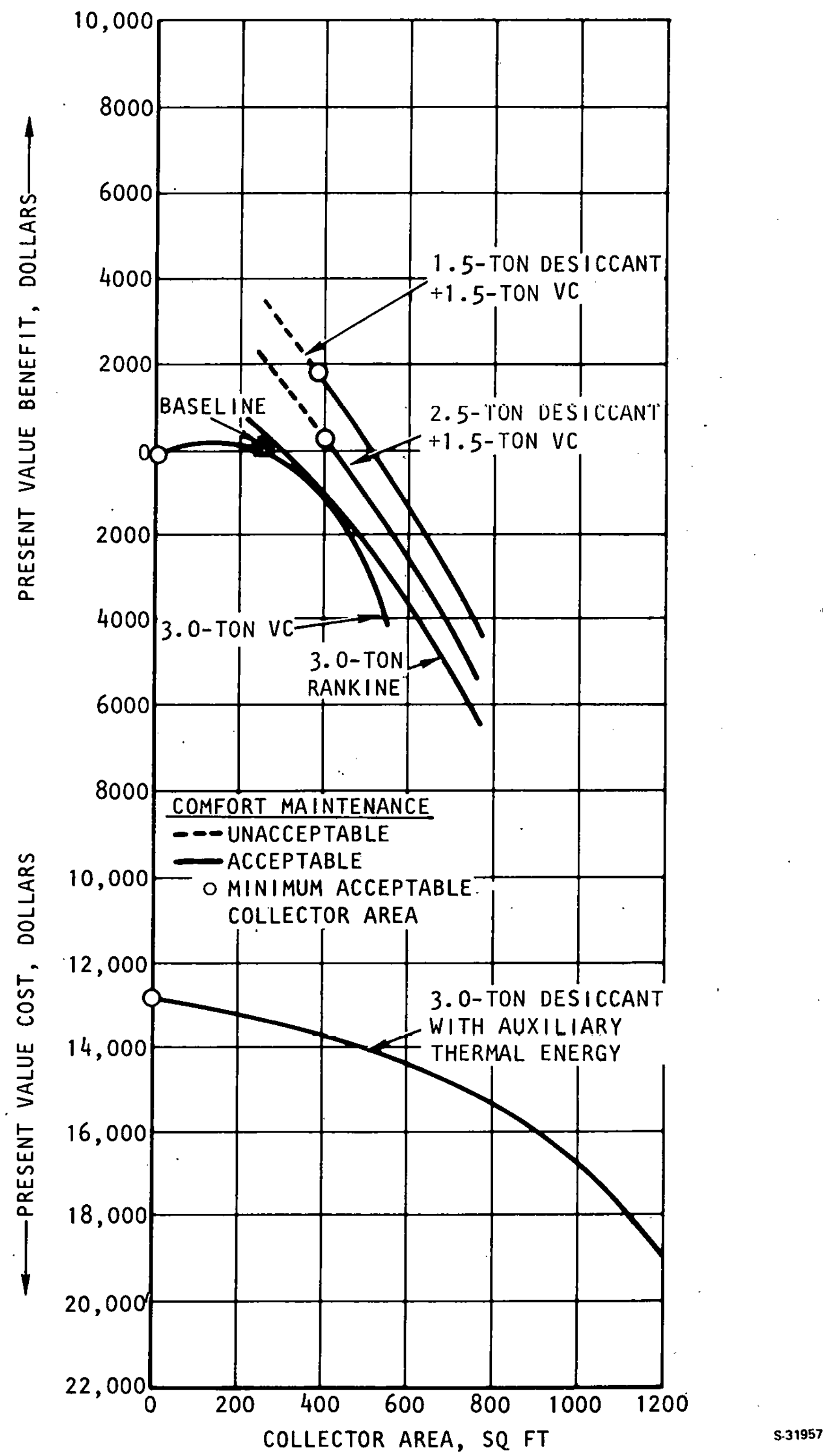


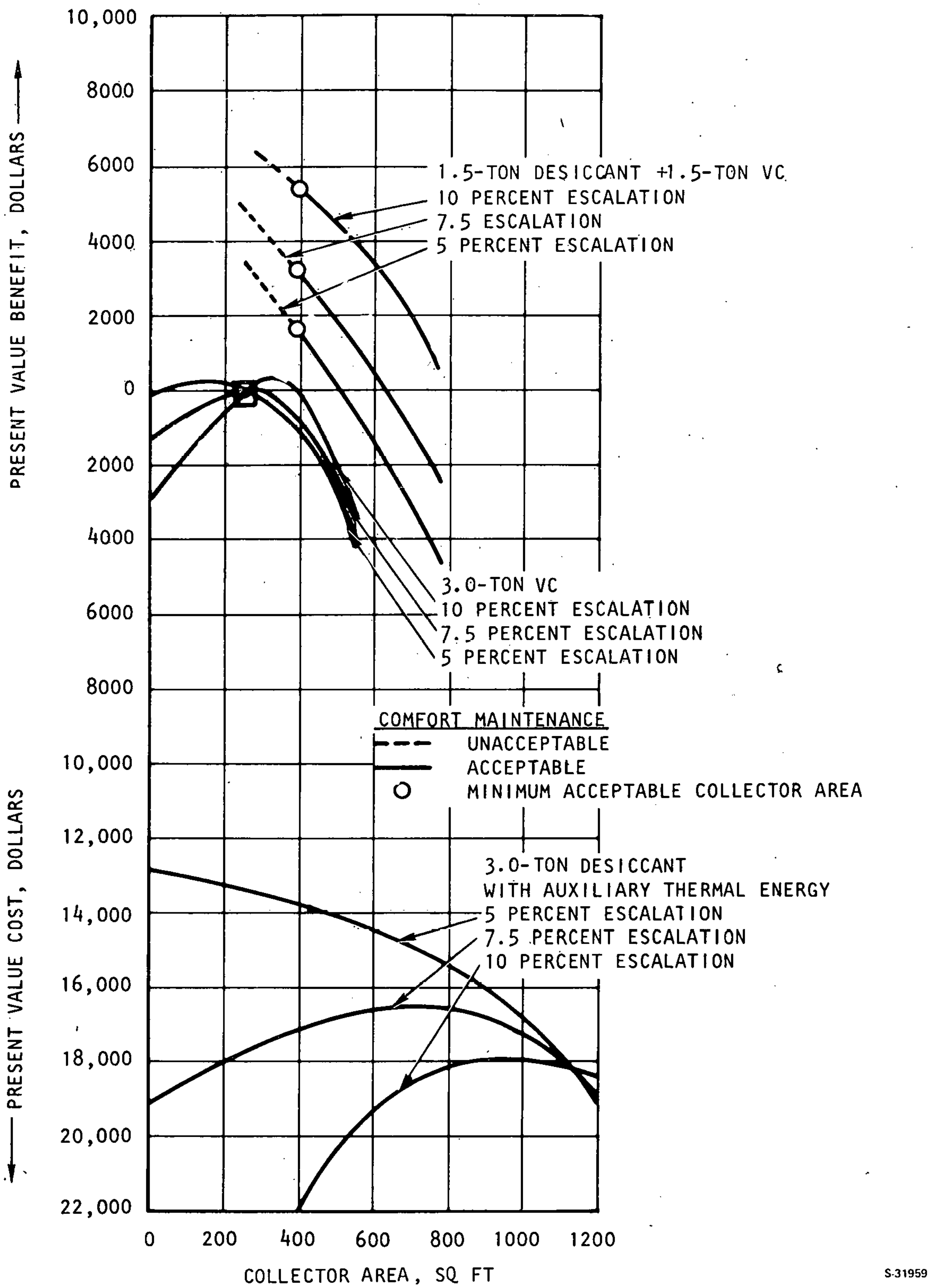
Figure 2-18. FOE Consumption of Apalachicola Systems



FIXED COSTS

$20/SQ FT
5 PERCENT ESCALATION
$0.05 KWHR
$3.00/10⁶ BTU

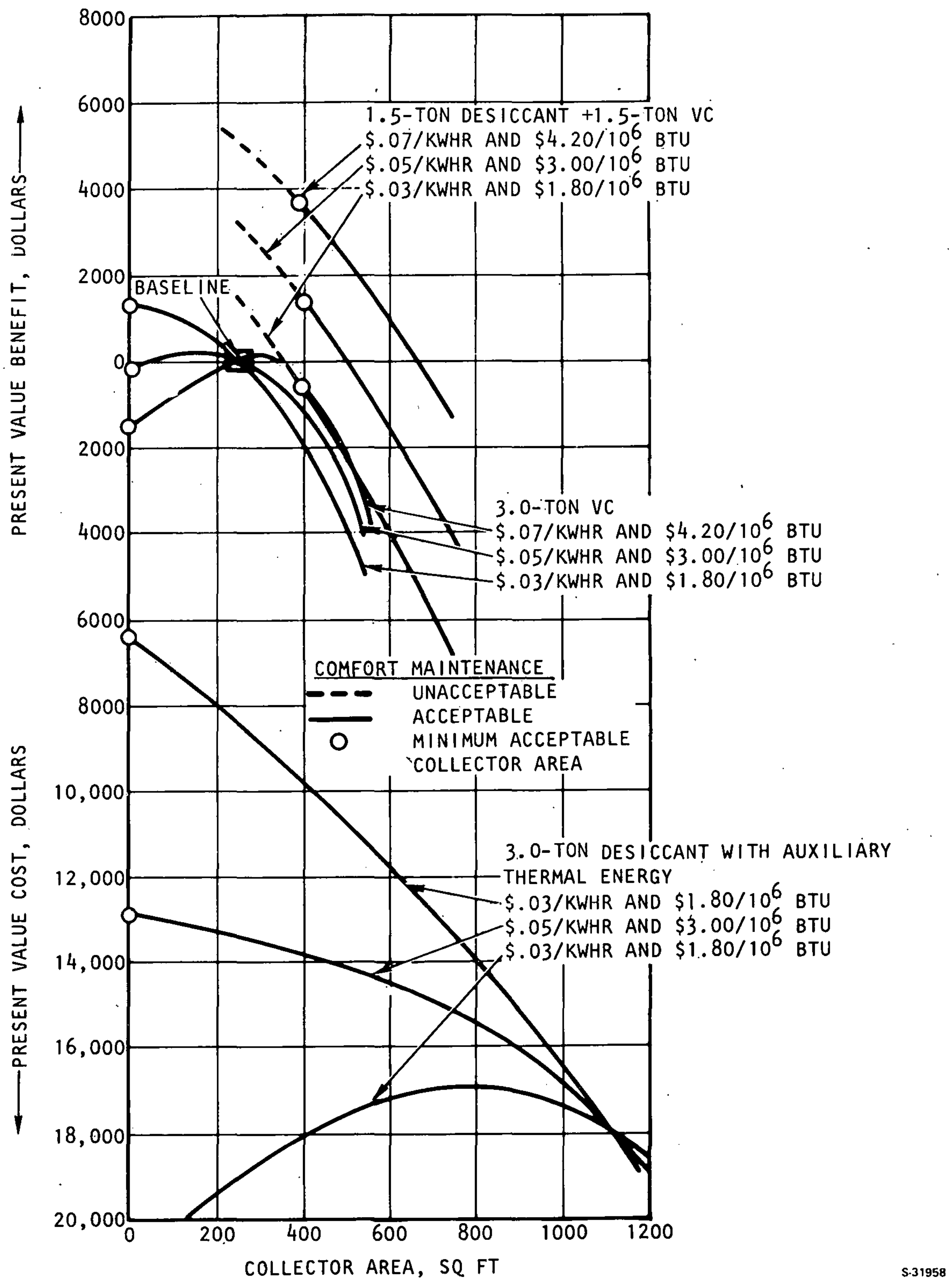
Figure 2-19. Present Value of Apalachicola Systems at Nominal Energy and Collector Costs



FIXED COSTS

$20/SQ FT
$0.05 KWHR
$\$3.00/10^6$ BTU

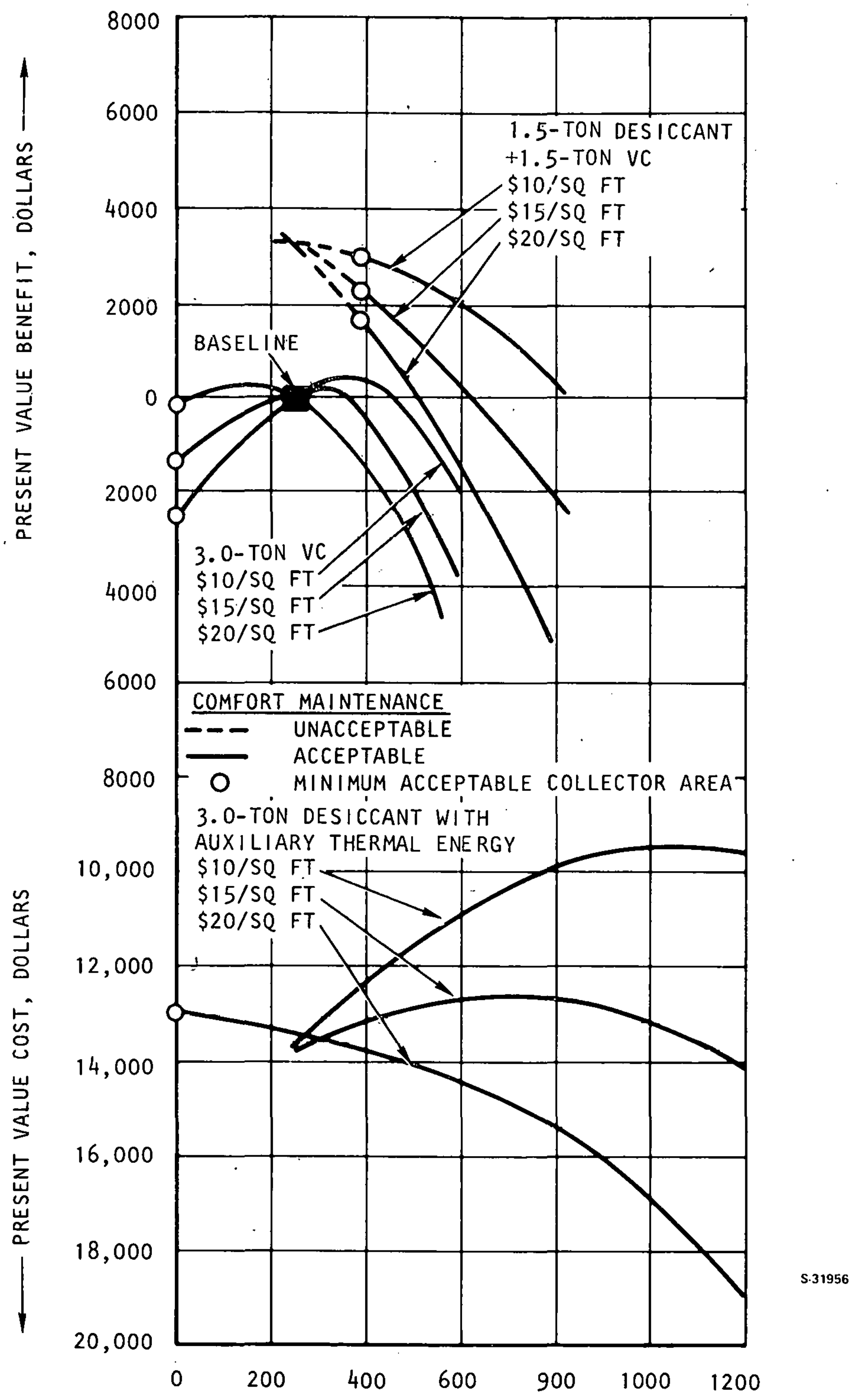
Figure 2-20. Present Value Sensitivity to Energy Escalation Rates for Apalachicola Systems



FIXED COSTS

$20/SQ FT
5 PERCENT ESCALATION

Figure 2-21. Present Value Sensitivity to Initial Energy Costs for Apalachicola System



FIXED COSTS

$0.5/KWHR
$2.00/10^6 BTU
5 PERCENT ESCALATION

Figure 2-22. Present Value Sensitivity to Collector Cost for Apalachicola Systems

Apalachicola has considerably lower space heating requirements than Phoenix, but due to the prevalent haze and cloud cover, much less solar energy is available for collection. Therefore, the baseline collector is somewhat larger (250 sq ft) than Phoenix (175 sq ft). This cloudiness has positive and negative effects with regard to the application of a solar cooling system. On the positive side, the high humidity yields a rather low sensible heat factor (about 54 percent, depending upon the system). In such high latent load applications, the desiccant system functions well.

On the other hand, a significant problem does arise on days when the cloud cover is present during most of the sunlight hours. If the energy storage tank temperature drops too low, the cooling capacity of the desiccant or absorption system decreases. Additionally, the COP of an absorption system decreases. This problem only can be solved with a pure solar cooling system by increasing the collector area so that more energy is collected. Figure 2-17 shows that the 3.0-ton desiccant and 3.0-ton absorption systems require 2450 sq ft and 4460 sq ft collector fields to achieve acceptable comfort. Of course, this is not practical.

The problem can be improved somewhat by enlarging the storage tank, so that cloudy period operation can be handled by stored energy from sunny periods. Figure 2-23 shows the change in comfort maintenance for the 3.0-ton desiccant system with 1000 sq ft of collectors, as the storage mass is increased from the standard 10 lb/sq ft to 70 lb/sq ft. Increases above 45 lb/sq ft do not result in an appreciable improvement. Even with a larger tank, the 3.0-ton desiccant system will require between 1500 and 2000 sq ft of collector. Combined with a 67,500- to 90,000-lb storage tank, this still does not yield a practical system. Therefore, nonaugmented solar cooling systems do not appear to be feasible in Apalachicola.

Fortunately, a simple hybrid system consisting of a vapor compression air conditioner and a desiccant system can effectively solve the cloudy day problem. Table 2-4 shows two such hybrids, a 2.5-ton desiccant plus 1.5-ton vapor compression, and a 1.5-ton desiccant plus 1.5-ton vapor compression. Despite its smaller capacity, the 1.5-ton desiccant system is capable of maintaining an acceptable level of comfort with a smaller collector area, as shown in Figure 2-17, and at larger FOE savings, as shown in Figure 2-18. This is an illustration of a possible effect of oversizing a system, a practice that is common in conventional air conditioning system design. Often a larger unit than is needed is selected to provide a safety factor, with resultant increased energy consumption and a decreased comfort level.

The 1.5-ton desiccant plus 1.5-ton vapor compression system provides significant FOE savings at its minimum comfort point collector area (390 sq ft). Since both FOE savings and comfort continue to improve steadily for larger collector areas, a somewhat larger collector could be justified. Perhaps 425 sq ft would provide a reasonable system.

The pure desiccant system with auxiliary thermal energy, as shown in Figure 2-17, is capable of maintaining a high level of comfort. However, this is done at a large FOE expenditure, as shown in Figure 2-18.

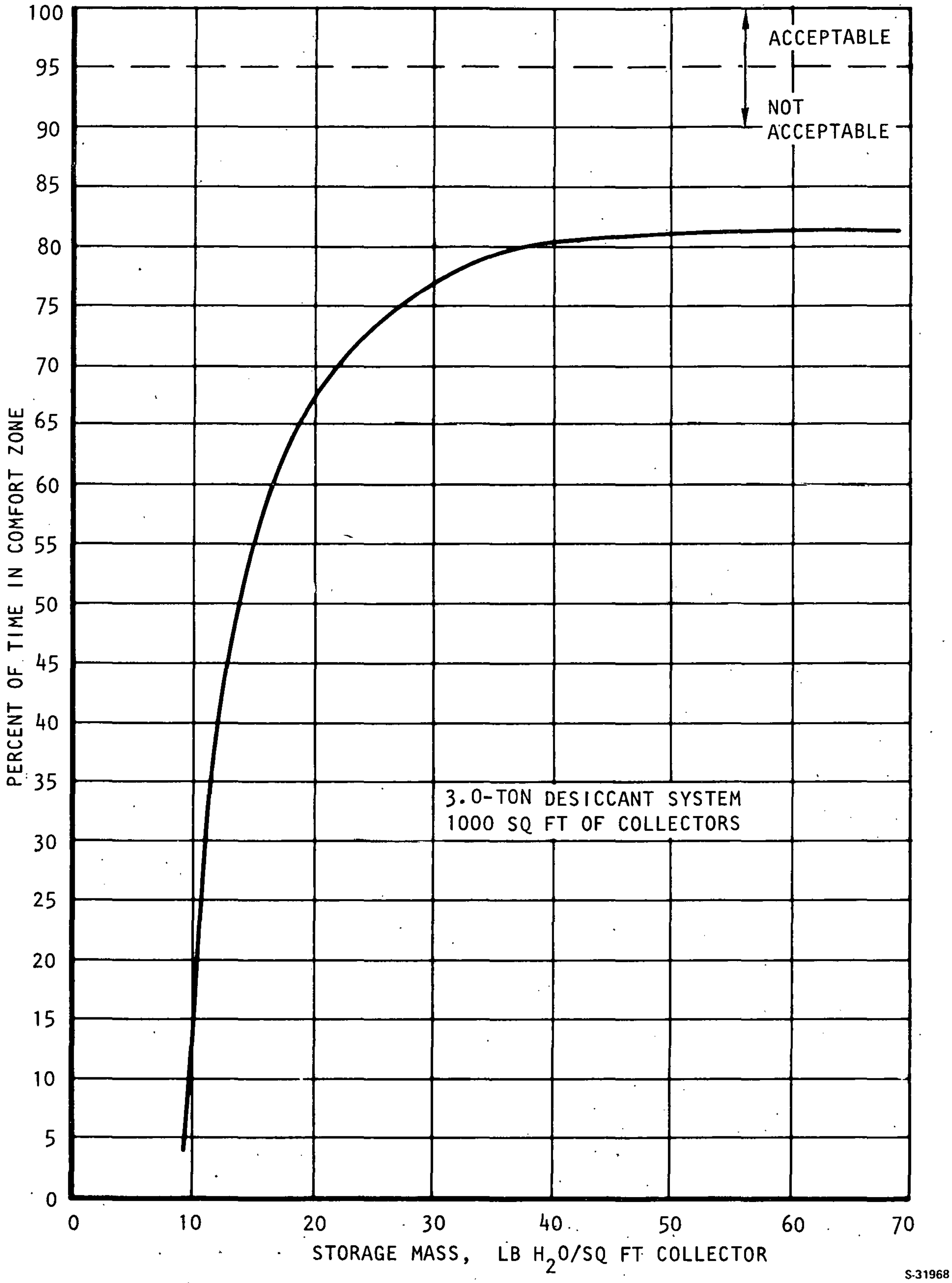


Figure 2-23. Comfort Maintenance Sensitivity to Storage Mass

The 3.0-ton Rankine system is capable of maintaining acceptable comfort levels regardless of collector area, as shown in Figure 2-17, due to the motor augmentation when there is insufficient solar thermal energy to drive the turbine. The FOE savings are slightly less than either of the two preferred desiccant/vapor compression hybrids, as shown in Figure 2-18.

The economic picture at nominal energy and collector costs favors the 1.5-ton desiccant plus 1.5-ton vapor compression hybrid. Figure 2-19 shows that the present value of the five systems shown is reasonable. Because of their very large collector requirements, the pure desiccant and absorption systems are not considered.

Figures 2-21, 2-22, and 2-23 show the sensitivity of several systems to variations in energy escalation rates, starting energy costs, and collector costs. Higher energy costs and lower collector costs yield significantly improved benefits.

The thermally augmented desiccant system performs particularly poorly in the economic comparison. The high cost is dominated by the very high energy consumption of this system. Even large collector areas at low ($10/sq ft) cost do not offset the high fuel bill, as shown in Figure 2-22.

At the present time, there are no economic incentives in effect in Florida for solar systems. However, the Florida legislature has discussed legislation of this type, and it is possible that programs could be enacted in the near future.

New York, New York

Table 2-5 shows the characteristics of the six systems presented for New York, New York. Figure 2-24 shows the percent of time in the comfort zone, and Figure 2-25 shows FOE savings and expenditures with respect to the baseline. Figures 2-26, 2-27, and 2-28 show present value benefit or cost for several energy escalation rates, initial energy costs, and collector costs, respectively.

New York has a very high space heating load, and therefore has a large baseline collector area (1100 sq ft). This is quite a benefit in the analysis of any solar cooling system, since the cooling system will not have to be charged with much, if any, additional collector area. For example, the 1.5-ton desiccant plus 1.5-ton vapor compression and the 1.0-ton desiccant systems both achieve acceptable comfort levels with collector areas less than the baseline system. This is the only one of the three cities investigated where any solar cooling system showed this benefit.

While the 1.5-ton desiccant plus 1.5-ton vapor compression and 1.0-ton desiccant systems have virtually identical performance in terms of maintaining comfort, the 1.0-ton desiccant system effects a slightly greater FOE savings. This gives the pure desiccant system a clear advantage over the hybrid in terms of reduced complexity. Therefore, New York appears to be a good application for the standard desiccant system.

AIRESEARCH MANUFACTURING COMPANY OF CALIFORNIA

TABLE 2-5

NEW YORK, NEW YORK AIR CONDITIONING SYSTEMS

Desiccant Capacity, tons	Vapor Compression Capacity, tons	Absorption Capacity, tons	Rankine Capacity, tons	Cooling Control Temperatures T_{vc}	T_{hi}	T_{lo}	T_{off}	Auxiliary Cooling Thermal Energy	Humidifier Bypass	Minimum Collector Area, sq ft*
0.0	1.0	0.0	0.0	73	--	--	--	--	--	0
1.0	0.0	0.0	0.0	--	74	73	72	Yes	Yes	0
1.5	1.5	0.0	0.0	75	74	73	72	No	Yes	1000
1.0	0.0	0.0	0.0	78	74	73	72	No	Yes	1000
0.0	0.0	1.0	0.0	72**	--	--	--	--	--	1150
0.0	0.0	0.0	1.0	72**	--	--	--	--	--	0

*Based on minimum acceptable comfort level.
**Cooling system switched on and off at T_{vc}.

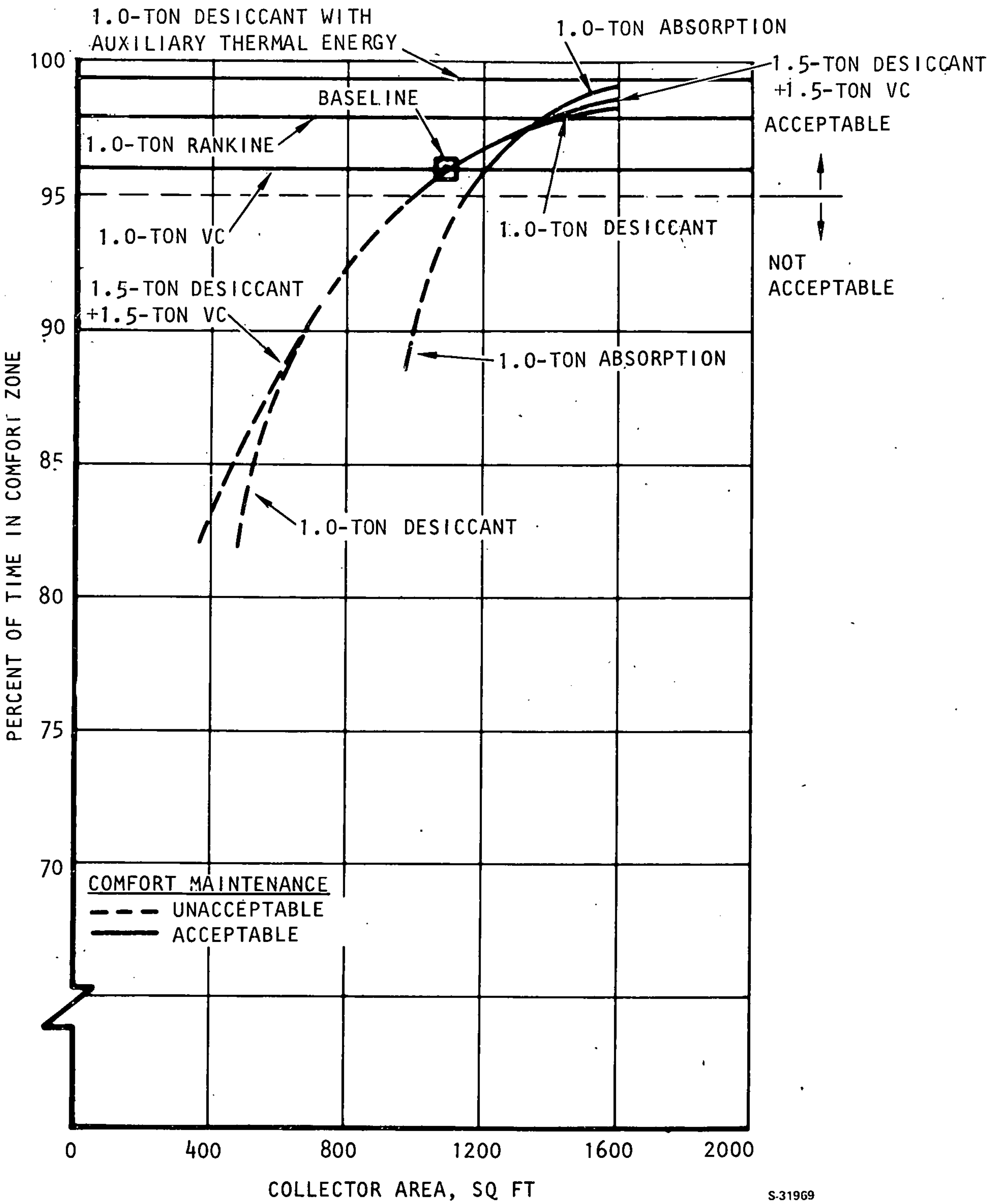


Figure 2-24. Percent Comfort for New York Systems

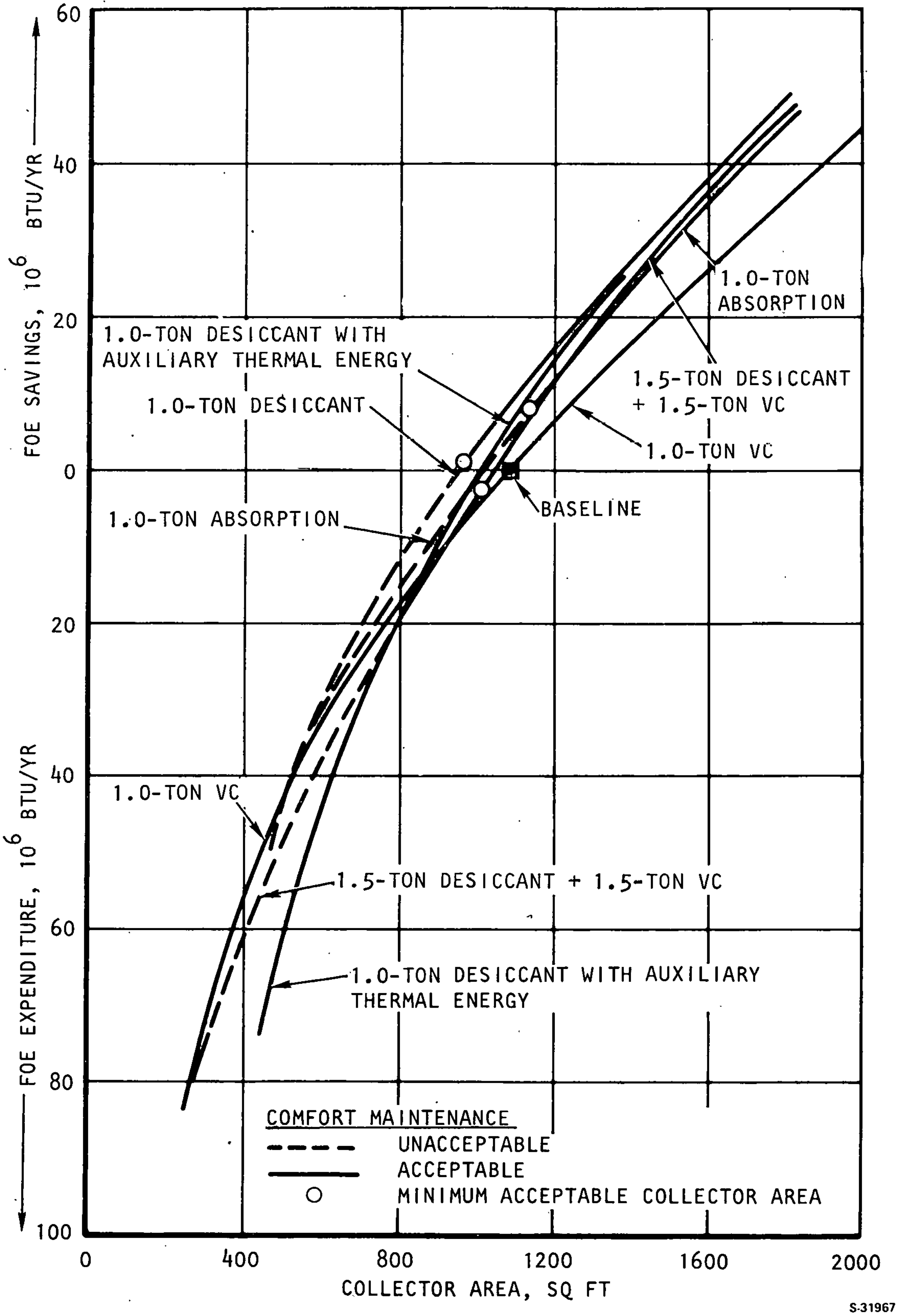
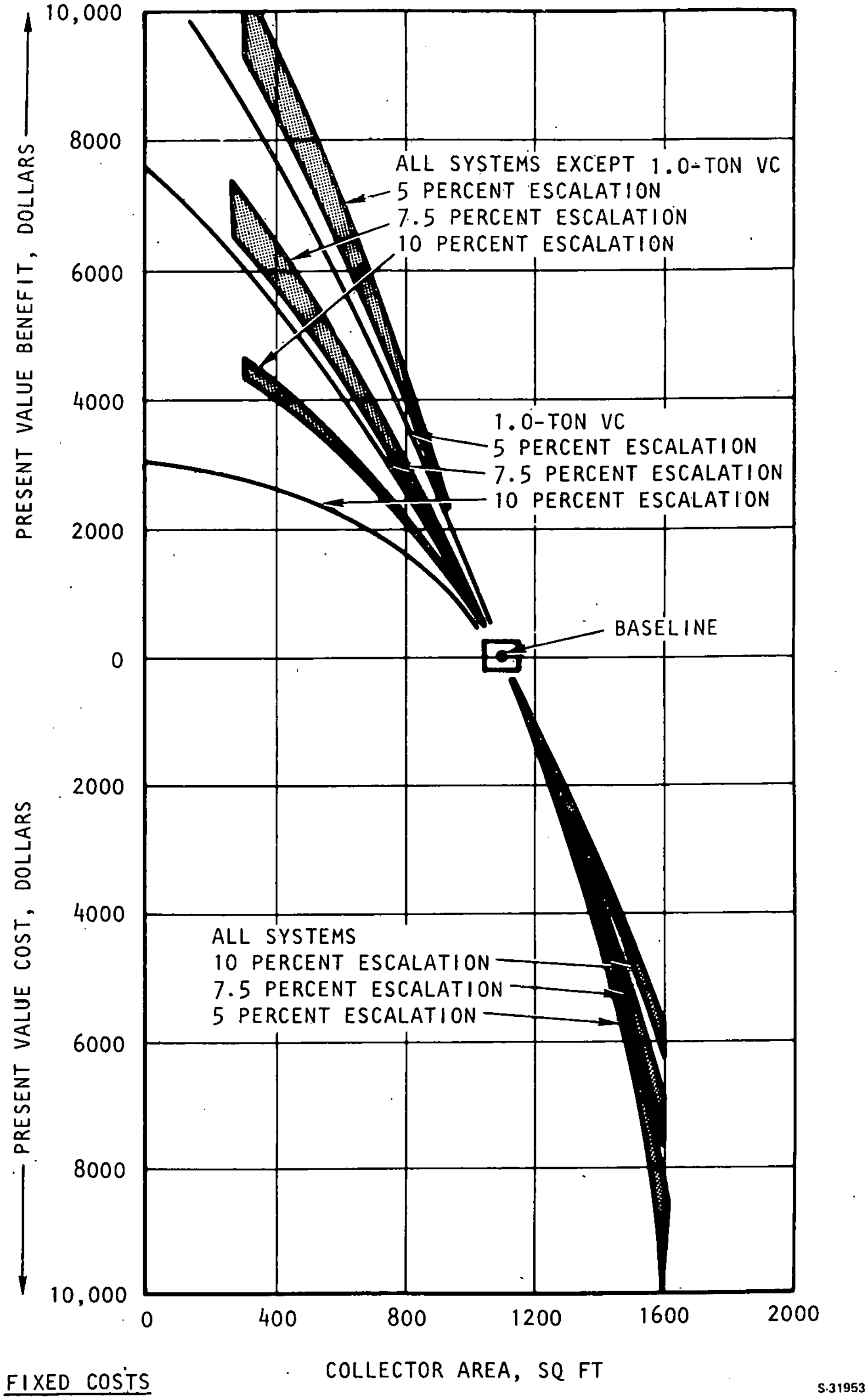


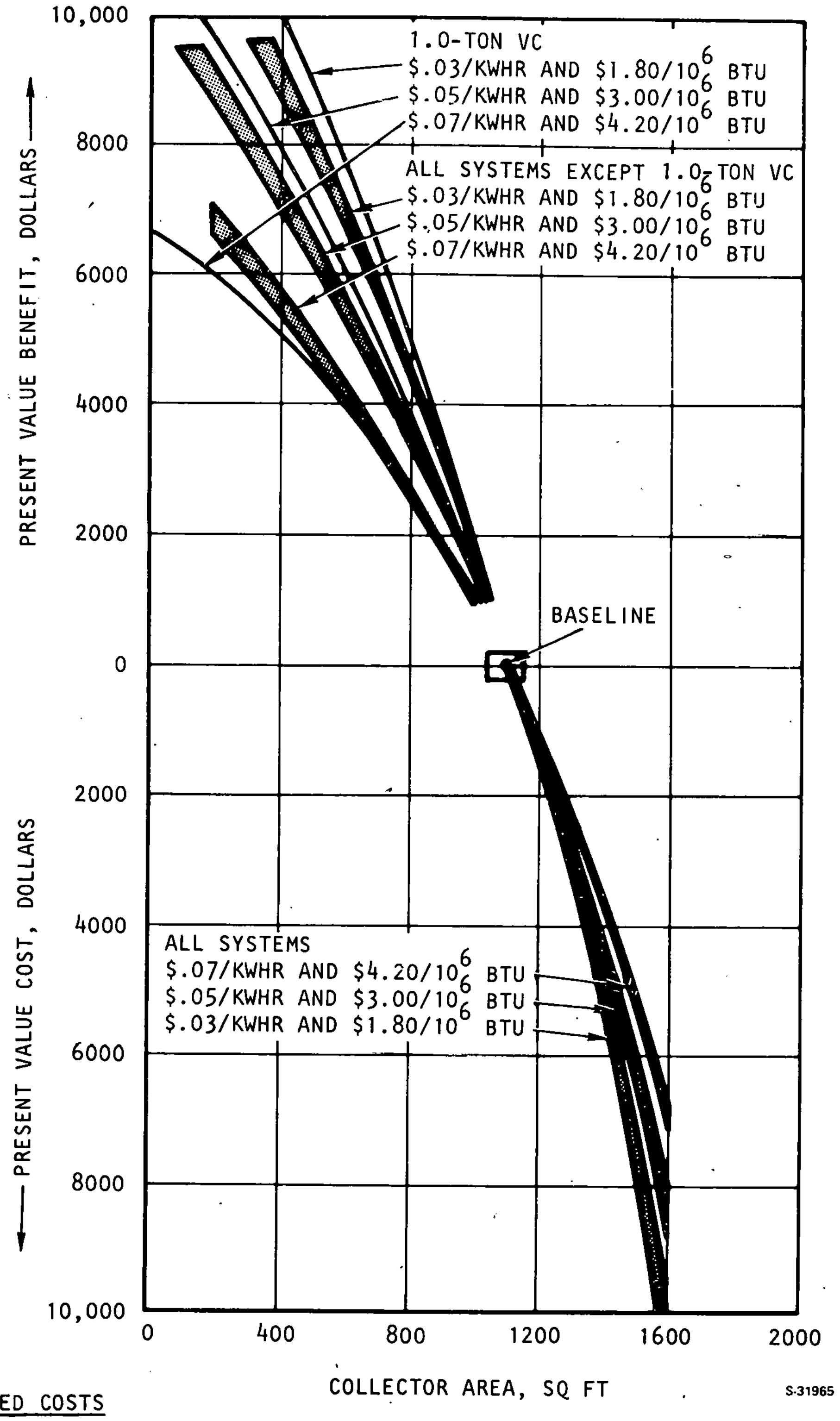
Figure 2-25. FOE Consumption of New York Systems



FIXED COSTS

$20/SQ FT
$0.05/KWHR
$3.00/10⁶ BTU

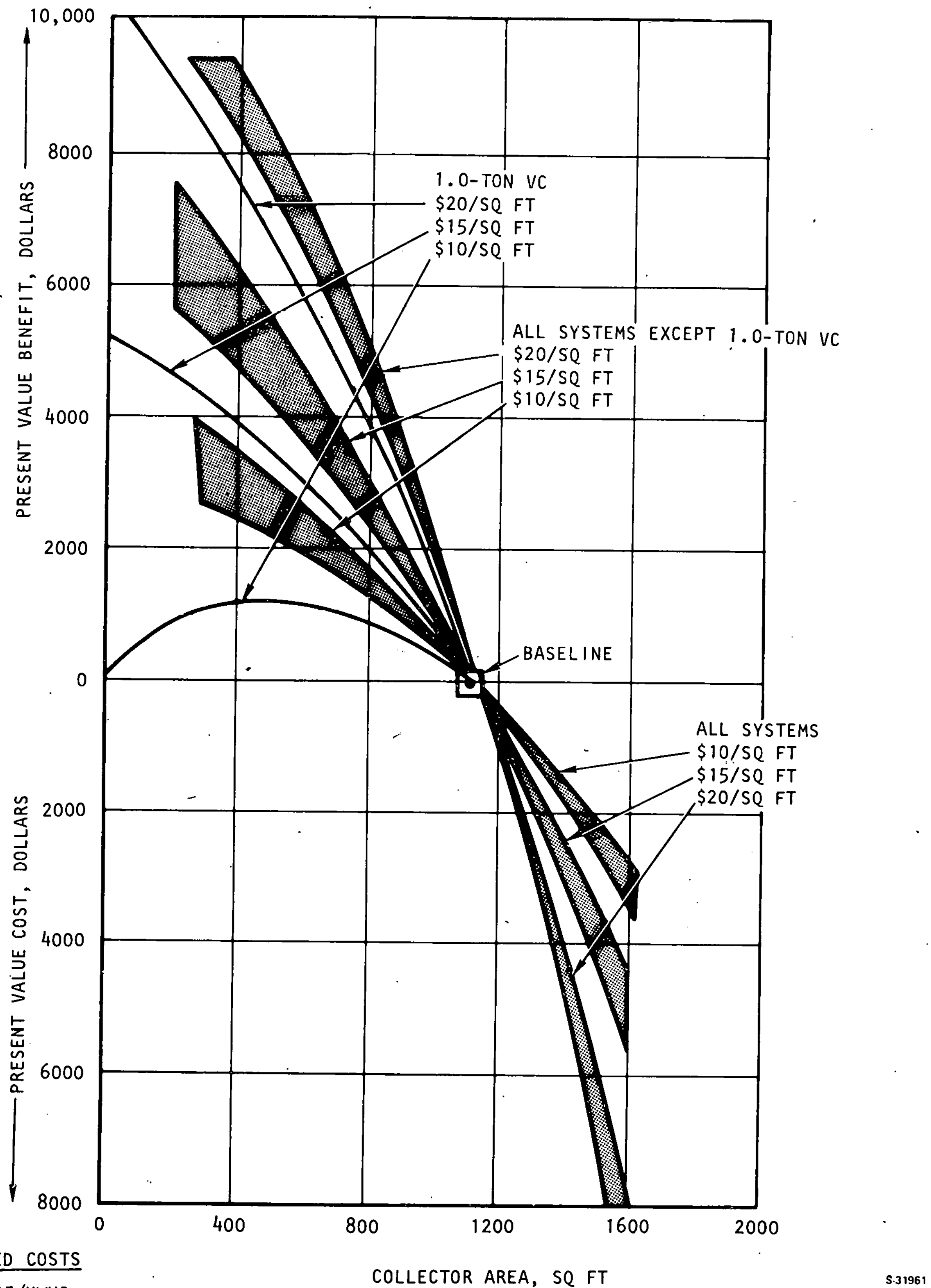
Figure 2-26. Present Value Sensitivity to Energy Escalation Rates for New York Systems



FIXED COSTS

$20/SQ FT
5 PERCENT ESCALATION

Figure 2-27. Present Value Sensitivity to Initial Energy Costs for New York Systems



FIXED COSTS

$0.05/KWHR
$3.00/10^6 BTU
5 PERCENT ESCALATION

Figure 2-28. Present Value Sensitivity to Collector Cost for New York Systems

The absorption system requires a small amount of extra collector over the baseline. As in Phoenix and Apalachicola, this is due to the sharp cutoff in COP at low inlet water temperatures. The Rankine system will maintain a high comfort level regardless of collector area by means of motor augmentation.

Although some differences exist in the FOE savings shown in Figure 2-25 all the systems, including the 1.0-ton vapor compression, have about the same behavior as the collector area is changed. This is because the large majority of the solar thermal energy demand is for water and space heating; the air conditioning power that is displaced by the solar energy is rather small. This relationship shows up again in Figures 2-26 and 2-27, which show that the price of the entire system is heavily dominated by the cost of the collectors; the lines for all the systems are virtually coincident on these three figures. However, Figure 2-25 shows how significant changes in collector costs may be. The thermally augmented desiccant system does not fare so badly in the FOE and economic comparisons in New York as did its counterparts in Phoenix and Apalachicola. This is because the air conditioning load is relatively small, and the baseline collector is large.

Due to the small incremental FOE savings and large present value costs associated with increased collector sizes, a reasonable system for New York would be the 1.0-ton desiccant system with the baseline collector field (1100 sq ft). In essence, this adds solar air conditioning to the baseline system at virtually no economic cost, and with a small savings in energy.

It is important to note, however, that although New York would be a good location for solar cooling when added to a solar heating system, it is difficult to make a valid economic argument for the basic solar heating system. Figures 2-26, 2-27, and 2-28 show that present value benefits with respect to the baseline are obtained by reducing the collector area to zero. Of course, this indicates that solar projects will, in general, not be too successful in New York City.

REFERENCES

1. Gunderson, M.E., K.C. Hwang, and S.M. Railing, Development of a Solar Desiccant Dehumidifier; Vol. 1, Summary; Vol. 2, Detailed Technical Information, AiResearch Technical Progress Report 78-14957-1 and 78-14957-2, March 31, 1978.

2. Rousseau, J., and K.C. Hwang, Preliminary Design of a Solar Desiccant Air Conditioner, AiResearch Technical Paper 78-14939, March 3, 1978.

SECTION 3

SEAL DEVELOPMENT

BACKGROUND

This section describes the seal development work done as part of Task II-3 (Subsystem Design). Early in the program, seals were identified as a potential problem area. Excessive seal leakage and friction would reduce system performance to unacceptable levels. Although the pressure differential across the seals is estimated to be below 1.0 in. H_2O, the system requires about 60 linear ft of seals to contain the various process airstreams. Under these conditions, careful attention must be paid to the design of the system seals. Three different types of seals are used in the system: (1) linear seals along the vertical wall of the desiccant and regenerator drums, (2) circular seals at the top and bottom surfaces of the drums, and (3) radial seals on the circular surface at the bottom of the regenerator drums.

To take advantage of existing technology, Mr. Stephen Fitch, President of Bry-Air, was engaged to act as a consultant on the program. Figure 3-1 shows a typical seal successfully used by Bry-Air on commercial desiccant systems. The seal consists of a stationary silicone rubber flap dragging on the rotary surface. This seal's life is reported to be in excess of five years of continuous operation. It is used by Bry-Air for all three types of seals defined above. The use of this seal concept on the AiResearch dehumidifier would result in a drum drive power requirement of about 0.75 kw. This value was estimated using torque data measured on Bry-Air desiccant wheels.

In an effort to reduce seal friction, an investigation was initiated involving alternate seal concepts and materials which would possess satisfactory leakage and friction characteristics while offering the potential for low production cost.

SEAL SELECTION

A number of seal configurations were examined including (1) labyrinth type seals, (2) roller type seals, (3) wiper seals, and (4) friction seals pressurized with extruded rubber sections. Because of the difficulty in maintaining a high dimensional accuracy on the desiccant and regenerator surfaces, the seal design must be able to accommodate run-out during operation. For this reason, the wiper type seal was selected. Other types of seals designed to allow for 1/8-in. movement of the drums would require complex mounting and/or result in relatively high pressure on the drum with attendant high friction and wear rate. The two basic seal configurations selected are depicted in Figures 3-2 and 3-3.

The linear seal is similar to that used by Bry-Air. It uses the same silicone rubber which acts to pressurize a Dacron felt pad on the sealing surface. The silicone rubber maintains adequate pressure on the seal with the air pressure in either direction relative to the direction of rotation.

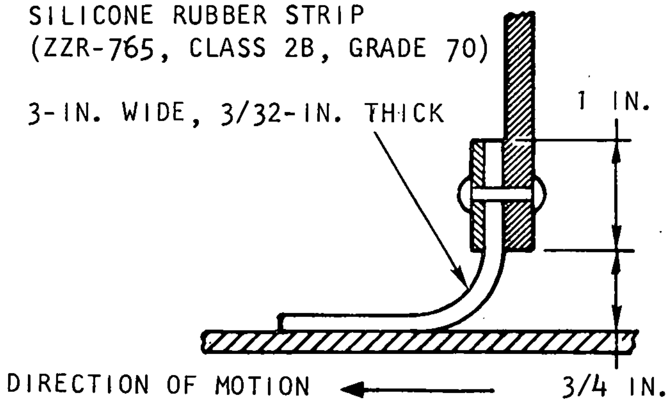


Figure 3-1. Typical Bry-Air Seal

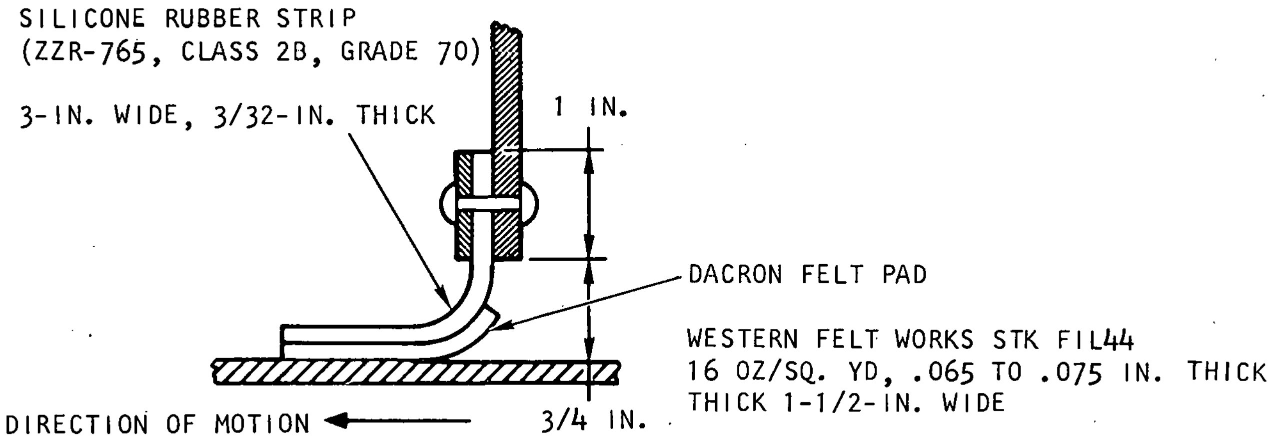


Figure 3-2. Linear Seal Configuration

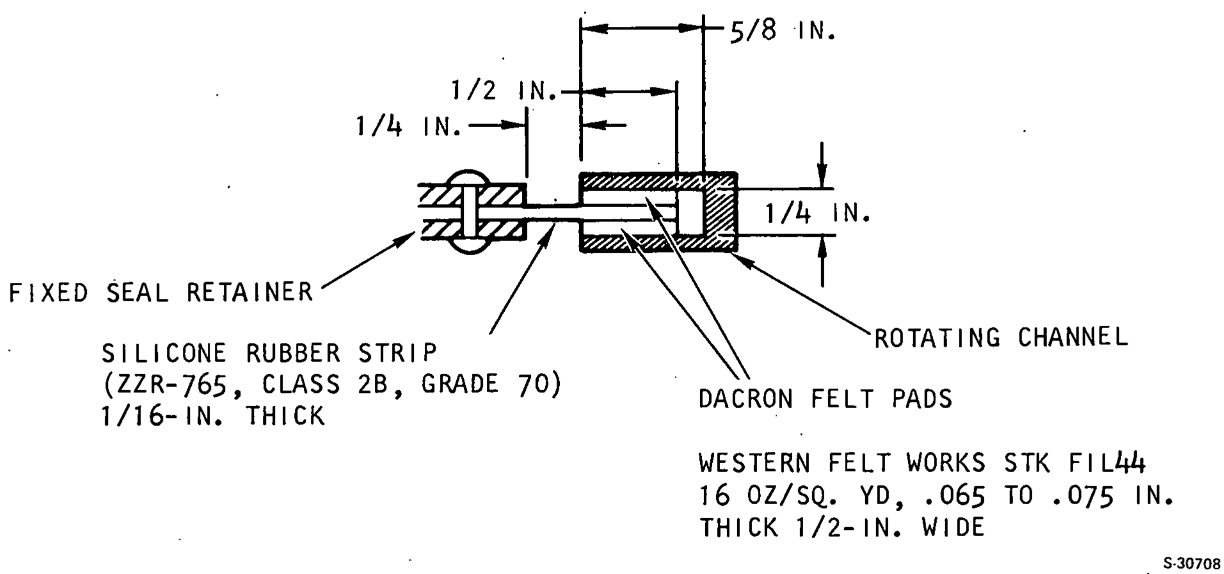


Figure 3-3. Circular Seal Configuration

For circular seals as shown in Figure 3-3, a thin strip of relatively soft silicone rubber is used with a Dacron felt pad on each side providing the sealing surface. The seal fits into a circular channel which rotates with the drum. The pressure across the seals changes direction as the drums rotate around their axis. The soft rubber is used to permit easy seating of the seal on either surface. This configuration also results in a smaller package in comparison to the design shown in Figure 3-2.

The selection of Dacron felt as the sealing surface was made as a result of a search for a material which would provide minimum leakage, minimum friction, and long life under operating conditions. Materials considered included (1) Teflon felt, (2) thin Teflon sheets, (3) Dacron felts of various densities (4) thin Celcon (Celanese) sheets, and (5) Teflon pile.

The Teflon materials were rejected because of their high cost. Preliminary tests on thin sheet materials showed high leakage in comparison to the felt materials. Also, a bonding problem between the silicone rubber and the pad may exist for the configurations selected. Scratching when handling thin sheet materials indicated that these materials may not have desirable wear characteristics. As a result, Dacron felt was selected as the pad material. Leakage and friction characteristics of this material (in the configurations shown in Figures 3-2 and 3-3) were determined.

SEAL TESTS

Leakage Tests

Two test rigs were designed and assembled to determine the leakage characteristics of the configurations selected. Figures 3-4 and 3-5 are photographs of the test rigs with the covers removed to show the seals under test. The ends of the seal strips were sealed at the interface with the test rig to eliminate end effects. Initial testing revealed that leakage through the ends overshadowed any leakage through the sealing surface under the Dacron felt pads. The lengths of the seal test section were 12 in. and 24 in. for the circular and linear seals, respectively.

Leakage test data are plotted in Figure 3-6. For the linear seal, two series of tests were conducted corresponding to conditions in which the air pressure on the silicone rubber is in the direction of rotation, or vice-versa.

Considering the very low pressure differential across the seals (see Figure 3-7), it is estimated that the maximum air leak through any one seal will be on the order of 0.5 cfm. Estimated leakage data are shown in Figure 3-8. The data show that seal leakage is insignificant in comparison to the process airflow streams listed below:

Return air from residence:	830 cfm
Humidified outside air:	830 cfm
Preconditioning air from outside:	120 cfm

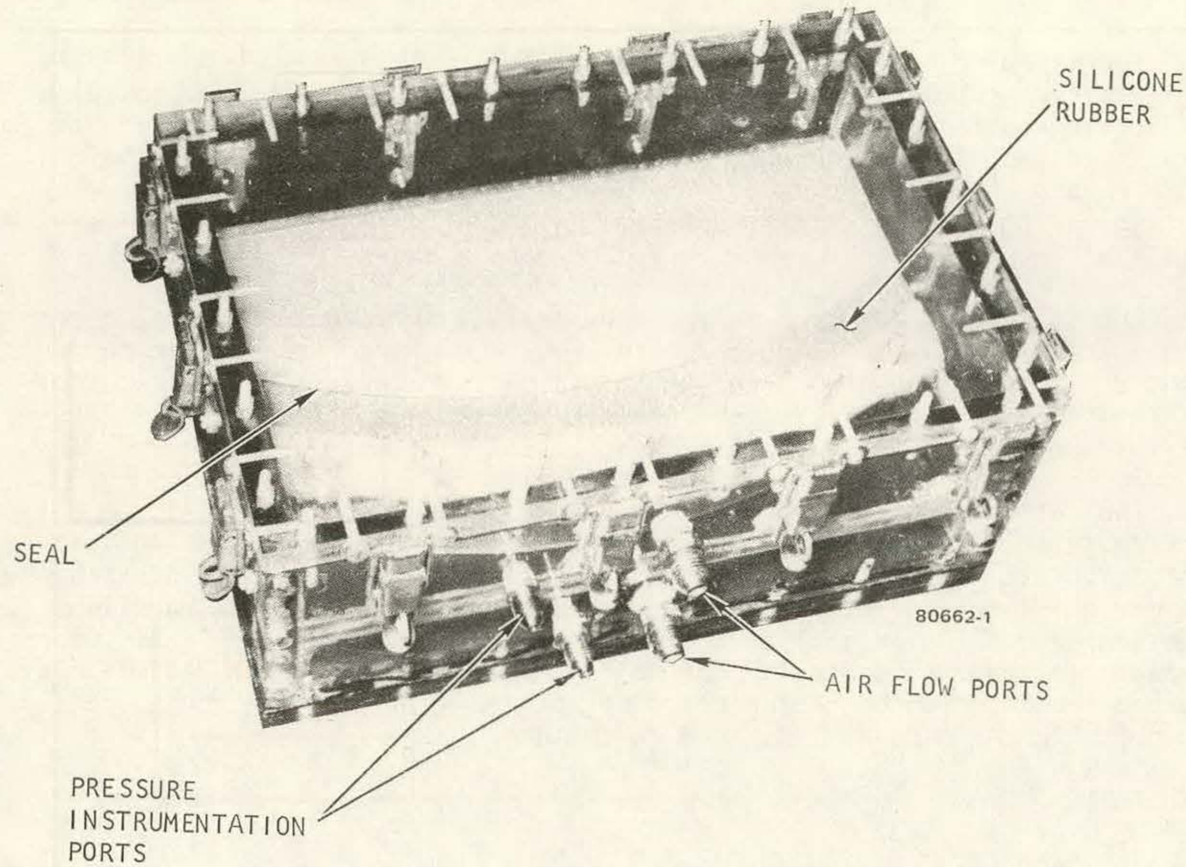


Figure 3-4. Circular Seal Test Rig

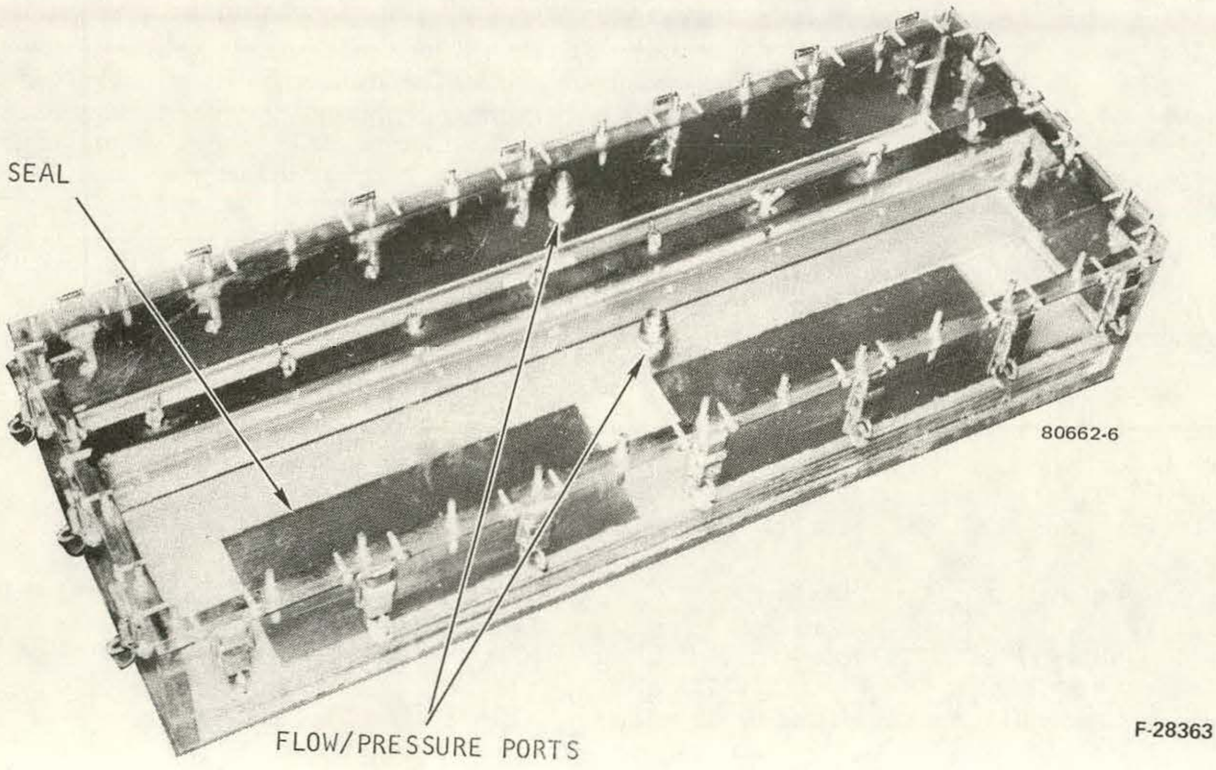


Figure 3-5. Linear Seal Test Rig

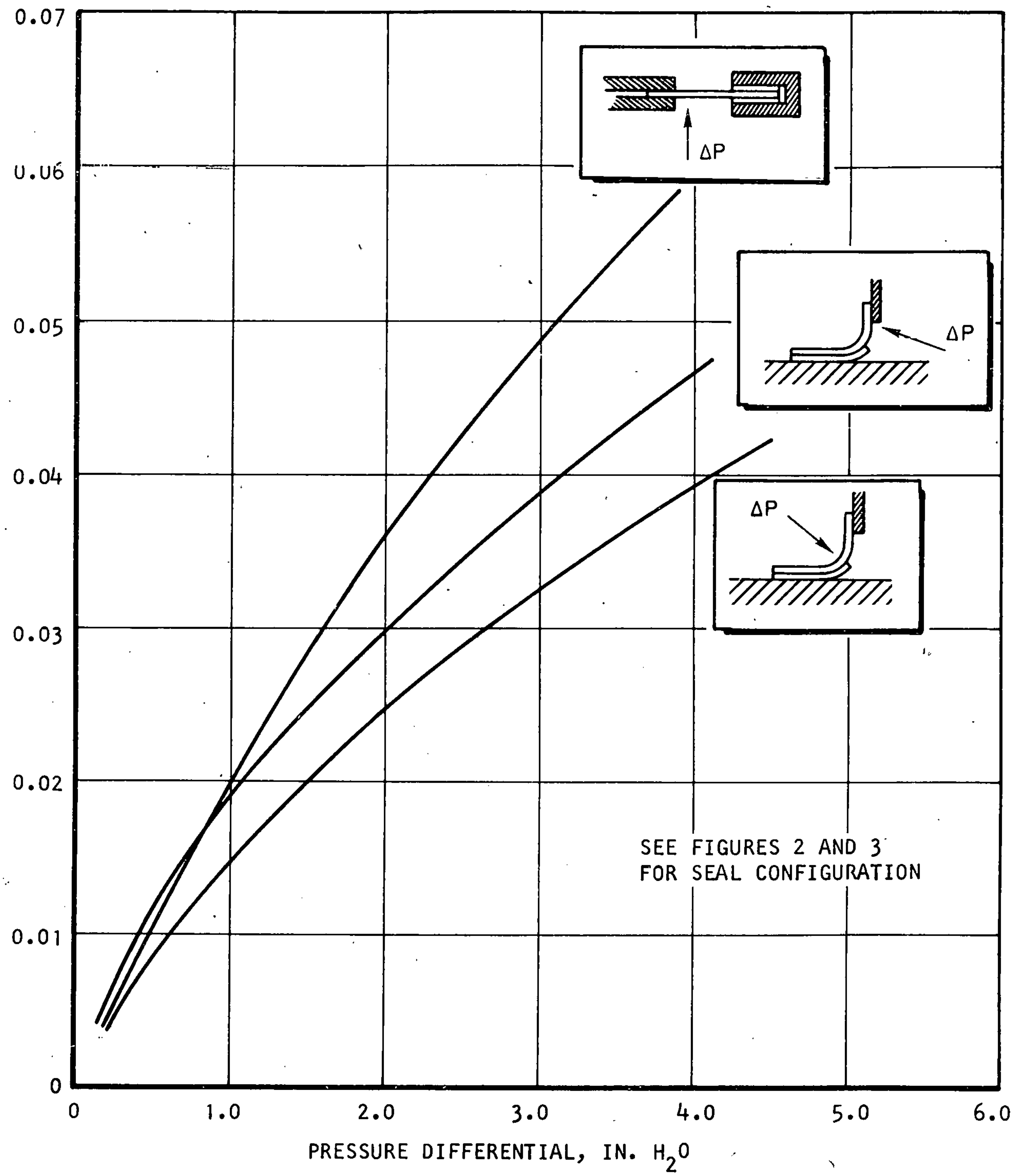


Figure 3-6. Leakage Test Data

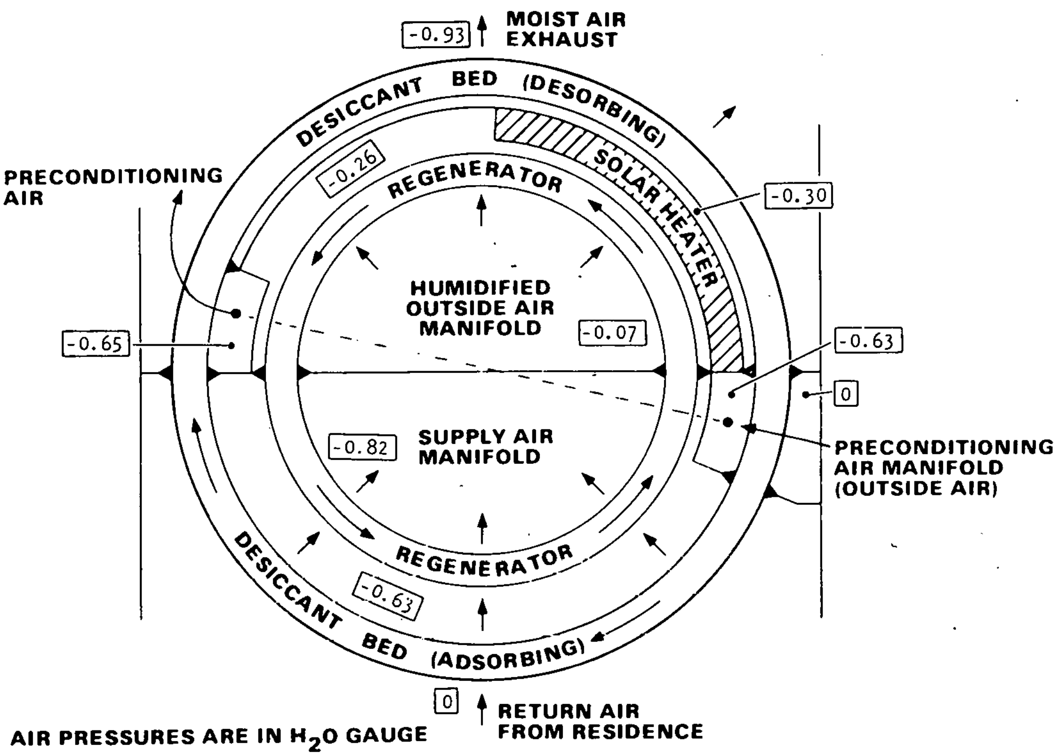


Figure 3-7. Estimated Air Pressure in the Dehumidifier

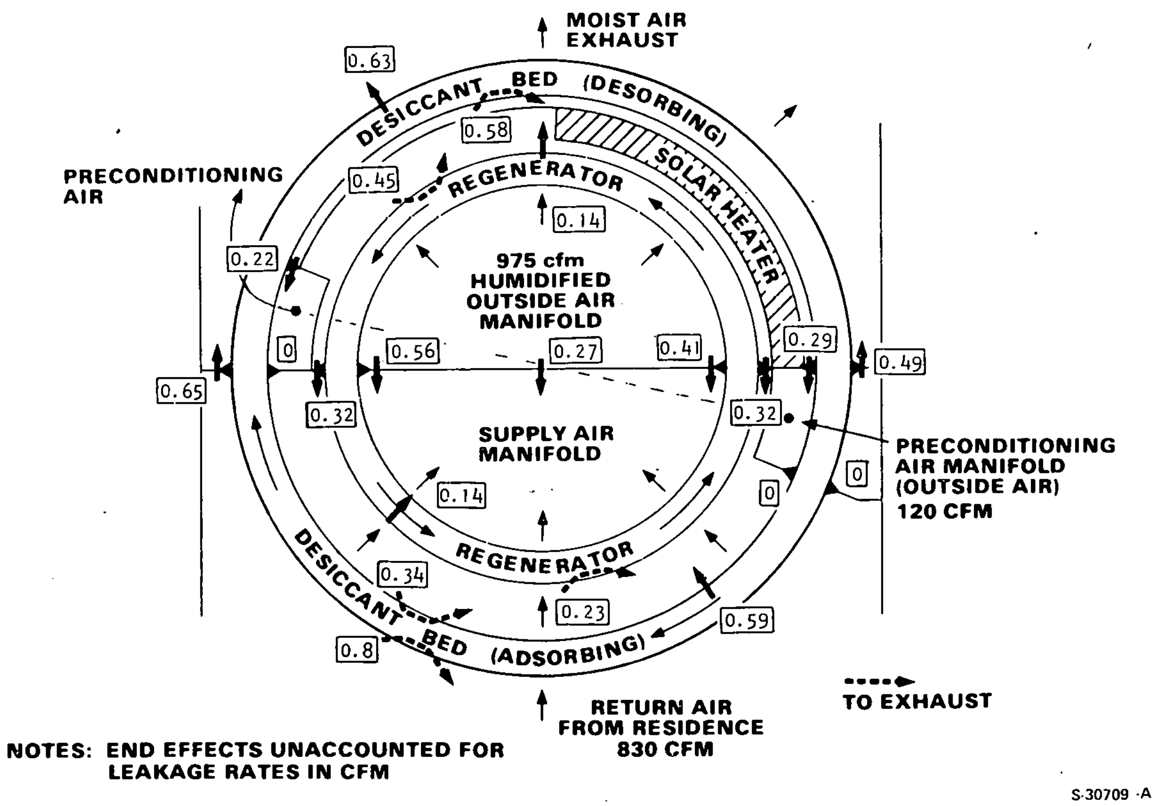


Figure 3-8. Estimated Seal Leakage

The data shown neglect linear seal end effects, which could yield leakage rates one order of magnitude higher than those shown. In the final seal design, careful attention will be paid to the detailed arrangement of the seal interfaces to assure positive sealing throughout.

Friction Tests

Samples of silicone rubber, Dacron felt, Teflon pile, and Celcon sheets were prepared and tested on inclined planes of aluminum and perforated steel to determine relative friction coefficients. The test data are contained in Table 3-1.

TABLE 3-1

COEFFICIENT OF FRICTION OF CANDIDATE MATERIALS

Material	Silicone Rubber	Dacron Felt	Teflon Pile	Celcon Sheet
Aluminum Plate	1.0	0.19	0.17	0.23
Perforated Steel Plate	0.8	0.23	0.23	0.23

Both the Dacron felt and the Teflon Pile displayed a higher coefficient of friction on the perforated plate than on the solid aluminum sheet. This is probably due to a slight penetration of these materials into the holes of the perforated plate. The pressure used on the test sample was 0.15 lb/in. which is representative of the pressure exercised by the deformation of the linear seal strips.

Using these data and previous estimates of the power required to drive the drum with silicone rubber seals, it is estimated that with the Dacron felt pads and the silicone and the seal configuration shown in Figures 3-2 and 3-3, the power will be reduced from 0.75 kw to 0.08 kw. This value is considered acceptable.

CONCLUSIONS

The seal concepts selected appear satisfactory in terms of leakage and friction. No wear tests were performed. However, an evaluation of seal wear during development is planned. This will be done by weighing the seals carefully prior to assembly in the system. Following development testing, the seals will be examined for signs of wear and will be weighed again to determine weight loss due to wear.

Should the Dacron felt be found unacceptable in terms of the 20-year life goal of the system, the material investigation will be started again and candidate materials will be checked for wear on a test rig designed for this purpose. All candidate materials will be tested concurrently at an accelerated rate.

SECTION 4

MARKET SURVEY

INTRODUCTION

A market survey was conducted by Dunham-Bush as part of Task II-2 (Commercialization). A questionnaire was prepared that Dunham-Bush residential equipment distributors were asked to fill and return. A total of 41 responses was received with most of the questions answered. The distributors contacted were primarily from the northeastern area of the United States where the largest portion of Dunham-Bush equipment is sold.

These distributors market a variety of products including HVAC equipment, plumbing and electrical supplies, etc. The distributors are not technically oriented as a rule, but are successful businessmen. In view of the background of the population to be surveyed, it was decided that the solar system would be treated as a black box and the explanation of the intricacies of the solar desiccant system would not be attempted.

The questions, answers, and comments are presented below. The following observations are pertinent:

(a) The distributors plan to market solar equipment if it could become a significant part of their business.

(b) First cost is a major factor in marketing this type of equipment.

(c) A payback period of about 5 years appears acceptable.

(d) The distributors expect low-cost equipment not commensurate with the present cost of solar collectors.

(e) A government incentive will be necessary to reduce first cost.

Survey Data

Question 1. Are you now marketing any solar equipment?

Yes	No
7	34

It is somewhat surprising to find that 17 percent of the distributors are presently carrying solar equipment lines.

Question 2. Do you plan to market solar equipment?

Yes	No
31	5

Only 36 answers were registered here. The remaining five answered "don't know" or are waiting to better assess the market potential.

Answers to questions 1 and 2 are indicative of the public awareness of solar systems and also of their confidence in the development of a marketplace. A minimum of 75 percent of the distributors surveyed plan on marketing solar equipment.

Question 3. Which is more important to your customers: solar heating or solar cooling?

Heat	Cool	Don't Know
31	2	8

The answers to this question reflect the geographical area of the U.S. where the distributors are located.

Question 4. Are customers willing to pay a premium over conventional equipment?

Yes	No	Don't Know
22	4	15

Of the five "no" answers to question 2, three also answered "no" to question 4. The 15 "don't know" answers are indicative of the small number of distributors currently marketing solar equipment. Of the seven distributors who currently market solar equipment, six answered "yes" to this question.

Question 5. What do you feel would be most effective concept in selling the extra cost?

Savings on Utility Bills	Return on Investment	Payback Period
28	5	8

For residential equipment, it appears that the "return on investment" and "payback period" concepts are not sufficiently clear or are not well understood terms. Yearly savings will have to be stressed and customers helped to understand simple payback.

Question 6. Do you feel a solar system is more important for retrofit (add-on) or new construction?

Retrofit	New Construction
16	24

Six of the seven distributors who currently market solar equipment answered "new construction". Considering the number of existing dwellings compared to new construction, it is understandable that a large portion of the solar market will involve retrofit.

Question 7. Do you feel there is a preference for roof mounted collectors or ground mounted collectors?

RETROFIT		NEW CONSTRUCTION	
Roof	16	Roof	30
Ground	16	Ground	6

The answers are as expected. The orientation and design of roofs in existing houses may not be suitable for collector mounting. In a large portion of retrofit installations, ground mounting would be necessary.

Question 8. What is the maximum payback period that you think you could sell? (This is the number of years it takes to get the added cost back.)

3 Years	10 Years
2	15
5 Years	**20 Years**
24	0

The acceptable payback period for a residential installation appears to be longer than five years and as long as ten years. The number of "10 years" answers is surprising.

Question 9. Assuming the payback is acceptable, what percentage of your sales of heating systems (or combined heating/cooling) would be solar?

Less than 2 percent	10 to 20 percent
5	9
2 to 5 percent	**20 to 30 percent**
12	3
5 to 10 percent	**Over 30 percent**
10	11

This question shows that solar installations would represent about 10 percent of the total market. This is sufficient to support a very healthy solar industry if the predictions are realized.

Question 10. Assuming the payback is acceptable, what percentage of cooling system sales would be solar?

Less than 2 percent	10 to 20 percent
9	2
2 to 5 percent	**20 to 30 percent**
9	1
5 to 10 percent	**Over 30 percent**
7	1

Again, the geographical area covered by the survey is evident, and the answers are consistent with the answers to question 3.

Question 11. How sensitive do you feel the market is to first cost rather than payback or some other criterion?

Most important	Important	Not important
16	25	0

Obviously, the first cost is a very important aspect of marketing residential systems. Studies conducted have shown that solar collectors are the highest cost item and the first cost of a solar system. A solar system is several times (as much as an order of magnitude higher) than a conventional system. In the present market, government intervention in the form of tax writeoffs or credits appears to be mandatory until the solar industry is geared for high volume production. Major efforts must be continued to develop low-cost collectors.

Furthermore, life cycle cost, payback period, or cash flow will have to be stressed in order to justify much higher first cost (even at minimum solar collector cost with government incentive).

Question 12. Do you have any preference for an air collector system or a wet collector system?

Air	Wet	Don't Care
15	12	14

The desiccant system could be designed to operate with either an air or liquid collector. In its present configuration, a wet collector is used.

Question 13. What percentage of your dealers are capable of installing (with some training) a solar system?

Less than 10 percent	25 to 50 percent
13	8
10 to 25 percent	**Over 50 percent**
12	7

These answers represent a judgment by people not familiar with the details of installing such solar systems. Of the seven distributors already marketing solar equipment, four answered 25 to 50 percent and one over 50 percent.

Question 14. Are customers becoming used to the idea of the increased bulk of solar equipment (water tanks, rock bins, collectors outside, etc.)?

Yes	No	Don't know
12	5	20

No comments.

Question 15. What do you feel should be the minimum solar contribution (savings) to a heating or cooling system to be salable?

10 percent	20 percent	30 percent	40 percent
0	5	14	7
50 percent	**60 percent**	**70 percent**	**80 percent**
8	4	1	0
90 percent	**100 percent**		
0	0		

The minimum acceptable appears to be near 40 percent. This is relatively low. Desiccant system augmentation presents a problem because of the low basic COP of the process. For relatively low solar contribution (on the order of 40 percent), a hybrid system with conventional vapor compression would be necessary.

Question 16. Say that the installed cost of a 3-ton heat pump system is $4000. How much additional installed cost would a typical customer pay for a solar system which would lower his utility bills by 40 percent year-round?

$2000	$6000
13	10
$4000	$8000
17	0

A 40 percent reduction in heating/cooling cost may correspond to a 400-sq ft collector. Using a $4000 added cost for solar installation would hardly pay for a $10/sq ft collector without any other equipment. Although a $10/sq ft collector is feasible, this type of economic inspection must be supplemented by more detailed analysis of economic factors and presented to the public in a simplified manner. As was ascertained earlier, first cost of solar equipment is a very important factor.

Question 17. How much additional cost would a consumer pay over and above a $10,000 solar assisted heat pump heating system for a solar cooling system (with an EER of 30)?

$1000	$4000
5	11
$2000	$6000
9	2

In the geographical area covered by the survey, the additional collector area necessary to provide cooling over and above that necessary for heating is relatively small, and the cooling loads are relatively low. It follows that the additional $2000 to $4000 involved in the installation of a dessicant system may be adequate.

SECTION 5

1.5-TON SOLAR DESICCANT DEHUMIDIFIER PROTOTYPE SPECIFICATIONS AND DRAWINGS

SUMMARY

This section constitutes the final documentation for Task II-3 (Subsystem Design) and Task II-4 (Desiccant Subsystem Specification). Performance predictions, which are the basis for the design, appear in Section 2 of this report. Design parameters are listed below, and the drawings from which the prototype will be built are included here.

Figure 5-1 schematically shows the major equipment that is included in the prototype unit. The rotary desiccant bed, rotary regenerators, solar heat exchanger, and humidifiers are all enclosed in the equipment package. Also included are secondary items, such as the mechanical drive mechanism. Supply and exhaust fans are to be externally mounted.

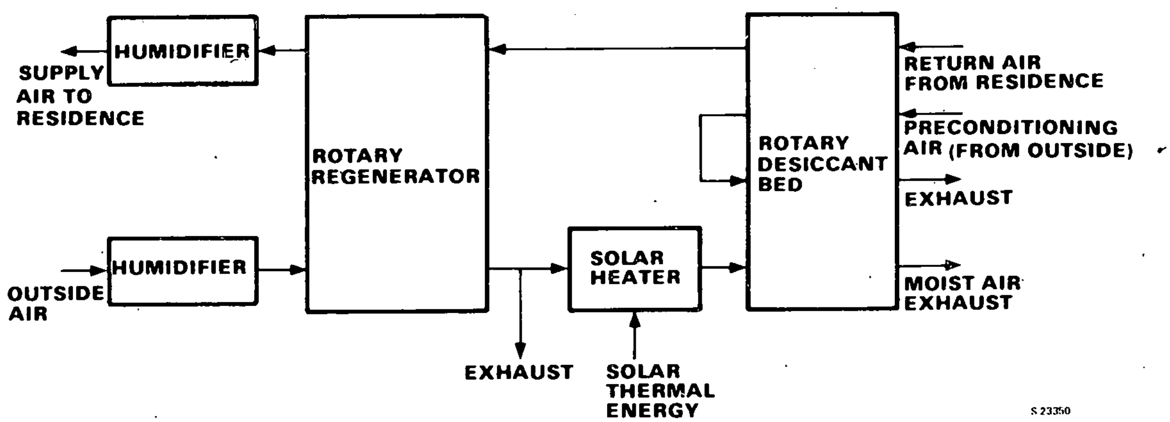


Figure 5-1. Air Conditioner Schematic

Performance and Design Parameters

Table 5-1 shows the optimized design parameters and design point performance for the 1.5-ton prototype. The optimization work was done at operating conditions that correspond to standard ratings for air conditioning systems.

Table 5-2 shows the fan power required for a typical installation. The supply air fan pressure drop given for full flow operation includes 0.10 in. H_2O for distribution system losses.

TABLE 5-1

1.5-TON DESICCANT DEHUMIDIFIER PROTOTYPE
OPTIMIZED DESIGN PARAMETERS AND DESIGN POINT PERFORMANCE

Cooling Capacity: 1.5 tons (18,000 Btu/hr)

Desiccant Bed	Regenerator
8 to 10 mesh silica gel	24 x 24 x 0.014 in. galvanized screen
Bed inside diameter: 31.3 in.	Matrix inside diameter: 19.0 in.
Bed active height: 34.7 in.	Matrix active height: 34.5 in.
Bed thickness: 1.25 in.	Matrix thickness: 1.13 in.
Bed weight (dry): 100 lb	Matrix weight: 175 lb
Rotating speed: 5 rph	Rotating speed: 20 rpm
Working capacity: 3.7 percent	Effectiveness: 90 percent
Pressure drop: 0.63 in. H_2O	Pressure drop: 0.19 in. H_2O
Airflow rates	**Solar Heater**
Residence airstream: 830 scfm	Effectiveness: 85 percent
Preconditioning airstream: 120 scfm	Arc: 86.6 deg
Outside airstream (without preconditioning air): 830 scfm	Heating rate: 35,000 Btu/hr
Solar heater airstream: 455 scfm	Water flow rate: 3600 lb/hr
	Pressure drop (air side): 0.04 in. H_2O
Preconditioning Air	**Mechanical Drive**
Manifold arc: 22.5 deg	Power requirement: 0.1 kw (maximum)

Note: Design point calculations made at following standard conditions:

Conditioned space: 78°F DB, 67°F WB

Outside: 95°F DB, 75°F WB

Hot water supply temperature: 200°F

Barometric pressure: 14.696 psia

TABLE 5-2

FAN POWER REQUIREMENTS

	Flow, cfm	Static Pressure Rise, in. H_2O	Air Horsepower	Electric Horsepower*
Supply Air Fan				
Full Flow	830	1.00	0.13	0.26
Half Flow	415	0.28	0.02	0.04
Exhaust Fan				
Full Flow	948	0.93	0.14	0.28
Half Flow	474	0.26	0.02	0.04

*Assuming 50-percent overall efficiency.

Drawings

The solar desiccant dehumidifier prototype design is shown on the following drawings:

Drawing No.	Title	Sheets
2202169-1	Heat Exchanger, Solar (SCD)	5
2202400-1	Dehumidifier Outline, Solar Desiccant	1
2202401-1	Dehumidifier Assembly	1
2202172-1	Frame Assembly, Dehumidifier	3
2201848-1	Cylinder Assembly, Desiccant	1
2201849-1	Cylinder Assembly, Regenerator	1
2202173-1	Saturator Assembly	1
2202183-1	Frame Assembly, Annular	3
2202184-1	Frame Assembly, Outer	3
2202185-1	Panel Assembly, Center	1
2202186-1	Frame Roller	1
2202187-1	Duct Assembly, Preconditioning Air	2
2202198-1	Idler Arm Assembly	1

Reduced-scale prints of the drawings follow this section. Due to the practical limitations of the printing format used for this report, the drawings are not all reduced by the same factor. Therefore, care should be used when scaling these reduced prints.

REVISIONS			
LTR	DESCRIPTION	DATE	APPROVAL

R

NOTES: UNLESS OTHERWISE SPECIFIED

1. PART NUMBER CHANGES AND/OR DESIGN CHANGES AFFECTING ITEM INTERCHANGEABILITY REQUIRE PRIOR AIRESEARCH APPROVAL AND AUTHORIZATION BY REVISION TO THIS DRAWING.

2. PACKAGING AND SHIPPERS SHALL INCLUDE THE AIRESEARCH CODE IDENTIFICATION NUMBER AND ITEM IDENTIFYING NUMBER.

3. ITEM TO BE PERMANENTLY MARKED WITH THE FOLLOWING MINIMUM IDENTIFICATION IN ACCORDANCE WITH MIL-STD-130:
   AIRESEARCH CODE IDENT NUMBER "70210"
   AIRESEARCH ITEM IDENTIFICATION NUMBER 2202169-1
   VENDOR CODE IDENT NUMBER OR TRADEMARK

4. ONLY THE ITEM DESCRIBED ON THIS DRAWING WHEN PROCURED FROM THE VENDOR(S) LISTED HEREON IS APPROVED BY AIRESEARCH MFG CO. OF CALIFORNIA, TORRANCE, CALIFORNIA, FOR USE IN THE APPLICATION(S) SPECIFIED HEREON. A SUBSTITUTE ITEM SHALL NOT BE USED WITHOUT PRIOR TESTING AND APPROVAL BY AIRESEARCH MANUFACTURING CO.

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SHEET INDEX						
REVISION LTR	—	—	—	—	—	
SHEET NO	1	2	3	4	5	

SOURCE CONTROL DRAWING	CONTRACT NO	AIRESEARCH MANUFACTURING COMPANY OF CALIFORNIA A DIVISION OF THE GARRETT CORPORATION TORRANCE, CALIFORNIA
	DFT S. RAILING 78-8-15	HEAT EXCHANGER, SOLAR
	CHK	
	APVD [signature] 78-10-17	
	APVD	
	AIRESEARCH APVD [signature]	SIZE A — CODE IDENT NO 70210 — DWG NO 2202169
	OTHER ACTIVITY APVD	SCALE — SHEET 1 OF 5

7. SCOPE

7.1 DEFINITION

This source control drawing (SCD) defines the solar heat exchanger and its specific technical requirements.

7.2 RESPONSIBILITY

The vendor shall be responsible for the design, development and manufacturing of the solar heat exchanger.

7.3 This SCD forms a part of the AiResearch purchase order and as such the vendor's compliance herewith is essential. The terms of the purchase order will not be closed out until all requirements of this document, as revised or otherwise amended by appropriate written notice, have been met.

8. DESCRIPTION

8.1 GENERAL

The heat exchanger will be installed in an annular sector between two rotating cylinders. Moist air will flow radially outward from the inner cylinder, through the heat exchanger, and out through the outer cylinder.

9. REQUIREMENTS

9.1 PERFORMANCE

9.1.1 Operating media: Moist air outside tubes, water inside tubes.

9.1.2 OPERATION

The unit must heat 33.3 lbm/min. of air from 107°F average inlet temperature to 182°F average outlet temperature, using 60.0 lbm/min. of water entering at 200°F.

9.1.3 Process Stream Pressure Drops

Air side: 0.03 to 0.05" H_2O static pressure drop (corrected to standard conditions)

Water side: 5 psi maximum static pressure drop.

9.2 PHYSICAL CHARACTERISTICS

9.2.1 DIMENSIONS

The heat exchanger shall be contained within the envelope as shown on sheet 4.

9.2.2 MOUNTING

The heat exchanger shall be supported by bolting to the adjacent assembly at the two radial surfaces shown on sheet 4.

9.2.3 WATER MANIFOLD CONNECTIONS

Water manifold connections shall be provided at the top surface of the mounting envelope as shown on sheet 4. Connections shall be plain end copper tubing.

9.2.4 HEAT EXCHANGER CORE

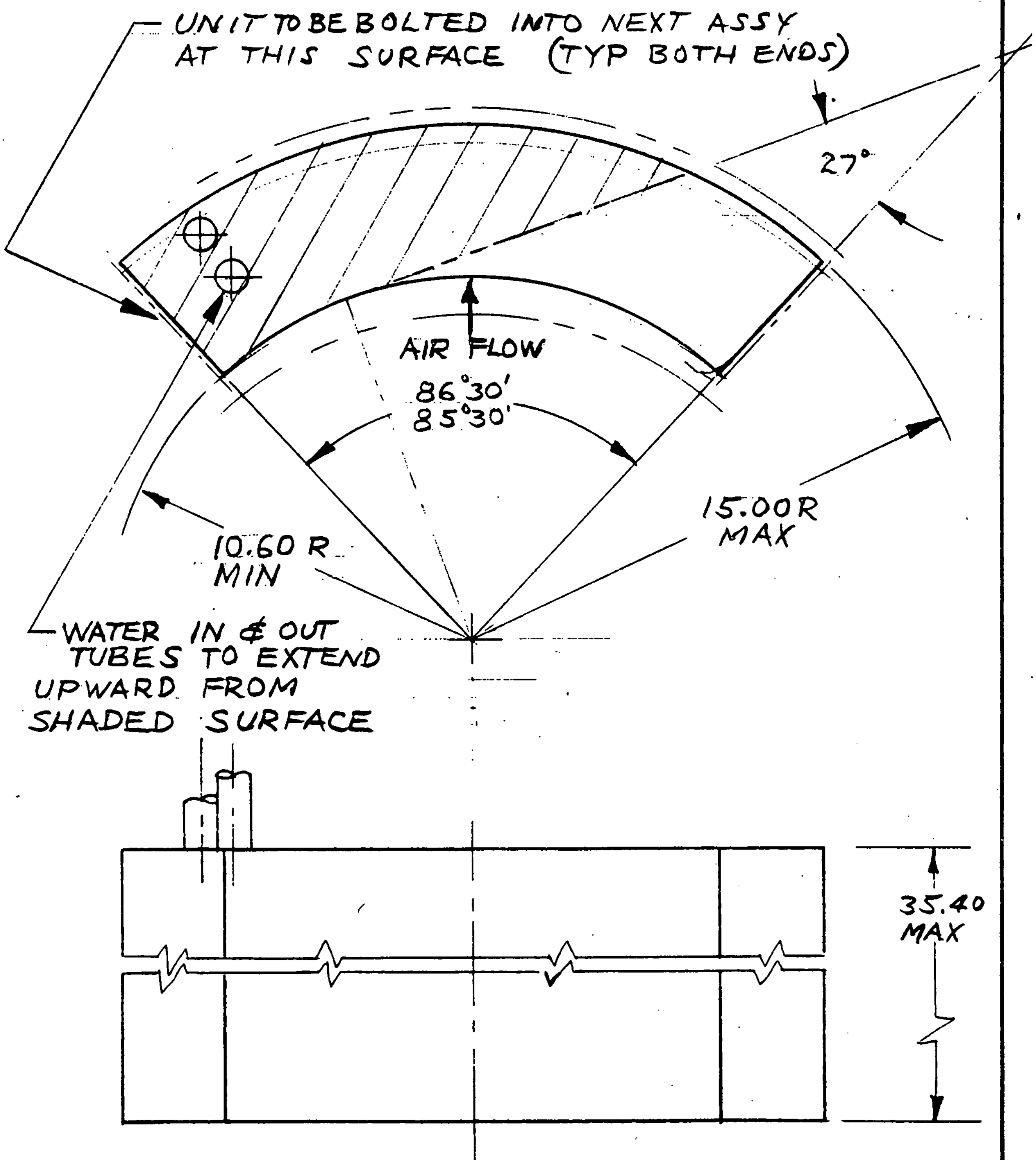
The core shall be constructed of three rows of straight vertical tubes, with twelve passes per row. Tubes shall be 0.625" OD X 0.022" wall copper. Fins shall be aluminum, 13/32" wide by 0.011" thick, mechanically bonded to tubes, with 14 fins per inch.

9.2.5 FRAME

The heat exchanger frame shall be constructed of galvanized steel.

9.2.6 CLEANLINESS AND PACKAGING

The vendor shall clean and package the heat exchanger per normal industrial practices.



HEAT EXCHANGER, SOLAR

AIRESEARCH MANUFACTURING COMPANY OF CALIFORNIA A DIVISION OF THE GARRETT CORPORATION TORRANCE, CALIFORNIA	SIZE A	CODE IDENT NO 70210	DWG NO 2202169
	SCALE 1/4	REV	SHEET 4

APPROVED SOURCE(S) OF SUPPLY

AIRESEARCH IDENTIFICATION NUMBER	VENDOR IDENTIFICATION NUMBER	VENDOR	VENDOR CODE IDENT NUMBER	END UNIT APPLICATION
2202169-1		AERO FIN		2202400-1

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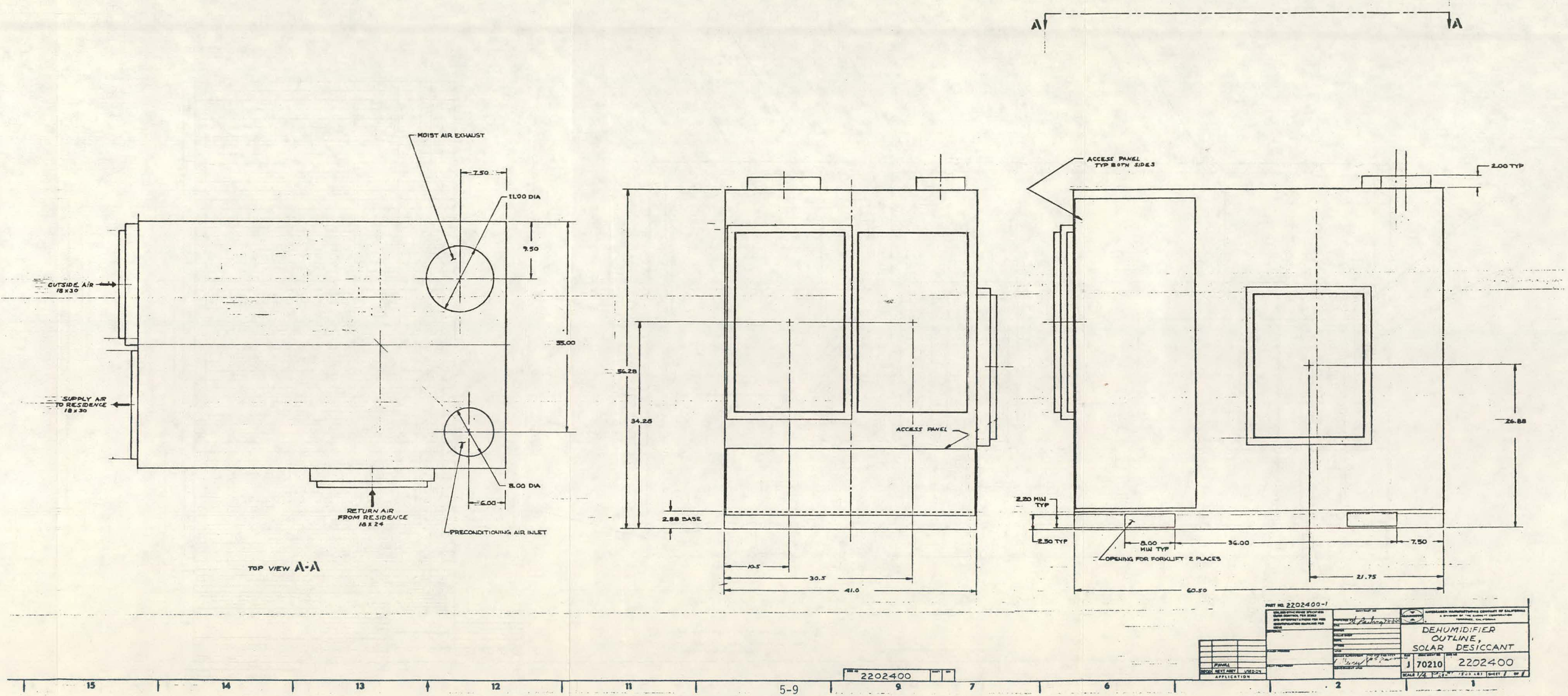
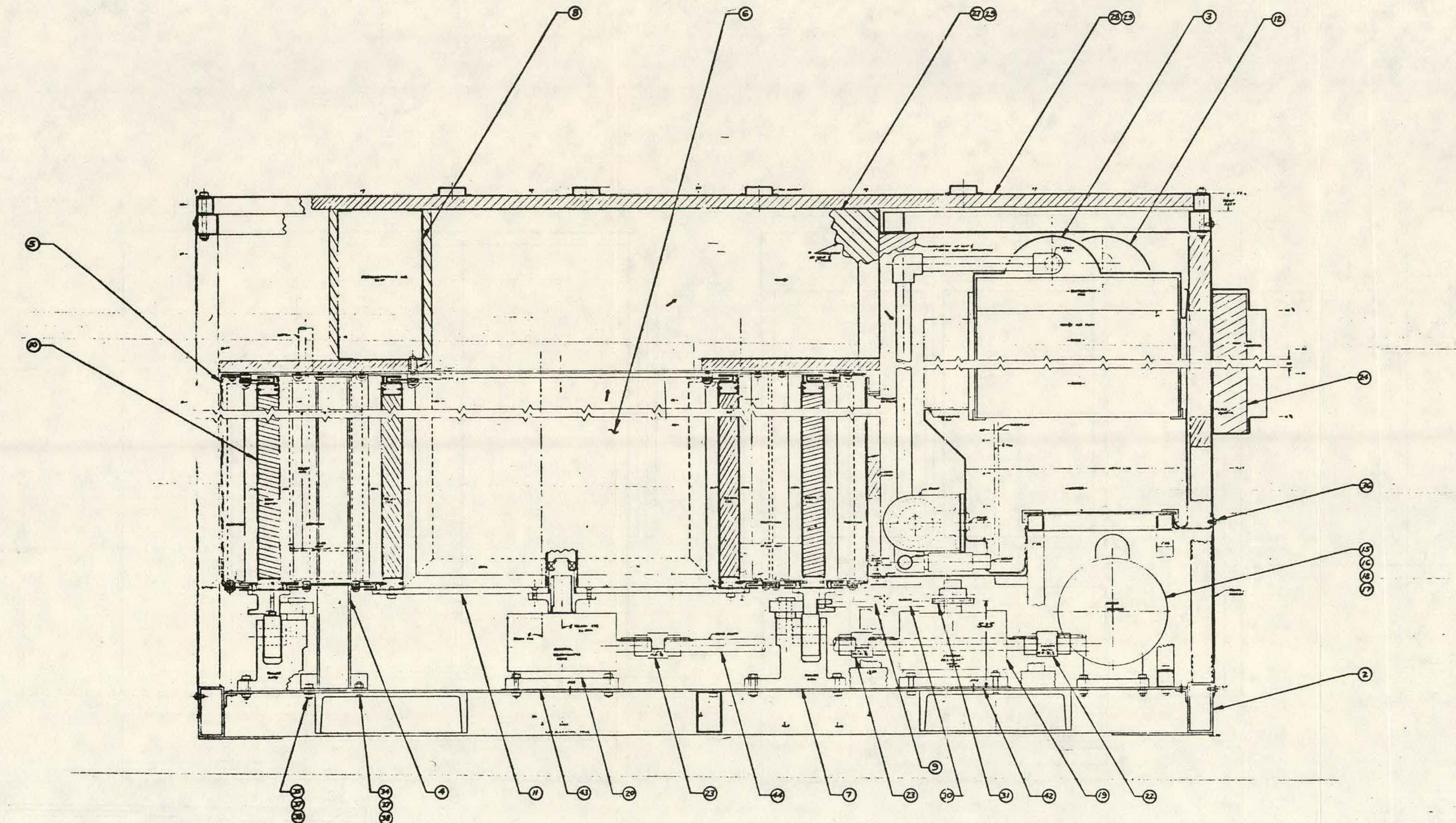
GARRETT AIRESEARCH MANUFACTURING COMPANY OF CALIFORNIA A DIVISION OF THE GARRETT CORPORATION TORRANCE, CALIFORNIA	SIZE A	CODE IDENT NO. 70210	DWG NO. 2202169
	SCALE	REV	SHEET 5


MOIST AIR EXHAUST
7.50
11.00 DIA
9.50
OUTSIDE AIR 18 x 30
35.00
SUPPLY AIR TO RESIDENCE 18 x 30
8.00 DIA
6.00
RETURN AIR FROM RESIDENCE 18 x 24
PRECONDITIONING AIR INLET
TOP VIEW A-A
ACCESS PANEL TYP BOTH SIDES
2.00 TYP
54.28
34.28
ACCESS PANEL
2.88 BASE
10.5
30.5
41.0
2.20 MIN TYP
2.50 TYP
8.00 MIN TYP
36.00
7.50
OPENING FOR FORKLIFT 2 PLACES
21.75
60.50
26.88
PART NO. 2202400-1
APPLICATION
USED ON
DEHUMIDIFIER OUTLINE, SOLAR DESICCANT
J
70210
2202400
SCALE 1/4
SHEET 1 OF 1



SEAL LEAKAGE PATHS WITH ITEM 48

PURCHASED ITEM SEE SYSTEM SPEC 78-15445

NOTES: UNLESS OTHERWISE SPECIFIED

2202401 | SHEET 1

QTY REQD	ITEM NO	CODE IDENT NO	PART OR IDENTIFYING NO	NOMENCLATURE OR DESCRIPTION	SYM
AR	48		RS184C01	SEALANT OR EQUIV	
1	44		-15	SHAFT .75 DIA KEYED	
1	43		-13	SPACER .062 PLATE	
1	42		-11	SPACER .094 PLATE	
AR	38			WASHER .25 DIA	
AR	37			NUT .25 DIA	
AR	36			SCREW NO.10 SELF TAPING	
AR	35			BOLT, HEX HD .25-20 VARIOUS LENGTHS	
AR	34			BOLT, FLAT HD .25-20	
1	31		40B13-1	SPROCKET 40-½ 13 TEETH BOSTON GR.	
1	30		40SL	CHAIN ½ PITCH BOST-LUBE BOSTON GR.	
AR	29		373	ADHESIVE RUBATEX	
AR	28			INSULATION .75 THICK RUBATEX	
AR	27			INSULATION 1.00 THICK RUBATEX	
1	25			FILTER 18X24X2 THK DURALAST	
1	24			FILTER 18X30X2 THK DURALAST	
2	23		FC15-¾	COUPLING BOSTON GEAR	
1	22		FC15-¾-⅞	COUPLING BOSTON GEAR	
1	20		2265-[illegible]-7.2:1	GEARBOX (REGENERATOR CYL)	1
1	19		2225-[illegible]-40:1	GEARBOX (DESICCANT CYLINDER)	1
1	18		2192-[illegible]	GEARBOX (MOTOR)	1
1	17		BC 290	CONTROLLER, MOTOR	1
1	16		504-36-040	MOTOR, DC	1
1	15		C48E677	MOTOR, AC	1
1	12		2202173-2	SATURATOR ASSY	
1	11		2201849-1	CYLINDER ASSY, REGENERATOR	
1	10		2201848-1	CYLINDER ASSY, DESICCANT	
1	9		2202198-1	IDLER ARM ASSY	
1	8		2202187-1	DUCT ASSY, PRECONDITIONING	
8	7		2202186-1	FRAME, ROLLER	
1	6		2202185-1	PANEL ASSY, CENTER	
1	5		2202184-1	FRAME ASSY, OUTER	
1	4		2202183-1	FRAME ASSY, ANNULAR	
1	3		2202173-1	SATURATOR ASSY	
1	2		2202172-1	FRAME ASSY	

PART NO -1 ASSY — PARTS LIST

NEXT ASSY	USED ON
-1 2202400	2202400-1

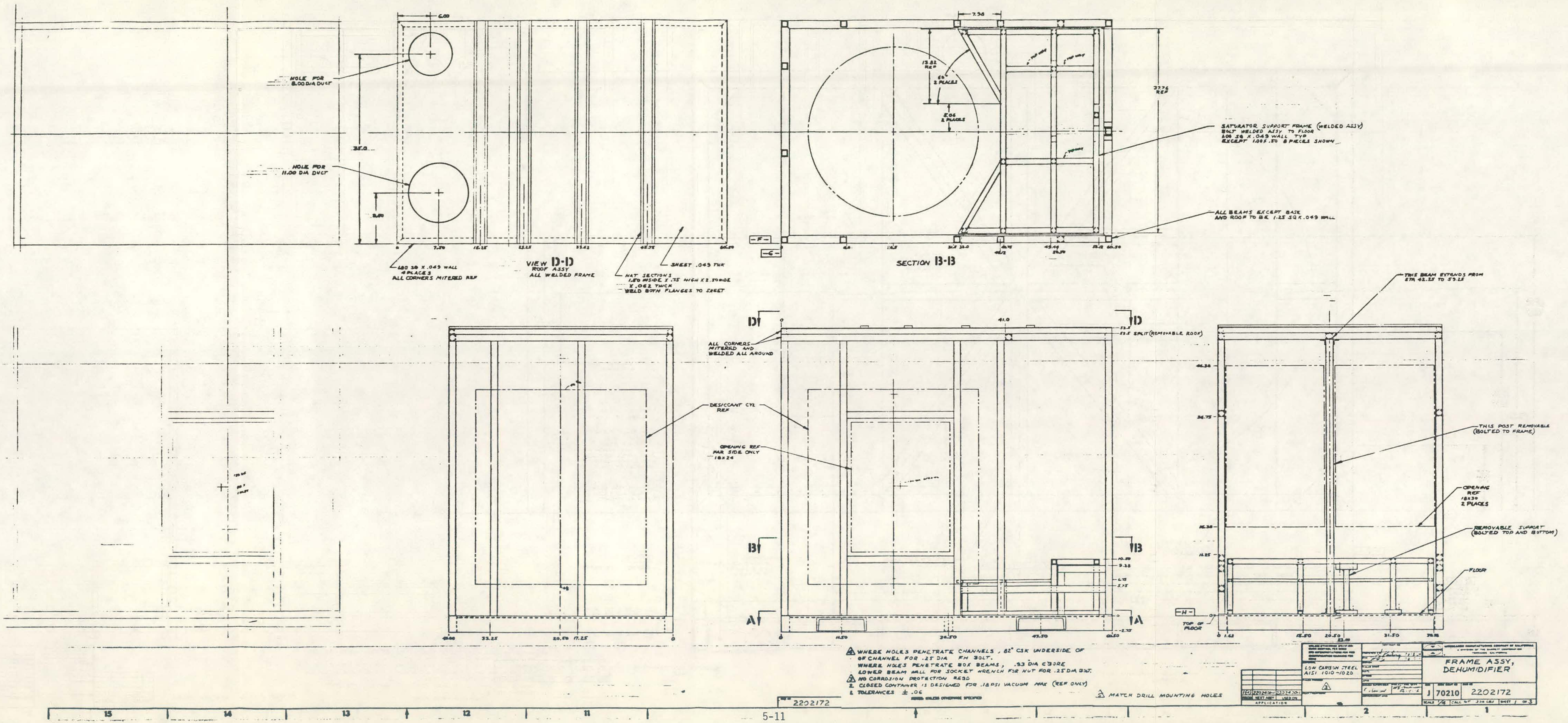
APPLICATION

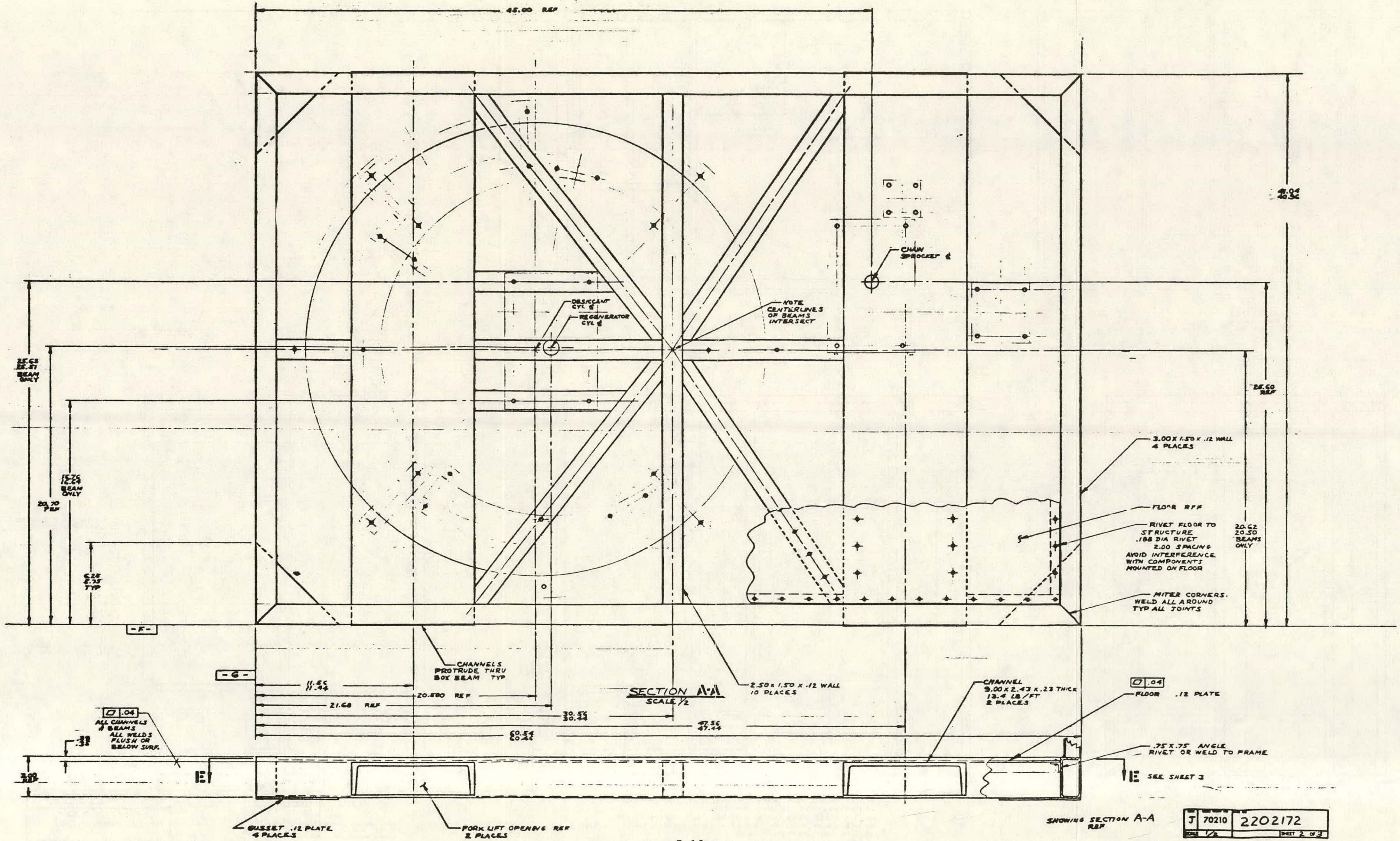
AIRESEARCH MANUFACTURING COMPANY OF CALIFORNIA
A DIVISION OF THE GARRETT CORPORATION
TORRANCE, CALIFORNIA

DEHUMIDIFIER ASSY

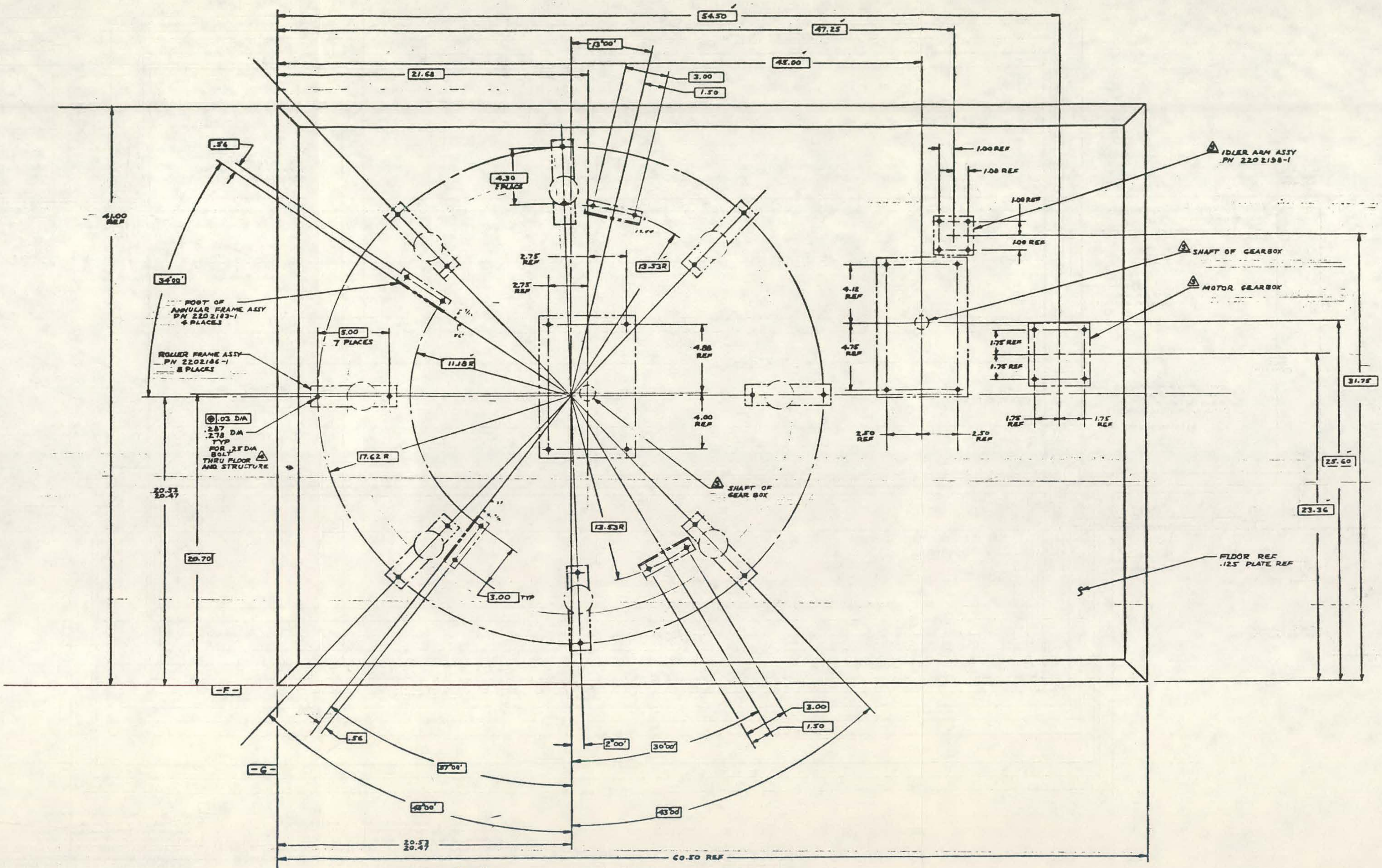
SIZE J | CODE IDENT NO 70210 | DWG NO 2202401

SCALE 1/2 | SHEET 1 OF 1


HOLE FOR 6.00 DIA DUCT
HOLE FOR 11.00 DIA DUCT
VIEW D-D
ROOF ASSY
ALL WELDED FRAME
SHEET .043 THK
SECTION B-B
SATURATOR SUPPORT FRAME (WELDED ASSY)
ALL BEAMS EXCEPT BASE AND ROOF TO BE 1.25 SQ X .049 WALL
THIS BEAM EXTENDS FROM STA 42.25 TO 53.25
ALL CORNERS MITERED AND WELDED ALL AROUND
53.5 SPLIT (REMOVABLE ROOF)
DESICCANT CYL REF
THIS POST REMOVABLE (BOLTED TO FRAME)
REMOVABLE SUPPORT (BOLTED TOP AND BOTTOM)
FLOOR
TOP OF FLOOR
WHERE HOLES PENETRATE CHANNELS, 82° CSK UNDERSIDE OF OF CHANNEL FOR .25 DIA FH BOLT.
WHERE HOLES PENETRATE BOX BEAMS, .93 DIA C'BORE
LOWER BEAM WALL FOR SOCKET WRENCH FOR NUT FOR .25 DIA BOLT.
NO CORROSION PROTECTION REQD
2. CLOSED CONTAINER IS DESIGNED FOR .18 PSI VACUUM MAX (REF ONLY)
1. TOLERANCES ± .06
3 MATCH DRILL MOUNTING HOLES
LOW CARBON STEEL AISI 1010-1020
FRAME ASSY, DEHUMIDIFIER
J 70210 2202172
SCALE 1/4
SHEET 1 OF 3
2202172



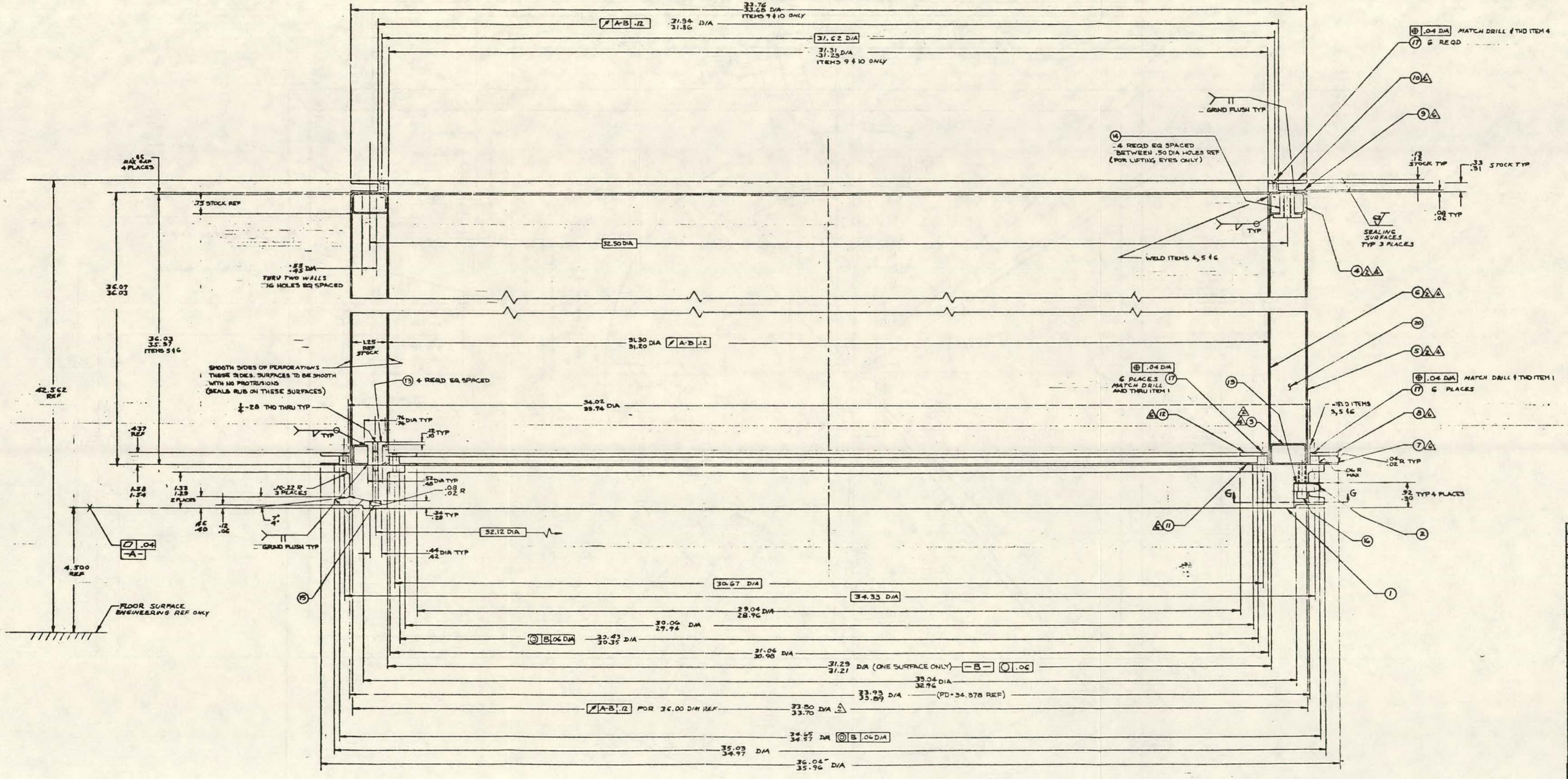
45.00 REF
CHAIN SPROCKET ℄
DESICCANT CYL ℄
REGENERATOR CYL ℄
NOTE CENTERLINES OF BEAMS INTERSECT
3.00 X 1.50 X .12 WALL 4 PLACES
FLOOR REF
RIVET FLOOR TO STRUCTURE .188 DIA RIVET 2.00 SPACING AVOID INTERFERENCE WITH COMPONENTS MOUNTED ON FLOOR
MITER CORNERS. WELD ALL AROUND TYP ALL JOINTS
CHANNELS PROTRUDE THRU BOX BEAM TYP
SECTION A-A
SCALE 1/2
2.50 X 1.50 X .12 WALL 10 PLACES
CHANNEL 3.00 X 2.43 X .23 THICK 13.4 LB/FT 2 PLACES
FLOOR .12 PLATE
21.68 REF
20.500 REF
ALL CHANNELS 4 BEAMS ALL WELDS FLUSH OR BELOW SURF.
.75 X .75 ANGLE RIVET OR WELD TO FRAME
SEE SHEET 3
GUSSET .12 PLATE 4 PLACES
FORK LIFT OPENING REF 2 PLACES
SHOWING SECTION A-A REF
J 70210 2202172
SHEET 2 OF 3

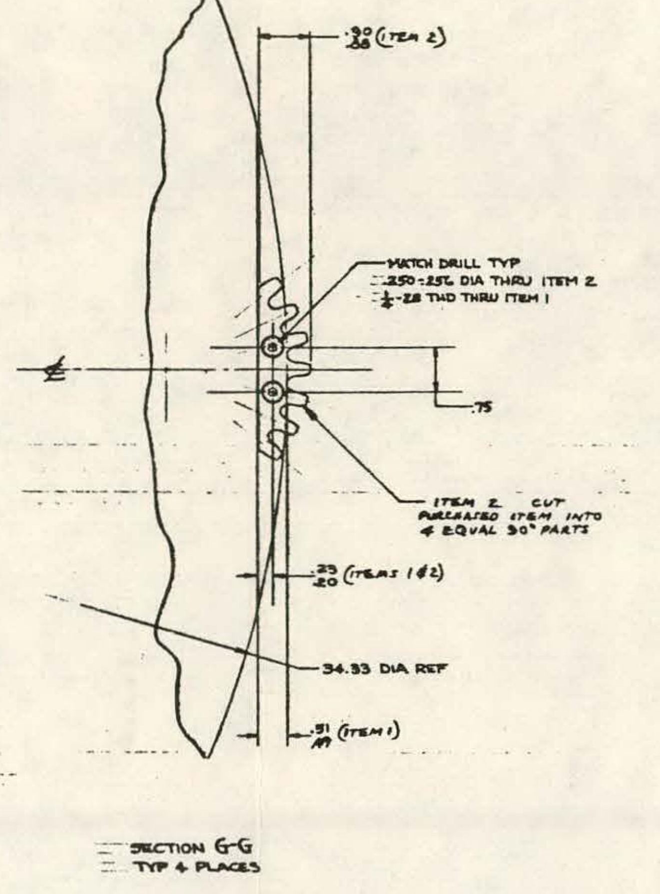


SECTION E-E
SHOWING BOLT PATTERN IN FLOOR

SHOWING SECTION E-E REF

J	70210	2202172
SCALE 1/2		SHEET 3 OF 7





AR	20			DESICCANT	3
4	19		-37	SHIM .020 SHEET	8
18	17			SCREW, FLUSH HD 6-32 X .75 LG 82°	
8	16		NAS1351-4-12	SCREW, CAP ¼-28 x .75 LG	
4	15		NAS1351-4-28	SCREW, CAP ¼-28 x 1.75 LG	
4	14		2201849-33	BOSS	
4	13		-35	BOSS	8
1	12		-33	SEAL PLATE .125 PLATE	9
1	11		-31	SEAL PLATE .312 PLATE	9
1	10		-29	SEAL PLATE .125 PLATE	9
1	9		-27	SEAL PLATE .312 PLATE	9
1	8		-25	SEAL PLATE .125 PLATE	9
1	7		-23	SEAL PLATE .312 PLATE	9
1	6		-21	CYLINDER, INNER	1
1	5		-19	CYLINDER, OUTER	1
1	4		-17	HOOP 1.25 x .75 x .049 WALL	8
1	3		-15	HOOP 1.25 x .75 x .049 WALL	8
4	2		-13	SPROCKET	5
1	1		-11	RING	8
QTY REQD	ITEM NO	CODE IDENT NO	PART OR IDENTIFYING NO	NOMENCLATURE OR DESCRIPTION	SYM

PART NO -1 ASSY — PARTS LIST

9. AL ALLOY 6061-T6 HARD ANODIZE PER MIL-A-8625 TYPE III
8. AISI 1010-1020 LOW CARBON STEEL.
7. NO CORROSION PROTECTION REQD
6. ITEMS 7 & 8 AND 9 & 10 AND 11 & 12 MAY BE MADE AS ONE PIECE. ITEMS 7,8,9,10,11,12 MUST BE REMOVABLE AND REASSEMBLED AT NEXT ASSY. ITEMS 7,8,11,12 MUST PASS EASILY OVER ITEMS 5 & 6.
5. MAKE FROM STEEL CHAIN SPROCKET 40-½ 28 TEETH (1 REQD) BOSTON GEAR CAT NO 40A28, ITEM NO 68016.
4. ROLL, BUTT WELD, GRIND WELD FLUSH
3. SILICA GEL 8 TO 10 MESH GRANULAR (106 POUNDS MIN) FILL ANNULUS FULL TO TOP WHILE VIBRATING.
2. MATCH DIAMETERS OF ITEMS 3, 4, 5 & 6
1. MAKE FROM .024 PERFORATED ZINC GRIP AISI 1010-1020 LOW CARBON STEEL SHEET .050 DIA HOLES ON STRAIGHT ROWS .066 APART 45% OPEN (PATTERN NO 214 DIAMOND PERFORATED METALS INC, GARDENA, CALIF) DO NOT PERFORATE AT 5/8 BORDERS EACH SIDE LEAVE ¼ BORDER EACH END AT BUTT WELD.

NOTES: UNLESS OTHERWISE SPECIFIED

-1 2202401 — NEXT ASSY — APPLICATION

AIRESEARCH MANUFACTURING COMPANY OF CALIFORNIA
A DIVISION OF THE GARRETT CORPORATION
TORRANCE, CALIFORNIA

CYLINDER ASSY, DESICCANT

J 70210 2201848

SCALE 1/1 SHEET 1 OF 1

2201848

9. MAKE PAN LEAK TIGHT WITH ITEM 21

8. ADDITIONAL COUPLINGS PERMITTED

7. TRIM ITEM 10 TO 12 = 32

6. CORE & DISTRIBUTION PAD MUST PASS FREELY THRU FRAME.

5. BEND RADII .05-.13 R FOR ITEMS 1, 3 & 4

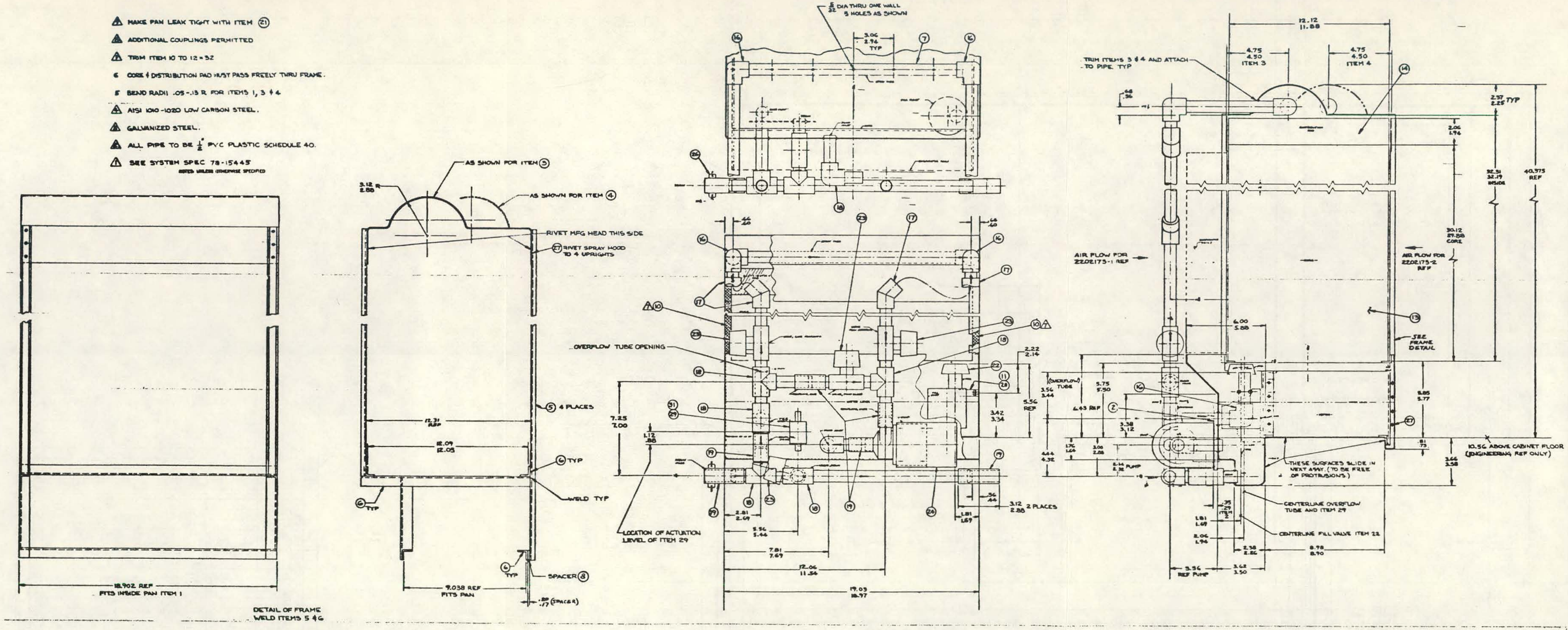
4. AISI 1010-1020 LOW CARBON STEEL.

3. GALVANIZED STEEL.

2. ALL PIPE TO BE 1/2" PVC PLASTIC SCHEDULE 40.

1. SEE SYSTEM SPEC 78-15445

NOTES: UNLESS OTHERWISE SPECIFIED



QTY REQD -2	QTY REQD -1	ITEM NO	CODE IDENT NO	PART OR IDENTIFYING NO	NOMENCLATURE OR DESCRIPTION	SYM
1	1	31		-33	BRACKET, LEVEL SWITCH MOUNT	
1	1	29			LEVEL SWITCH (LS-1800)	1
1	1	28			BAND CLAMP	
A/R	A/R	27			RIVET	
1	1	26			GROMMET	
1	1	24		08512-000	PUMP	1
4	4	23			VALVE, MANUAL	
1	1	22		400-A	VALVE, FILL	1
A/R	A/R	21		RS184C01	SEALANT	9
10	10	20			CLAMP, HOSE FOR 1" HOSE	
6	6	19			COUPLING, HOSE	8
9	9	18			TEE	2
4	4	17			ELBOW 45°	2
5	5	16			ELBOW 90°	2
AR	AR	15			PIPE 1/2 (.840 OD)	2
1	1	14			PAD DISTRIBUTION 2 X 12 X 18 CELDEK	1
1	1	13		6560/15	CORE 12 X 18 X 30 CELDEK	1
A/R	A/R	12		373	ADHESIVE, RUBATEX	
1	1	11		-31	SADDLE	
2	2	10		-29	INSULATION .50 THICK RUBATEX	
2	2	8		-25	SPACER .188 THK X .75 X .75	4
1	1	7		-23	PIPE, SPRAY	2
A/R	A/R	6		-21	ANGLE .75 X .75 X .125	4
4	4	5		-19	CHANNEL .375 X .75 X .125	4
1	—	4		-17	COVER .049 SHEET	3
—	1	3		-15	COVER .049 SHEET	3
1	1	2		-13	BRACKET, PUMP SHEET	3
1	1	1		-11	PAN .049 SHEET	3
-2	-1	ASSY			PARTS LIST	

REQD	NEXT ASSY	USED ON
(-2)	2202401-1	2202400-1
(-1)	2202401-1	2202400-1
APPLICATION		

2 3

CORROSION PROTECTION REQD ON ALL ITEMS

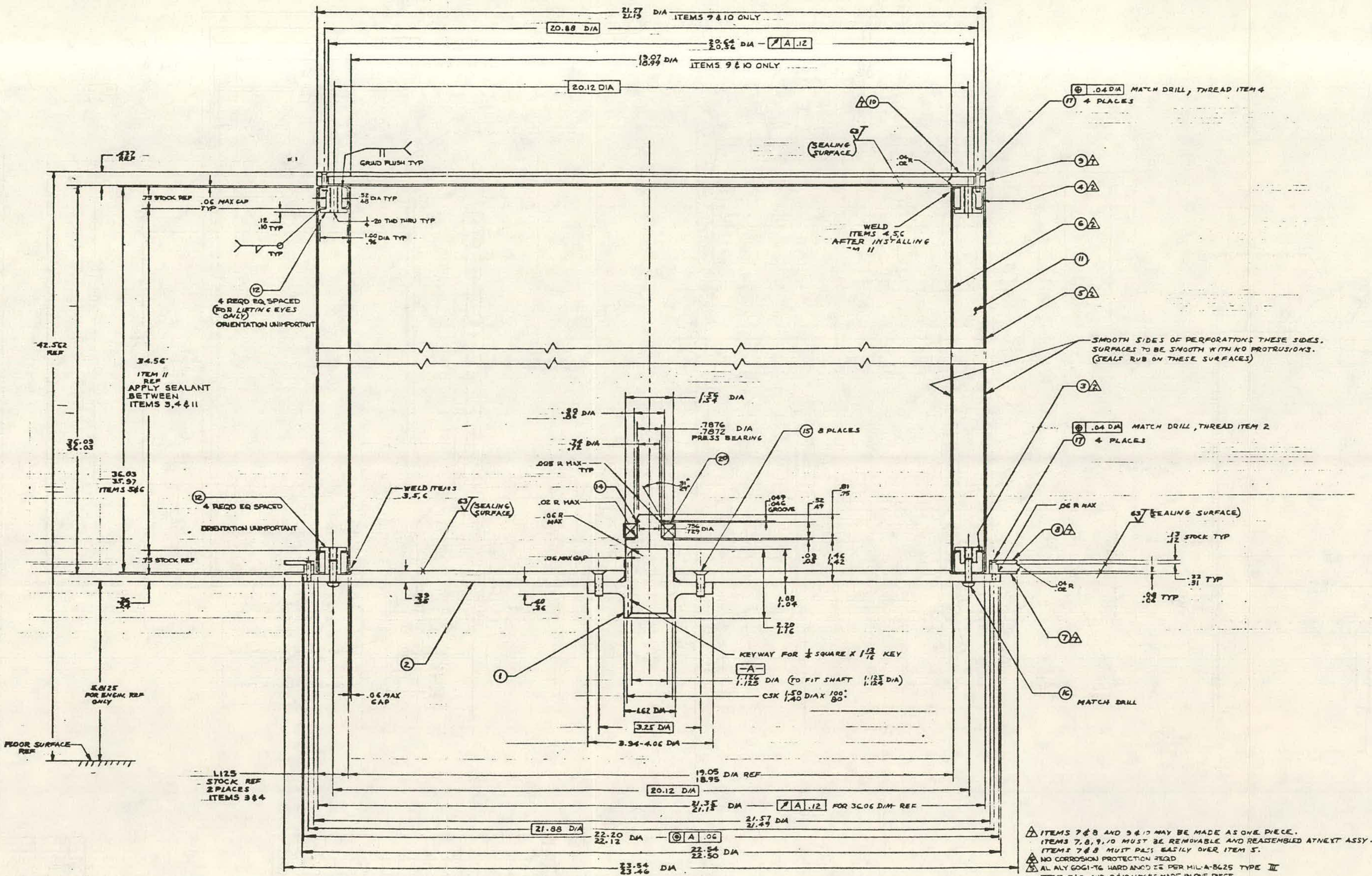
AIRESEARCH MANUFACTURING COMPANY OF CALIFORNIA
A DIVISION OF THE GARRETT CORPORATION
TORRANCE, CALIFORNIA

SATURATOR ASSY

SIZE J | CODE IDENT NO 70210 | DWG NO 2202173

SCALE 1/4 | SHEET 1 OF 1

2202173



⚠1 ITEMS 7 & 8 AND 9 & 10 MAY BE MADE AS ONE PIECE.
ITEMS 7, 8, 9, 10 MUST BE REMOVABLE AND REASSEMBLED AT NEXT ASSY.
ITEMS 7 & 8 MUST PASS EASILY OVER ITEM 5.
⚠6 NO CORROSION PROTECTION REQD
⚠5 AL ALY 6061-T6 HARD ANODIZE PER MIL-A-8625 TYPE III
ITEMS 7 & 8 AND 9 & 10 MAY BE MADE IN ONE PIECE.
⚠4 AISI 1010-1020 LOW CARBON STEEL.
⚠3 CORE TO BE CARBON STEEL SCREEN, GALVANIZED .014 DIA WIRE 24 x 24 STRANDS PER INCH WIND 180 POUNDS ONTO ITEM 6.
⚠2 ROLL AND BUTT WELD. MATCH DIAMETERS OF ITEMS 3, 4, 5 & 6
⚠1 MAKE FROM .024 PERFORATED ZINC GRIP AISI 1010-1020 LOW CARBON STEEL SHEET .050 DIA HOLES ON STRAIGHT ROWS .066 APART 45% OPEN (PATTERN NO 214 DIAMOND PERFORATED METALS INC, GARDENA CA.) DO NOT PERFORATE AT 5/8 BORDERS EACH SIDE LEAVE 1/4 BOARDER EACH END AT BUTT WELD.

NOTES: UNLESS OTHERWISE SPECIFIED

QTY REQD	ITEM NO	CODE IDENT NO	PART OR IDENTIFYING NO	NOMENCLATURE OR DESCRIPTION	SYM
1	20			BEARING 20 MM SERIES 10 SEALED	
8	17			SCREW, FLUSH HD, 6-32 X .75 LG 82°	
4	16			SCREW .25-20 X .75 LG	
8	15			SCREW, FLUSH HD, .25-20 X .88 LG 82°	
1	14		MS16624-1078	RING, RETAINING	
8	12		-33	BOSS	4
1	11		-31	CORE	3
1	10		-29	SEAL PLATE .125 PLATE	5
1	9		-27	SEAL PLATE .312 PLATE	5
1	8		-25	SEAL PLATE .125 PLATE	5
1	7		-23	SEAL PLATE .312 PLATE	5
1	6		-21	CYLINDER INNER	1
1	5		-19	CYLINDER OUTER	1
1	4		-17	HOOP 1.125 x .75 x .049 WALL	4
1	3		-15	HOOP 1.125 x .75 x .049 WALL	4
1	2		-13	FLOOR .375 PLATE	4
1	1		-11	CAP 4.00 DIA	4

PARTS LIST

AIRESEARCH MANUFACTURING COMPANY OF CALIFORNIA
A DIVISION OF THE GARRETT CORPORATION
TORRANCE, CALIFORNIA

CYLINDER ASSY, REGENERATOR

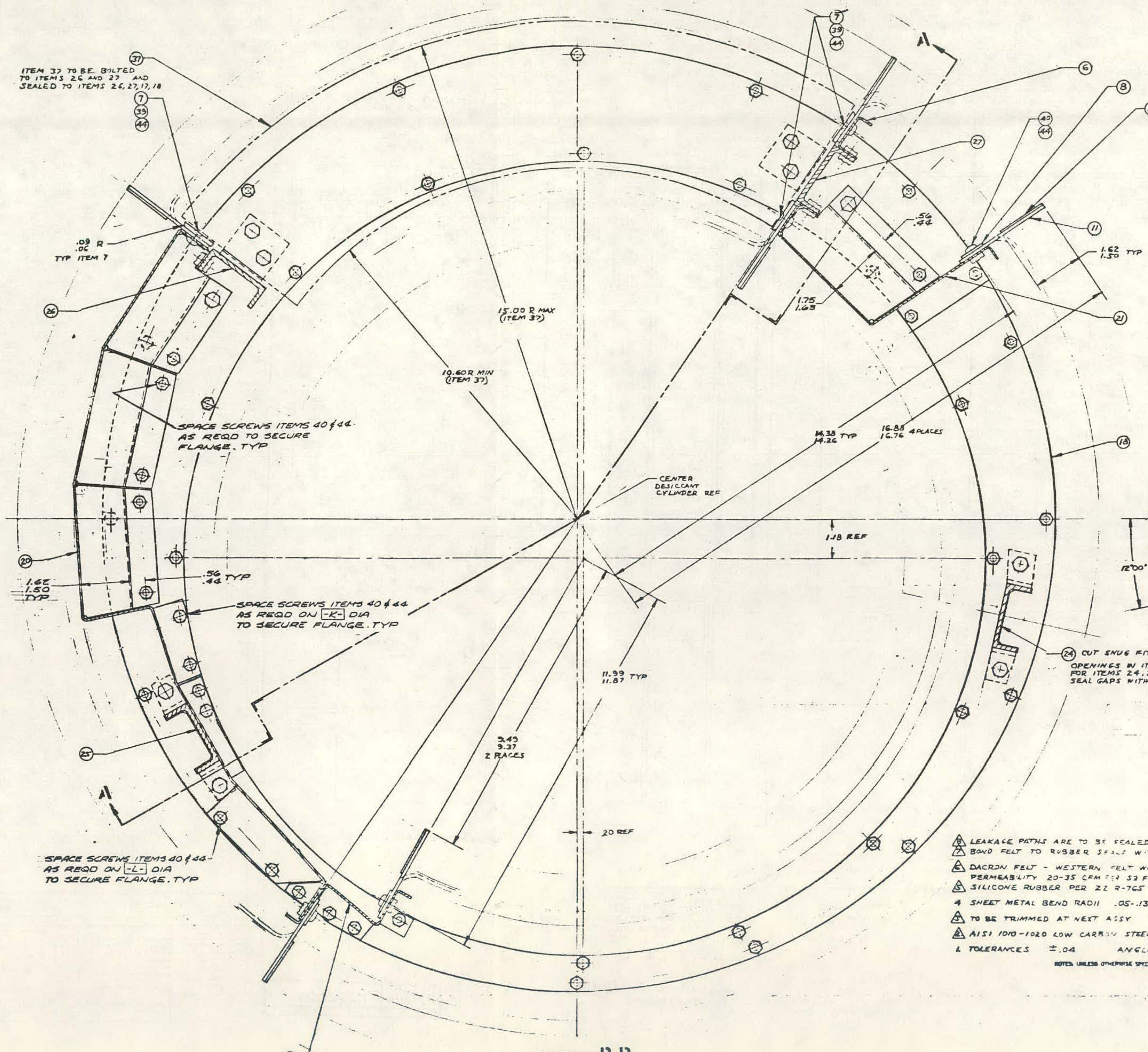
SIZE	CODE IDENT NO	DWG NO
J	70210	2201849

SCALE 1/1 CALC WT 257 LBS SHEET 1 OF 1

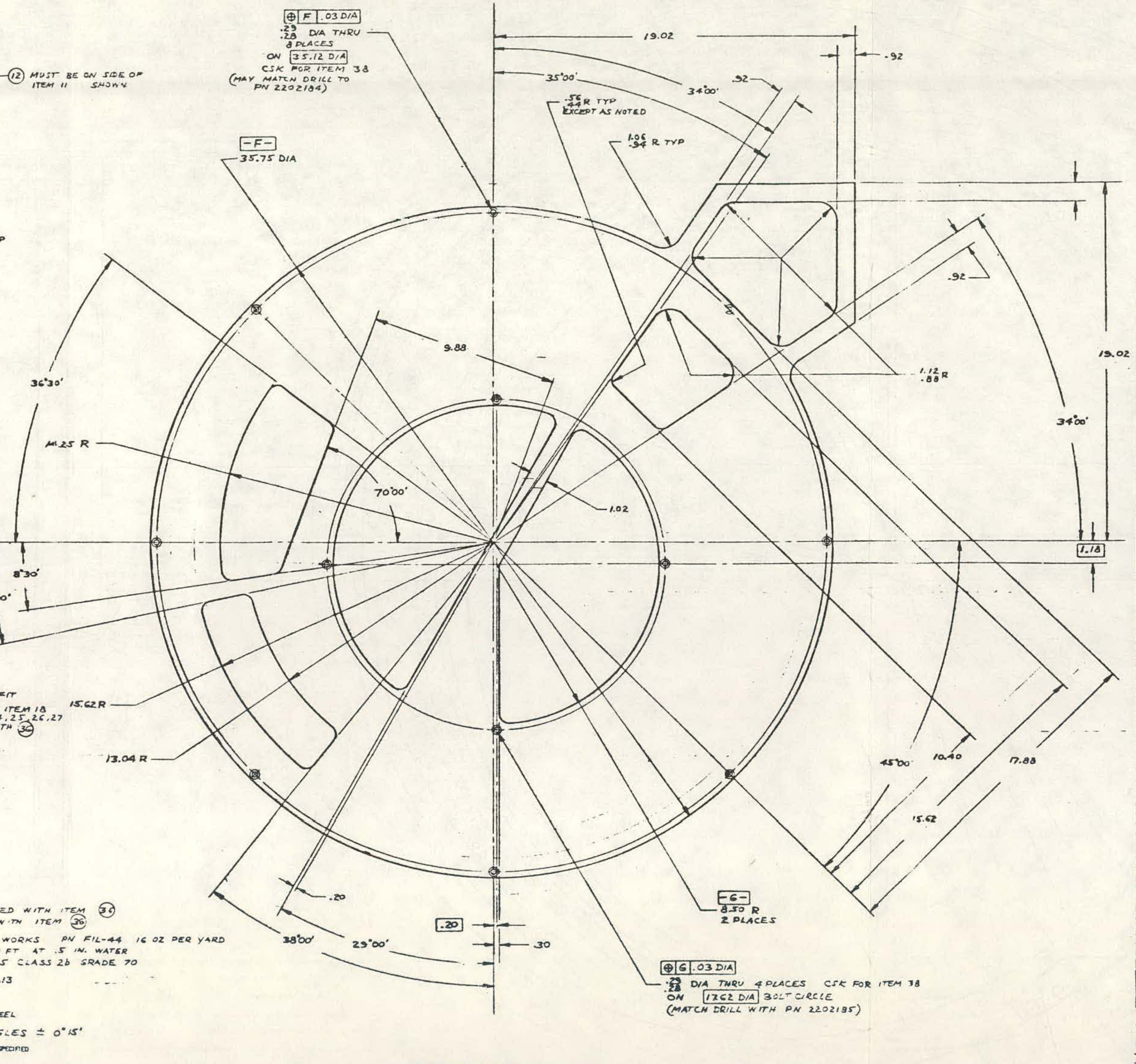
NEXT ASSY	USED ON
2202401-1	2202400-1

APPLICATION

2201849



SECTION 13-13
LOOKING DOWNWARD



DETAIL ITEM 17
SCALE 1/2
SEE SECT H-H FOR ADDITIONAL HOLES

8 LEAKAGE PATHS ARE TO BE SEALED WITH ITEM 36
7 BOND FELT TO RUBBER SEALS WITH ITEM 36
6 DACRON FELT - WESTERN FELT WORKS PN FIL-44 16 OZ PER YARD PERMEABILITY 20-35 CFM PER SQ FT AT .5 IN. WATER
5 SILICONE RUBBER PER ZZ R-765 CLASS 2b GRADE 70
4 SHEET METAL BEND RADII .05-.13
3 TO BE TRIMMED AT NEXT ASSY
2 AISI 1010-1020 LOW CARBON STEEL
1 TOLERANCES ±.04 ANGLES ± 0°15'

NOTES: UNLESS OTHERWISE SPECIFIED

QTY REQD	ITEM NO	CODE IDENT NO	PART OR IDENTIFYING NO	NOMENCLATURE OR DESCRIPTION	SYM
AR	44			NUT 10-32	
AR	43			NUT .25-28	
AR	42			BOLT 10-32 HEX HD ALY STEEL	
AR	41			BOLT .25-28 HEX HD ALY STEEL	
AR	40			SCREW 10-32 PAN HD ALY STEEL	
AR	39			SCREW 10-32 FLAT HD 100°	
AR	38			SCREW .25-28 FLAT HD 100°	
1	37		2202169-1	HEAT EXCHANGER	
AR	36		RS184C01	ADHESIVE	7 8
AR	33			ANGLE 1.00 x 1.00 x .125	2
AR	32			ANGLE 1.50 x 1.50 x .125	2
4	28		-65	FOOT, ANGLE 1.00 x 1.00 x .125	2
1	27		-63	SUPPORT, CHANNEL 2.00 x .75 x .12	2
1	26		-61	SUPPORT, CHANNEL 2.00 x .75 x .12	2
1	25		-59	SUPPORT, CHANNEL 2.00 x .75 x .12	2
1	24		-57	SUPPORT, CHANNEL 2.00 x .75 x .12	2
	23				
	22				
1	21		-51	PARTITION .062 SHEET	2
1	20		-49	PARTITION .062 SHEET	2
1	19		-47	PARTITION .062 SHEET	2
1	18		-45	PLATE, FLOOR .062 SHEET	2
1	17		-43	PLATE, CEILING .125 PLATE	2
1	16		-41	FELT, CIRCULAR .062 SHEET	7 6
1	15		-39	SEAL, CIRULAR .062 SHEET	5
1	14		-37	FELT, CIRCULAR .062 SHEET	7 6
1	13		-35	SEAL, CIRCULAR .062 SHEET	5
6	12		-33	FELT, STRIP .062 THK x 1.56 WIDE	7 6
6	11		-31	SEAL, STRIP .094 THK x 3.00 WIDE	5
3	10		-29	RING, SEAL BACK UP .062 SHEET	2
3	9		-27	RING, SEAL BACK UP .062 SHEET	2
AR	8		-25	STRIP, SEAL BACK UP .062 THK x .75 WIDE	2
3	7		-23	STRIP, SEAL BACK UP .125 THK x 1.00 WIDE	2
1	6		-21	CHANNEL, SEAL BACK UP .049 SHEET	2
	5				
	4				
	3				
	2				
	1				

PART NO. -1 ASSY
PARTS LIST

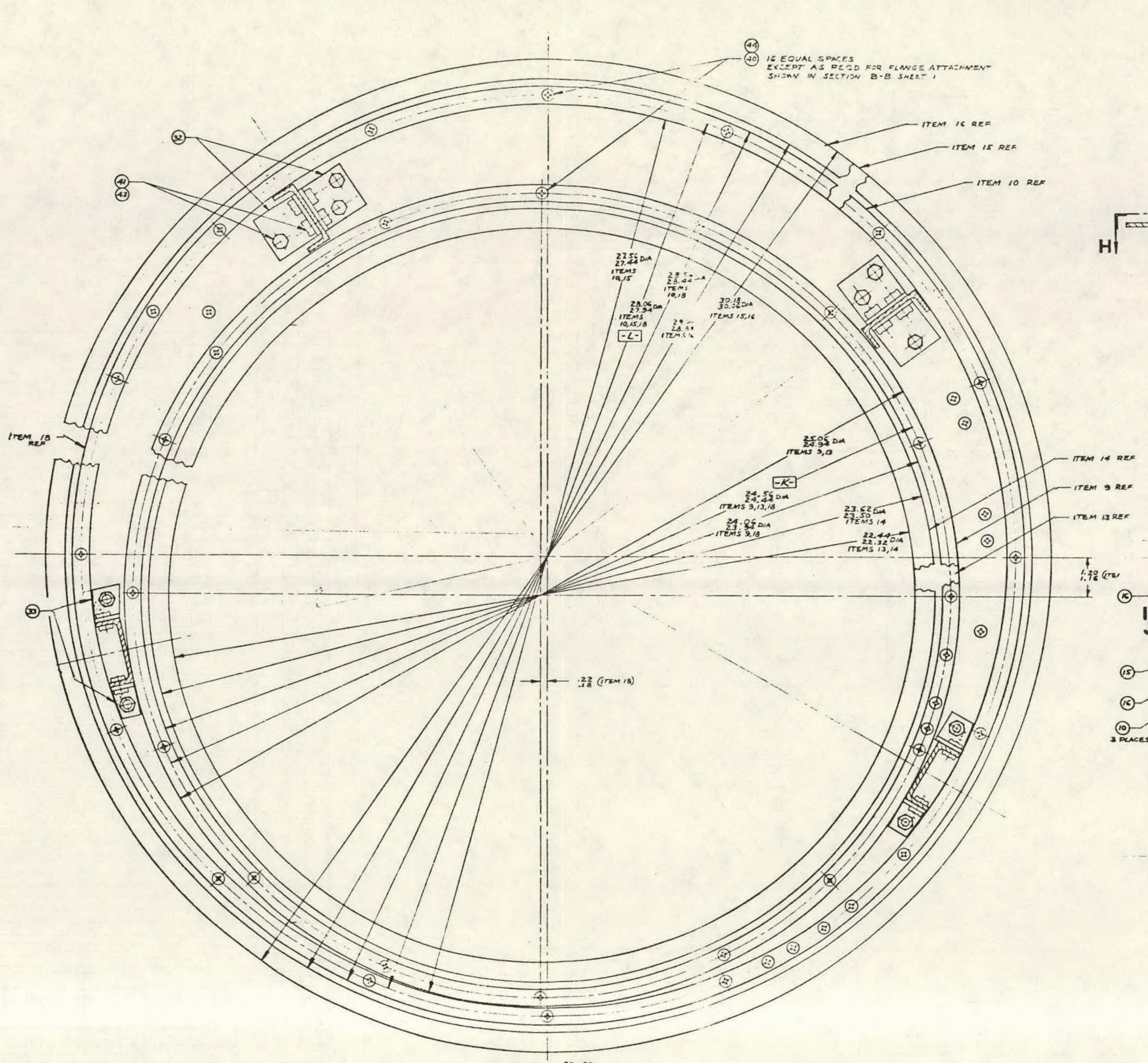
MATERIAL: 2, 5, 6
NO CORROSION PROTECTION REQD

AIRESEARCH MANUFACTURING COMPANY OF CALIFORNIA
A DIVISION OF THE GARRETT CORPORATION
TORRANCE, CALIFORNIA

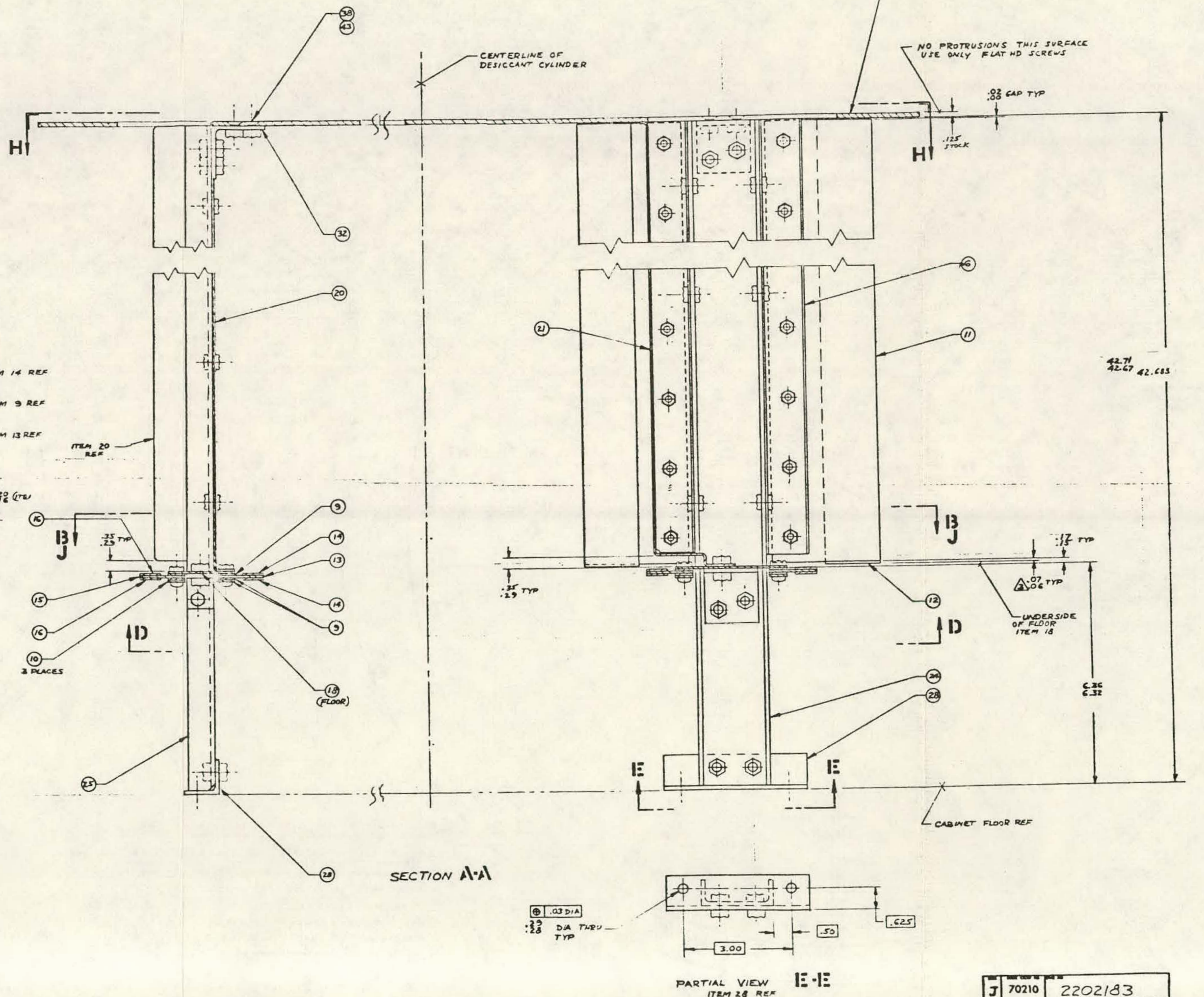
FRAME ASSY, ANNULAR

J | 70210 | 2202183

SCALE 1/1 SHEET 1 OF 3



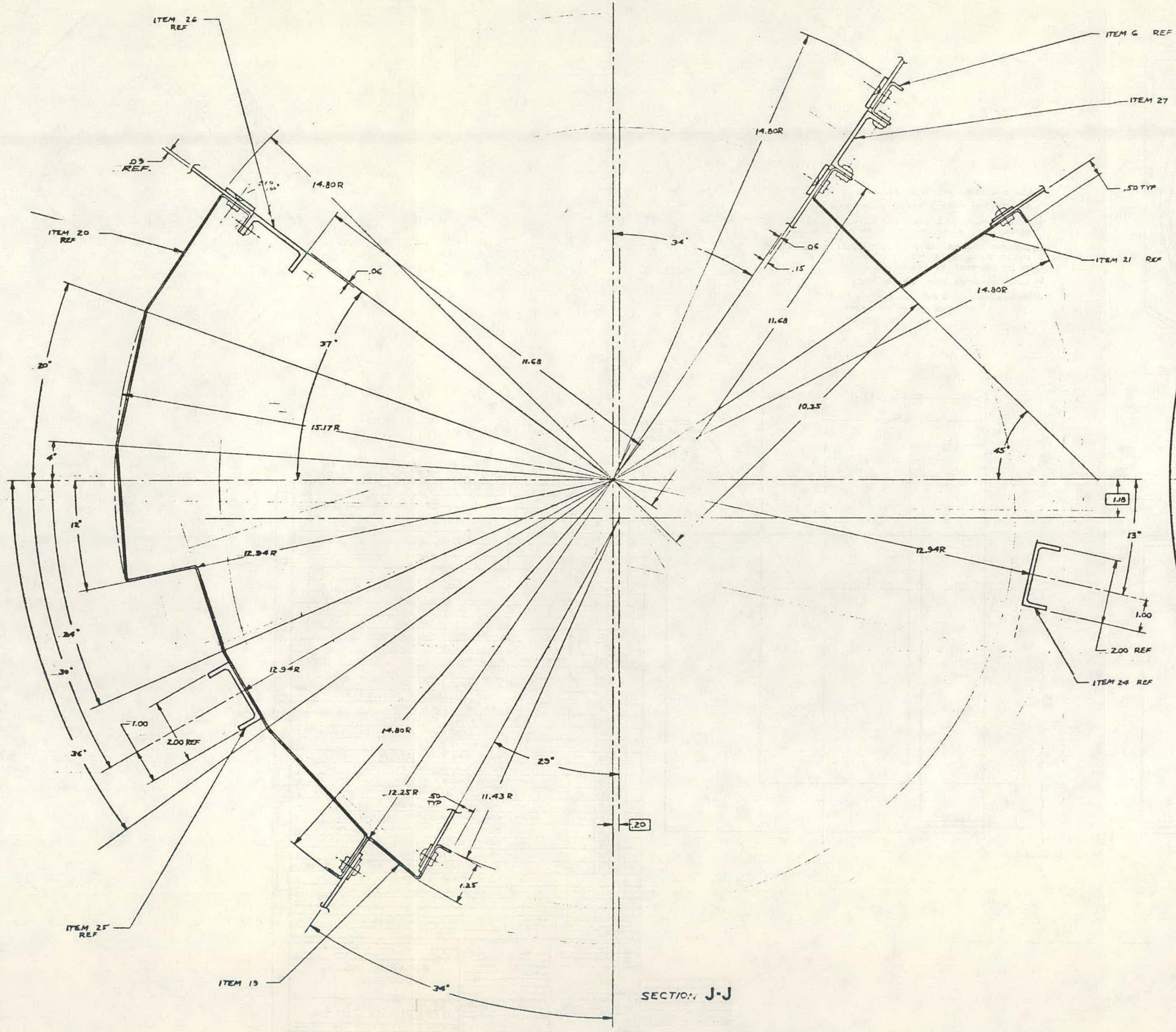
SECTION D-D
LOOKING UPWARD
NOTE: THIS SECTION IS ROTATED 180° FROM SECT B-B



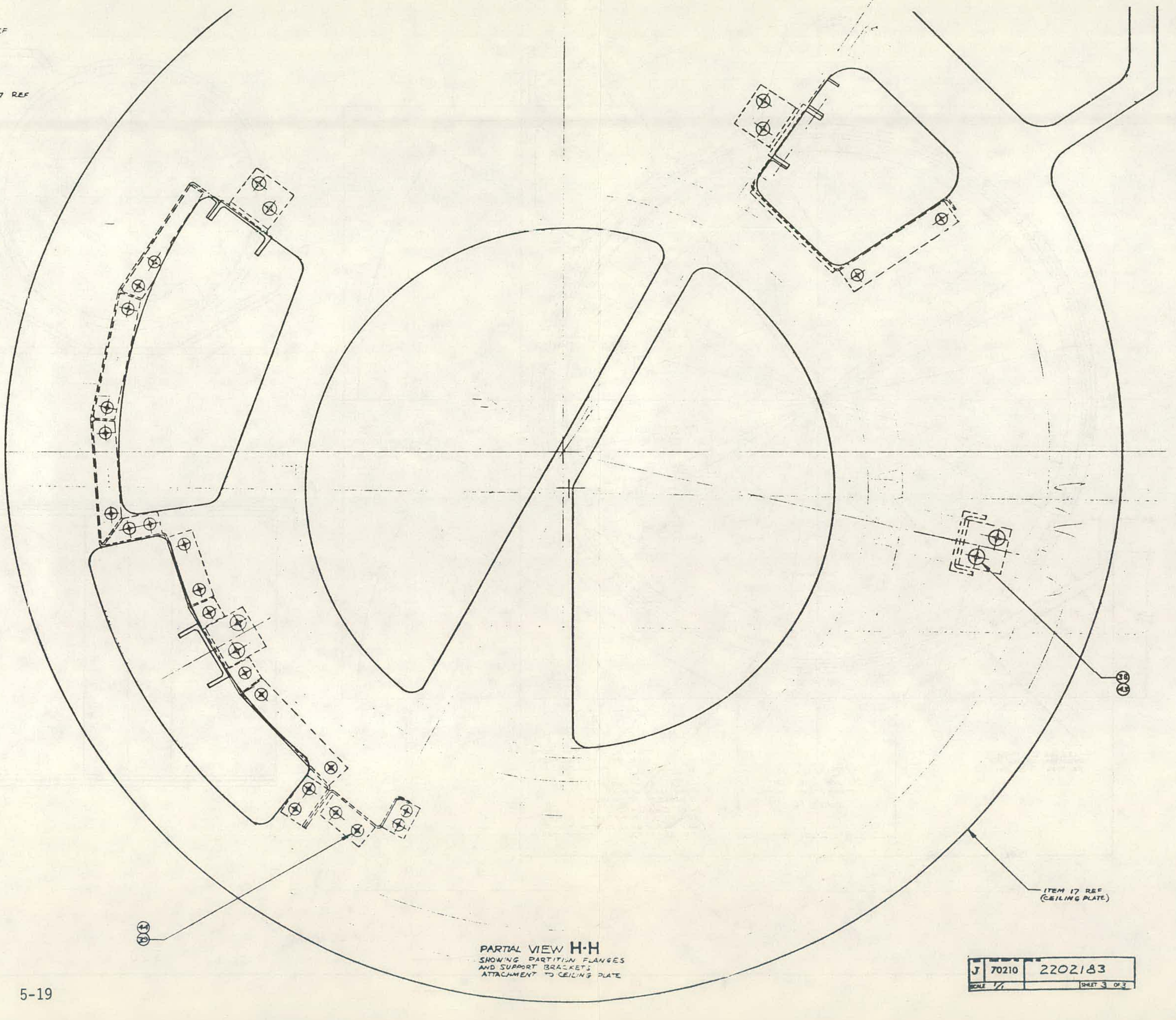
SECTION A-A

PARTIAL VIEW E-E
ITEM 28 REF

J	70210	2202183
SCALE 1/1		SHEET 3 OF 3

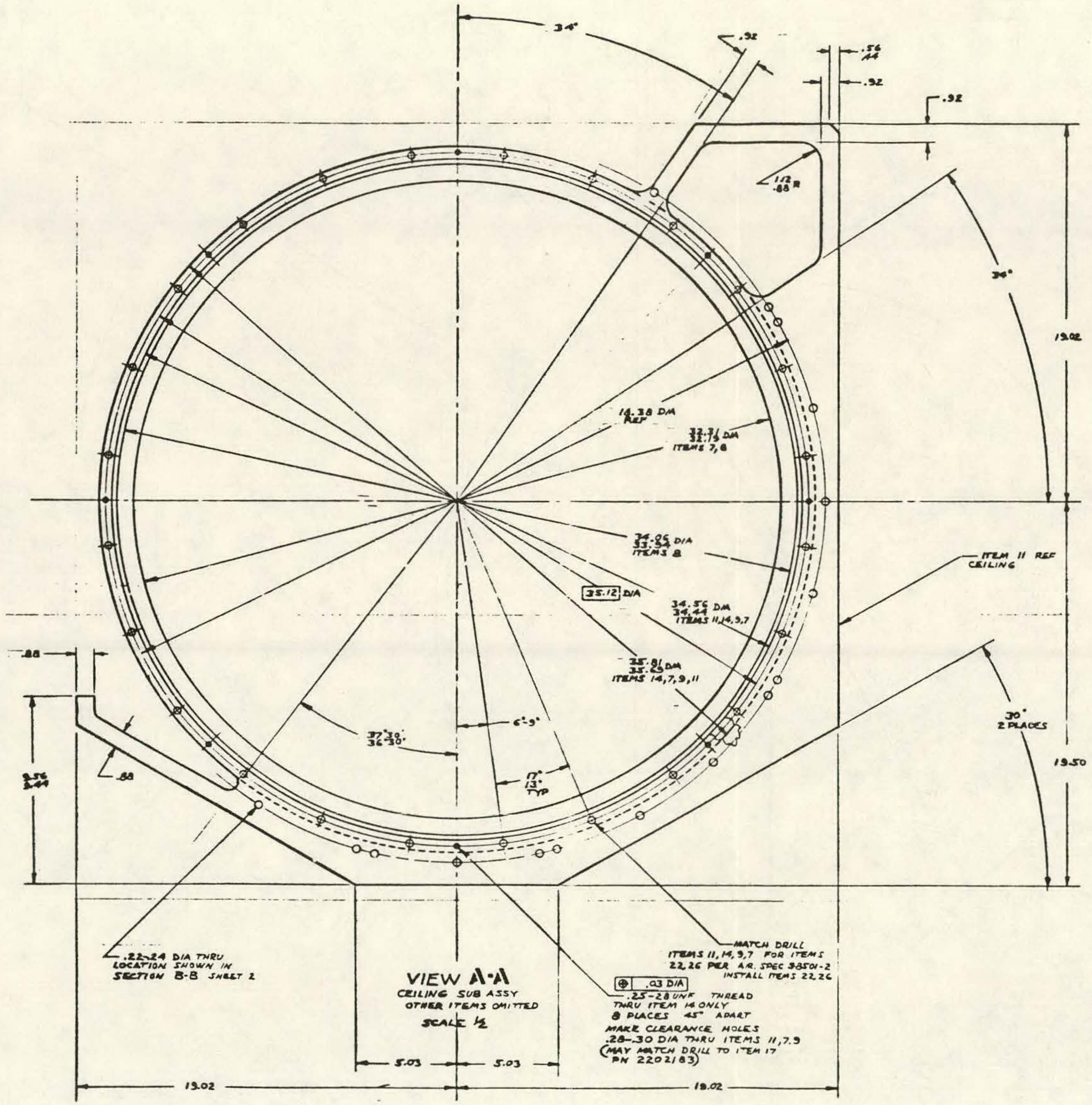


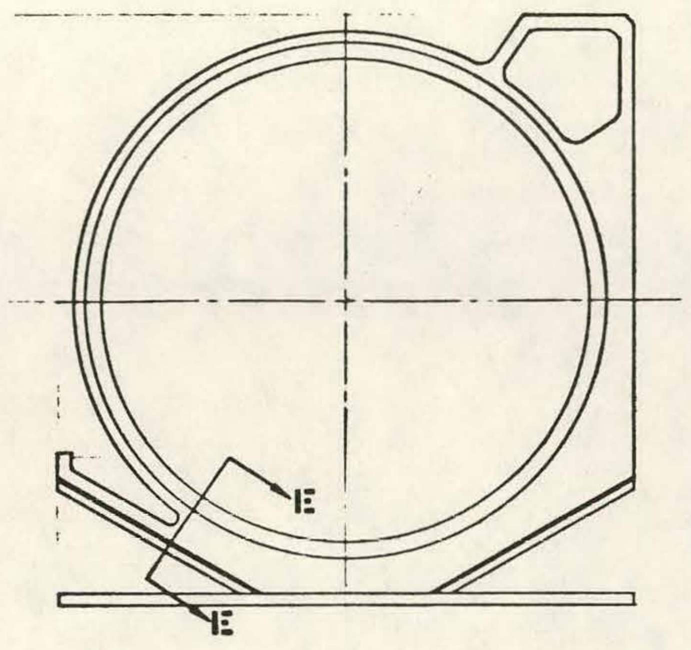
SECTION J-J

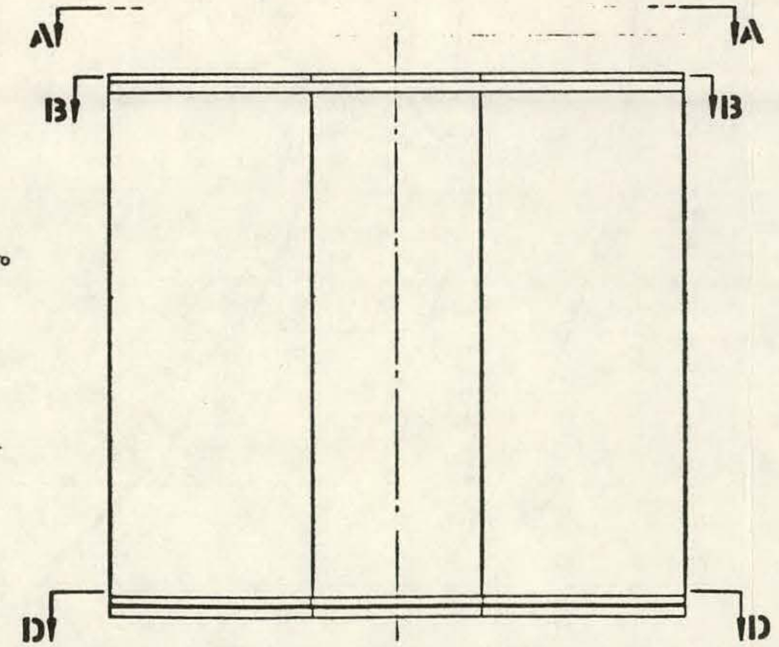


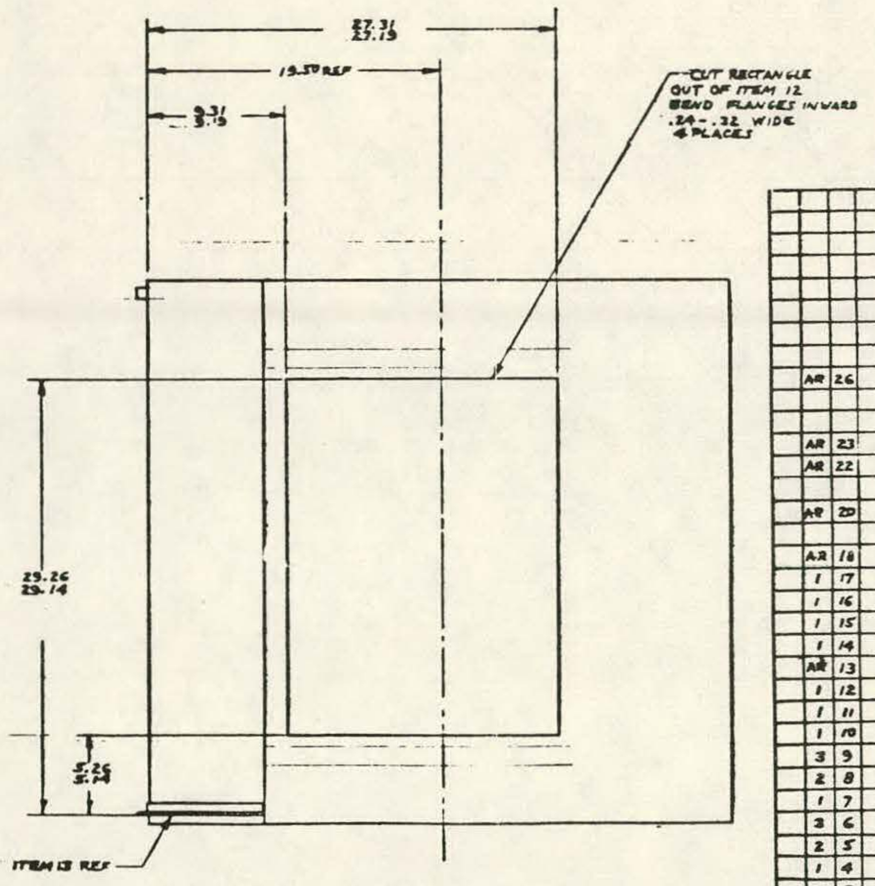
PARTIAL VIEW H-H
SHOWING PARTITION FLANGES AND SUPPORT BRACKETS ATTACHMENT TO CEILING PLATE

J	70210	2202183
SCALE 1/1		SHEET 3 OF 3









1. LEAKAGE PATHS ARE TO BE SEALED WITH ITEM (20)
2. BOND FELT TO RUBBER SEALS WITH ITEM (20)
3. DACRON FELT - WESTERN FELT WORKS PN. FIL-44 16 OZ PER YARD PERMEABILITY 20-35 CFM PER SQ FT AT .5 IN. WATER
4. SILICONE RUBBER PER ZZ-R-765 CLASS 2b GRADE 70
5. SHEET METAL BEND RADIUS .05-.13
6. TO BE TRIMMED AT NEXT ASSY
7. AISI 1010-1020 LOW CARBON STEEL
8. TOLERANCES ±.04 ANGLES ±0°15' UNLESS OTHERWISE SPECIFIED

QTY REQD	ITEM NO	CODE IDENT NO	PART OR IDENTIFYING NO	NOMENCLATURE OR DESCRIPTION	SYM
AR	26			NUT 10-32	
AR	23			SCREW 10-32 PAN HD	
AR	22			SCREW 10-32 FLAT HD 100°	
AR	20		RS184C01	ADHESIVE	7 6
AR	18		-45	ANGLE 1.25 x .75 x .125	2
1	17		-43	TUBE, SQUARE .75 SQ x .049 WALL	2
1	16		-41	PARTITION .049 SHEET	2
1	15		-39	PARTITION .049 SHEET	2
1	14		-37	RING, SEAL BACK UP, UPPER .22-.24 THICK	2
AR	13		-35	ANGLE .75 x .75 x .125	2
1	12		-33	WALL .049 THK SHEET	2
1	11		-31	PLATE, CEILING .062 THK SHEET	2
1	10		-29	PLATE, FLOOR .062 THK SHEET	2
3	9		-27	RING, SEAL BACK UP, UPPER .062 SHEET	2
2	8		-25	FELT, CIRCULAR, UPPER .062 SHEET	6
1	7		-23	SEAL, CIRCULAR, UPPER .062 SHEET	5
3	6		-21	RING, SEAL BACK UP, LOWER .062 SHEET	2
2	5		-19	FELT, CIRCULAR, LOWER .062 SHEET	6
1	4		-17	SEAL, CIRCULAR, LOWER .062 SHEET	5
3	3		-15	STRIP, SEAL BACK UP .062 THK x .75 WIDE	2
3	2		-13	FELT, STRIP .062 THK x 1.56 WIDE	6
3	1		-11	SEAL, STRIP .094 THK x 3.00 WIDE	5

PARTS LIST

2,5,6

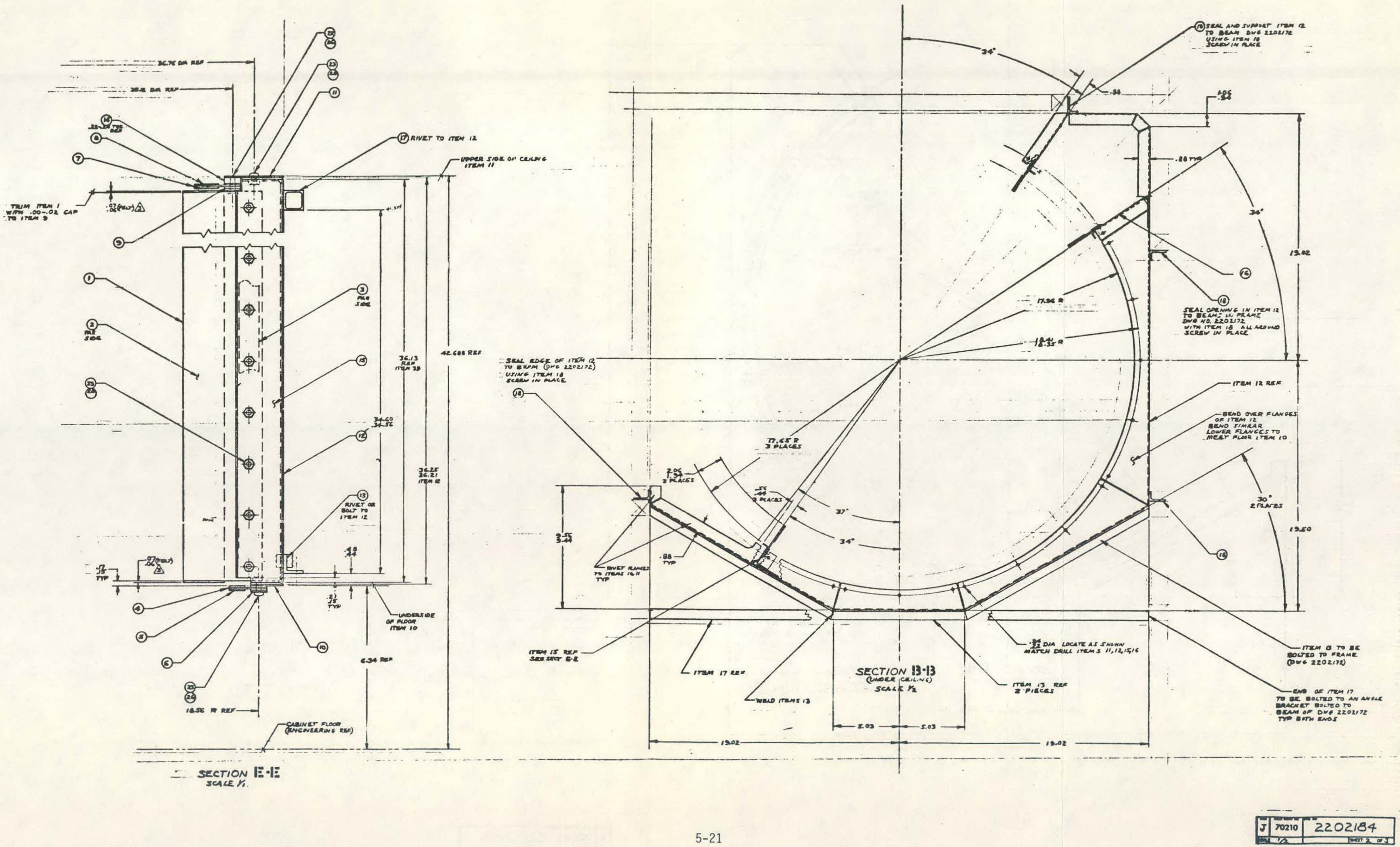
NO CORROSION PROTECTION REQD

2202401-1 | 2202400-1
NEXT ASSY | USED ON
APPLICATION

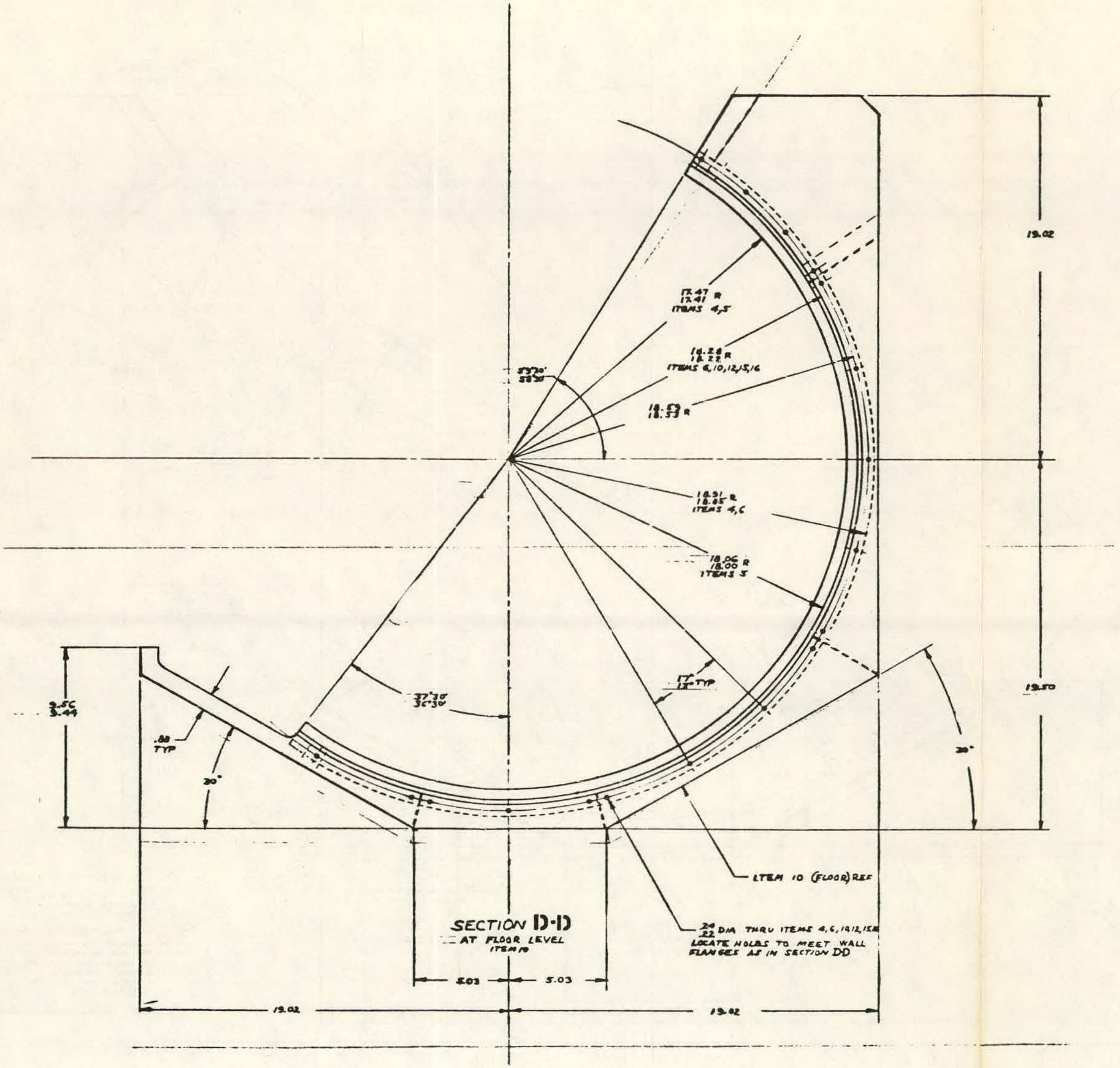
FRAME ASSY, OUTER

70210 | 2202184

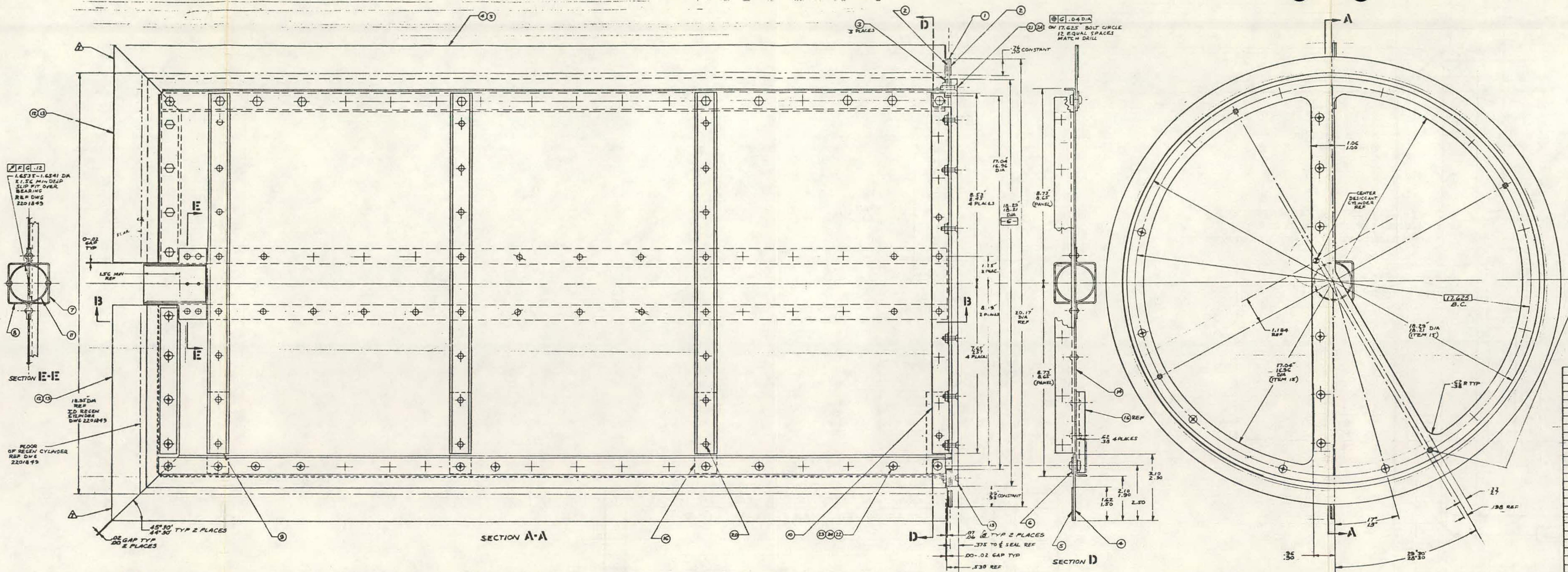
SCALE 1/4 | SHEET 1 OF 3

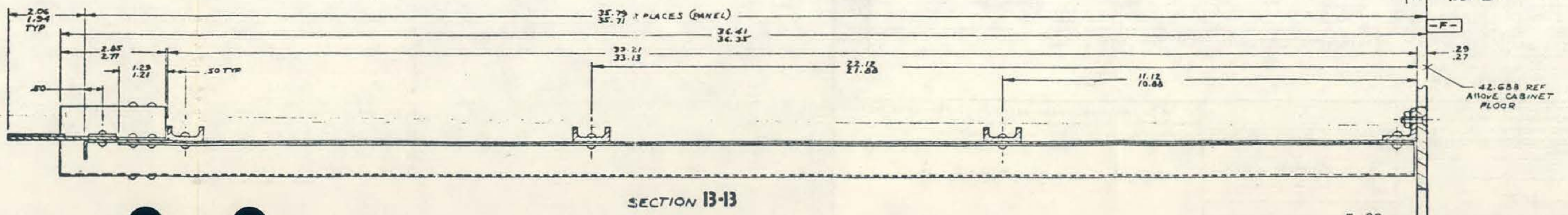


SECTION E-E
SCALE 1/2
SECTION B-B
(UNDER CEILING)
SCALE 1/2
RIVET TO ITEM 12
UPPER SIDE OF CEILING ITEM 11
TRIM ITEM 1 WITH .00-.02 GAP TO ITEM 9
42.688 REF
6.34 REF
18.56 R REF
CABINET FLOOR (ENGINEERING REF)
UNDERSIDE OF FLOOR ITEM 10
RIVET OR BOLT TO ITEM 12
SEAL AND SUPPORT ITEM 12 TO BEAM DWG 2202172 USING ITEM 18 SCREW IN PLACE
SEAL OPENING IN ITEM 12 TO BEAMS IN FRAME DWG NO. 2202172 WITH ITEM 18 ALL AROUND SCREW IN PLACE
SEAL EDGE OF ITEM 12 TO BEAM (DWG 2202172) USING ITEM 18 SCREW IN PLACE
ITEM 12 REF
BEND OVER FLANGES OF ITEM 12 BEND SIMILAR LOWER FLANGES TO MEET FLOOR ITEM 10
17.65 R 3 PLACES
RIVET FLANGES TO ITEMS 16,11 TYP
ITEM 15 REF SEE SECT E-E
ITEM 17 REF
WELD ITEMS 13
MATCH DRILL ITEMS 11,12,15,16
ITEM 13 REF 2 PIECES
ITEM 13 TO BE BOLTED TO FRAME (DWG 2202172)
END OF ITEM 17 TO BE BOLTED TO AN ANGLE BRACKET BOLTED TO BEAM OF DWG 2202172 TYP BOTH ENDS
30° 2 PLACES
19.62
19.50
19.02
5.03
J 70210 2202184



J	70210	2202184
1/2		SHEET 3 OF 3





10. LEAKAGE PATHS ARE TO BE SEALED WITH ITEM 19
9. ALL SHEET METAL BEND RADII : .05-.13.
△8 FINAL TRIM OF ITEM 5 TO BE AT FINAL ASSY.
△7 FINAL TRIM OF ITEMS 12 & 13 CORNERS TO BE AT FINAL ASSY.
△6 NO CORROSION PROTECTION REQD
△5 DACRON FELT - WESTERN FELT WORKS PN. FIL-44 16 OZ PER YARD
PERMEABILITY 20-35 CFM/SQFT AT .5 IN. WATER
△4 SILICONE RUBBER PER ZZ-R-765 CLASS 2b GRADE 70
△3 BOND FELT ITEMS 5 & 13 TO SEALS ITEMS 4 & 12 WITH ITEM 19
△2 AISI 1010-1020 STEEL.
△1 TRIM LENGTH AT NEXT ASSY

NOTES: UNLESS OTHERWISE SPECIFIED

QTY REQD	ITEM NO	CODE IDENT NO	PART OR IDENTIFYING NO	NOMENCLATURE OR DESCRIPTION	SYM
46	24			NUT, 10-32	
34	23			WASHER FOR NO. 10 SCREW	
46	22			SCREW, PAN HD 10-32	
18	21			SCREW FLAT HD 82° 10-32	
62	20			RIVET, UNIVERSAL HD .125 DIA	
AR	19		RS184C01	ADHESIVE	△3
4	16	△2	-41	CHANNEL 1.00 X .38 X .12 THK X 3.62 LG	
1	15	△2	-39	PLATE, .281 THK X 18.29 DIA	
1	14	△2	-37	PANEL, .049 THK X 13.70 X 36.41	
2	13	△5	-35	FELT, STRIP .062 THK X 1.56 WIDE X 10.00 LG	△7
2	12	△4	-33	SEAL, STRIP .094 THK X 3.00 WIDE X 10.00 LG	△1 △7
1	11	△2	-31	TUBE, 1.75 OD X .049 WALL X 2.85 LG	
1	10	△2	-29	ANGLE .88 X .88 X .12 THK X 16.18 LG	
4	9	△2	-27	CHANNEL 1.00 X .38 X .12 THK X 16.18 LG	
1	8	△2	-25	HAT SECTION .049 SHEET X 36.14 LG	
1	7	△2	-23	HAT SECTION .049 SHEET X 2.85 LG	
AR	6	△2	-21	STRIP, SEAL BACK UP .062 THICK X .75 WIDE	
2	5	△5	-19	FELT, STRIP .062 THK X 1.56 WIDE X 37.57 LG	△8
2	4	△4	-17	SEAL, STRIP .094 THK X 3.00 WIDE X 37.50 LG	△1 △3
3	3	△2	-15	RING, SEAL BACK UP .062 SHT X 18.29 DIA	
2	2	△5	-13	FELT, CIRCULAR .062 THK 20.23 OD	△3
1	1	△4	-11	SEAL, CIRCULAR .062 SHEET 20.23 OD	△3

PARTS LIST

AIRESEARCH MANUFACTURING COMPANY OF CALIFORNIA
A DIVISION OF THE GARRETT CORPORATION
TORRANCE, CALIFORNIA

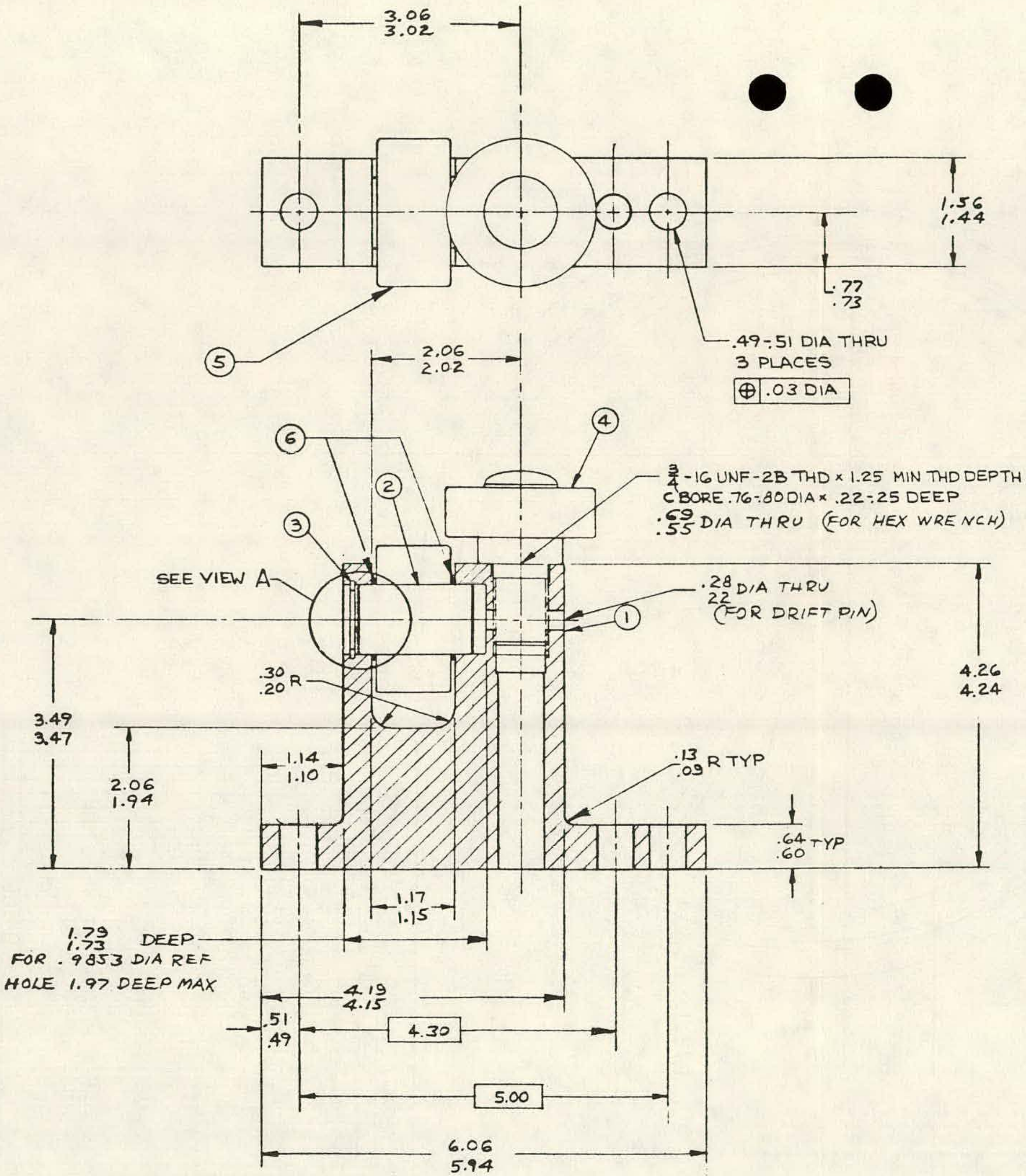
PANEL ASSY, CENTER

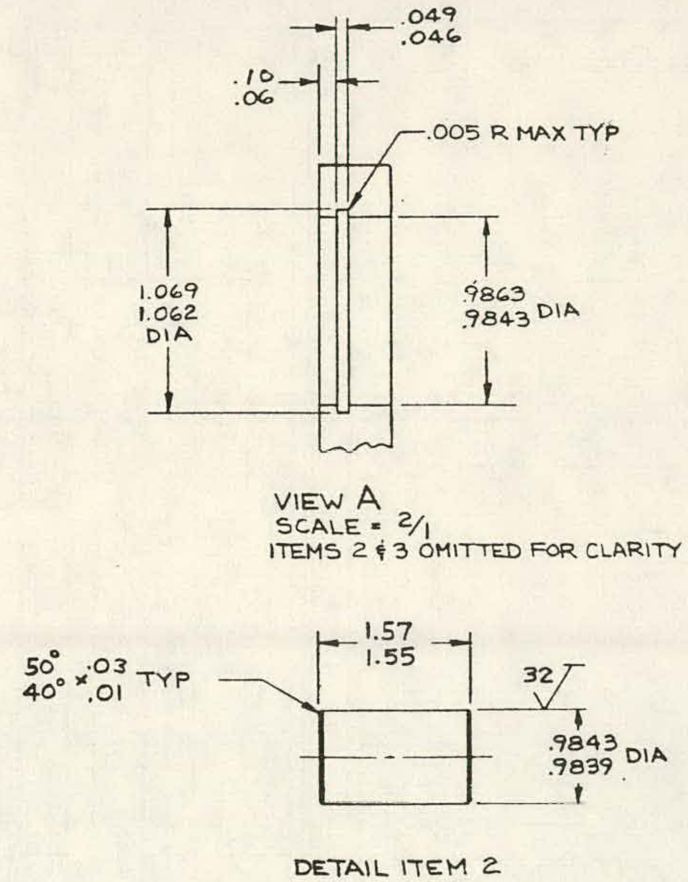
NOTED

REQD	NEXT ASSY	USED ON
R-1	2202493-1	2202400-1

APPLICATION

SIZE J | CODE IDENT NO 70210 | DWG NO 2202185
SCALE 1/4 | SHEET 1 OF 1





QTY REQD	ITEM NO	PART NO	NOMENCLATURE OR DISCRIPTION	SYM
2	6	S8157CX332-062	WASHER .985-1.000 ID × 1.11-1.15 OD × .062 THICK	1
1	5	5305SB	BEARING, BALL, SEALED 25MM (MRC)	
1	4	905969	BEARING	3
1	3	MS16625-1100	RING, RETAINING	
1	2	2202186-5	SHAFT	2
1	1	2202186-3	FRAME	1

PART NO 2202186-1

3. NEW DEPARTURE BEARING
2. MATL- STEEL
1. MATL- AISI 1010-1020 LOW CARBON STEEL.

NOTES: UNLESS OTHERWISE SPECIFIED.

8(-1)	2202401-1	2202400-1
REQD	NEXT ASSY	USED ON
APPLICATION		

UNLESS OTHERWISE SPECIFIED:
BURR CONTROL PER SC653
STD INTERPRETATIONS PER PI86
IDENTIFICATION MARKING PER MC16

MATERIAL: 1

FINISH PROCESS: NO CORROSION PROTECTION REQD

HEAT TREATMENT

CONTRACT NO

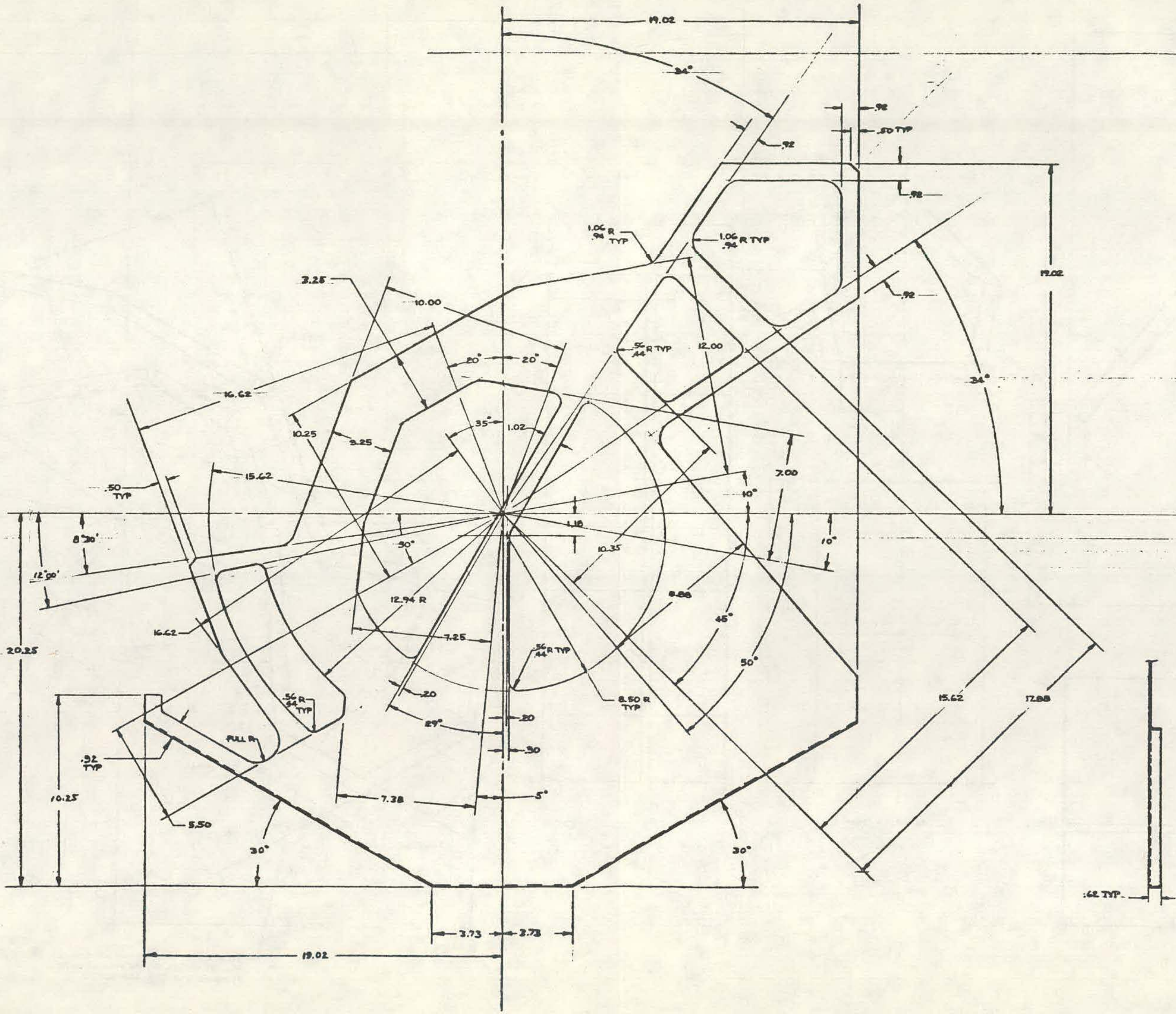
PREPARED BY [signature] 10-19-76
CHK
DESIGN [signature] 78-10-20
VALUE ENGR
MATL
STRESS
APVD
DESIGN SUPERVISOR
PROJECT ENGINEER
GOVERNMENT APVD

AIRESEARCH MANUFACTURING COMPANY OF CALIFORNIA
A DIVISION OF THE GARRETT CORPORATION
TORRANCE, CALIFORNIA

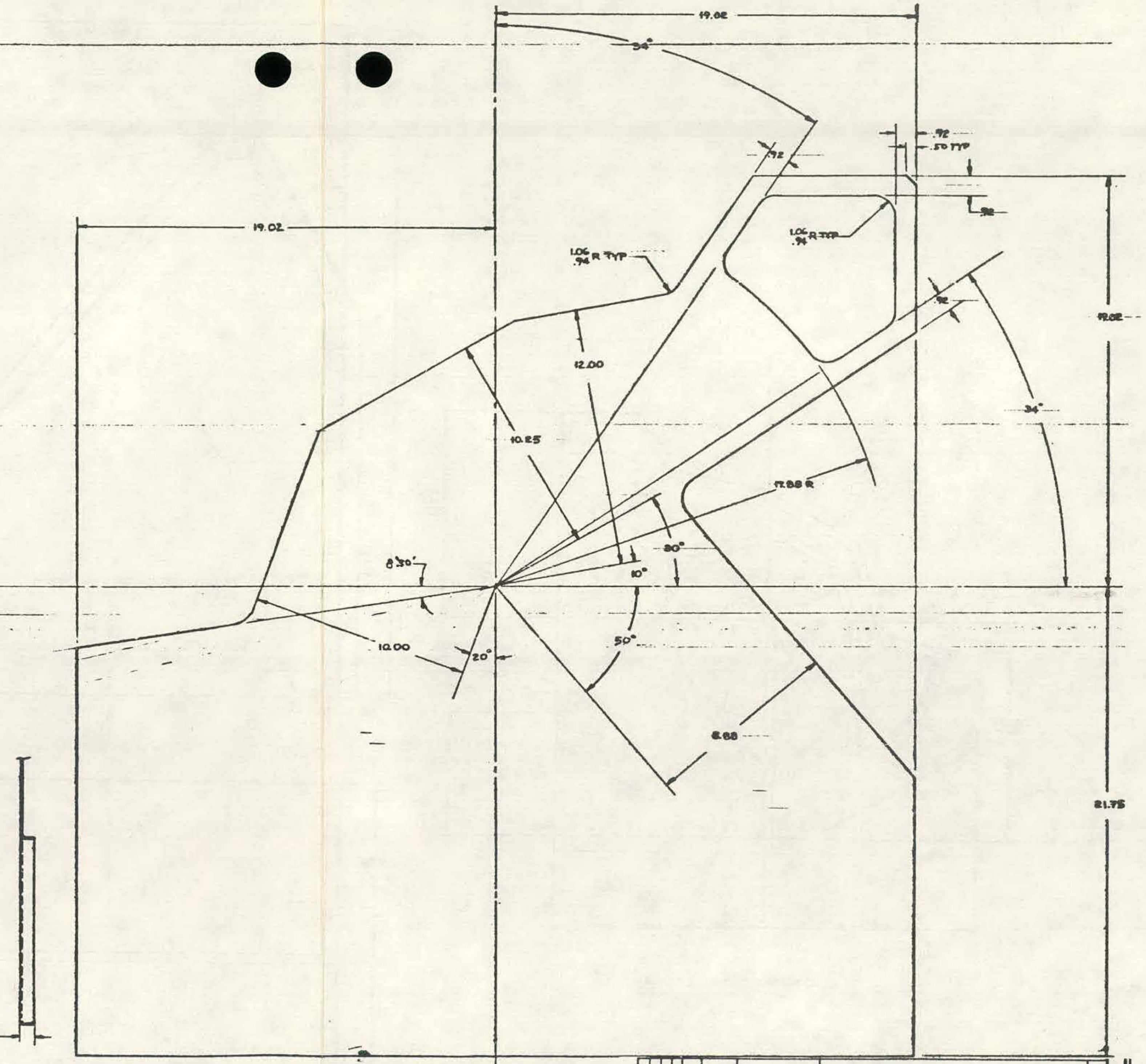
FRAME ASSY, ROLLER

SIZE D | CODE IDENT NO 70210 | DWG NO 2202186

SCALE 1/1 | SHEET | OF



DETAIL OF ITEM 2



DETAIL OF ITEM 1

3. APPLY ITEM 10 BETWEEN FLANGES & ITEMS 1 & 2 AT ASSY.
3. APPLY ITEMS 8 & 9 TO SURFACES INDICATED BY "Δ".
2. METHOD OF FABRICATION OPTIONAL.
1. TOLERANCES ±.04 ANGLES ± 0°15'

NOTES: UNLESS OTHERWISE SPECIFIED

QTY REQD	ITEM NO	CODE IDENT NO	PART OR IDENTIFYING NO	NOMENCLATURE OR DESCRIPTION	SYM
A/R	10		RS184C01	ADHESIVE	
A/R	9		373	ADHESIVE RUBATEX	
A/R	8			INSULATION .50 THK RUBATEX	
1	5		-11	PARTITION .049 SHEET	
1	4		-9	DUCT .049 SHEET	
1	3		-7	PARTITION .049 SHEET	
1	2		-5	PLATE, FLOOR [illegible] SHEET	
1	1		-3	PLATE, [illegible] SHEET	
	-1			ASSY	

PARTS LIST

AIRESEARCH MANUFACTURING COMPANY OF CALIFORNIA

MATERIAL: AISI 1010-1020 GAL SHEET STL

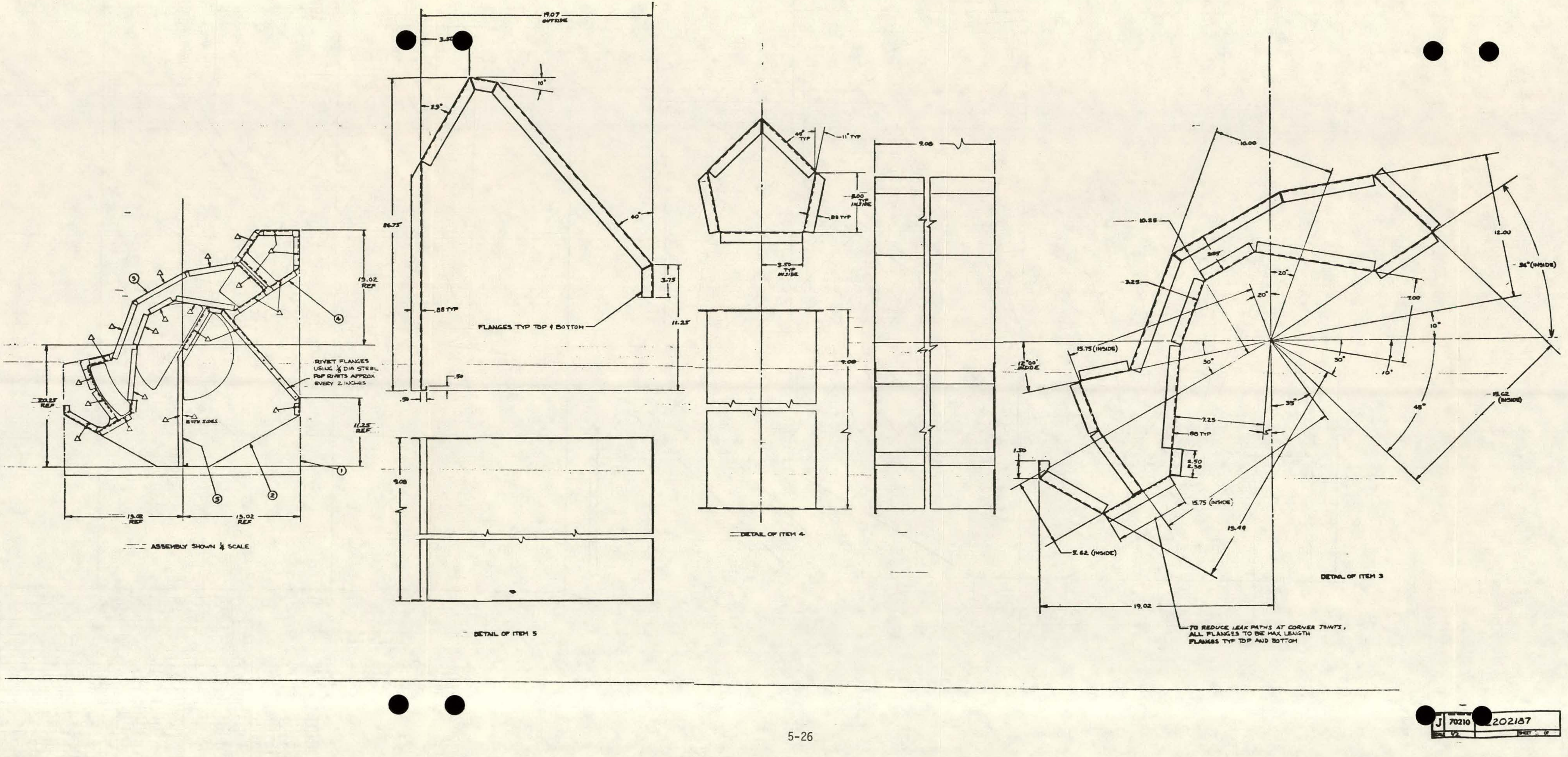
DUCT ASSY, PRECONDITIONING AIR

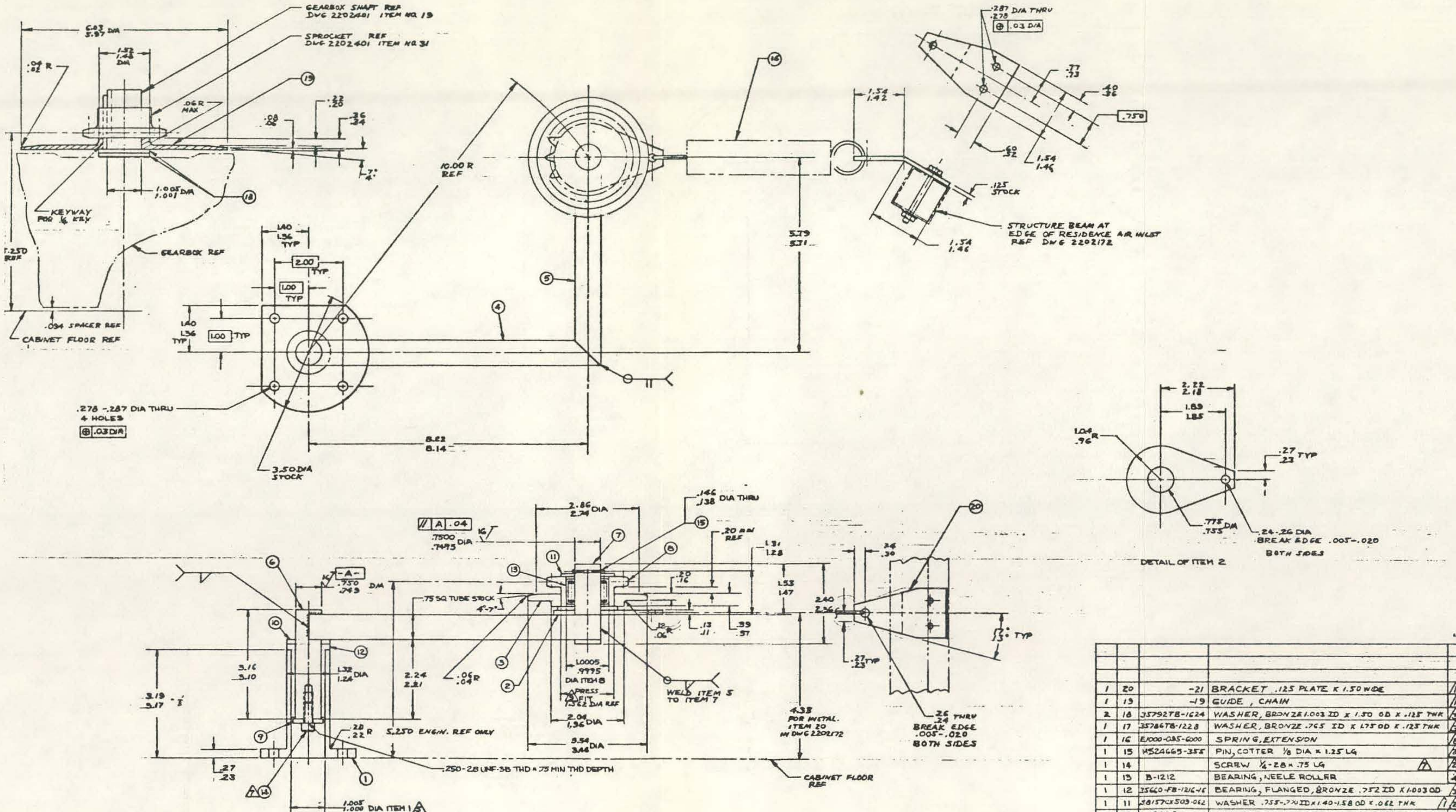
J 70210 2202187

SCALE 1/8 SHEET 1 OF 2

1	2202401-1	2202400-1
REQD	NEXT ASSY	USED ON

APPLICATION


RIVET FLANGES USING 1/8 DIA STEEL POP RIVETS APPROX EVERY 2 INCHES
BOTH SIDES
ASSEMBLY SHOWN 1/4 SCALE
FLANGES TYP TOP & BOTTOM
DETAIL OF ITEM 5
DETAIL OF ITEM 4
DETAIL OF ITEM 3
TO REDUCE LEAK PATHS AT CORNER JOINTS, ALL FLANGES TO BE MAX LENGTH FLANGES TYP TOP AND BOTTOM
J 70210 2202187



QTY REQD	ITEM NO	PART NO.	NOMENCLATURE OR DESCRIPTION	
1	20	-21	BRACKET .125 PLATE X 1.50 WIDE	1
1	19	-19	GUIDE , CHAIN	1
2	18	35792TB-1624	WASHER, BRONZE 1.003 ID X 1.50 OD X .125 THK	3
1	17	35786TB-1228	WASHER, BRONZE .765 ID X 1.75 OD X .125 THK	3
1	16	E1000-085-6000	SPRING, EXTENSION	8
1	15	MS24665-355	PIN, COTTER 1/8 DIA X 1.25 LG	2
1	14		SCREW 1/4-28 X .75 LG 7	2
1	13	B-1212	BEARING, NEELE ROLLER	4
1	12	35660-FB-1216-16	BEARING, FLANGED, BRONZE .752 ID X 1.003 OD	3
1	11	58157CX503-062	WASHER .755-.770 ID X 1.40-1.58 OD X .062 THK 1	2
1	10	18858	WASHER THRUST .75-.77 ID X 1.25-1.35 OD X .188 THK	3
1	9	58153-48-125	WASHER .255-.270 ID X .85-.97 OD X .125 THK 1	2
1	8	-17	SPROCKET 40-1/2 PITCH 13 TEETH 40B13 15692	5
1	7	-15	SHAFT .75 DIA	2
1	6	-13	SHAFT .75 DIA	2
1	5	-11	ARM .75 SQUARE X .049 WALL	1
1	4	-9	ARM .75 SQUARE X .049 WALL	1
1	3	-7	GUIDE .375 PLATE	1
1	2	-5	PLATE .25 PLATE	1
1	1	-3	HOUSING 3.50 DIA BAR	1

9 .0008-.0024 INTERFERENCE FIT REQD

8 ASSOCIATED SPRING CORP

7 USE POSITIVE THREAD LOCKING DEVICE (LOCKTITE, DEFORMED THD, ETC)

6 .001-.003 CLEARANCE REQD AFTER PRESSING BUSHING INTO HOUSING (REF)

5 PRESS FIT .0015-.0030 INTERFERENCE REQD (REF)

4 PURCHASE FROM THE TORRINGTON CO, TORRINGTON, CONNECTICUT.

3 PURCHASE FROM BOSTON GEAR

2 MATL- STEEL

1 MATL- AISI 1010-1020 LOW CARBON STEEL

NOTES, UNLESS OTHERWISE SPECIFIED

PART NO 2202198-1

NO CORROSION PROTECTION REQD

AIRESEARCH MANUFACTURING COMPANY OF CALIFORNIA
A DIVISION OF THE GARRETT CORPORATION
TORRANCE, CALIFORNIA

IDLER ARM ASSY

E 70210 2202198

SHEET 1 OF 1