TR7020-009-91

DE/FC07-89ID12826-1

DOE/ID/12826--1

DE91 009987

RESEARCH AND DEVELOPMENT OF INDUSTRIAL DRYING CONCEPTS USING A SUPERHEATED STEAM ATMOSPHERE WITH EXHAUST RECOMPRESSION

PHASE I – FINAL REPORT
Report Period: October 1989 to December 1990

January 1991

Work Performed Under Contract DE/FC07-89ID12826
October 1989 to June 1992

Prepared for
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1. EXECUTIVE SUMMARY

In the United States, industrial drying accounts for approximately 1.5 quads of energy use per year. Annual industrial dryer expenditures are estimated to be in the $500 million range. Clearly, industrial drying is a significant energy and monetary expense for the United States industrial complex. For the thermal drying processes where water is to be removed via evaporation from the feedstock, attempts have been made to reduce the consumption of energy using exhaust waste heat recovery techniques, improved dryer designs, or even the deployment of advanced mechanical dewatering techniques. Despite these efforts, it is obvious that a large amount of thermal energy is often still lost if the latent heat of evaporation from the evaporated water cannot be recovered and/or in some way be utilized as direct heat input into the dryer.

Under Contract No. DE/FC07-89ID12826 with the Department of Energy, Tecogen Inc. is conducting research and development on an industrial drying concept. The concept utilizes a superheated steam drying atmosphere with exhaust steam recompression to recover the latent heat in the exhaust that would otherwise be lost. This approach has the potential to save 55 percent of the energy required by a conventional air dryer. Work on Phase I: Feasibility Investigation, has been completed and the results of this work are given in this Phase I Final Report.

One of the steam drying systems with recompression studied in this first phase is illustrated in Figure 1.1. Superheated steam is circulated by a fan through the heat exchangers and the drying chamber. In the drying chamber, moisture is thermally driven from the product and carried off by the recirculating superheated steam. A portion of the superheated steam exhausted from the drying chamber (equal to the amount of moisture removed) is taken off and compressed. The compressed steam then gives up its heat of compression and latent heat of evaporation in the heat exchanger. This is the primary, and possibly only, source of heat for the drying process. If economics dictate, an auxiliary fossil-fuel-fired heater may be provided to reduce the size and cost of the steam recompression system. The liquid condensed in the heat exchanger passes through an expansion valve and is then vented.

An alternative to the directly heated steam atmosphere drying system identified previously is an indirectly heated steam drying system, which also utilizes a steam recompression system.

A flow schematic of the proposed indirectly heated superheated steam drying system is illustrated in Figure 1.2. In operation, wet feed is introduced into the drying chamber where a superheated steam atmosphere enables water to be evaporated from the particles. In addition, a steam jacket around the dryer also transfers heat via conduction to the particles to further enhance evaporation.

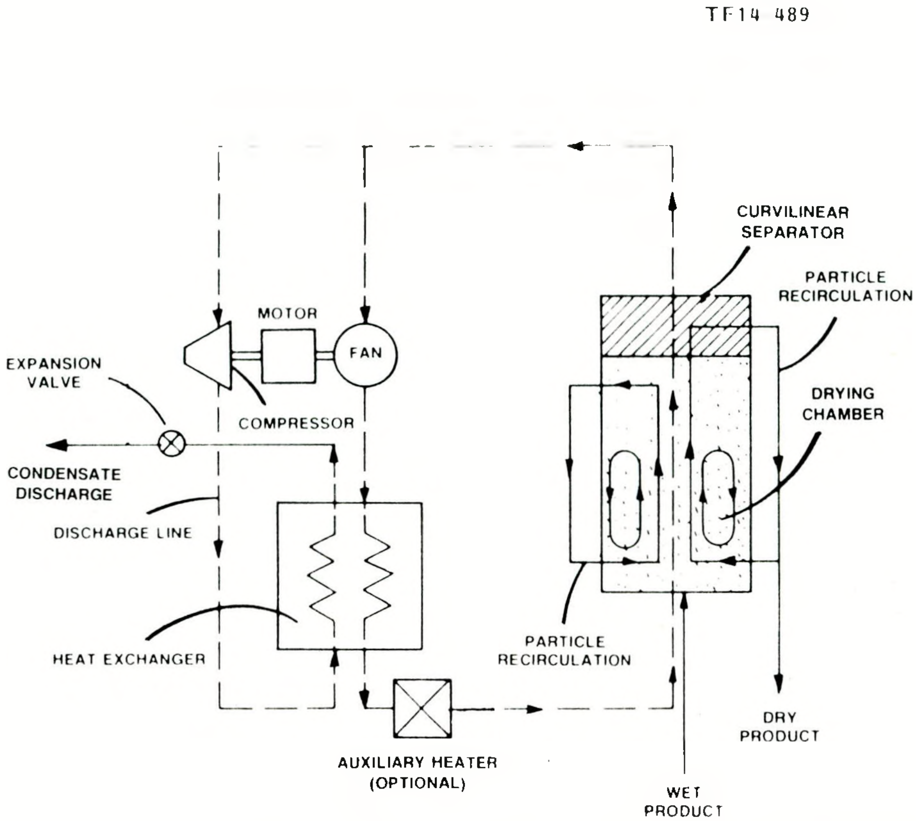

Figure 1.1 Concept for Direct Superheated Steam Dryer with Steam Recompression

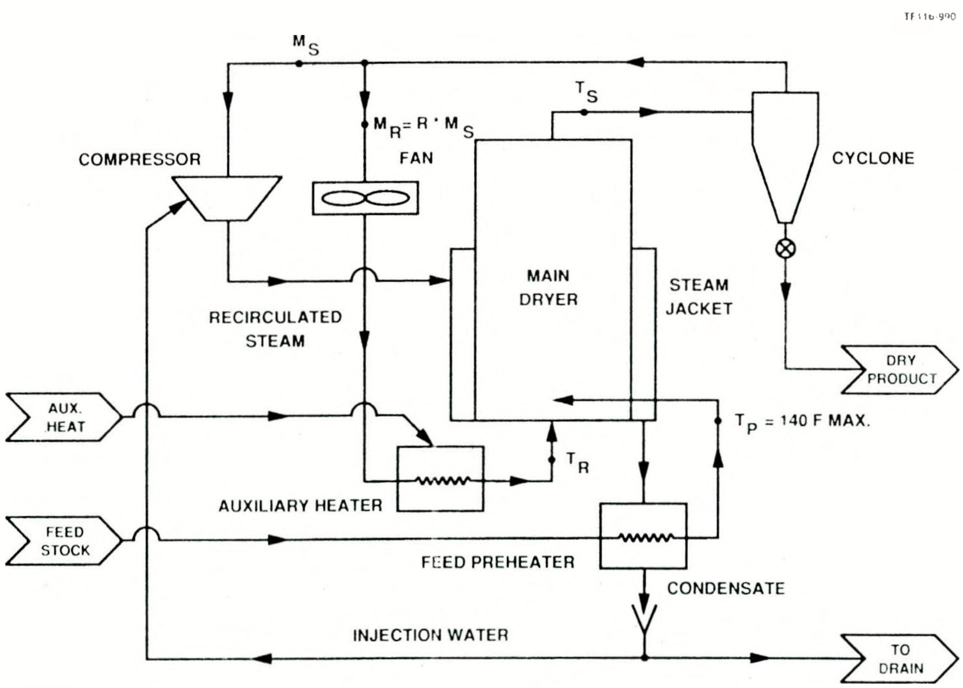

Figure 1.2 Indirect Steam Atmosphere Dryer System with Steam Recompression

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The steam that leaves the dryer is recirculated through a fan, and if necessary, is heated by an auxiliary heater. This circulating steam is introduced into the chamber in such a manner that a vortex pattern is set up, enabling the feedstock particles to circulate in an internal and external flow field (see Steam Dryer Design). Once the particles are dry, they are able to escape and can then be separated from the transport steam by a cyclone separator for collection.

A portion of the transport steam, about 20 percent, is compressed and sent to the vessel jacket where it is condensed as it gives up its latent heat to the feedstock. Thus, all the latent heat in the moisture evaporated from the feedstock is recycled, and only cold condensate leaves the system.

An innovative feature used in both concepts is the unique design of the drying chamber. High heat transfer and drying rates are achieved by intimate contact of the superheated steam with the particles being dried. Through high internal and external recirculation rates, the residence time of the particles in the drying atmosphere is increased substantially over levels normally obtained in a circulating fluid bed. All internal and external recirculation is accomplished by the pressure differentials created in the drying chamber by a forced vortex flow pattern. No separate fans are required to achieve the recirculation. Tecogen's new steam dryer can be identified as an inertial reactor with internal and external separation or IRIS.

A second innovative feature of the drying chamber design relates to the way the particles are efficiently separated from the superheated steam. This is accomplished by the use of a specially designed curvilinear separator. Good separation of the particles is important not only to prevent escape of dried particles but also to prevent fouling of the heat exchanger.

The work on Phase I of the program also concentrated on identifying the most significant industrial applications for this superheated steam drying concept. The work consisted of evaluating information gathered from a literature search, a survey of industrial dryer manufacturers product brochures, and material provided by APV Crepaco, Inc. APV Crepaco is a major industrial dryer manufacturer and has now agreed to be a project partner and contributor in the engineering work that must be conducted for the proposed Phase II.

The thermodynamic performance of the steam atmosphere dryer system with steam recompression is illustrated in Figure 1.3 as a function of system overall pressure drop and inlet dryer temperature. The heat input requirements are in units of heat (Btu's including electric power parasitics) per pound of water evaporated. The best standard air dryer systems can attain only 1400 to 1800 Btu/lb_{evap}; the former only if exhaust heat recovery methods are employed,

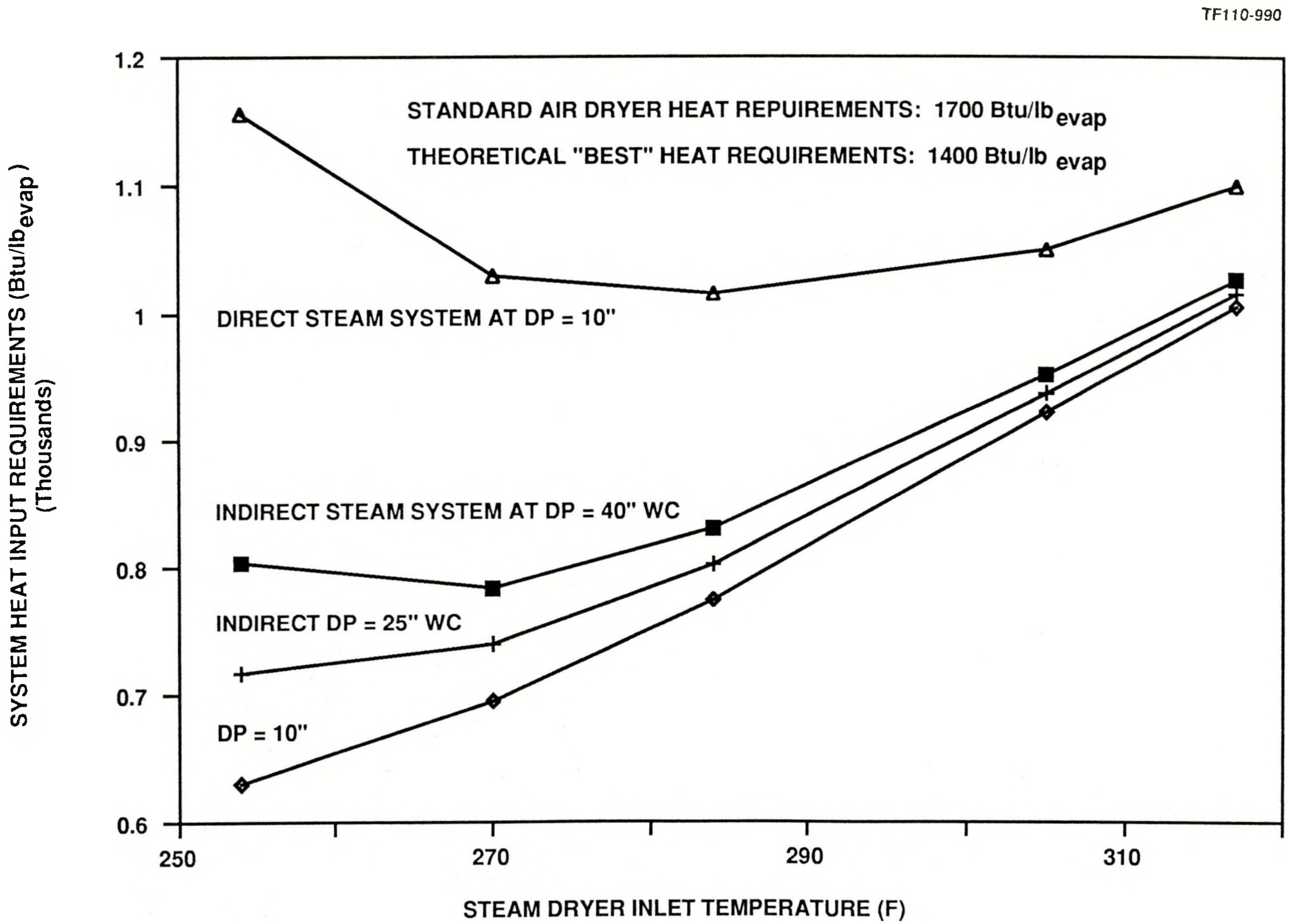

Figure 1.3 Steam Dryer System Performance for Various System Pressure Drops

such as if the dryer exhaust air is recirculated back into the air heater or if feedstock preheating is performed. Clearly, the steam atmosphere dryer system can provide a 55-percent energy savings over the standard air dryer system.

The energy savings brought about by utilizing the Steam Atmosphere Drying System are significant. The energy saved per year will pay back the steam system cost in less than 2 years, as shown in Figure 1.4.

In addition to the energy savings of the steam dryer system there are several other advantages that are important to its being accepted by the industrial dryer user. These advantages are:

- A 35-percent reduction in the yearly cost per lb_{evap} to dry wet feedstock
- Reduced airborne emissions of both feedstock effluent and combustion exhaust products by 75,000 tons/yr and by 5,800 tons/yr, respectively
- Reduced dry dust fire/explosion risks
- Hot product not exposed to oxygen
- Constant rate drying in steam atmosphere
- Product quality enhanced
- Dryer controls simplified
- Water mass transfer in product enhanced
- Reduced dryer size
- Reduced dryer cost
- Reduced dryer heat losses due to lower dryer inlet temperatures

Based on these clear advantages Tecogen has projected that the steam atmosphere drying system is most suitable as a replacement technology for state-of-the-art spray, flash, and fluidized bed drying systems. Such systems are utilized in the Food and Kindred Products (SIC 20); Rubber Products (SIC 30); Chemical and Allied Products (SIC 28); Stone, Clay, and Glass (SIC 32); Textiles (SIC 22); and Pulp and Paper (SIC 26) industrial sectors.

A more detailed list of the types of feedstocks that could be treated with Tecogen's steam atmosphere dryer is identified in Table 1.1. Thus, the 0.201 quad/yr of energy presently consumed in the U.S. with air dryers can be reduced by 55 percent, using Tecogen's steam atmosphere drying system with recompression. This results in a net energy savings of 111 x 10^{12} Btu/yr if 100 percent of the industrial dryer market in the U.S. is converted to a steam atmosphere dryer system or still a significant 1.1 x 10^{12} Btu/hr if only 1 percent of the market is developed.

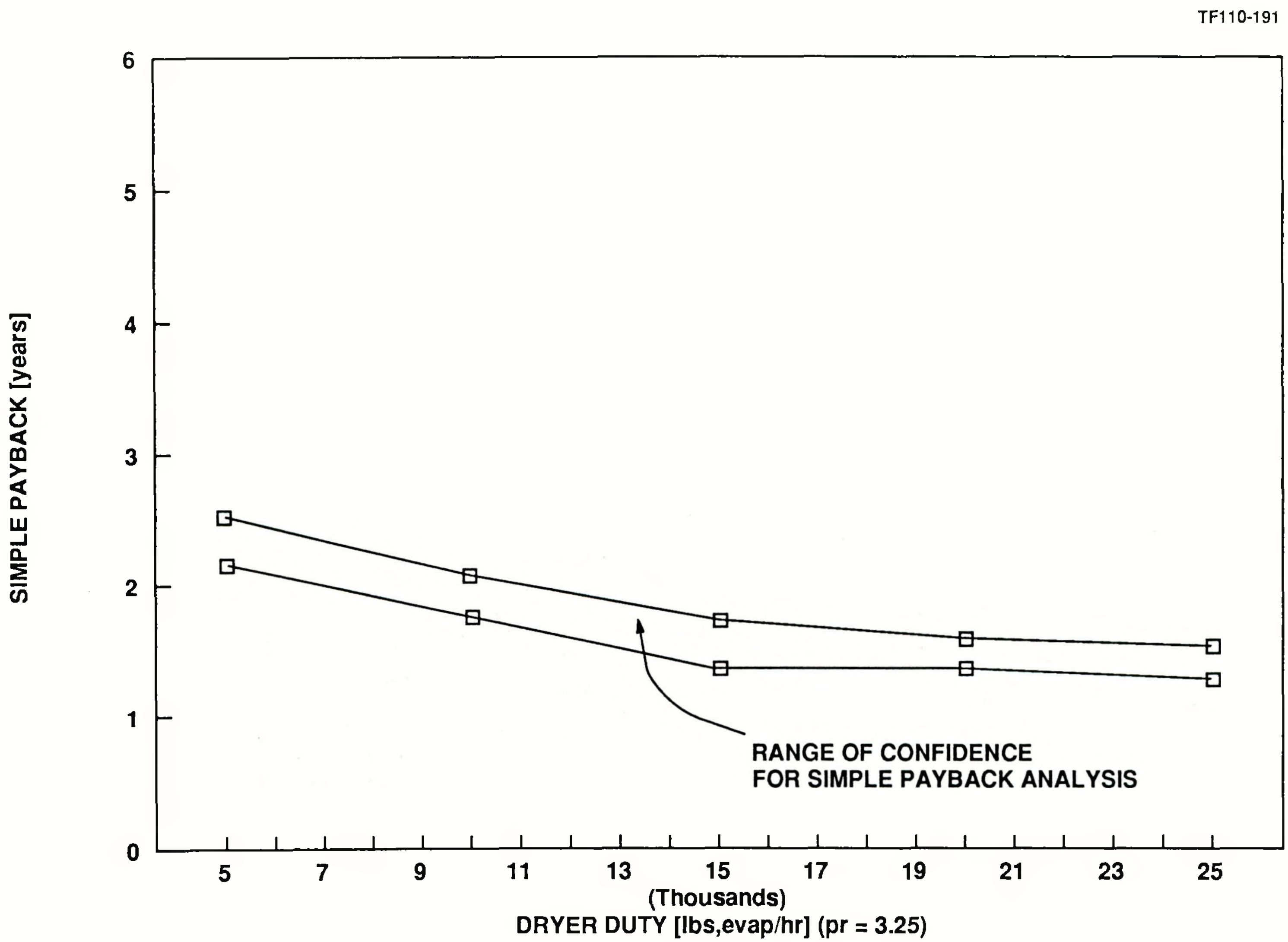

Figure 1.4 Simple Payback for Steam Dryer System
Hours = 6525, Elec. = $0.05/kWh, Gas = $3.5/MMBtu

TABLE 1.1

SPECIFIC FEEDSTOCKS TO BE DRIED

		Energy Requirements (quads/yr)
I. FOOD AND KINDRED PRODUCTS (SIC 20)		
1. Condensed and Evaporated Milk		
2. Coffee		
3. Dehydrated Food		
4. Pet Food		
5. Prepared Feeds		
6. Cane and Beet Sugar		
7. Malt		
8. Whey		
	SUBTOTAL	0.0628
II. CHEMICAL AND ALLIED PRODUCTS (SIC 28)		
1. Rubber		
2. Synthetic Fiber		
3. Activated Carbon		
4. Chalk Powder		
	SUBTOTAL	0.0389
III. STONE, CLAY AND GLASS (SIC 32)		
1. Structural Clay Products		
2. Gypsum Products		
	SUBTOTAL	0.032
IV. PULP AND PAPER (SIC 26)		
1. Pulp		0.05
V. MINING (SIC 10, 11, 12, 14)		
1. Bituminous Coal		0.0168
	SUBTOTAL	0.2005

Based on a market penetration of 10 percent, the potential number of superheated steam dryers that could be sold is estimated to be:

	Size	Number	Nominal Capacity lbs/hr Water Removal	Hours of Operation/yr
	Small	336	5,000	6,525
or	Large	93	20,000	6,525

The estimated cost of the proposed steam dryer system is shown in Figure 1.5 as a function of dry product flow rate. From this figure, a typical capital cost for 1000 lbs/hr of water removal capacity has been estimated to be $100,000 to $210,000. On this basis, the total value of the potential dryer sales would be $190 to $353 million. Spread over 10 years, this represents a sales level of $19 million to $35 million per year.

The recent survey of present dryer manufacturers found that they fall into the following size ranges:

Large Manufacturer	$10 million to 25 million/year
Average Manufacturer	$1 million to 5 million/year
Small Manufacturer	less than $1 million/year

Hence, the potential market of $19 million to $35 million per year for the superheated steam dryer represents a substantial new business opportunity for one or more manufacturers. The relatively small size of the individual manufacturers, however, makes it extremely difficult for any one manufacturer to underwrite the cost of developing and introducing the product without outside support.

A principal task in Phase I of the Steam Atmosphere Drying Project was the performance of cold (with air as the medium) and hot (with steam as the medium) testing of bench-scale models of the IRIS-type dryer. The purpose of these tests was twofold. For the cold testing the objective was to visualize the fluid dynamic processes of the IRIS vortex flow field while measuring dryer pressure drop and particle collection efficiency. For the hot testing the objective was to verify the ability of the steam IRIS dryer to dry slurried feedstock while measuring its heat transfer characteristics; i.e., heat transfer coefficient, drying performance with respect to dryer size, slurry flow rate, and conformity to known heat transfer coefficient relationships.

In order to accomplish both these project objectives, considerable effort was put forth with regard to apparatus design, fabrication, and laboratory testing. This was necessary in order to be confident in the results obtained as well as to be able to provide a more permanent bench-scale laboratory test facility for testing the IRIS dryer for future development interests.

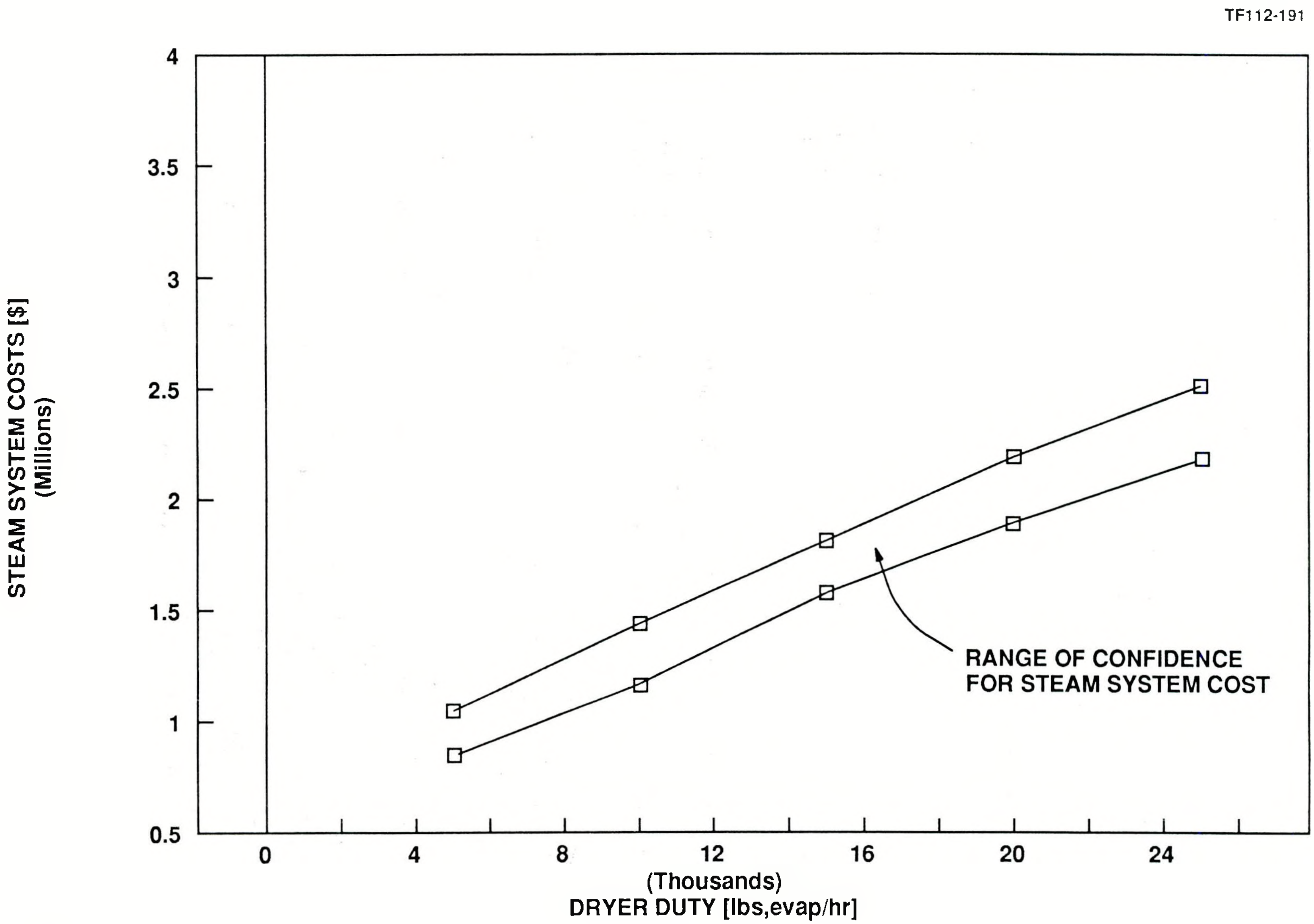

Figure 1.5 Steam Dryer System Costs

As a result of these cold (fluid dynamic) and hot (heat transfer) tests, the following accomplishments are cited. These results provided considerable new information to the field of steam atmosphere drying with respect to the IRIS dryer performance.

From the Cold Testing:

1. Measured recirculation collection efficiency as a function of particle diameters inlet gas steam velocity and dryer vortex flow field pressure drop. Recirculation ratios as high as 100:1 were recorded.

2. Visually observed dryer vortex flow field (up to full saturation) as a function of particle loading (L_p). Particle loading is defined as the ratio of particle mass flow rate to gas stream mass flow rate.

3. Observed and measured vortex field pressure drop as a function of particle loading and enabled a verification of the predicted overall dryer pressure drop. Pressure drops were found to be 60 to 70 percent of the predicted values.

4. Determined a dimensionless relationship for the vortex (dryer) pressure drop as a function of four (4) dimensionless groups. These groups were developed from a dimensional analysis of the measured cold test data.

5. Verified an increase in the dryer's recirculation ratio (R) as the dryer's particle loading increased. Test demonstrates that there will be adequate particle exposure time in the drying medium.

6. Determined a theoretical fluid dynamic and heat transfer model for the IRIS dryer and identified the optimum zone of operation with respect to particle size and gas velocity. The fluid dynamic and heat transfer limitations identified by this model have been tested and demonstrated to be conservative in this application to IRIS modeling.

7. Tested effects of increased lengths of the recirculation spout tube on the dryer collector efficiency and verified this method as a means of controlling dryer collector efficiency. A patent disclosure has been filed with DOE.

From the Hot Testing:

8. Measured actual local particle heat transfer coefficient h_p and dryer sizing coefficient H ($Q/Vol/\Delta T_{LM}$). Confirmed that the local heat transfer coefficient (h_p) is not dependent on particle flow rate or dryer loading.

9. Successfully dried slurry (liquid) feedstock with a wetness of 0.5 lb water/lb particle. An average of 90 percent reduction in wetness was recorded. The dryer's drying performance of 0.04 (D.B.) exceeded the predicted values of 0.05 (D.B.).

10. Succeeded in developing a preliminary design for a steam atomizer for slurry feedstock.

11. Succeeded in testing the steam atomizer nozzle in two different locations in the IRIS steam dryer and was thus able to compare the drying ability as a result of testing in these two locations. Recommended location is at the dryer inlet.

12. Tested several slurry feedstocks:
Temperature insensitive (clay) as well as temperature sensitive (non-dairy coffee creamer and maltodextrin-100, a food sweetener).
The steam dried clay powder exhibited no damage as a result of drying.

13. Verified the contributions of conduction heat transfer via vessel wall heating to the overall heat transfer mechanism in support of the indirect drying design.

Each of these testing developments will be discussed in detail in this report. These accomplishments are all positive in that they have contributed to the development of the IRIS-type steam dryer. The results from these tests, however, have also identified several additional engineering development areas that are recommended for further investigation. These areas of study are:

(1) Further study of an indirectly heated (i.e., jacketed) steam dryer vessel.

(2) A study of the IRIS-type dryer tested under a partial vacuum steam atmosphere in order to reduce saturated steam drying temperatures.

(3) A study of a larger scale IRIS dryer which will allow larger steam flow rates and therefore larger slurry flow rates.

(4) The continued design and development of a colder steam atomizer nozzle.

(5) The further development of better particle extraction methods (e.g., via the curvilinear louvered separator – C.L.S.) and/or a study of alternative locations for dried particle extraction from the recirculation line or from the bottom of the dryer.

(6) Additional study of the effects of alternative slurry atomizer nozzle locations on the effects of drying.

(7) A study of the effects of dryer size changes on the dryer's drying ability to determine the scale-factor relationship for heat and mass transfer.

(8) Complete the testing of a curvilinear louvered separator (C.L.S.).

(9) Testing with hot air in order to measure the comparative heat transfer coefficients for steam and air atmosphere drying using the IRIS dryer.

(10) Testing with additional feedstocks including specifically temperature-sensitive materials.

All of these studies can be incorporated into the next phase of the steam atmosphere drying project. The extensive hot test bench-scale-size IRIS laboratory developed in this first phase can be used to continue this basic research while a larger prototype-scale-size IRIS is being constructed. The larger scale IRIS dryer can then be used to repeat the heat transfer coefficient measurements between the steam atmosphere and the slurry feedstock in order to determine the scale factor relationship between large and small steam dryers.

An important goal will also be to demonstrate the successful drying of temperature-sensitive products (i.e., food and/or protein feedstocks). There are numerous references in the engineering literature of the successful steam drying of temperature-sensitive materials and Tecogen is confident of achieving similar results with drying temperature-sensitive materials. There has been considerable success in drying a non-temperature-sensitive feedstock (i.e., clay slurry). Tecogen is convinced of the ability of the IRIS dryer to dry temperature-sensitive materials and the second phase of this program provides the opportunity to continue the steam dryer testing with a wide range of feedstock products.

As a result of the Phase I work, Tecogen Inc. has been given commitments from APV Crepaco, Inc. and the New York State Energy Research Authority (NYSERDA) (with Tecogen's contribution) to share in the Phase II work effort.

NYSERDA Interests

New York State contains many of the industrial dryer manufacturers and industrial dryer users that Tecogen has identified (in Phase I) as being able to take advantage of the energy improvements afforded by the steam atmosphere dryer with steam exhaust recompression system. In a survey of industrial dryers for solids prepared by the Idaho National Engineering Laboratory (32), New York State was found to have 42 of the 301 reported industrial dryer manufacturers in the United States in 1976. Of the 42 industrial dryer manufacturers reported, seven were manufacturers of spray, pneumatic (flash), fluidized bed and tower dryers; dryers that have been identified by Tecogen's Phase I study to be replaceable by the more efficient steam atmosphere dryer system. Based on the INEL report's inventory of U.S. energy usage, and assuming that the energy consumption by each state is in proportion to the number of dryer manufacturers, New York State consumes approximately 25 to 58 x 10^{12} Btu/yr of (thermal) drying energy.[1]

Tecogen's steam atmosphere dryer with exhaust steam recompression can save approximately 55 percent of this energy.

Thus, a potential energy savings of 14 to 28 x 10^{12} Btu/hr can be realized if 100 percent of the dryers are replaced. Assuming that only 5 percent of these old dryers are replaced with the new steam atmosphere dryer in the first year, then an energy savings of 0.7 to 1.4 x 10^{12} Btu/hr can be projected. These energy savings are recognized by NYSERDA as well as DOE as substantial even if they were savings projected for the entire United States. The manufacturing of these steam dryers to meet this energy savings would be a $12M-per-year business including such steam dryer components such as the dryer, steam fan, steam compressor, and steam and feedstock preheat heat exchangers – some of which are exclusively available from New York State manufacturers.

New York State is also particularly attractive as a locale for field testing the steam atmosphere dryer system because it hosts many of the SIC industrial sectors that Tecogen has identified as being appropriate users of the steam atmosphere drying system. For example, Tecogen has found (from Phase I) that the Food and

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[1]This assumption is thought to be conservative for the highly industrialized New York State. Thus, a more reasonable drying energy usage for New York State is approximately 75 to 100 x 10^{12} Btu/yr.

Kindred Products Sector; the Textile Sector; the Sand, Stone, and Cement Sector; as well as the Pulp and Paper Sector all are represented in New York State. The likelihood, therefore, of identifying a cooperative field site for demonstrating a pilot-plant-size system (in Phase III) is very high.

Clearly, given New York State's many industrial dryer manufacturers and users, a joint DOE/NYSERDA steam atmosphere dryer development project could benefit the New York State economy and energy conservation efforts.

APV Crepaco, Inc. Interests

APV Crepaco, Inc. is a leading manufacturer of industrial dryers and evaporators as well as mechanical vapor recompression systems in the United States and Europe. The DOE/Tecogen steam atmosphere drying system program affords APV Crepaco, Inc. an opportunity to be in the vanguard of the development of a new dryer design and a novel steam atmosphere drying system.

APV's engineering facilities are located in Attleboro Falls, Massachusetts (a 1-hour drive from Tecogen Inc.) and in Tonawanda, New York. The locations of these engineering facilities will greatly expedite the project engineering and management communications required between Tecogen (and, hence, DOE) and NYSERDA during Phase II and later during Phase III. For example, APV Crepaco's dryer and evaporator business experience in New York State will clearly help to find a field test site for Phase III.

2. OVERALL PROJECT SCOPE OF WORK

The overall program is divided into three phases: Phase I Feasibility Investigation, Phase II Engineering Development, and Phase III Proof-of-Principle Testing. The Work Breakdown Structure and the Schedule for the complete program as originally proposed to DOE are presented in Figures 2.1 and 2.2, respectively. As currently planned, the program will take 42 months to complete.

Under Phase I of the program, Tecogen Inc. initiated a feasibility evaluation of the proposed drying concept. This was accomplished through a series of analytical, design, and laboratory testing tasks. The results of Phase I are summarized in this report.

The primary objective of the Phase II: Engineering and Development Phase is to design, build, and test a pilot-scale superheated steam drying system for the purpose of obtaining basic engineering data needed before a full-scale system can be designed and built for Proof-of-Principle Testing under Phase III of the program.

Based on the system design concepts and component specifications developed in Phase I, a complete pilot-scale system will be designed for drying granular materials in a laboratory environment. Additional laboratory testing will be performed on the Phase I facilities as needed to support the design effort. The system will be built and installed in a Tecogen test facility. A data acquisition system will be provided to measure and record all data. Prior to initiating tests, a test plan will be prepared and submitted to DOE for approval.

The main purpose of the tests will be to explore the effects of critical parameters on the performance of the dryer and to make improvements to the system where appropriate. Tests will be performed at various operating conditions with one or more granular materials. Important performance parameters will be measured including drying rates, particle separation efficiency, steam recompression power input, auxiliary power input, and final moisture contents. Variables that will have an effect on performance include particle size, particle size distribution, particle density, particle loading, superheated steam flow, drying chamber geometry, separation chamber geometry, steam recirculation rates, and recompression pressure ratio. The test data will be evaluated and the results will be used to update the performance and economic predictions.

The Phase III Proof-of-Principle Testing has as its main objective the testing and evaluation of a full-scale system to confirm the in-service performance and operability and to finalize the energy, economic, and environmental analyses.

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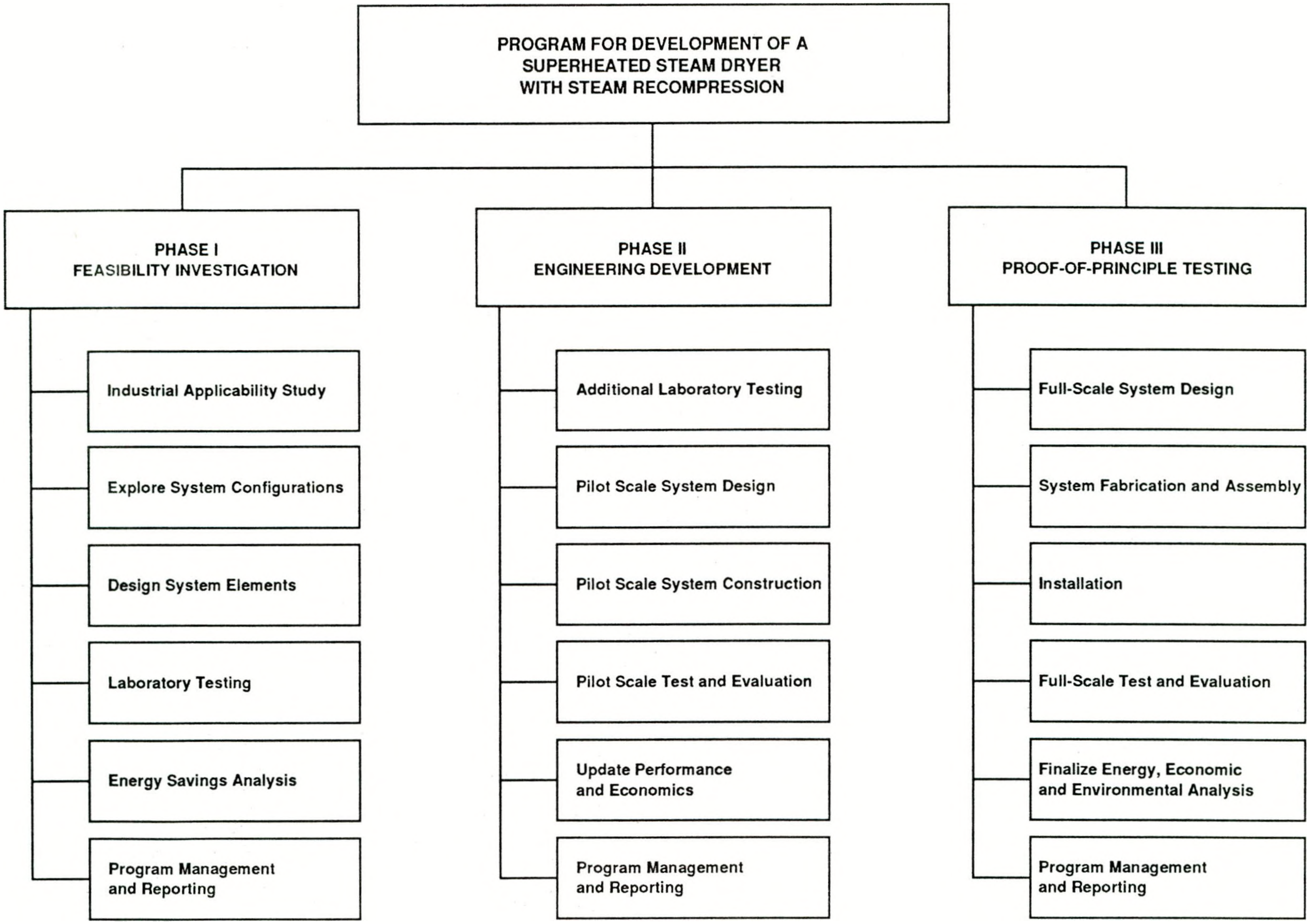

Figure 2.1 Work Breakdown Structure for Program

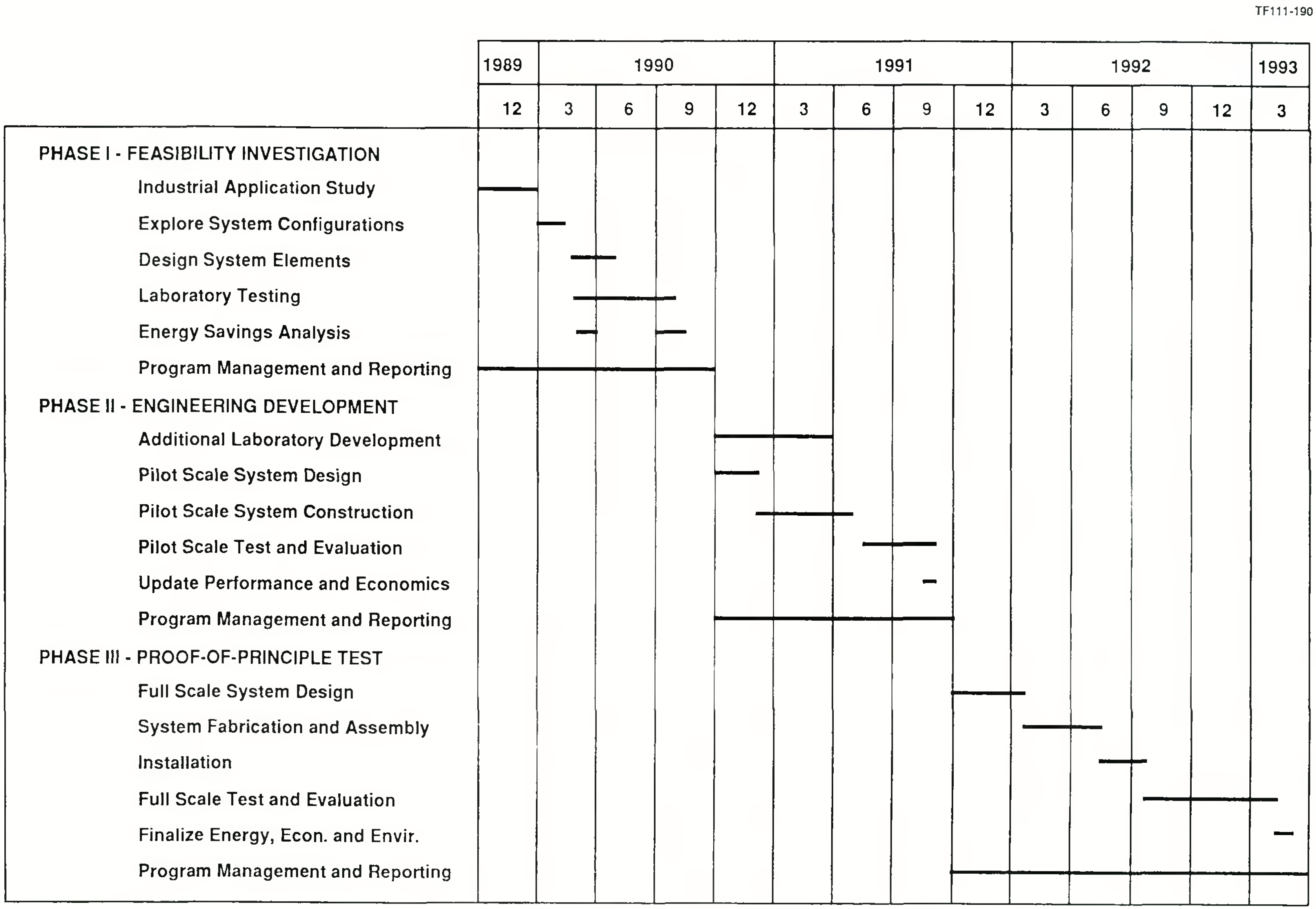

Figure 2.2 Program for Development of a Superheated Steam Dryer with Steam Recompression

Design of the full-scale system will be based on the test results obtained in Phase II with the pilot-scale system. The full-scale system will be built and installed at a site to be selected prior to the initiation of Phase III. To meet the Proof-of-Principle Testing objective, the full-scale system will be operated at the industrial site for a period of six months. The test results will then be evaluated and the energy, economic, and environmental analyses finalized. Successful Proof-of-Principle Testing will be the basis for commercialization of the system by Tecogen and APV Crepaco, Inc. without continued DOE funding.

Monthly reports will be submitted to DOE summarizing the technical, budget, and schedule status during Phase III of the program, and a final report documenting the results will be prepared and submitted to DOE at the completion of the technical work.

2.1 PHASE I STATEMENT OF WORK

A detailed Statement of Work for the just completed Phase I Feasibility Investigation is given in this section. The primary objective of this phase of the program was to establish the feasibility of Tecogen's proposed steam atmosphere drying concept using steam recompression and Tecogen's new IRIS steam dryer. This first phase included an industrial survey to identify the most suitable feedstock to dry, an analytical analysis of the proposed thermodynamic steam dryer cycle, and a laboratory hot and cold test of a bench-scale-size IRIS steam dryer model. The laboratory hot testing was to demonstrate the potential for the IRIS dryer design to dry a wet feedstock and to characterize the dryer's heat transfer performance.

2.1.1 Task 1 – Industrial Applicability Study

The purpose of this task was to identify the most significant industrial applications for drying granular solids using the proposed superheated steam drying concept. This was accomplished through a literature search, an industrial survey, and by using the drying knowledge and experience available through Thermo Electron Web Systems Inc. One or more of the most promising candidates was then selected and system design criteria established for later tasks.

2.1.2 Task 2 – Explore System Configurations

An analytical model of the drying system was prepared and a parametric analysis was conducted to determine the sensitivity of system performance to key parameters. The key parameters were found to include: steam pressure ratio, number of reheat stages, ratio of direct heating to recompression heating, dryer recirculation rates, dryer residence time, dryer inlet temperature, separator efficiency, compressor efficiency, reheater effectiveness, particle size and distribution, and particle loading.

Using the results of the analytical modeling, conceptual layouts were prepared to explore different design configurations for the overall drying system. Based on the results of the analytical and design studies, specifications were prepared for the major components or subsystems.

2.1.3 Task 3 – Design System Elements

Using the component specifications and system layouts prepared in Task 2, analysis and design work were conducted at the component or subsystem level for the major elements of the system. The dryer section, separator, and reheater are the main components requiring detailed analysis and design. The steam recompression subsystem can use available compressors and drives of the type Tecogen offers for its Mechanical Vapor Recompression Systems. Therefore, work on the steam recompression subsystem under this task consisted of sizing the rotary screw compressor, evaluating drive types (natural gas and diesel engines, variable and fixed speed electric motors, and steam or gas turbines), and designing the heat exchanger.

The primary component of the system is the steam dryer. The design of this component required careful evaluation of many interacting design parameters, which include recirculation rate, gas velocity, residence time, and pressure drop. Detailed design analyses were carried out in this regard to assure that the design will meet the overall system design criteria and performance specifications.

The second key component to the success of the dryer system is the high-efficiency, low-pressure-drop curvilinear particle separator. While cyclone separators have typically been used as solids collectors, their large size, low efficiency, and high pressure loss strongly limit their applicability for this application. For this system, a Curvilinear Louver Separator (CLS), which has significantly improved performance, will be analyzed.

2.1.4 Task 4 – Laboratory Testing

Laboratory testing concentrated on the key technical issues associated with the novel drying system proposed. As a result, the testing focused on the two new elements in the proposed system, the Inertial Recirculation with Internal Separation (IRIS) dryer and the curvilinear louver separator (CLS).

Cold tests were performed with transparent test sections to visually observe the flow patterns and (at a later time) to determine the particle separation efficiency of the CLS and the retention time for particles in the IRIS dryer.

Scaling rules were used to select air flow velocities and particle properties (density, size, size distribution) that simulate the conditions anticipated in an actual dryer.

The hot tests were conducted with superheated steam generated from the direct fired laboratory boiler. Heat and mass transfer rates were measured as a function of operating parameters in a test loop simulating the IRIS dryer and CLS. Data obtained in the cold flow tests were used to finalize the design of the hot flow components.

Prior to beginning the laboratory test, a test plan was prepared and submitted to DOE for approval. The test plan described the test loops and instrumentation, defined the tests and measurement to be made, and identified the range over which key parameters would be varied.

2.1.5 Task 5 – Energy Savings Analysis

An initial energy savings analysis was performed at the completion of Task 1 and Task 2. The energy savings analysis took into account the industrial applications and potential market share identified for the proposed concept in Task 1 and the specific energy savings calculated in Task 2 to estimate the national energy savings potential. At the completion of the laboratory testing, the analysis was updated to incorporate new information gained from the testing and the component analysis and design efforts.

2.1.6 Task 6 – Program Management and Reporting

The purpose of this task was to ensure the timely completion of the program objectives within the budgeted cost and to report on the work performed. To accomplish this, the program manager set goals, made the plans to accomplish these goals, and maintained effective use of program personnel. The program manager was responsible for the delivery of the contract commitments and for responding to requests from the project manager at DOE.

Brief monthly progress reports and detailed interim reports were submitted to DOE summarizing the technical, schedule, and budgetary status of the program. At the completion of the Phase I activities, a final report was prepared to document the results of the work performed.

This report details the results obtained from this Phase I work effort.

3. TASK 1 – INDUSTRIAL APPLICATION STUDY

3.1 TECHNICAL APPROACH FOR TASK 1 STUDY

The technical approach taken in performing Task 1 had two principal aspects. The first was identifying the sources and gathering information on industrial drying and on prior work on superheated steam drying. The second was establishing criteria for identifying significant industrial applications.

3.1.1 Sources of Information

Three main sources of information were used for the study. These were a literature and patent search, information gathered from industrial dryer manufacturers, and information provided by the Web Systems Division of Thermo Electron, a manufacturer of industrial dryers.

The electronic portion of the literature and patent search was conducted through Dialog Information Services Inc. and Orbit Information Technologies. The data bases searched from Dialog included Agricola, Chemical Engineering Abstracts (England), Coffeeline (England), Compendex Plus, Conference Papers Index, Dissertation Abstracts Online, DOE Energy, Electric Power Database, Energyline, Food Science and Technology Abstracts (England), NTIS, Scisearch, World Patents Index, INPADOC (International Patent Documents), Claims Patent Files, Thomas Register Online, PTS PROMT (Overview of Markets and Technology), PTS U.S. Forecasts, and PTS New Product Announcements. From ORBIT, the data bases searched included Biotechnology, Energy Biblio, Power, and Tropical Agriculture.

Separate from the electronic search, relevant information was gathered from the publications: Drying (volumes containing papers from the International Symposiums on Drying), the volumes of Advances in Drying, Drying of Solids – Recent International Developments, and preprints from the Electricity Council Research Centre.

The most important literature and patent references gathered from the search are listed in Appendix A.

A selective list of industrial dryer manufacturers has been compiled and is presented in Appendix B. Technical literature has been obtained from a representative sampling.

3.1.2 Applications Identification Criteria

For a drying application to be a candidate for the superheated steam drying concept under evaluation, it must meet the following criteria:

1. Superheated steam must be an acceptable drying medium for the application;

2. There must be a potential advantage to using superheated steam such as improved product quality, reduced energy costs and/or reduced first cost;

3. There must be sufficient market potential to make the application attractive to a dryer manufacturer; and

4. The material to be dried must be pumpable or slurried feedstocks, wet particles, or wet powders.

In addition, the total potential energy savings for all the candidate applications must meet the Department of Energy goal of saving 10^{12} Btu/yr.

In order to identify those applications where superheated steam drying has the greatest potential to make an impact, statistics were compiled on the industrial drying market regarding energy use, market size, dryer type, dryer efficiency, and material dried. These statistics were then examined with respect to the applications identification criteria listed above and the inherent advantages of superheated steam drying described previously. On this basis, a list of dryer and market characteristics was developed that identifies promising applications. The superheated steam dryer can be marketed to replace current dryer technology that:

1. Is low in thermal efficiency, uses large amounts of energy, and is expensive for the user to operate;

2. Is expensive for the user to purchase and is a type that sells in most industrial sectors;

3. Is large in size or difficult and costly to operate and maintain;

4. Requires an inert or oxygen-free atmosphere;

5. Is dedicated to specialty feedstocks; or

6. Has had some prior demonstrated success with a steam drying atmosphere.

The results, recommendations, and conclusions from the study are presented in the following chapters.

3.2 RESULTS OF STUDY

3.2.1 Energy Use Profile

Table 3.1 shows the energy used for drying in various industries. For 1990, the total energy used for drying is projected to be 1.51 Quads. The pulp and paper industries are by far the largest user and will consume 47 percent of the total. The food, agriculture, and lumber industries will consume approximately 10 percent each, and the textile, stone, and clay industries will consume about 7 percent each. Chemicals and mining will use an additional 5 percent and 3 percent, respectively. Except for the pulp and paper industries, these statistics indicate that a new type of dryer should be marketable across several industries in order to make a significant impact on energy use.

Drying energy use also has been categorized by dryer type. This information is presented in Table 3.2. Flash dryers are the largest energy users at 42 percent and are followed by cylinder dryers at 34 percent. Many other types use varying shares of the remaining drying energy. Table 3.3 shows in which industries the different types of dryers are used.

3.2.2 Applicable Industrial Dryer Types

A comparison of the performance and economic characteristics of several of the important types of dryers is presented in Table 3.4. The superheated steam drying concept under evaluation can most easily be adapted to replace spray, flash, and fluidized bed dryers. As Table 3.3 shows, these three types of dryers find wide use in the food, chemical, and rubber and plastics industries. Spray and fluidized bed dryers are also used in the stone, clay, and glass products industries. Although not shown in Table 3.3, additional applications for the spray, flash, and fluidized bed dryers are found in the agriculture and mining sectors of the economy.

As the wide use of spray dryers would suggest, these dryers are used to dry a broad range of products. Table 3.5 lists some of the many materials that have been dried with spray dryers. Many of these materials are also good candidates for superheated steam drying.

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TABLE 3.1

DRYING ENERGY-INTENSIVE INDUSTRIES: 1990 PROJECTIONS
(Ref. 1, pg. A-8)

Industry	SIC Code	U.S. Drying Energy (quads/year)	% U.S. Energy By Sector	% of Total Energy Used for Drying in Sector
Paper/Pulp	26	0.71	47.0	26
Food	20	0.17	11.3	29
Agriculture	01	0.16	10.5	6
Lumber	24	0.15	9.7	N/D
Stone and Clay	32	0.10	6.9	5–8
Textiles	22	0.10	6.9	N/D
Chemicals	28	0.07	4.9	N/D
Mining	10,12,14	0.04	2.8	N/D
TOTALS		1.51	100.0	

TABLE 3.2

TYPICAL DRYER EFFICIENCIES AND 1990 PROJECTIONS FOR ENERGY REQUIREMENTS BY DRYER TYPE (Ref. 13 and 32)

Dryer Type	Dryer Eff.	Dryer Energy Reqs. (quads/year)	"±" Accuracy (quads/year)	% of Total Energy
Tower	20-40	0.163	0.038142	10.8
Flash	50-75	0.629	0.252	41.7
Sheeting	50-90	0.003		0.2
Conveyor	40-60	0.002		0.1
Rotary	40-70	0.079		5.2
Spray	50	0.011		0.7
Tunnel	35-40	0.001		0.1
Fluidized Bed	40-80	0.027		1.8
Tray (Batch)	85	0.001		0.1
Drum (Indirect)	85	0.003		0.2
Rotary (Indirect)	75-90	0.063		4.2
Cylinder (Indirect)	90-92	0.509	0.063	33.7
Batch:				
Agitated Pan	90	0.001		0.1
Vacuum Rotary	Up to 70	0.013		0.9
Vacuum Tray	—	0.001		0.1
Infrared	30-60	0.001		0.1
Dielectric	60	0.001		0.1
		1.510	0.353	100.0

TABLE 3.3

INDUSTRIAL DRYER UTILIZATION SUMMARY
(Ref. 32)

SIC Code	Industrial Sector	Tower	Flash	Tray	Sheeting	Conveyor	Rotary	Spray	Through Circ.	Tunnel	Fluidized Bed
20	Food & Kindred Prod.	X	X	X	X	X	X	X	X	X	X
21	Tobacco Manufacturers			X		X			X		
22	Textile Mill Prod.					X			X		X
24	Lumber and Wood Prod.			X						X	
26	Paper and Allied Prod.					X					
28	Chemicals and Allied Prod.		X	X		X	X	X	X	X	X
30	Rubber and Misc. Plastics		X	X	X	X		X	X	X	X
32	Stone, Clay, and Glass Prod.					X	X	X			X
49	Electric, Gas, & Sanitary					X	X	X	X		

TABLE 3.4

COMPARISON OF INDUSTRIAL DRYERS: PERFORMANCE AND ECONOMICS
(1975 DATA)
(Ref. 32)

	Spray Dryer	Flash Dryer	Fluidized Bed Dryer	Tunnel Dryer	Rotary Dryer	Sheeting Dryer	Tower Dryer
1. Price Range	\$1.8K to \$1M	\$2 – \$200K	\$10K – \$300K	\$25K – \$250K	\$250K – \$1.5M	Up to \$250K	\$5K–\$200K
2. Operating Life (years)	Up to 25	8 – 20	Up to 25	Up to 30	Up to 10	25 – 30	10 to 20
3. Maintenance Cost (\$ of dryer cost/year)	1–2%	LT. 5%	5%	2–5%	Up to 5%	5%	1 – 3%
4. Sales Volume			\$300K to \$500K	About 5 units/yr		\$250,000/yr	\$10M – \$15M
5. Number of Units in Operation	Approx. 2000	3500	300	50	35,000	50	4000 Commercial (30,000 on farms)
6. Range of Capacities	Up to 30 tons/hr water	5 – 20 tons/hr, prod.	0.05 to 30 tons/hr prod.		0.05 – 600 tons/hr	40 – 50 ft/hr	450 – 7500 Bushels of grain/hour
7. Range of Operating Temperatures	95 – 700C	38 to 700 C	38 to 540 C	Up to 150 C	95 – 1315 C	Up to 204 C	66 to 204 C
8. Energy Requirements	E,G,O, STM Coal, Waste Heat	E,G,O, STM Coal, Wood	E,G,O, STM Coal, Waste Heat	E,G	E,G,O, Coal, Wood	E,G	E,G,O, Propane
9. Moisture Removal Capacity	0.1 – 3.0 $lbw/hr/ft^3$						
10. Energy Requirements (Btu/lb Prod.)		400 – 1000				1500	10 – 15 MMBtu/ 1000 Bushels
(Btu/lb Water)		1600 – 3750				——	
11. Dryer Efficiency	50%	50 – 75%	40 – 80%	35 – 40%	40 – 70%	50 – 90%	20 – 40%
12. Water or Solvent Removal	Both	Both	Both	Both	Both	Generally Solvent	Water

TABLE 3.5

TYPICAL FEEDSTOCKS USED IN SPRAY DRYERS
(Ref. APV Crepaco)

DAIRY PRODUCTS
Baby Food
Butter
Buttermilk
Caseinates
Casein Hydrolysate
Cheese
Chocolate Milk
Cream
Dietetic Products
Fat-Enriched Milk
Ice Cream Mix
Lactates
Lactose
Malted Milk
Milk-Cocoa
Peptones
Skim Milk
Whey
Whey Non-Hygroscopic
Whole Milk

EGG PRODUCTS
Egg White
Egg Yolk
Whole Egg

ANIMAL PRODUCTS
Albumen
Beef Extract
Bile Extract
Blood, Albumen
Blood, Plasma
Blood, Serum
Bouillon
Brain
Fish Meat, Hydrolyzed
Fish Pulp
Fish Solubles
Gelatine
Glands
Glue
Hormones
Liver Extract
Meat Extract
Pancreas
Pepsin
Proteins
Rennet
Thymus

VEGETABLE PRODUCTS
Agar-Agar
Alfala
Alginates
Aloe
Antibiotics
Bacitracin
Bananas
Bark Extract
Beer Wort
Carrageen
Champignon
Chlorophyll
Chlortetracyclin
Colors
Corn Steep Liquor
Corn Starch
Corn Syrup
Dextrane
Dextrin Maltose
Dextrose
Diastase
Distillers' Waste
Enzymes
Flavors
Fruit Juice
Fruit Pulp
Garlic
Glucose
Glue
Gluten
Gum Arabic
Hydrolysates
Latex
Lignin
Licorice
Malt Extract
Mango
Molasses with Filler
Olive Paste
Oxytetracyclin
Papain
Peanut Milk
Pectin
Penicillin
Pollen Extract
Potatoes
Potato Waste Liquor
Quebracho
Resin Soap
Rubber Latex
Saponin
Seaweed Extract
Senna
Sorbose
Soups
Soy Flour
Soy Bean Milk
Soy Bean Protein
Starch Products
Streptomycin
Tannin Extract
Tapioca
Tea Extract
Tomato
Vegetable Extracts
Vegetable Proteins
Yeast
Yeast Autolysate
Yeast Hydrolysate

ORGANIC CHEMICALS
Acetates
Alcoholic Extracts
Alizarin Carmine
Alkyl-Aryl-Sulfonate
Amino Acids
Benzoate
Carbamide Resins
Carboxy-Methyl Cellulose
Cellulose Acetate
Cellulose Hydrate
Chelates
Cholin Chloride
Citrates
Colors
Detergents
Dyes
Emulsifiers
Fatty Alcohol Sulfonates
Flavors
Formiates
Glucoheptonate
Glycerol Monostearate
Herbicides
Hexamine
Insecticides
Melamine Resins
Penta-Erythritol
Pesticides
Phenolic Resins
Plastic Emulsions
Polyvinyl Acetate
Polyvinyl Chloride
Quaternary Salts
Sequestering Agents
Soap
Sorbate
Sodium Adipate
Sodium Phenate
Stearates, Metallic
Stearic Acid
Stearyl-Tartrate
Sulfonates
Thiamine
Urea Formaldehyde
Wax
Weed Killers
Wetting Agents

INORGANIC CHEMICALS
Alumina Gels
Aluminates
Aluminum Silicate
Aluminum Sulfate
Ammonium Phosphate
Barium Chloride
Barium Sulfate
Barium Titanate
Bleaching Agents
Borates
Boric Acid
Calcium Carbonate
Calcium Salts
Carbon, active
Carbon, black
Catalysts
Chromium Sulfate
Copper-Oxy-Chloride
Ferric Oxide
Ferrous Oxide
Glauber Salt
Hypochlorites
Magnesium Carbonate
Magnesium Oxide
Magnesium Salts
Manganese Sulfate
Metallic Soaps
Nickel Compounds
Nitrates
Petroleum Catalyst
Phosphates
Pigments
Potassium Sulfite
Silica-Alumina Gels
Silicates
Titanium Dioxide
Water Glass
Zinc Ammonium Chloride
Zinc Chromate
Zinc Stearate

CERAMIC AND OTHER MINERAL MATERIALS
Abrasive Slurries
Alumina
Bentonite
Ceramic Colors
Ceramic Enamels
Cermets
China Clay
Clay
Cryolite
Diatomaceous Earth
Ferrites
Fullers Earth
Glazes
Graphite
Kaolin
Metal Powders
Quartz Slurries
Steatite Slurries
Titanates
Wall Tile Slurries
Zirconium Silicate

3.2.3 Potential Energy Savings

The annual U.S. energy use and efficiency for the three types of dryers selected as candidates for replacement by the superheated steam drying system are:

Dryer Type	Drying Energy (Quads)	Typical Dryer Efficiency
Flash	0.629	0.63
Spray	0.011	0.50
Fluid Bed	0.027	0.60
TOTAL	0.667	

The calculated energy weighted average efficiency for these dryers is 0.626. It should be noted that these efficiencies are defined as the minimum energy required to evaporate the water removed during the drying process divided by the actual energy used by the dryer. This efficiency definition assumes that the water leaves the system as a vapor. With a superheated steam drying system that employs exhaust steam recompression, the water removed in the drying process leaves the system mostly as a liquid. As a result, a superheated steam dryer with steam recompression can have a dryer efficiency greater than 100 percent.

A further refinement of the likely types of feedstocks that could be treated with Tecogen's steam atmosphere dryer is identified in Table 3.6. It will be shown from the analysis conducted in Tasks 2 and 5 that the 0.201 quad/yr of energy presently consumed in the U.S. with air dryers can be reduced by 55 percent. This would result in a net energy savings of 111×10^{12} Btu/yr if 100-percent market utilization is to be realized.

3.2.4 Market Potential

Based on a market penetration of 10 percent, the potential number of superheated steam dryers that could be sold is estimated to be:

	Size	Number	Nominal Capacity lbs/hr Water Removal	Nominal Hours of Operation/yr
	Small	336	5,000	6,525
or	Large	93	20,000	6,525

TABLE 3.6

SPECIFIC FEEDSTOCKS TO BE DRIED

		Energy Requirements (quads/yr)
I. FOOD AND KINDRED PRODUCTS (SIC 20)		
1. Condensed and Evaporated Milk		
2. Coffee		
3. Dehydrated Food		
4. Pet Food		
5. Prepared Feeds		
6. Cane and Beet Sugar		
7. Malt		
8. Whey		
	SUBTOTAL	0.0628
II. CHEMICAL AND ALLIED PRODUCTS (SIC 28)		
1. Rubber		
2. Synthetic Fiber		
3. Activated Carbon		
4. Chalk Powder		
	SUBTOTAL	0.0389
III. STONE, CLAY AND GLASS (SIC 32)		
1. Structural Clay Products		
2. Gypsum Products		
	SUBTOTAL	0.032
IV. PULP AND PAPER (SIC 26)		
1. Pulp		0.05
V. MINING (SIC 10, 11, 12, 14)		
1. Bituminous Coal		0.0168
	SUBTOTAL	0.2005

The estimated cost of the proposed IRIS steam atmosphere dryer system is shown in Figure 3.1 as a function of dry product flow rate. From this figure, a typical capital cost for 1000 lbs/hr of water removal capacity has been estimated to be $100,000 to $210,000. On this basis, the total value of the potential dryer sales would be $186 to $353 million. Spread over 10 years, this represents a sales level of $19 million to $35 million per year.

Tecogen's survey of present dryer manufacturers found that they fall into the following size ranges:

Large Manufacturer	$10 million to 25 million/year
Average Manufacturer	$1 million to 5 million/year
Small Manufacturer	less than $1 million/year

Hence, the potential market of $19 million to $35 million per year for the superheated steam dryer represents a substantial new business opportunity for one or more manufacturers. The relatively small size of the individual manufacturers, however, makes it extremely difficult for any one manufacturer to underwrite the cost of developing and introducing the product without outside support.

3.2.5 Review of Prior Superheated Steam Drying Work

The literature search identified 18 initiatives where superheated steam had been considered or used for drying. Although one program dates back to 1920, the majority of the work has been in the 1980's. Table 3.7 summarizes this work. It lists the names of the researchers, references from Appendix A, dryer configurations, feedstocks tested, level of development (analysis, laboratory testing, pilot plant, or commercial installation), successes, problems, and stated conclusions.

Most of the recent research on superheated steam drying has been conducted outside of the United States, particularly in Europe, Canada, Australia and South Africa. While some of the programs considered steam recompression, only two have proceeded to a pilot plant and none are commercially available either inside or outside the U.S. Several pilot plants have been built and operated on superheated steam <u>without</u> steam recompression. The types of dryers investigated have been quite varied and have included spray, flash, fluidized bed, yankee, tray, kiln, convective, and film. Equally varied have been the products dried, including milk, whey, cabbage, hay, soybean flakes, soy sauce cake, sugar beet pulp, coffee nutrient, coffee slip, paracetomal, bone protein, detergent whitener, clay, tissue paper, pulp, timber, textiles, alumina, activated carbon pellets, coal, sodium nitrate, and cellulose acetate. The successes, problems, and conclusions from this

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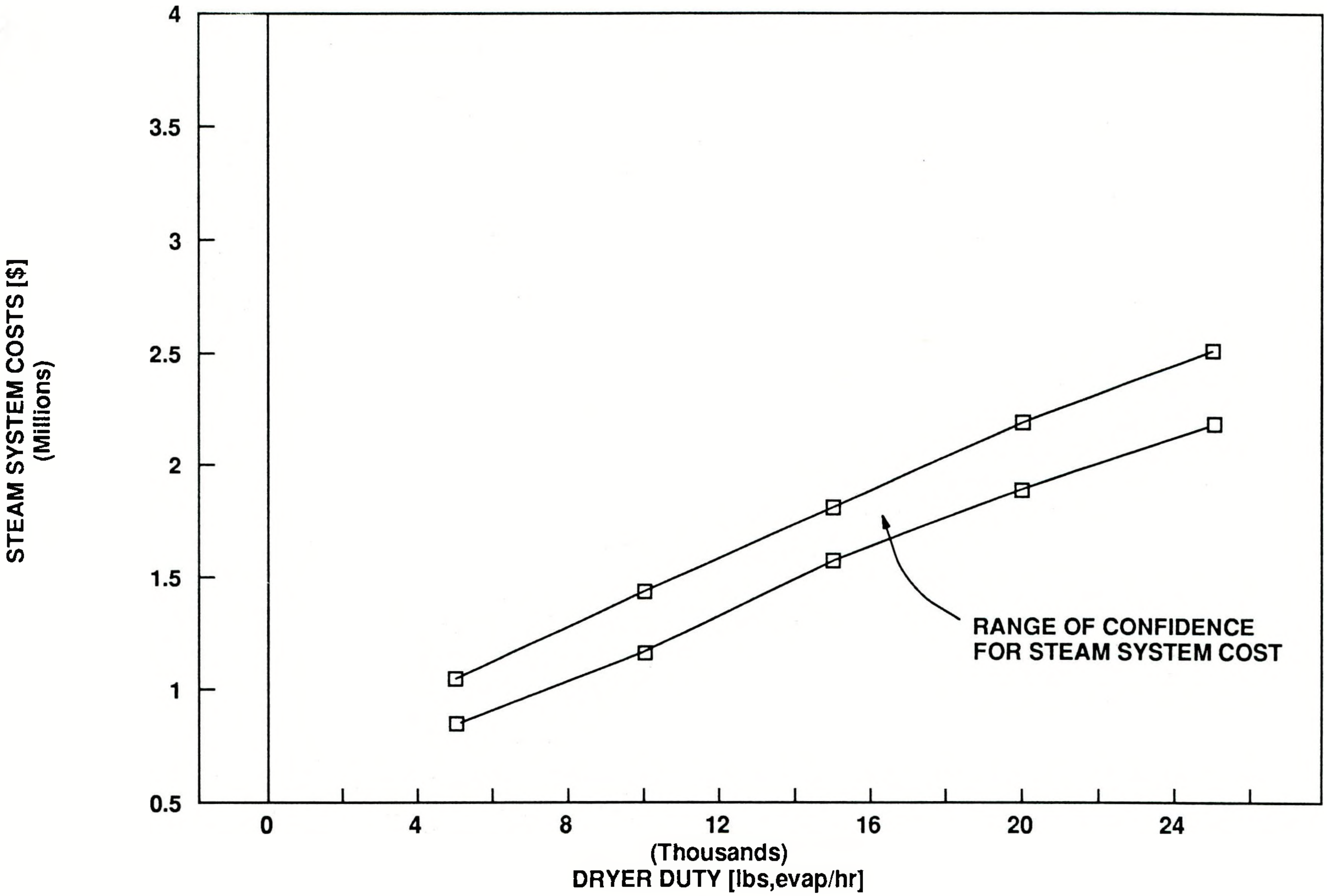

Figure 3.1 Steam Dryer System Costs

TABLE 3.7

SUMMARY OF STEAM DRYING R&D, PILOT PLANT, AND COMMERCIAL INSTALLATIONS

No.	Researcher	Ref. No.	Year	Dryer Configuration	Feedstock Tested	Level of Activity	Successes	Problems	Stated Conclusions
1.	E.C.R.C. (G.B.) R. Benstead (see also No. 6)	36	1965	Spray dryer w/recompression & NIRO atomizer	Milk, whey, coffee nutrient, coffee slip & detergent whitener, clay paracetomal 800 micron particles and 60% moisture typical	L	Clay slip and detergent gave no problems; paracetomal & bone protein, fair results depending on conditions; pilot plant to be built (2 tonnes/hr)	Dairy products coated walls and degraded at Twb = 100 C; steam dryer is larger and wet bulb temps. higher than air dryer counterpart at steam temps. less than 180 C; stopping distance for steam particle is also longer	Steam atmosphere spray dryer may be uneconomic except where it replaces an expensive or sophisticated system such as inert atm. Stack temperatures should be 175 C for optimum performance
2.	Messrs. Cui and Mujumbar McGill Univ.	13		Yankee dryer	Tissue paper	L	Mathematical model of dryer verified by lab tests; drying rates enhanced by 25–30%	Effects of steam on paper quality unknown at time but suggested to be beneficial to strength properties	Quality of paper must be assured; concept is technically and economically viable and thus attractive for user
3.	H.N. Rosen AICHE Symposium Ser. 77 (207)	40	1981	Tray dryers	Timber	CI	Capital and operating costs of drying in steam lower than drying in air; 1272 Btu/lbw removed	Excessive steam temp. may discolor timber	Installations are operating satisfactorily

LEVEL OF ACTIVITY
- A: Analysis
- L: Laboratory Testing
- P: Pilot Plant Facility
- CI: Commercial Installation

TABLE 3.7 (Continued)

SUMMARY OF STEAM DRYING R&D, PILOT PLANT, AND COMMERCIAL INSTALLATIONS

No.	Researcher	Ref. No.	Year	Dryer Configuration	Feedstock Tested	Level of Activity	Successes	Problems	Stated Conclusions
4.	W. Miller Forest Products Journal 27 (9) pg. 54–58	40	1977	Kiln dryers with steam recompression	Timber	A	COP overall = 1.6 compared to typical dryer efficiencies of 60–65%	Specially designed steam compressor required for 405 C superheat and 7 atm pressures; contaminated steam also a problem with fouling equipment	
5.	Gauvin & Costin McGill Univ.	2	1980	Spray dryer with steam atmosphere (5000 lb/hr with evaporator)	Sodium-nitrate	A	Capital costs for steam drying were 33% lower than air and operating costs were 21.5% lower; steam dryer performance and cost model were developed; dryer cost: $745,000 (1978 $)	Attention must be given to heat sensitive materials; there is a risk of condensation in product handling equipment; without steam recompression spray dryer effectiveness drops from 51.8% to 48.4%	Results should encourage further modeling and a full-scale or pilot plant system; caution is needed with respect to temperature-sensitive materials
6.	A.V. Heaton R. Benstead E.C.R.C. (see item 1)	34	1984	Spray dryer with steam recompression (aka.: HITREC)	Clay slip and various others; 16–40% t.s.; drying from 50% to 1% wet basis	L P	Operating temps. and pressures were 140 C inlet and 4 bars compressor discharge pressure; system built and tested for 360 hrs using clay slip	Compressor selection difficult, rotary vane oil; lubricated; heat losses were substantial and affected efficiency	Steam recompression is a viable energy saving technique; acceptable payback periods are possible

LEVEL OF ACTIVITY
- A: Analysis
- L: Laboratory Testing
- P: Pilot Plant Facility
- CI: Commercial Installation

TABLE 3.7 (Continued)

SUMMARY OF STEAM DRYING R&D, PILOT PLANT, AND COMMERCIAL INSTALLATIONS

No.	Researcher	Ref. No.	Year	Dryer Configuration	Feedstock Tested	Level of Activity	Successes	Problems	Stated Conclusions
7.	British Patent I 558 513 W.R.S. Baxter	35	1980	Convective dryer specifically identified but all types of dryers claimed	Unspecified granular wet materials	A			
8.	A.S. Mujumbar "Survey of R&D in Canada"	20		Convective film dryers without steam recompression	Pulp and textiles	CI	Installations in India and Sweden are operating satisfactorily	Cautions given toward temperature-prohibited materials used with steam drying	In general, steam drying is viable
9.	E.F. Faber M.D. Heydenrych (South Africa)	10	1986	Convective dryer	Alumina drying from 40% to 2%; activated carbon pellets drying from 50% to 8% (final drying to 2%)	A L	Technical and economic comparison of air and steam dryers given; South Africa commercial installation showed to be 40% less expensive to build compared to an air dryer system	Steam is a more complex dryer system; cautions given concerning temperature-sensitive materials and condensation effects; steam dryers must operate at high temps. (greater than 160 C) if drying rates are to exceed those of air	Computer model available for technico-economic modeling; energy is saved with capital costs reduced; steam system particularly attractive for combustible atmospheres

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LEVEL OF ACTIVITY
- A: Analysis
- L: Laboratory Testing
- P: Pilot Plant Facility
- CI: Commercial Installation

TABLE 3.7 (Continued)

SUMMARY OF STEAM DRYING R&D, PILOT PLANT, AND COMMERCIAL INSTALLATIONS

No.	Researcher	Ref. No.	Year	Dryer Configuration	Feedstock Tested	Level of Activity	Successes	Stated Problems	Conclusions
10.	Claes Swenson Swedish Exergy Tech., Inc. with Chalmers Univ. of Tech.; affl. Mo Do-Chemetics	19	1981	Staged shell and tube heat exchangers with steam outside tubes; fans used without recompression; also with steam recompression	Pulp drying from 50% to 10%; hog fuel drying from 70% to 30%; sugar beet drying from 80% to 10%	A P CT	150 ton/day capacity with 30% reduction in drying costs compared with flash dryer; low power, very short drying times, no fire risk; four CT's in operation; 62% less energy used here than in flash dryer	Minor or no effect on pulp quality; needs relatively small particles (5–10 mm); pressurized equipment and SST construction required	Steam drying of pulp, hog fuel, and sugar beets have shown to be viable; with steam recompression added marginal benefits considering the added complexity and cost of compressor and subsystems
11.	R. Zylla and C. Strumillo Inst. of Chem. Eng. Lodz Tech Univ. Poland	16		Film dryer (after Villalobos and Sakhuja)		A			
12.	Owen Potter Colin Beeby Dept. of Chem. Monash Univ. Australia	5	1985	Fluid bed without steam recompression and with steam recompression	Brown coal drying from 67% moisture	A P A	Batch drying of foundry sand successful in 1920; control is simpler by needing to monitor only temp.; no fire risk; constant rate drying period is longer with steam	Cautions against temperature-sensitive materials; it is also more difficult to achieve low moisture levels	

LEVEL OF ACTIVITY
- A: Analysis
- L: Laboratory Testing
- P: Pilot Plant Facility
- *CI: Commercial Installation*

TABLE 3.7 (Continued)

SUMMARY OF STEAM DRYING R&D, PILOT PLANT, AND COMMERCIAL INSTALLATIONS

No.	Researcher	Ref. No.	Year	Dryer Configuration	Feedstock Tested	Level of Activity	Successes	Problems	Stated Conclusions
13.	J. Karner, Chu, et al.	5	1920	Batch drying without steam	Cabbage and hay	A			
			1953	recompression	Soybean flakes	P			Product quality increased
	Yoshida & Hyopo	5	1963		Cellulose acetate				
14.	O. Potter, Keogh	40		Fluidized bed without steam recompression	Brown coal drying from 67% moisture	P	System worked well		Steam drying is encouraged
15.	T. Akao T. Fukurawa T. Watanase	40	1982	Fluidized bed without steam recompression and cylindrical tubes dryer without steam recompression; steam at 200–250 C	Soy sauce cake drying from 12% to 5% moisture	A P CI	Objectionable odors were removed by using steam as drying medium	Erosion, corrosion, and oil fires (due to oil condensation and dust formation)	Drying was successful but some problems did occur
16.	J. Meunier McGill Univ. Canada	12	1966	Flash dryer	Generic materials with 50% moisture and 300 micron particle sizes	A	Successfully modeled a steam atmosphere flash dryer; size reductions predicted with very high steam temperatures; narrow size distribution in feedstock shown to help drying performance		Modelling was successful; parametric study suggested design features of a steam flash dryer
17.	French Patent A.D. Passey J.R. Moreau		1969	Steam spray drying with recompression					
18.			1989	Steam conveyor drying with recompressor		P	Successful with sugar beet pulp	Poor steam seals cause loss of steam from dryer	Drying sugar beets can be successful with steam

LEVEL OF ACTIVITY
- A: Analysis
- L: Laboratory Testing
- P: Pilot Plant Facility
- CI: Commercial Installation

prior work provide a valuable guide in our selection of applications that offer a high probability of commercial success. It is interesting to note the encouraging successes in drying both food and non-organic materials using steam as the drying medium.

The literature also identified an important limitation that must be considered when superheated steam is substituted for air in conventional flash or spray dryers. Particles will travel faster and have longer trajectories in steam because steam is less dense than air. If the dryer design approach is not modified, the drying chamber would have to be made larger to obtain the same drying capacity with superheated steam as with air. The IRIS drying chamber design provides a way to overcome this limitation and obtain increased residence times for wet particles in the drying zone.

3.2.6 Summary of Study Results

The most important results from the study can be summarized as follows:

1. The superheated steam drying concept with the IRIS drying chamber can compete best against the spray, flash, and fluidized bed types of dryers.

2. Spray, flash, and fluidized bed dryers annually use 0.667 Quads of energy, which is 55 percent of the total drying energy consumed in the United States.

3. The typical weighted efficiency for these types of dryers is 62 percent.

4. There are approximately 10,000 spray, flash, and fluid bed dryers in use in the United States, with a nominal size of 6,000 to 10,000 lbs/hr of water evaporated.

5. The capital costs for these types of dryers are high, and competitive systems would be welcome.

6. Only spray and flash dryers can be used for the many and varied pumpable (slurried) feedstocks.

7. A 10-percent penetration of the market for spray, flash, and fluidized bed dryers by superheated steam dryers with exhaust recompression would save 11 x 10^{12} Btu/yr.

8. Successes with drying food (i.e., temperature sensitive) as well as non-organic (i.e., temperature unsensitive) materials have been documented in the literature.

3.3 RECOMMENDATIONS

Based on the results of the study, it is recommended that work on the superheated steam drying concept be directed at applications where spray, flash, and fluidized bed dryers are currently used in industry. In addition, the following specifications are recommended for the superheated steam dryer analysis and design work to be performed under the remaining tasks of the Phase I Feasibility Investigation:

1. Moisture Content:	
At Inlet:	0.45 – 0.60 (wet basis) 0.80 – 1.50 (dry basis)
At Outlet:	0.005 – 0.10 (wet basis) 0.005 – 0.09 (dry basis) or 2 – 5% higher than final moisture content for first stage of a two-stage dryer
2. Particle Size:	Less than 500 micron (10 – 100 micron: spray dryer) (100 – 500 micron: flash dryer)
3. Drying Capacity:	
Full-Scale:	5,000 to 15,000 lbs/hr water evaporated
Pilot-Scale:	2,000 to 3,000 lbs/hr water evaporated
Bench-Scale:	500 lbs/hr water evaporated
4. Maximum Operating Temperature:	320°F
5. Typical Residence Times:	0.5 – 3.5 seconds
6. Feedstock Preheat Temperature:	140°F (maximum)
7. Particle Recovery Efficiency:	99% typical 99.9% (fatty powders)

3.4 CONCLUSIONS

The most important conclusions from Task 1 are the following:

1. Our current conservative estimate for potential energy savings ranges from 1 to 11 times the DOE goal of 10^{12} Btu/yr.

2. Most recent research on superheated steam drying has been conducted outside of the United States, particularly in Europe, Canada, Australia, and South Africa. This work has identified some of the potential applications and limitations of superheated steam drying.

3. Flash, spray, fluidized bed, and tower types of dryers are good candidates for replacement by superheated steam dryers and are among the most widely used types in industry.

4. The superheated steam drying concept can be readily adapted to drying pumpable or slurried feedstocks, wet particles, and powders. Many of these applications exist in the chemical, pharmaceutical, mining, agriculture, food, textile, and pulp and paper industries.

5. The potential advantages of superheated steam drying compared to air drying include: lower energy costs, improved product quality, no airborne emissions, reduced fire or explosion risk, elimination of oxygen exposure, prolonged constant-rate drying, denser product, and higher drying rates.

6. The need for longer particle residence times is a limitation identified in the literature when superheated steam is substituted for air in flash or spray dryers. The IRIS drying chamber design provides a way to overcome this limitation and to obtain increased residence times for wet particles in the drying zone through use of its internal as well as external particle recirculation flow fields.

4. TASK 2 – EXPLORE SYSTEM CONFIGURATIONS

4.1 TECHNICAL DISCUSSION

The following section summarizes the results obtained in comparing the performance of a steam atmosphere (with exhaust recompression) dryer system with a conventional air atmosphere drying system. These comparisons were made using Tecogen's thermodynamic computer models for a conventional air-dryer system (with air recirculation) and the proposed steam dryer and steam recompression system. These computer models were developed under Phase I of the program.

4.1.1 Conventional Air-Dryer System

A simplified schematic of a typical air-drying system is presented in Figure 4.1. A supply fan and burner are used to provide hot air to the dryer where moisture is evaporated from the product. The humid outlet air from the dryer can then be exhausted to the atmosphere, or parts of it can be recirculated to be mixed with the air-dryer's inlet air stream. For this system's analysis the following system component efficiencies and operating parameters were used:

Fan/Blower Pressure Rise	10 in. wc
Fan/Blower Efficiency	0.60
Elec. Motor Efficiency	0.95
Elec. to Thermal Conversion Efficiency	0.3 (11,376 Btu/kWe-hr)
Dryer Ambient Inlet Temperature and R.H.	70°F; 50% RH
Feedstock Inlet Water Content	50% (D.B.)
Feedstock Outlet Water Content	5% (D.B.)
Feedstock Preheat Temperature (When Used)	140°F
Air-Dryer Heat Transfer Loss	1.5%
Fueled Air Heater Efficiency	85%

The total energy entering the process is equal to the sum of the burner input and the equivalent fuel input used to provide the electrical energy to drive the fan.

The theoretical performance range of a typical air-dryer system is summarized by the solid lines in Figure 4.2. It can be observed that the best dryer heat requirements can vary from 1400 Btu/lb_{evap} to 1700 Btu/lb_{evap} depending on the exhaust recirculation ratio, dryer exhaust humidity limits, or the limitations on the dryer inlet and exhaust temperatures. For example, dryer inlet and exhaust temperatures may need to be controlled in order not to damage temperature-sensitive feedstocks. Thus, given a specific drying duty and/or air-dryer design

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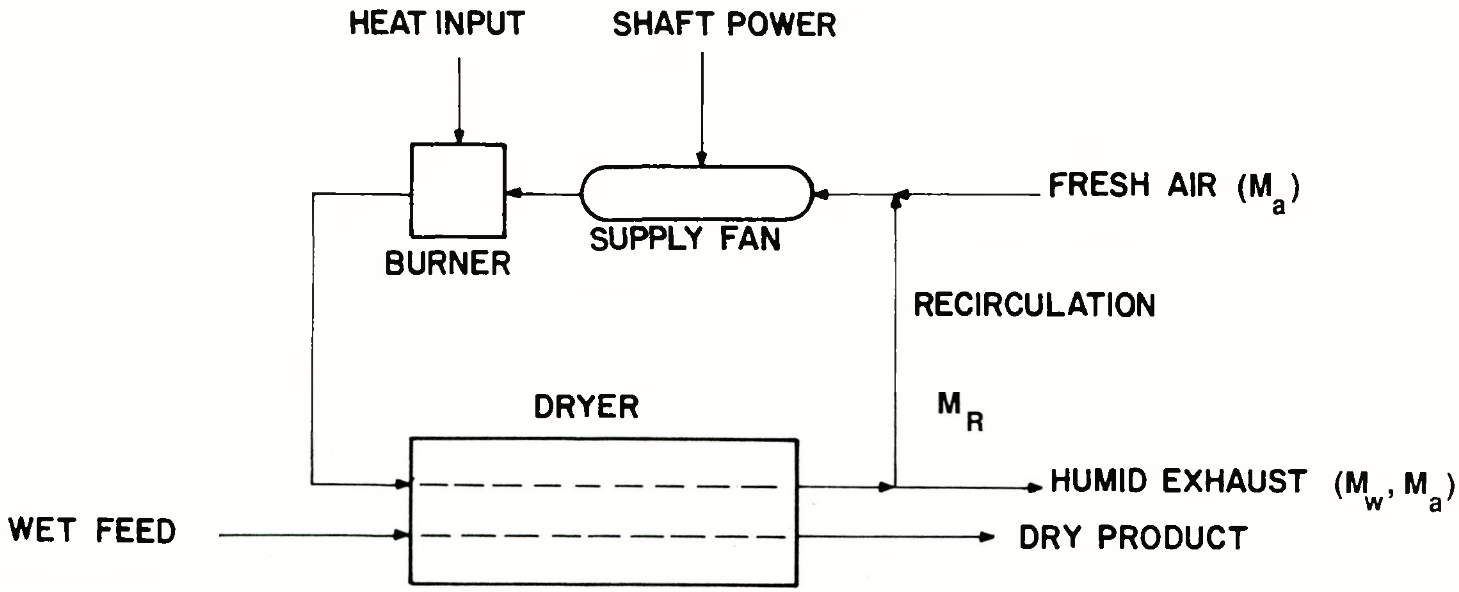

DEFINE: RECIRC. RATIO $(R_f) = \frac{M_R}{M_a}$

HUMIDITY RATIO $(R_w) = \frac{M_w}{M_a}$

Figure 4.1 Schematic of Air-Drying System

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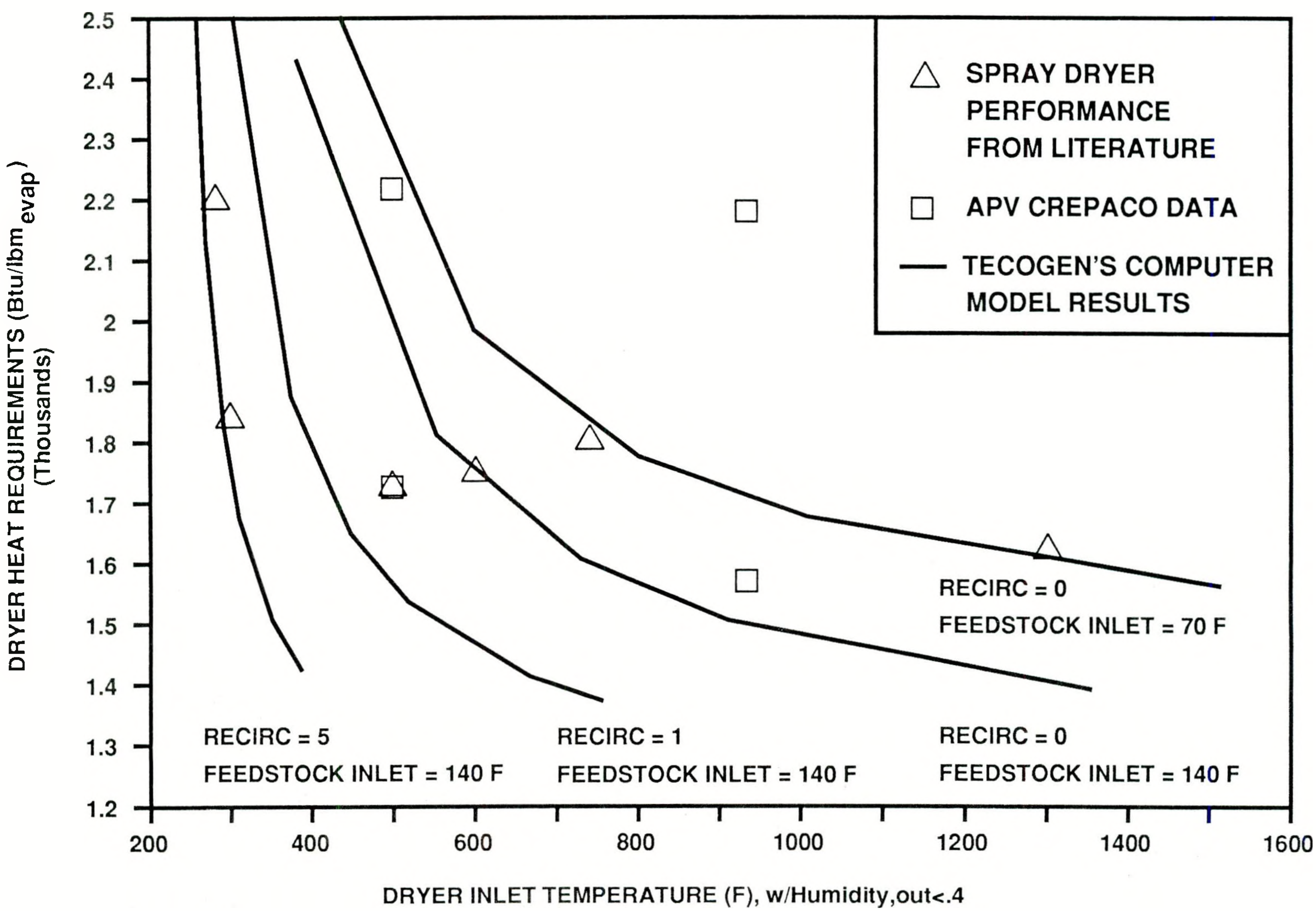

Figure 4.2 Air Dryer (Standard) Cycle Performance Heat Input vs. Dryer Inlet Temperature at 220°F Stack Temperature

criteria, the operating state point for the air-dryer system can be identified using Figure 4.2. Superimposed on the theoretical performance (solid) curves of Figure 4.2 are the reported performances of actual air dryers from the literature (triangular data points). Data reported by a dryer manufacturer for their state-of-the-art spray dryer performance is identified with the square data points.

In general, the approach that has been taken in designing air dryers for high drying rates and low energy requirements has been the following:

1. Using as high a supply temperature as possible without damaging the feedstock during drying or incurring substantial heat transfer losses. An inlet temperature limit of 800 to 1000°F is typical for non-heat-sensitive materials. An inlet temperature of 400 to 500°F is typical for heat-sensitive materials.

2. Using a high dryer supply flow rate. This is ultimately limited by the acceptable amount of electrical power needed to drive the fans.

3. Using high recirculation rates to raise the exit humidity, but also allowing lower dryer supply temperatures. Maximum humidity levels are normally limited to about 0.4 pound of water per pound of air because of combustion air requirements.

4. Using advanced heat transfer techniques to get high levels of heat and mass transfer per unit area.

As shown in Figure 4.2, the highest allowable exhaust humidity (R_w) should be used to minimize dryer heat input requirements. Unfortunately, combustion limitations usually require humidity levels to be below 0.4 pound of water per pound of air. Dryer inlet temperatures must also be considered when selecting the air-dryer operating state point. Figure 4.2 indicates relatively high dryer inlet temperatures are necessary if high humidity levels and no recirculation of exhaust air are to be used. In fact, humidities above 0.2 are not typical due to the high dryer inlet temperatures required. Dryer inlet temperatures can be reduced by providing recirculation of dryer exhaust into the dryer inlet. However, this also increases the size of the air dryer as the volume flow rate through the dryer increases.

The use of exhaust recirculation also increases the dryer heat requirements (albeit very slightly) due to the increased fan power requirements. Figure 4.3 displays the air-dryer inlet temperature as a result of recirculating some of the air-dryer exhaust back into the dryer inlet. A dryer inlet temperature decrease of

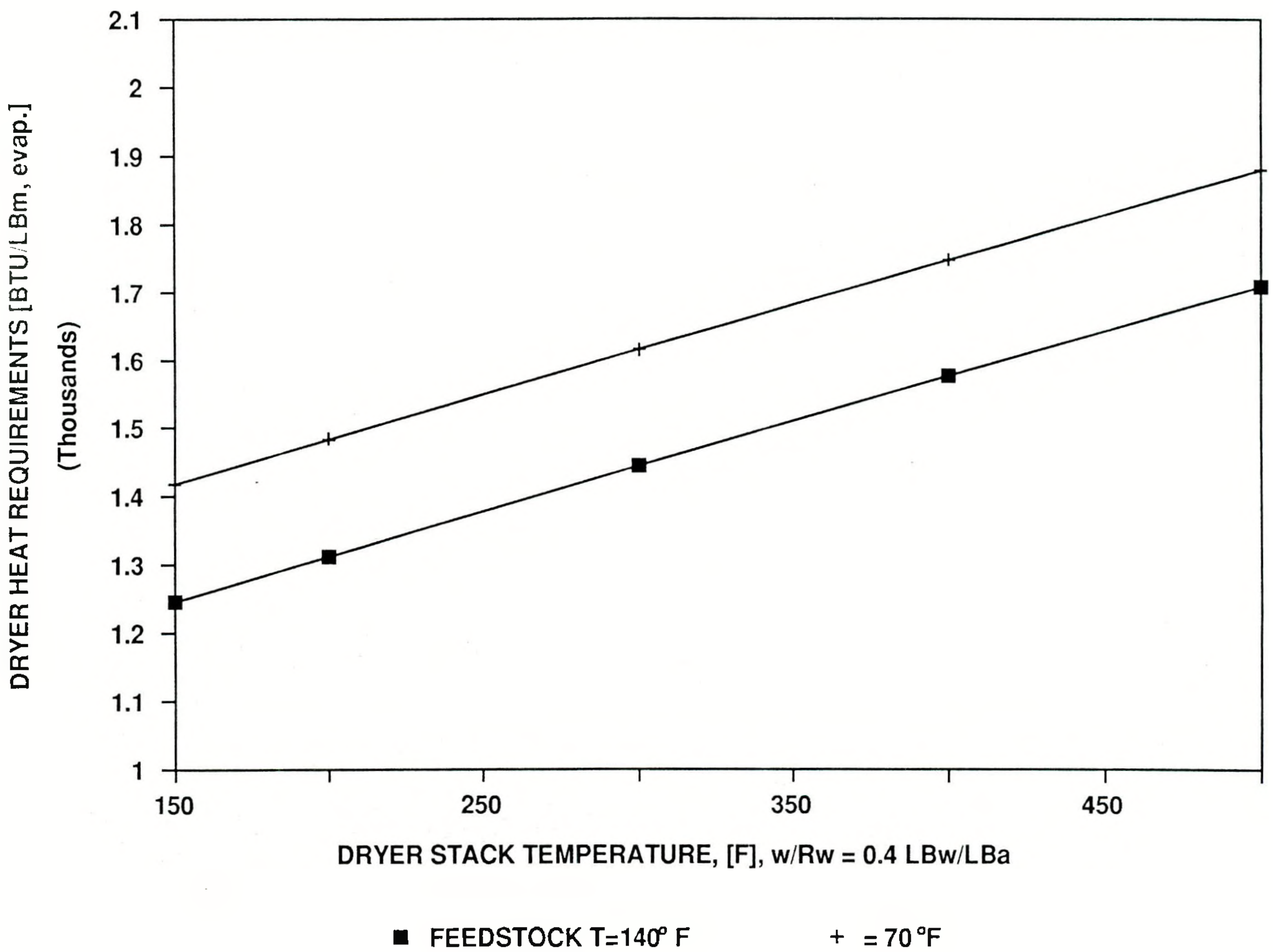

Figure 4.3 Air-Dryer Cycle Performance
Effect of Preheating Feed., w/Recirc. = 0

800 to 1000°F is possible using a recirculation ratio of as little as 1:1; i.e., the ratio of recirculation flow rate to the fresh air inlet flow rate. Feedstock preheating with waste exhaust air from the dryer would have a significant effect on the air-dryer heat requirements, as can be observed in Figure 4.2. Unfortunately, preheating the feedstock to 140°F cannot be performed with the air-dryer exhaust at temperatures below 200°F without requiring very large preheaters. As will be observed in the next section, steam atmospheric dryers can utilize latent as well as sensible waste heat from the condensate of the steam dryer's waste steam stream. The availability of this waste steam for feedstock preheating is seen as another advantage of the steam dryer system.

4.1.2 Tecogen's Direct Steam Atmosphere Dryer (DSAD) With Exhaust Steam Recompression

Drying in a superheated steam atmosphere with exhaust steam recompression offers a better approach for reducing the energy requirements in many applications where air is used now. In addition, superheated steam has several other potential advantages as a drying medium. It eliminates oxidation damage to sensitive products, eliminates explosive hazards with flammable materials, and provides a high humidity environment which is required in some specialized drying applications.

A proposed direct steam dryer with exhaust steam recompression concept is illustrated in Figure 4.4 (an alternative to the direct steam dryer system, i.e., an indirect steam dryer, has also been identified). Superheated steam is circulated by a fan through the heat exchangers and Tecogen's novel drying chamber. In the drying chamber, moisture is thermally driven from the product and carried off by the recirculating superheated steam. A portion of the superheated steam exhausted from the drying chamber (equal to the amount of moisture removed) is taken off and compressed. The compressed steam then gives up its heat of compression and latent heat of evaporation in the heat exchanger. This is the primary, and possibly only, source of heat for the drying process. If economics dictate, an auxiliary fossil-fuel-fired heat may be provided to reduce the size and cost of the steam recompression system. The liquid condensed in the heat exchanger passes through an expansion valve and is used for preheating the feedstock and then vented.

One of the most innovative features of the proposed concept is the use of a unique design for the steam dryer chamber. A schematic of this steam dryer design (with an integral particle separator) is illustrated in Figures 4.5 and 4.6. A photograph of Tecogen's cold flow laboratory test model is shown in Figure 4.7. High heat transfer and drying rates are achieved by intimate contact of the

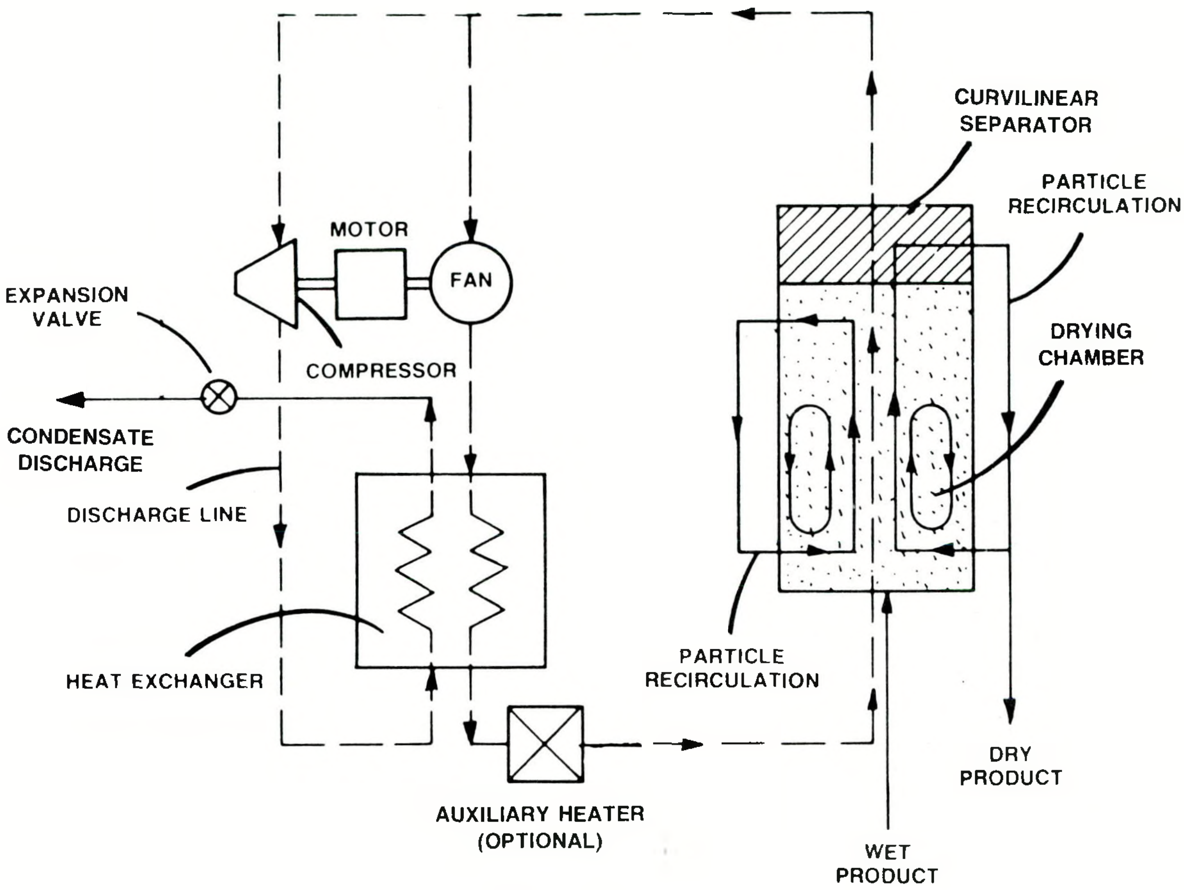

Figure 4.4 Concept for Superheated Steam Dryer with Steam Recompression

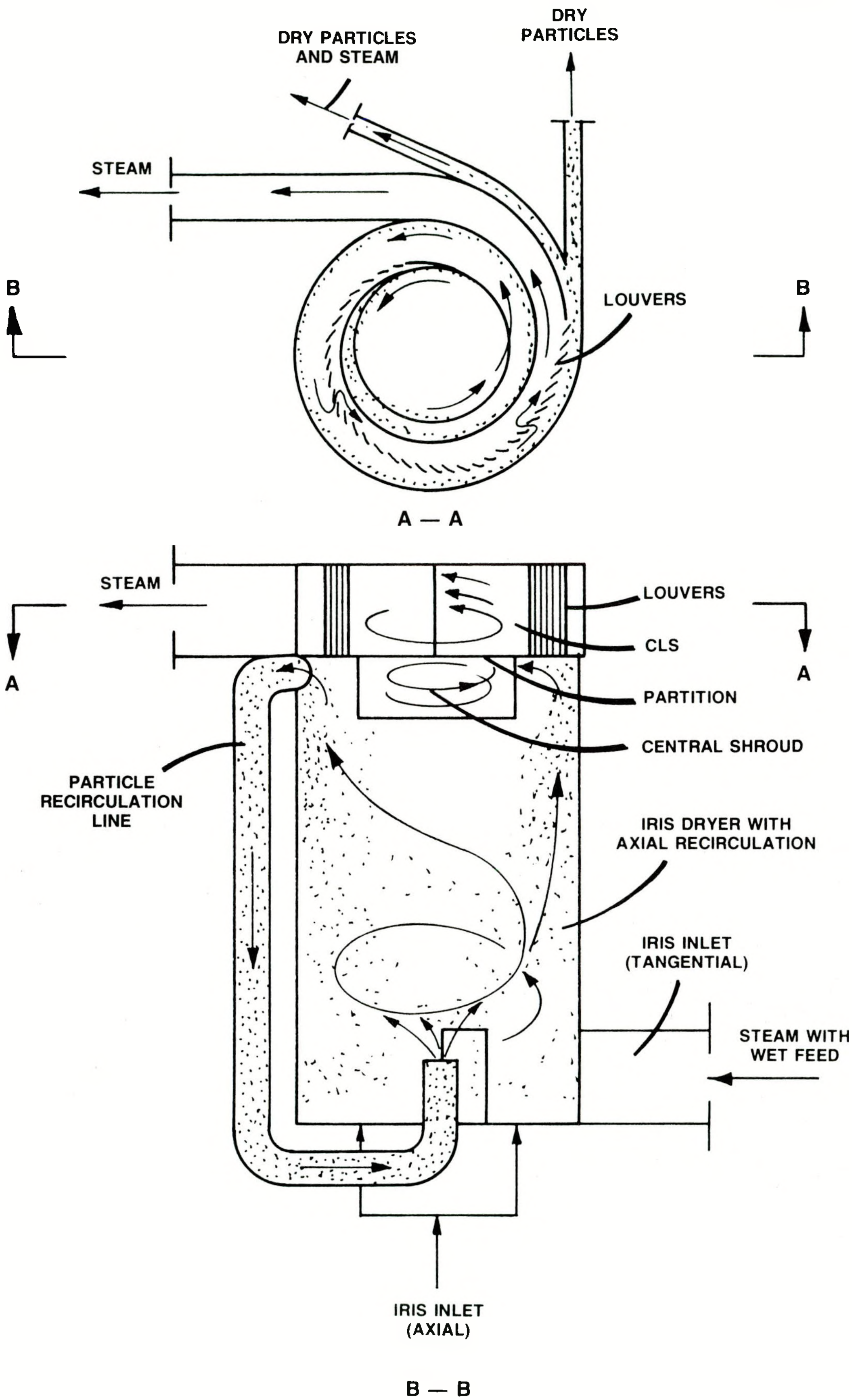

Figure 4.5 CLS in Conjunction with Steam Dryer

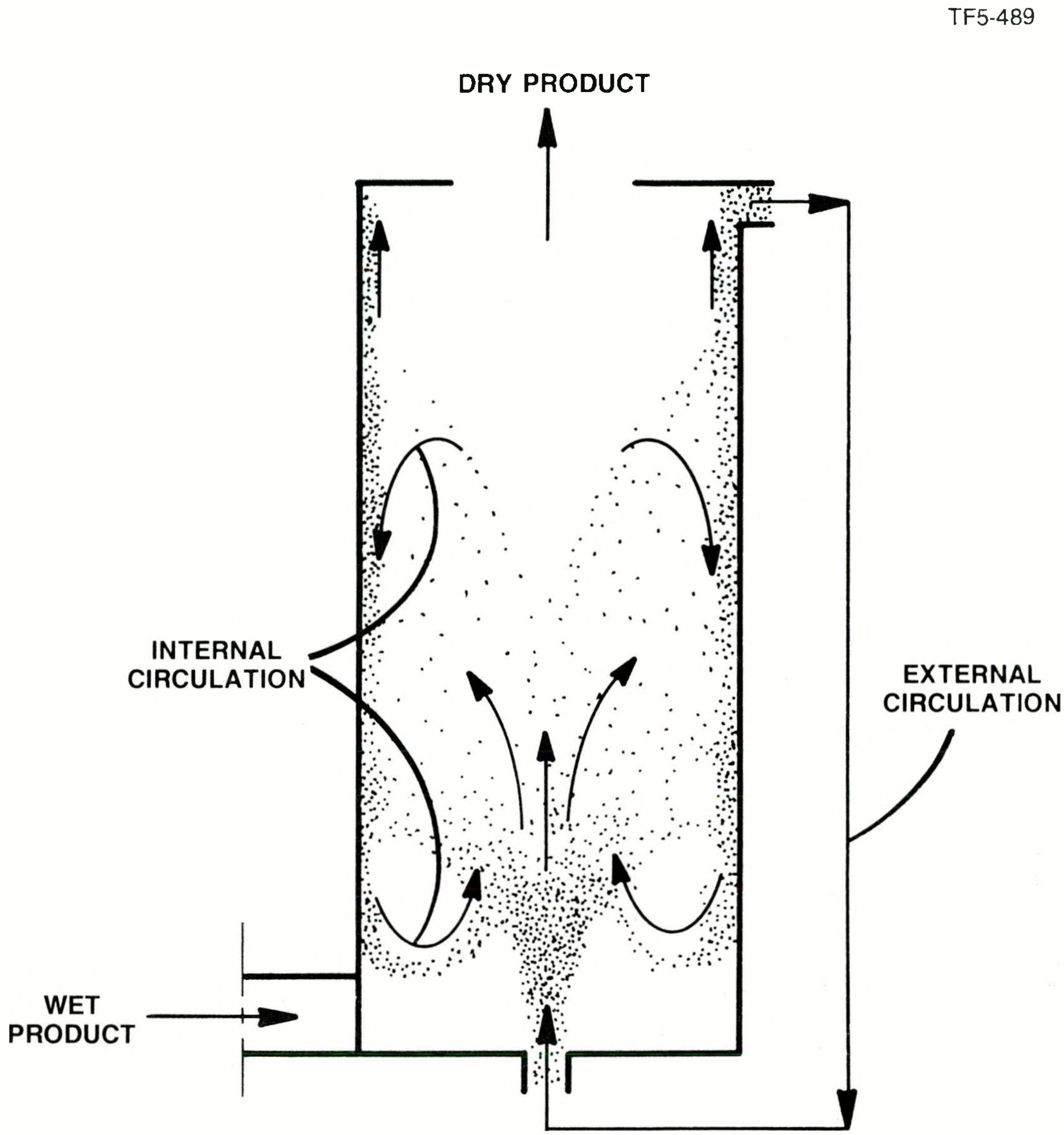

Figure 4.6 External and Internal Circulation in the Steam Atmosphere Dryer

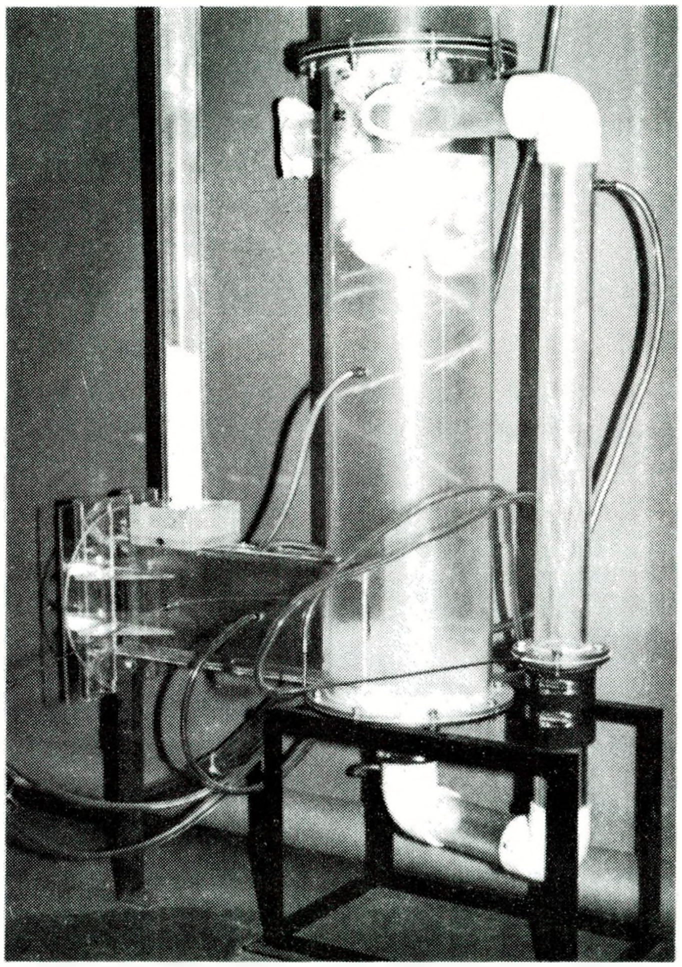

Figure 4.7 Tecogen's Phase I, Cold Flow Laboratory Test Model

superheated steam with the particles being dried. Through high internal and external recirculation rates, the residence time of the particles in the drying atmosphere is increased substantially over levels normally obtained in a circulating fluid bed. All internal and external recirculation is accomplished by the pressure differentials created in the drying chamber by a forced vortex flow pattern. No separate fans are required to achieve the recirculation.

Tecogen Inc. developed a computer model of this unique steam dryer component during Phase I of the project. The computer model is used to determine the interdependence of four basic design parameters that have been determined to characterize the performance of this unique drying chamber. These four design parameters or criteria, listed below, define a region of operation for the dryer.

1. The pressure drop allowable in the dryer from overall system considerations;

2. The minimum percentage of particles that are to be captured and recycled each pass through the dryer;

3. The minimum velocity required to ensure that the particles are carried through the drying zone; and

4. A sufficient residence time in the dryer to dry the particles from the initial moisture content to the final moisture content.

Figure 4.8 shows the acceptable region of operation with respect to inlet velocity and particle size for a dryer of the dimensions of our laboratory test unit with steam entering at 280°F and exiting at 220°F. These limits have been calculated based on a maximum pressure drop of 10 in. H_2O, a 99.5-percent particle capture per pass, a minimum vertical velocity equal to two times the terminal velocity for the particle, and a residence time sufficient to fully dry material entering the dryer with a 50-percent moisture content on a dry basis.

This model will help design and size the pilot-scale steam dryer. Empirical data from both Phase I and Phase II's laboratory testing of this dryer design will then be used to confirm or modify the dryer model relationships shown above, thus providing a more accurate empirical (computer) model of Tecogen's steam dryer concept. Ultimately, this refined model will be used to design and size the prototype-size steam dryer chamber that will be used in Phase III of the DOE/Tecogen project.

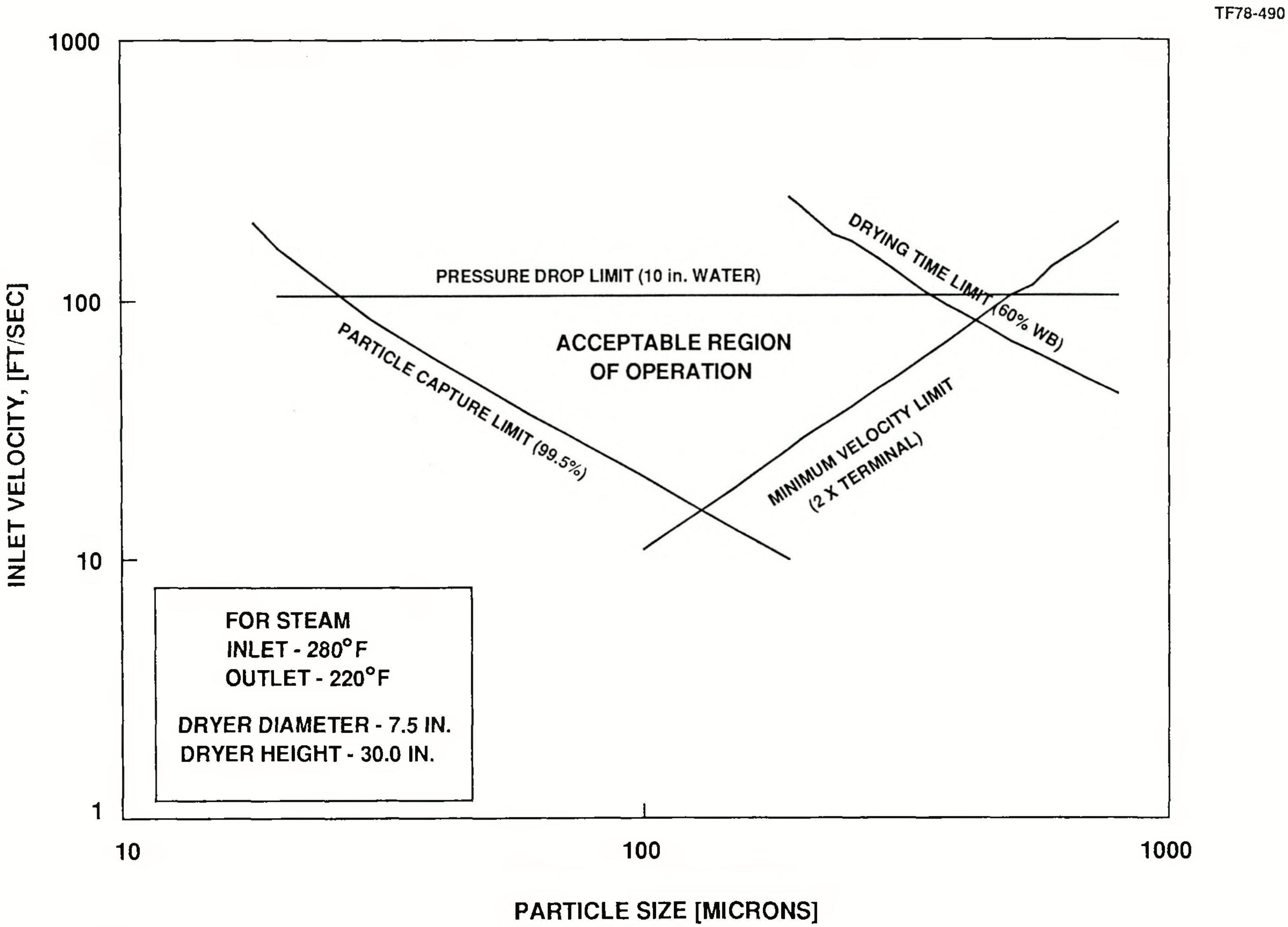

Figure 4.8 Interdependence of Design Parameters for the New Steam Dryer Design

A second innovative feature of the drying chamber design relates to the way the particles are efficiently separated from the superheated steam. This is accomplished by using a specially designed Curvilinear Louvered Separator (CLS). Good separation of the particles is important not only to prevent escape of dried particles but also to prevent fouling of the heat exchanger. The CLS is an efficient particle separator that can be readily integrated with the new steam dryer design.

Some of the advantages of superheated steam drying compared to air drying include:

1. Reductions of 55 percent in the net energy consumption when steam recompression is used (verified with Phase I, Task 2 analysis);

2. Reduced airborne emissions;

3. Improved product quality because hot product is not exposed to oxygen or combustion products;

4. Reduced risk of fire or explosion, particularly for combustible dusty products and products from which a combustible solvent is being removed;

5. Prolonged constant-rate drying zone because diffusion is not a source of resistance to drying and mobility of water in the pores of the material is enhanced;

6. Density of the dried product is increased; and

7. Higher drying rates for wet granular solids.

Tecogen's thermodynamic analysis of the steam atmosphere dryer and steam recompression system used the following system component efficiencies and operating parameters:

Fan/Blower Pressure Rise*	10 to 40 in. wc
Fan/Blower Efficiency	0.60
Elec. Motor Efficiency	0.95
Auxiliary Boiler Efficiency	0.85

*Principal design parameter used in parametric analysis.

Elec. to Thermal Conversion Efficiency	0.3 (11,376 Btu/kWe-hr)
Dryer Ambient Inlet Temperature and R.H.	70°F; 50%
Feedstock Inlet Water Content	50% (D.B.)
Feedstock Outlet Water Content	5% (D.B.)
Auxiliary Steam Boiler Thermal Efficiency	85%
Feedstock Preheated Temperature (Maximum)	140°F
Steam Compressor Thermal Efficiency	0.70
Heat Transfer Loss (Steam Dryer and Steam Reboiler)	1.5% each
Reboiler Pinch Point	8°F
Reboiler Subcool Temperature Drop	25°F
Steam Dryer Outlet Temperature	220°F
Compressor Pressure Ratio*	2.0 to 6.0
Dryer Inlet Temperature*	240° to 600°F

The total energy (reported as Btu per pound of water evaporated from the feedstock) is the sum of the energies required to drive the steam compressor and the steam circulating fan/blower, added to the auxiliary heat needed to sensibly heat the recirculation steam flow to the desired steam dryer inlet temperature.

A computer program was prepared to model the steam atmosphere dryer system using the constants shown above. An extensive parametric analysis was performed. It was found that the system pressure drop, compressor pressure ratio, and the dryer inlet temperature were the three principal design parameters that most significantly affected the steam dryer system's energy requirements and system cost. An example of a typical computer program printout for the direct steam atmosphere dryer system is displayed in Table 4.1.

A discussion of the results of the thermodynamic analysis obtained for the direct steam atmosphere dryer (DSAD) is presented in this section. The cost analysis discussion is presented in Section 7 of this final report.

The thermodynamic advantage of using a steam compressor can be seen as comparable to using a heat pump between a low-temperature reservoir (i.e., the dryer exhaust stream) and a higher-temperature reservoir (i.e., the dryer inlet stream). The coefficient of performance (COP), or heat pump efficiency, for this heat pumping action is defined as the ratio of heat delivered to the dryer recirculating steam to the work of compression required by the steam compressor. Thus, high COP's are desirable, but as will be seen, can be achieved only if low (less than 300°F) steam dryer inlet temperatures are used.

TABLE 4.1

TYPICAL COMPUTER PRINTOUT FOR THE DIRECT STEAM ATMOSPHERE SYSTEM ANALYSIS

	ELEC. MOTOR EFF.	0.95
	AUX. BOILER EFF.	0.85
	SUBCOOL DT PINCH	25
	ELEC./FUEL CONV. EFF.	0.3
	DTSUPER	25
	DTPINCH	8
	COMP. PRES. RATIO	3.5
	COMP. EFF.	0.7
	BLOWER PRES. DP (IN. H20)	10
	BLOWER EFF.	0.6
PREHEAT:Y/N=1/0	1 FEEDSTOCK TEMP.	140
	STACK TEMP.	220
	DRYER OPER. PRES.(PSIA)	14.696
OPT.CYCLE:Y/N=1/0	0 DRYER INLET TEMP.(Tr; F)	280
	% WATER,in (d.b.)	0.5
	% WATER,out (d.b.)	0.05
	WET BULB TEMP.	212
	PARTICLE Cp	0.42
	STEAM Cp	0.485
	STEAM SP. HT. RATIO	1.32
	DRYER OPERATING TEMP.(F)	212
	FRACTION HEAT LOSS FROM DRYER	0.015
	FRACTION HEAT LOSS FROM REBOILER	0.015
	Hv(Tr,Pr)	1168.57
	Hv(Tso,Pso)	1139.47
	(Wi-Wo) x Hv(Tr,Pr)	525.86
	(Wi-Wo) x Hv(Tso,Pso)	512.76
	CPp x (Tin- Twb)	-30.24
	Wi x (Tin-32) - Wo x (Twb-32)	45.00
CALCULATED RECIRCULATION "R"		38.61
BLOWER POWER REQ.S (BTU/Lbw,evap)		118.62
BLOWER CIRC. OUTLET TEMP.		226.33
STEAM COMP. POWER (Btu/LBm-Hr,evap)		167.19
STEAM COMP. POWER (Btu/LBm/Hr,recirc.)		4.33
	Hfg @PCO	924.20
	Tissen. F	564.72
	Tsat. @ Pco	283.28
FLASH Stm. Cond. Temp.		212.00
COMPRESSOR OUTLET TEMP. (F)		308.28
CALCULATED INJ. WATER RATIO (R2)		0.12
Qsource (Btu/Lbm,recirc.)		27.75
REBOILER HOT OUTLET SUBCOOLED TEMP.		251.33
MAX. REBOILER TEMP. OUT. (F)		283.56
(iterated pinch point temp.)		275.28
(iterated pinch point temp.)		275.28
(original ratio of heat pump ht req.d)		0.86
ACTUAL FRACTION OF HEAT PUMP HEAT USED		0.86
FINAL PINCH POINT TEMP.		275.28
FINAL DT PINCH		8.00
ACTUAL HEATED OUTLET REBLR TEMP.		275.82
COMPRESSOR POWER REQ.S (Btu/Lbm,evap.)		144.58
AUX. BLR. HT. IN @Tr (Btu/Lbm,evap.)		92.18
C.O.P. RECOMP. SYSTEM (w/conv. eff=1)		6.41
TOTAL Q1(Btu/Lbw,evap.;with conv.eff.]		1015.68
DRYER EFF.(w/conventional eff. defns.)		1.16
TWO STAGE DRYER REHEATING RATIO:Q2/Q1		833.88
3RD STAGE DRYER REHEATING RATIO:Q3/Q1		773.28
FOUR STAGE DRYER REHEATING RATIO:Q4/Q1		742.98

The performance of a direct steam atmosphere dryer with recompression is summarized in Figures 4.9 through 4.16. The minimum heat requirements can be observed (Figure 4.9) to be 1015 Btu/lb_{evap} at approximately 275 to 280°F dryer inlet temperatures. This optimum performance state point requires no additional heat input from the auxiliary heater. The energy savings between this state point and the comparable air-dryer operating condition is approximately 40 percent at an air-dryer stack temperature of 150°F, and approximately 85 percent at an air-dryer stack temperature of 500°F (assuming air recirculation ratio equals 0 and humidity equals 0.4). Clearly, this is a substantial performance improvement.

In order to achieve the locus of state points shown in Figure 4.9, the steam compressor pressure ratio was varied until no auxiliary heat input was required to attain the dryer inlet temperature shown. The steam compressor's pressure ratio and, hence, the steam recompression system's COP can consequently be observed to change as the desired dryer steam inlet temperature increases from 240°F to 350°F. Figures 4.10 and 4.11 show the increase in steam compressor ratio and the subsequent decrease in the system's COP.

Although there is a clear minimum heat requirement identified in Figure 4.9, the performance of the steam dryer cycle at the low dryer inlet temperatures requires that a large amount of recirculated steam flow rate be maintained through the steam dryer. Consequently, the fan/blower power requirements for the steam cycle are significant and are comparable to those of the steam compressor. Figure 4.12 displays the Btu/lb_{evap} of mechanical energy required to drive the fan/blower and steam compressor. For example, at the 275°F (minimum) state-point operating condition, the fan/blower and steam compressor would each require a 180-kW electric motor drive for 5000 pounds of evaporation duty.

The volume of steam flowing through the dryer is found to be very high at dryer inlet temperatures below 400°F when compared with the flow rate of air in standard air dryers. This implies that the size of the dryer system piping and components would need to be made larger if equivalent system pressure drops were to be maintained. Figure 4.13 displays the significantly greater volume flow rate of steam required at 200 to 400°F as compared with air volume requirements in dryers that typically operate at temperatures above 500°F. It is important therefore to design the steam dryer system with lower volume flow rates (i.e., reduce the steam recirculation ratio (R)).

The steam recirculation rate (R) can be <u>decreased</u> by <u>increasing</u> the steam dryer inlet temperature. However, as can be observed in Figures 4.10 and 4.11, the steam compressor pressure ratio must be increased with a subsequent reduction in the steam compressor's COP. Thus, although an increase in dryer inlet

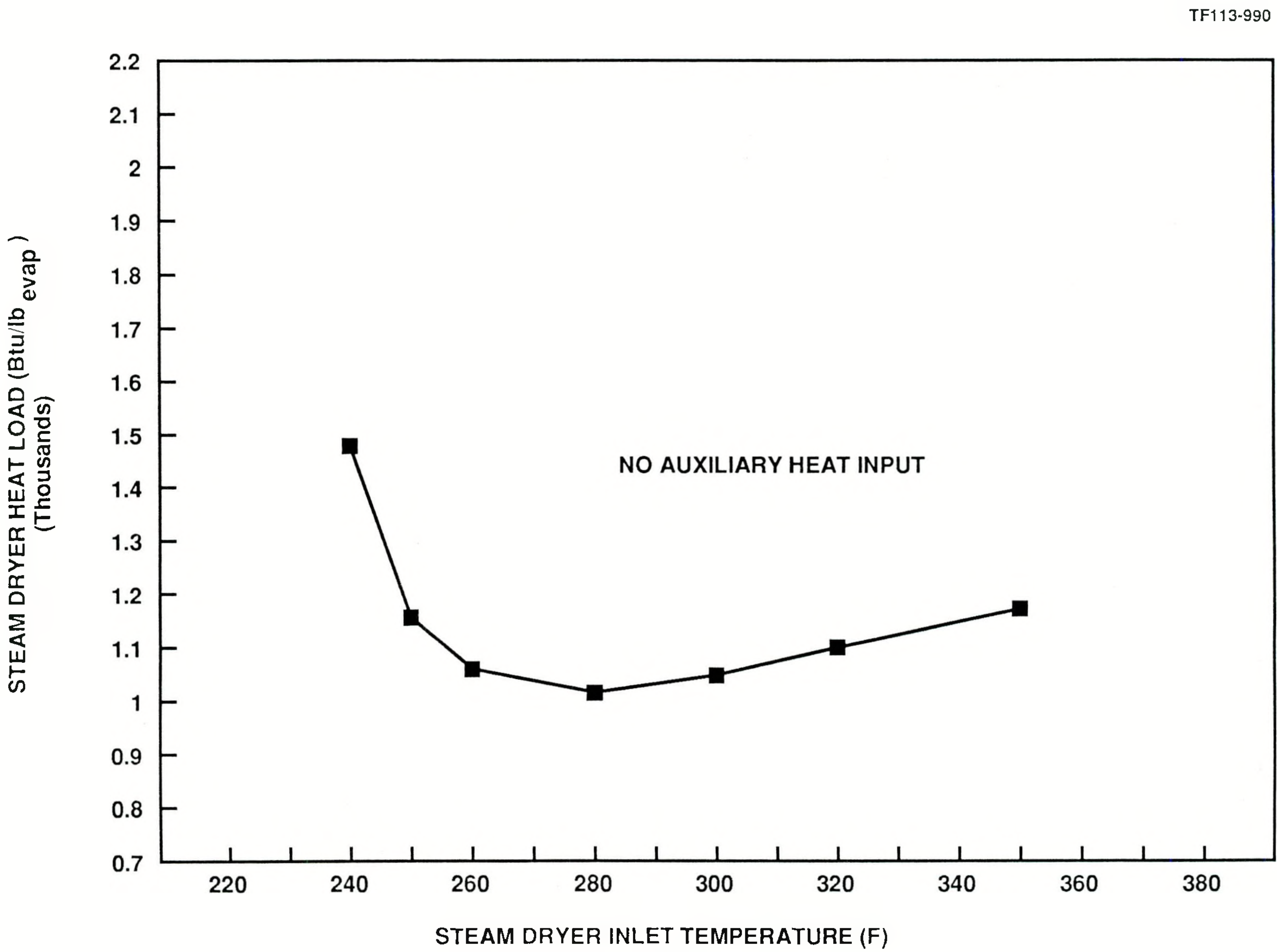

Figure 4.9 Direct Steam Dryer Cycle Performance (Single-Stage Dryer System)
(System Pressure Drop = 10 in. wc)

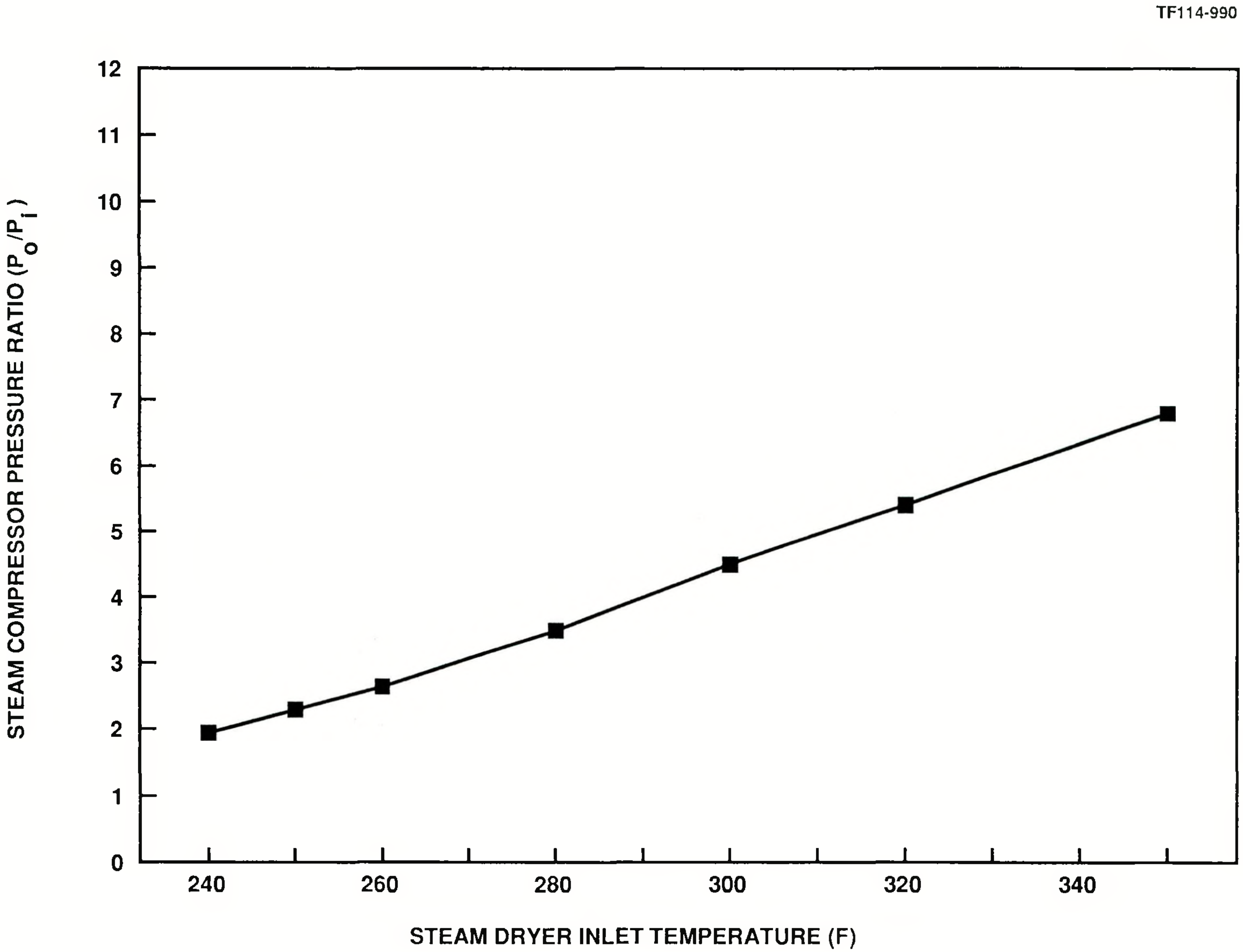

Figure 4.10 Direct Steam Dryer Cycle Steam Compressor Pressure Ratio (P_o/P_i) (System Pressure Drop = 10 in. wc)

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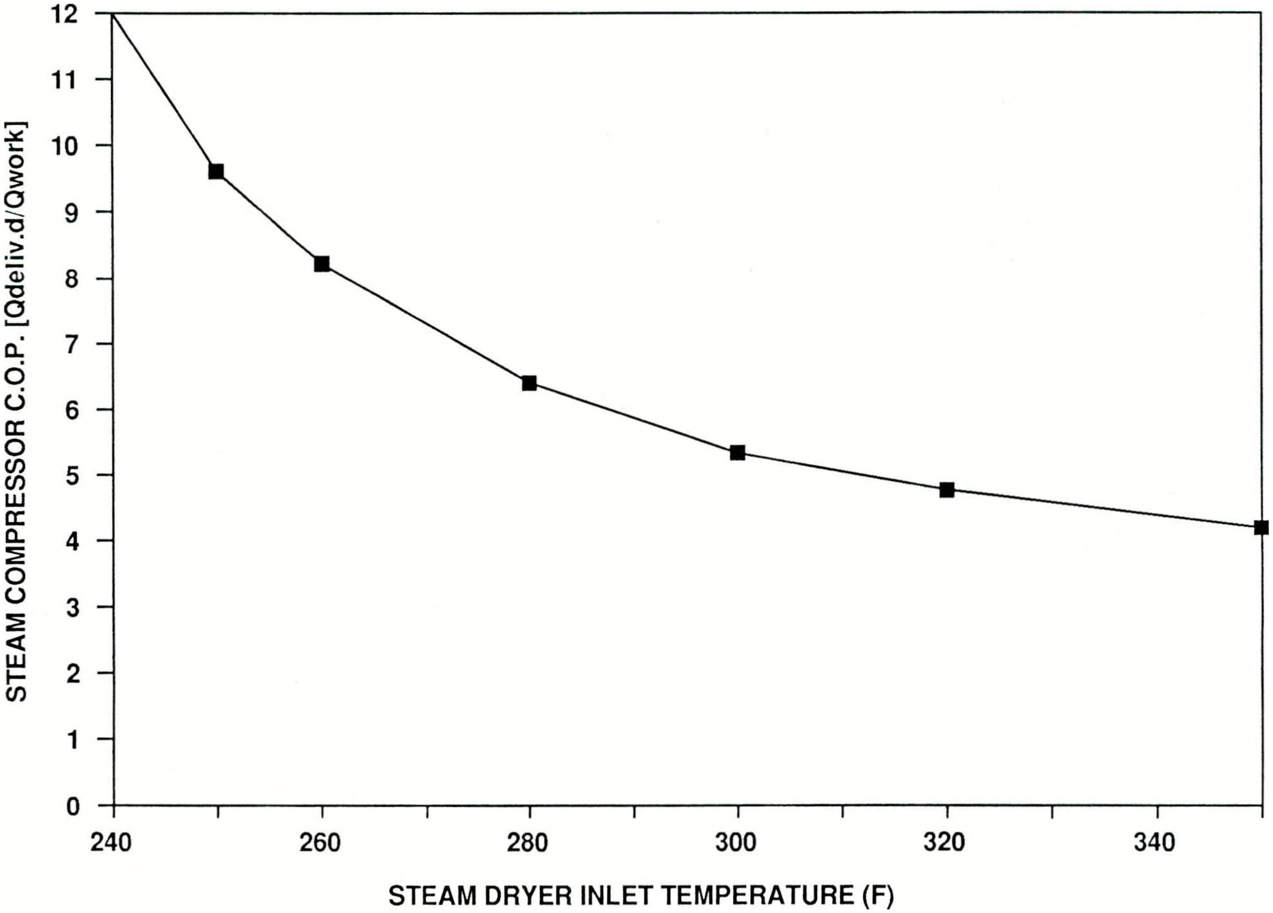

Figure 4.11 Steam Dryer Cycle Design Point
Steam Comp. COP (w/Fuel Conv. = 1)

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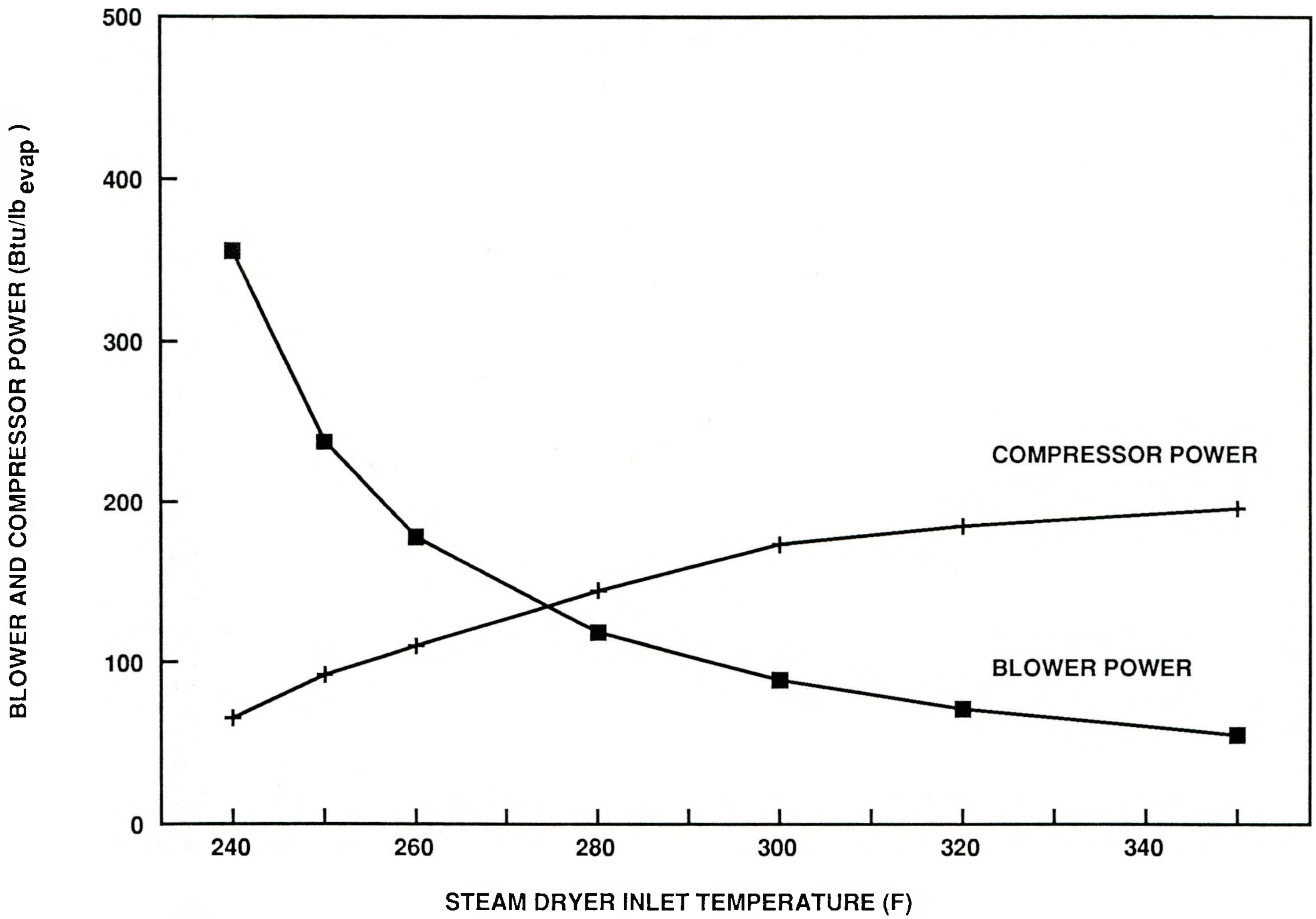

Figure 4.12 Direct Steam Dryer Cycle Design Point Blower and Compressor Power Consumption (System Pressure Drop = 10 in. wc)

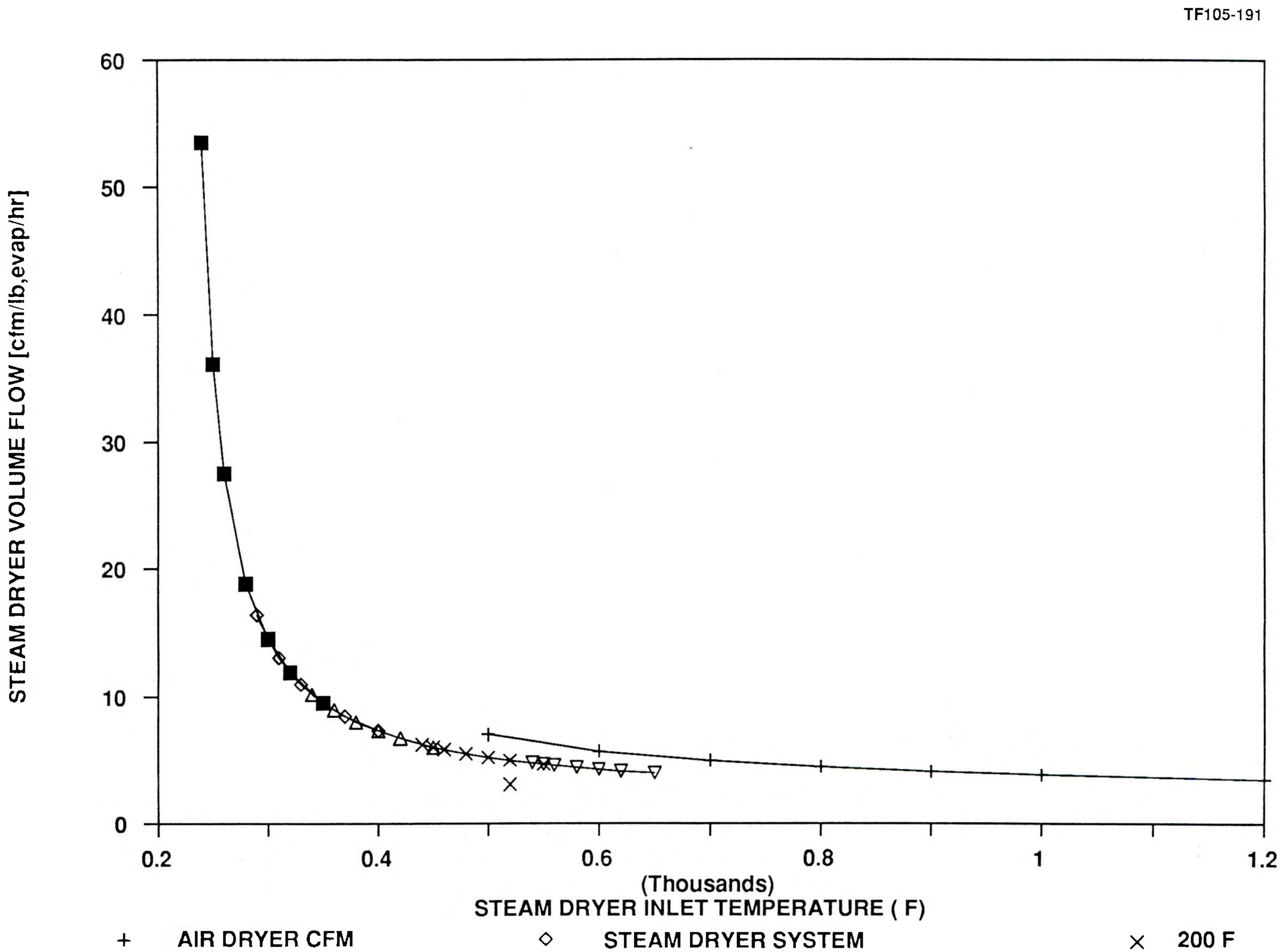

Figure 4.13 Comparison of Typical Volume Flow Rates for DSAD and the Air Standard Dryer System

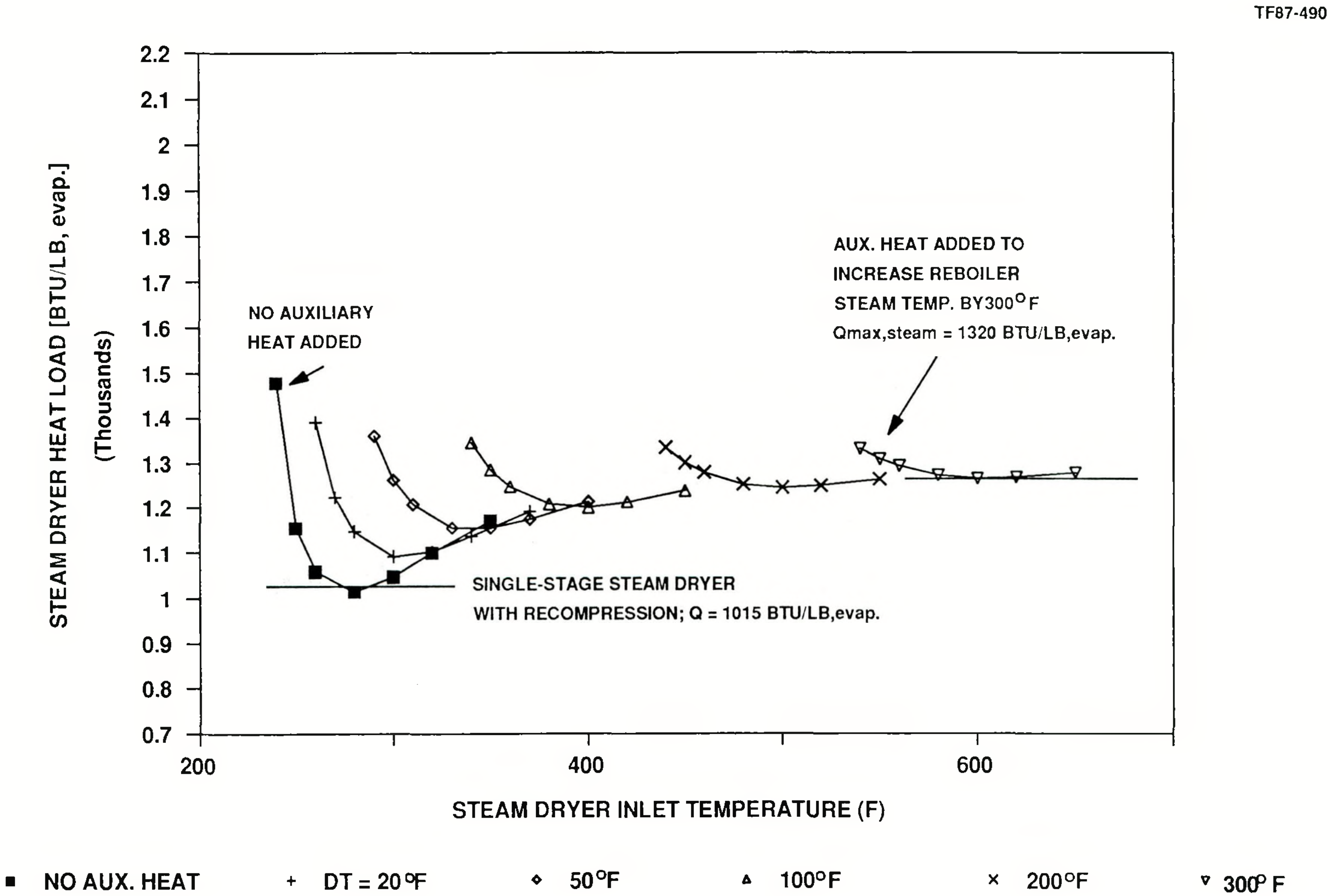

Figure 4.14 Optimum Steam Dryer Cycle
Effect of Aux. Heat Input on Single-Stage Steam Dryer System

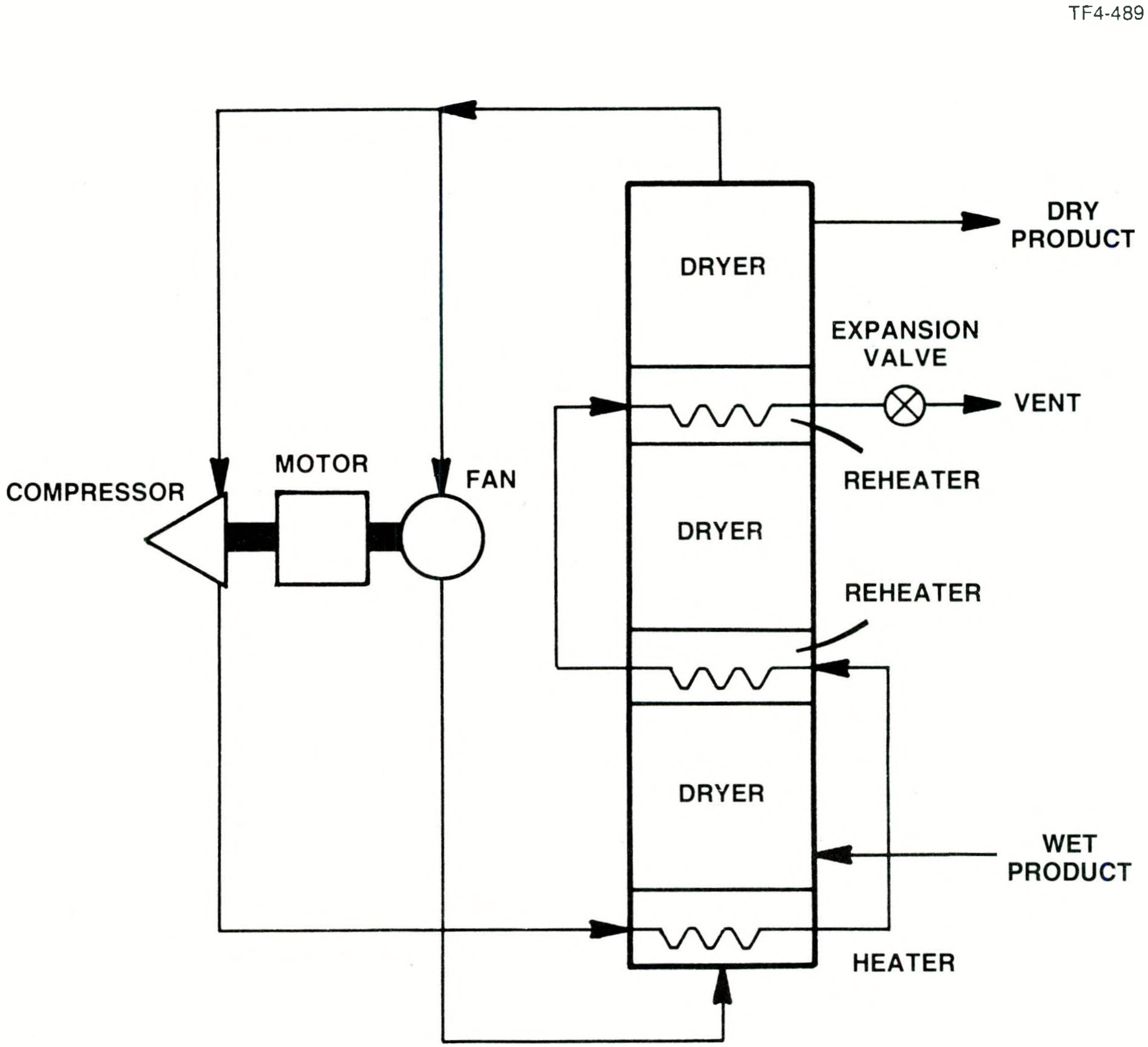

Figure 4.15 Superheated Steam Heat Pump Dryer with Two Reheats (Schematic)

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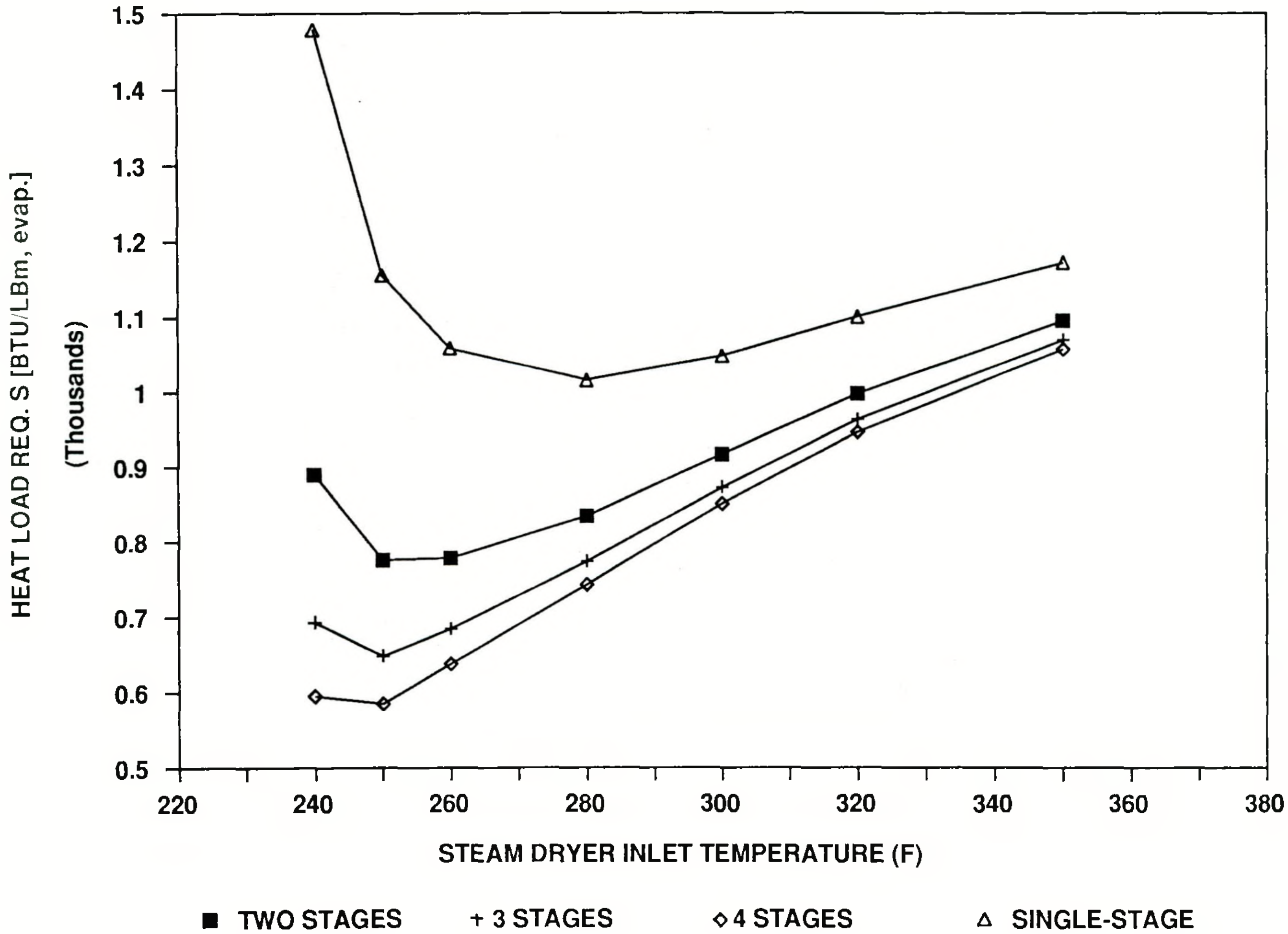

Figure 4.16 Optimum Steam Dryer Cycle Performance with Staged Dryer Heating

temperature results in a decrease in the recirculation rate, the use of auxiliary heat input from a fuel-fired steam heater would be required. This results in an increase in the amount of heat required to evaporate one pound of water from the feedstock.

In the event that higher steam inlet dryer temperatures are desired and auxiliary heat input is used to increase the steam temperatures exiting the steam reboiler, the use of steam as the drying medium still provides an advantage over the conventional air-dryer system in that a maximum heat input limit is reached ($Q_{max,steam}$) that is still 7 to 42 percent lower in energy requirements than the best air-based dryer performance. For example, as shown in Figure 4.14, the need for higher steam dryer inlet temperatures results in the locus of state points (with the same compressor pressure ratio as originally displayed in Figure 4.9) to equilibrate to a steady state and maximum heat input of approximately 1320 Btu/lb_{evap} as auxiliary heat input to the steam is increased. Shown in Figure 4.14 are six curves, each the locus of state points originally displayed in Figure 4.9. However, now each curve displays an increase of superheating steam temperature of 20°F, 50°F, 100°F, 200°F, and 300°F greater than the original optimum steam cycle reboiler discharge temperatures shown in Figure 4.15. The effect observed in Figure 4.14 is the transition from a system that has 100 percent of the dryer recirculation steam heated with a steam heat pump to one that has 90 to 95 percent of its heating derived from the auxiliary heating system. For such a system, the fan/blower and steam compressor power requirements are considerably lower, requiring as little as 30 kW for the blower and steam compressor for 5000 pounds of evaporation duty.

It is interesting to note, therefore, that two distinct regimes of steam dryer operation can now be discerned: steam dryer temperatures above and below 450°F. For heat-sensitive feedstock materials that typically require 450°F or cooler dryer inlet temperatures, a steam recompression system is clearly beneficial and is recommended. For less heat-sensitive feedstock materials that typically can tolerate higher inlet dryer temperatures (i.e., above 450°F), a steam recompression system is less desirable and perhaps should not be included as part of the steam atmosphere industrial dryer system. In each instance, however, the use of a steam atmosphere dryer is still more efficient than the comparable air-based dryer. For the state points described here, an average energy reduction of 25 percent is still attainable at steam dryer inlet temperatures of 650°F.

Another means of reducing the recirculation rate and thus reducing the fan/blower power requirements is available by employing a staged dryer system, as shown in Figure 4.14. The effect of using up to four stages of steam dryer heating is shown in Figure 4.16. The benefit of a staged system is apparent, reducing the steam dryer heat input from 1015 Btu/lb_{evap} in a single-stage dryer to 600 Btu/lb_{evap} in a four-stage dryer. As can be observed in Figure 4.15, the incremental benefit of using a staged heating process is significant for the second and perhaps third stages. However, the incremental energy saving for more than three stages probably will not outweigh the cost for the added dryer system complexity and, hence, cost. Therefore, it is recommended that at least a two-stage process should be studied as a means of decreasing the dryer heat requirements.

4.1.3 Proposed Indirect Steam Atmosphere Drying System (ISAD) With Exhaust Steam Recompression

An alternative means of reducing the steam recirculation flow rate and thereby reducing the steam fan power parasitic is to use an indirect steam atmosphere drying system as outlined in Figure 4.17. In this alternative steam dryer the feedstock drying is produced by conduction heat transfer from the dryer's steam-heated jacket in addition to convection drying from the recirculated or transport steam medium.

A thermodynamic analysis of this system reveals that the very low recirculation (transport) steam flow rate requirements cause this system to be less sensitive to the dryer's fan power parasitic. In fact, the very low recirculation flow rate allows the overall system pressure drop to increase without critically increasing the power required for the steam fan. The heat input requirements for the indirect steam atmosphere dryer are displayed in Figure 4.18 and can be compared with the previously quoted performance of the direct steam atmosphere dryer system. Thus, it can be observed that the heat input requirement for an indirectly heated steam system (i.e., one that utilizes a steam jacket) with a 40-in. system pressure drop results in a 20-percent decrease in heat requirements when compared with the direct steam atmosphere dryer with a 10-in. overall system pressure drop.

The low recirculation flow rate (R) is a net result of providing most of the dryer's heat requirements via conduction heat transfer between the dryer's jacketed wall and the feedstock's water-laden particle. The amount of heat required to preheat, vaporize, and superheat the water entrained in the feedstock is fixed at approximately 1120 Btu/lb_{evap} whether the heating is to be done with the DSAD or the ISAD system. If a large portion of this heating is available via conduction then less heat will be needed from the superheated transport steam medium. If less heat is required from the transport steam its flow rate through the dryer can be reduced. The net result is a decrease in the recirculation flow rate ratio (R) and thus a decrease in the impact of the system's pressure drop on steam fan parasitics.

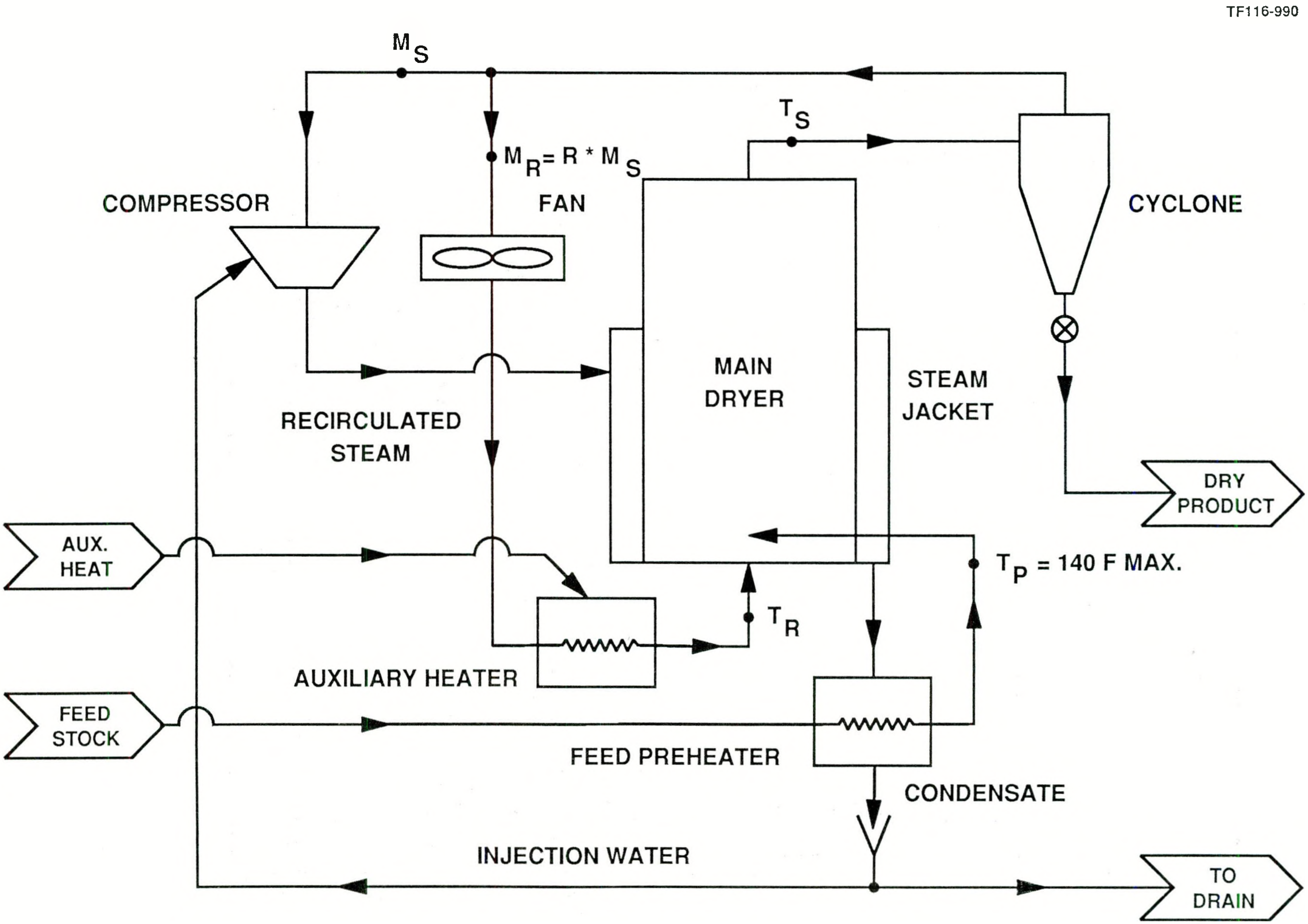

Figure 4.17 Indirectly Heated Steam Dryer

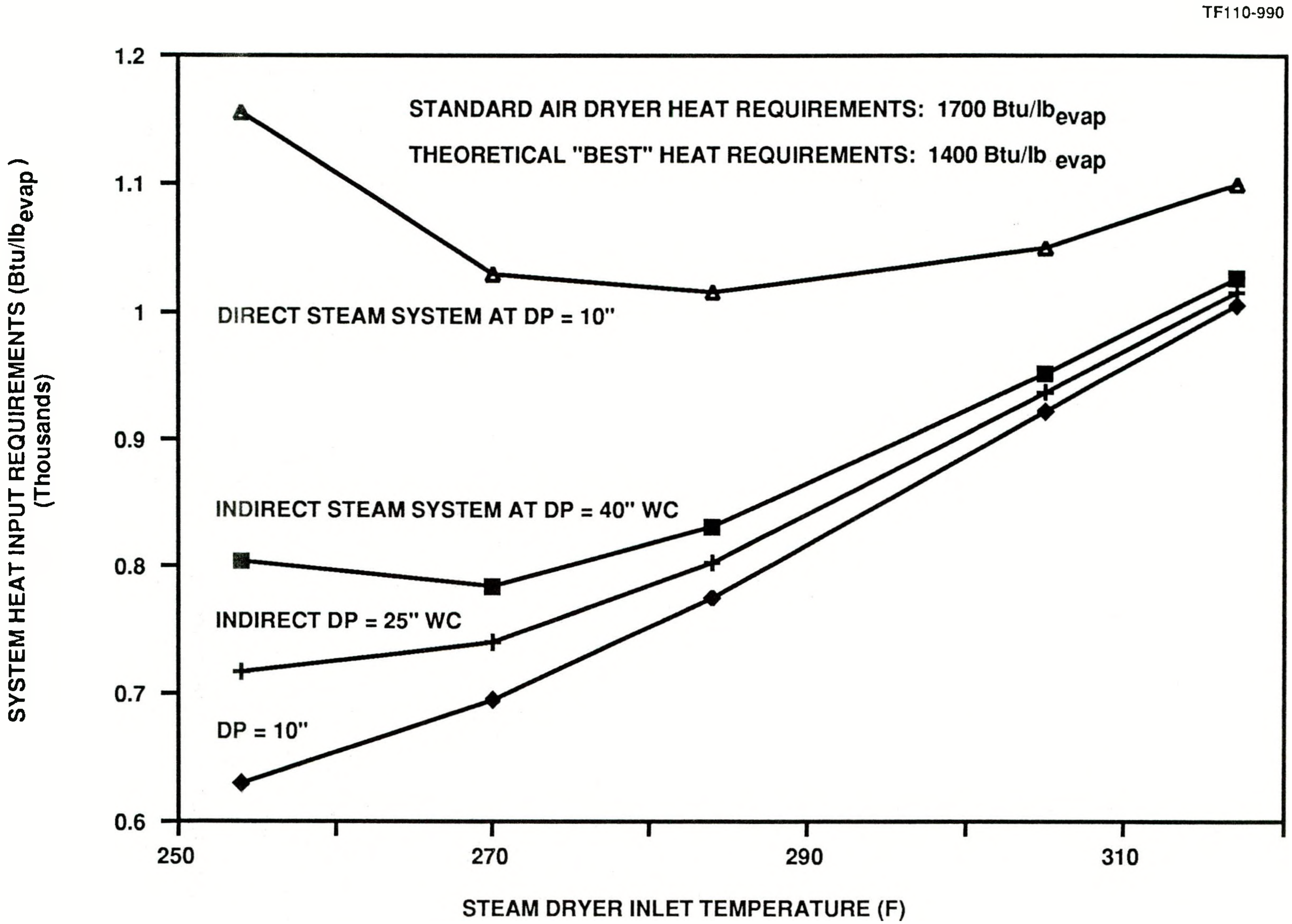

Figure 4.18 Steam Dryer System Performance for Various System Pressure Drops

The thermodynamic analysis conducted for the ISAD revealed that 90 percent of the heating requirements of the feedstock can be met by the steam compressor's steam output. A typical display of this finding is shown in Figure 4.19. The direct consequence of this is a drastic reduction in the dryer's recirculation ratio (R) when compared with a DSAD system. This effect can be observed in Figure 4.20.

With a lower recirculation ratio the dryer system's overall volume flow rate is reduced. Consequently, the fan power parasitic is reduced, which ultimately provides a more efficient steam atmosphere drying system. For example, Figure 4.21 displays a plot of the system's steam compressor and fan power consumption. The fan power is drastically reduced compared to the fan power requirements previously displayed in Figure 4.12. In fact, the lower fan parasitic in the ISAD system results in the constant need for an auxiliary steam heater for the transport steam. Unlike the DSAD's system, the ISAD's fan power (i.e., heat) input to the system is not sufficient to avoid the use of an auxiliary steam heater. Certainly, this is an advantage for the ISAD system as the thermodynamic irreversibility of the heat input into the transport steam is reduced if it is performed via heat transfer rather than by producing heat energy input from the mechanical energy required to rotate the steam fan.

The ISAD does have a small drawback in that it requires a slightly larger steam compressor discharge pressure, as shown in Figure 4.22. This also results in a slight decrease in the system COP when compared to the DSAD system, as can be observed in Figure 4.23. These consequences, however, are minor compared to the significant improvements in energy requirements. Compared to the best air standard system where energy heat inputs may range from 1400 to 1700 Btu/lbm, the ISAD system requires an energy input of only 625 to 800 Btu/lb_{evap} for an energy savings of from 43 to 63 percent.

A parametric analysis performed on the ISAD system was also able to detect only small effects of changes in the system pressure drop or compressor discharge pressure (i.e., steam dryer pinch point ΔT) on the system's energy input. Figures 4.24 and 4.25 display the results of several parametric studies using the system pressure drop (DP) and the system pinch point (ΔT). The pinch point is the temperature difference between the dryer's inlet steam temperature and the compressor's saturation temperature corresponding to the compressor discharge temperature.

Similarly, a parametric study of the effect on system energy input requirements caused only by a change in the dryer's discharge temperature is given in Figure 4.26. A change in dryer energy input of less than 2 percent is discerned.

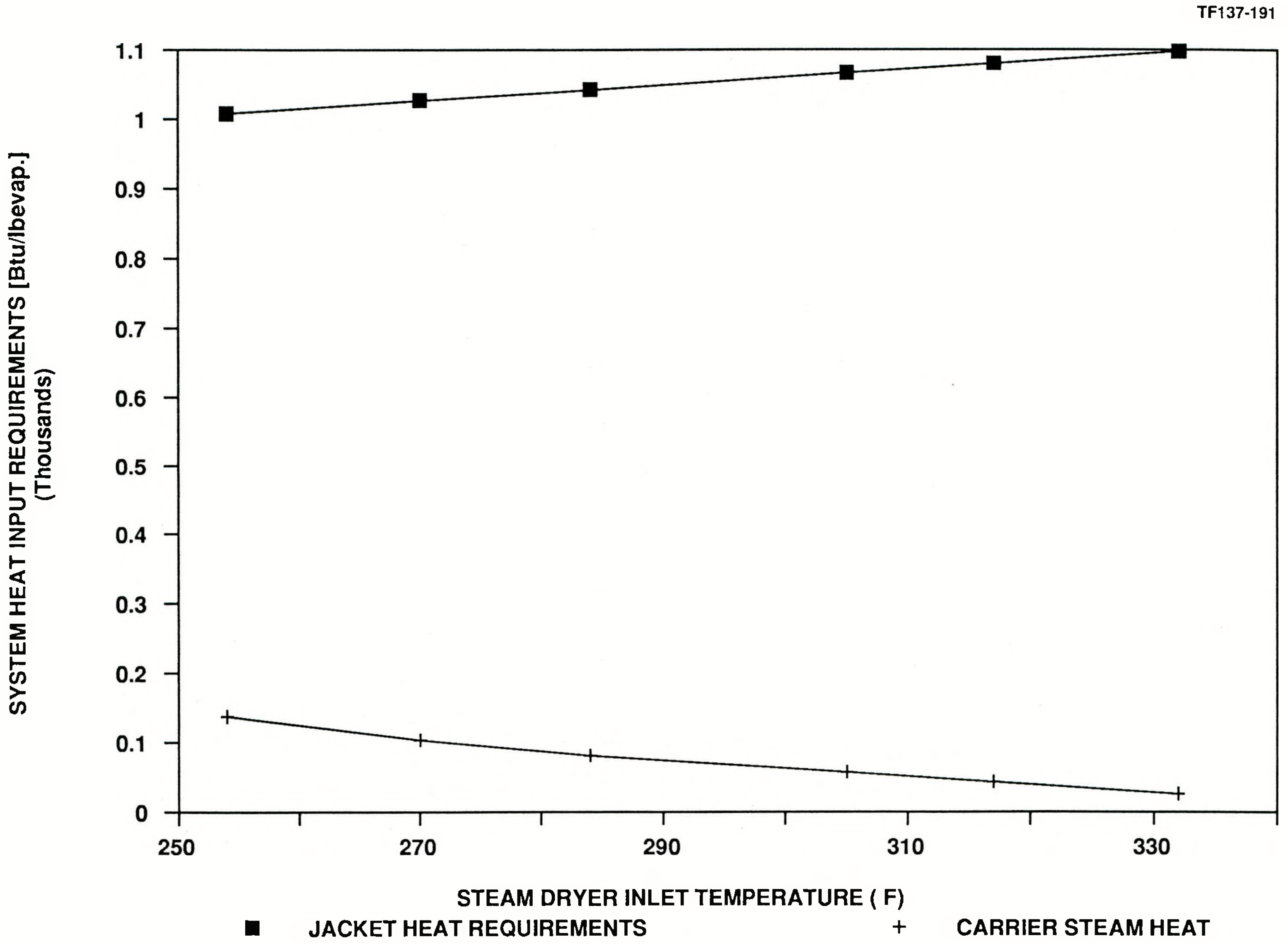

Figure 4.19 Indirect Steam Dryer Cycle System
Dryer Jacket and Carrier Steam Heat

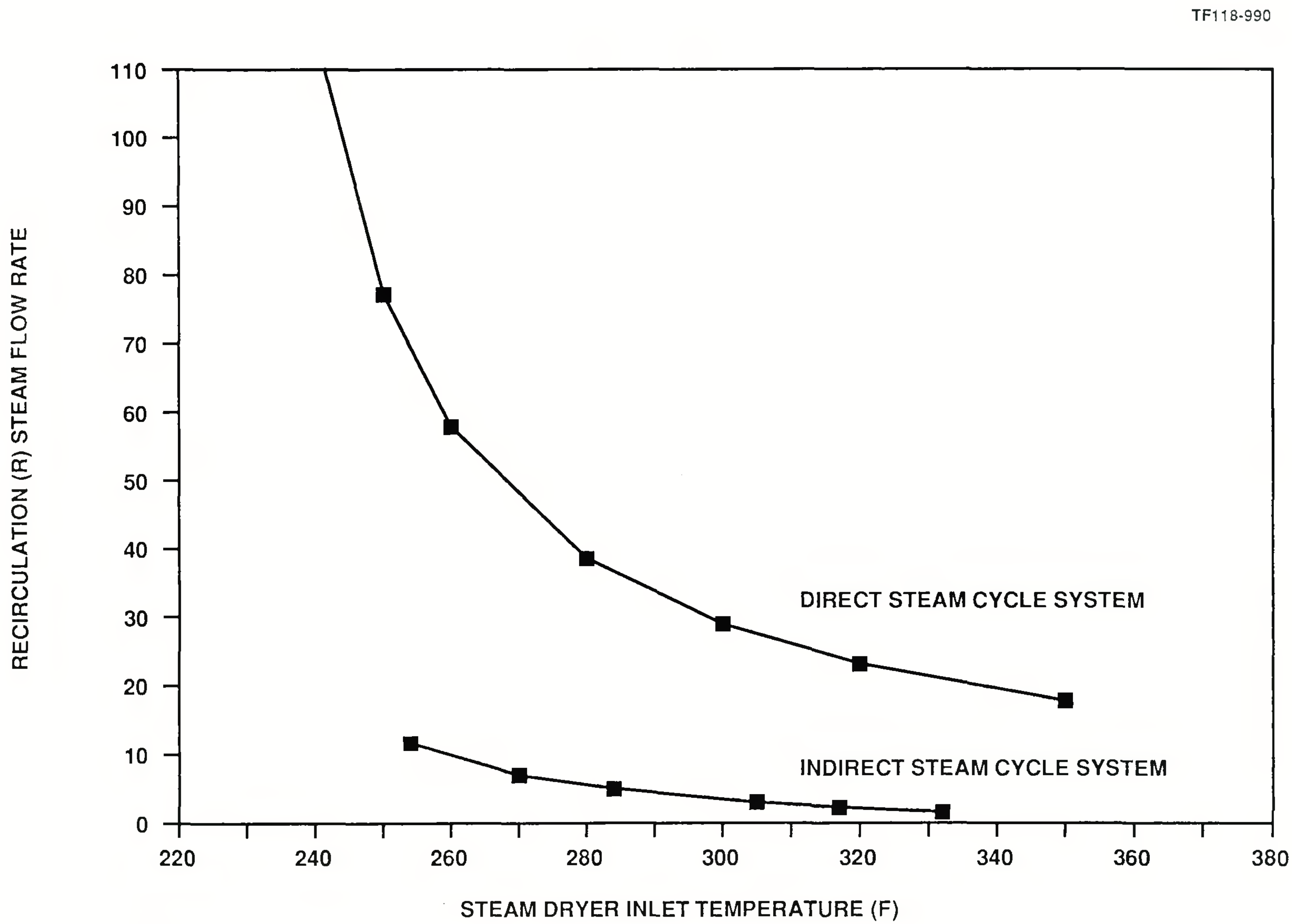

Figure 4.20 Comparison of Recirculation Ratio (R) for Indirect and Direct Steam Cycles

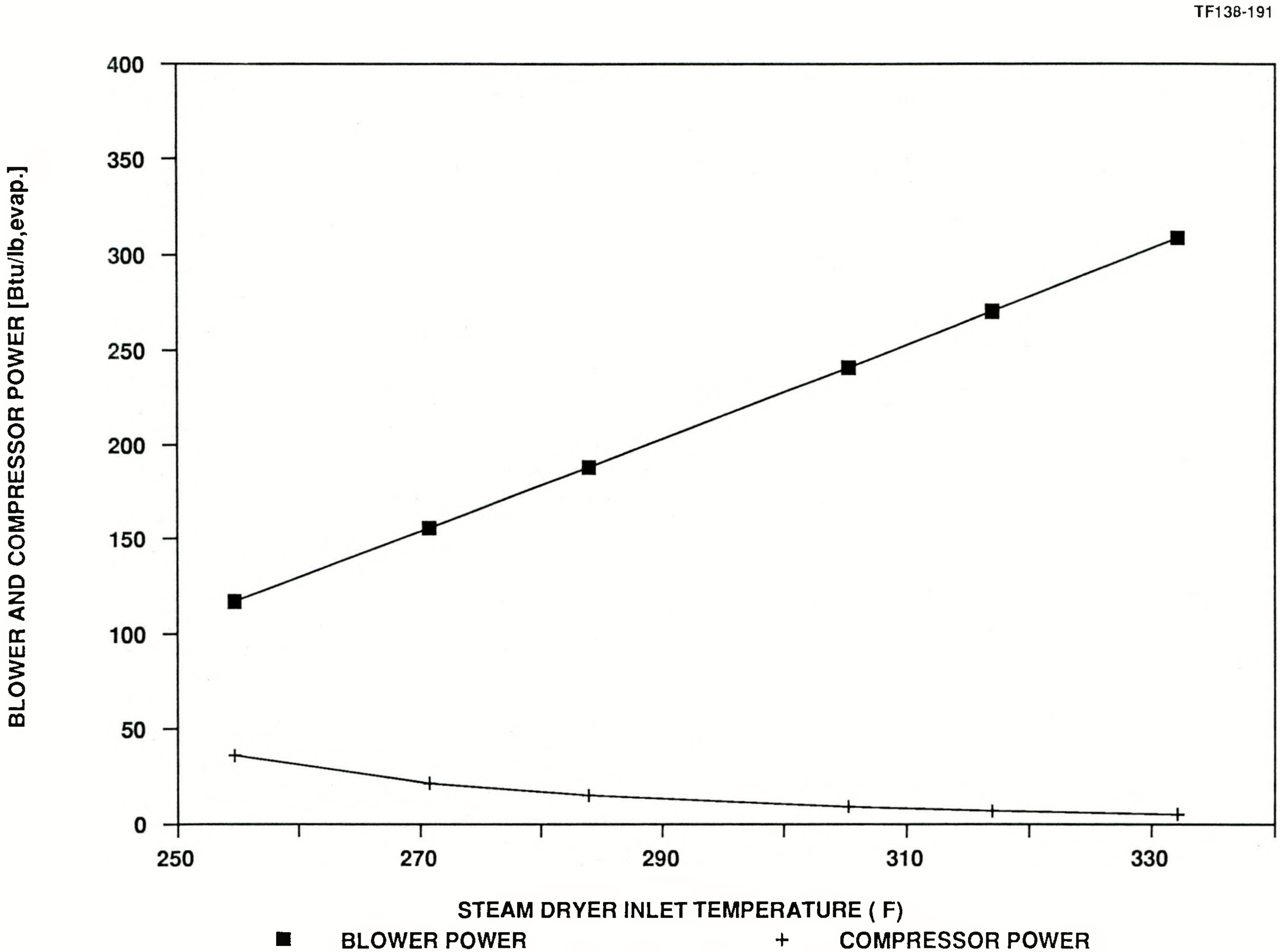

Figure 4.21 Indirect Steam Dryer Cycle
Blower and Comp. Power; w/DP = 10 in. and DT = 8°F)

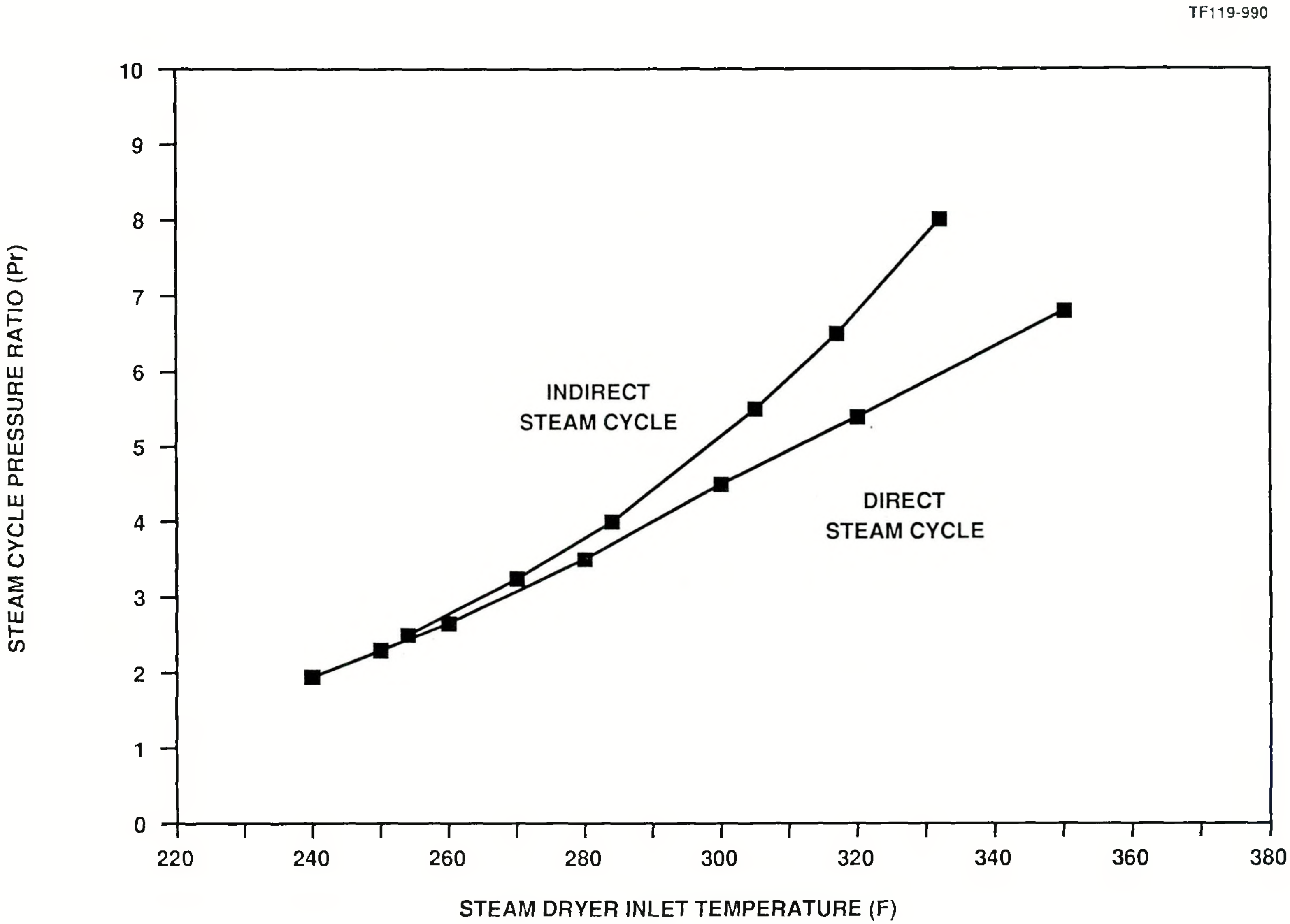

Figure 4.22 Comparison of Compressor Pressure Ratio (Pr) for Direct and Indirect Steam Dryer Cycle

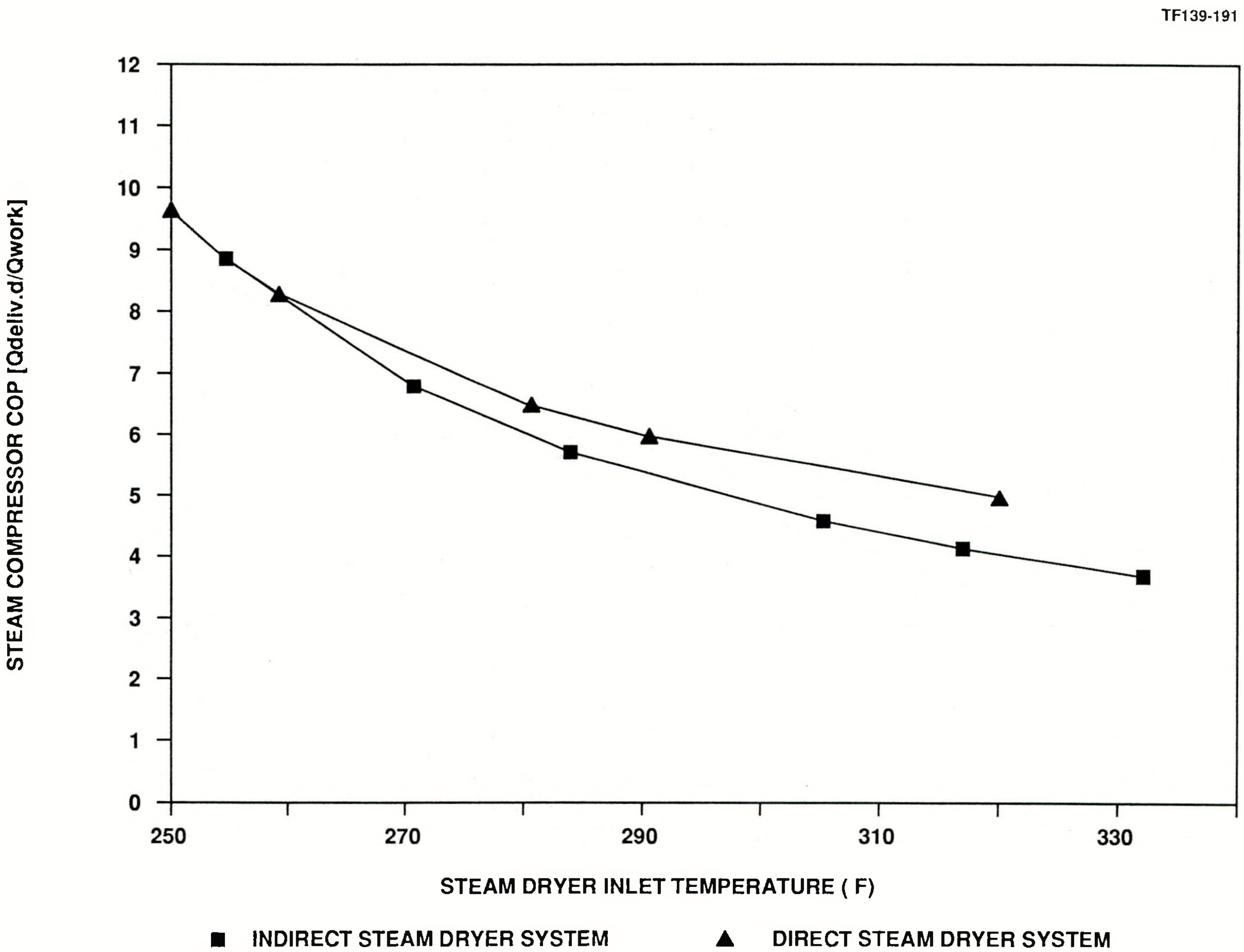

Figure 4.23 Indirect Steam Dryer Cycle Design Point
Steam Comp. COP (w/Fuel Conv. = 1)

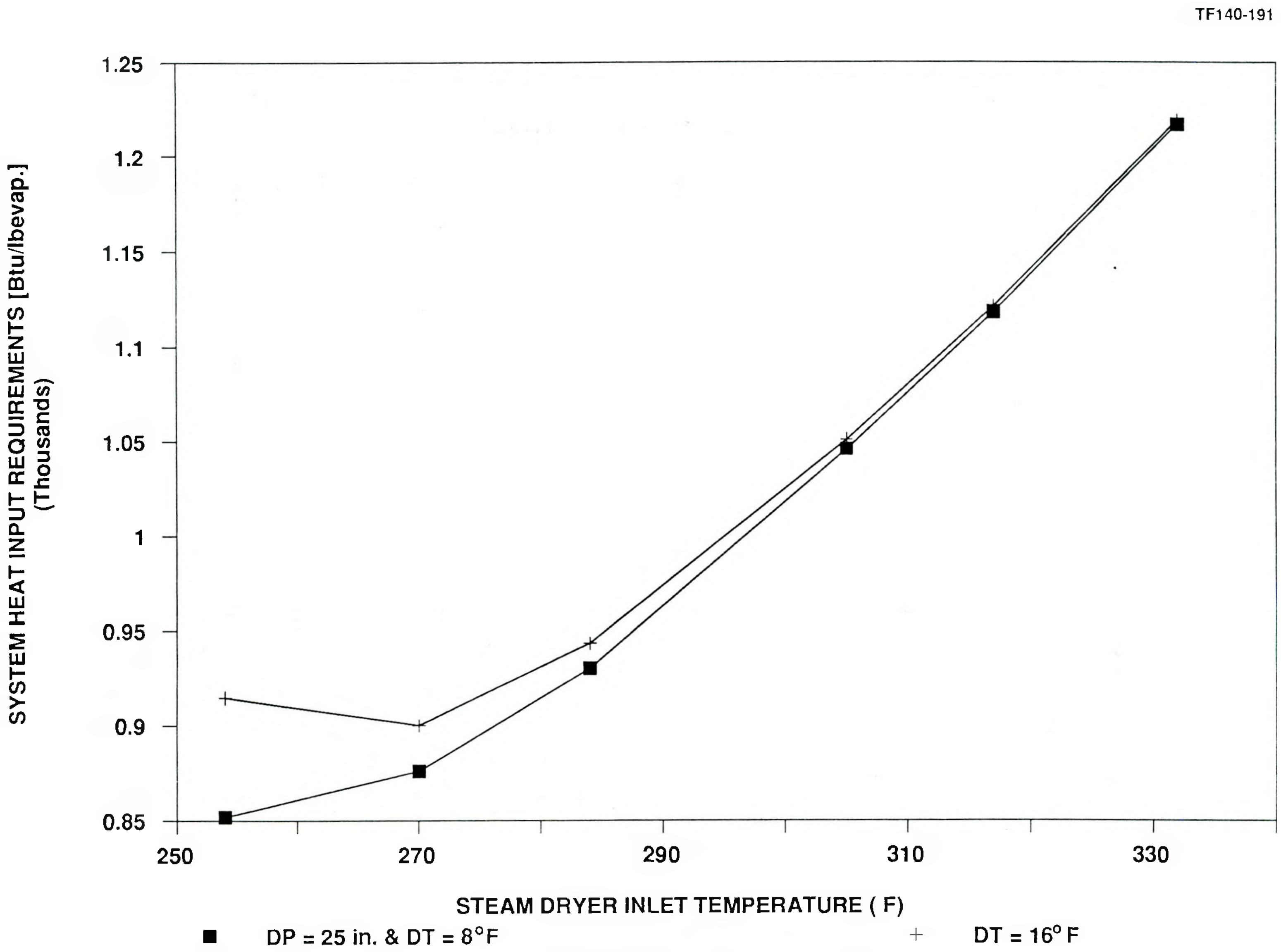

Figure 4.24 Indirect Steam Dryer Cycle System
with Respect to System Dryer Pinch Point

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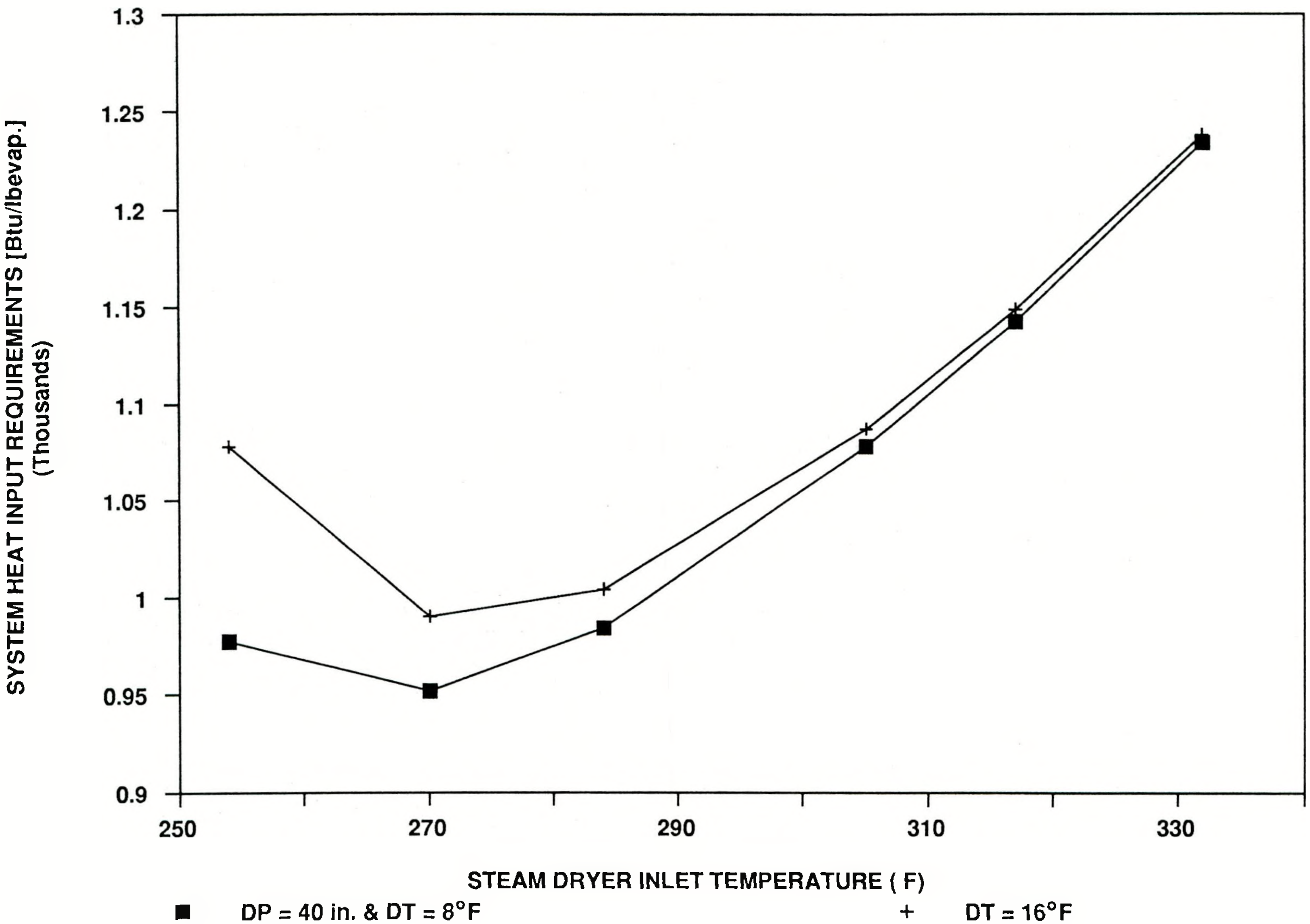

Figure 4.25 Indirect Steam Dryer Cycle System with Respect to System Dryer Pinch Point

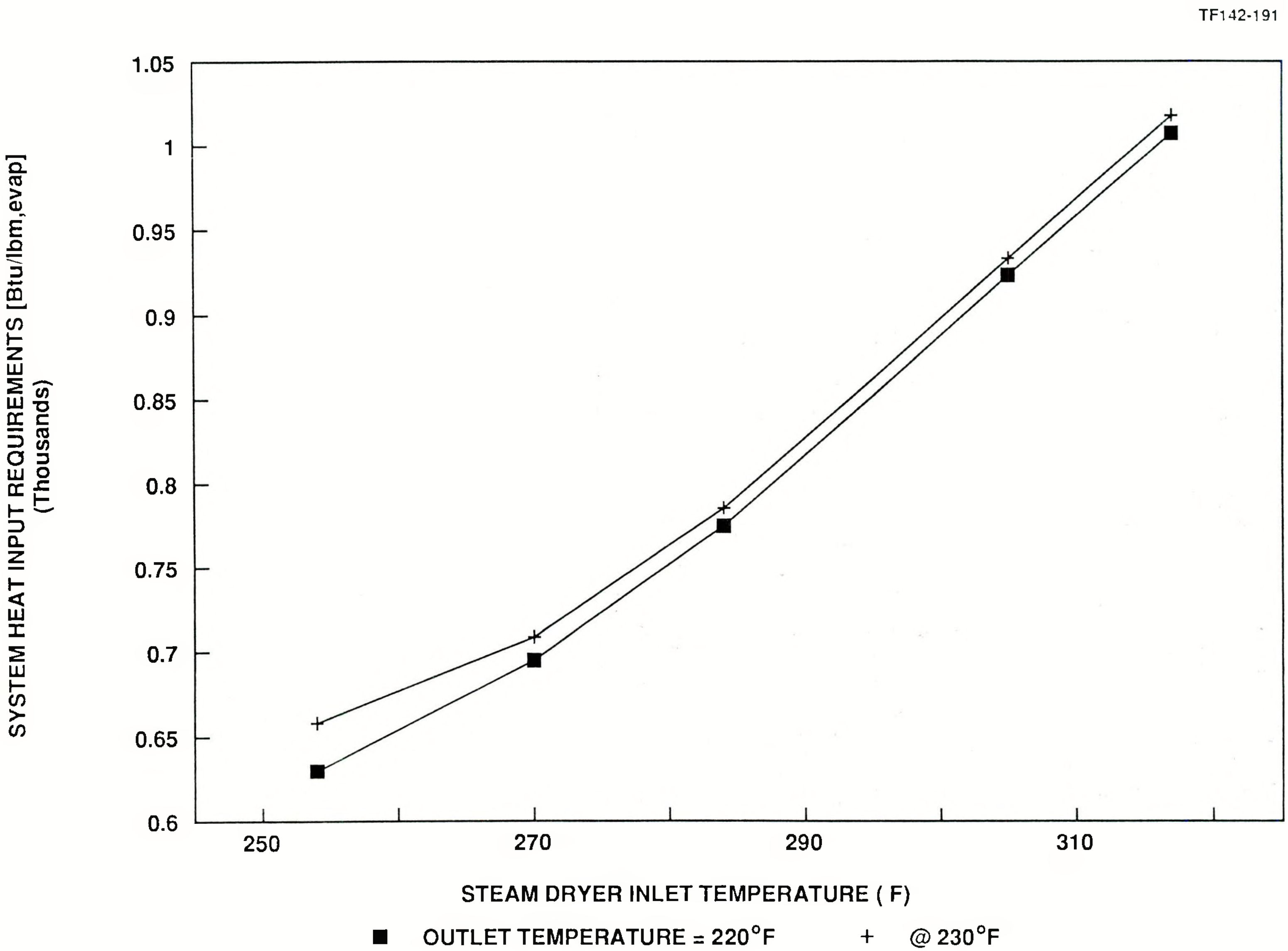

Figure 4.26 Indirect Steam Dryer Cycle System
Inlet Temperature Effect on Dryer Energy Requirements

However, the effect that this small change in discharge temperature has on the dryer's size is significant. This effect is displayed in Figure 4.27, in which a graph of the normalized product of the heat transfer coefficient and dryer volume (H_n x Vol) with respect to dryer inlet temperature is displayed. (See Chapter 6 for a discussion of H_n x Vol.) Figure 4.27 indicates that for a given steam dryer inlet temperature, the size of the steam dryer (as indicated by H_n x V) can be reduced by 26 percent if the dryer's discharge temperature is increased from 220°F to 230°F. These relationships between system energy requirements, dryer outlet temperature, and dryer size become critical in selecting a steam dryer operating design point that provides the minimum simple payback or a maximum in yearly energy and cost savings.

4.2 CONCLUSION

Two steam dryer system configurations were identified and studied in detail. Each of these systems provides a significant improvement in energy input requirements when compared with the standard air dryer system, i.e., 43 to 63 percent. The indirect steam atmosphere dryer (ISAD) requires lower energy inputs than the direct steam atmosphere dryer (DSAD) system. However, its requirement for conduction as well as convection heat transfer makes it more complicated than the simpler DSAD design. A study of the conduction heat transfer mechanism of the ISAD system is required. For example, the conduction heat transfer coefficients must be measured in a laboratory test for the IRIS dryer. Also, the use of extended surface within the dryer to promote the conduction heat transfer must also be considered. It is recommended that both of these tasks can be conducted in Phase II of the project.

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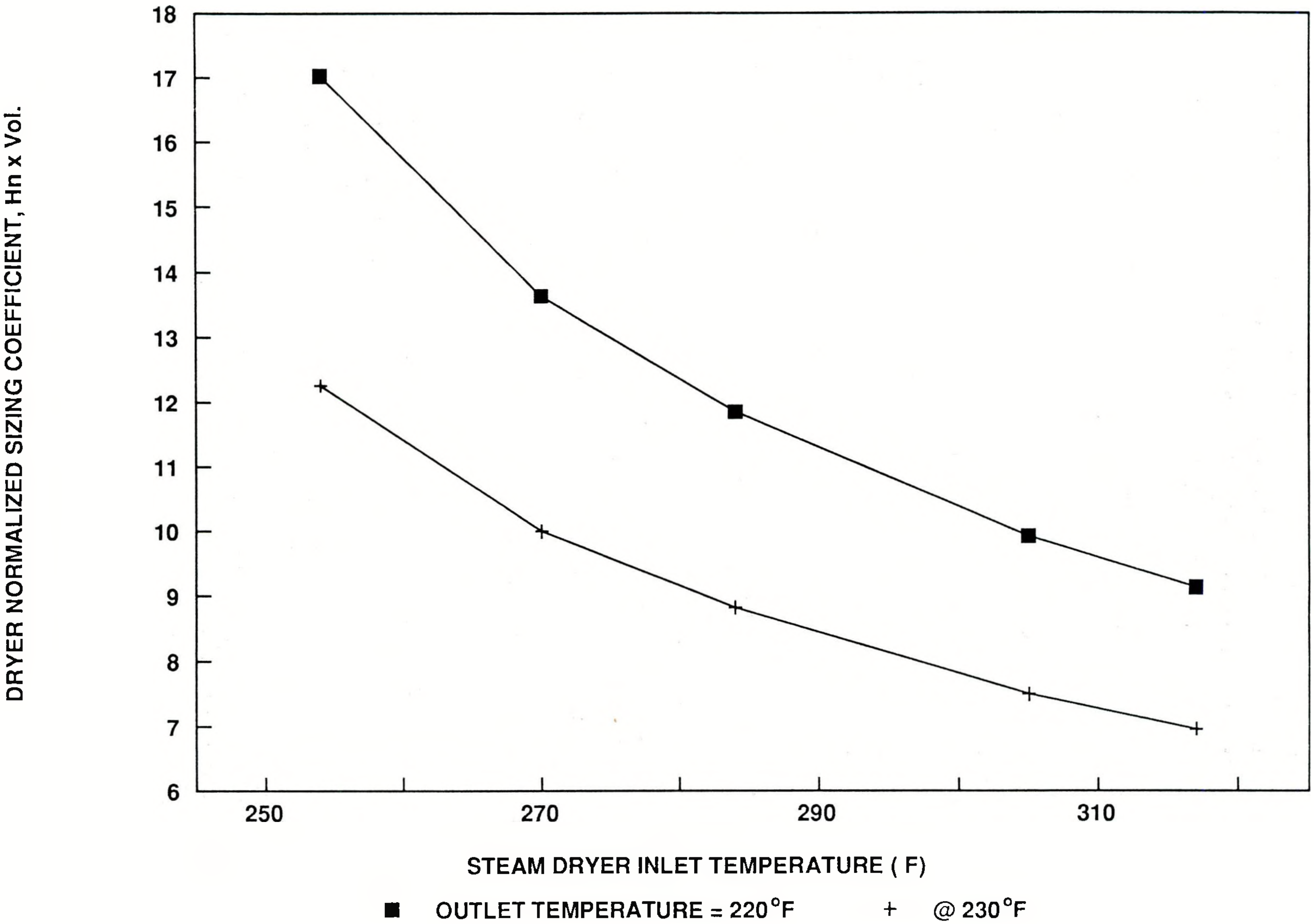

Figure 4.27 Indirect Steam Dryer Cycle System
Inlet Temperature Effect on Dryer Sizing Coefficient

5. TASK 3 – DESIGN SYSTEM ELEMENTS

The steam atmosphere dryer system with recompression consists of two principal components: the steam compressor system and the new steam atmosphere dryer, or Tecogen's Inertial Recirculation with Internal (and External) Separation (IRIS) dryer. The preparation of the design specifications for these components was the objective of this Task 3 of the feasibility analysis.

5.1 STEAM ATMOSPHERE DRYER DESIGN POINT SELECTION

Based on the thermodynamic analysis conducted in Task 2 (as revised by the economic analysis conducted in Task 5) a design point operating condition for the direct steam atmosphere dryer system and the indirect steam atmosphere dryer system is identified. The state point conditions for these two systems are shown in Figures 5.1 an 5.2, respectively. These state point conditions were used to determine the size and cost of the steam compressor and the IRIS steam dryer systems. It must be noted that the indirect steam atmosphere dryer system is the preferred steam dryer configuration due to its lower energy requirements. However, a two-stage direct steam atmosphere dryer system also provides a comparable energy efficiency. Consequently, its design point operating conditions are also given in Figure 5.3 for reference purposes. The operation of the steam compressor and IRIS steam atmosphere dryer are similar for each of these steam dryer configurations and thus their design specifications are similar in nature. However, the work conducted in Task 3 concentrated on the steam compressor and jacketed IRIS steam dryer as used in the indirect steam atmosphere dryer.

5.2 STEAM COMPRESSOR SPECIFICATIONS

The function of the steam compressor is to increase the pressure of the steam so that the higher steam temperature may be used to transfer the latent steam's heat to the dryer. The steam compressor provides essentially adiabatic steam compression and requires some means of desuperheating the discharge steam's temperature. The desuperheating is usually performed by water injection at either the suction or discharge of the compressor. Thus, the mechanical work of compression is transformed into thermal energy represented in the form of pressurized and saturated steam.

In the selection of the compressor, there are three important factors: (1) mechanical reliability, (2) efficiency, and (3) cost. The compressor must be capable of compressing steam and be reliable while doing so. Although a number of compressors are available for steam recompression, there are some unique problems associated with each type. For example, centrifugal compressors are

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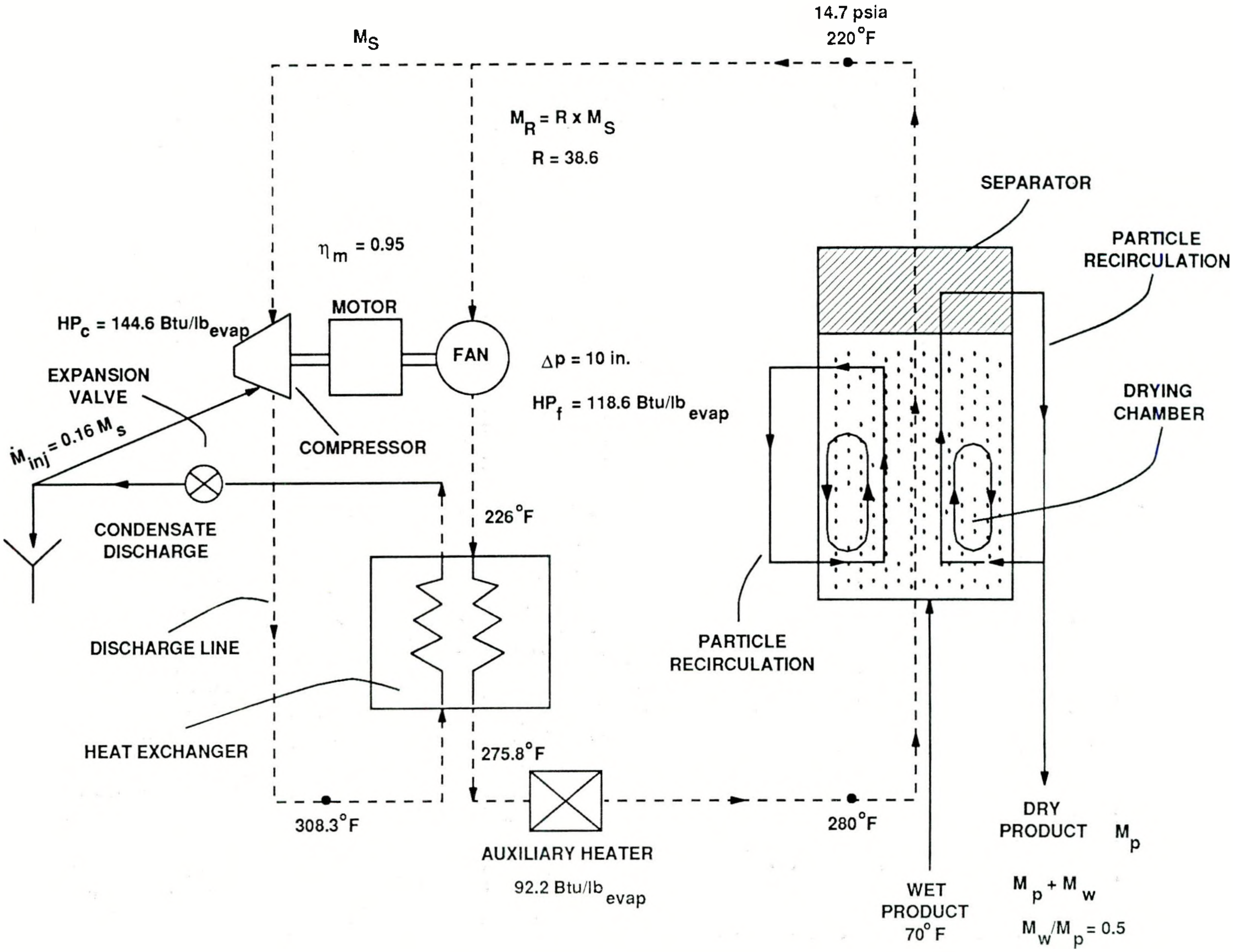

Figure 5.1 Direct Steam Atmosphere Dryer State Point Conditions

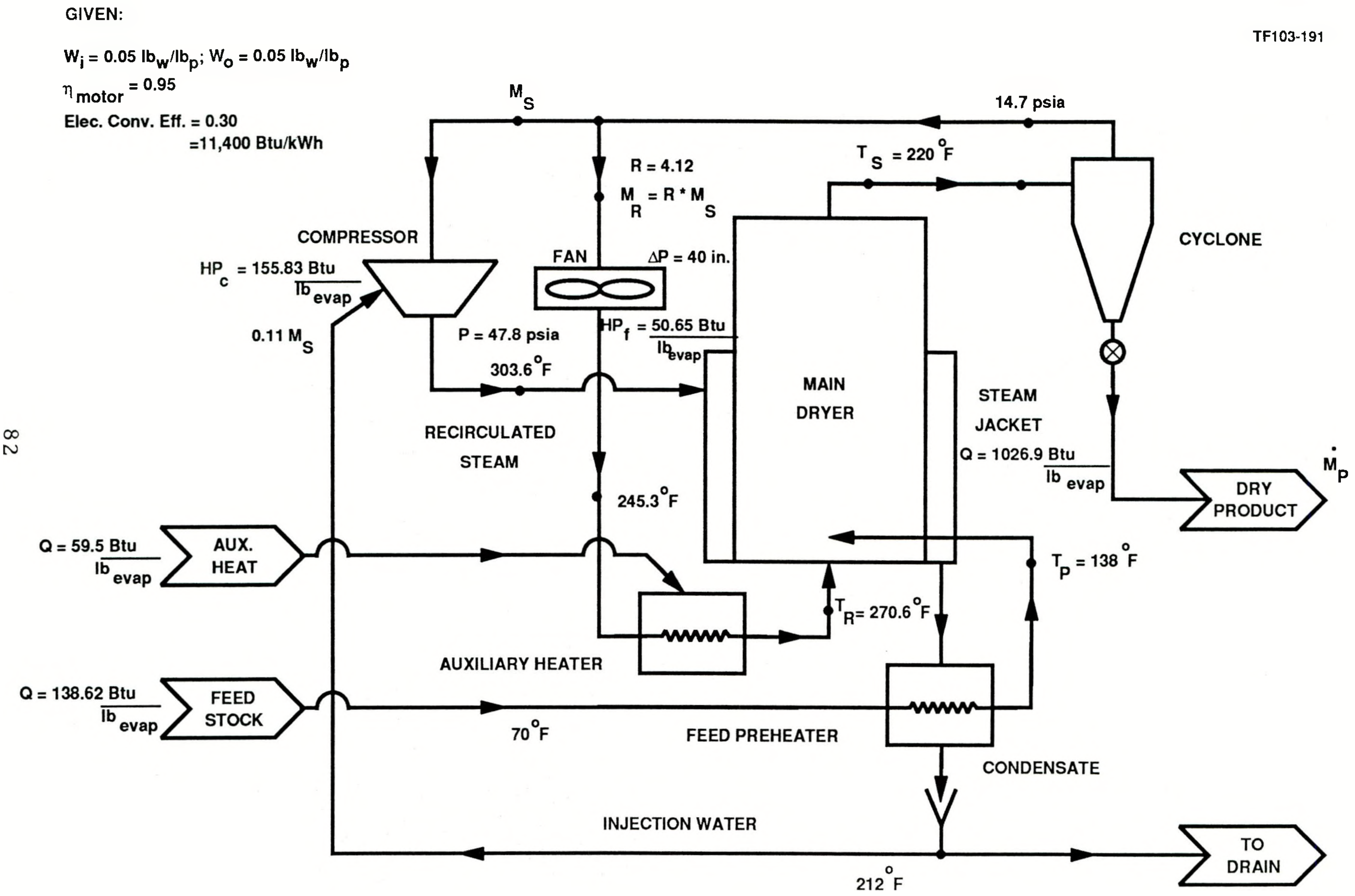

Figure 5.2 Indirectly Heated Steam Dryer State Point Conditions

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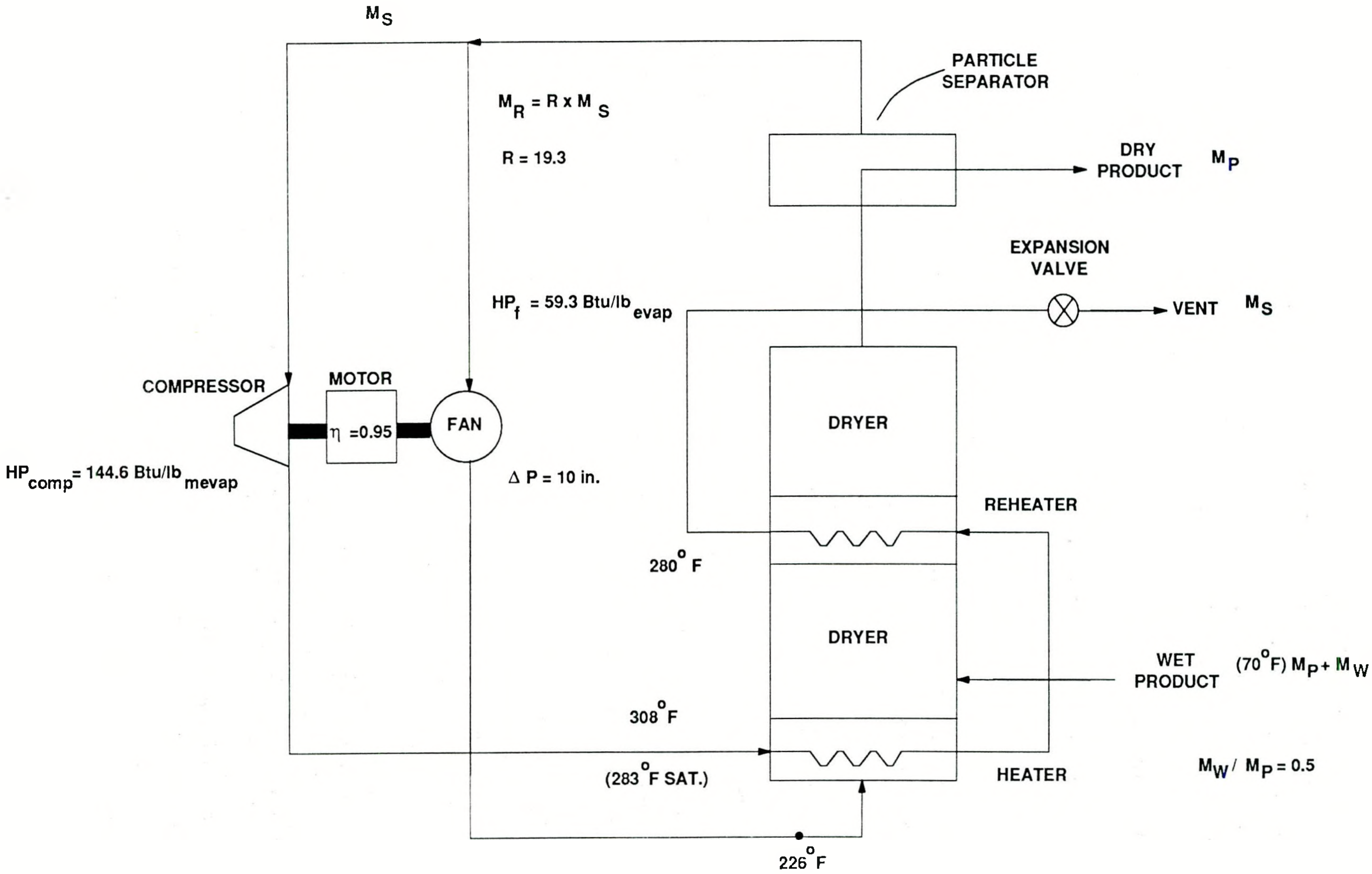

Figure 5.3 Direct Steam Atmosphere Dryer with Two Reheater Stages

susceptible to erosion from entrained water droplets. Because the compressor will be operating continuously, the consequence of a failure can be very costly from the standpoint of lost product, thus high reliability is very important. The second factor, efficiency, relates to the power requirements of the compressor. The third factor, cost, must be examined in terms of the compressor's mechanical reliability and efficiency and the specific application for which the machine is most suited. Some compressors have high reliability and efficiency and low cost, but are only available for special applications. A discussion of several steam compressor types with their advantages and disadvantages noted is given in the following section.

Compressor Characteristics

Compressors can generally be classified into two types, aerodynamic and positive displacement, as illustrated in Figure 5.4. Among the various types of compressors, centrifugal, Roots, and screw have been used for steam recompression.

Centrifugal compressors are often selected for steam recompression. Figure 5.5 is a sectional view of a typical single-stage centrifugal compressor. These machines achieve compression based on aerodynamic operating principles. To achieve the desired pressure rise, the machines run at relatively high impeller tip velocities, up to 1200 ft/sec. The machine is very sensitive to water droplets that can cause blade erosion, rotor imbalance, and potential failure. To avoid this problem, about 10 percent of the outlet superheated vapor is recirculated back to the inlet to dry the incoming steam.

Pressure ratios for centrifugal compressors are generally limited to approximately 1.5 to 2. The machines are essentially constant pressure devices, and the power consumption is almost directly proportional to the volume delivered.

Centrifugal compressors are manufactured in capacity ranges from 3,000 to 180,000 cfm. Capacity control is most efficiently achieved by speed variation or the use of inlet guide vanes. Figure 5.6 illustrates the operating characteristics typical of these control techniques. With speed control, there is a minimum speed below which operation becomes unstable and a phenomenon known as surging occurs. The surge limit is set by the impeller discharge angle and is generally around 50 percent of the flow capacity at the maximum efficiency point. In addition to speed control, the use of inlet vanes is common practice. The guide vanes act to provide prerotation ahead of the impeller to reduce entrance losses and as a throttle to reduce the flow rate by reducing the vapor density.

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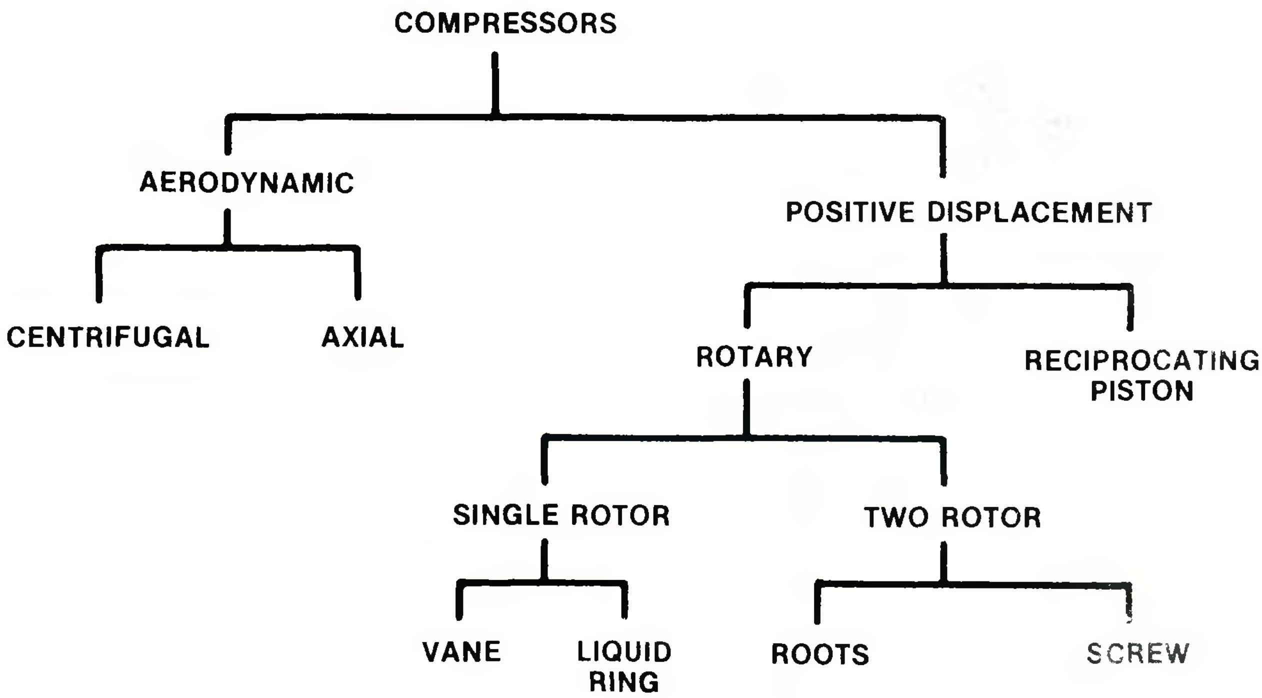


Figure 5.4 Classification of Mechanical Compressor Types

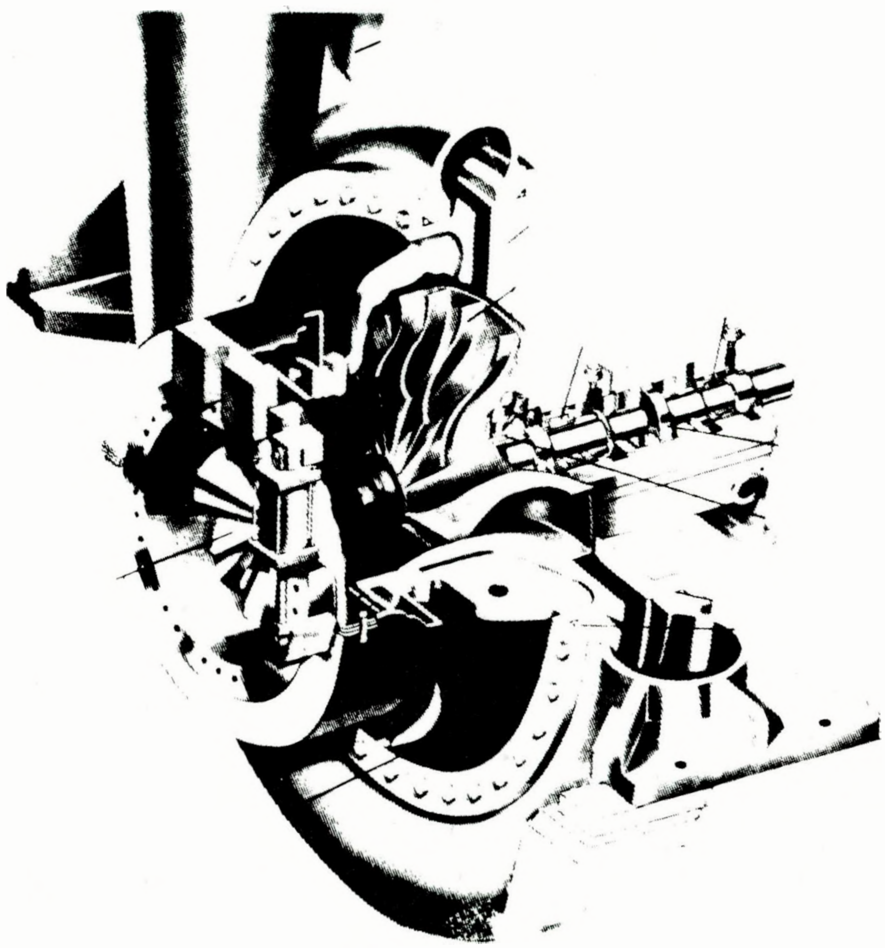
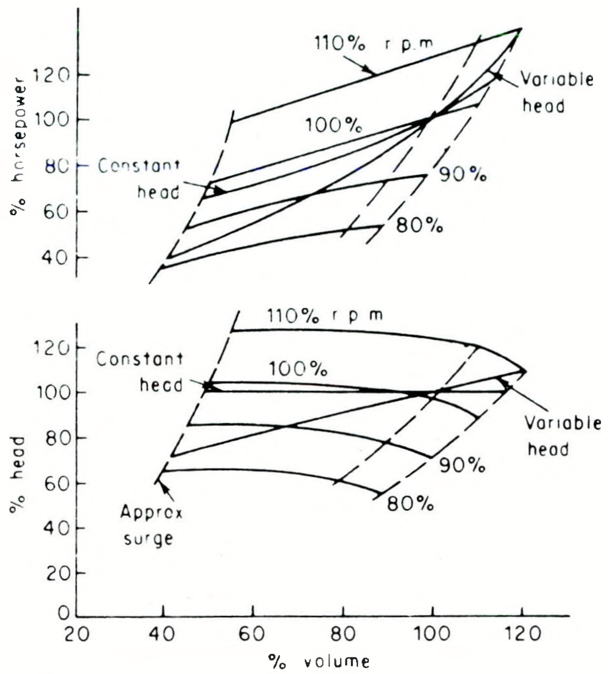
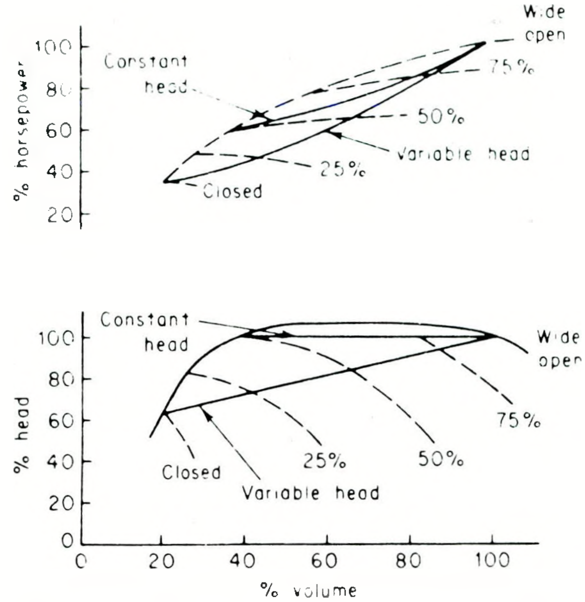


Figure 5.5 Single-Stage Centrifugal Compressor



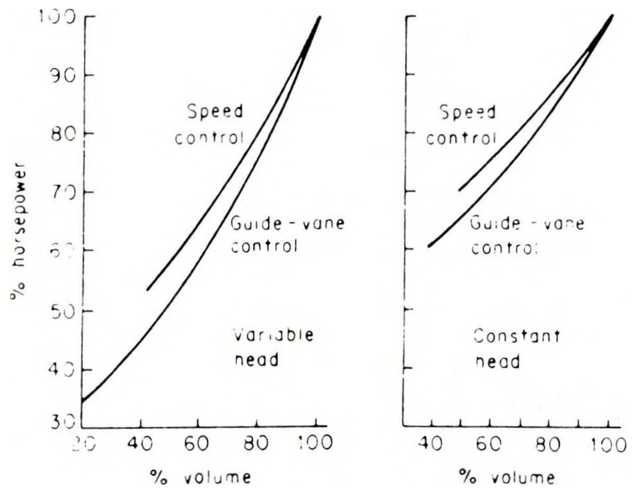
Operating characteristics of centrifugal compressor, illustrating turbine speed control.

(a)



Operating characteristics of guide-vane control for centrifugal compressors

(b)



Horsepower comparison between speed and guide-vane control of centrifugal compressors.

(c)

Figure 5.6 Centrifugal Compressor Performance Curves

Roots blowers are positive displacement machines. Shown in Figure 5.7 is a two-impeller blower. These machines are constant volume devices. The discharge pressure is determined by the resistance of the system. However, Roots blowers are generally used at low pressures and low pressure rises (typically 20 to 25 psi) due to excessive leakage, "slip," and rotor deflection. Roots blowers can, however, handle wet inlet steam. Capacity ranges vary from 200 to 25,000 cfm. Flow delivery is varied by changing the speed or bypassing some of the machine capacity. Figure 5.8 is a typical performance curve for a Roots blower. Bypassing for the purpose of effecting volume control offers no energy savings. The major limiting factor for Roots blowers is the limited pressure differential. During dryer startup, shutdown, or during an upset, where pressure differentials could be very large, serious compressor damage could result. For the steam atmosphere dryer system, the Roots blowers are not suitable due to the high pressure rises (pressure ratio = 3.25:1) that are required.

The helical rotary screw compressor is a positive displacement machine consisting of two mating helically grooved rotors, one male and the other female, operating in a stationary housing with suitable inlet and outlet ports. Figure 5.9 shows a cutaway of a typical screw compressor. No inlet or discharge valves are required. In operation, inlet gas is pulled into a void created by a pair of spiral rotors, as shown in Figure 5.10. As rotation occurs, the incoming gas is cut off from the inlet and compressed by the meshing rotors as it is moved along the axis of the machine. At some point, depending on the compression ratio built into the machine, discharge ports are uncovered and the compressed gas is discharged. As with Roots blowers, the discharge pressure is a function of the resistance on the discharge side.

The machine is designed with a built-in pressure ratio, and operation above or below this value does reduce performance. Pressure ratios up to 7 to 1 can be achieved in a single stage with pressure differences up to 200 psi. Screw compressors are generally not utilized at pressure ratios less than 1.5 to 1.

Screw compressors are built as oil-flooded, water-flooded, and dry machines. They are highly suited for wet steam compression. At pressure ratios greater than 2 to 1, water injection is usually required to minimize the steam temperature rise. The machine is built in a wide range of sizes, from 400 to 23,000 cfm, and characterized by good efficiency over a 2-to-1 turndown range. Flow and power are basically proportional to speed. Speed control is the most efficient method of capacity control.

The work of compression for reciprocating and rotary compressors can be determined with reasonable accuracy assuming an ideal gas by the following relation:

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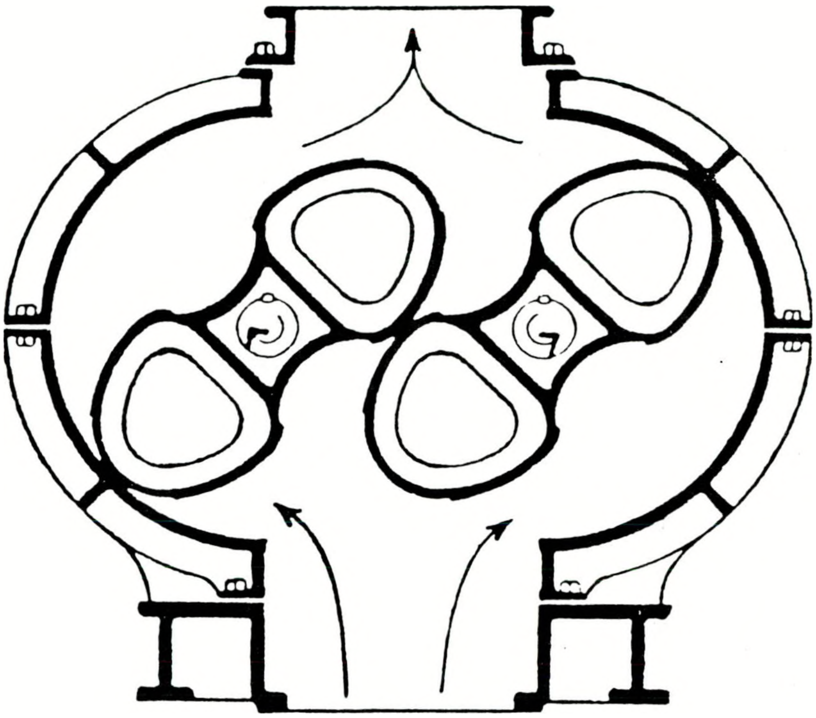

Figure 5.7 Two-Impeller Type of Positive Rotary Blower

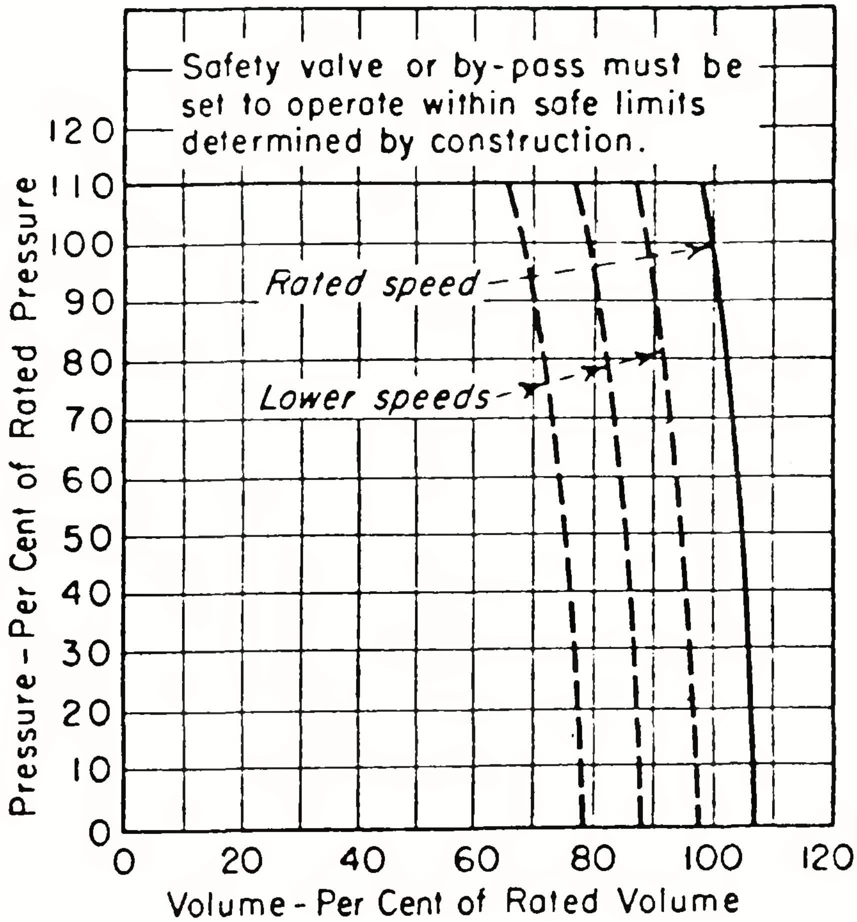

Figure 5.8 Approximate Type of Performance Curve for Rotary Compressors

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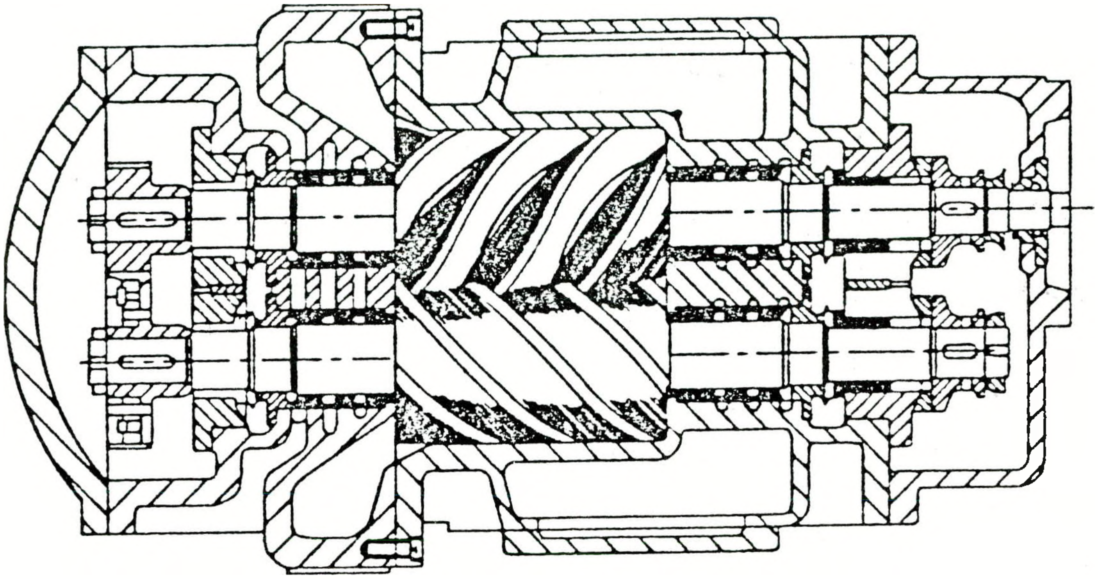

Figure 5.9 Screw-Type Rotary Compressor

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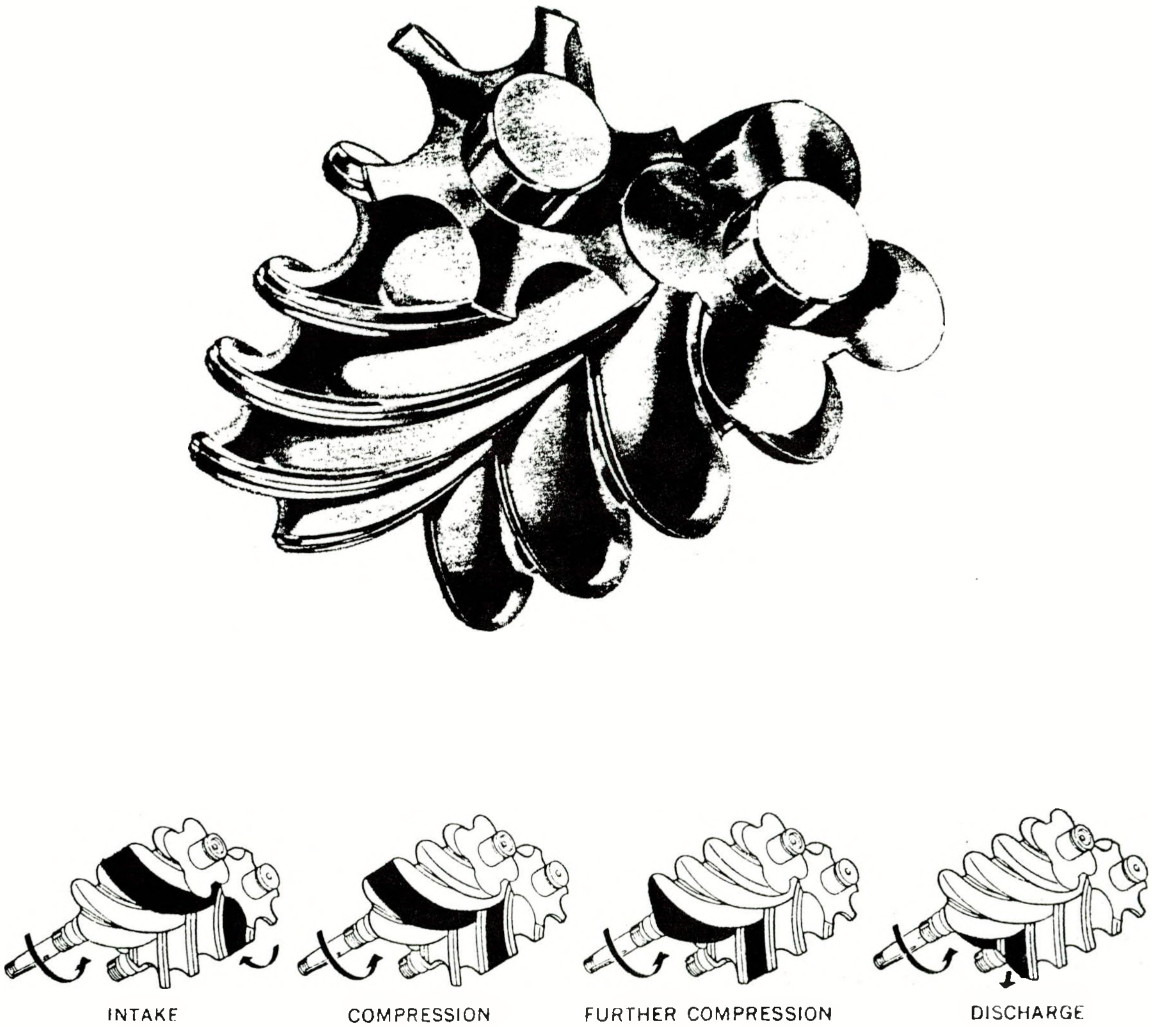

Figure 5.10 Screw Compressor Operation

$$\text{Power (hp)} = [m \times Cp \times (460 + T_1) \times [Pr^{(k-1)/k} - 1]]/$$
$$[n_m \times n_s \times 2545];\ Pr = P_2/P_1$$

where:

- Cp = specific heat of fluid (approximately 0.5 Btu/lb-F)
- k = specific heat ratio (cp/Cv) of fluid ($k \cong 1.32$)
- m = fluid flow rate (lbm/hr)
- P_1 = compressor inlet pressure (psia)
- P_2 = compressor outlet pressure (psia)
- T_1 = compressor inlet temperature (F)
- W = compressor power requirement
- n_m = compressor mechanical efficiency
- n_s = compressor isentropic efficiency

The power requirement is a strong function of the pressure ratio, and thus it is important to keep it as low as possible to minimize the input power. Also, in selecting the compressor, a high mechanical efficiency (low lubrication or friction loss) and high isentropic efficiency (low deviation from the ideal thermodynamic process) are desired.

Of the various compressors discussed, the centrifugal compressor has been used by far in the greatest number of installations for steam recompression. However, these applications are low pressure and low pressure ratios – generally around 1 atmosphere and less than 2, respectively – and would require much higher volume flow rates than would be typically found in the steam atmosphere dryer application. For these reasons, a centrifugal compressor is not the best choice for the steam dryer system.

In the past, Roots blowers have been offered for use as steam recompressors. They come in size ranges ideally suited to dryer applications and can handle wet steam directly. As with centrifugal compressors, they are generally used at low pressures, around 1 atmosphere, but at higher pressure ratios. These machines would be a good choice for the steam dryer application. However, the Roots blowers cannot operate at pressure differentials greater than 15 to 18 psi. Also, there are few Roots compressor manufacturers available for the size range required for the steam dryer system.

Tables 5.1 and 5.2 summarize the design and operating characteristics of the mechanical compressors suitable for steam compression service.

TABLE 5.1

SUMMARY OF MECHANICAL COMPRESSOR DESIGN AND OPERATING CHARACTERISTICS

Type	Pressure Ratio (Rise) Per Stage	Absolute Pressure Range	Capacity (cfm)	Speed	Problems
Centrifugal	1.5:1	Any	3,000 to 180,000	High	Minimum capacity is much larger than required for a single crepe wadding machine
Reciprocating	4:1 to 8:1	Any	15 to 15,000	Low	Oil carryover
Roots Blower	(18 psi)	Generally used at low pressures	200 to 25,000	Medium	Nonstandard, special casing required
Helical Screw	7:1	Any	400 to 23,000	Medium	Overbuilt for some applications

TABLE 5.2

SUMMARY OF MECHANICAL COMPRESSORS

Type	Method of Control	Reliability	Efficiency* (bhp)
Centrifugal	Varying speed or inlet guide vanes	Good; must recycle about 10% of the outlet superheated vapor to dry the incoming steam	Compressor not available at these pressures and flow rates
Reciprocating	Unloading or varying speed	Fair; must superheat the incoming vapor and/or separate out the liquid droplets; its reciprocating nature entails more maintenance	27.3
Roots Blower	Varying speed or bypassing machine capacity	Fair; can handle wet steam; damaged by large pressure differentials	26.2
Helical Screw	Varying speed or bypassing machine capacity	Best; highly suited for wet steam recompression	37.0

*Efficiency expressed as bhp required to compress 4000 pph of 150 psia steam to 165 psia.

In terms of reliability and performance, the screw compressor is considered the best suited for vapor recompression in general, and for the steam dryer application in particular. It can handle wet steam directly and operates at high pressure and pressure differences.

A list of the available manufacturers of steam screw compressors is shown in Table 5.3. After contacting these manufacturers Tecogen identified the following steam compressor design features:

- Helical rotary screw
- No oil injection (i.e., dry unit)
- Carbon steel rotor and housing; rotor coatings preferred
- Timing gear required
- Sleeve type journal bearings and tilted pad thrust bearing (roller bearings optional)
- Carbon ring shaft seal with air buffer
- Integral, one-step gearbox, flex-coupling, and electric motor drive
- Integral lubrication system
- Rigid support frame
- Instrumented safety system

The compressor's complete performance map was generated for 5000 lb/hr (2300 acfm) and 20,000 lb/hr (9200 acfm) by means of Tecogen's helical rotary screw compressor modeling computer program. This provided an estimate of the compressor's required speed and available adiabatic compressor efficiency. Plots of two compressor maps for a 10 1/2-in. and a 20-in. diameter rotor are given in Figures 5.11 and 5.12, respectively. From Figures 5.11 and 5.12 it may be observed that the screw compressor efficiencies are above 70 percent, which agrees with the previous assumptions used in the modeling.

An electric motor drive was chosen for the prime mover for the steam compressor. This choice was based on both equipment cost and fuel efficiency considerations. While it is possible to utilize an internal combustion (IC) or gas turbine engine as the compressor's prime mover, the cost per kW for these systems is typically 2.5 to 3.5 times higher. This would result in an increased system cost of 10 to 15 percent. The fuel efficiency of the steam dryer system, however, could be increased if all of the IC or gas turbine engine's exhaust gas heat recovery could be effectively utilized; for auxiliary dryer steam heating, for example.

For the steam atmosphere dryer system with recompression, the requirements for auxiliary steam heating are low. Consequently, the fuel efficiency for the gas turbine and IC engines would typically be less than 28 to 31 percent and

TABLE 5.3

LIST OF STEAM COMPRESSOR MANUFACTURERS CONTACTED DURING PROJECT

A-C Compressor Corporation (Bridgewater, New Jersey)

Atlas Copco Compressor Corp. (Vorhesville, New York)

Mycom Compressor (Torrance, California)

Aerzen (Exton, Pennsylvania)

Kobe Compressor (Japan)

STI Sulzer Turbosystems (Houston, Texas)

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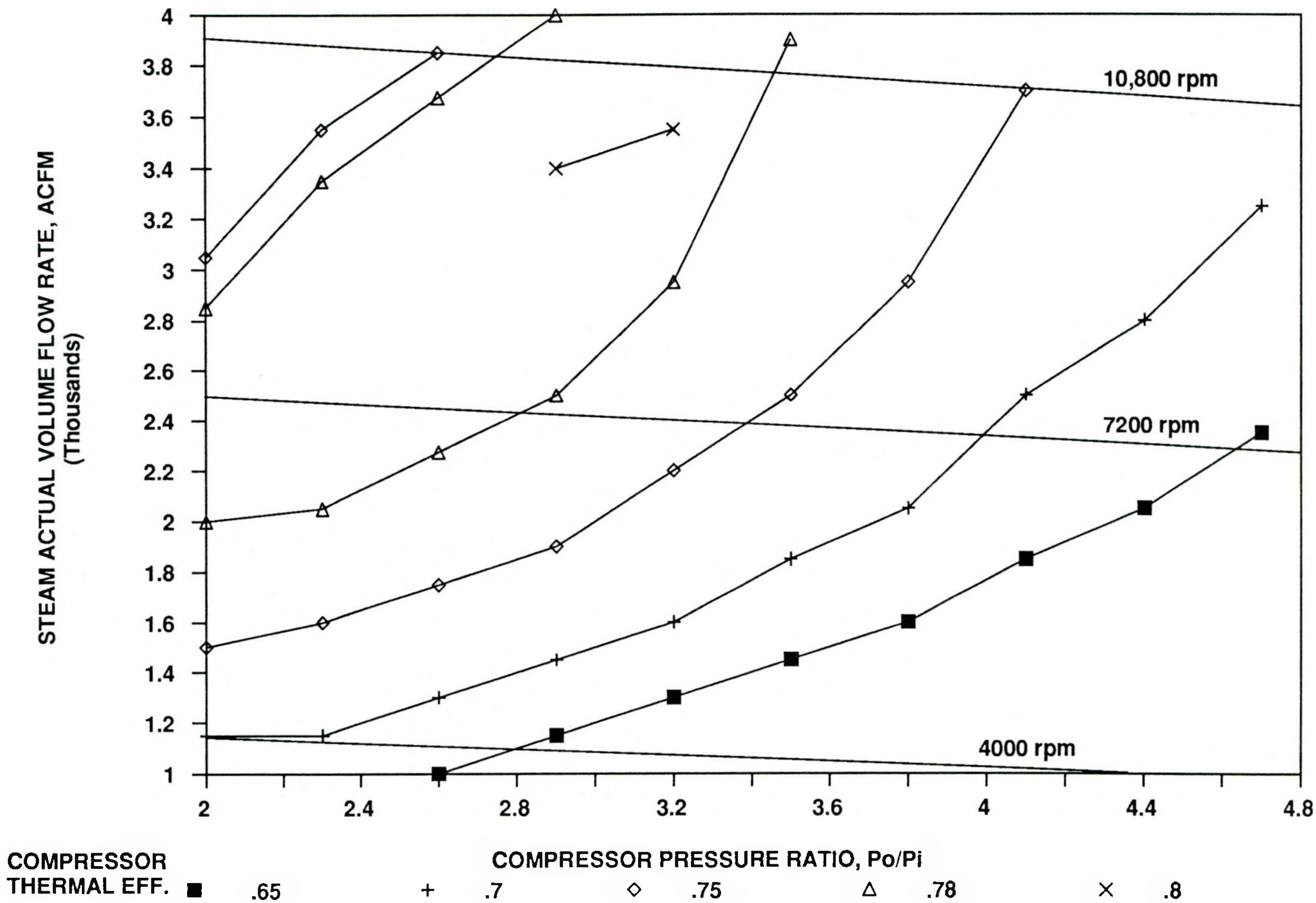

Figure 5.11 10-in. Diameter Steam Screw Compressor Performance Map
Thermal Efficiency and Speed Curves

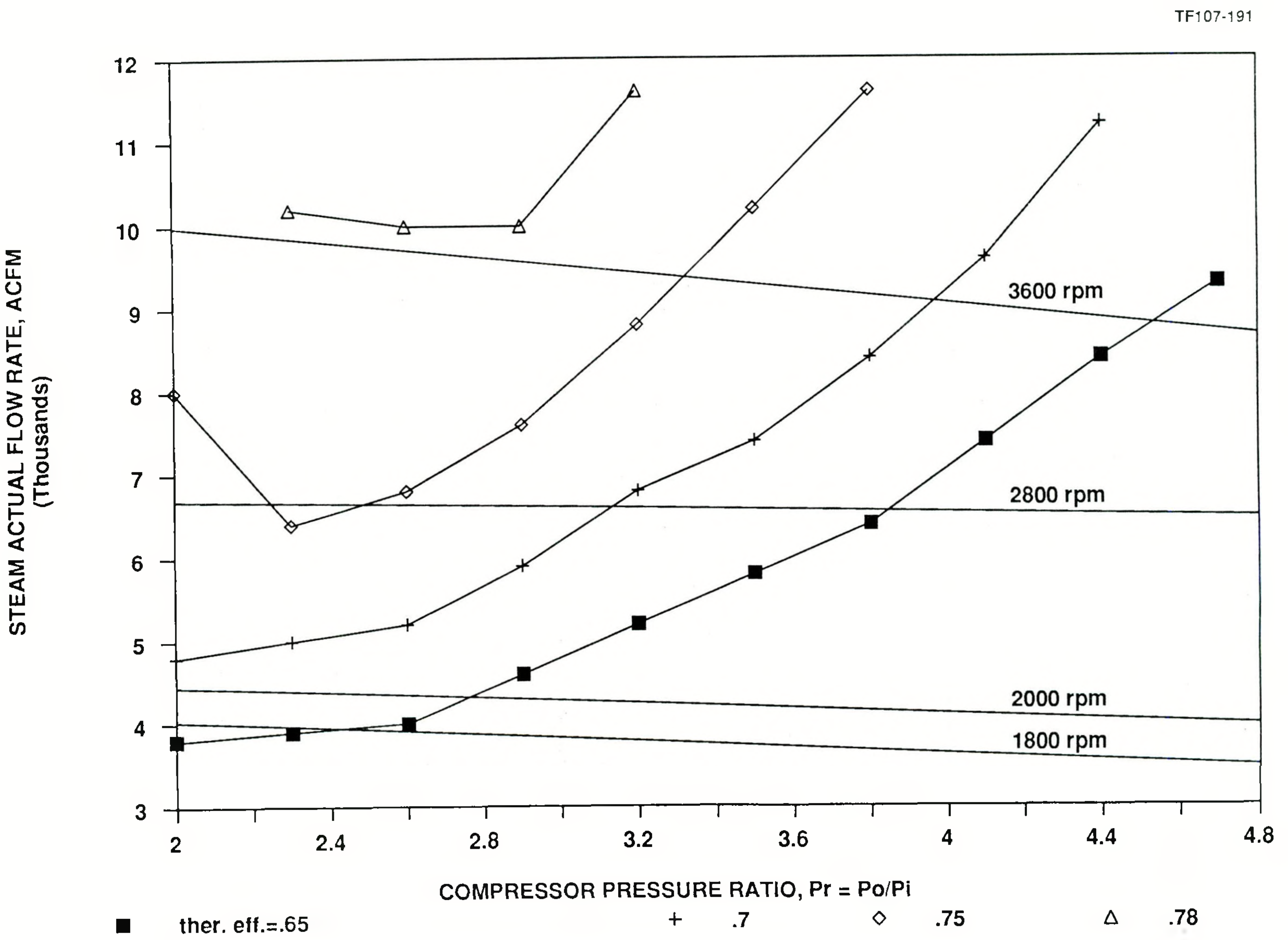

Figure 5.12 20-in. Diameter Steam Screw Compressor Performance Map
Thermal Efficiency and Speed Curves

thus only comparable to the 30 percent assumed in this analysis for the conversion of primary fuel to electric power.

Thus, the electric motor drive is the logical choice for this application.

5.3 IRIS STEAM DRYER

The basic IRIS steam dryer design is shown schematically in Figure 5.12. It consists of a cylindrical vessel with an external recirculation line. The top section of the dryer is also fitted with a concentric cylinder sized to be approximately 60 percent of the vessel's diameter. For reference purposes, the IRIS bench-scale model has been reproduced in Figure 5.13 and represents the typical IRIS steam dryer configuration. The relative proportions of the vessel length to diameter, steam inlet flow area to vessel cross-sectional area, and the steam exhaust outlet or "vortex collector" aperture diameter to the vessel diameter are critical dimensions and are essential to the success of the IRIS as a good dryer.

The IRIS design shown in Figure 5.13 is typical of what would be used in the <u>direct</u> atmosphere dryer system.

The IRIS steam dryer design used with the <u>indirect</u> steam atmosphere dryer configuration consists of a jacketed IRIS vessel and, if required, internal extended heat transfer surface. The jacketed vessel is required to provide the direct conduction heat transfer to the dryer's feedstock. This conduction amounts to 80 to 90 percent of the feedstock heating requirements.

During this task, particular attention was paid to the design of the jacketed IRIS vessel, given its importance to the operation of the indirect steam atmosphere dryer system. Several design concepts for the jacketed IRIS dryer were made. The two basic concepts are displayed in Figures 5.14 and 5.15.

Figure 5.14 presents a multiple array of jacketed IRIS dryers of reduced diameters. These dryers when paralleled together provide the proper surface area for conduction heat transfer while maintaining the correct proportion of dryer diameter to length. This arrangement also allows each dryer to maintain the proper steam inlet velocity and feedstock particle loading. The inlet steam and the jacket condensate for the dryers are all manifolded together. This arrangement also presents some versatility in the means of controlling the dryer's overall part-load performance. Most importantly, the use of internal, extended heat transfer surface area is eliminated, or greatly minimized.

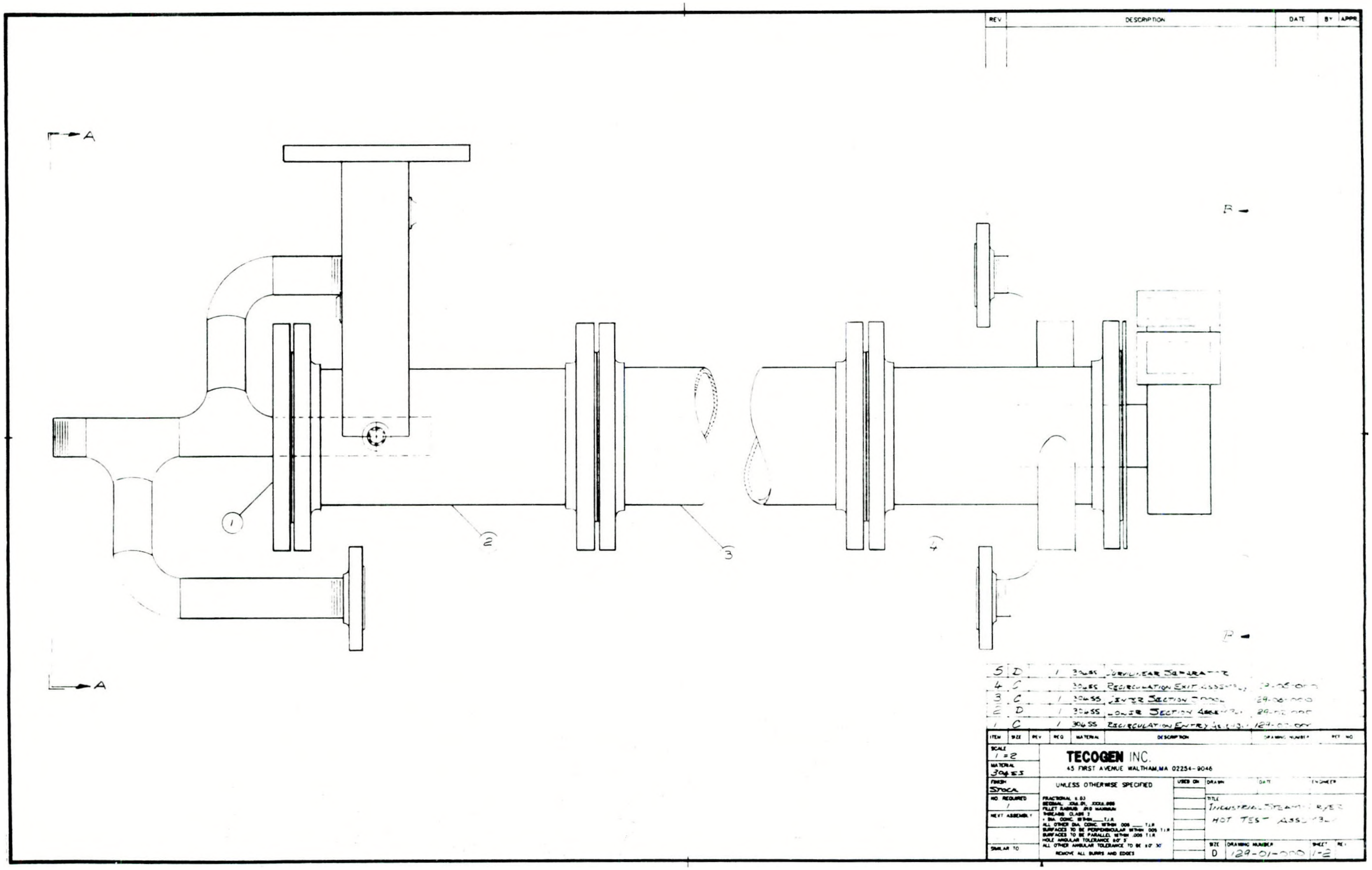

Figure 5.13 IRIS Design Drawing

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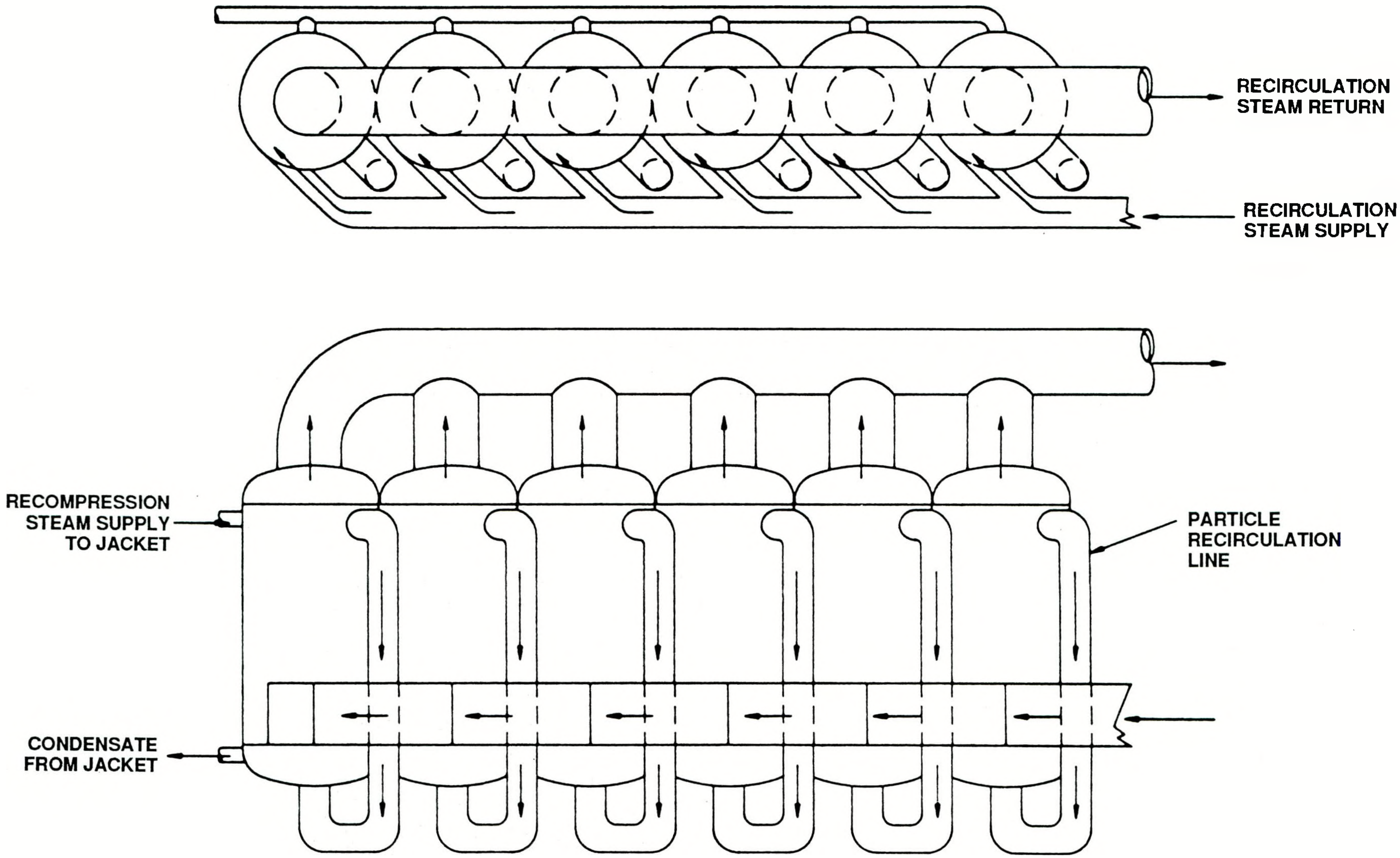

Figure 5.14 An Array of Multiple IRIS Steam Dryers

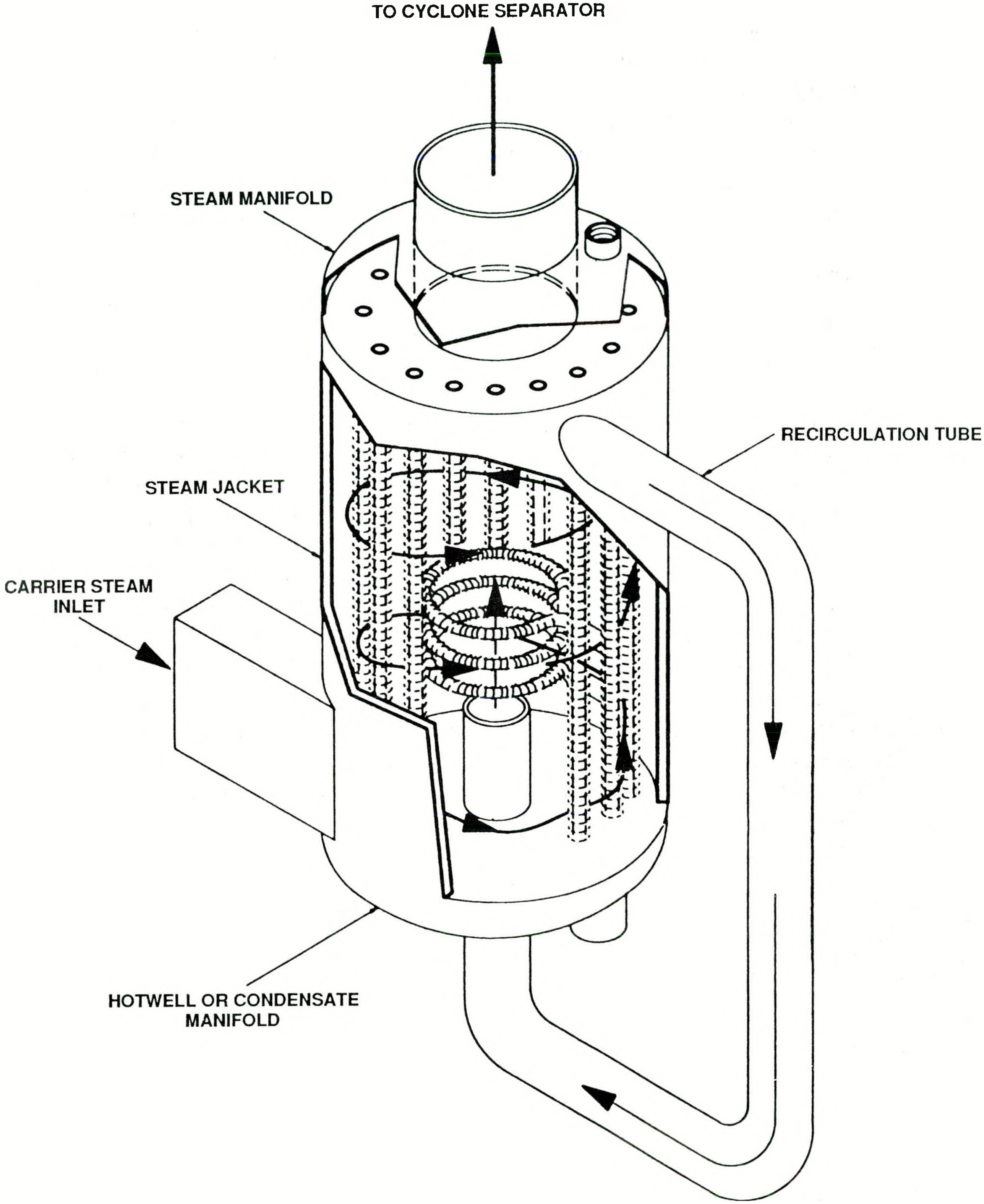

Figure 5.15 IRIS Steam Dryer with Steam Jacket and Extended Surface for Enhanced Heat Transfer

Figure 5.15 presents an alternative jacketed IRIS dryer design. By using internal surface area, a single IRIS can be used to dry all of the feedstock. It is not clear at this time what amount of extended surface is required to provide the necessary conduction heat transfer to the feedstock particle streams. The analysis conducted by Tecogen indicated that a heat transfer coefficient (h) of 100 Btu/hr-^{o}F-ft^2 would be sufficient to have a simple rippled surface provide the necessary heat transfer. Values of h = 100 Btu/hr-^{o}F-ft^2 have been reported in the literature. However, the exact values for the overall heat transfer coefficient (h) for this IRIS dryer configuration are best determined from heat transfer (i.e., feedstock drying) experiments. The Phase II of this project should be devoted, in part, to determining these values. As shown in Figure 5.15, an internal finned tube surface could be installed, if necessary, to complement the wet feedstock stream's flow field. Although Figure 5.15 displays both a vertical array of finned tubes (along the vessel's walls) and a conically wound finned tube coil (above the steam's recirculation spout tube), only one of these finned tube configurations has been calculated to be necessary to provide heat transfer to the wet feedstock. The choice for either the vertical or the conical tubes should be determined from the effects of each tube's position on the steam and feedstock flow streams. Once again, this is recommended as a test to be conducted in Phase II of the project. Although the jacketed IRIS dryer with internal surface area looks complicated, it must be remembered that the reboiler heat exchanger needed in the direct atmosphere dryer is eliminated by the use of this internal surface area (and the small auxiliary steam boiler). Any concerns about the clogging of the finned tubing by the drying feedstock particles can be resolved by utilizing 2 to 3 fins per inch (as was done in Figure 5.15) or by utilizing suitable steam injection cleaning methods. The concepts presented in Figures 5.14 and 5.15 are recognized as being viable alternative designs for the IRIS dryer for use with the indirect steam atmosphere dryer.

6. TASK 4 – LABORATORY TESTING

6.1 LABORATORY TEST OBJECTIVES

A principal task in Phase I of the Steam Atmosphere Drying Project was the performance of cold (with air as the medium) and hot (with steam as the medium) testing of bench-scale models of the IRIS-type dryer. The purpose of these tests was twofold. For the cold testing the objective was to visualize the fluid dynamic processes of the IRIS vortex flow field while measuring dryer pressure drop and particle collection efficiency. For the hot testing the objective was to verify the ability of the steam IRIS dryer to dry slurried feedstock while measuring its heat transfer characteristics; i.e., heat transfer coefficient, drying performance with respect to dryer size, slurry flow rate, and conformity to known heat transfer coefficient relationships.

In order to accomplish both these project objectives, considerable effort was put forth with regard to apparatus design, fabrication and laboratory testing. This was necessary in order to be confident in the results obtained as well as to be able to provide a more permanent bench-scale laboratory test facility for testing the IRIS dryer for future development interests.

This section details the laboratory testing work.

6.1.1 <u>Cold Test Laboratory Results of Tecogen's Bench-Scale IRIS Model</u>

The bench-scale IRIS model cold test apparatus was assembled to provide visualization of the fluid dynamics that occur in an IRIS-type dryer. This laboratory apparatus also provided an opportunity to measure the effects of particle loading (L_p) (defined as the ratio of the particle mass flow rate to transport gas mass flow rate) on the performance of the IRIS dryer. Particle loading (L_p) was identified as a critical design variable in Tecogen's Systems' Analysis and Design Study conducted in Tasks 2 and 3 of this first phase of the project. A high particle loading is a characteristic of the indirectly heated steam atmosphere dryer system. It allows lower dryer steam flow rates through the dryer with a corresponding decrease in steam fan power parasitics and hence an increase in drying energy savings. These cold fluid tests afforded the first opportunity to measure the effects of high particle loading on the dryer's performance. Changes in the "performance" of the dryer are noted by the changes occurring in the dryer's vortex pressure drop, effects on the recirculation ratio (R) or dryer collector efficiency, and the witnessing of dryer saturation or stalling of the IRIS dryer vortex as particles become trapped in the dryer or as they simply escape from the dryer without becoming recirculated in the external recirculation line.

The laboratory apparatus built for the cold testing of the IRIS model consisted of the following principal components:

1. A clear Plexiglass IRIS-type dryer assembly with inlet, outlet, and external recirculation lines (see Figure 6.1).

2. A notched, variable orifice area slide valve and a gravity particle feed tube assembly and vibrator. This assembly allowed a means of gravity-feeding the dry test material into the IRIS dryer.

3. A curvilinear louvered separator (C.L.S.) assembly (see Figure 6.2).

4. An induction fan and motor installed downstream from the IRIS dryer and particle baghouse.

5. A dry baghouse assembly to separate the dry particles from the air to prevent the particles from being exhausted into the atmosphere.

6. Test instrumentation consisting of five (5) pitot tube and static pressure manometers. This instrumentation recorded (in inches of water column) the pressure drops: (a) across the entire unit (inlet to outlet); (b) across the IRIS vortex flow field (i.e., the pressure drop between the center of the dryer chamber and the interior dryer wall), (c) between the inlet of the IRIS and the inlet to the induction fan motor, (d) across the baghouse (i.e., monitoring the onset of the baghouse clogging with dried particles), and (e) at the inlet and outlet of the external recirculation line.

7. Dry alumina, non-dairy coffee creamer, and clay powder test particles ranging in size from 30 microns to 203 microns of clay material and ranging in bulk density from 55 lb/ft^3 to 165 lb/ft^3.

8. An adjustable length recirculation spout tube, which was found to affect control over the dried recirculation ratio.

A piping and instrument (P&I) diagram for the cold test apparatus is displayed in Figure 6.3. A test plan was developed by Tecogen Inc. for conducting the cold test (as well as the hot test). The experimenter made only minor changes in the manner in which these tests were conducted in order to provide the best possible data. This became necessary as new experimental methods and new test objectives were identified to improve the experimental technique and increase the experiment's results. A typical experiment procedure was conducted as follows:

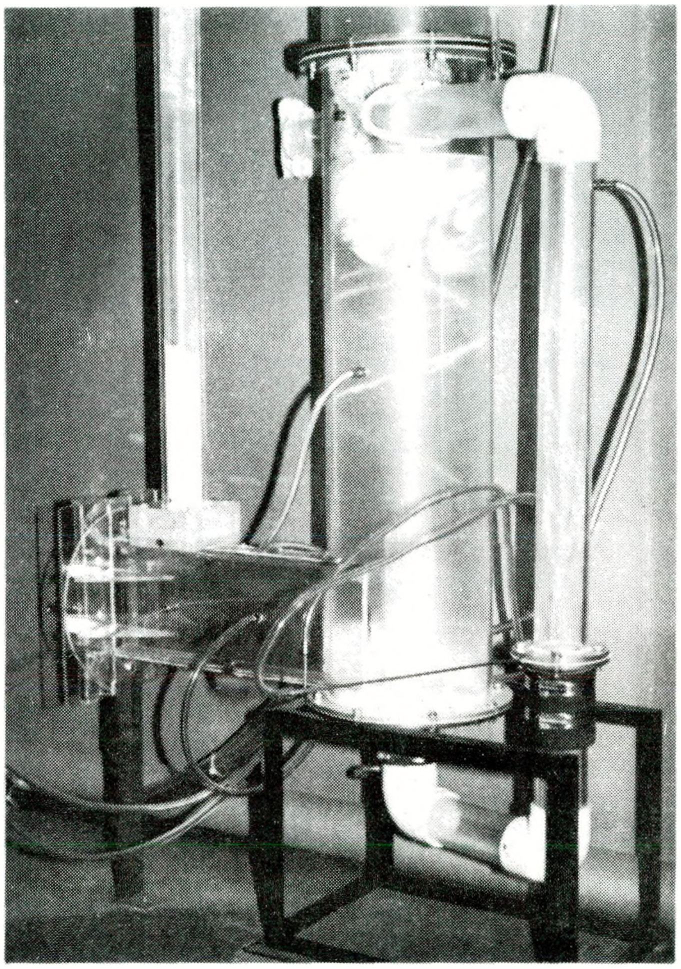

Figure 6.1 Tecogen's Phase I, Cold Flow Laboratory Test Model

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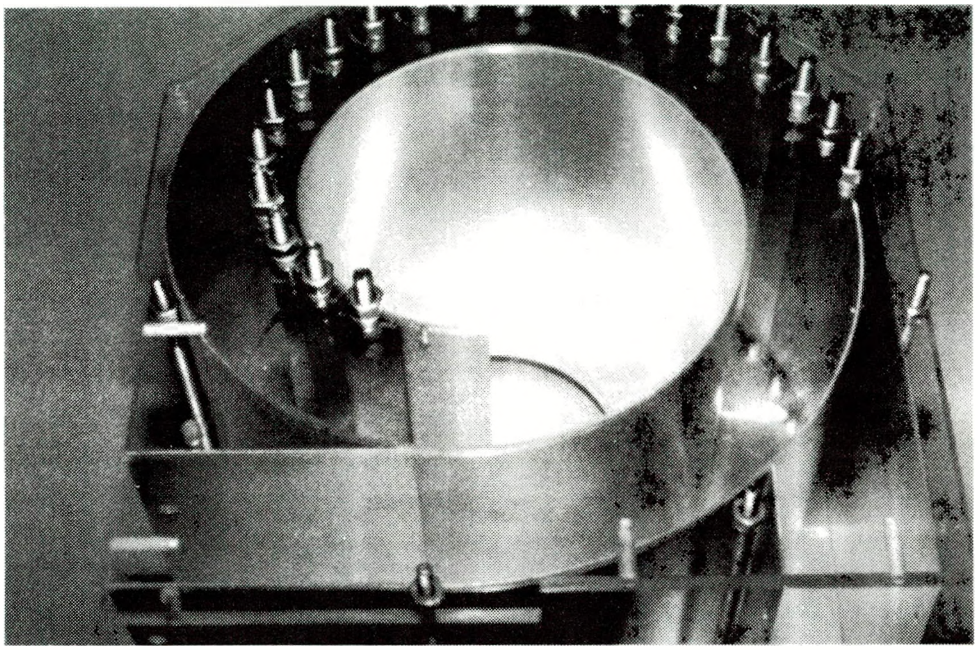

Figure 6.2 Curvilinear Louvered Separator

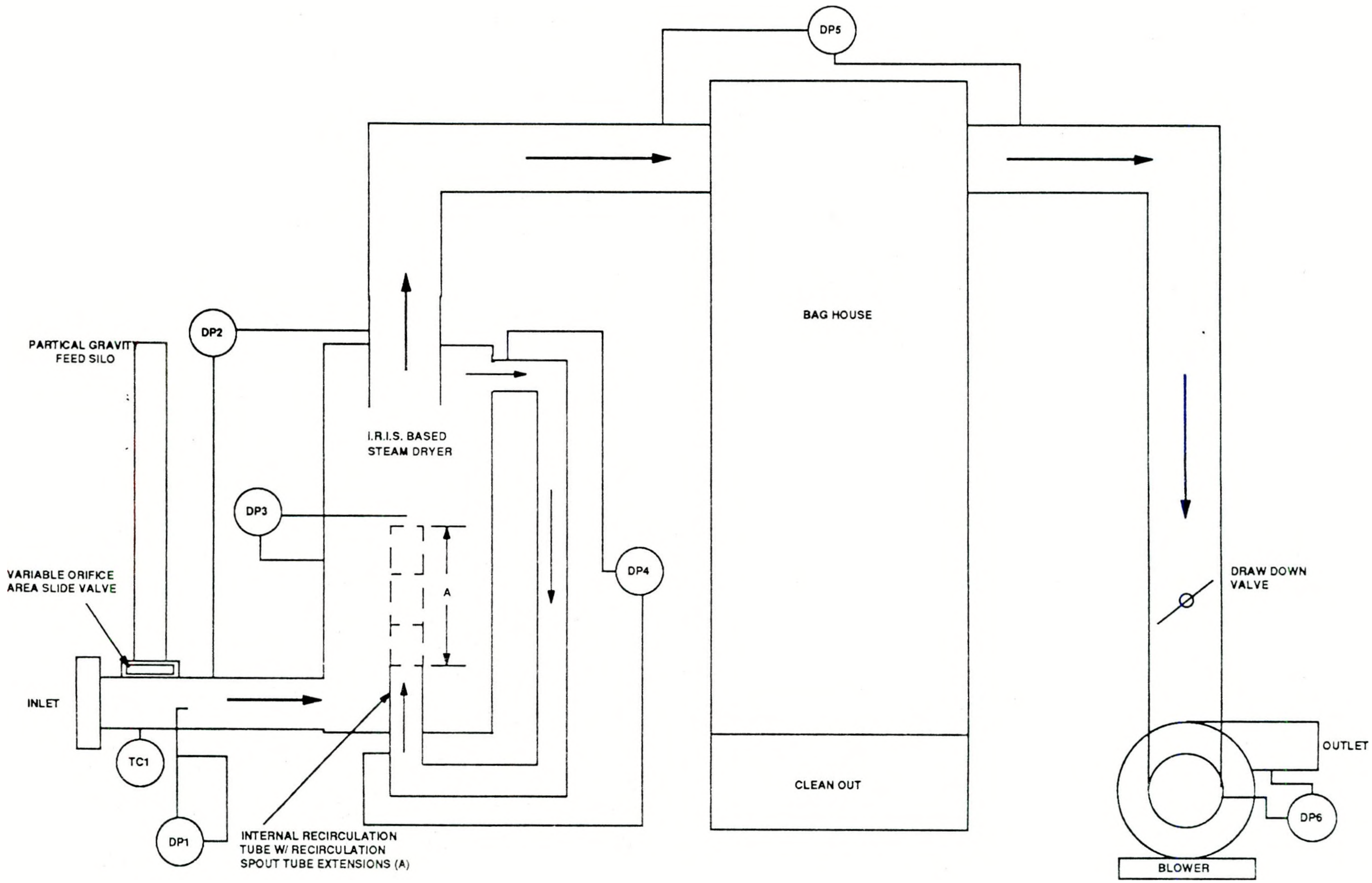

Figure 6.3 Steam Dryer Cold Flow Test Loop

1. Weigh and measure a batch of dry feedstock (alumina, creamer, or clay particles) of a given micron size and fill the gravity feed tube.

2. Turn on the induction fan motor and (with no particles flowing in the system) record the system's initial pressure drops.

3. Begin the gravity feed of dried particles by selecting one setting of the variable orifice-area slide valve. The time duration for the feedstock delivery and the change in height of the feedstock in the delivery tube were recorded to determine the mass flow rate of the dried feedstock into the dryer.

4. Observe carefully the ability of the IRIS dryer to externally recirculate the feedstock and also to eventually expel the particles from the dryer. The time required to saturate the dryer was recorded. The dryer is saturated when the selected feedstock mass flow rate is too large; thus, the dryer cannot expel the particles as fast as the feedstock is entering the dryer. This "fill time" ranges from zero to many minutes depending upon the inlet air velocity, particle size, and particle flow rate. The determination of the fill time is somewhat subjective. However, the same test personnel were used throughout the test and graduations on the dryer wall were used as references as to when the dryer is completely "filled."

5. Note and record the fill time duration. When this was done, the air inlet to the dryer was completely shut off but the induction fan was kept operational and thus the system was allowed to evacuate. The dryer evacuation time was also recorded and represents the time required for the system to rid itself of the particles that filled the dryer during the previous procedure (step 4).

   The fill time and evacuation time can both be shown to be a function of the dryer's recirculation or particle collection efficiency; that is, how efficiently does the external recirculation line collect a particle from the wall of the dryer and return it to the center of the dryer vortex via the external recirculation line? An efficiency of zero (0) implies the recirculation line is not recovering particles. An efficiency of 99 percent implies that the recirculation line is recovering 99 percent of the particles flowing through the dryer and is returning them to the dryer (for further drying). It can be shown that the recirculation ratio (R) of the dryer – defined as the number of recirculations of a given particle per single passing of the carrier gas through the dryer – is related to the dryer's collection efficiency by the following relationship.

$$R = \frac{1}{1 - \eta} \tag{1}$$

where:

η = the dryer's collection efficiency.

From a fluid dynamic analysis on the IRIS-dryer configuration it was also shown that the fill time and evacuation time are both measures of the dryer's particle collection efficiency. That is, given an initial dryer particle mass content of ($\dot{m}_o$) the change in the dryer's mass content with respect to time is given by:

$$\dot{m}_{(t)} = \dot{m}_o \text{ EXP } \frac{\left[- (1 - \eta)\ W_{gas}(t)\right]}{(\text{Den.}) \times V_d} \tag{2}$$

where:

W_{gas} = gas flow rate in the dryer (lb/min), and
V_d = total dryer geometric volume (ft^3)

The term inside the exponent is essentially a time constant. The final mass within the dryer is reduced to 0.67 percent of the dryer's initial mass content if this time constant is equal to 5. Given this, it is only a matter of measuring the time duration (i.e., evacuation time) required to empty the drying chamber to find the dryer's collection efficiency (η). The recirculation ratio (R) follows from Equation 1. An interesting and necessary condition for the proper measurement of the dryer's particle collection efficiency measurement is that the "fill time" must equal or exceed the "evacuation time." This necessary theoretical fluid dynamic condition provides a check for the otherwise subjective measurements for these time durations. This check was used to qualify all of the data collected and becomes useful in interpreting the test results.

6. A change in the gravity feed rate of the feedstock was made by changing the variable orifice area slide valve position (open or closed). Machined indentations in the slide valve mechanism helped to ensure that the same position of the slide valve could be repeated from one test to another.

7. A change in the recirculation spout tube length was also made. Increments of 1, 3, 5, 7, and 9 inches were used. Changes in these tube lengths drastically changed the dryer's fill and evacuation times and hence reflected changes in the recirculation ratio or dryer collection efficiency.

8. Changes in the gas inlet velocity with all of the above were also made.

In all, 150 data points were collected, representing how the bench-scale IRIS dryer performed fluid dynamically when several principal design parameters – inlet gas velocity, particle size and density, recirculation spout tube length, and particle mass flow rate – were changed and their effects on the vortex flow field pressure drop, recirculation ratio (collection efficiency), and dryer core saturation were measured.

6.2 DRYER FLUID DYNAMIC THEORY AND ESTABLISHING TEST OBJECTIVES

A major accomplishment of the Phase I dryer analysis was the first-time development of a fluid dynamic and heat transfer computer model of the IRIS dryer. This model determined the operating zone for the IRIS dryer as a function of the particle size and dryer gas inlet velocity. Figure 6.4 displays one such operating zone for a specific dryer. The analysis identified four principal operational constraints that must be satisfied by an effective IRIS dryer:

1. The system overall pressure drop, which was also found to be essentially equal to the vortex flow field pressure drop. The vortex flow field pressure drop is important as it establishes the pressure differential in the external recirculation line which induces the external recirculation flow rate and thus creates the dryer's ability to collect and recirculate particles. A high system drop results in increased fan power parasitics. Obviously, it is desirable to maintain the dryer's pressure drop high enough to allow recirculation but not so high to result in high fan parasitics. The question to be answered is: How does the vortex pressure drop vary with gas inlet velocity?

2. The dryer's particle collector efficiency is a function of gas inlet velocity. This constraint is empirically available from previous state-of-the-art studies of conventional cyclone fluid dynamics. A principal relationship developed for cyclone designs is given in Figure 6.5 which displays cyclone collection efficiency with respect to a diameter ratio: d_p/dp_{th}, or particle diameter to theoretical particle diameter. The theoretical particle diameter is given by the well established expression (after an analysis by Rosin, Rammler, and Intelmann: "Principles and Limits of Cyclone Dust Removal"):

$$dp_{th} = \sqrt{9\ \mu_g \left(\frac{W}{12}\right) / \pi\ N\ V_i\ (\varphi_p - \varphi_g)}\ x\ \frac{12}{0.00003937} \tag{3}$$

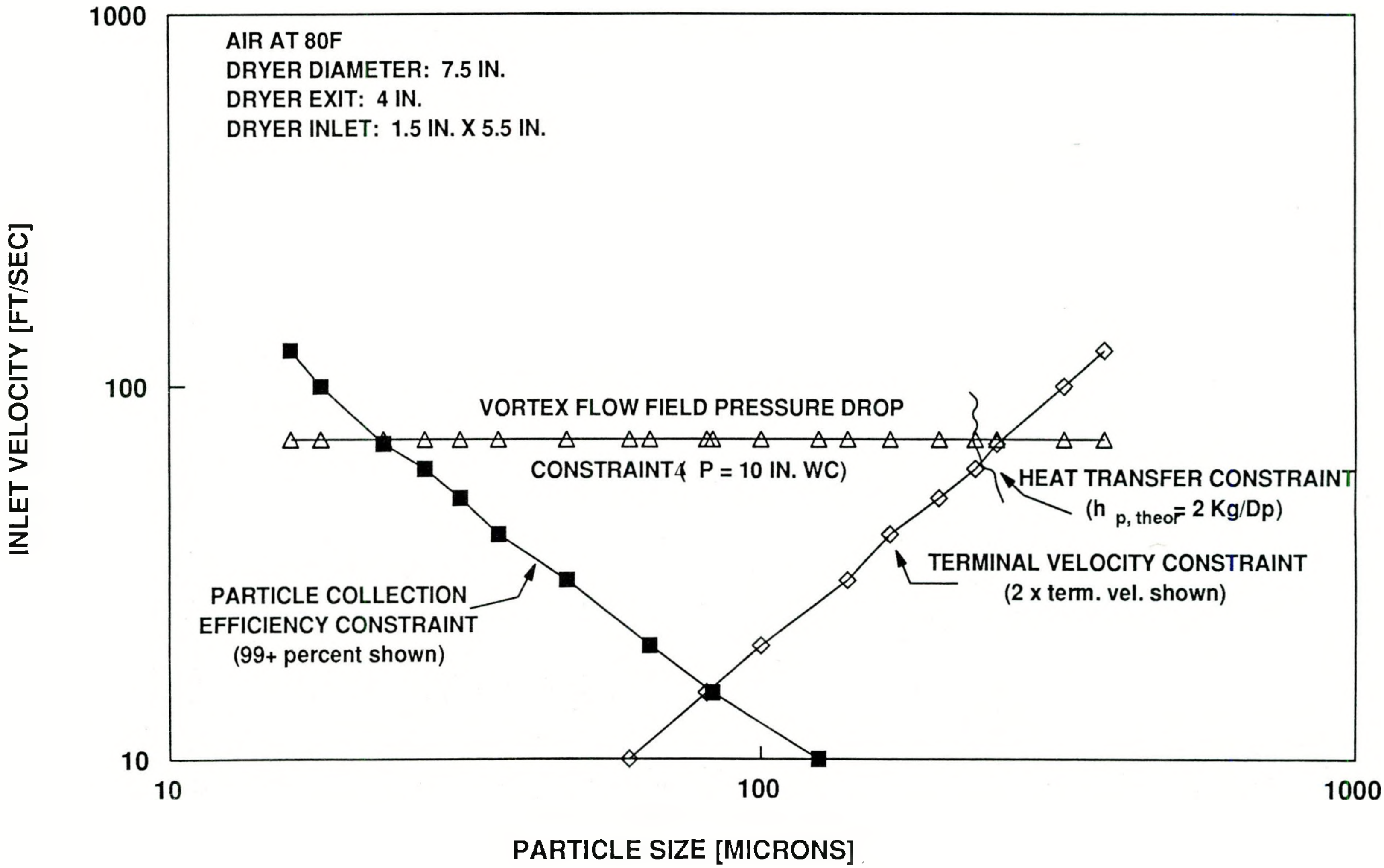

Figure 6.4 Operating Zone for Cold Test IRIS Model Based on IRIS Computer Model

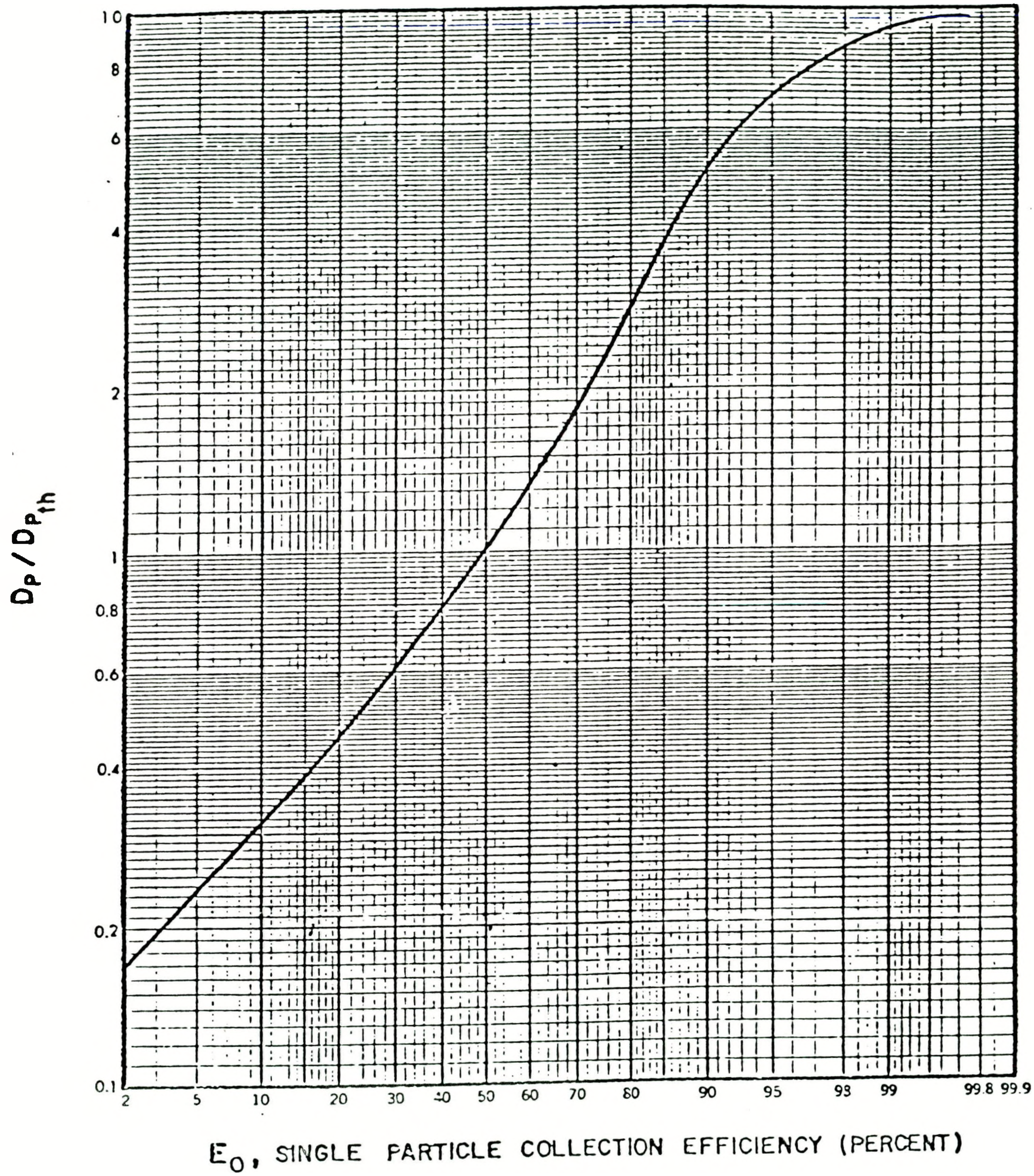

Figure 6.5 Typical Normalized Fractional Collection Efficiency Curve

where:

μ_g = gas viscosity (lb/sec-ft)
W = gas entrance width (inch)
N = number of spiral turns
V_i = inlet gas velocity (ft/sec)
φ_p = particle density (lb/ft^3)
φ_g = gas density (lb/ft^3)
dp_{th} = theoretical particle size (microns)

Using this expression with Figure 6.5, and given the cold test operating parameters of particle size, gas velocity, gas viscosity, particle density and entrance geometry, it is possible to identify what the gas inlet velocity to particle size relationship must be to maintain a high (99+ percent) dryer collector efficiency. The question to be answered is: Does this relationship accurately represent the constraints for the IRIS dryer's particle collection efficiency, and if not, what is the constraint relationship (if any) for the IRIS dryer?

3. An additional fluid dynamic constraint on the performance of the IRIS dryer is one associated with the particle's terminal velocity. For a given size particle, there exists a minimum dryer gas velocity that will suspend or fluidize the particles that have been injected into the flow stream. The terminal velocity is that velocity which causes an exact balance between the particle's weight and the drag imposed on it by the (fluidizing) gas stream. The computer model calculated the terminal velocity and multiplies this value by 2 as a sufficient minimum condition for particle suspension. This identifies the inlet gas velocity requirements for a given particle diameter if particle carry-over or recirculation is to be possible. The question to be answered is: Is the calculation of terminal velocity with respect to gas inlet velocity and particle size correctly modeled, and if not, what are the constraints with respect to the particle's terminal velocity or entrainment limits?

4. A fourth IRIS-dryer system constraint used in the IRIS computer model involves a heat transfer constraint that can only be addressed in the actual hot (steam) testing of the IRIS-type bench-scale model. This constraint requires that the particle's exposure time within the superheated steam medium should be long enough to completely dry the water-laden particle. If the local heat transfer coefficients are not high enough then the wet particle will require longer exposure times, and thus the dryer will need to have higher recirculation ratios (higher dryer particle collection efficiency). If higher recirculation ratios cannot be achieved, then larger IRIS dryer vessels (i.e., larger dryer diameters and/or dryer lengths) may be necessary to compensate for this heat transfer design requirement.

The operation zone that these four constraints map out (as shown in Figure 6.4) can be made larger or smaller depending on the validity of the computer models used for these constraints. The verification of these models or the empirical determination of better models is thus a principal objective of the cold and hot testing.

In order to determine the pressure drop (for constraint number 1), collection efficiency vs. particle size and gas velocity (constraint number 2), terminal velocity (constraint number 3) and heat transfer coefficient (constraint number 4), several straightforward fluid dynamic measurements of the system pressure drop, dryer "fill time" and dryer "evacuation time" (for constraint number 2) as well as visual observation of the feedstock saturation limits (constraint number 3) for the dryer were performed during the hot tests (for constraint number 4) and cold tests (for constraints number 1, 2, and 3). Reporting these test results and the conclusions drawn from the results were the principal goals of the laboratory testing (Task 4) of Phase I's steam atmosphere drying project.

6.2.1 Cold Test Results

A tabulation of all of the data points collected during the cold testing is displayed in Table 6.1 for reference purposes. All subsequent data analysis presented in this section can be referenced back to this database as necessary. The data have been sorted with respect to particle size and recirculation tube length for data analysis purposes only, and thus the order of the data presented in Table 6.1 is not the order in which the data were collected. The measured data are identified with a single asterisk in the header column; two asterisks mark the columns that have their entries calculated based on the measured values.

Some of the following data analysis shown in the graphs and tables is often identified as being "qualified." That is, by applying one of the theoretical guidelines discussed earlier concerning the relative magnitudes of the "fill time" and "evacuation time" (fill time must be equal to or greater than the evacuation time) some of the measured data were excluded and only the qualified data was used to generate the figure or table.

Dimensional Analysis Results

It is of interest to consider the results of a dimensional analysis conducted with the data shown in Table 6.1. Several dimensionless groups can be and were discerned from a strict theoretical application of dimensional analysis concepts (Buckingham-PI Theorem, etc.). However, the following dimensional groups and their corresponding equation best fit the measured data:

TABLE 6.1
COMPILATION OF ALL COLD TEST DATA

FEEDER POSITION (CLICKS)	INLET VELOCITY (fps)	PARTICAL DIAMETER (micron)	PITOT TUBE #1 ("h2o)	RECIRC TUBE PLUS (in)	DIFF PRES IN-OUT ("h2o)	DIFF PRES RECIRC ("h2o)	DIFF PRES VORTEX ("h2o)	MATERIAL USED (in)	PARTICAL FLOW TIME (sec)	EVAC TIME (sec)	MATERIAL FLOW RATE (LB min)	SEPERATION EFFICIENCY (%)	RECYCLE NUMBER (#)	PARTICAL BULK (S.G.)	MASS LOAD RATIO	VORTEX DELTA P RATIO	THEOR.	PART.DIA.	Dp/Dp.rn
0	74.99	42	1.25	1	5.75	1.8	5.4	0	0	0	0	0	0	0	0.0000	0.48	8.534043	4.921465	0.180
1	74.99	42	1.25	1	5.2	2	4	7	422	900	0.122492	99.916937	1203.906	1.45	0.0062	0.36	7.957505	5.278035	0.133
0	86.25	76	1.65	1	7.83	2.41	7.11	0	0	0	0	0	0	0	0.0000	0.48	7.957505	9.550731	0.237
1	86.25	76	1.65	1	7.4	3.4	7.4	2.5	86	218	0.229472	99.701848	335.3995	1.55	0.0102	0.50	8.015803	9.481270	0.247
2	85	76	1.6	1	6.8	3.2	5.8	12	130	495	0.728662	99.866761	750.5351	1.55	0.0327	0.40	7.844627	9.688158	0.193
0	88.75	89	1.75	1	8.34	2.57	7.57	0	0	0	0	0	0	0	0.0000	0.48	7.844627	11.34534	0.252
1	88.75	89	1.75	1	8	3.8	7.8	4.5	113	163	0.320440	99.612477	258.0494	1.58	0.0138	0.49	7.900462	11.26516	0.260
2	87.5	89	1.7	1	6.6	2.4	5.6	12	120	8	0.804662	91.991428	12.48662	1.58	0.0351	0.37	7.957505	11.18440	0.187
0	86.25	102	1.65	1	7.83	2.41	7.11	0	0	0	0	0	0	0	0.0000	0.48	7.957505	12.81808	0.237
1	86.25	102	1.65	1	7.6	3.2	7.4	4	123	88	0.264991	99.261396	135.3906	1.6	0.0117	0.50	8.015803	12.72486	0.247
2	85	102	1.6	1	6.2	2.2	5.4	12	142	376	0.688603	99.824593	570.1034	1.6	0.0339	0.37	8.097184	12.59697	0.180
0	83.3	122	1.55	1	7.31	2.25	6.66	0	0	0	0	0	0	0	0.0000	0.48	8.097184	15.06696	0.222
1	83.3	122	1.55	1	7.2	3	7.2	3	101	60	0.246572	98.878351	89.15447	1.63	0.0113	0.52	8.097184	15.06696	0.240
2	83.3	122	1.55	1	6.6	2.8	5.4	7.5	77	256	0.808564	99.737113	380.3924	1.63	0.0370	0.39	7.768411	15.70462	0.180
0	90.5	145	1.8	1	8.6	2.65	7.8	0	0	0	0	0	0	0	0.0000	0.48	7.768411	18.66533	0.260
1	90.5	145	1.8	1	8	3.6	8	3	81	35	0.311226	98.230149	56.50196	1.65	0.0131	0.49	7.844627	18.48398	0.267
2	88.75	145	1.75	1	6.6	2.2	5.6	13	148	272	0.738111	99.767771	430.6100	1.65	0.0317	0.35	7.844627	18.48398	0.187
0	88.75	165	1.75	1	8.34	2.57	7.57	0	0	0	0	0	0	0	0.0000	0.48	7.844627	21.03350	0.252
1	88.75	165	1.75	1	7.8	3.6	7.6	2	69	8	0.246521	92.104225	12.66500	1.67	0.0106	0.48	7.844627	21.03350	0.253
2	88.75	165	1.75	1	6	1.8	5	8.5	88	254	0.821503	99.777583	449.6075	1.67	0.0353	0.32	8.309373	19.85709	0.167
0	79.1	203	1.4	1	6.31	2.03	6	0	0	0	0	0	0	0	0.0000	0.48	8.309373	24.43024	0.200
1	79.1	203	1.4	1	6.4	2.9	6.2	5.5	183	11	0.255614	93.557062	15.52087	1.67	0.0123	0.49	8.309373	24.43024	0.207
2	79.1	203	1.4	1	0	0	0	9.25	110	765	0.715191	99.907356	1079.405	1.67	0.0345	0.00	8.534043	23.78708	0.180
0	74.99	42	1.25	3	5.75	1.8	5.4	0	0	0	0	0	0	0	0.0000	0.48	8.534043	4.921465	0.180
1	74.99	42	1.25	3	5.6	2	4.2	4.25	246	249	0.127578	99.699772	333.0808	1.45	0.0065	0.37	8.534043	4.921465	0.140
2	74.99	42	1.25	3	3.5	0.4	1	13.75	199	461	0.510239	99.837838	616.6676	1.45	0.0260	0.09	7.957505	5.278035	0.033
0	86.25	76	1.65	3	7.83	2.41	7.11	0	0	0	0	0	0	0	0.0000	0.48	7.900462	9.619690	0.237
1	87.5	76	1.7	3	7.7	3.4	7.4	2	64	58	0.246682	98.895369	90.52800	1.55	0.0108	0.48	7.957505	9.550731	0.247
2	86.25	76	1.65	3	7.2	3	6.8	5	72	170	0.548183	99.617664	261.5501	1.55	0.0243	0.46	8.015803	9.481270	0.227
3	85	76	1.6	3	6.2	2	5	9	72	271	0.98673	99.756631	410.8990	1.55	0.0443	0.35	7.844627	9.688158	0.167
0	88.75	89	1.75	3	8.34	2.57	7.57	0	0	0	0	0	0	0	0.0000	0.48	7.844627	11.34534	0.252
1	88.75	89	1.75	3	7.8	3.6	7.8	3.5	102	31	0.276109	97.962380	49.07688	1.58	0.0119	0.49	7.900462	11.26516	0.260
2	87.5	89	1.7	3	7.4	3.2	7	7.5	92	146	0.655974	99.561174	227.8808	1.58	0.0286	0.46	7.957505	11.18440	0.233
0	86.25	102	1.65	3	7.83	2.41	7.11	0	0	0	0	0	0	0	0.0000	0.48	7.957505	12.81808	0.237
1	86.25	102	1.65	3	7.4	3.3	7.2	4.25	115	16	0.301139	95.937681	24.61648	1.6	0.0133	0.48	8.015803	12.72486	0.240
2	85	102	1.6	3	5.8	2	5.2	11.5	136	288	0.689025	99.649186	285.0517	1.6	0.0309	0.36	8.097184	12.59697	0.173
0	83.3	122	1.55	3	7.31	2.25	6.66	0	0	0	0	0	0	0	0.0000	0.48	8.097184	15.06696	0.222
1	83.3	122	1.55	3	7	3.2	6.8	2.5	72	10	0.288238	93.270108	14.85907	1.63	0.0132	0.49	8.097184	15.06696	0.227
2	83.3	122	1.55	3	6.6	3.2	6.2	6.5	68	69	0.793503	99.024653	102.5276	1.63	0.0363	0.45	8.181095	14.91242	0.207
3	81.6	122	1.5	3	5.6	2	4.6	7	50	122	1.162176	99.436877	177.5811	1.63	0.0543	0.34	7.957505	15.33143	0.153
0	86.25	145	1.65	3	7.83	2.41	7.11	0	0	0	0	0	0	0	0.0000	0.48	7.957505	18.22178	0.237
1	86.25	145	1.65	3	7.6	3.6	7.6	2.5	73	7	0.287778	90.714699	10.76971	1.65	0.0127	0.51	7.957505	18.22178	0.253
2	86.25	145	1.65	3	7	3.4	6.6	6.5	61	59	0.895414	98.898354	90.77327	1.65	0.0396	0.44	8.015803	18.08926	0.220
3	85	145	1.6	3	6	1.8	4.6	10	67	134	1.254197	99.507813	203.1751	1.65	0.0563	0.32	7.768411	18.66533	0.153
0	90.5	165	1.8	3	8.6	2.65	7.8	0	0	0	0	0	0	0	0.0000	0.48	7.768411	21.23986	0.260
1	90.5	165	1.8	3	8.4	4	8	2	53	2	0.320942	69.027624	3.228683	1.67	0.0135	0.49	7.768411	21.23986	0.267
2	90.5	165	1.8	3	7.8	3.6	7.2	5.5	58	32	0.806506	98.064226	51.65893	1.67	0.0340	0.44	7.844627	21.03350	0.240
3	88.75	165	1.75	3	6.2	2	5	7	44	101	1.353064	99.374592	159.8956	1.67	0.0582	0.32	8.309373	19.85709	0.167
0	79.1	203	1.4	3	6.31	2.03	6	0	0	0	0	0	0	0	0.0000	0.48	8.309373	24.43024	0.200
1	79.1	203	1.4	3	6.4	3.2	6.2	5.25	194	12	0.230160	94.093973	16.93185	1.67	0.0111	0.49	8.309373	24.43024	0.207
2	79.1	203	1.4	3	5	1.2	4.2	10.25	49	142	1.779102	99.504389	201.7713	1.67	0.0858	0.34	8.374204	24.24110	0.140

TABLE 6.1
COMPILATION OF ALL COLD TEST DATA

0	77.88	42	1.35	5	6.12	1.96	5.8	0	0	0	0	0	0	0	0.0000	0.48	8.374204	5.015401	0.193
1	77.88	42	1.35	5	6.2	2.4	4.8	5	300	60	0.123076	98.800291	83.35354	1.45	0.0060	0.40	8.440576	4.975963	0.160
2	76.66	42	1.3	5	3.5	0.4	1	9.25	147	230	0.464674	99.682051	314.5165	1.45	0.0231	0.08	7.957505	5.278035	0.200
0	86.25	76	1.65	5	7.83	2.41	7.11	0	0	0	0	0	0	0	0.0000	0.48	7.900462	9.619690	0.237
1	87.5	76	1.7	5	7.8	3.6	7.6	3.5	80	19	0.345355	96.627969	29.65572	1.55	0.0151	0.50	7.957505	9.550731	0.253
2	86.25	76	1.65	5	7.4	3.2	7	4.75	60	66	0.624929	99.015195	101.5429	1.55	0.0276	0.47	7.957505	9.550731	0.233
3	86.25	76	1.65	5	6.6	2.4	5	11	70	126	1.240460	99.484149	193.8547	1.55	0.0549	0.34	7.900462	9.619690	0.167
0	87.5	89	1.7	5	8.09	2.49	7.34	0	0	0	0	0	0	0	0.0000	0.48	7.900462	11.26516	0.245
1	87.5	89	1.7	5	7.6	3.7	7.6	2.5	76	6	0.264691	89.321904	9.364966	1.58	0.0115	0.50	7.957505	11.18440	0.253
2	86.25	89	1.65	5	7.2	3.4	6.8	6.5	79	41	0.662064	98.414704	63.07973	1.58	0.0293	0.46	8.015803	11.10306	0.227
3	85	89	1.6	5	6.2	2.2	5	13	111	110	0.942397	99.400427	166.7855	1.58	0.0423	0.35	8.181095	10.87873	0.167
0	81.6	102	1.5	5	7.06	2.17	6.42	0	0	0	0	0	0	0	0.0000	0.48	8.181095	12.46776	0.214
1	81.6	102	1.5	5	7	3.6	7	3	111	5	0.220229	86.259803	7.277916	1.6	0.0103	0.52	8.181095	12.46776	0.233
2	81.6	102	1.5	5	6.4	3.2	6.2	8	87	39	0.749285	98.238436	56.76774	1.6	0.0350	0.46	8.242947	12.37421	0.207
3	80.38	102	1.45	5	5	1.2	4.2	15.25	96	142	1.29442	99.508847	203.6025	1.6	0.0614	0.32	8.097184	12.59697	0.140
0	83.3	122	1.55	5	7.31	2.25	6.66	0	0	0	0	0	0	0	0.0000	0.48	8.097184	15.06696	0.222
1	83.3	122	1.55	5	7.2	3.6	7.2	2.5	84	3	0.247061	77.567026	4.457723	1.63	0.0113	0.52	8.015803	15.21993	0.240
2	85	122	1.6	5	6.8	3	6.4	4.5	52	17	0.718378	96.120415	25.77595	1.63	0.0322	0.44	8.097184	15.06696	0.213
3	83.3	122	1.55	5	6.6	3.2	6	9	64	46	1.167365	98.597939	71.32358	1.63	0.0535	0.43	8.097184	15.06696	0.200
4	83.3	122	1.55	5	5.8	2	4.6	7	34	255	1.709083	99.736082	378.9065	1.63	0.0783	0.33	7.957505	15.33143	0.153
0	86.25	145	1.65	5	7.83	2.41	7.11	0	0	0	0	0	0	0	0.0000	0.48	7.957505	18.22178	0.237
1	86.25	145	1.65	5	7.6	3.8	7.4	2	56	4	0.300111	83.750724	6.154120	1.65	0.0133	0.50	7.957505	18.22178	0.247
2	86.25	145	1.65	5	7.2	3	6.6	4.5	47	18	0.804554	96.389049	27.69354	1.65	0.0356	0.44	7.957505	18.22178	0.220
3	86.25	145	1.65	5	6	2	4.2	16	114	80	1.179385	99.187536	123.0824	1.65	0.0522	0.28	7.900462	18.35335	0.140
0	87.5	165	1.7	5	8.09	2.49	7.34	0	0	0	0	0	0	0	0.0000	0.48	7.900462	20.88485	0.245
1	87.5	165	1.7	5	7.4	3.6	7	1.5	43	1	0.296685	35.931428	1.560827	1.67	0.0129	0.46	7.900462	20.88485	0.233
2	87.5	165	1.7	5	7	3.2	6.4	6	72	10	0.708748	93.593142	15.60827	1.67	0.0309	0.42	7.900462	20.88485	0.213
3	87.5	165	1.7	5	5.8	1.6	4.2	7	47	49	1.266698	98.692478	76.48055	1.67	0.0552	0.27	8.309373	19.85709	0.140
0	79.1	203	1.4	5	6.31	2.03	6	0	0	0	0	0	0	0	0.0000	0.48	8.242947	24.62711	0.200
1	80.38	203	1.45	5	6.5	3.4	6.4	5	197	13	0.215862	94.635098	18.63967	1.67	0.0102	0.49	8.309373	24.43024	0.213
2	79.1	203	1.4	5	6.2	2.8	5.5	12	152	19	0.671445	96.269878	26.80877	1.67	0.0324	0.44	8.309373	24.43024	0.183
3	79.1	203	1.4	5	5	1.2	4.2	23.25	147	143	1.345174	99.504389	201.7713	1.67	0.0649	0.34	8.374204	24.24110	0.140
0	77.88	42	1.35	7	6.12	1.96	5.8	0	0	0	0	0	0	0	0.0000	0.48	8.374204	5.015401	0.193
1	77.88	42	1.35	7	6.2	2.5	4.8	4.5	255	30	0.130315	97.600582	41.67677	1.45	0.0064	0.40	8.309373	5.054532	0.160
2	79.1	42	1.4	7	6	2.2	4.4	10	240	51	0.30769	98.610346	71.96039	1.45	0.0148	0.35	8.181095	5.133787	0.147
3	81.6	42	1.5	7	5.2	1	1.8	19	145	147	0.967632	99.532646	213.9707	1.45	0.0452	0.13	7.900462	5.316144	0.190
0	87.5	76	1.7	7	8.09	2.49	7.34	0	0	0	0	0	0	0	0.0000	0.48	7.844627	9.688158	0.245
1	88.75	76	1.75	7	7.8	4	4.8	5	149	12	0.264893	94.736150	18.99750	1.55	0.0114	0.30	7.900462	9.619690	0.160
2	87.5	76	1.7	7	7.8	3.8	7.8	5	61	13	0.647036	95.071648	20.29075	1.55	0.0282	0.51	8.015803	9.481270	0.260
3	85	76	1.6	7	6.2	2	7.6	18	129	72	1.101466	99.083986	109.1687	1.55	0.0494	0.53	7.900462	9.619690	0.253
0	87.5	89	1.6	7	7.57	2.34	6.88	0	0	0	0	0	0	0	0.0000	0.45	7.957505	11.18440	0.229
1	86.25	89	1.65	7	7.4	4	7.4	3	81	4	0.298023	83.750724	6.154120	1.58	0.0132	0.50	8.015803	11.10306	0.247
2	85	89	1.6	7	7	3.4	6.6	6	64	18	0.754371	96.335947	27.29218	1.58	0.0339	0.46	8.015803	11.10306	0.220
3	85	89	1.6	7	6.6	3.4	6.2	10	83	37	0.969472	98.217488	56.10060	1.58	0.0435	0.43	8.015803	11.10306	0.207
4	85	89	1.6	7	5.8	1.8	4.2	8	45	51	1.430510	98.706805	77.32786	1.58	0.0642	0.29	8.181095	10.87873	0.140
0	81.6	102	1.5	7	7.06	2.17	6.42	0	0	0	0	0	0	0	0.0000	0.48	8.097184	12.59697	0.214
1	83.3	102	1.55	7	6.8	3.6	6.8	4	130	2	0.250722	66.350540	2.971815	1.6	0.0115	0.49	8.181095	12.46776	0.227
2	81.6	102	1.5	7	6.4	3	6	7	79	13	0.722017	94.715309	18.92258	1.6	0.0338	0.45	8.270259	12.33335	0.200
3	79.85	102	1.47	7	6	3	5.4	12.75	90	36	1.154368	98.152456	54.12593	1.6	0.0552	0.42	8.097184	12.59697	0.180
0	83.3	122	1.55	7	7.31	2.25	6.66	0	0	0	0	0	0	0	0.0000	0.48	8.097184	15.06696	0.222
1	83.3	122	1.55	7	7	3.6	7	2	56	1	0.296473	32.701080	1.485907	1.63	0.0136	0.50	8.097184	15.06696	0.233
2	83.3	122	1.55	7	6.8	3.2	6.4	3.5	39	3	0.744985	77.567026	4.457723	1.63	0.0341	0.46	8.097184	15.06696	0.213
3	83.3	122	1.55	7	6.4	3.2	5.6	7	50	22	1.162176	96.940958	32.68997	1.63	0.0532	0.40	8.097184	15.06696	0.187
4	83.3	122	1.55	7	6	2.4	4	6	32	32	1.556487	97.896908	47.54905	1.63	0.0713	0.29	7.957505	15.33143	0.133
0	86.25	145	1.65	7	7.83	2.41	7.11	0	0	0	0	0	0	0	0.0000	0.48	7.957505	18.22178	0.237
1	86.25	145	1.65	7	7.6	4	7.6	2	50	1	0.336124	35.002898	1.538530	1.65	0.0149	0.51	7.957505	18.22178	0.253

TABLE 6.1
COMPILATION OF ALL COLD TEST DATA

2	86.25	145	1.65	7	7.2	3.4	6.8	4.5	51	5	0.741451	87.000579	7.692650	1.65	0.0328	0.46	7.957505	18.22178	0.227
3	86.25	145	1.65	7	6.8	3.2	6	9	62	14	1.219807	95.357349	21.53942	1.65	0.0540	0.40	7.957505	18.22178	0.200
4	86.25	145	1.65	7	6	2.6	4.5	8	36	30	1.86736	97.833429	46.15590	1.65	0.0826	0.30	7.900462	18.35335	0.150
0	87.5	165	1.7	7	8.09	2.49	7.34	0	0	0	0	0	0	0	0.0000	0.48	7.900462	20.88485	0.245
1	87.5	165	1.7	7	7.4	4	7.2	3	69	1	0.369781	35.931428	1.560827	1.67	0.0161	0.47	7.900462	20.88485	0.240
2	87.5	165	1.7	7	7	3.4	6.4	4	47	2	0.723827	67.965714	3.121655	1.67	0.0316	0.42	7.957505	20.73514	0.213
3	86.25	165	1.65	7	6	2	4.4	10	68	31	1.250731	97.903319	47.69443	1.67	0.0553	0.30	8.181095	20.16844	0.147
0	81.6	203	1.5	7	7.06	2.17	6.42	0	0	0	0	0	0	0	0.0000	0.48	8.181095	24.81330	0.214
1	81.6	203	1.5	7	6.5	3.4	6.4	4.95	142	7	0.296476	90.185574	10.18908	1.67	0.0139	0.48	8.181095		0.213
2	81.6	203	1.5	7	6.6	3.2	6.2	9	162	7	0.472498	90.185574	10.18908	1.67	0.0221	0.46	8.181095		0.207
3	81.6	203	1.5	7	4	0.4	1.8	18	132	5.14	1.159769	86.634050	7.481698	1.67	0.0542	0.13	8.374204		0.060
0	77.88	42	1.35	9	6.12	1.96	5.8	0	0	0	0	0	0	0	0.0000	0.48	8.309373		0.193
1	79.1	42	1.4	9	7.2	3.3	6.4	3.5	240	12	0.107691	94.093973	16.93185	1.45	0.0052	0.51	8.309373		0.213
2	79.1	42	1.4	9	7	2.6	6	21.75	283	22	0.567541	96.778531	31.04174	1.45	0.0274	0.48	8.015803		0.200
3	85	42	1.6	9	7	2	5	13	113	48	0.849551	98.625980	72.77916	1.45	0.0381	0.35	8.015803		0.167
4	85	42	1.6	9	6.5	1.4	3.5	23	173	68	0.981762	99.030103	103.1038	1.45	0.0441	0.24	7.844627		0.117
0	88.75	76	1.75	9	8.34	2.57	7.57	0	0	0	0	0	0	0	0.0000	0.48	7.768411		0.252
1	90.5	76	1.8	9	8.4	4.6	8.4	4	127	7	0.248624	91.150749	11.30039	1.55	0.0105	0.51	7.844627		0.280
2	88.75	76	1.75	9	7.7	3.6	7.2	10.25	98	20	0.825631	96.841690	31.66250	1.55	0.0355	0.46	7.860141		0.240
3	88.4	76	1.73	9	7.2	3.2	6.4	13.75	92	36	1.179785	98.238436	56.76774	1.55	0.0509	0.41	7.873513		0.213
4	88.1	76	1.72	9	6.6	2.8	5.4	15	90	39	1.31564	98.368404	61.28968	1.55	0.0570	0.35	7.900462		0.180
5	87.5	76	1.7	9	5.8	2	4.2	18.5	91	64	1.604791	98.998928	99.89297	1.55	0.0700	0.27	8.097184		0.140
0	83.3	89	1.55	9	7.31	2.25	6.66	0	0	0	0	0	0	0	0.0000	0.48	8.015803		0.222
1	85	89	1.6	9	7.4	4	7.4	2	62	1	0.259568	34.047058	1.516232	1.58	0.0117	0.51	8.015803		0.247
2	85	89	1.6	9	6.8	3.2	6.4	4	44	5	0.731511	86.809411	7.581163	1.58	0.0328	0.44	8.097184		0.213
3	83.3	89	1.55	9	6.6	3.4	5.8	12	96	28	1.005828	97.596467	41.60542	1.58	0.0461	0.42	8.097184		0.193
4	83.3	89	1.55	9	5.6	2	4	15	80	64	1.508742	98.948454	95.09810	1.58	0.0691	0.29	8.181095		0.133
0	81.6	102	1.5	9	7.06	2.17	6.42	0	0	0	0	0	0	0	0.0000	0.48	8.097184		0.214
1	83.3	102	1.55	9	7.2	3.8	7	3	87	2	0.280982	66.350540	2.971815	1.6	0.0129	0.50	8.181095		0.233
2	81.6	102	1.5	9	6.2	3	6.2	4.25	48	5	0.72148	86.259803	7.277916	1.6	0.0337	0.46	8.257341		0.207
3	80.1	102	1.48	9	6.2	3	5.6	6.5	48	21	1.10344	96.667261	30.00535	1.6	0.0526	0.44	8.257341		0.187
4	80.1	102	1.48	9	5	1.6	3.6	15	68	54	1.797458	98.703934	77.15661	1.6	0.0856	0.28	8.097184		0.120
0	83.3	122	1.55	9	7.31	2.25	6.66	0	0	0	0	0	0	0	0.0000	0.48	8.097184		0.222
1	83.3	122	1.55	9	7.2	3.8	7.2	1.5	72	1	0.172943	32.701080	1.435907	1.63	0.0079	0.52	8.181095		0.240
2	81.6	122	1.5	9	6.6	3.2	6.2	3.5	37	2	0.785254	65.649509	2.911166	1.63	0.0367	0.46	8.181095		0.207
3	81.6	122	1.5	9	6.4	3	5.6	8	58	12	1.145001	94.274918	17.46699	1.63	0.0535	0.42	8.181095		0.187
4	81.6	122	1.5	9	5.2	1.8	3.6	15	75	51	1.660252	98.652921	74.23474	1.63	0.0776	0.27	7.957505		0.120
0	86.25	145	1.65	9	7.83	2.41	7.11	0	0	0	0	0	0	0	0.0000	0.48	7.900462		0.237
1	87.5	145	1.7	9	7.8	4	2.8	2	55	1	0.305568	35.931428	1.560827	1.65	0.0133	0.18	7.900462		0.220
2	87.5	145	1.7	9	7.4	3.4	6.8	4.5	50	2	0.756280	67.965714	3.121655	1.65	0.0330	0.44	7.900462		0.227
3	87.5	145	1.7	9	6.8	3	6.2	7.5	53	10	1.189120	93.593142	15.60827	1.65	0.0519	0.40	7.900462		0.207
4	87.5	145	1.7	9	6	2	4.4	18	101	41	1.497585	98.437351	63.99393	1.65	0.0653	0.29	7.957505		0.147
0	86.25	165	1.65	9	7.83	2.41	7.11	0	0	0	0	0	0	0	0.0000	0.48	7.900462		0.237
1	87.5	165	1.7	9	7.4	4	7.2	2	50	1	0.340199	35.931428	1.560827	1.67	0.0148	0.47	7.957505		0.240
2	86.25	165	1.65	9	6.8	3.2	6.2	6.5	67	2	0.825109	67.501449	3.077060	1.67	0.0365	0.42	7.957505		0.207
3	86.25	165	1.65	9	5.4	1.4	3	18.25	122	43	1.272260	98.488439	66.15679	1.67	0.0563	0.20	8.309373		0.100
0	79.1	203	1.4	9	6.31	2.03	6	0	0	0	0	0	0	0	0.0000	0.48	8.309373		0.200
0	79.1	203	1.4	9	0	0	0	15.25	127	72	1.108554	99.015662	101.5911	1.67	0.0535	0.00	ERR		

$$\frac{\Delta P}{H} = K\left[\frac{V\ \varphi_m\ D_o}{\mu_m}\right]^a [\varphi_p/\varphi_m]^b \left[1 - \frac{\dot{m}_p}{\dot{m}_m}\right]^c [d_p/l \times 12 \times 3.28 \times 10^{-6}]^d \left[\frac{H}{d_o}\right]^e \quad (4)$$

where:

- V = gas stream velocity (ft/sec)
- φ_m = gas density (lb/ft^3) = 0.075
- μ_m = gas viscosity (lb/ft-sec) = 0.00001477 ft-sec
- φ = particle density (lb/ft^3)
- m_p = particle flow rate (lb/hr)
- m_m = gas stream flow rate (lb/hr)
- l = recirc. spouting tube length (ft)
- d_p = particle size (microns)
- H = dryer height (inch) = 30 in.
- D_o = dryer diameter (inch) = 8 in.

The following values of the constants were determined from applying this dimensionless equation to the data in Table 6.1.

Using all of the Data	Using Qualified Data Only
a = 0.333	a = 0.333
b = 0.200	b = 0.200
c = 4.00	c = 3.00
d = -0.03	d = -0.035
e = 0 (the dryer height & lengths did not vary)	e = 0
k = 0.00065	k = 0.00065
Average Percent Differential: 16.5%	Average Percent Differential: 15.2%

Thus, it would appear that the loading dimensionless term: $[1 - m_p/m_m]$ has a significant effect upon the dryer's vortex flow field m_m pressure drop dimensionless group. The first dimensionless group is in actuality a Reynold's number. The power coefficient for the fourth term: $[d_p/l]$ was observed to clearly require a negative coefficient and thus seemingly establishes a dependence (albeit a very small one) of vortex pressure drop as inversely proportional with particle diameter.

A simple plot of the Table 6.1 data for vortex pressure drop with respect to dryer particle loading ($L_p = m_p/m_m$) is given in Figure 6.6a (using all of the data) and Figure 6.6b (for the qualified data). Some of the data is scattered but a distinct pattern emerges for vortex pressure drop as a function of particle loading (L_p). Some of the vertical scatter can be attributed to a degradation in system pressure

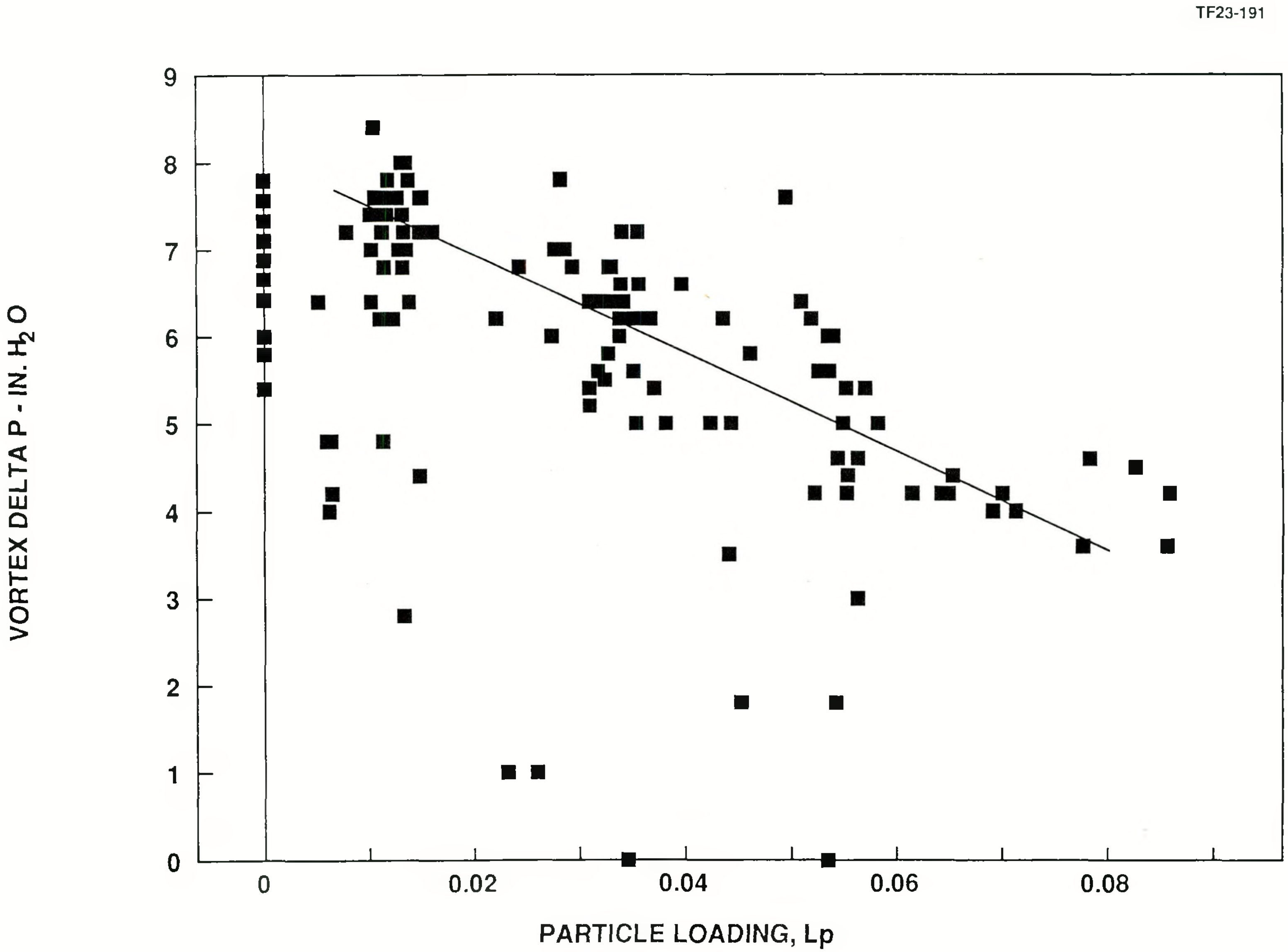

Figure 6.6a Cold Test Vortex ΔP vs. Particle Loading
(All Measured Data Used)

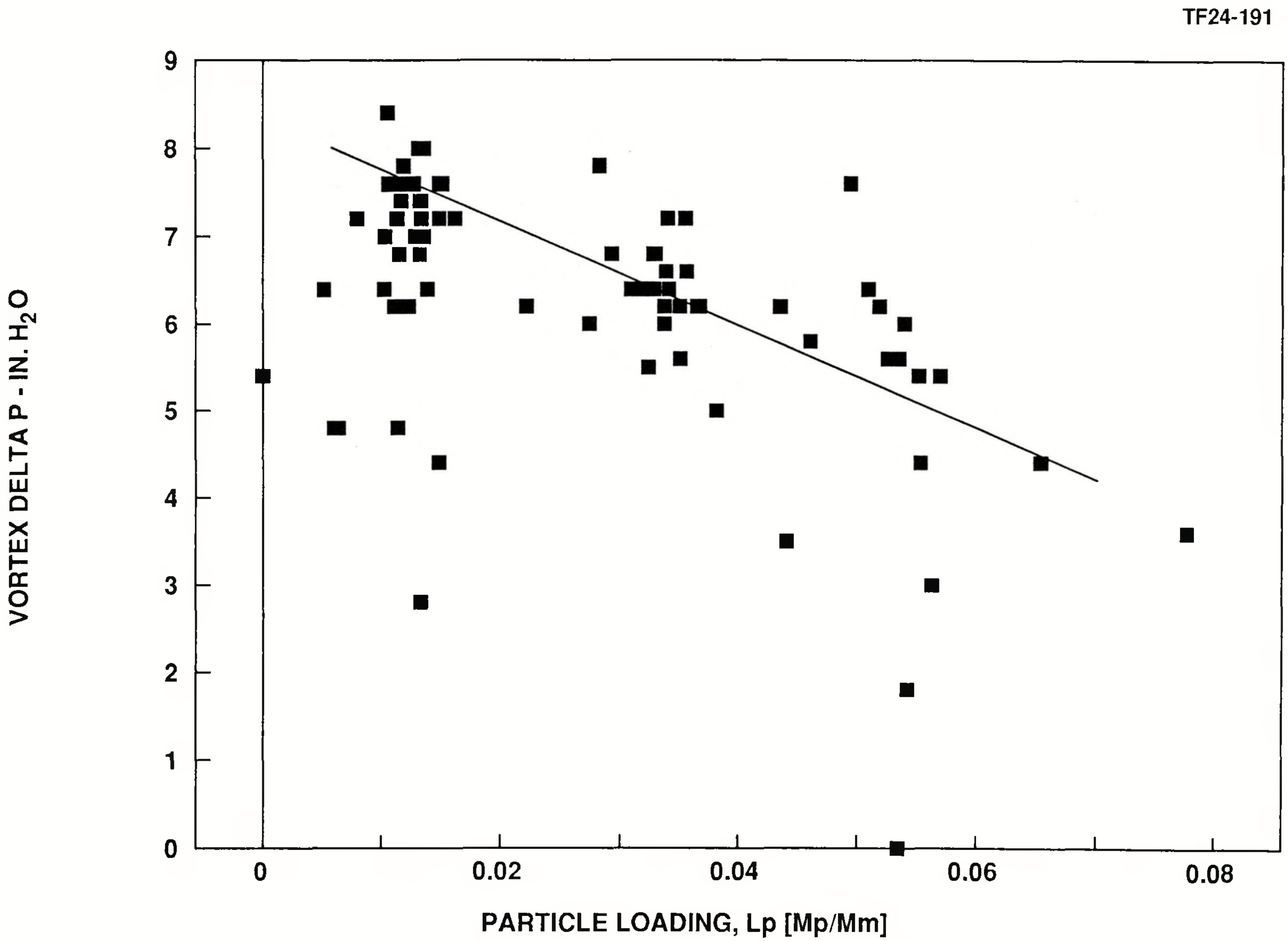

Figure 6.6b Vortex ΔP vs. Loading Using Qualified Data Only

drop due to the clogging of the baghouse during and between test points. The clogging of the baghouse results in a decrease in velocity through the system with a corresponding decrease in the system pressure drop that is independent of the dryer particle loading (L_p).

IRIS Dryer Pressure Drop Measurements (Constraint No. 1)

Figures 6.7a and 6.7b were constructed to determine how this measured pressure drop compared with the computer model's predicted value for the vortex pressure drop. These figures clearly show that the original pressure drop models for the IRIS dryer were conservative, and thus the dryer pressure drop constraint is not as severe as originally thought. Thus, the IRIS operating zone (shown in Figure 6.4) can be increased if a 10-in. wc pressure drop is thought to result in an acceptable fan power parasitic.

Prior to these tests there was some concern that an increase in the particle loading L_p would adversely affect the vortex pressure drop and hence reduce the dryer's recirculation ratio or particle collection efficiency. The cold test results provided the first opportunity to investigate this claim. Based on the data base (Table 6.1) a plot of recirculation ratio vs. vortex pressure drop was made, and this plot is presented in Figure 6.8 using only the qualified data. This interesting test result clearly reveals an increase in the recirculation ratio (and hence dryer particle collection efficiency) as the dryer's vortex pressure drop is decreased. Moreover, it has been found that this experimental revelation does have some basis in conventional cyclone separator design theory. A graphical representation of the effects of particle loading on the particle collection efficiency of cyclones is reproduced in Figure 6.9, which is from the studies of Rosin, Rammler, and Intelmann. This figure also shows an increase in expected cyclone particle efficiency even as the particle loading is increased. This conformity of Tecogen's cold test results to established cyclone theory is very encouraging and lends more confidence to the validity of the testing procedures and the results obtained.

The observation made in this experiment that the collection efficiency does not "go to zero" as the external recirculation line flow rate decreases is an important discovery and a confirmation of other independent experiments also conducted at Tecogen. The dryer's particle capture efficiency is affected by internal circulation flow streams set up within the dryer's vortex flow field. These internal flow streams, like the external recirculation flow streams, serve the same purpose: the retention of the feedstock particle within the drying medium until it is dried of its water content.

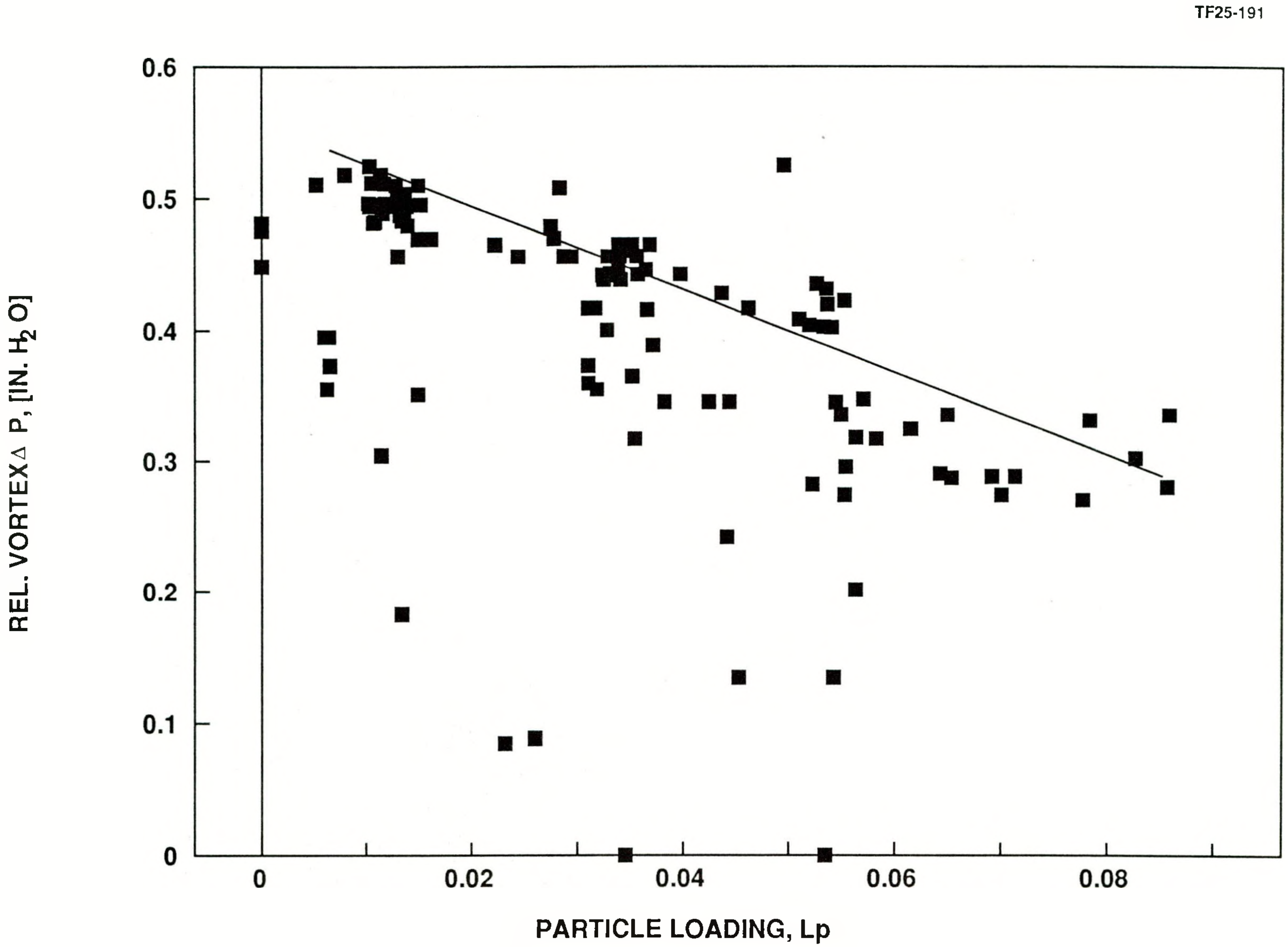

Figure 6.7a Relative Vortex ΔP vs. Particle Size
(All Measured Data Used)

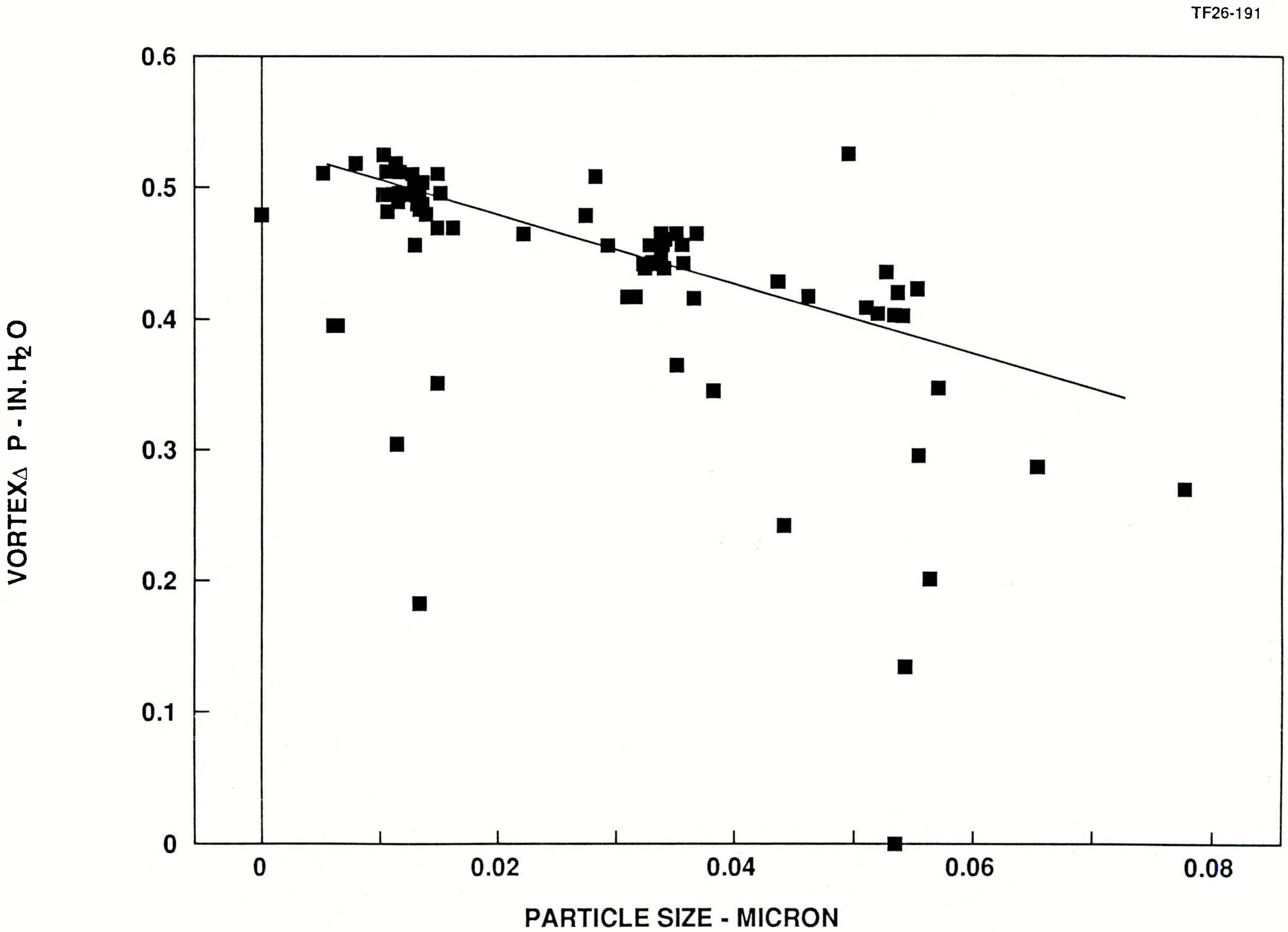

Figure 6.7b Relative Vortex ΔP vs. Loading, Lp
Dryer Loading = $M_{parts}/M_{gas\ stream}$; Qualified Data Only

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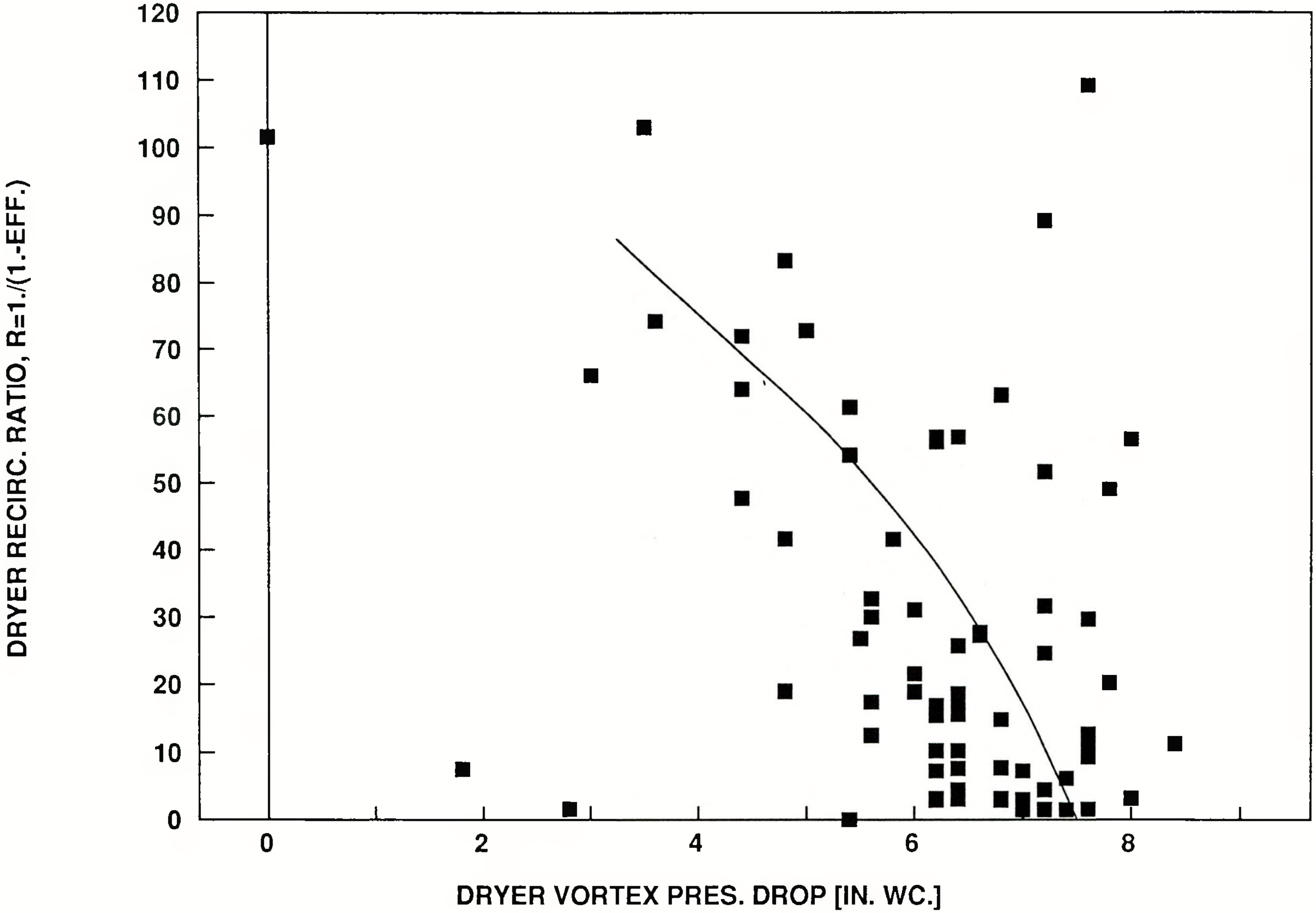

Figure 6.8 Recirculation Ratio (R) vs. Vortex ΔP Using Qualified Data Only

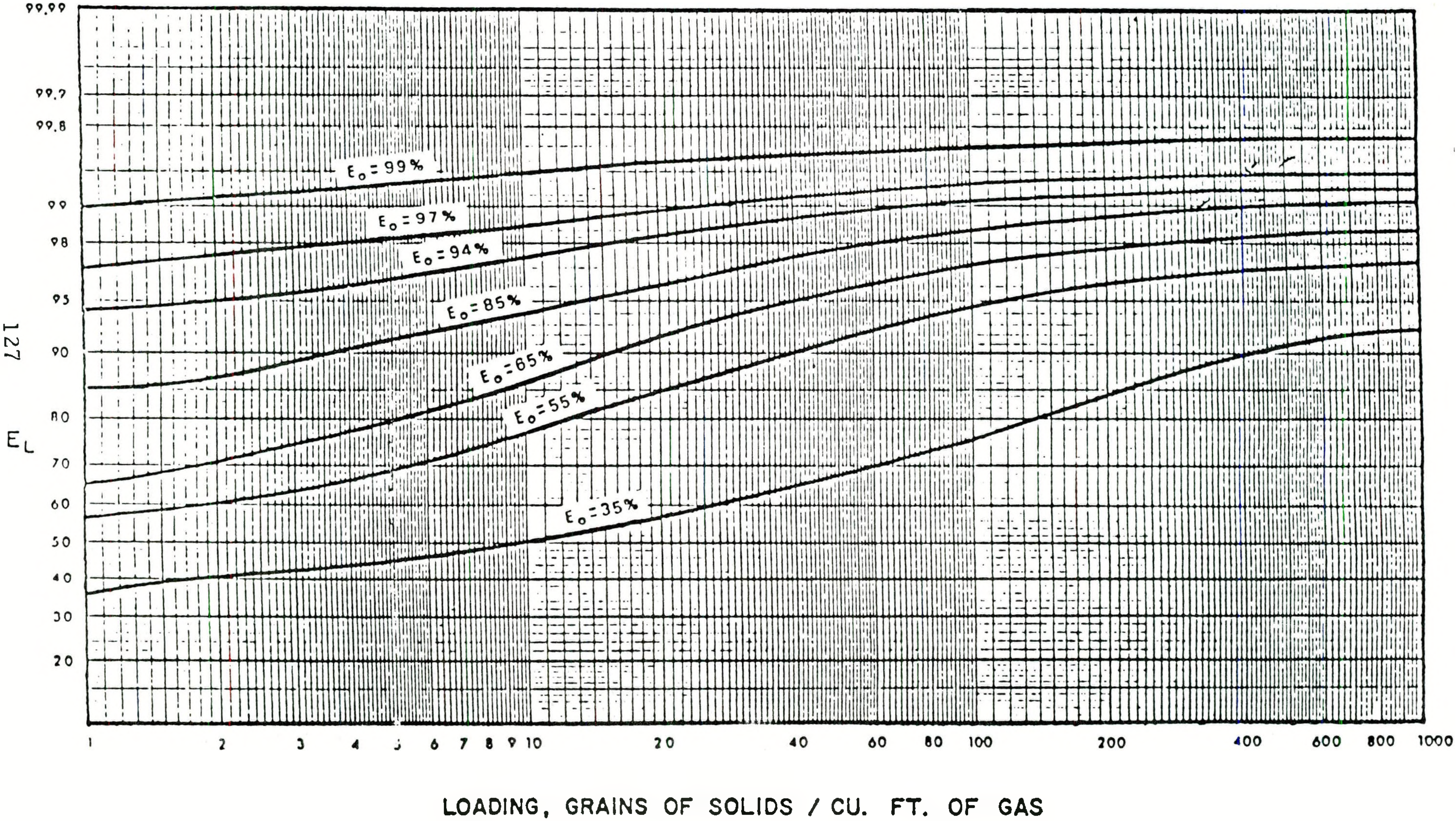

Figure 6.9 Correlation of Loading Effect on Collection Efficiency

It should also be noted that in the very first tests, the dryer vortex pressure drop vs. dryer inlet velocity was also measured with and without the dryer's external recirculation line blocked. The results of these tests conducted with the cold test apparatus are shown in Figure 6.10. It is interesting to note that the dryer's vortex pressure drop is the same whether the external recirculation line is open or partially blocked; however, a completely blocked recirculation line requires a larger vortex field pressure drop and hence larger fan parasitics.

IRIS Dryer Particle Collector Efficiency Measurements (Constraint No. 2)

One of the principal calculations made from the data measured in the cold test was the calculation of the dryer's particle collection efficiency or the dryer's recirculation ratio (R). As indicated previously, the dryer's particle collection efficiency is a function of gas inlet velocity and particle diameter and presents a second major constraint to the fluid dynamic operation of the dryer. A plot of the capture efficiency vs. the particle to theoretical particle ratio (d_p/dp_{th}) is presented in Figures 6.11a (using all of the data) and 6.11b (using only the qualified data). It may be observed that these measured data reveal almost universally very high dryer collection efficiencies as a function of d_p/d_{th} – values that are typically above 90 percent. These collection efficiencies are adequate to maintain sufficient particle residence times within the dryer's superheated steam medium to permit full particle drying. For example, a plot of the minimum dryer particle collection efficiency with respect to particle diameters is shown in Figure 6.12. This figure was calculated using the computer heat transfer model to determine the minimum time required to dry a wet particle and then to extrapolate back to determine what minimum dryer collection efficiency would be required to keep the particles inside the dryer for this time period. Based on the measured collection efficiency performance with the 30- to 200-micron particles, it appears that the collection efficiency constraint (shown previously in Figure 6.4) was also conservative; that is, the collector efficiency constraint boundary can be moved slightly to the left, thus allowing a larger dryer operational zone.

An alternative means of visualizing this shifting of the dryer collection efficiency constraint curve can be done by observing Figure 6.13. Figure 6.13 plots gas inlet velocity with respect to true particle diameter (microns), as is done in Figure 6.4. Using the data in Table 6.1 it can be shown that for a given gas inlet velocity, the measured collection efficiency is equal to or higher than the predicted values. The minimum particle collection efficiency for the particle sizes used is given in Figure 6.13.

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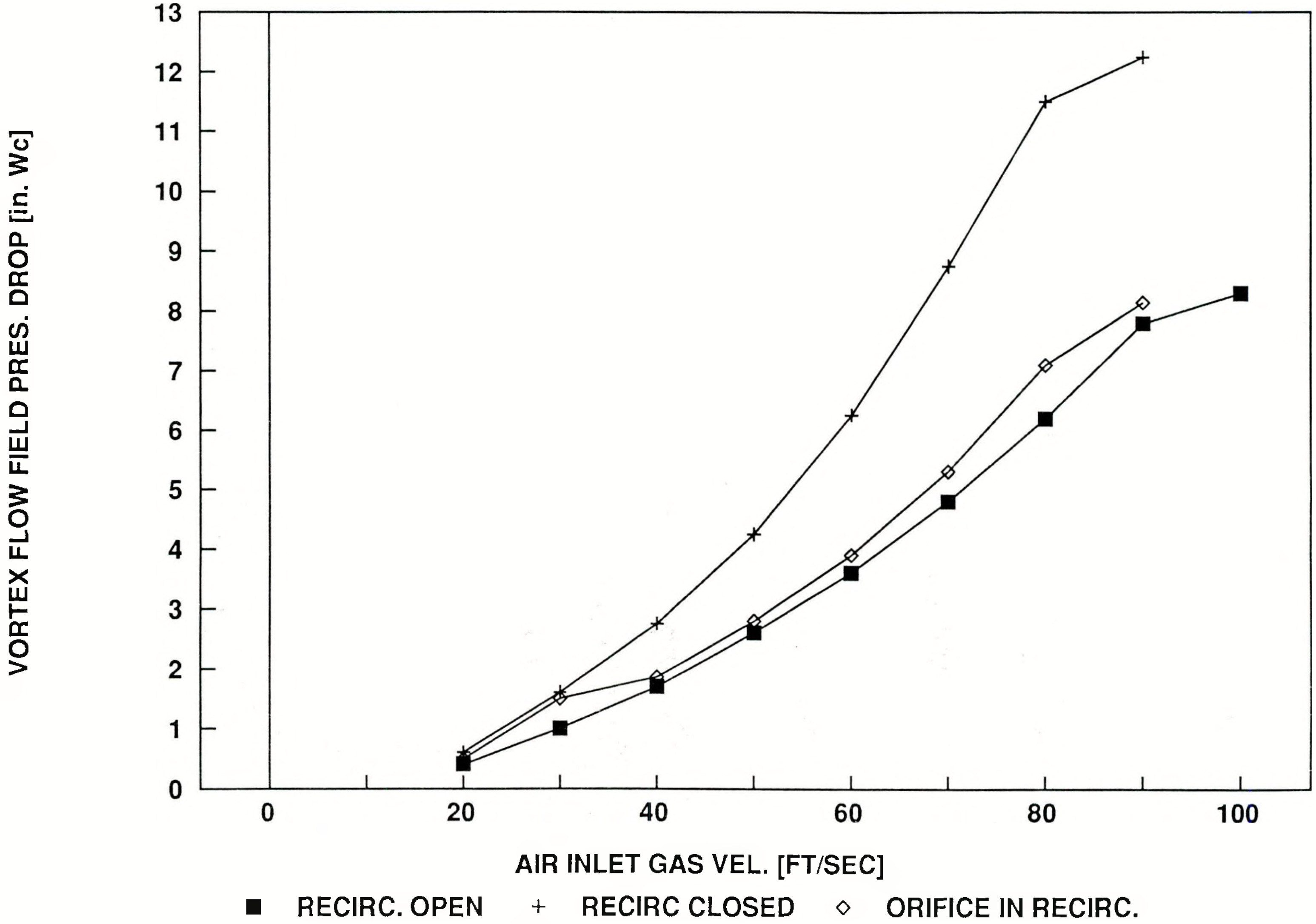

Figure 6.10 Cold Test IRIS Dryer Model Pressure Drop with Particle Loading (Lp) = 0

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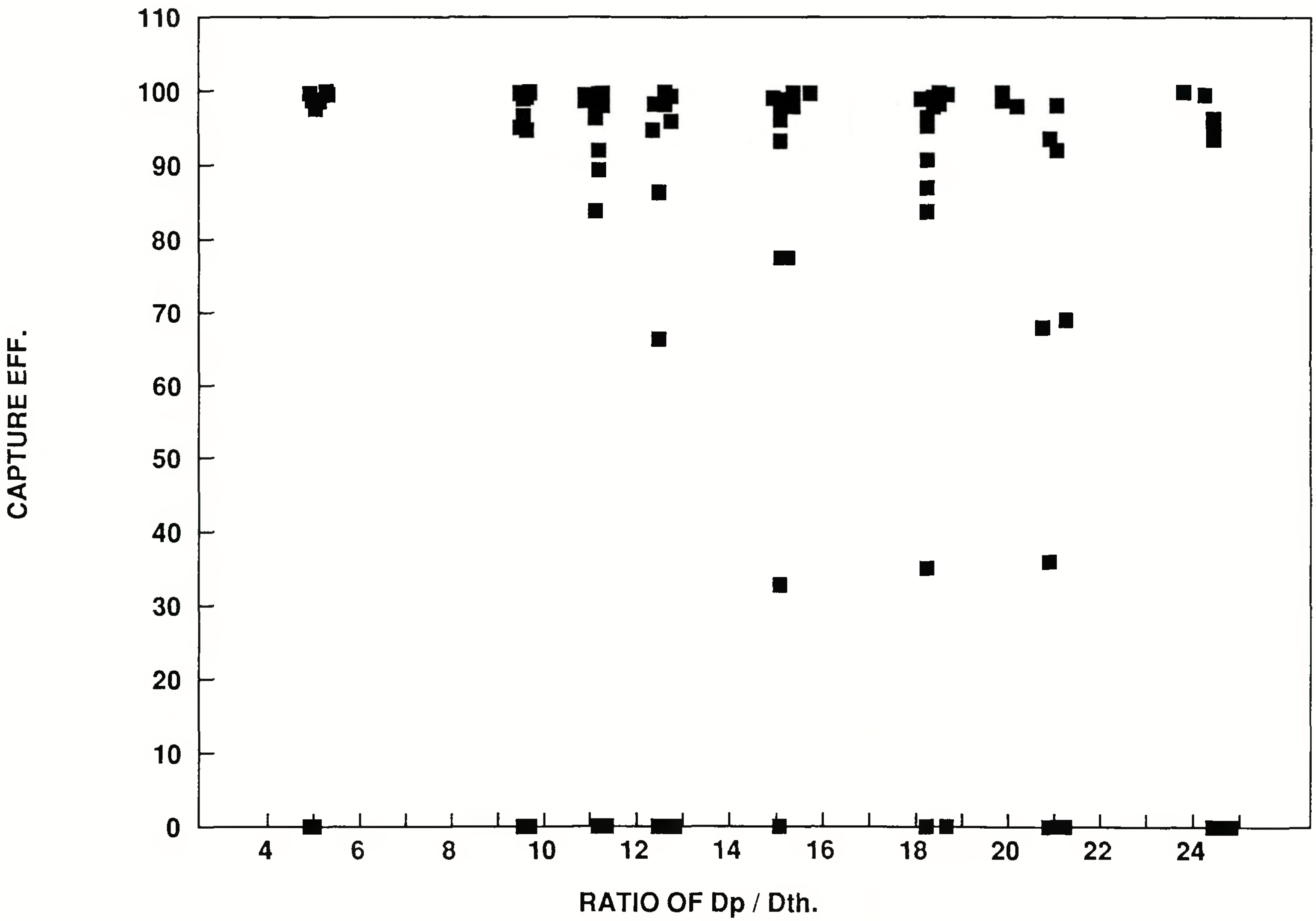

Figure 6.11a Cold Test Capture Efficiency vs. Dp/Dth
(All Measured Data Used)

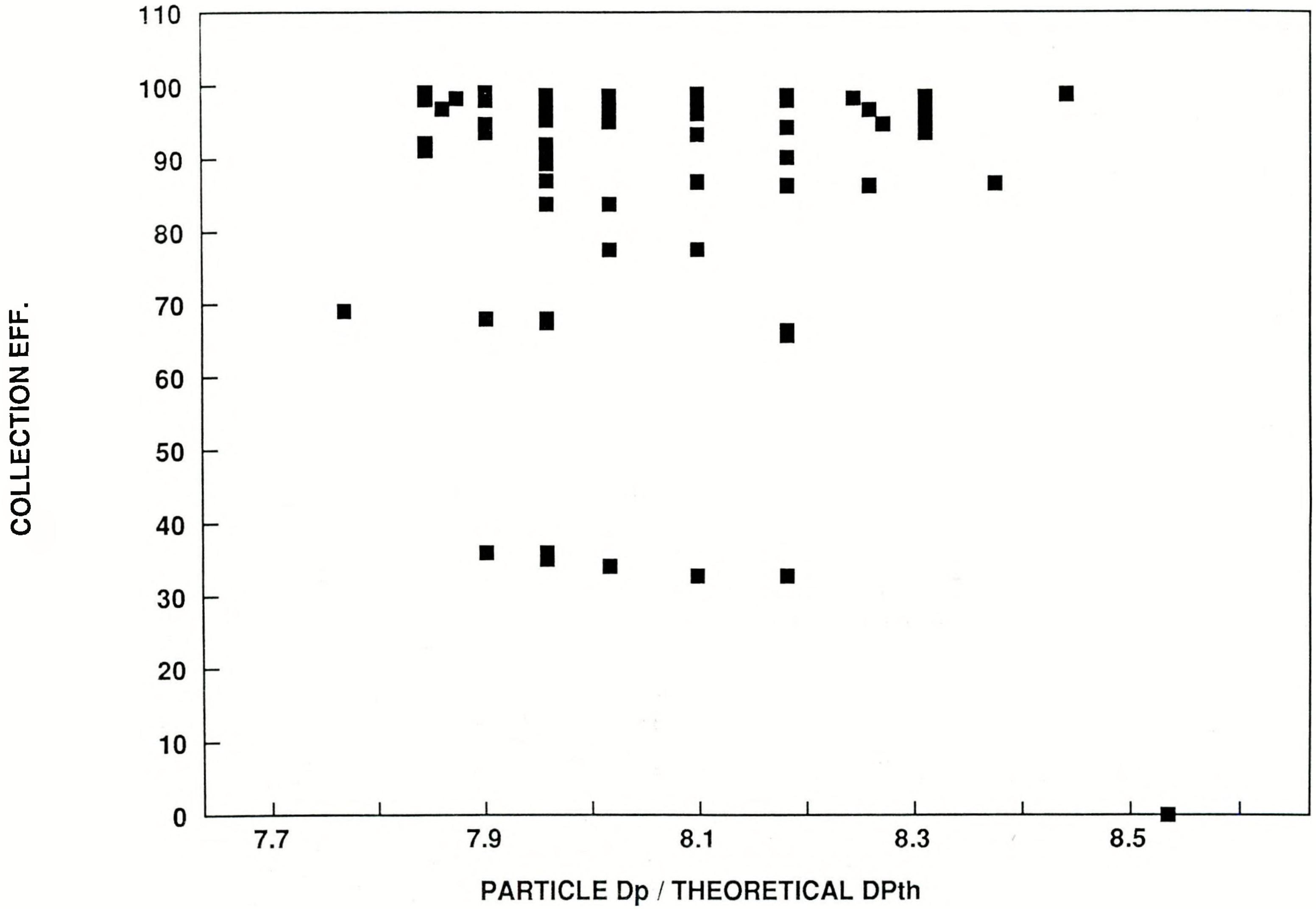

Figure 6.11b Collection Efficiency vs. Dp/DPth
Using Only Qualified Data

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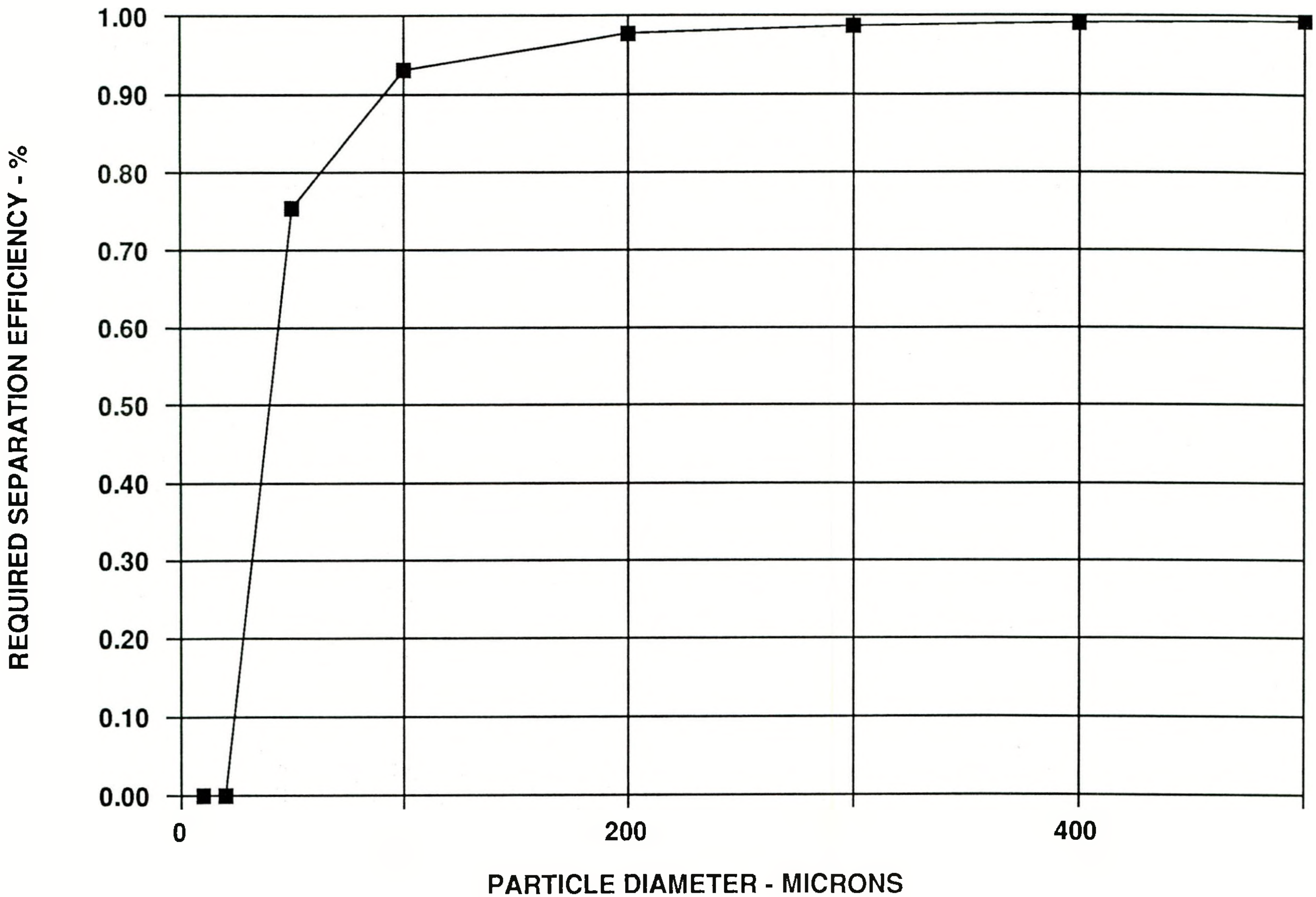

Figure 6.12 Required Separation Efficiency
(50%/5% DB, 75 lb/ft^3)

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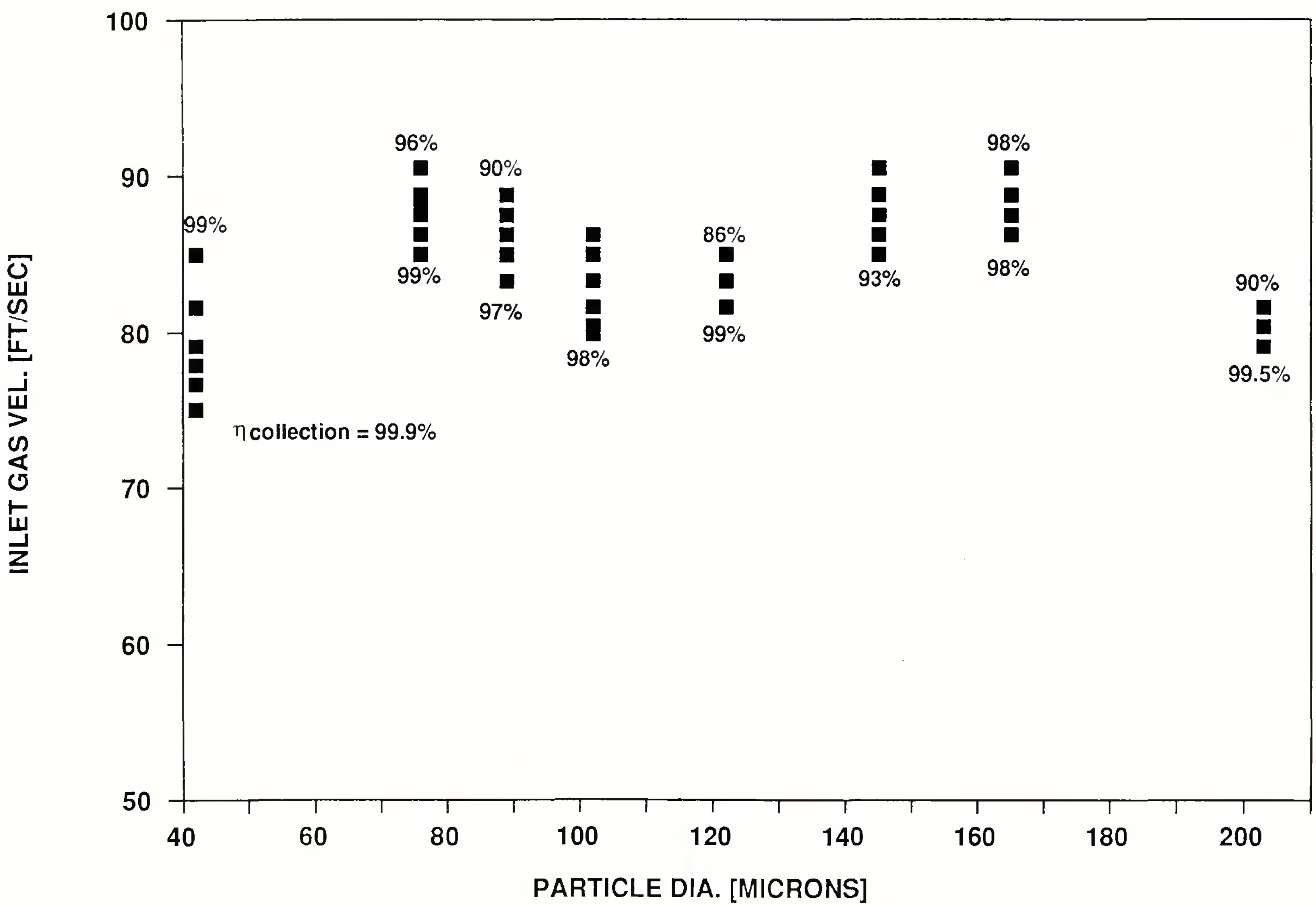

Figure 6.13 Inlet Gas Velocity vs. Particle Diameter
IRIS Cold Test Experiment

IRIS Dryer Terminal Velocity Measurements (Constraint No. 3)

The measurements that could be made to verify the terminal velocity constraints were limited by the availability of only 200-micron-size particles and the need to maintain a relatively high gas inlet velocity. Gas velocities between 70 and 90 ft/sec were used through the testing in order to ensure the correct particle loading (L_p). At low particle loadings and throughout the range of alumina particle sizes (30 to 200 microns) tested, there was no evidence of particle terminal velocity limitations. That is, the gas stream at these velocities could completely fluidize the particles that were injected into the stream; particles did not "rain" down through the dryer and collect at the bottom. Thus, it is concluded that with an inlet gas velocity of 70 to 90 ft/sec and a 200-micron (or smaller) particle, the terminal velocity constraint is still not exceeded. This observation falls within the computer model's prediction that a 70- to 90-ft/sec gas inlet velocity should be able to fluidize at least a 250-micron-size particle, and most likely a 300-micron-size particle. These cold tests thus have verified that the terminal velocity constraint boundary is no smaller than what is shown in Figure 6.4.

It should be noted that the terminal velocity limits on particle fluidization are not to be confused with the dryer's saturation limit, the time for which was dutifully recorded as the "flow" time for each experiment. The saturation limit is the point at which the dryer's particle intake flow rate exceeds the dryer's particle outlet flow rate and is thus a function of particle flow rate and gas velocity, and not a function of fluidization velocity.

6.3 INDIVIDUAL TEST RUN RESULTS

In addition to combining all of the measured tests into the same figure (see Figures 6.6a and 6.6b), the measured data have been plotted for individual tests conducted for the various alumina and clay particle sizes and the various lengths of the recirculation spouting tube heights. Figures 6.14 through 6.26 present these data for additional reference. These figures are in two distinct presentation formats: Figures 6.14 through 6.21 display each particle size with the recirculation spout tube heights changing from 1 in. to 9 in. Figures 6.22 through 6.26 display each recirculation spouting tube height with the particle sizes varying. Several observations may be discerned from these graphs that have an influence on future IRIS-type dryer designs. These observations are:

1. A longer recirculation spouting tube length reduces the change in the vortex pressure drop as the dryer particle loading increases.

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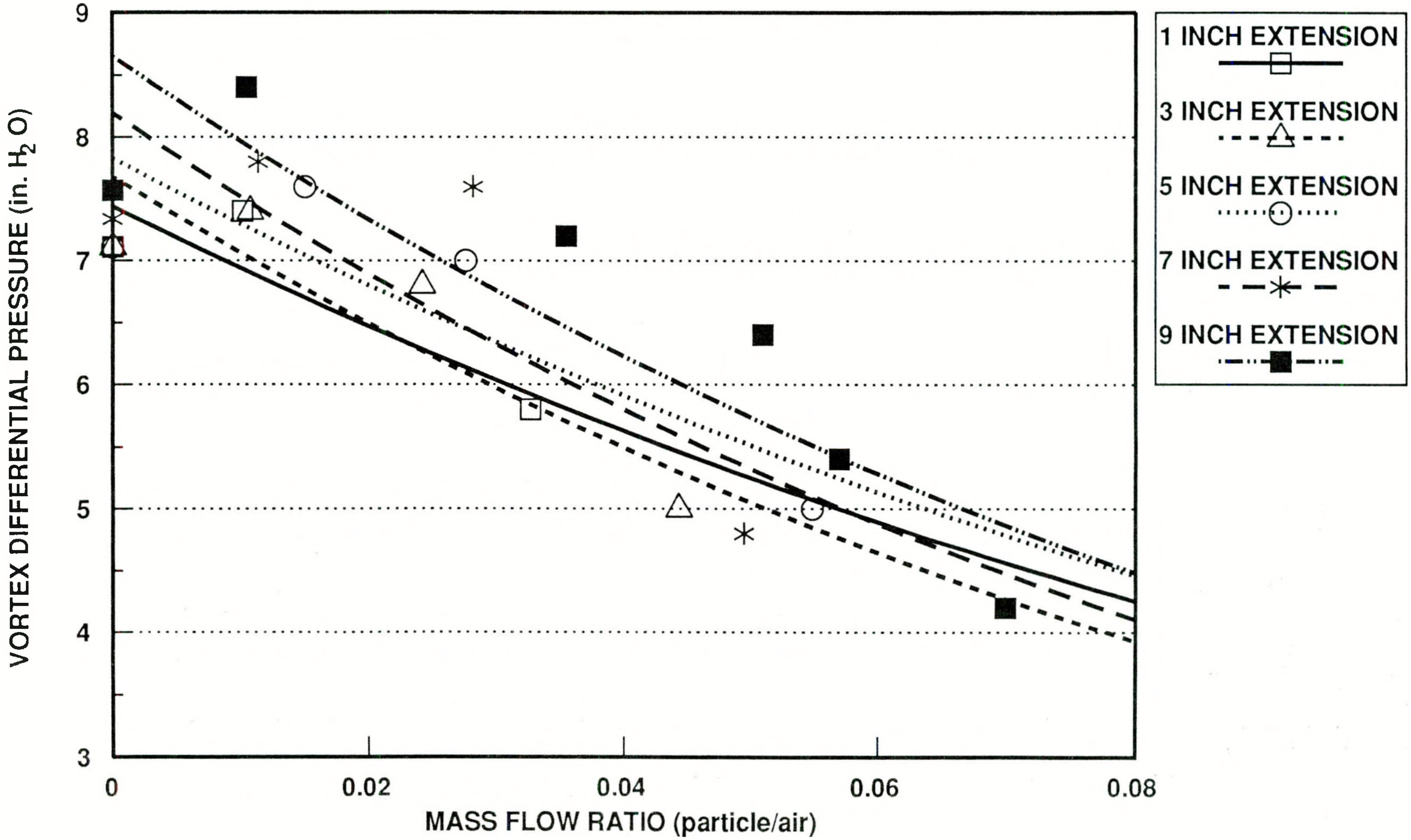

Figure 6.14 Cold Flow Test Data with Varied Recirculation Tube
Aluminum Oxide at 76 microns

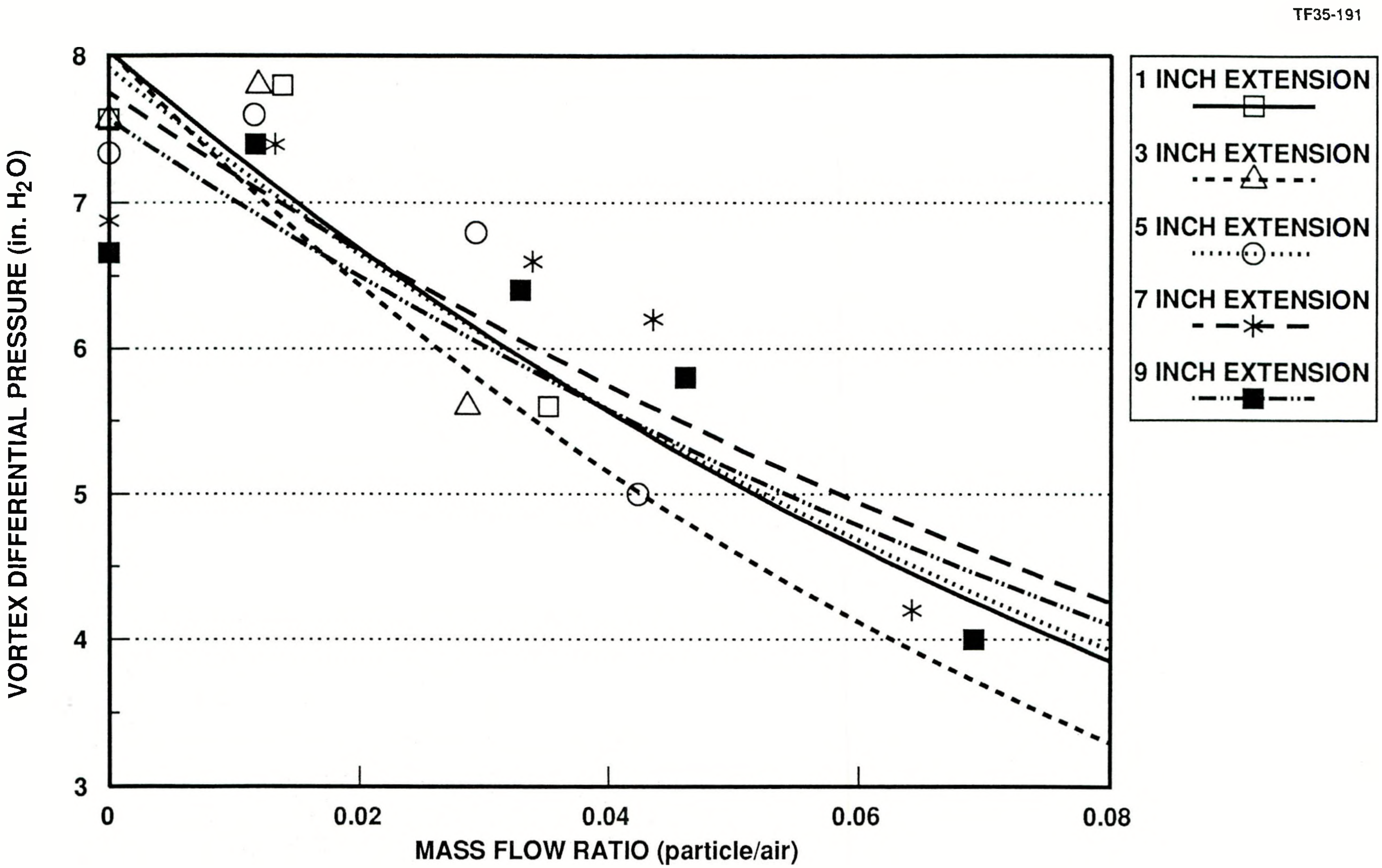

Figure 6.15 Cold Flow Test Data with Varied Recirculation Tube Aluminum Oxide at 89 microns

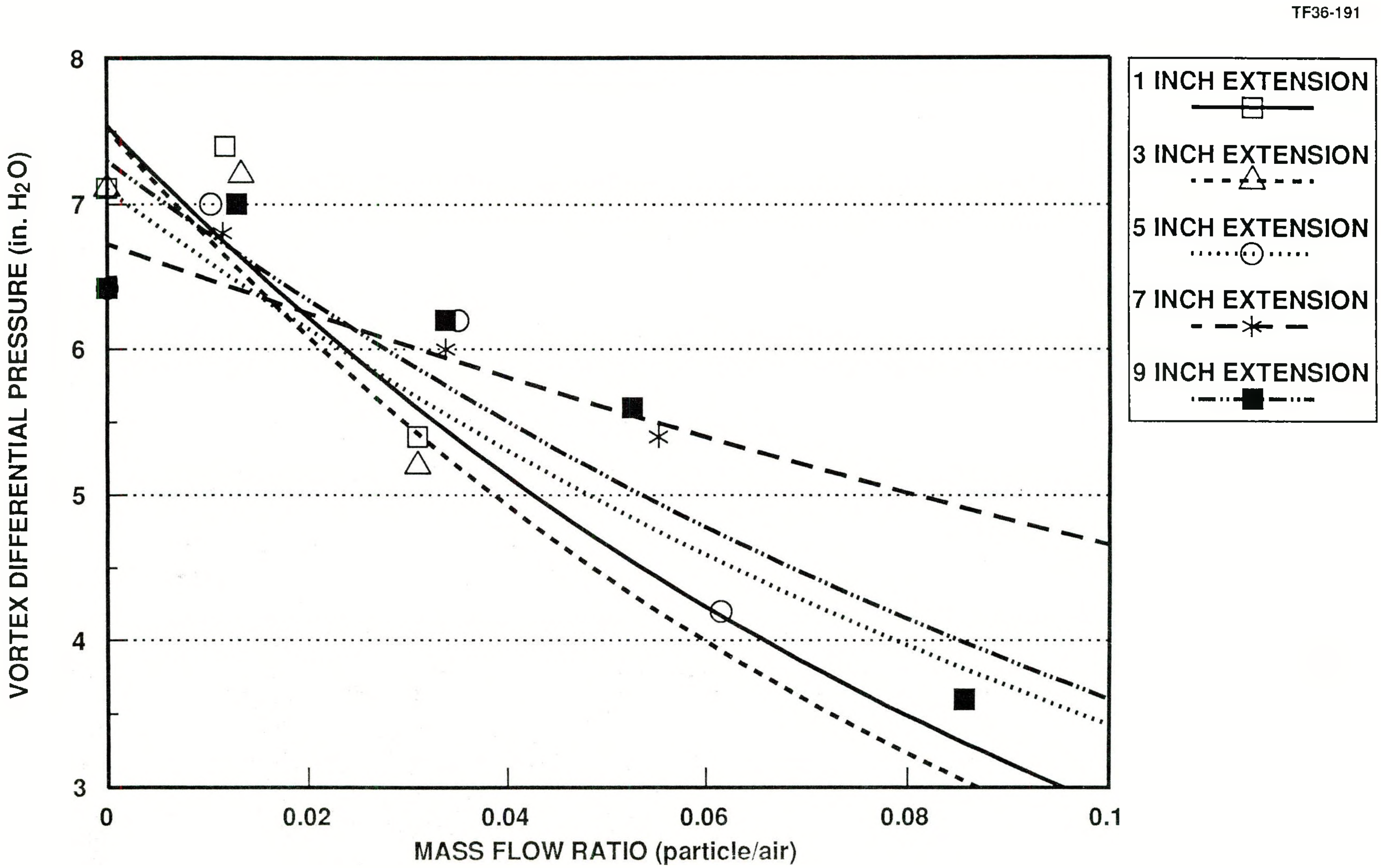

Figure 6.16 Cold Flow Test Data with Varied Recirculation Tube Aluminum Oxide at 102 microns

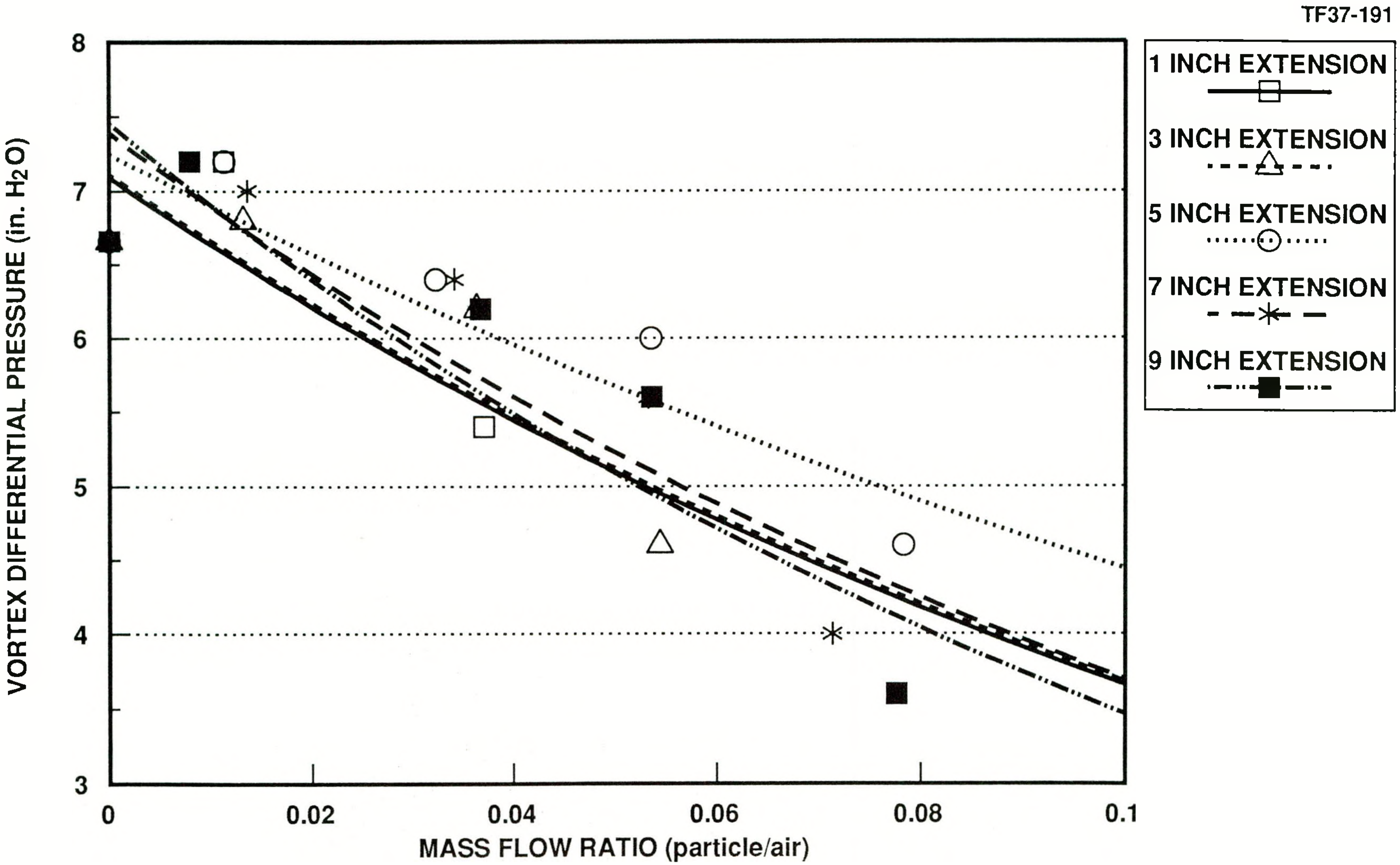

Figure 6.17 Cold Flow Test Data with Varied Recirculation Tube
Aluminum Oxide at 122 microns

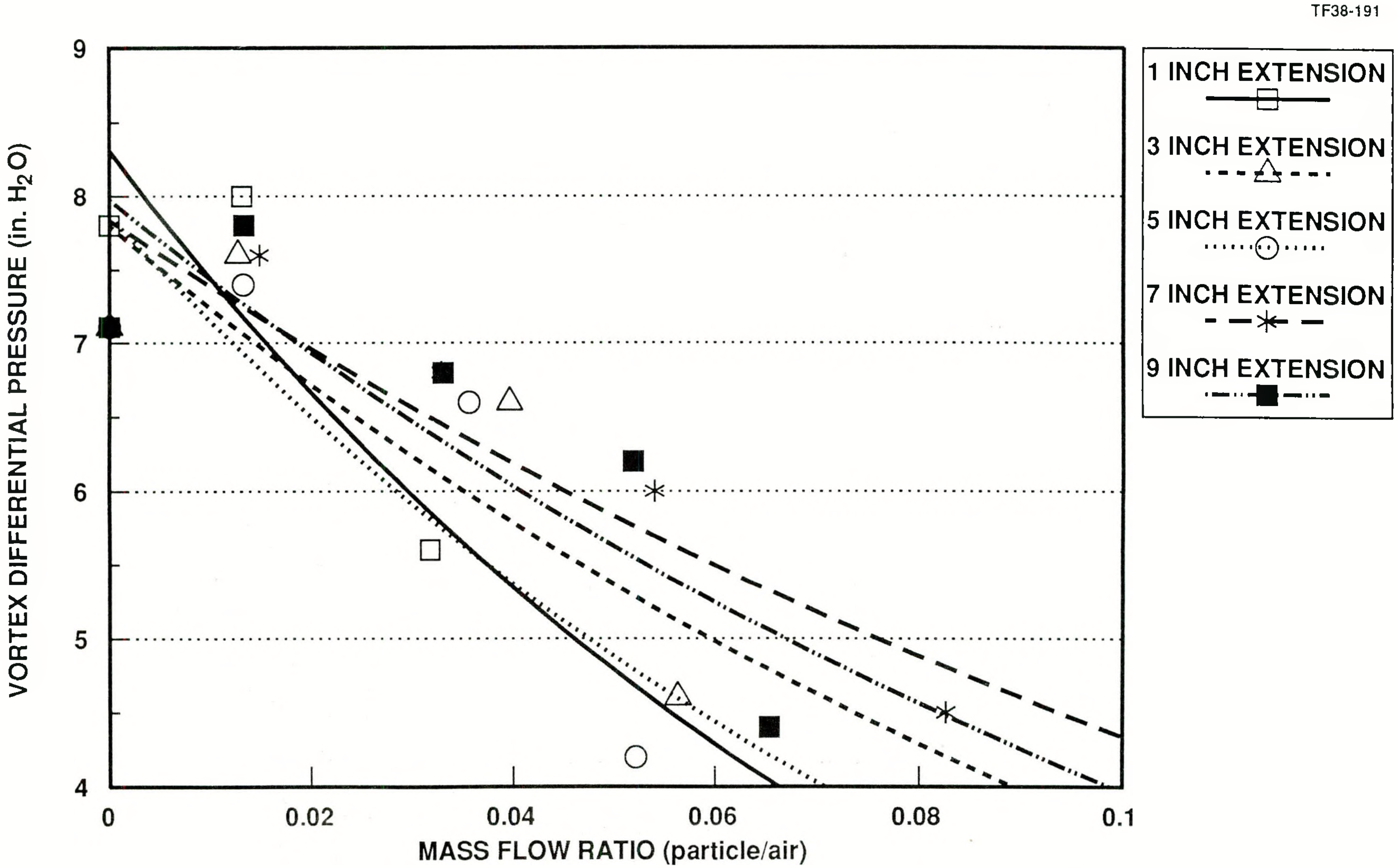

Figure 6.18 Cold Flow Test Data with Varied Recirculation Tube
Aluminum Oxide at 145 microns

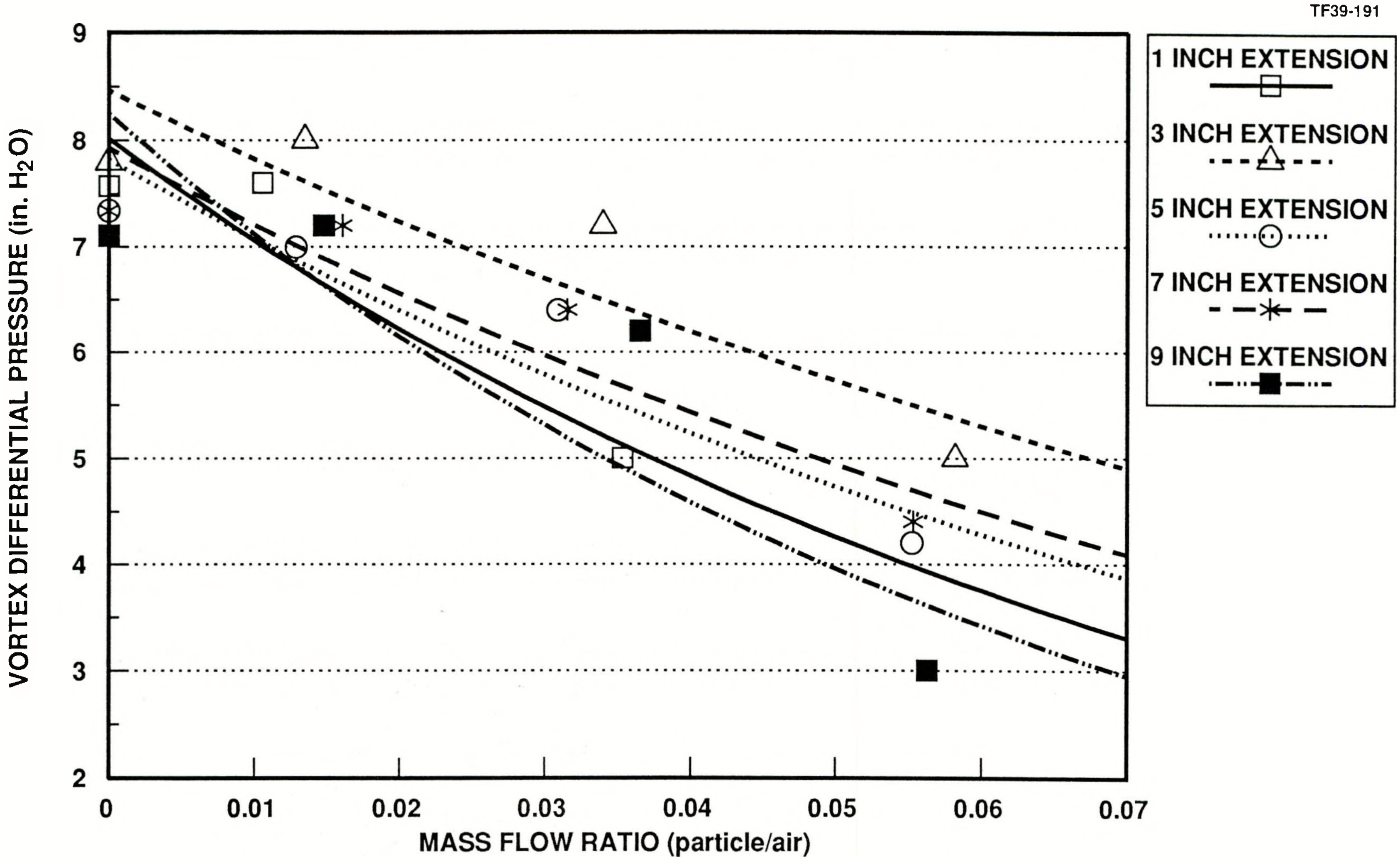

Figure 6.19 Cold Flow Test Data with Varied Recirculation Tube Aluminum Oxide at 165 microns

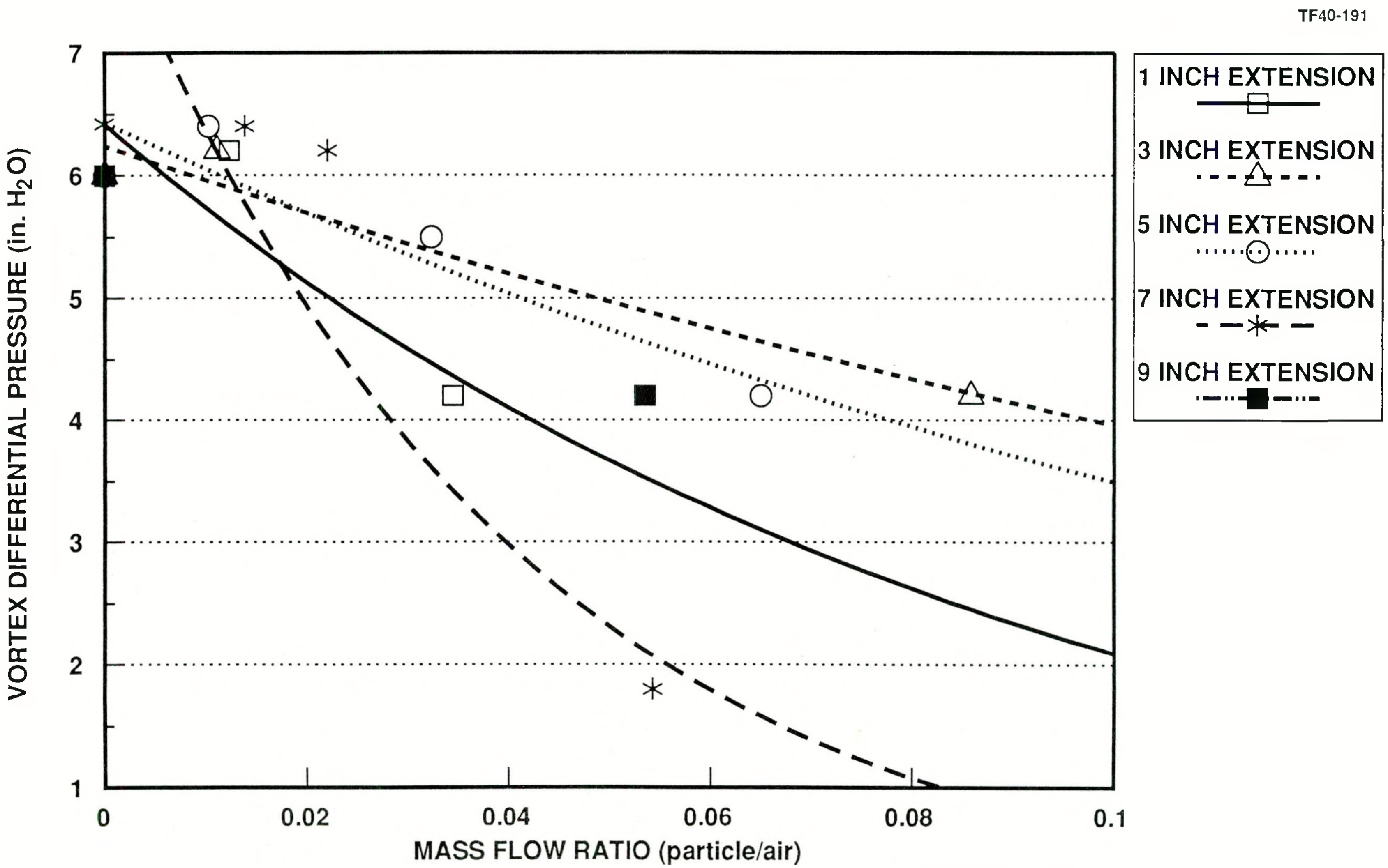

Figure 6.20 Cold Flow Test Data with Varied Recirculation Tube
Aluminum Oxide at 203 microns

TF41-191

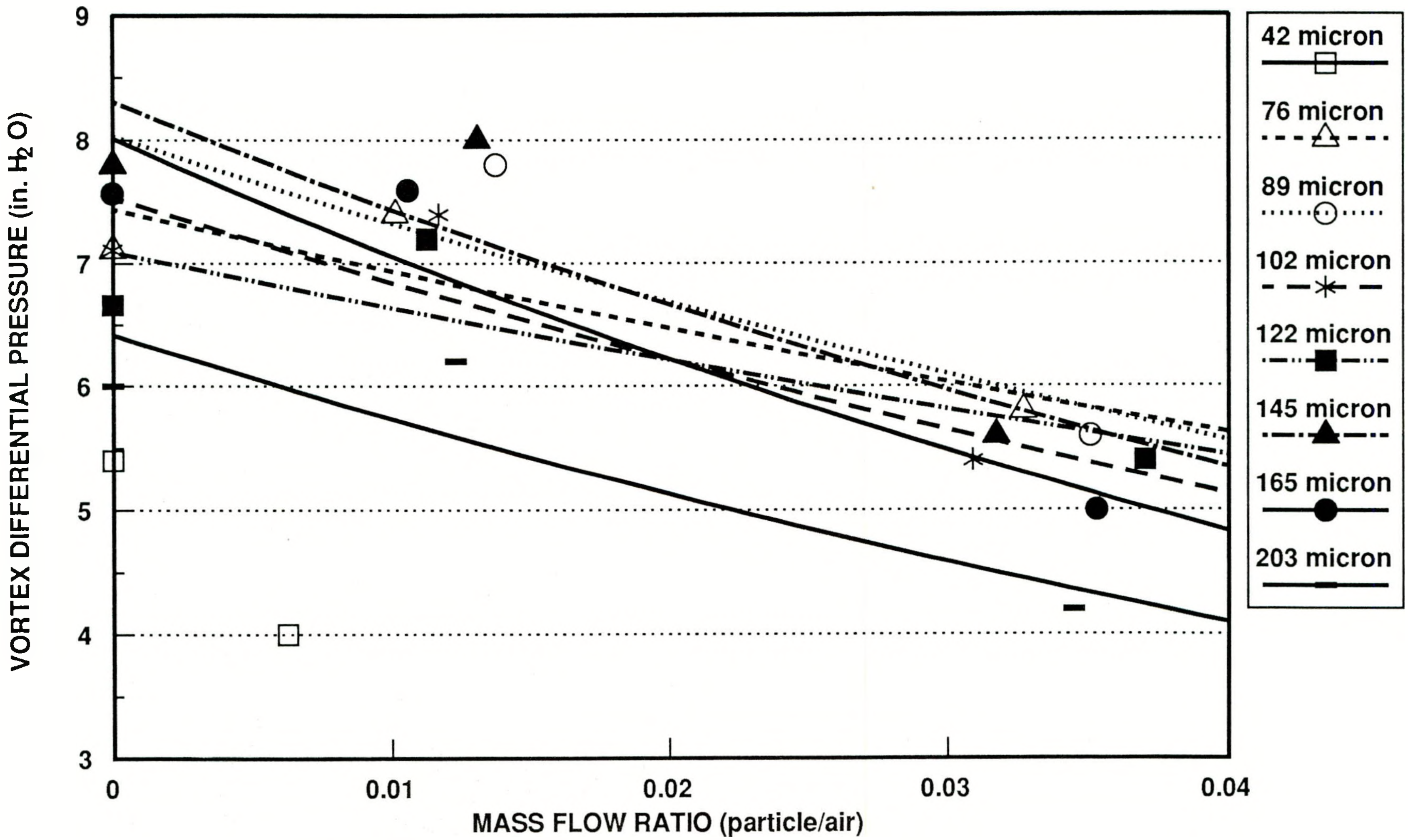

Figure 6.21 Cold Flow Test Data with Varied Particle Diameters
Recirculation Tube Length at 1 Inch

TF42-191

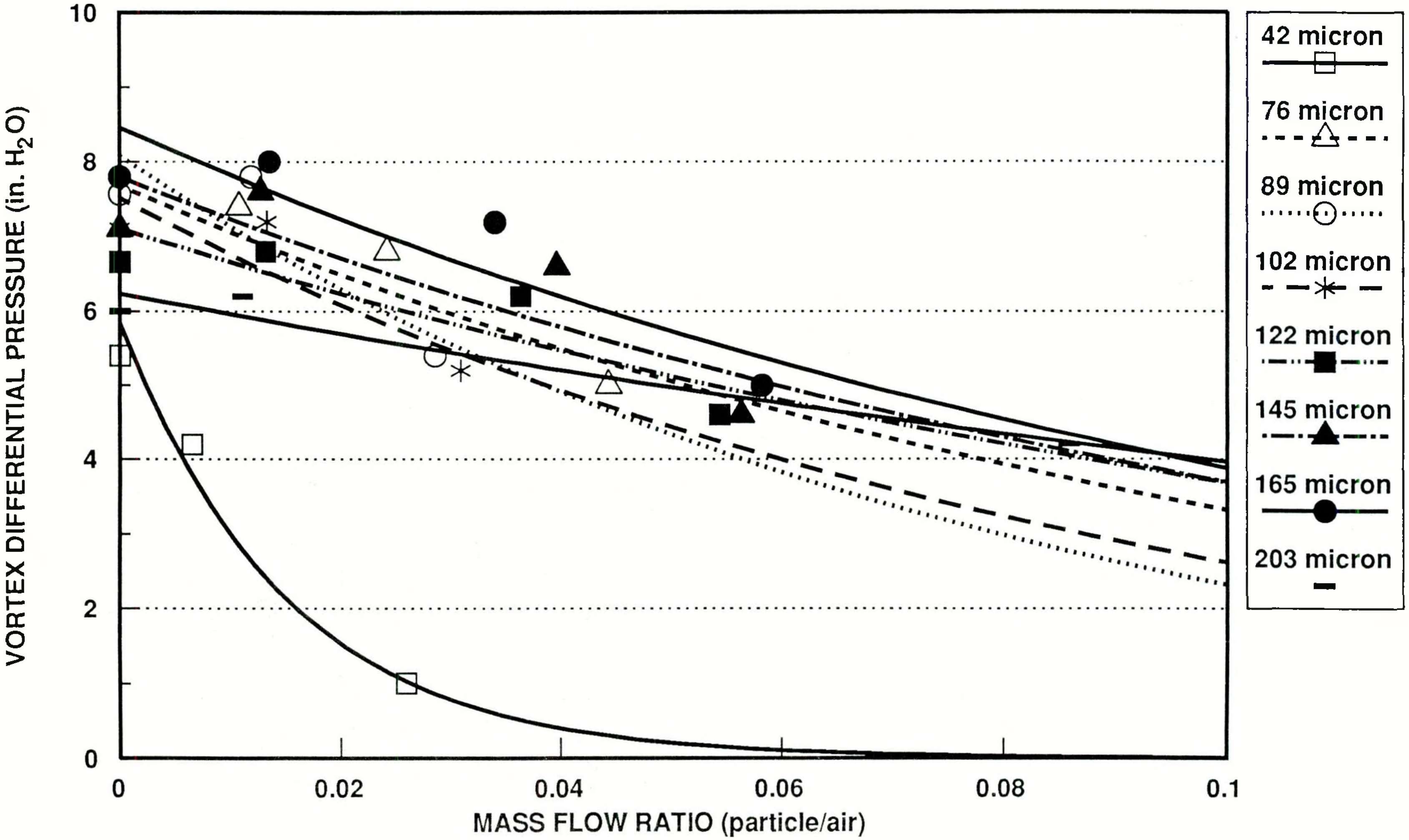

Figure 6.22 Cold Flow Test Data with Varied Particle Diameters
Recirculation Tube Length at 3 Inches

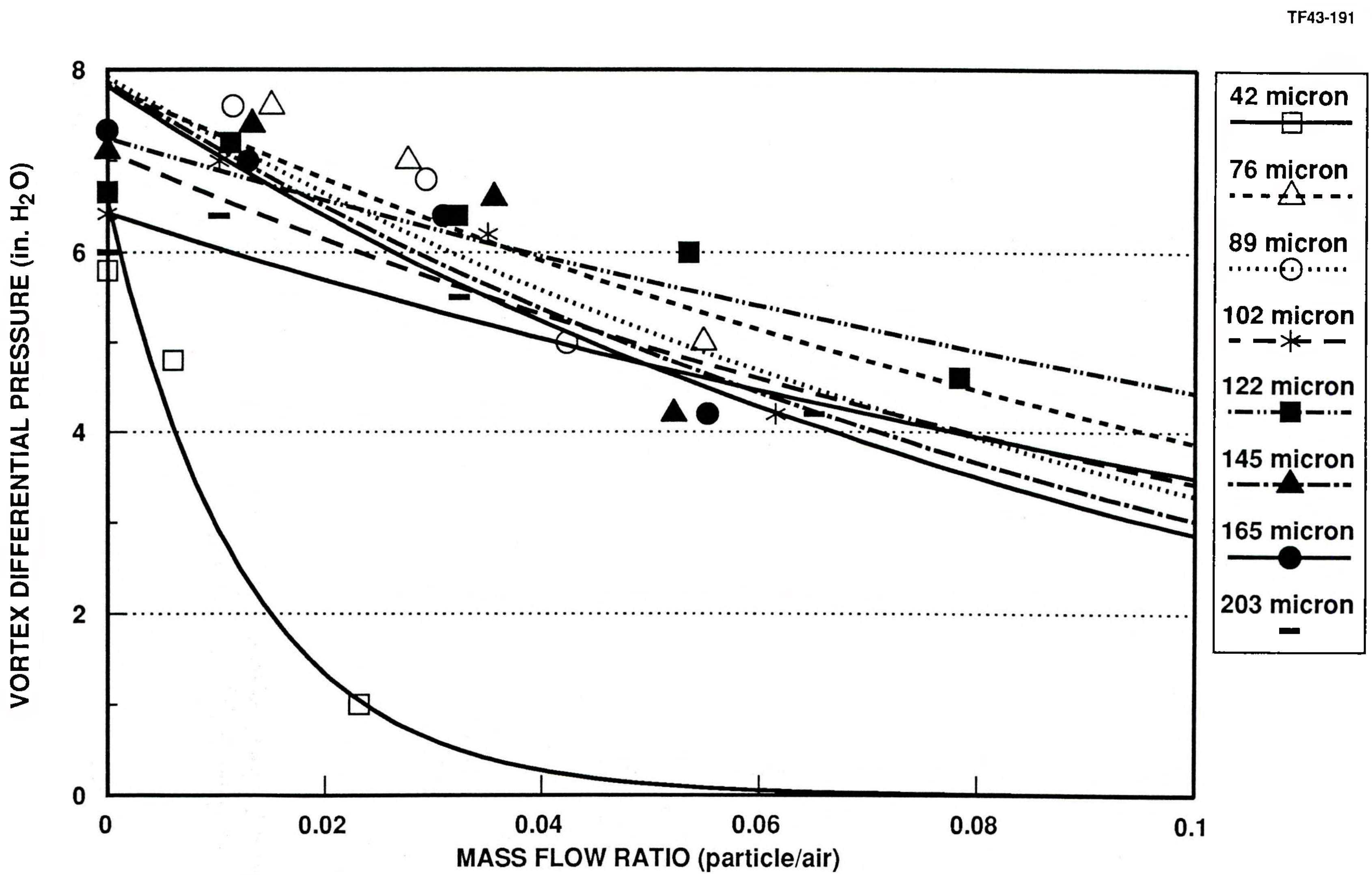

Figure 6.23 Cold Flow Test Data with Varied Particle Diameters
Recirculation Tube Length at 5 Inches

TF44-191

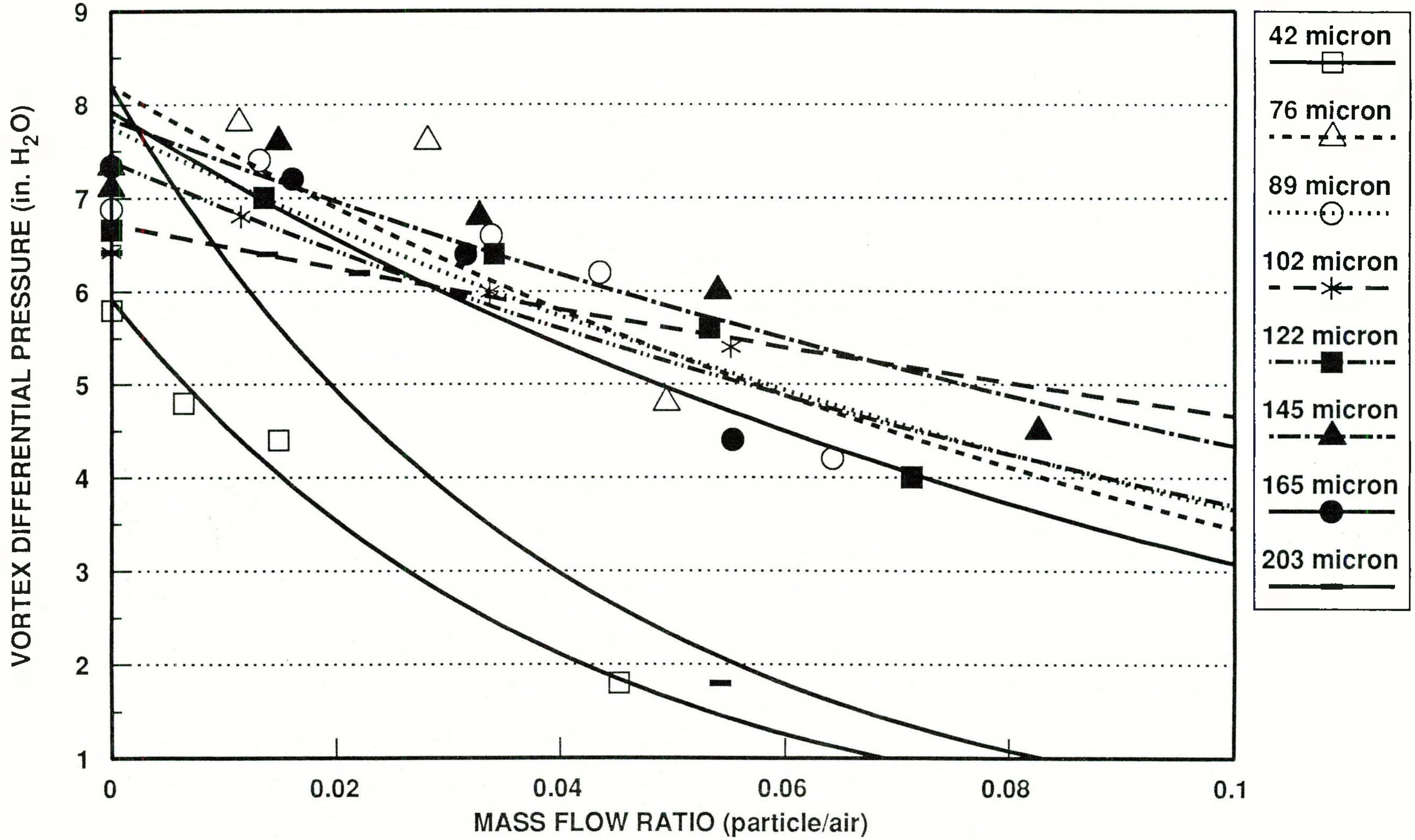

Figure 6.24 Cold Flow Test Data with Varied Particle Diameters
Recirculation Tube Length at 7 Inches

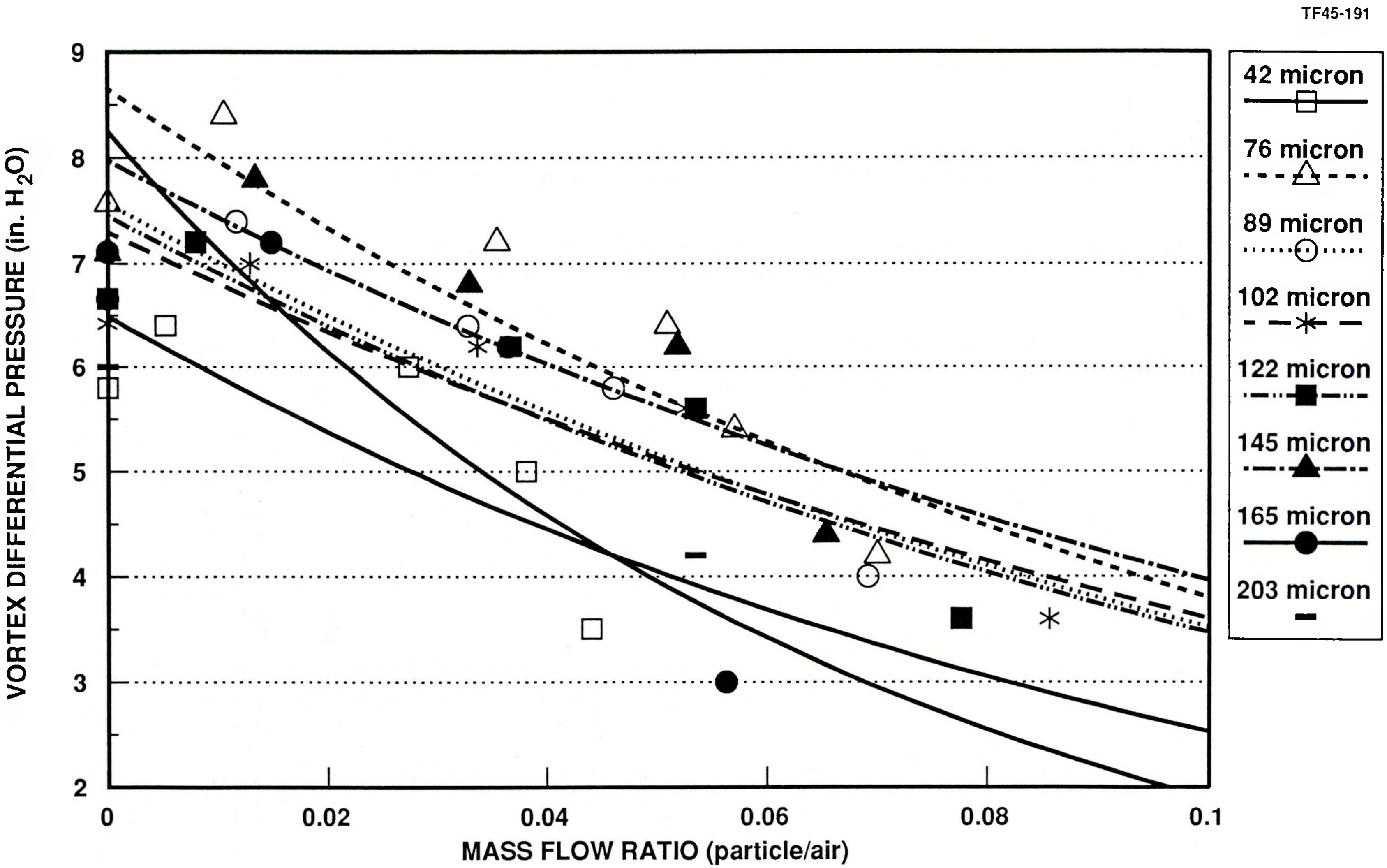

Figure 6.25 Cold Flow Test Data with Varied Particle Diameters
Recirculation Tube Length at 9 Inches

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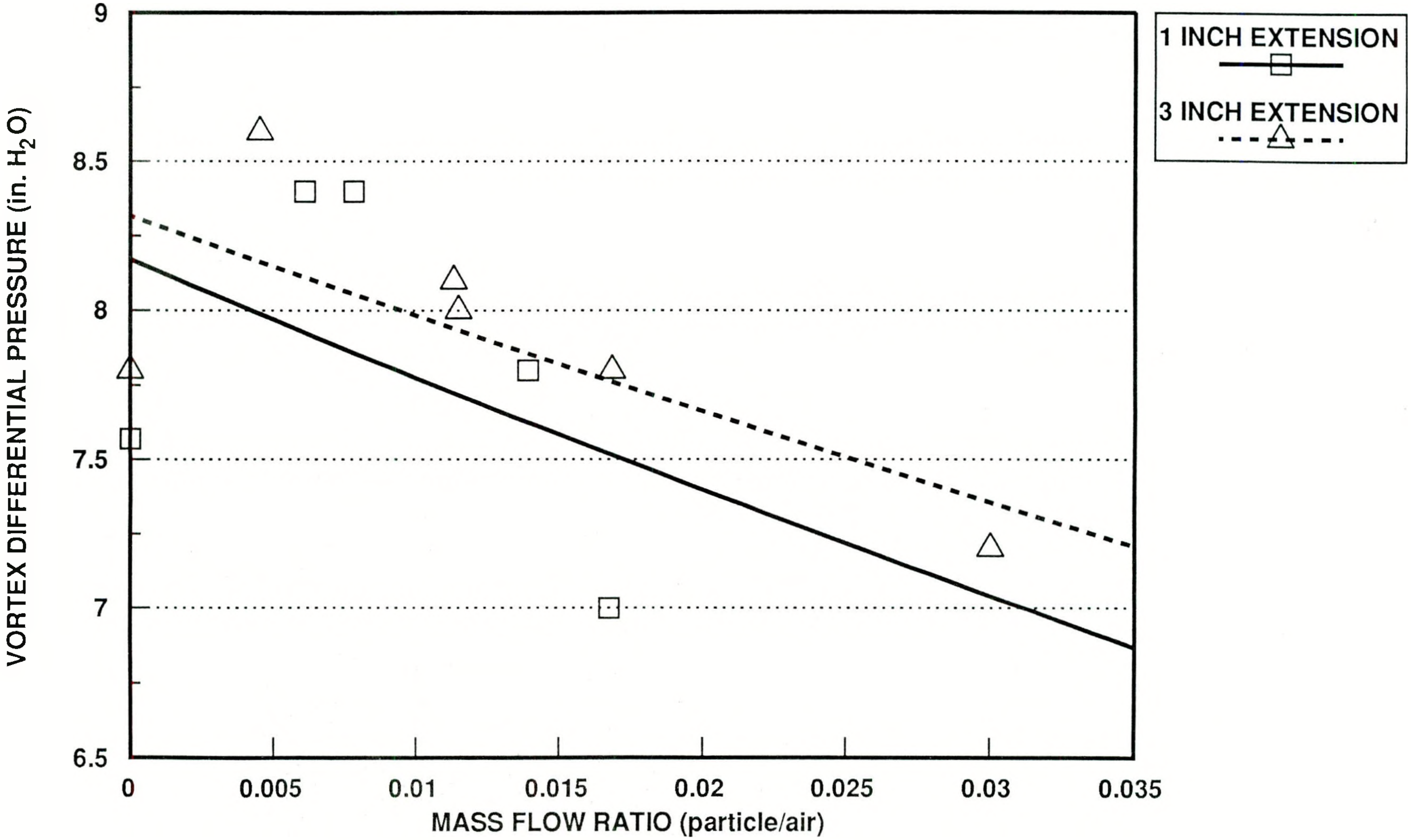

Figure 6.26 Cold Flow Test Data with Varied Recirculation Tube
Powdered Clay (100 microns)

2. For a given recirculating spouting tube length the smaller diameter particle reduces the change in the vortex pressure drop as the dryer particle loading increases. This would seem to imply that in the hot test, a large (water-laden) particle may not adversely affect the vortex pressure drop for it will be continually reduced in size until the particle is dry and is then able to leave the drying chamber.

3. Based on the powdered (dry) clay measurements and dairy creamer observations in the cold test apparatus, it would appear that lighter (less dense) materials will not adversely affect the vortex flow field pressure drop as severely as the higher density materials as the dryer loading is increased. This is also particularly relevant if one considers the effect that water loss has on the particle's density. Depending upon whether the feedstock particle's density is initially lighter than or heavier than the water it holds, the dried particle's combined density can increase or decrease, respectively, as drying proceeds. This effect is displayed in Figure 6.27. As (dry base) wetness (w) decreases as drying continues, a wet food particle feedstock, for example, will become less dense as drying continues. A clay particle (whose specific gravity is greater than one) will become more dense and may tend to disrupt the vortex flow field pressure drop.

4. Changing the length of the recirculation spouting tube height can be an effective means of instantaneously controlling the dryer's recirculation ratio or particle collection efficiency. A patent disclosure has been filed with DOE for this control mechanism.

6.4 STEAM ATMOSPHERE IRIS-DRYER HOT TESTS

6.4.1 Steam Atmosphere Laboratory Equipment and Operating Characteristics

The steam atmosphere drying hot test apparatus was designed and assembled specifically for the hot testing of a bench-scale model of an IRIS steam atmosphere dryer. The purpose of these tests was:

- To dry a slurry mixture of feedstock and water to its constituent powder and observe the effects of steam drying on the powder properties.

- To characterize the heat transfer capability of the IRIS dryer, measuring heat transfer coefficients and comparing this performance with Tecogen's computer heat transfer model.

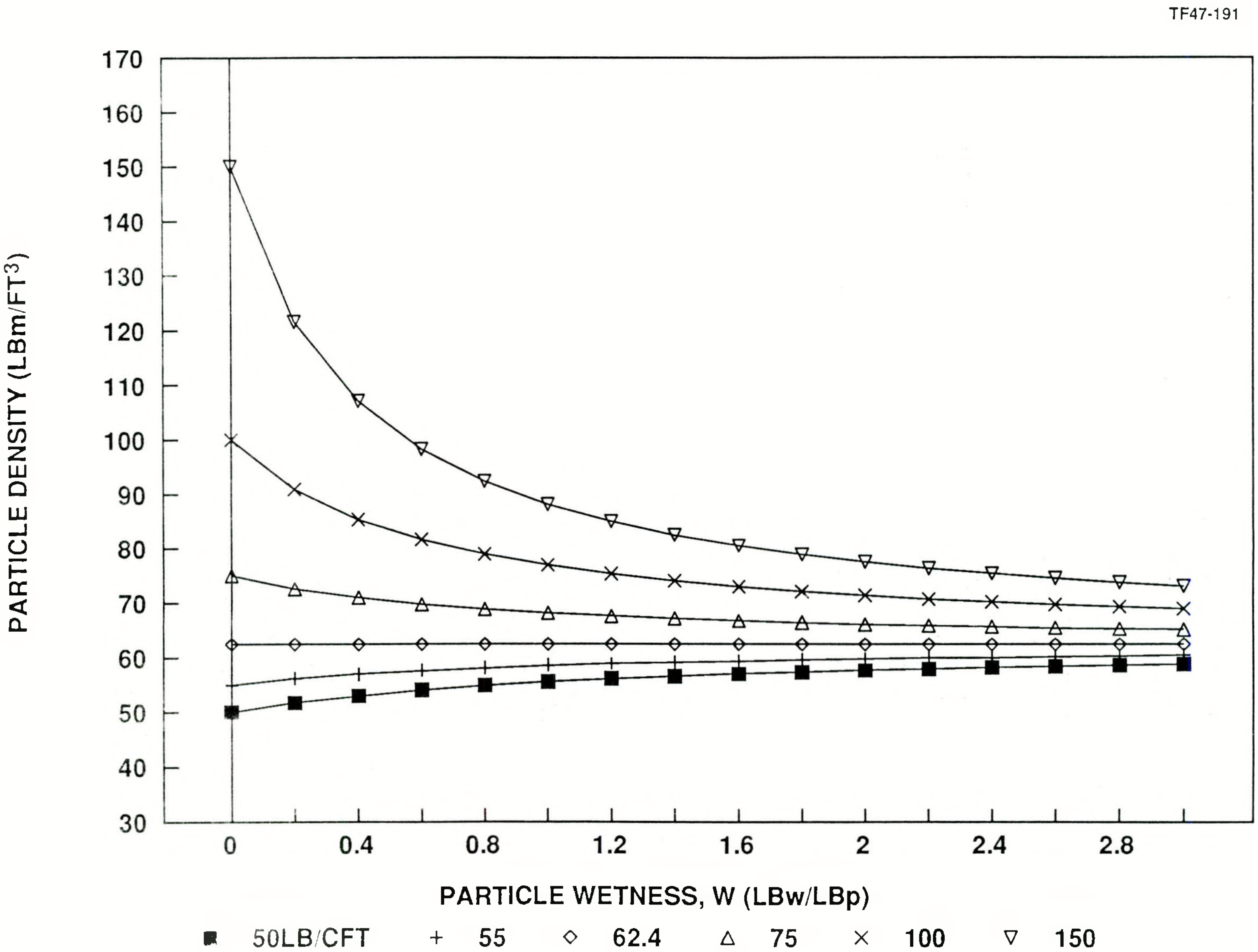

Figure 6.27 Effect of Wetness Changes on Particle Density
Constant Particle Densities Shown

The test apparatus consisted of the following principal components:

1	50-in. tall x 6-in. dia., stainless steel IRIS-type, steam atmosphere dryer (see Figure 6.28).
1	450 lb/hr, 125-psig Tecogen facility steam boiler required to produce saturated steam (see Figure 6.29).
1 ea	Steam pressure regulator, throttling valve, and steam flow orifices for producing and measuring superheated steam (typically at 290°F, 14.7 psia) for use with steam dryer (see Figure 6.29).
1	Steam cyclone particle separator; 12-in. dia. x 4-ft long (see Figure 6.30).
3	Foxboro steam pressure transducers, calibrated for 0 to 10 in. wc; these were used to monitor the vortex pressure drop and the two flow orifice pressure drops; that is, the main steam inlet flow and the steam atomizer injection flow.
1	Peristaltic pump with a variable speed motor; used to pump slurry to the injector.
1	Steam nozzle atomizer for pumpable slurry feedstocks (see Figure 6.31).
3	Sets of electric heater wrap each at 600 watts; used to increase and/or maintain the dryer wall temperatures.
1	140°F hot water heater to allow preheating of the boiler make-up water into the facility boiler.
1	Roth condensate return station (i.e., boiler feedpump, tank, and level controls) to provide pressurized water (125 psig) to the facility boiler.
1	Weighing scale with digital weight readout.
13	Temperature thermocouple readings, including various wall temperatures, but most importantly, steam temperature into and out of the steam dryer and steam temperature out of the cyclone particle separator.
3	Categories of feedstock were tested – clay powder, nondairy coffee creamer, and maltodextrin-100 (a food sweetener).

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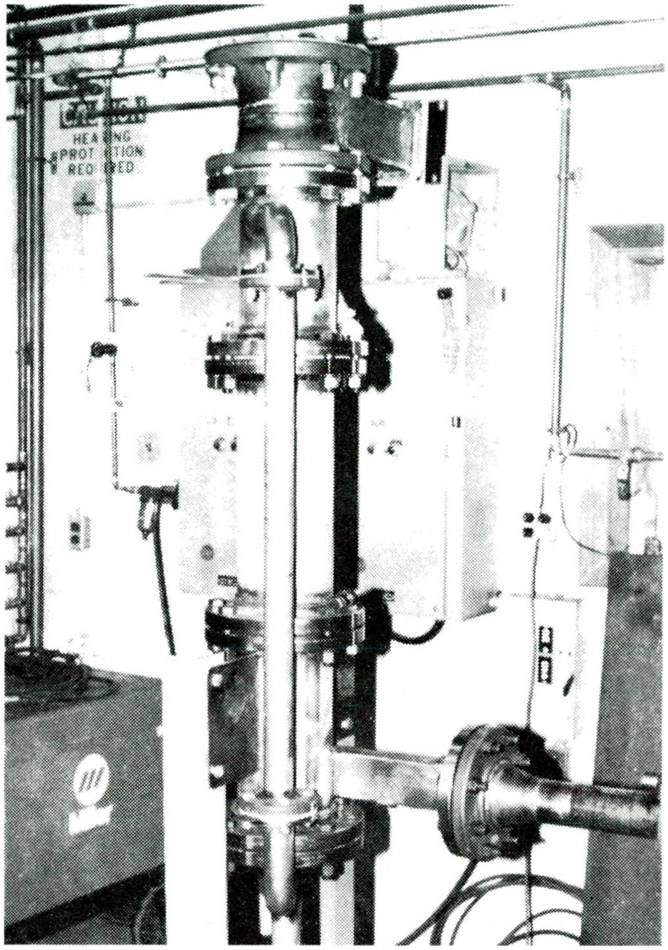


Figure 6.28 Bench-Scale IRIS Dryer
Used in Tecogen's
Hot Testing

TF49-191

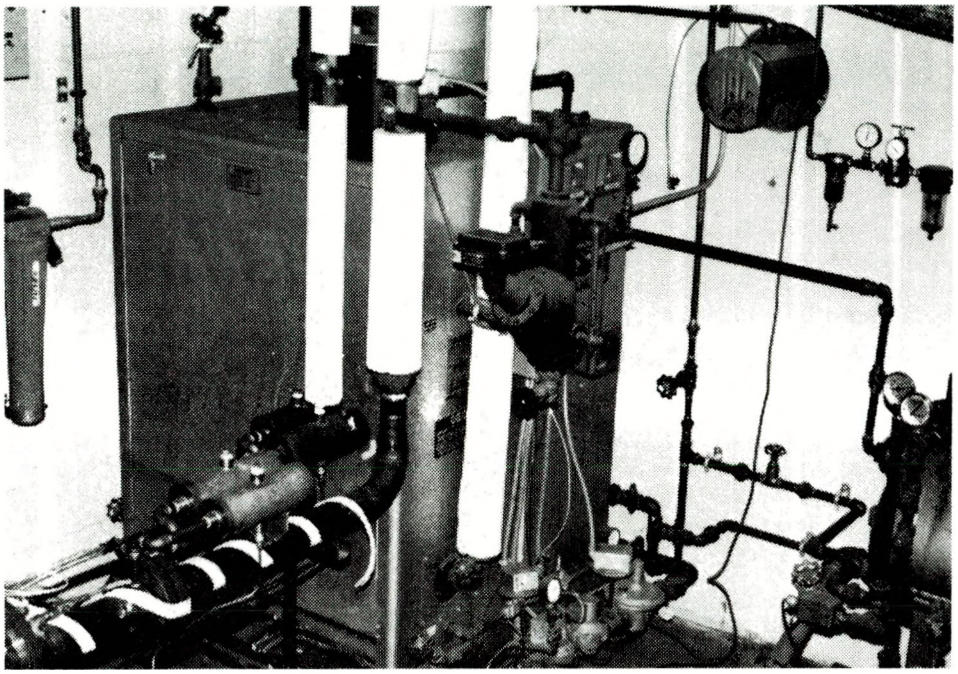


Figure 6.29 Facility Steam Boiler with IRIS Hot Test Steam Loop Control Valves and Steam Orifice Metering

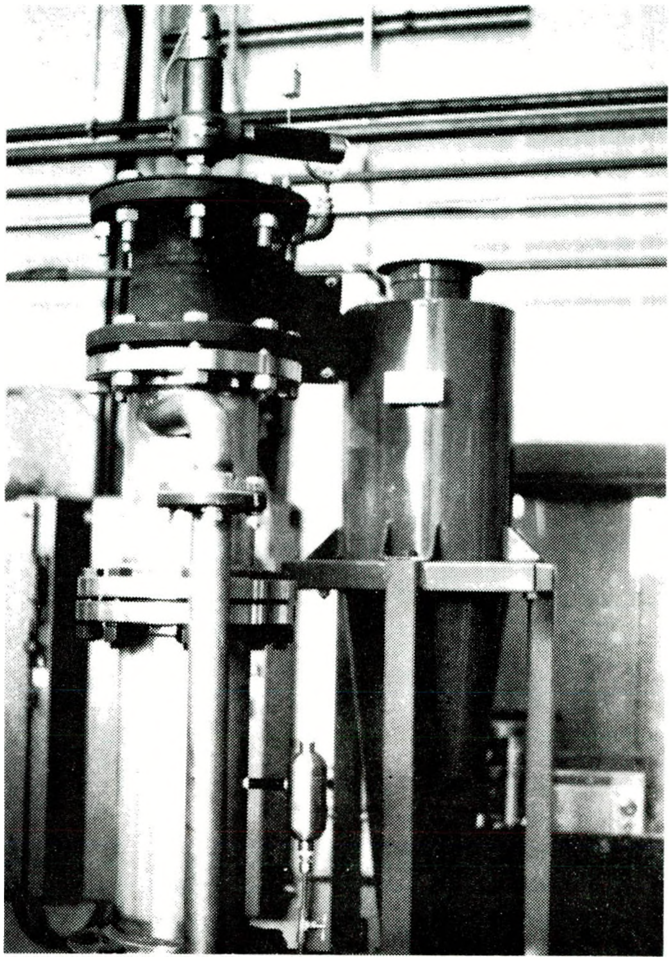


Figure 6.30 Cyclone Particle Separator Shown Installed Next to the Rear of the Tecogen IRIS Steam Dryer Test Model

TF8-191

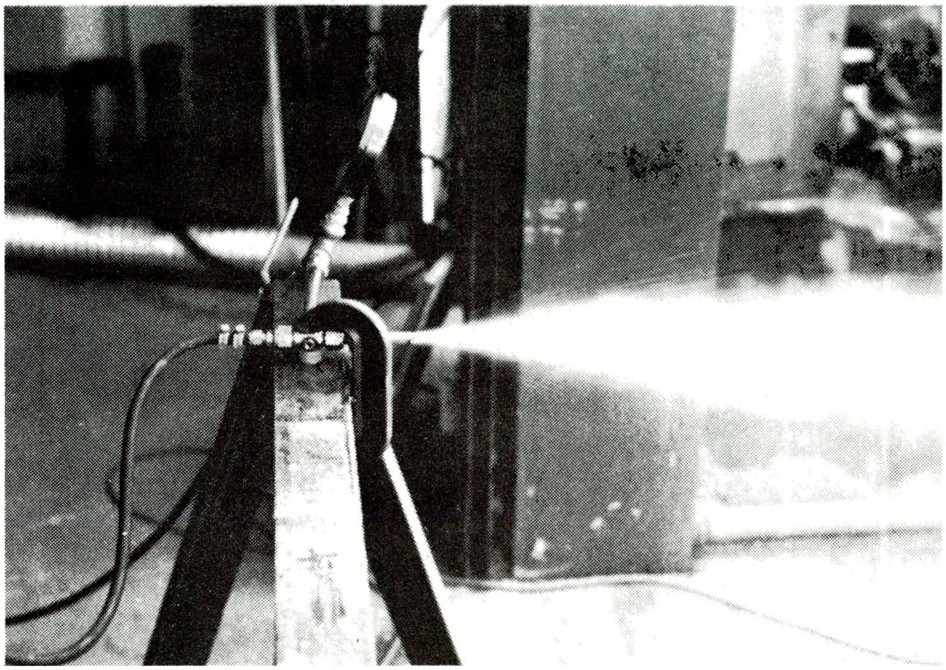


Figure 6.31 One of Two Slurry Steam Nozzle Atomizers Used in Tecogen's Hot Testing

A Piping and Instrumentation (P&I) diagram of the steam atmosphere dryer laboratory facility is displayed in Figure 6.32. As was the case for the cold test apparatus, the steam atmosphere dryer system was designed and assembled as a permanent testing laboratory. It will enable a continuation of steam atmosphere drying tests. Photograph of the bench-scale IRIS-type and the entire test facility are shown in Figures 6.33a and 6.33b.

The steam atmosphere drying facility generates high-pressure saturated steam at 125 psig which can be immediately flashed to atmospheric pressure and hence to a superheated temperature. Typically, 375 to 400 lb/hr of steam at 285 to 290°F could be maintained for use by the bench-scale IRIS-type steam dryer. There was some difficulty in maintaining constant maximum flow rate through the dryer. Approximately 5 to 8 percent of this steam was used in a specially designed steam atomizer nozzle that would atomize the liquid feedstock slurry that was to be tested. Both the main dryer steam flow rate and the steam injector flow rate were individually measured using two ASME-designed steam orifices. The pressure drops were measured by a Foxboro-series pressure transducer.

The IRIS steam atmosphere dryer used the steam to vaporize the water-laden atomized feedstock. Liquid (slurry) drains were installed into the bottom of the steam dryer. Drainage from the steam dryer was collected in a 100-ml beaker, and its contents were recorded after each test run. This drainage represented slurry that could not be dried (for whatever reason) by the steam dryer and was not carried out of the dryer in the form of wet steam and dried particles. The dried feedstock particles leaving the IRIS dryer were separated from the transport steam by the cyclone particle separator. The separator included a collection box at its bottom where dried feedstock collected and could be removed. Three heater tapes (each at 600 watts) were installed on the walls of the IRIS dryer and cyclone. In all of the tests the cyclone heater remained in operation, while the IRIS dryer heater tape was in use in only a limited number of the tests. The test numbers marked "H" or "NH" indicate if the wall heater tapes were or were not in operation, respectively.

The wall temperatures of the IRIS and cyclone were monitored very closely during each hot test. In all of the testing temperatures were found to be hotter than 212°F, and usually the walls were at a temperature of 245°F <u>without</u> requiring wall heaters. The wall temperatures of the external recirculation line were typically 240 to 245°F and clearly indicated that recirculation was present in this line. The only exception to these high temperatures was found on the walls near where the slurry was injected. Those wall temperatures were typically 212°F.

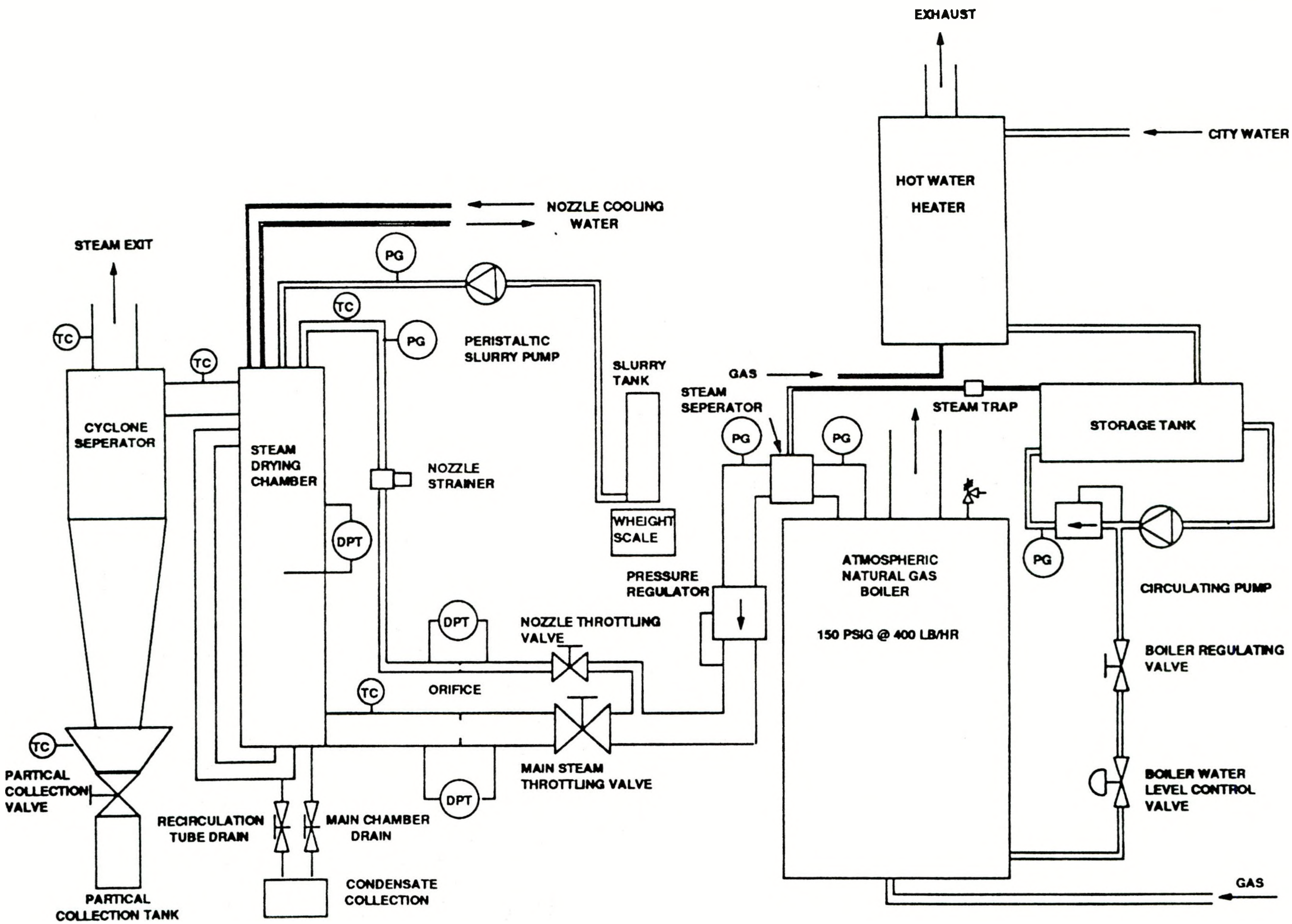
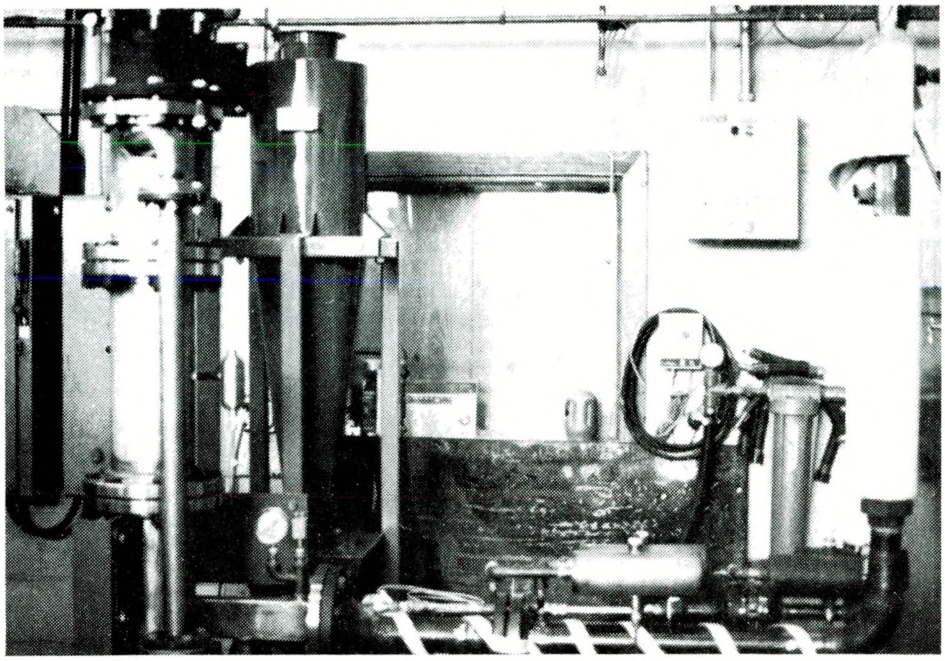
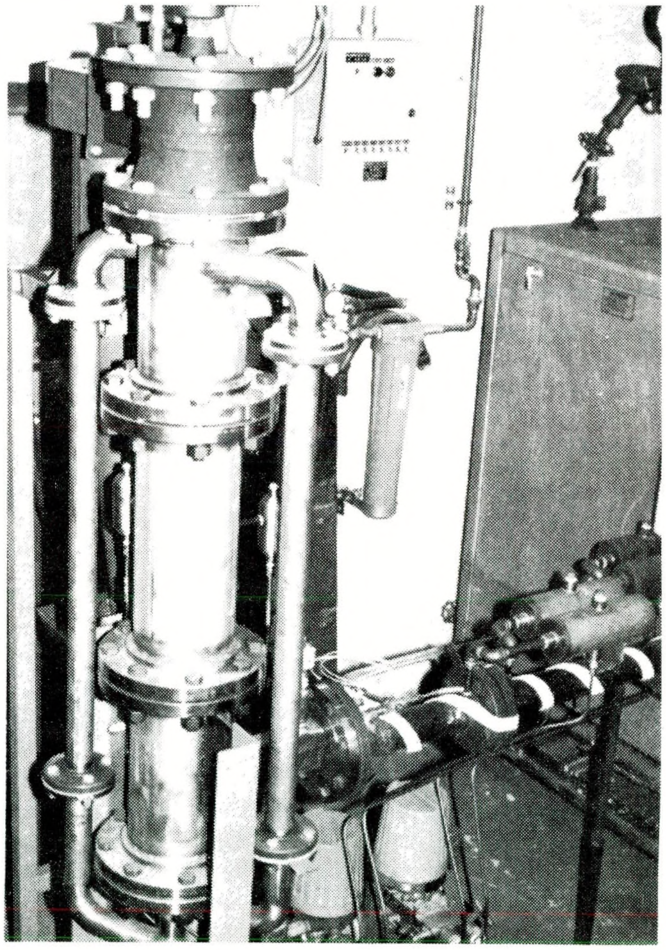


Figure 6.32 Steam Drying Hot Test System



(a)



(b)

Figure 6.33 Steam Hot Test Apparatus

The entire steam atmosphere apparatus was insulated with 4-in. thick Fiberfrax insulation. This insulation was adequate to maintain a minimum of heat loss through the system. The heat loss was estimated by measuring the steady-state temperatures into and out of the steam dryer before slurry flow rate was started in each test. An average of 1625 Btu/hr heat loss was measured, and this value was used in the data reduction computer program for each test point.

6.4.2 Testing Procedure

A typical testing procedure consisted of the following:

(1) Heat up the entire steam atmosphere dryer test apparatus using the facility boiler steam. The entire system was allowed to reach steady-state temperatures (approximately 280 to 285°F) by not injecting water or slurry. This procedure usually took 40 to 60 minutes.

(2) A batch of pumpable slurry mixture was prepared. A 15-lb, 50-percent (dry basis) sample of a water and feedstock mixture was typically enough for a 1- to 1 1/2-hour test, allowing 2 or 3 data points to be taken.

(3) Slurry was pumped to the injector nozzle at a controlled rate (between 6 and 40 lb/hr) using a peristaltic, variable speed pump. The slurry's atomizer nozzle used steam to atomize the slurry into a stream of microscopic particles. The atomizer was placed in two (2) different locations during the testing: at the dryer inlet and at the top of the dryer. This testing concluded that the preferred location for the atomizer is in the dryer inlet position.

(4) The dryer inlet and outlet steam temperatures and the cycle discharge temperatures were continuously monitored and recorded at 1- to 5-minute intervals during the test. Each test ran until steady-state dryer operation was reached, which usually took 15 to 25 minutes.

    During the first series of tests (1 through 22) the test time was limited (to typically 10 minutes) by the clogging of the steam-atomizer nozzle.

    During the testing, slurry weight as a function of time was recorded as were the steam flow rates (via the orifice pressure drops). The slurry flow rate measurements, the steam dryer inlet and outlet temperatures, and the cyclone discharge temperatures are considered highly accurate. The boiler steam flow rate, however, was observed to fluctuate between 2 and 3.5 in. wc as recorded by the pressure transducers. This corresponds to between 250 and 400 lb/hr of steam. Thus, there was a variation of steam flow rate through the steam dryer during each test due entirely to the limited capacity of the facility steam generator. This fluctuation was unavoidable and entries of steam flow rate to the Data Reduction Program were equal to the weighted average of the steam flow rate with respect to time. During the analysis of the measured hot test data, it was found to be worthwhile to determine the steam flow rate based on a heat balance from each test run. When used, this recalculation of the steam is identified in the subsequent figures as "corrected" steam data.

(5) The dried powder was collected in plastic airtight bags from the cyclone's collection box. These samples were immediately sealed, cataloged, and weighed. The sample collection procedure did not require the system to be shut down.

(6) The testing continued by repeating steps 3, 4, and 5 using higher slurry flow rates.

(7) The wetness of each dried powder sample was measured by removing approximately 30 grams and weighing this smaller sample before and after it was dried in a baking oven. The oven's temperature was set at 220°F.

(8) The data collected from each test was entered into a data reduction computer program to determine such important information as:

- heat balance
- actual heat transfer in the dryer
- calculation of the dryer's heat transfer coefficient
- particle loading ratio

A typical output from the Data Reduction Program is shown in Table 6.2 for reference purposes. The heat balance is checked in two ways:

1. Summation of $\dot{m}$ x h for dryer inlet and outlet streams (see line A) and
2. By determining how much slurry <u>could have been</u> evaporated given the measured steam flow rate and dryer inlet and outlet temperatures (line B).

This potential slurry flow rate is then compared with the actual slurry used in the test run (see Line C) to determine a (+ or –) percent difference.

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TABLE 6.2

SAMPLE OUTPUT FROM THE TEST'S DATA REDUCTION COMPUTER PROGRAM

TEST DATE: 12/27/90 CASE NO.29

	SLURRY TYPE	NEW CLAY			
	SLURRY DENSITY (LBm/cu. ft)	165.5			
	DRYER INLET TEMP. (F)	290			
	DRYER OUTLET TEMP. (F)	253			
	PRODUCT SP. HT. (Btu/LBm-F)	0.38			
	SLURRY TEMP. IN (F)	70			
	STEAM FLOW (Dp and **LBm/hr)	3.8	399.66	Measured Water Removed	
	SLURRY WATER CONTENT (Mwater/Msolids)	0.5			
	SLURRY INJEC. FLOW (LBm/hr)	21.66	**Wo	0.124	
	**INJ. STEAM FLOW (LBm/hr)	11.99	AT TEMP. (F)	270	
	SLURRY DRAIN AMOUNT (ml)	30	equals**	0.11 (LBmix.)	
	AMOUNT OF SOLIDS RECOVERED (LBm)	0.45			
	DURATION OF TEST (min.)	10			
	**DRAINED SOLIDS FLOW (No./hr)	0.45	**DRAINED WATER FLOW	0.23 (LBm/hr)	
A	**INLETΣ(Mi x hi) (Btu/hr)	489,741.0	(Heater = 2048, Qloss = 1625 Btu)		
	**OUTLETΣ(Me x he) (Btu/hr)	489,814.3	Net Amount of Solids Pumped (LBm)	2.41	
	**TOTAL HEAT BALANCE (%)	0.0	0.0		
	**SOLIDS LOADING (Ls)	0.035	0.035		
	**WATER LOADING (Lw)	0.018	**Meas'd. Ht. Trans. (Btu/hr)	8434.214	D
	**HEAT BALANCE w.r.t. SLURRY (%)	-0.9	-0.9		
B	**MAXIMUM SLURRY THAT COULD BE EVAP. FROM AVAIL. STEAM (LBm/hr)	23.24	23.24		
C	**PERCENT DIFF. OF ACTUAL SLURRY EVAP'D.	-10.8	Dryer Dia. (in.)	6	
	**DRYER HT. AND MASS COEF.		Dryer Height (in.)	50	
E	(Btu/hr/DTLMF/VOL)	170.6627	Dryer Area (cu. ft)	0.86	

NOTE: ** DENOTES CALCULATED VALUES

The calculated heat transfer is given in line D and the calculated dryer heat transfer coefficient in units of Btuh/Vol x ΔT_{LM} is given in line E.

6.4.3 Chronology of Significant Testing Events

The following chronology of the laboratory's testing is provided in order to familiarize the reader with the extensive laboratory preparations, modifications, and testing conducted in support of the Task 4's laboratory testing program. This chronology has been excerpted from the hot test laboratory notebook, which provides complete details for the hot testing conducted to date.

Entry 1: First Test

The first test of the steam dryer with clay feedstock proved to be successful. A small beaker containing only 1 liter of a 50-percent (D.B.) clay-water slurry was used. After thoroughly heating the system until steady state was achieved (as evidenced by equal dryer inlet and outlet temperatures), the clay slurry was injected into the dryer. Very little was seen to be deposited into the cyclone's collection bin; however, some dry clay was found there and was observed to be dry, still white, but slightly dirty and virtually identical to the initial product mixed with the water.

In future tests it has been decided to use a cloth bag (for example) at the outlet of the cyclone to collect the dried particles. It was also observed that the recirculating return leg needs to be kept slightly opened in order to avoid condensate from accumulating in the stand pipe and eventually choking off the recirculating line until it is ineffective.

The hot testing will continue, first with a simple set of experiments to determine the heat balance across the device with simply water (and no feedstock particle) being injected. These tests will also shake out problems with the instrumentation and/or system configuration.

Entry 2: Steam Loop Apparatus and Instrumentation Checks

The steam dryer was tested for longer durations today. Testing consisted of the operation of the dryer system at different injection water and main dryer steam inlet flow rates. Several interesting observations were made.

The injector flow rate has limited range; perhaps two or three flow rate settings will be able to be run with repeatability. This should be adequate for these hot test experiments.

The steam boiler valves were finally opened full with the result that the boiler maintained approximately 60 psig before the steam separator and forward pressure regulator while the inlet dryer was at atmospheric pressure. Thus, there would seem to be 60 psig pressure drop across the separator and regulator. This results in a dryer inlet temperature of only 265°F instead of the 285 to 290°F preferred. Operation at this condition is unsteady and does not allow for a constant steam flow at sufficiently high pressure for these tests. Thus, the testing will be performed with the boiler valves set at an intermediate position, one determined from additional calibrations that gives a constant steam flow rate for longer periods of time.

The vortex pressure drop seemed to drop steadily as the testing time proceeded. There is a suspicion that the bottom of the dryer is collecting condensate and that this pool of water is affecting the vortex in this bad manner. It was already determined that the collection of condensate in the recirculating tube chokes the recirculation around the dryer and eliminates the vortex pressure drop. This was remedied by keeping the drain valve at the bottom of the tube slightly opened at all times. It is interesting to note that this may need to be a collection point for the dried particles and that a collection system from this point should be considered. A drain and a "false" bottom will be installed into the bottom of the dryer to eliminate the pooling of condensate into this area. The next series of tests will help determine if these remedies worked as planned.

The pressure drop instrumentation will also be moved to two new locations to try to get better readings. The negative vortex pressure will be monitored by tapping into the recirculating line just before the 90-degree elbow. The positive vortex pressure drop will be monitored from a pressure tap at the exit of the dryer, near where the TC1 measurement is made.

The thermocouples on the dryer shell are working well to indicate that the wall temperatures near the main steam inlet are really at or close to saturation conditions (215°F, for example). Clearly, the physical space available for steam inlet and water injection are too small for the evaporation of the injected water to occur and thus the inside walls are "seeing" saturated water temperature conditions until the heat transfer has a chance to happen. It will be difficult to "scale" the injector and steam nozzle inlet conditions for the larger steam dryer sizes. This "cooling" effect should <u>decrease</u> as the dryer sizes approach more typical dryer sizes. Unfortunately, this effect will be a negative effect on the results we hope to obtain from these preliminary "proof-of-concept" tests.

The injector nozzle will probably need to be moved to the top and/or bottom to experiment on how the different nozzle locations affect drying.

Entry 3: Water Injection Tests A Through G

Today's testing started late due to the need to modify the dryer assembly and reposition the pressure transducers that measure the vortex pressure drop.

I have also started to attempt to collect the condensate that is drained from the dryer during the testing. A small 1-liter beaker is placed at the bottom of the dryer directly below the two drain lines installed there. The drains are from the recirculation line standpipe and the dryer bottom, respectively. I believe this should be done during all future tests of the dryer as a means of monitoring how much of the injected water or slurry is not evaporated.

Today's testing lasted for only one hour. During this time there was not any noticeable deterioration of the vortex; perhaps a sign that the modifications to the dryer to improve condensate draining have worked. An inspection of the data collected during this brief one-hour test reveals that the dryer does stabilize to steady-state quicker than when the drains were not in place.

Entry 4: Tests No. 1 to 5

The dryer was tested for several hours. These tests were the first "valid" tests of the dryer since the dryer modifications were completed. Tests with and without the heater tapes ON were performed. The data from these tests have been studied; specifically cases 3, 4, and 5 (without the heater tapes on) were tested. The heat balances were very close. All measurements have been checked against calibrations performed specifically for the testing. For example, the orifice was calibrated for a DP of 2 and 3 in. wc, and all of the tests were conducted with a steam flow through the orifice that recorded 3 in. wc. During the testing this pressure drop would be seen to vary (up or down: to 2.6 in. wc, for example) but this was seen to be affected by the boiler operating pressures. The average reading of 3 in. wc was maintained throughout the testing.

The next set of runs will be performed with the clay powder mixed with water to a 50-percent (dry basis) mixture.

Entry 5: Tests No. 6 and 7

Today's test was the first serious attempt to dry slurry. SUCCESS... A considerable amount of dried powder was recovered from the cyclone particle bin. The bin was literally filled with the dried powder. After shutting off the slurry injection, steam was forced through the dryer and cyclone on two separate occasions and each time a little more dried clay powder was removed. The powder was collected in two separate bags. The powder will be weighed and its dryness measured using an oven drying technique. The clay powder appears to be whiter than the original product and somewhat more "soft" or "fluffy" to the touch. It was expected that the steamed dried powder would be denser; this test seems to indicate that the steam dried powder is less dense. Specimens will be given to APV Crepaco for further analysis and their objective opinion as to the quality of the dried product.

Additional testing will now be performed with a little more care to collect the dried product quickly from the cyclone particle bin. It will also be necessary to try wetter slurry to avoid the clogging of the injection nozzle as was observed during this testing. This test was a GREAT SUCCESS and proved that the equipment can steam dry the clay product. Now it will be necessary to get more quality data for assessing the efficiency of the IRIS dryer configuration. The principal question remains to be answered: Does the dryer performance match the computer predictions for necessary size?

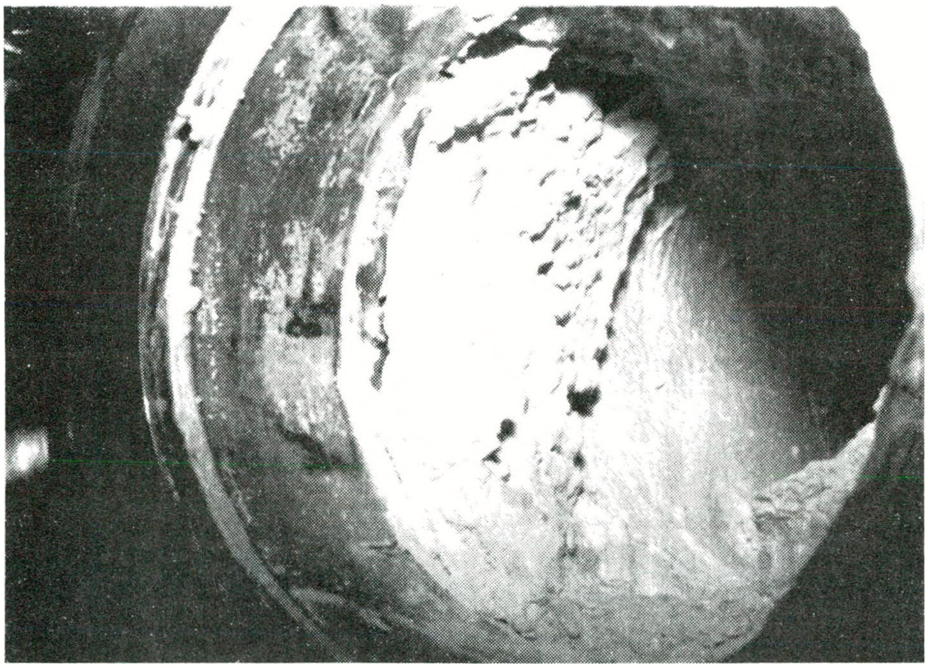
POSTSCRIPT

The following observations were made on the first morning after the testing of data point no. 7NH.

The bottom of the recirculating tube was found to be completely clogged with dried clay powder. The bottom of the steam dryer was found to have agglomerated dried clay around the drain leg of the recirculating spout in the center of the dryer, directly across from the steam nozzle inlet. The steam/slurry injection nozzle was also clogged (see Figure 6.34). It is clear that the spray nozzle at the steam inlet will continue to give an agglomeration of clay powder at the inlet. Hence, the nozzle will be moved to the top of the dryer and inject slurry against the axial steam velocity in the dryer chamber. At this time a variable cross-sectional nozzle inlet area assembly will also be installed. This will allow the cross-sectional area to be varied (made smaller for higher inlet velocities) during a test.

Positioning the slurry nozzle into the bottom of the dryer (up through the recirculating spout tube) is also a possibility. However, there is some concern that the spray from the nozzle will spray directly into the outlet nozzle of the dryer and leave the dryer wet or at least not as dry as it should be.

View of Bottom of Dryer Removed
From the Dryer Vessel



View From Bottom Looking Up Into Dryer

Figure 6.34 Clay Powder Clogging Bottom of IRIS Dryer

Entry 6: Tests No. 8 to 11

The bench-scale steam dryer is back together with a new steam-slurry nozzle injector relocated at the top of the dryer (see Figure 6.35). The new injector is water cooled and is installed through the top dryer orifice plate. The orifice plate needed to be increased in diameter in order to accommodate the larger diameter nozzle. The total flow area for the steam was kept the same, however. There is some concern that the larger diameter orifice plate may affect the vortex pressure drop by "locating" the vortex differently.

The tests numbered 8, 9, 10, and 11 were conducted with water (8 and 9) and 50-percent slurry (10 and 11) injection. The data have been reduced and are available. Slurry at 50-percent D.B. may still be too thick to avoid clogging the nozzle. The first attempts did clog the nozzle, but it was cleaned quickly and the test proceeded.

Entry 7: Tests No. 12 to 18

The testing continues with data points 12 through 18. At this time slurry with 60-percent D.B. will be tested. Observations are made that the cyclone bin should have a collection pipe installed into it to make it easier to recover the dried powder and without needing to shut down the steam flow. This will be done at the earliest opportunity. It was also learned that the plastic bags used to collect the powder and thought to be usable in the microwave oven for drying the collected powder are not suitable. A petri dish or crucible must be used to dry the powder and thus determine the dryness of the powder coming out of the steam dryer. Some data have been collected, however (runs numbered 14, 15, and 16). The first attempt at drying a temperature-sensitive slurry made of non-diary creamer was not successful. No powder at all was recovered and the creamer slurry could only be recovered by flushing the steam dryer rig with steam and high water injection. A sizable quantity of the slurry was recovered (albeit diluted) – approximately 1.5 to 2 gallons. After additional flushing a final water-only test was run (test no. 18). The data are available for review.

Entry 8: Tests No. 19 to 21

The dryer rig was partially disassembled for inspection after the weekend testing. The last slurry run in the dryer was the non-diary creamer, and it coated the top nozzle and top sections of the steam dryer (see Figure 6.36). The cyclone and the steam separator inside surfaces were clean. A crown of glazed (creamer) material was found covering the top of the recirculating spout. Approximately 2 inches of dried powder was found under this dome, hence the clogging and

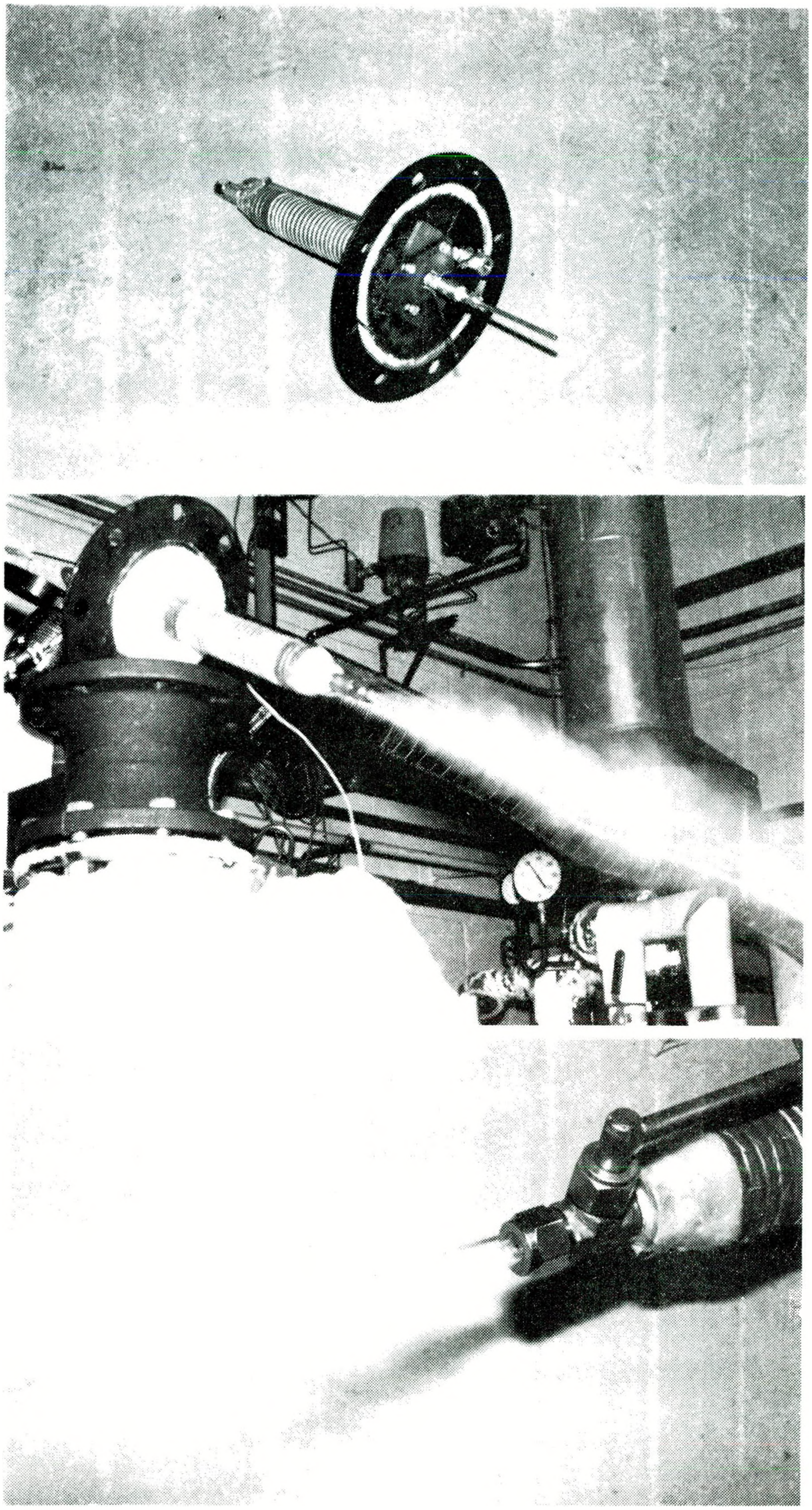
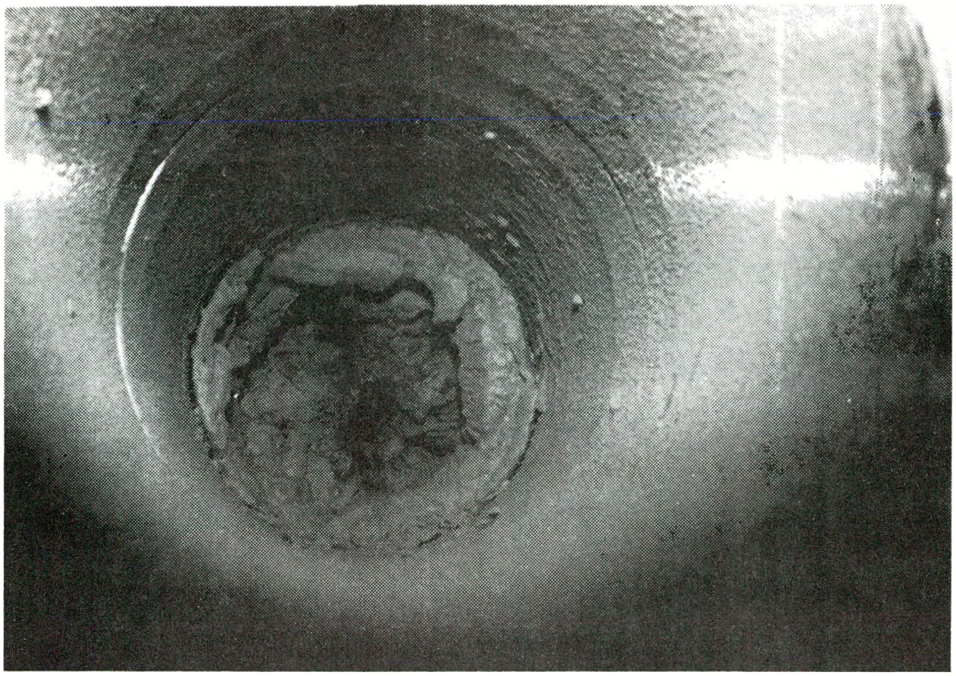
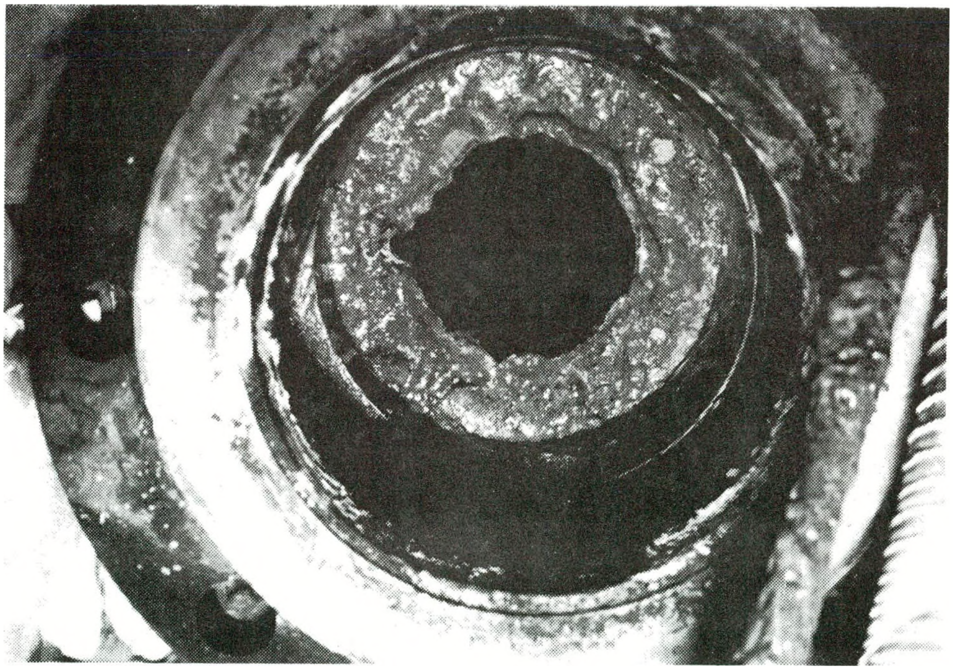


Figure 6.35 New Steam Nozzle Atomizer Modified for Installation at the Top of the Steam Dryer



View of the Dryer's Interior From the Top



View of Dryer's Outlet Orifice From the Top

Figure 6.36 Photos Showing the Effects of Drying the Non-Dairy Creamer stopping of 21 were obtained and are available for review. The last two runs (20 and 21) used slurry. A new collection can was used for the first time during these tests to collect samples of dried powder without shutting down the dryer. The collection procedure works well. Thus, two consecutive runs were possible for the first time and these runs performed very well; approximately 1 lb was collected and the powder was wetter with a faster slurry flow rate, as would be expected. This is encouraging, and it is hoped that the remaining tests can be conducted smoothly.

Entry 9: Test No. 22

Some difficulty was had in procuring more dry clay powder. Previous testing depleted our supply and APV Crepaco did not have any more to give Tecogen. We purchased some "over the counter" pottery clay but found that this powder has had some binder material added to it that causes the powder to require much more water before it will flow easily and allow pumping. An attempt was made to use the smallest grain size alumina powder that Tecogen used in the cold testing of the IRIS model. However, these powders would not stay in suspension long enough for our peristaltic pump to work properly. Ultimately, a batch of approximately 8 lbs of "old," used steam dried powder was remixed with water to 50 percent (D.B.) and used in a brief test. The steam nozzle during this test (no. 22) clogged and had to be cleaned out before the testing could continue. The testing continued with a relatively large flow rate of slurry, but no dried powder was recovered after 30 minutes of drying. The slurry flow rate was slowed to almost one half and the test lasted about 17 minutes before the slurry was depleted. No dried powder was collected, however. The top of the dryer was removed to reveal that the top half of the steam dryer was seriously clogged with dried powder (a sketch and photos are available); enough so that the full diameter of the dryer was "bridged" with powder and the recirculating line was literally flowing backwards in order to get steam out of the dryer. This would seem to explain why the vortex pressure drop decreased to zero and then went negative (off the scale) during the last part of the testing. At this time a review of the testing indicates that the relocation of the steam injector atomizer to the top of the dryer may have been ill-advised. The steam nozzle will now be returned to the inlet with some modifications to allow some water cooling. It is also recommended that the slurry mixture be increased to a water content of 60 to 70 percent to help avoid pre-drying in the hot atomizer. It is interesting to note that 8.35 lbs of dried material was collected from the dryer during this clean-up but only a maximum of 5.4 lbs was pumped into the dryer during tests no. 20 and 21. Thus, almost 3 lbs had to be deposited during the last runs providing for the start of the "bridging" that was observed in today's test. The deflector installed into the dryer for these tests has been removed for the next tests. Also, prior to starting the slurry tests again, the outlet vortex orifice will be changed to try to discern some improvements in the vortex pressure drop. The vortex pressure drop has not been holding very steady during the last tests. The reason for this "lack of vortex pressure drop" is not for the lack of recirculating flow, rather it seems now that the particle collection at the bottom of the dryer is causing an interruption in the readings. By evidence of the recirculating wall temperatures it is very clear that the external recirculating line does have recirculation (i.e., all the wall temperatures are above 212°F). Unfortunately, there is nothing that can be done quickly to measure the amount of recirculation without a good reading of recirculating line pressure drop.

The Monday night test was encouraging in that it seemed that consecutive tests can be run now that the collection tube was installed at the bottom of the cyclone to collect any dried powder. Monday night's runs collected two such powder samples back-to-back without shutting down the steam dryer and varying the slurry flow between dryer tests. These samples are collected and labeled as runs 20 and 21. These runs also saw the use of the new deflector cones for the first time. These last runs were made with the last of the APV Crepaco clay powder samples. More has been requested but the powder has been sent back to APV's customer. An alternative supplier is being sought.

The testing conducted on Tuesday night was to use pottery clay bought locally from a clay wholesaler. However, the binder in the clay seems to require more water to be added if it is to maintain the same liquid consistency as APV's clay. A water-to-particle ratio of 3.7:1 was found to be necessary to match the consistency. For this reason it was decided to re-use the clay that has been removed from the steam dryer and to mix it once again to a slurry with a water-to-particle ratio of 0.5:1. It is not clear that the clay powder once steam dried (or air dried) can be remixed with the same properties.

Tuesday night's testing resulted in no samples collected. The data is stored in runs 22a and 22b. Although the testing went relatively smoothly, no samples of dried powder could be collected. Apparently all of the slurry stayed in the dryer. An inspection of the dryer will determine where in the dryer this slurry finally was collected.

Entry 10: Tests No. 23 to 25

An inspection of the steam dryer revealed that considerable clogging of the dryer inlet had occurred. Approximately 8.35 lbs of hard clay powder was recovered. This is to be compared to the 5.4 lbs of slurry that was pumped into the dryer during the night's testing (runs 22a and 22b). The balance of the dried powder apparently comes from previous testing.

The day is spent redesigning the new nozzle to be water cooled using a small capillary cooling tube to fit into the inlet of the steam dryer. It has been reasoned that the nozzle installation at the top of the dryer is not working. Apparently the nozzle steam injection significantly affects the upward velocity of the carrier (main) steam. The result is the depositing of the powder on the dryer walls, which in the last test caused a complete bridging of the walls. This resulted in the backward flow of steam up through the recirculation tube for the steam to leave the system. This was evident during the testing by observing a negative pressure difference across the dryer vortex. It was decided to return the steam nozzle atomizer to the inlet where it once performed well. However, cooling of the nozzle was deemed still necessary and thus a redesign of the nozzle was performed and its construction completed for testing in the evening.

With the lack of fresh clay powder, the already dried powder was re-slurried to a 60-percent mixture for testing. Tests no. 24 and 25 were run and results recorded. A computer data collection system was also installed during this time to record the dryer skin temperatures and to provide the operators with more of an opportunity to observe the testing.

Entry 11: Test No. 26

The last runs also resulted in the clogging of the inlet. However, it seems that the agglomeration is more easily cleaned when at the inlet by a combination of steam blasts from the boiler and water back-filling of the dryer and flushing. It is becoming more evident that the reuse of the dried clay is not working. Perhaps the dried clay changes its surface tension, size, and/or some other property in such a manner as to make it useless for retesting. It is decided therefore to wait for the new clay powder that has been promised for delivery on Friday morning. An attempt was made to use the pottery clay mixed to a 3.7:1 ratio with water. This mixture ratio gives the same liquid consistency as the APV clay-water mixture. However, the testing performed this evening (run no. 26) clearly demonstrated that this mixture is too wet for proper drying. It is also clear from the data that the slurry flow rate was purposely held high (perhaps too high) in order to avoid clogging of the nozzle during slurry injection. It is clearer now that the testing should revert back to the very first testing conducted in test no. 7NH and should duplicate as much as possible those successful test conditions.

Entry 12: Tests No. 27 to 31

The testing was conducted using fresh HUBER clay powder that seems to be of the same general characteristics of the original APV clay. The slurry was mixed to a 50 percent by dry weight solution. The nozzle is at the inlet and tests will be conducted with the nozzle not cooled.

The testing seemed to go very well. Test points 27, 28, 29, 30, and 31 were recorded. All but test point no. 27 were with slurry. Various flow rates were used. On several occasions the nozzle clogged but it was cleared easily. Samples were taken in each case often after the dryer was blown out with a large rush of steam from the boiler. The dryer was also allowed to steam clean itself before runs if the nozzle was clogged or if there were signs of the inlet becoming clogged with dried clay powder. In summary, these tests would seem to be long enough to get quality data.

Entry 13: Tests No. 32 to 38

The data collected during the last week of testing has been reduced and appears to be very revealing and interesting. Heat transfer coefficients have been discerned and plotted vs. loading (L_p). At this time, another redesign of the steam atomizer nozzle has been conducted. A glass insulating tube has replaced the steel tube that served to introduce slurry into the steam nozzle for atomizing. This tube has been put into place using RTV for expediency purposes and is first tested in the December 13, 1990 tests. These tests result in a substantial increase in the time that slurry can be injected into the dryer. In fact, for the first time, the testing did not need to be suspended in order to unclog the steam atomizer. This has resulted in seven successful tests (runs 32 through 38), all with 50 percent slurry. The test results are being analyzed at this time. It should be noted that the first five tests (32 through 36) all were conducted with the heaters on and thus somewhat simulating the indirect dryer heat transfer mechanism. The results may indicate for the first time the benefits and/or abilities for the walls to conduct heat to the drying particle.

Entry 14: Test No. 39

The food sweetener maltodextrin-100 finally arrived from the Grain Processing Corporation. The steam dryer test facility is prepared for a second attempt to dry a food feedstock. Approximately 14 lbs of slurry are mixed, but to a wetness of 63 percent. This higher wetness is required so as to make the feedstock-water mixture less viscous and thus pumpable by the peristaltic pump. After approximately 55 minutes of pumping at two different slurry flow rates, 6 and 9.5 lb/hr (a slow-to-moderate mass flow rate), the experiment was terminated. No dried samples were collected at the bottom of the cyclone particle separator. The IRIS dryer was cooled long enough to have its top section removed so that an internal inspection could be quickly made of the dryer. The bottom of the dryer was coated with a glazing of the slurry. The glazed slurry looked clear (no discoloration) and was being contoured in a helix pattern by the steam. The depositing of the slurry would have likely continued if the experiment was not stopped. This pattern was observed for the clay slurry but in the case of the clay, clay particles would be carried off by the steam and eventually deposited in the cyclone's collection bin. In this case, the sweetener did not get entrained by the steam. Perhaps this would have happened if the experiment were continued longer. The dryer system was closed and a combination of hot water washing (facilitated by using the steam atomizer nozzle and injecting only water) and hot steam cleaning was able to clean the dryer to its original condition. Thus, it should be noted that the glazed sweetener that has been deposited upon the dryer walls was washed off easily by no mechanical means, but by the cleaning action of the hot water and steam.

This experiment should be tried again with the same dryer conditions to eliminate the possibility that some experimental procedure had been missed which may have caused this poor result. Tecogen is also expecting the delivery of autolyzed yeast from a local brewery, and this "food" feedstock should be tested as soon as possible in order to provide a third sample of food-type feedstock for the laboratory test.

6.4.4 Steam Dryer Data Summary

A total of 39 tests have been run to date. Twenty-four tests were performed with slurry feedstock; fifteen were tests run with just water injection (for purposes of instrument calibration and equipment debugging); twenty two tests were conducted with a clay powder feedstock; and two tests were conducted with temperature-sensitive materials – coffee creamer and a food sweetener, maltodextrin-100.

A complete summary of the materials tested is given in Table 6.3, including their initial and final wetness and the amount of sample collected from each test.

Figure 6.37 displays photographs of a typical clay slurry (with a wetness of 50 percent (dry basis) before its injection into the dryer and the dried powder resulting from the steam atmosphere drying. The final measured wetness is plotted in Figure 6.38 as a function of particle loading. This figure displays the important result that the powder exit dryness remains low despite an increase in particle loading; an encouraging result.

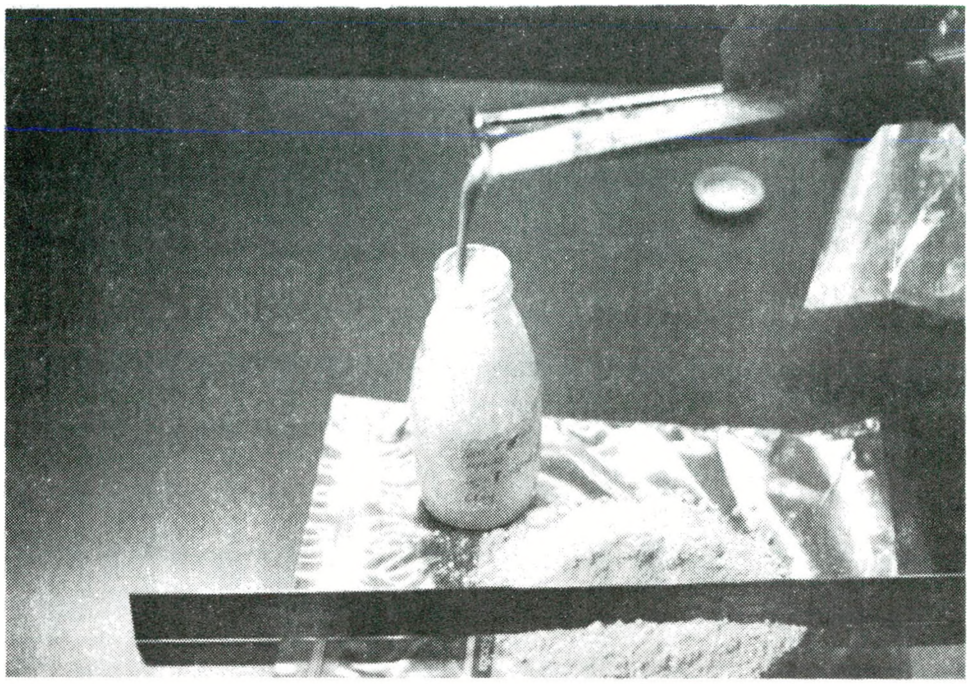
A plot of the percent powder dryness as determined from: $(1 - W_o/W_i) \times 100$ where W_o and W_i are the final and initial powder moisture contents, provides a different perspective and is given in Figure 6.39. Figure 6.39 also identifies the test points that were tested with or without the wall heaters on. The powder dryness is observed to be typically above 85 percent and does display the expected trend of lower powder dryness as more slurry is injected into the dryer.

TABLE 6.3

POWDER TEST RESULTS

VORTEX	Dp TEST NO.	MATERIAL	Wi	$LB_{collected}$	Wo	NOZZLE LOCATION
	7	CLAY	0.50	1.03	0.0100	INLET
	10a	CLAY	0.50	0.22	0.0560	INLET
	10b	CLAY	0.50	0.38	0.0610	INLET
0.2	11a	CLAY	0.60	0.17	0.4000	TOP
	12	CLAY	0.60	0.93	0.0710	TOP
0.5	17	CREAMER	0.50	0.00		TOP
0.3	20	CLAY	0.56	0.44	0.1330	TOP
0.0	21	CLAY	0.56	0.33	0.0670	TOP
0.3	22a and b	CLAY*	0.50	0.00		TOP
	24	CLAY*	0.60			TOP
	25	CLAY*	0.60	0.06	0.2560	TOP
	26	CLAY**	3.67			TOP
	28	CLAY***	0.50		0.0420	INLET
	29	CLAY***	0.50	0.65	0.1240	INLET
	30	CLAY***	0.50	0.08	0.7970	INLET
	31	CLAY***	0.50	0.28	0.0640	INLET
	32	CLAY***	0.50	0.29	0.0168	INLET
	33	CLAY***	0.50	0.25	0.0240	INLET
	34	CLAY***	0.50	0.11	0.0360	INLET
	35	CLAY***	0.50	0.31	0.0640	INLET
	36	CLAY***	0.50	0.53	0.0460	INLET
	37	CLAY***	0.50	1.12	0.0330	INLET
	38	CLAY***	0.50	1.41	0.0380	INLET
	39	MALTODEXTRIN-100	0.03	0.00		INLET

* REUSED CLAY
** POTTERY CLAY
*** NEW CLAY



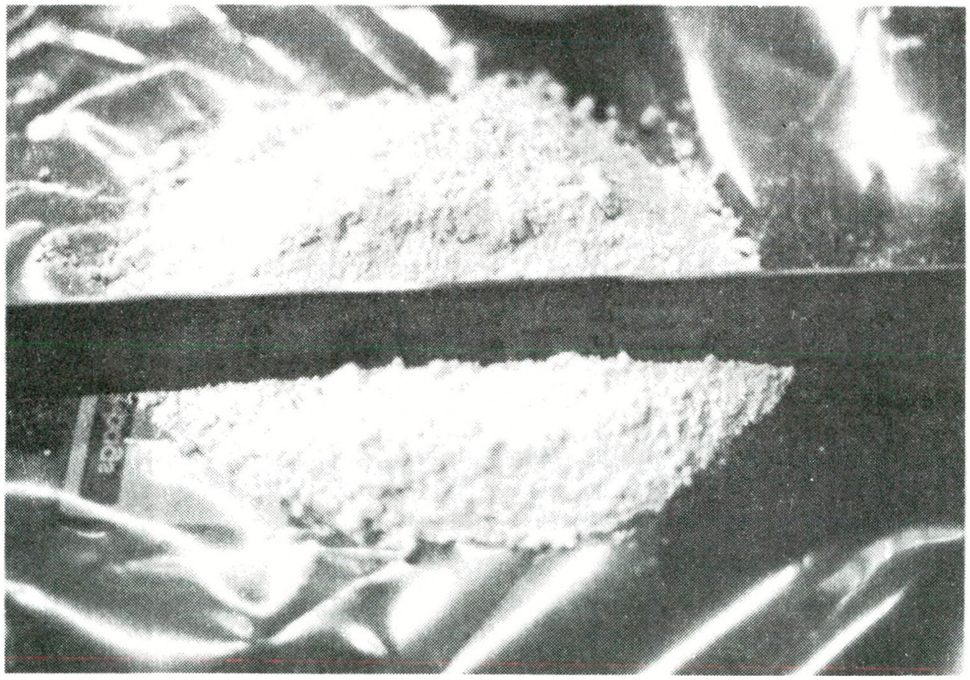


Figure 6.37 Photos of Clay Slurry Feedstock (Top) and Steam Dried Clay Powder

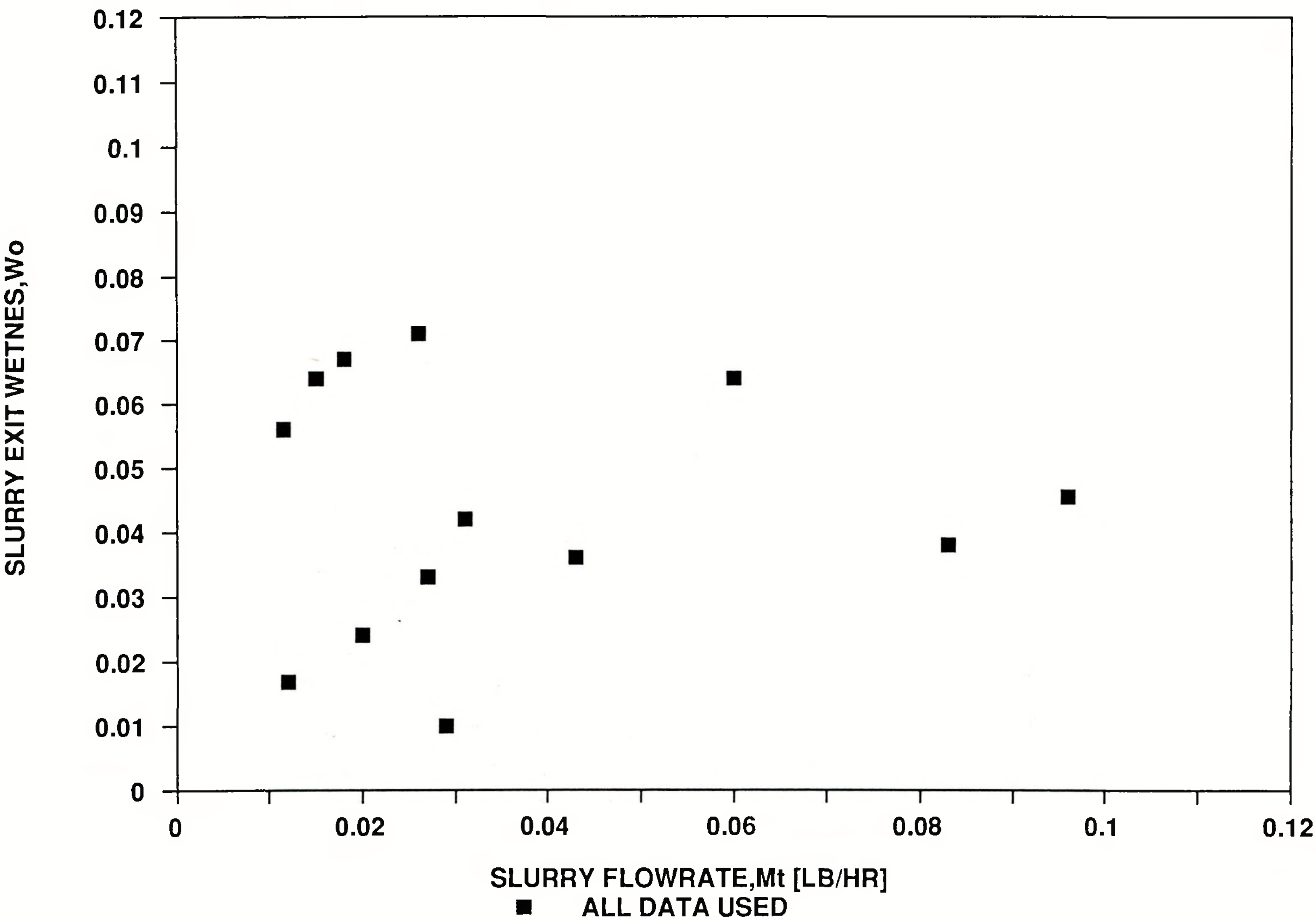


Figure 6.38 Steam Hot Test Data
Clay Slurry Wi = 0.5

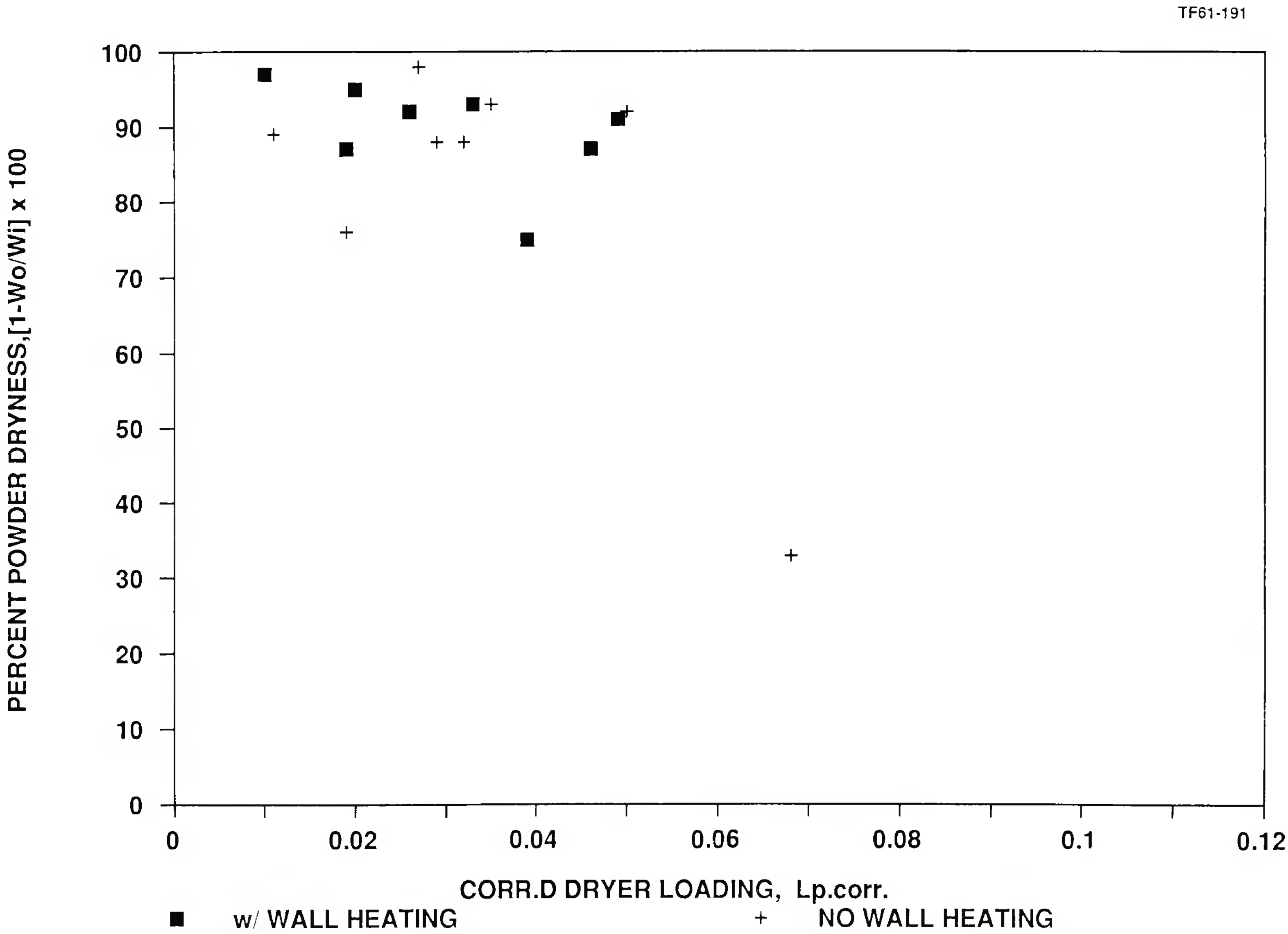


Figure 6.39 IRIS Drying Ability with Clay Slurry; All Data Used

Fortunately, this decrease in drying performance is slight. However, the powder dryness with wall heating appears to be higher than those test points that had no wall heating. Although this test sample is small, this is the first evidence of the benefits of wall (i.e., jacket vessel) heating in the drying of the slurry.

The high drying performance is also evidence that the dryer's external recirculation is in effect and perhaps retains the dried particles longer than expected, consequently the powders become very dry in the process even as the particle loading (L_p) is increased.

Given the evidence of the ability of the IRIS-type dryer to completely dry a pumpable slurry, a major goal of the first phase of the steam atmosphere drying project has been realized – the IRIS dryer does indeed dry a slurry back to its constituent powder with no signs of product deterioration. Samples of the steam dried powder are being studied by Tecogen's project partner APV Crepaco, Inc. and their report should be available for inclusion in the Final Report. Tecogen's inspection of the dried powder revealed no deterioration of any kind with respect to color, density, or odor. The dried powder was found to have an average weighted size of 160 microns compared to its original size of 106 microns. This is perhaps due to the loss of the smaller grains through the cyclone as well as the agglomeration of the particles due to the drying process. The final dried product is from all appearances identical to the original powder.

The capability to both dry and recover two food products, a coffee creamer and a food sweetener, maltodextrin-100, was found to be more difficult. In both instances, the slurry was found to "glaze" the inside of the steam dryer, forming a thick deposit on the walls without being carried off by the steam. Although the clay powder slurry also tended to deposit along the bottom of the dryer (beginning near the slurry injection point), the surface powders were known to blow off with an increase in steam velocity. The food feedstocks could not be similarly dislodged but rather continued their glazed buildup on the bottom of the interior dryer walls, beginning at the point where the slurry was injected into the dryer.

The tendency of the slurries to deposit on the dryer wall is not completely unexpected for this size model dryer. Due to the close proximity of the slurry injection point and the dryer wall, particles could easily contact the surface while still in a wet, sticky state. Additional testing with different atomizers, injection locations, and a larger dryer, along with different food products (in either the granular or slurried form) can likely alleviate this problem. In future designs, mechanical scrubbers may also serve to remove the buildup.

A complete summary of the measured and calculated test data is presented in Table 6.4. This table was constructed from test data derived from the output of each test's data reduction computer program. Table 6.4 also includes calculations required to conduct the steam atmosphere dryer analysis given in the following sections of this report. Thus, all of the measured steam dryer test data and the relevant analysis data derived from the steam hot testing are given in Table 6.4.

Several analytical observations of the data in Table 6.4 must be noted to facilitate the understanding of the data analysis that is presented in the next sections. These observations are as follows:

(1) The dryer main steam and slurry atomizer steam flow rates are listed under the column headings "m_{steam}" and "m_{inj}," respectively. These steam flow rates are based on weighted time averages of the steam flow rate through the dryer. Attempts were made to keep these flow rates constant and as high as possible. However, limitations in the size of the facility boiler and the desire to maintain high steam flow rates often caused changes in the steam flow rates during the data collection for a test.

(2) The most accurate test data recorded during all of the testing was the dryer inlet and outlet temperatures and slurry mass flow rates. Unlike the fluctuations in mass steam flow rate (see item 1 above), the slurry mass flow rate was easily fixed via the variable speed positive displacement pump.

(3) The heat balance values shown in the column heading "Ht. Bal. % Diff.," reflect a comparison of actual slurry flow rate (see column "m_{slurry}") with a slurry flow rate which could be theoretically possible if the given measured steam mass flow rates (m_{inj} plus m_{steam}) are cooled from the dryer inlet temperature to the dryer outlet temperature. Thus, a positive value in this column indicates that the measured slurry flow rate is higher than the theoretical potential slurry flow rate. That there is a difference in these two slurry flow rates is indicative of the potential errors in the measurement of the test's steam mass flow rate (see item 1). This percent difference calculation is therefore useful in calculating a "corrected" particle loading quantity (L_p) (see column heading "Corrected Loading"). This corrected parameter is used in some of the following analysis and will be properly noted in the affected tables and figures. It is interesting to note that the percent differences are relatively small, which implies fairly good accuracy in measuring the dryer processes.

TABLE 6.4

SUMMARY OF MEASURED AND CALCULATED HOT TEST DATA FOR CLAY SLURRY

TEST NO.	IRIS PARTICLE COLLECTOR EFF.	NO. OF ATOMIZED PARTICLES	MAX. REC. RATIO (R) REQ. TO PRODUCE THEOR. COEFF. hp, theor.	CORRECTED LOADING TEMP. Lp, corr.	DRYER INLET TEMP. (F)	MEAS'D HT. TRANS. (BTUH)	LOC. PART. hp, meas'd (w/R=1) (BTUH/sq. ft /dtlm)	DRYER OUT TEMP (F)	PARTICLE WETNESS AT INLET Wi	PARTICLE WETNESS AT OUTLET Wo	SLURRY FLOWRATE Mslurry (Mt)	STEAM INJ. Minj	STEAM FLOWRATE Msteam (Ms)	HT. BAL. % DIFF.	PARTICLE LOADING (Lp)	WATER LOADING (Lw)	SIZING HT. COEF. (BTUH/VOL /DTLM) (H)	NORMALIZED HT. COEF. (BTUH/VOL/ DTLM/Mt) (Hn)	VORTEX FLOW FIELD Dp (in. wc)
7NH	0.92	249,597	11.8	0.027	297	6,839	870.103	255	0.50	0.0100	17.40	6.0	390	5.2	0.0290	0.0150	130	7.5	2.4
10NH	0.91	98,978	11.0	0.011	289	2,700	809.624	268	0.50	0.0560	6.84	6.0	390	8.2	0.0115	0.0058	48	7.0	0.6
11NH	0.94	609,361	18.1	0.068	287	16,968	1,336.244	231	0.60	0.4000	38.30	13.0	390	-15.9	0.0590	0.0360	485	12.7	0.2
12NH	0.93	268,532	14.2	0.029	293	7,549	1,042.824	244	0.60	0.0710	16.80	12.0	390	-13.1	0.0260	0.0160	166	9.9	0.2
20NH	0.92	279,549	12.8	0.019	292	7,053	942.628	244	0.56	0.1330	17.47	13.0	390	35.2	0.0290	0.0140	157	9.0	
21NH	0.91	173,513	11.8	0.032	293	4,314	866.530	249	0.56	0.0670	10.87	13.0	390	-76.1	0.0180	0.0090	90	8.3	0.1
28H	0.92	266,811	12.1	0.026	285	6,838	894.450	254	0.50	0.0420	17.34	10.7	358	15.3	0.0310	0.0160	142	8.2	0.9
29H	0.92	301,238	12.9	0.039	290	8,434	952.484	253	0.50	0.1240	21.66	12.0	400	-10.8	0.0350	0.0180	171	7.9	
30H	0.90	619,689	9.9	0.050	288	11,694	726.078	244	0.50	0.7970	35.06	9.5	316	30.4	0.0720	0.0360	268	7.6	
31H	0.91	129,102	11.5	0.019	288	3,686	846.084	269	0.50	0.0640	9.33	12.0	400	-24.0	0.0150	0.0080	65	7.0	2.0
32H	0.87	103,282	7.6	0.010	294	2,286	559.593	285	0.50	0.0168	5.68	9.1	302	20.6	0.0120	0.0060	34	6.0	1.3
33H	0.90	172,136	9.6	0.020	287	3,955	709.675	265	0.50	0.0240	10.08	9.5	316	-1.0	0.0200	0.0100	73	7.2	1.1
34H	0.88	370,092	8.2	0.033	288	6,604	602.746	255	0.50	0.0360	17.06	8.2	275	22.2	0.0430	0.0200	133	7.8	0.5
35H	0.93	516,408	13.4	0.046	282	10,596	986.738	233	0.50	0.0640	27.16	8.7	288	23.5	0.0600	0.0300	303	11.2	
36H	0.93	826,253	13.4	0.049	281	15,871	990.528	230	0.50	0.0455	42.86	8.7	288	48.8	0.0960	0.0480	487	11.4	
37NH	0.89	232,384	9.4	0.035	291	4,782	695.143	253	0.50	0.0330	11.94	8.7	288	-27.9	0.0270	0.0130	96	8.0	
38NH	0.92	714,364	13.1	0.050	289	14,901	965.531	232	0.50	0.0380	36.90	8.7	288	39.3	0.0830	0.0410	411	11.1	

(4) The measured heat transfer (Btu/hr) in the dryer is calculated by the data reduction computer program by means of the following equation. This accounts for changes in enthalpy for the feedstock particle, water, and dryer steam mass flow streams going into and out of the dryer.

$$Q_{meas'd} = \left[\frac{m_T * Cp}{(1 + W_i)} + \dot{m}_{drained\ water} + \dot{m}_{entrained\ water}\right](212 - T_{in}) + \left[\dot{m}_T\left(\frac{W_i}{1 + W_i}\right) - \dot{m}_{drained} - \dot{m}_{entrained\ water}\right](\Delta H_{fg}) \quad (5)$$

(5) The heat transfer coefficient shown in Table 6.4 ("H, Ht. Trans. Coef., Btuh/Vol/ΔT_{LM}") is calculated based on the measured heat transfer, the log mean temperature difference between the steam temperature profile and the assumed particle temperature of 212°F and the actual volume of the bench-scale dryer model (0.859 ft^3). The dryer volume is based on the dryer height of 50 in., a diameter of 6 in., and a 1 1/2-in. by 40-in. long external recirculation line. No credit is given for heat transfer that could have occurred in the cyclone particle separator (installed downstream of the IRIS-type dryer) as the steam temperatures measured at the discharge of the cycle and the steam dryer consistently showed no change in steam temperatures during each test and thus no additional heat transfer was discernible in the cyclone.

(8) The actual particle or local heat transfer coefficient ("Particle hp (Btuh/ft^2/ΔT_{LM}") is calculated based on the assumption that each individual feedstock particle and entrained water mass can be represented by a sphere of water encapsulating each feedstock particle. Given that the initial slurry wetness was mixed to 50 percent (dry basis) and that the clay powder has a size of 106 microns, it can be shown that the spherical water shrouded particle has a diameter of 139 microns. The local particle heat transfer coefficient calculation assumes that the average temperature difference between the heat source (i.e., the steam) and the heat sink (i.e., the feedstock particle) can be represented by the log mean temperature difference between the steam temperature end states and the particle's water evaporation temperature of 212°F (corresponding to atmospheric dryer pressures). The total particle surface area is based on an average of the spherical particle 106 microns plus water diameter of 139 microns (consistent for the clay powder under test) and the generation of the number of (calculated) atomized particles shown in the column. The formulae used in these calculations are derived in the following section 6.4.5: Analytical Study of Steam Atmosphere Dryer Performance.

(9) The column heading "Max. Recirc. Required to Produce Theoretical h_p" represents the maximum number of external recirculations required by the IRIS-type dryer before the local particle heat transfer coefficient (h_p) is calculated to be equal to what is theoretically expected, given the particle's diameter and steam thermal conventional conductivity (Kg) and using the heat transfer coefficient expression:

$$h_{p_{(th.)}} = \frac{2\ Kg}{dp} \tag{6}$$

For example, for test no. 7NH, the local particle heat transfer coefficient is calculated to be 870.1 Btu/hr-ft^2-^{o}F. This calculation, however, (as will be shown in the derivation of the equation for $h_{p,\ meas'd}$) is based on the minimum time for the particle to have had its shroud of water evaporated from the particle. This minimum time corresponds to an IRIS-dryer recirculation ratio (R) equal to one (1). If this heat transfer coefficient were calculated based on an assumed IRIS recirculation ratio (R) of 11.8, the value of the local heat transfer coefficient would exactly match the theoretical heat transfer coefficient ($h_{p\ th}$) calculated from Equation 6 above. Fortunately, a recirculation ratio of 11.8 corresponds to a dryer collector efficiency of only 92 percent, which the IRIS dryer has little difficulty in providing.

Table 6.4 is the sole source of the measured and calculated values used in the analysis that follows. It is therefore the principal result of the hot testing conducted to date.

6.4.5 Analytical Study of Steam Atmosphere Dryer Performance

Before proceeding to the analysis of Table 6.4 and the measured performance of the bench-scale IRIS-type dryer, it is necessary to discuss the theoretical performance of a generic steam atmosphere dryer. This will provide a basis with which to compare the test's measured performance.

A steam atmosphere dryer is in effect a direct contact heat exchanger. The superheated atmosphere steam is to exchange its sensible heat to the latent and sensible heating of the wet feedstock slurry. The feedstock's water is evaporated via boiling heat transfer and subsequently sensibly heated to the dryer's discharge temperature. Some of the entrained water within the particle must be driven to the particle surface via a mass transfer mechanism. For simplicity, the combined heat and mass transfer processes will be combined into a single heat transfer coefficient (H) such that it satisfies the following principal dryer (i.e., heat exchanger) design equation:

$$\dot{Q}_{transfer} = H\ (Vol)\ \Delta T_{LM} \tag{7}$$

where:

$\dot{Q}_{transfer}$ = the rate of heat transfer within the dryer (Btu/hr).

ΔT_{LM} = The log mean temperature difference between the sensible steam heat source temperature and the (assumed) constant feedstock temperature of 212°F. (°F)

or ΔT_{LM} = $[(T_{Di} - 212) - (T_{Do} - 212)]/\ln (T_{Di} - 212/T_{Do} - 212)$.

Vol = total volume of the dryer (ft^3).

This expression is analogous to the conventional heat exchanger sizing design equation: $\dot{Q} = (UA)\ \Delta T_{LM}$. The use of the dryer volume (V_o) is expedient for Tecogen's analysis and it will be shown that the volume of the dryer is a direct measure of the number of atomized particles that must fill the dryer's volume and thus provide sufficient particle surface area for local particle heat transfer to occur. Without a sufficient wet particle surface area and/or a low local heat transfer coefficient (h_p) the dryer performance will be reduced, resulting in a lower heat transfer per dryer size. It is interesting to note here that conventional dryer manufacturing practice is to use the cross-sectional area (ft^2) of the dryer rather than its total volume in forming a dryer heat transfer coefficient. However, proportional relationships between conventional dryer volumes and cross sections exist with these manufacturers, and hence Tecogen's use of the volume (V_o) as the characteristic dryer dimension instead of the cross-sectional area for use in the overall heat transfer coefficient (H) serves the same purpose – the ability to size the dryer based on the desired dryer heat transfer rate and the discharge temperature.

The heat transfer sizing coefficient (H) as defined previously should not be confused with the local particle heat transfer coefficient (h_p) which is defined by the equation:

$$h_p = \frac{Q_{transferred}}{N_p\ A_p\ \Delta T_{LM}} \tag{8}$$

where:

N_p = the number of atomized particles whose water must be evaporated in the dryer.

A_p = the average surface area of a single water-feedstock particle. A spherical surface is assumed with the entrained water completely enveloping the feedstock particle.

ΔT_{LM} = the log mean temperature difference between the sensibly cooling superheated steam and the (assumed) constant water-feedstock particle at 212°F.

The value of N_p must be determined if the magnitude of the local heat transfer coefficient h_p is to be calculated and later compared with its value as determined from a form of the Ranz-Marshall formulation for the theoretical local particle heat transfer as given by:

$$h_{p,theor} = \frac{K_{gas}}{dp} [2 + 0.37 \, Re^{0.6} \, P_R^{0.333}] \tag{9a}$$

For small relative velocities between the gas stream and particle the Reynolds No. = 0 and thus:

$$h_{p,theor} = 2 \, K_{gas}/dp \tag{9b}$$

(i.e., no dependence on slurry flow rate or loading)

The value for N_p can be estimated by assuming that all of the water-feedstock particles created by the dryer's atomizer are destroyed via water evaporation in the dryer, leaving only the core particle to be transported by the steam to the cyclone where it is separated from the steam and collected.

The slurry atomizer produces the particles at a rate described by the following equation:

$$\left[\frac{\text{Particles}}{\text{hr}}\right]; \; \dot{N}_p = \frac{\dot{m}_p \, (W_i)}{\varphi_w \, \frac{4}{3} \, \pi \, (R_w^3 - R_p^3)} \tag{10}$$

where:

R_w = outside radius of the spherical water particle (ft)
R_p = radius of the spherical (core) feedstock particle (ft)
$\dot{m}_p$ = particle feedstock flow rate (lb/hr)
W_i = initial slurry wetness (lb_w/lb_p)
φ_w = density of water (lb/ft^3)

The time period for the water evaporation to occur is given by the time that it takes the hot steam to travel through the dryer multiplied by the IRIS dryer's recirculation ratio (R) or:

$$t_{drying} = \frac{V_o \times R}{\left(\frac{\dot{m}_{STM}}{\varphi_{STM}}\right)} \tag{11}$$

where:

V_o = the dryer volume (ft^3)

$\dot{m}_{STM}$ = the dryer's superheated steam flow rate (lb/hr)

Substituting these equations into Equation 8 results in:

$$\left[\frac{Btu}{hr-ft^2-{}^\circ F}\right];\ h_{p,meas'd} = \frac{\dot{Q}_{meas'd}\ (R_w^3 - R_p^3)\ \varphi_w}{(W_i)\ R\ (L_p)\ (V_o)\ (3)\ \varphi_{STM}\ (\bar{R}_w)^2\ (\Delta T_{LM})} \tag{12}$$

where:

$\bar{R}_w$ = $(R_w + R_p)/2$

L_p = dryer particle loading, m_p/m_{STM}

R = recirculation ratio (R)

W_i = initial slurry water content, $lb_w/lb_{particle}$

The ratio of $h_{p,meas'd/H}$ reduces to:

$$\frac{h_{p,meas'd}}{H} = \varphi_w\ (R_w^3 - R_p^3)/(3\ W_i\ L\ R\ \varphi_{STM}\ \bar{R}_w^2)$$

For the testing conducted at Tecogen, the following experimental values are known:

φ_w = 62.4 lb/ft^3

φ_{STM} ≅ 0.0346 lb/ft^3

D_w ≅ 139 microns

D_p ≅ 106 microns

W_i = 0.50

V_o = 0.859 ft^3

which requires that:

$$h_{p,meas'd} = H_{meas'd} \times \frac{0.195}{(L_p)\ R} \tag{13}$$

$$\text{or } H_{p,meas'd} = h_{p,meas'd}\ (L_p)\ R \times 5.13$$

In order to evaluate $h_{p,meas'd}$, it is necessary to know the IRIS recirculation ratio (R). However, this is not available directly from the hot test experimental results. It is possible to determine the maximum value that R must be before $h_{p,meas'd}$ has a value of less than $h_{p,theoretical}$ as given by Equation 9b ($2\ K_{gas}/d_w$). This calculation has been carried out for each test and the results are given in Table 6.4 and displayed in Figure 6.40. These values have a mean and average value of 11.8. This recirculation rate is relatively small corresponding to an average IRIS dryer collector efficiency of only 91.5 percent. A calculation of the theoretically necessary IRIS collector efficiency for the tests conducted at Tecogen is shown in Figure 6.41. This required IRIS collector efficiency should be within the capability of a full-size dryer where the effects of bench-scale sizing are not as influenced by the effects of the dryer particle loading as witnessed during the cold test experiments.

It is interesting to observe a plot of the particle's measured local heat transfer coefficient, assuming a recirculation ratio (R) = 1 is used in Equation 13. This plot is shown in Figure 6.42. Although the values of h_p shown in Figure 6.42 are high due to the assumption of R = 1, the near horizontal locus of the measured values does exhibit another important conformity of Tecogen's laboratory testing to conventional heat transfer theory; namely, the independence of the local heat transfer coefficient to particle loading. That is, the local particle heat transfer coefficient should be independent of the Reynolds number and hence not dependent upon slurry flow rate or particle loading according to Equation 9. Figure 6.42 verifies this conventional theory, for it appears that Tecogen's measured local heat transfer coefficient has a constant average value of 900 Btu/hr-ft^2-^{o}F (with a data scattering of +150 and –200 Btu/hr-ft^2-^{o}F) that is independent of the particle loading. This is even more dramatically displayed if Figure 6.42 is replotted using the corrected values of particle loading (as defined by Equation 15) as shown in Figure 6.43.

Dryer Effectiveness and Sizing Coefficients

If one selects the dryer's inlet steam temperature, the feedstock initial (W_i) and final (W_o) moisture content, and the dryer's desired discharge temperature, two distinct dryer performance curves may be identified that, used together, completely identify the heat transfer sizing coefficient (H) and thus define how the dryer will

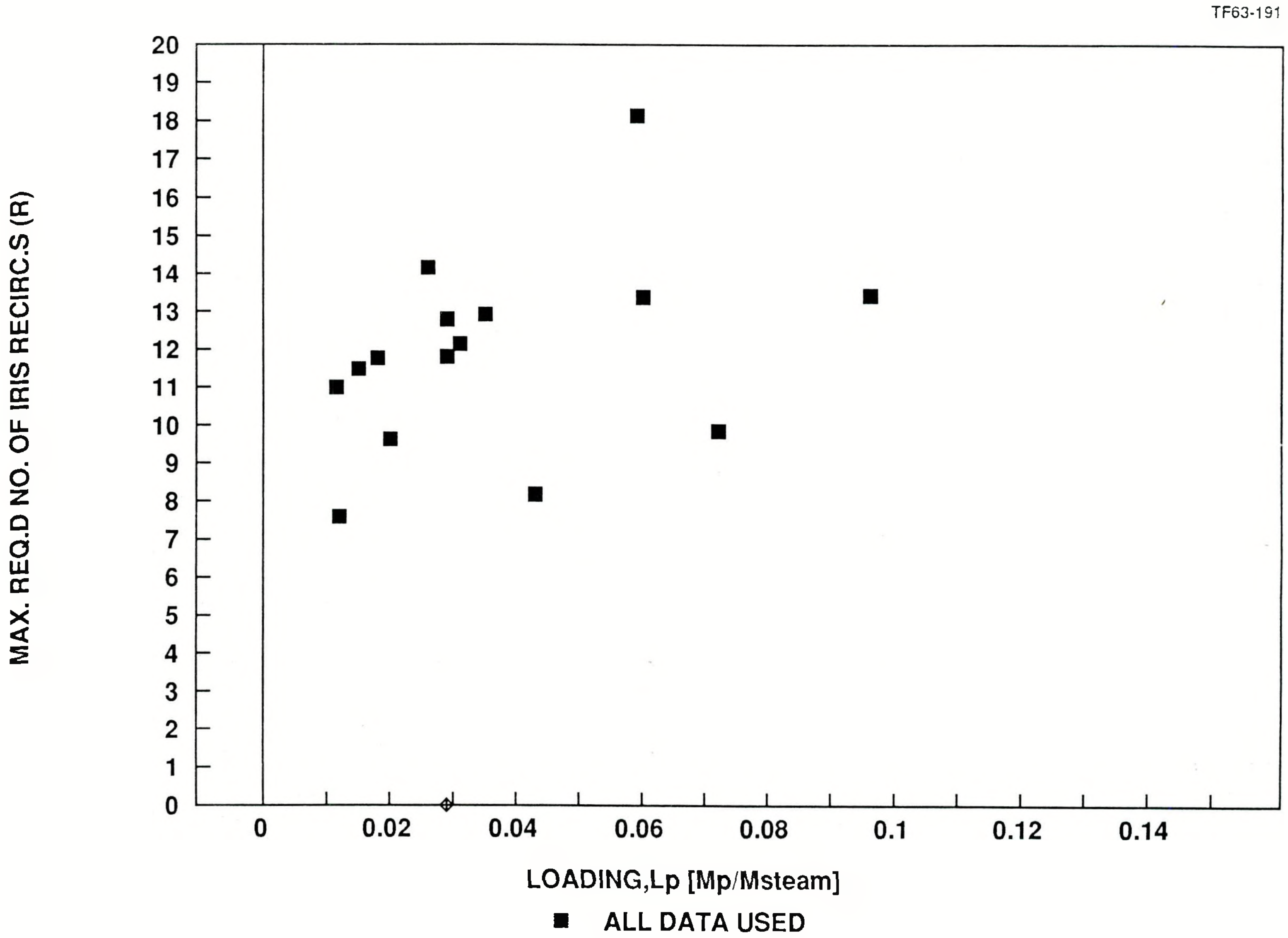


Figure 6.40 Steam Hot Test Data
Maximum Recirculation Ratio Requirements

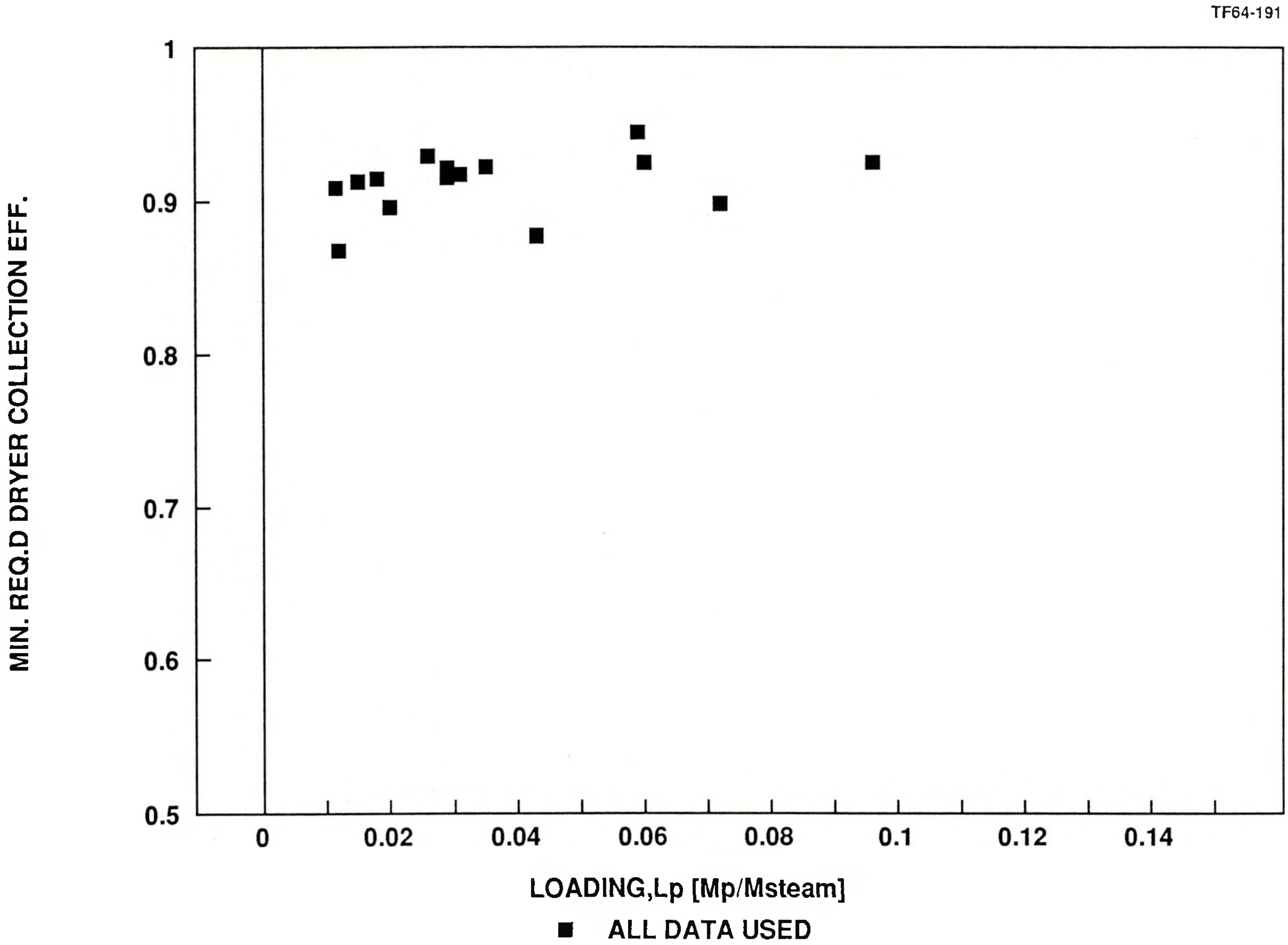


Figure 6.41 Steam Hot Test Data
Minimum Required Dryer Collection Efficiency

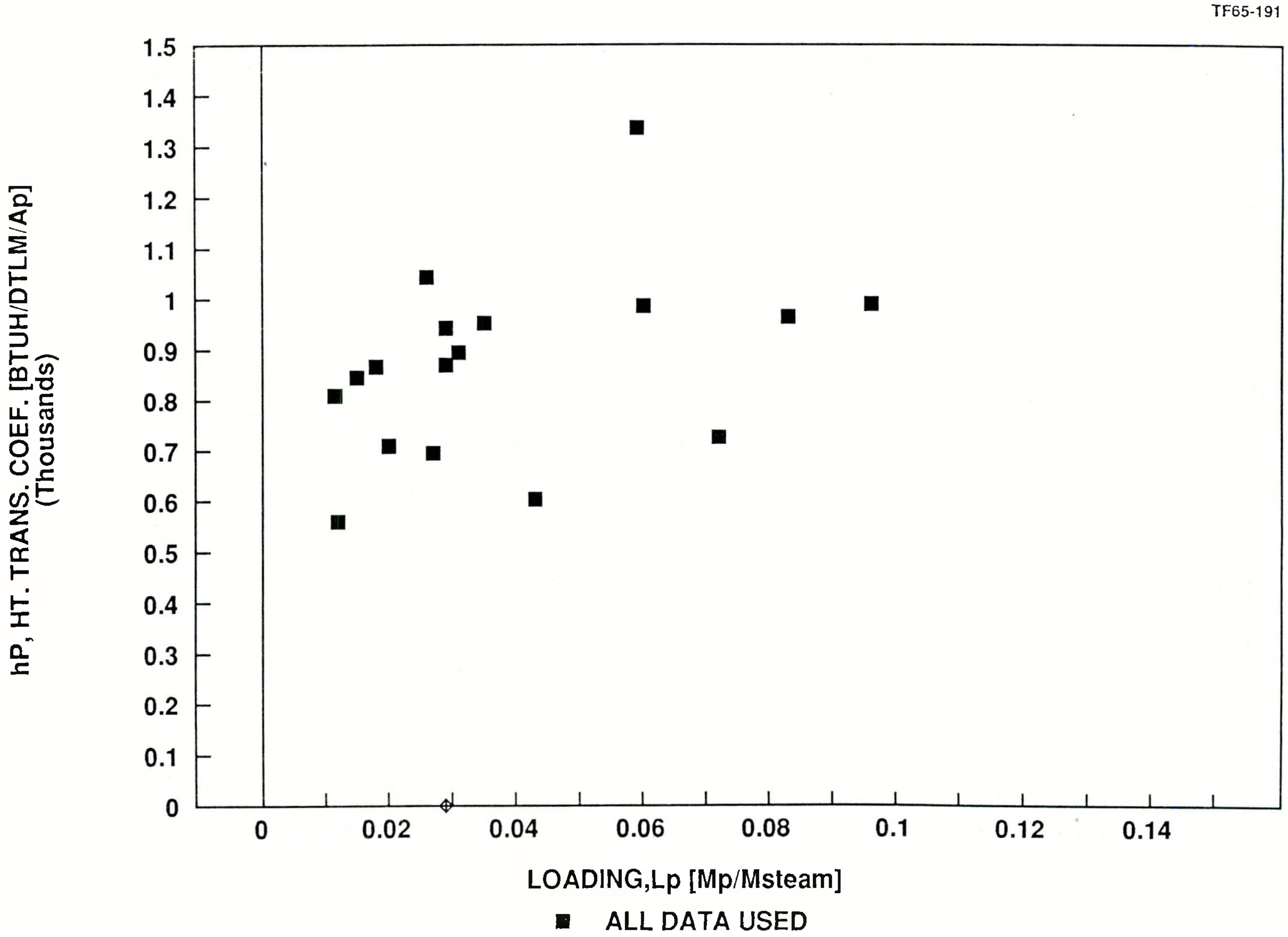


Figure 6.42 Steam Hot Test Data
Measured Heat Transfer Coefficient, hp, with Recirculation, R = 1

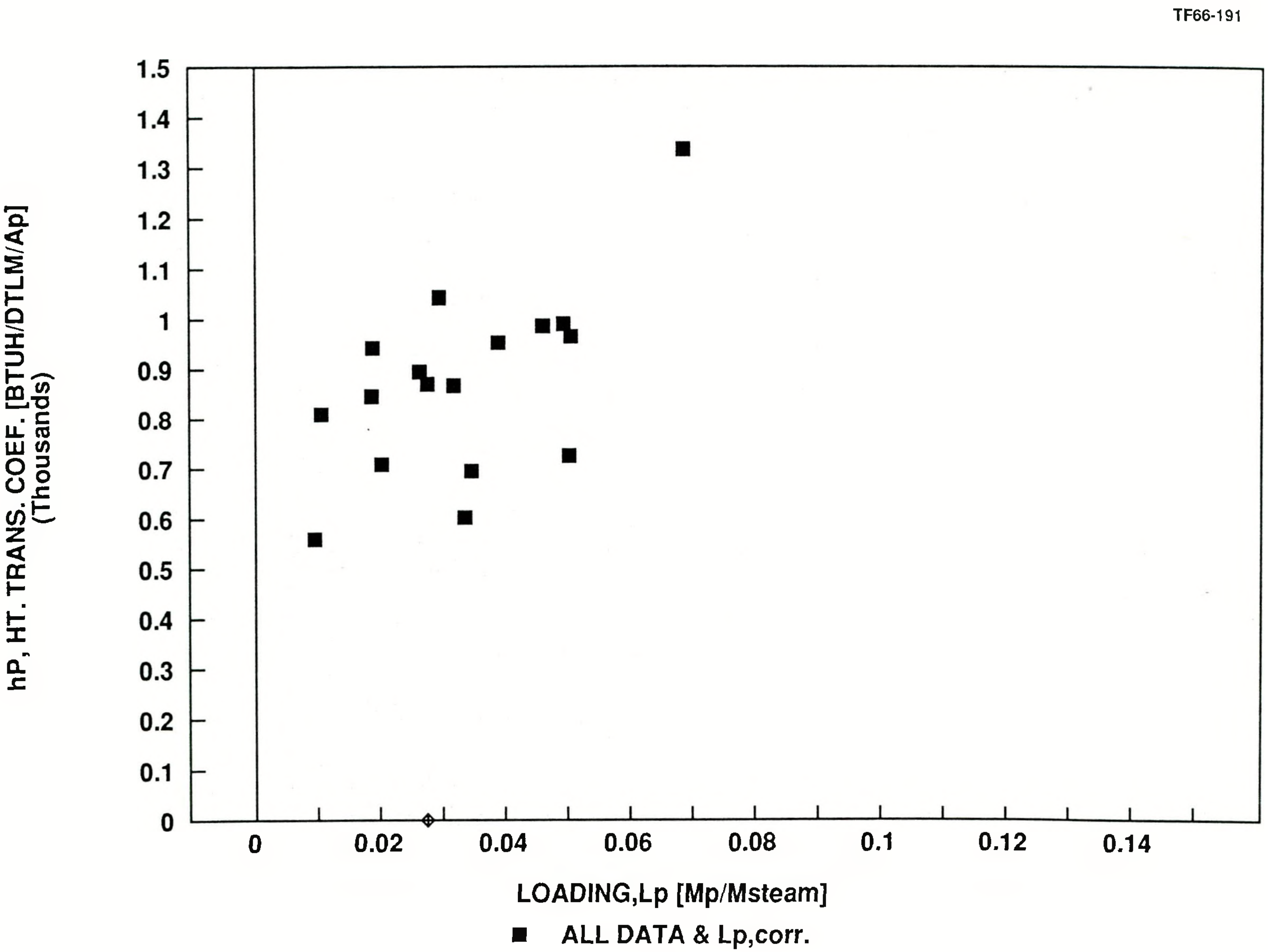


Figure 6.43 Steam Hot Test Data
Measured Heat Transfer Coefficient, hp, with Recirculation, R = 1

perform. These two theoretical performance curves are shown in Figures 6.44 and 6.45. These curves have been generated using the following dryer design conditions selected to correspond to those held constant during the testing of the bench-scale IRIS-type dryer at atmospheric pressure:

Steam Inlet Temp.	290°F
Steam Discharge Temp.	Varies from 290°F to 212°F (212°F is the minimum possible dryer discharge temp.)
Feedstock Initial Wetness	50 percent (dry basis)
Feedstock Final Wetness	0 percent (This is the minimum possible discharge wetness from the dryer and thus serves to identify the best dryer performance.)
Slurry Inlet Temp.	65°F
Slurry Feedstock Flow Rate	10 to 35 lb/hr

Figure 6.44 identifies the theoretical dependence upon the dryer particle loading (L) as a function of dryer discharge temperature and particle feedstock flow rate (m_p). The dryer particle loading (L) is defined as the ratio between the particle feedstock flow rate and the dryer steam flow rate that is used to both transport the slurry into the dryer as well as to evaporate the entrained particle moisture. The graphical relationship is defined by the equation:

$$L = \frac{\Delta h_{steam}}{\left[\Delta h_{particle} + (\Delta h_{latent} * \Delta W) + W_o \, \Delta h_{water} - \dfrac{Q_{wall\ conductor}}{m_p}\right]} \tag{14}$$

where:

ΔW = change in feedstock moisture
= $W_i - W_o$

W_o = outlet feedstock moisture = 0 (assumed)

Δh_{steam}, $\Delta h_{particle}$, Δh_{water} ≡ changes in the steam, particle, and entrained water enthalpy

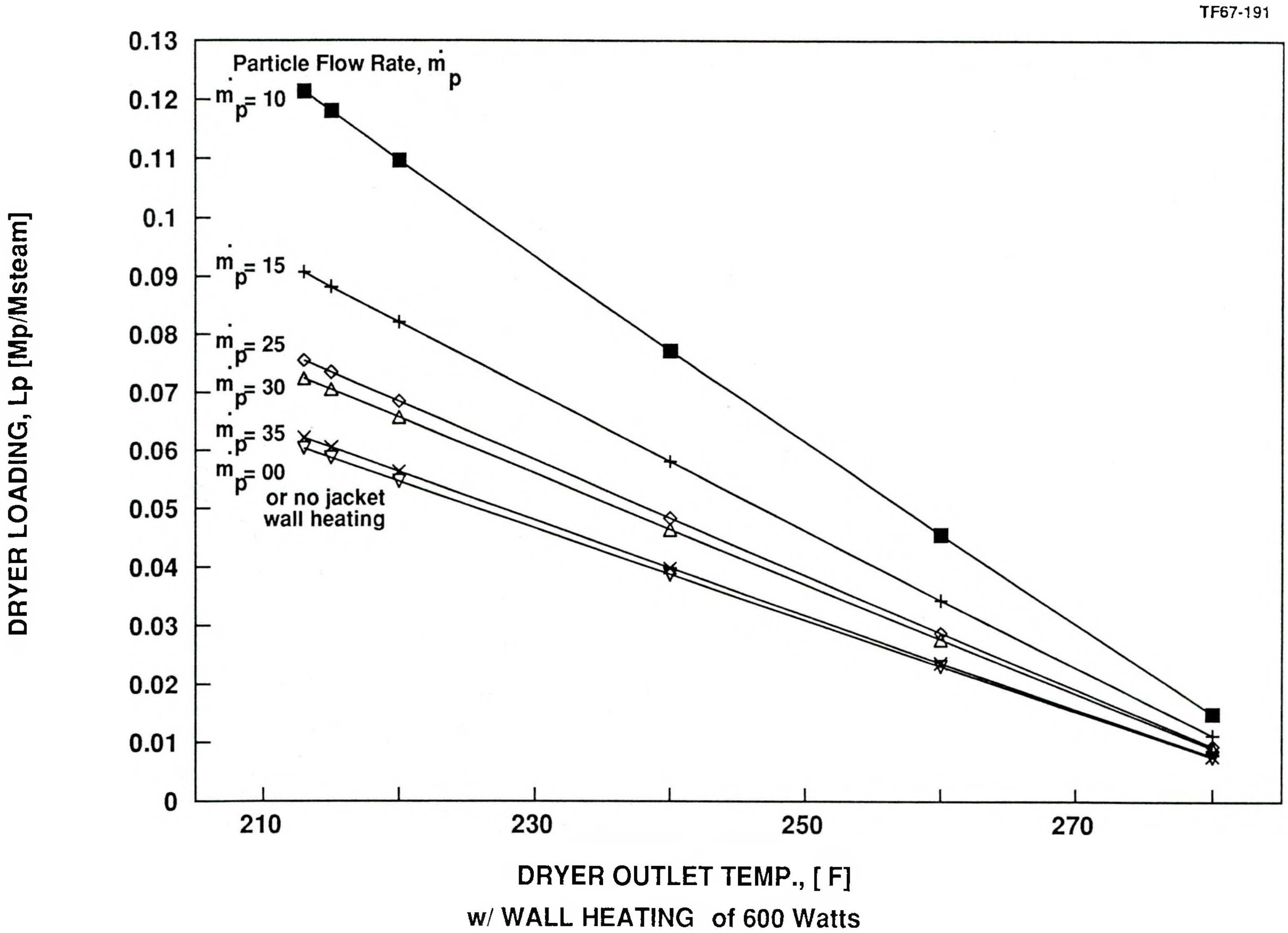


Figure 6.44 Steam Dryer Loading, Mp/Msteam vs. Dryer Outlet Temperature (F)

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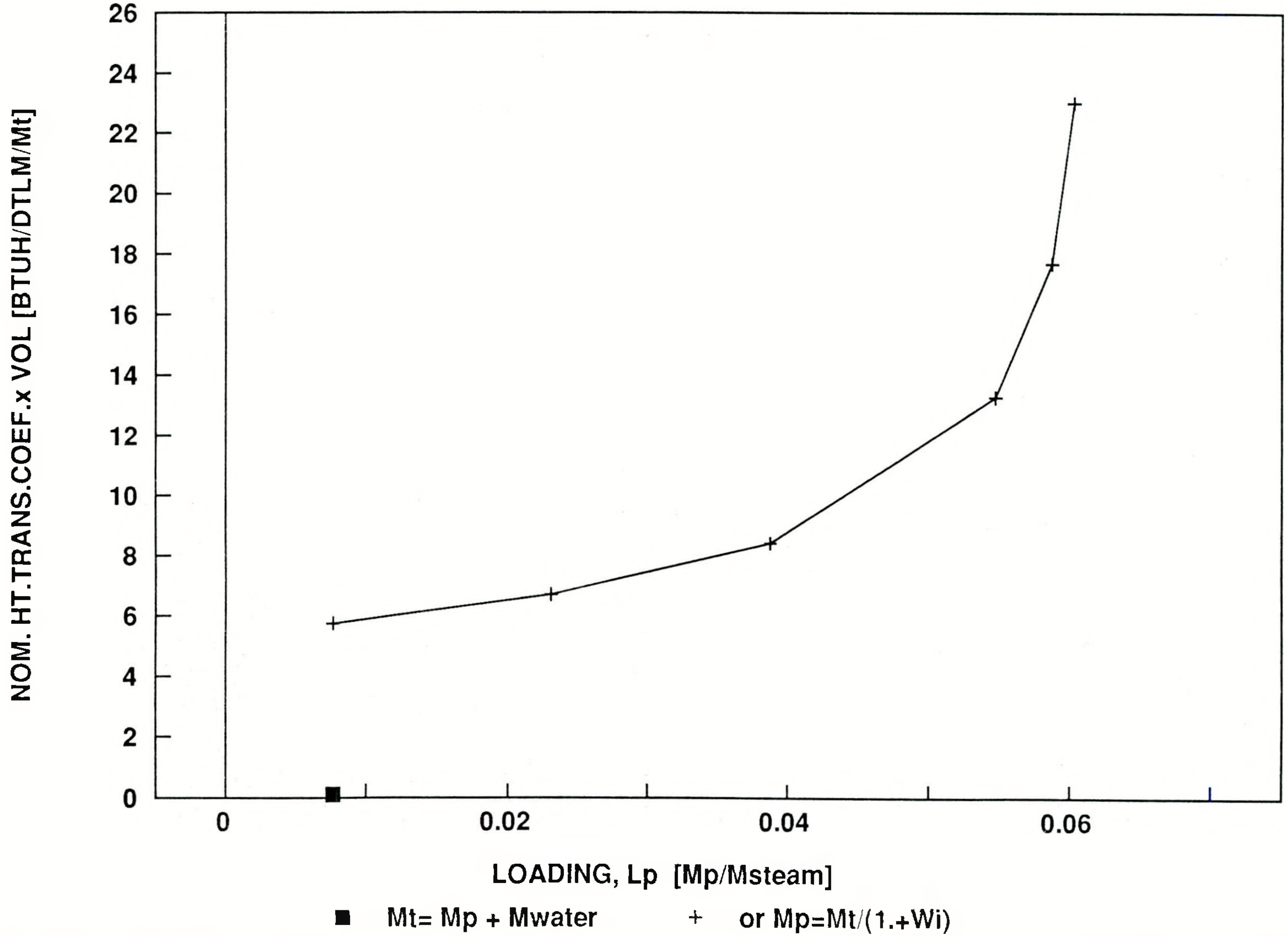

Figure 6.45 Steam Dryer Normalized Heat Transfer Coefficient x Volume vs. Dryer Loading, L (Mp/Msteam)

$Q_{wall\ conduction}/m_p$ = the amount of conduction heating available from a jacketed dryer with respect to the particle flow rate (m_p) used. (This quality is approximately 600 watts for these experiments.)

It is necessary to note the influence of conduction heat transfer (via a steam-heated, jacketed vessel) on the value of the dryer loading (L). Figure 6.44 clearly shows the need for less superheated transport steam (m_{steam}) per particle feedstock flow rate (and hence a larger value for (L)) if the latent heating of the water from the wet particles can be accomplished by wall conduction in addition to the convection heat transfer available from the sensible cooling of the superheated steam. Figure 6.44 uses a fixed heat conduction of 600 watts, corresponding to the magnitude of the wall heater tapes used during <u>some</u> of the drying experiments. Different values for the magnitude of the available conduction heat transfer would change these curves according to Equation 14. Certainly, the conduction heat input provided by the heater tapes in these bench-scale experiments is analogous (albeit much smaller in magnitude) to the conduction heat input that would be provided by the steam compressor in the indirectly heated steam atmosphere dryer design (as studied in Tasks 2 and 3 of Phase I). The question to be answered in the later phases of the project is whether higher particle dryer loadings can be maintained by the IRIS-type dryer while still enabling external recirculation. The external recirculation is necessary as a means of extending the particle's exposure time in the drying steam environment rather than requiring an increase in the size of the IRIS dryer.

Figure 6.45 identifies the normalized product of heat transfer sizing coefficient times dryer volume (V_o) as a <u>sole</u> function of the dryer loading (L). The <u>normalized</u> heat transfer coefficient – volume product ($H_N \times V_o$) is defined as the ratio of the heat transfer coefficient (H) (as defined earlier) multiplied by the dryer's volume divided by the total slurry flow rate, $\dot{m}_T$. The feedstock flow rate (m_p) can be determined from: $m_p = \dot{m}_T/(1 + W_i)$; where W_i is the initial feedstock moisture content. Thus, given a desired (or known) dryer discharge temperature and slurry flow rate, it is possible to determine the necessary product of heat transfer coefficient (H) x volume (V_o) required to perform the dryer's desired heat transfer duty.

For example, assuming a slurry flow rate of 15 lb/hr and an initial wetness of 50 percent (D.B.), the particle feedstock flow rate would be 10 lb/hr. If the dryer's desired discharge temperature is 230°F, then the necessary particle loading is approximately 0.094. However, in determining the magnitude of the normalized heat transfer sizing coefficient H_N from Figure 6.45, the particle loading (L_p) of 0.048 is used, which is obtained by referring to the lowest curve in Figure 6.44 that corresponds to a dryer operating without wall conduction heating. Using Figure 6.45 with a particle loading value of 0.048, a normalized heat transfer coefficient-volume of 10 Btu/hr-^{o}F-lb/hr is obtained. Given the slurry flow rate of 15 lb/hr, the dryer sizing heat transfer coefficient-volume of 15 x 10 = 150 Btu/h^{o}F can be determined, enabling the volume of the dryer to be known given either the dryer's measured or theoretically determined heat transfer coefficient (H) or h_p (Equation 13).

It must be recalled that Figures 6.44 and 6.45 were derived assuming a dryer inlet temperature (290^{o}F) and a slurry wetness (50 percent D.B.). Therefore, a dryer operating at other temperature and wetness conditions would require different curves to be used. Similarly, using the measured performance of a given size dryer, it is possible to use Figures 6.44 and 6.45 to discern how well a given fixed size steam dryer (i.e., a direct contact heat exchanger) performed in comparison to its theoretical expectations. As indicated previously, the dryer inlet steam temperatures, inlet wetness, and dryer volume used to generate Figures 6.44 and 6.45 correspond to the test conditions used in Tecogen's steam atmosphere dryer hot test experiments. Therefore, Figures 6.44 and 6.45 form the basis for determining how well the bench-scale IRIS steam atmosphere dryer performed in comparison to the dryer's required performance if it is to achieve a desired discharge temperature. Thus, the principal question to be addressed by these hot steam bench-scale size dryer experiments – What are the heat transfer characteristics of the IRIS dryer and how do they compare with Tecogen's computer model? – now has a means for objective evaluation. Furthermore, the measured heat transfer sizing coefficient (H) can be used to evaluate the dryer size requirements for larger, and hence more useful steam atmosphere dryers, assuming that the heat transfer coefficients measured in these experiments are scaleable (i.e., can be extrapolated) for the larger scale IRIS dryers.

6.4.6 Measured IRIS Dryer Performance Results

Using the measured test data summarized in Table 6.4, a good characterization of the heat transfer performance of the bench-scale IRIS dryer has been made and is represented in the following section. The following graphs use all measured data with a single exception: a corrected particle loading, or $L_{p,corr}$. It has been found to be worthwhile to consider a correction to the measured dryer steam mass flow rate (m_{steam}) and thus a recalculation of the measured particle loading according to the correction formula:

$$L_{p,corr} = L_{p,meas'd} \times \left[1 - \frac{\%\ \text{Ht. Trans. Diff.}}{100}\right] \tag{15}$$

The term "% Ht. Trans. Diff." appears in Table 6.4 and accounts for the discrepancy found in the heat balances (albeit small) from the experiments between the slurry heat input and the superheated heat output. The correction formula modifies the measured steam flow rate up or down forcing the heat balance and thus compensating for, perhaps, the variations in the steam flow rate that could have affected a true measurement of steam mass flow rate. The clarifying effects of this particle loading correction is evident by observing Figures 6.46 and 6.47 – a plot of particle loading vs. dryer discharge temperature. Figure 6.46 used the uncorrected particle loading, L_p; Figure 6.47 used the corrected particle loading, $L_{p,corr}$. The scatter in the data is greatly reduced by employing the corrected particle loading. Corrected particle loading is used in a similar manner in several other figures that follow and these figures are always clearly labeled.

An analytical interpretation of Figure 6.46 remains, however, and is given here. Figure 6.46 is a superposition of the dryer's measured performance on Figure 6.44, which was presented earlier in this section. Recalling that each straight line represents the effect of wall conduction heating (in this case, 600 watts) on the particle loading vs. dryer discharge temperature, this graph suggests that there has been some (albeit small) measurable effect of the 600-watt wall heaters on the slurry feedstock heating. This is particularly evident when each point is identified as to whether the dryer wall heaters were operating (designated as "H") or not operating (designated as "NH"). This is more evidence that wall (jacket) heating will have a positive effect in contributing to the slurry drying, an effect that is critical to the success of the indirectly heated steam atmosphere dryer system. A simpler interpretation of Figures 6.46 and 6.47 will at least indicate that the locus of measured data does conform (with allowances given for some experimental scatter) to a straight line with a slope similar to what would be expected of the dryer's performance, as shown previously in Figure 6.44.

Using the data from Table 6.4, it is also possible to present a comparison of the measured product of normalized heat transfer-volume or $(H_n \times V_o)_{meas'd}$ vs. particle loading, L_p with the predicted values for $(H_n \times V_o)_{predicted}$. This has been done in Figure 6.48 (using measured particle loading, $L_{p,meas.}$) and Figure 6.49 (using corrected particle loading, $L_{p,pred.}$). Once again, the use of the corrected particle loading decreases the experimental scatter and allows for a better comparison to predicted dryer performance. From Figure 6.49, it appears that the measured and predicted performances are relatively close. It was predicted that the bench-scale IRIS dryer could achieve a discharge temperature of 220°F under fully

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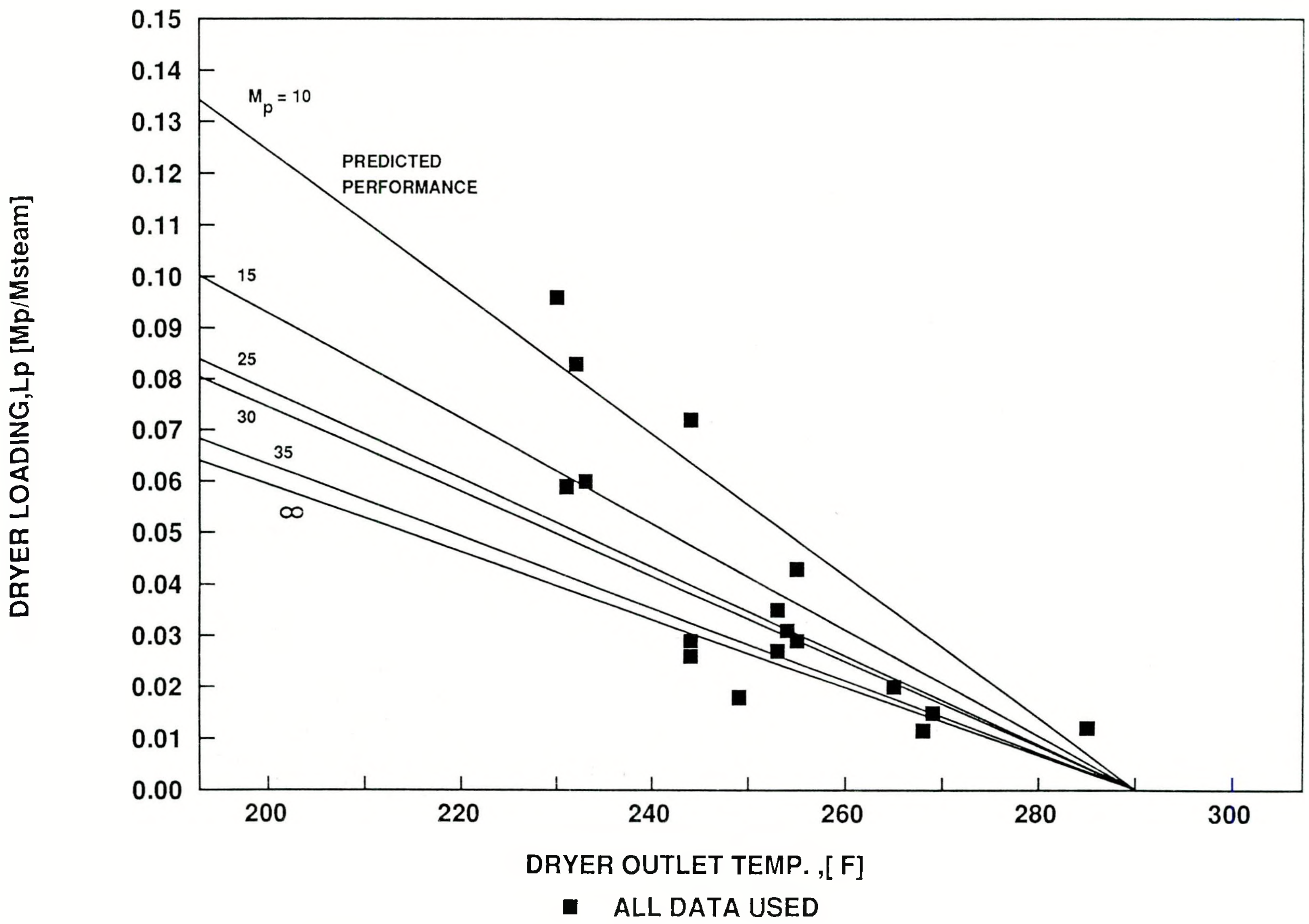

Figure 6.46 Measured Steam Hot Test Data
Dryer Loading vs. Dryer Outlet Temperature

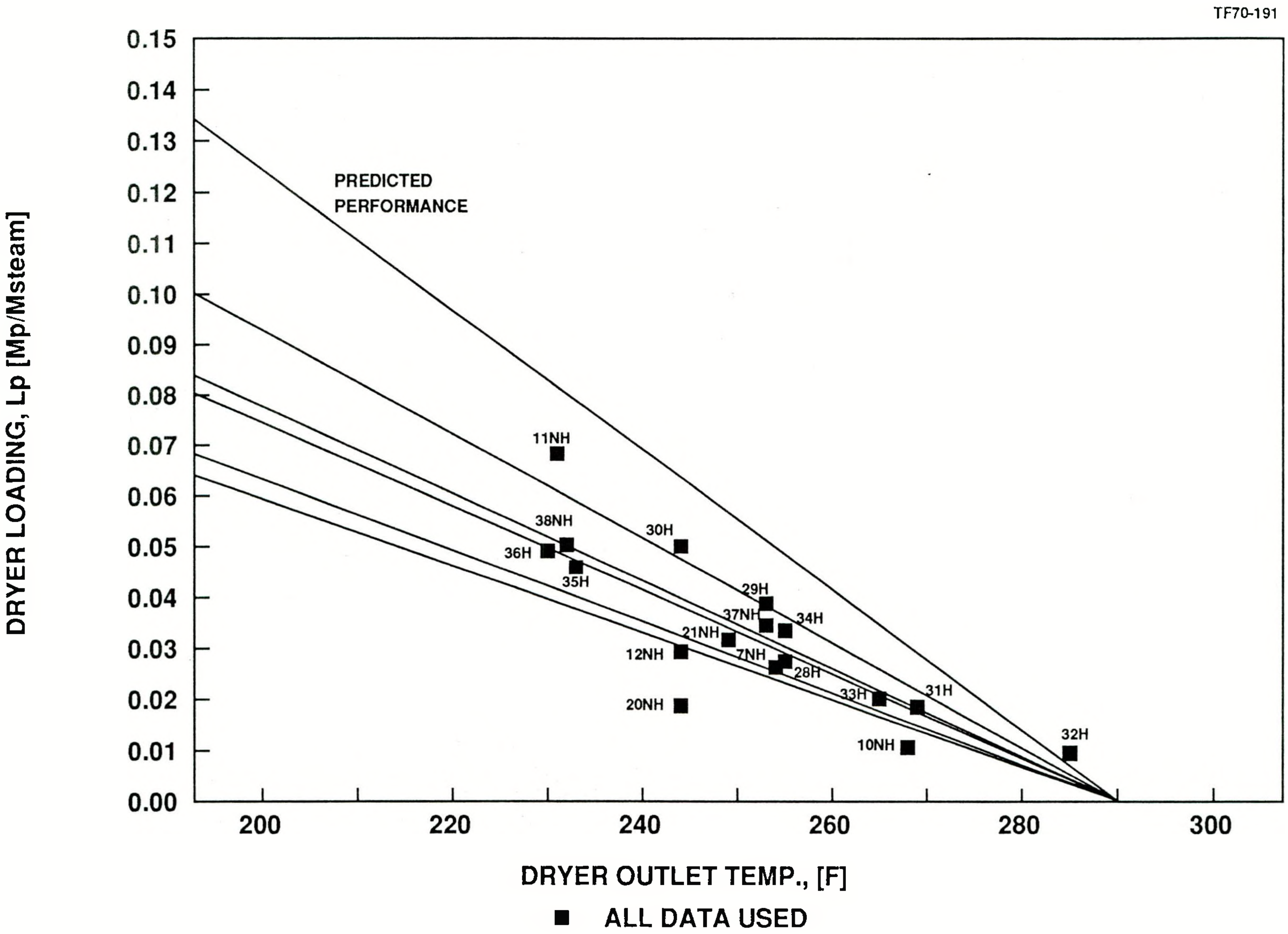

Figure 6.47 Steam Hot Test Data with Corrected Lp

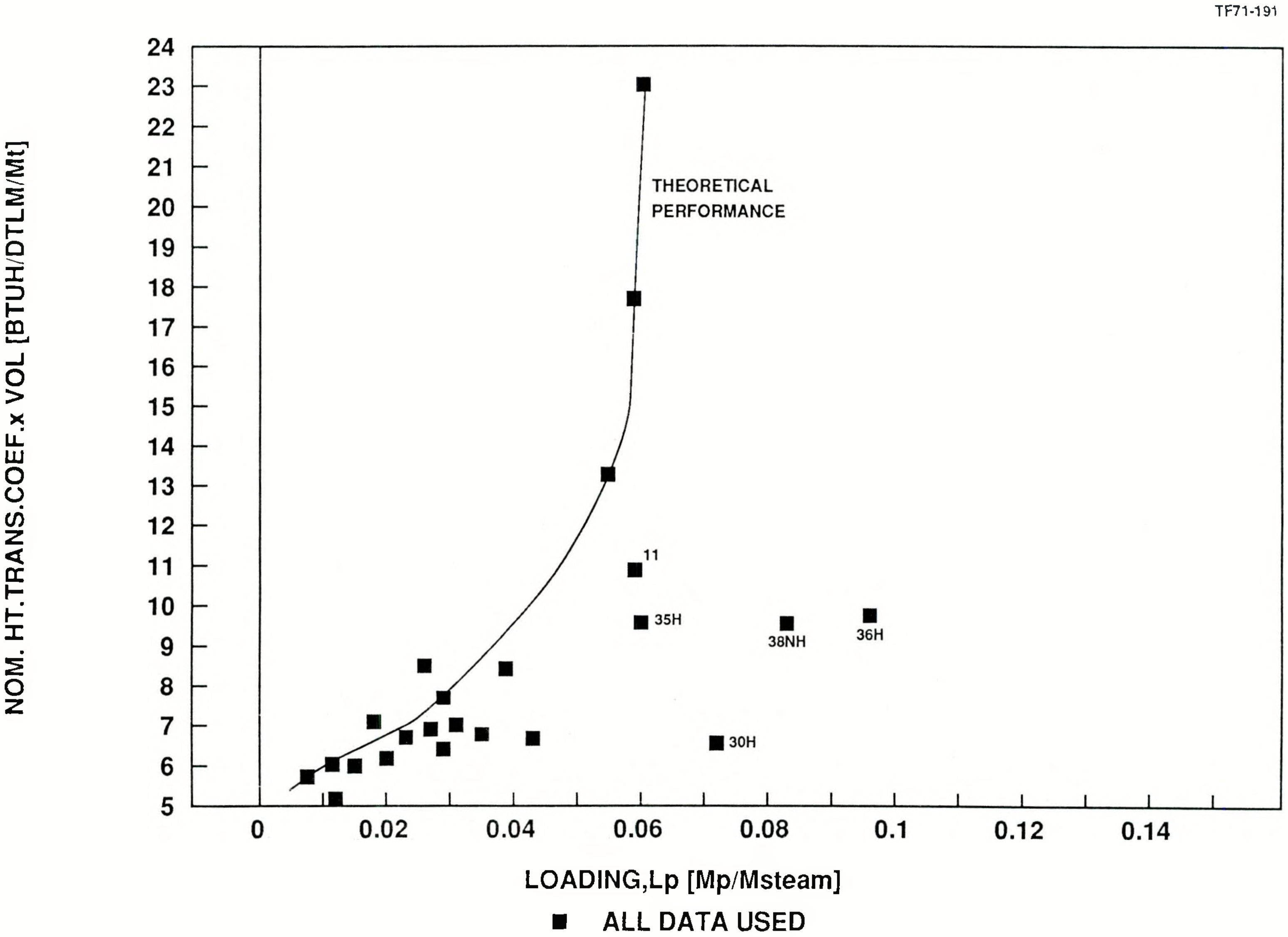

Figure 6.48 Steam Hot Test Data
Normalized Heat Transfer Coefficient, Hn x Volume

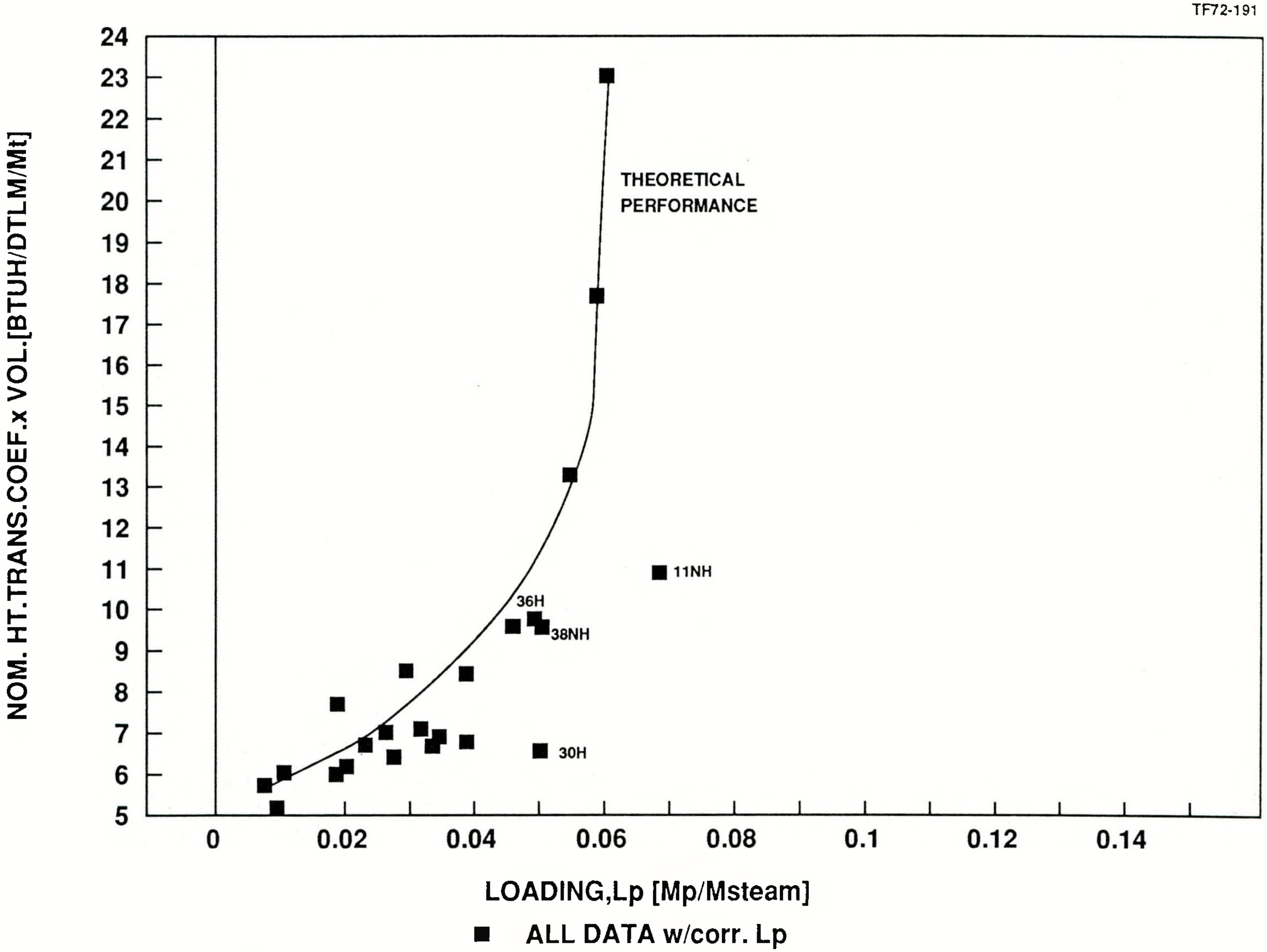

Figure 6.49 Steam Hot Test Data
Normalized Heat Transfer Coefficient x Dryer Volume

loaded conditions. This corresponds to a particle loading of (approximately) 0.05 (from Figures 6.44 or 6.46). Using Figure 6.49 (with corrected particle loading considered) it is clear that this has been achieved. It is interesting to note that at particle loadings greater than 0.05 the bench-scale IRIS dryer begins to lose some of its (heat exchange) effectiveness. For example, the data points labeled 30 H and 11 NH on Figure 6.49 or the data points labeled 30 H, 35 NH, 36 H, and 38 NH on Figure 6.48, reveal lower H_n x Vol products than expected. This is due to the higher dryer discharged temperatures actually measured in the tests which result in larger log mean temperature differences for these data points. Consequently, the values of H_n x Vol are lower. To achieve higher values of H_n x Vol would require therefore either higher heat transfer sizing coefficients (H) or a larger dryer. Fortunately, the direct and indirect steam atmosphere drying system analysis conducted in Tasks 2 and 3 (systems analysis and design) and Task 6 (systems cost analysis) assumed a discharge temperature of only 220°F. Thus, the recalculation of the steam dryer energy savings and cost analysis using these actual measured heat transfer performance data should show only minor differences (if any) based on Figures 6.48 and 6.49.

An alternative means of comparing the measured dryer performance with its expected performance is to use the temperature effectiveness relationships for the steam dryer treated as an evaporator. That is, the measured temperature effectiveness for the evaporator can be identified from:

$$\varepsilon_{meas'd} = \frac{T_{D_i} - T_{D_o}}{T_{D_i} - 212} \tag{16}$$

where:

$T_{D\,i}$ = IRIS dryer steam inlet temperature
$T_{D\,o}$ = IRIS dryer steam discharge temperature
212 = evaporation temperature of entrained feedstock water at atmospheric dryer pressure

However, the theoretical effectiveness can be discerned using:

$$\varepsilon_{pred} = 1 - \exp\,(-DNTU) \tag{17}$$

where:

DNTU = $[(H_n \times Vol)_{pred.} \times (1 + W_i) \times L_p]/C_{p\,steam}$
and $(H_n \times Vol)_{pred.}$ is obtained from Figure 6.45

Plots of the measured and predicted effectiveness using Equations 16 and 17, respectively, are given in Figures 6.50 and 6.51 (using corrected steam flow rates).

These plots also clearly display how well the bench-scale IRIS dryer performed in comparison to the predicted performance. For example, the measured dryer effectiveness matches the predicted effectiveness for dryer NTU (DNTU) less than or equal to approximately 1.5 (or $\varepsilon \cong 0.78$). Measured performance above a DNTU of 1.5 was less than the predicted performance. Using Figures 6.50 or 6.51, a bench-scale IRIS dryer would need to have a DNTU of approximately 2.2 in order to achieve a dryer discharge temperature of 220°F (as was assumed in the Tasks 3, 4, and 6 analysis). A maximum measured dryer effectiveness of 0.775 implies that the bench-scale dryer's H_n x Vol must be 47 percent larger. To be conservative, a recalculation of simple payback (Task 6) for the steam atmosphere dryer system should perhaps use a larger (and hence more expensive) dryer. It is, however, not absolute certain at this point in Tecogen's dryer study if the scaling laws for the heat transfer coefficient x Vol. would not have provided a better dryer performance had a larger bench-scale size IRIS dryer been used. In that event, the prototype size IRIS dryer used in Task 6 may be adequate.

Certainly, the better design procedure would be to identify the effect that the dryer discharge temperature has on the energy efficiency of the direct and indirect steam atmosphere system and weigh this effect against the need for additional dryer volume. For example, if a dryer discharge temperature of 230°F (and not 220°F) is chosen as the design point, then the required dryer effectiveness of 78 percent can be met by the present IRIS dryer design with an increase in energy consumption of only 1 to 2 percent. Clearly, the proper design decision would be to use a higher stack discharge temperature rather than to increase the dryer volume. In either case, the hot testing results have provided Tecogen a means of confidently designing the IRIS dryer's volume in order to obtain a desired discharge temperature.

The final presentation of the measured IRIS dryer performance is given in an additional graph and illustrated in Figure 6.52. Figure 6.52 presents the measured heat transfer sizing coefficient (H) (for use in Equation 7) as a function of corrected particle loading, $L_{p,corr}$. This graph shows how the heat transfer is expected to vary with respect to the dryer's particle loading. A principal question remains, however. Can this measured normalized/heat transfer sizing coefficient be interpolated to larger dryers (i.e., how does dryer scaling affect this coefficient)?

The dryer industry is unanimous in their opinion that such scaling laws can only be verified by building dryers that are close to if not exactly the size intended for the industrial drying duty under consideration.

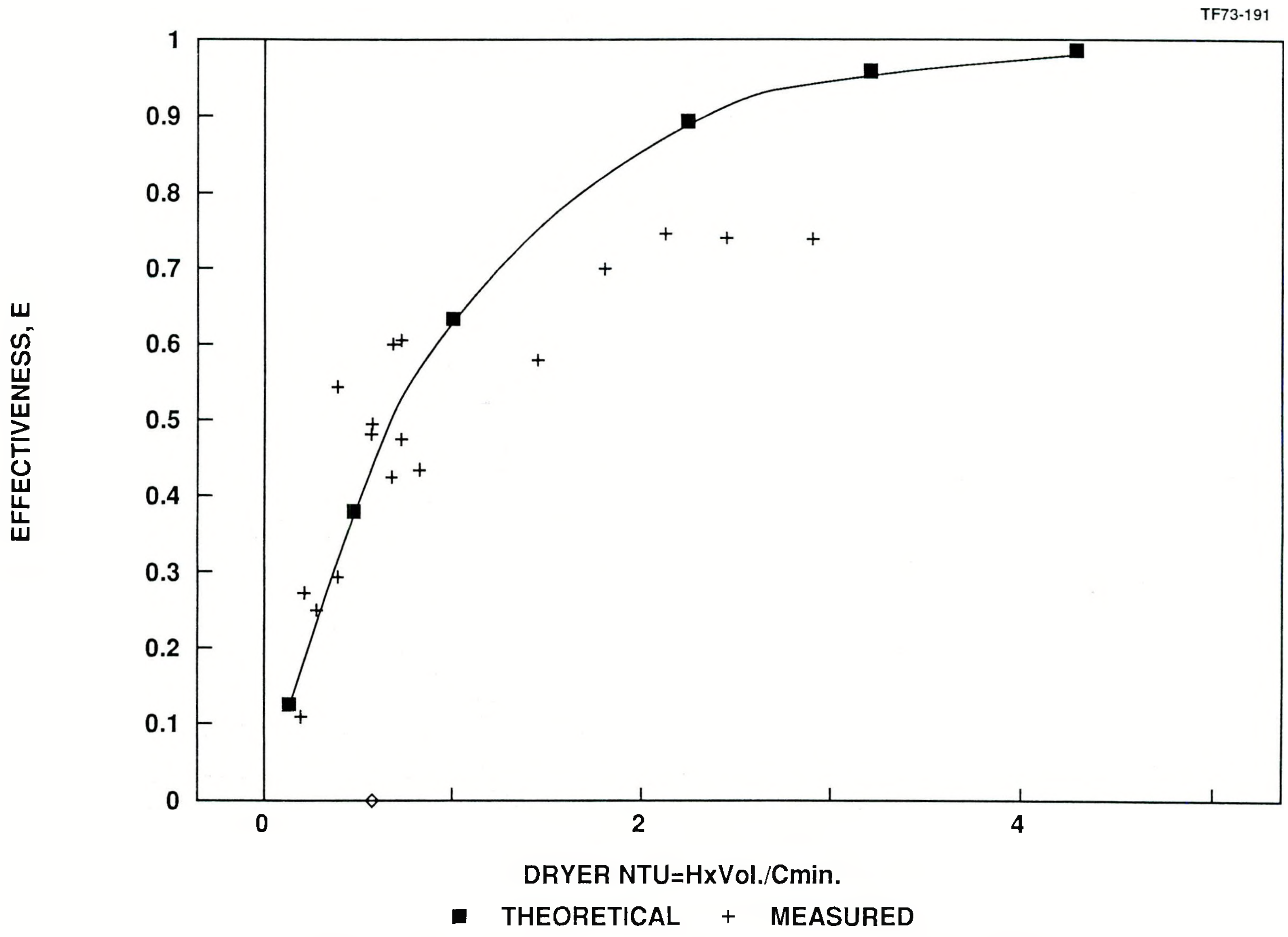

Figure 6.50 Steam Hot Test Data: Dryer Effectiveness Measured vs. Theoretical; All Data Used

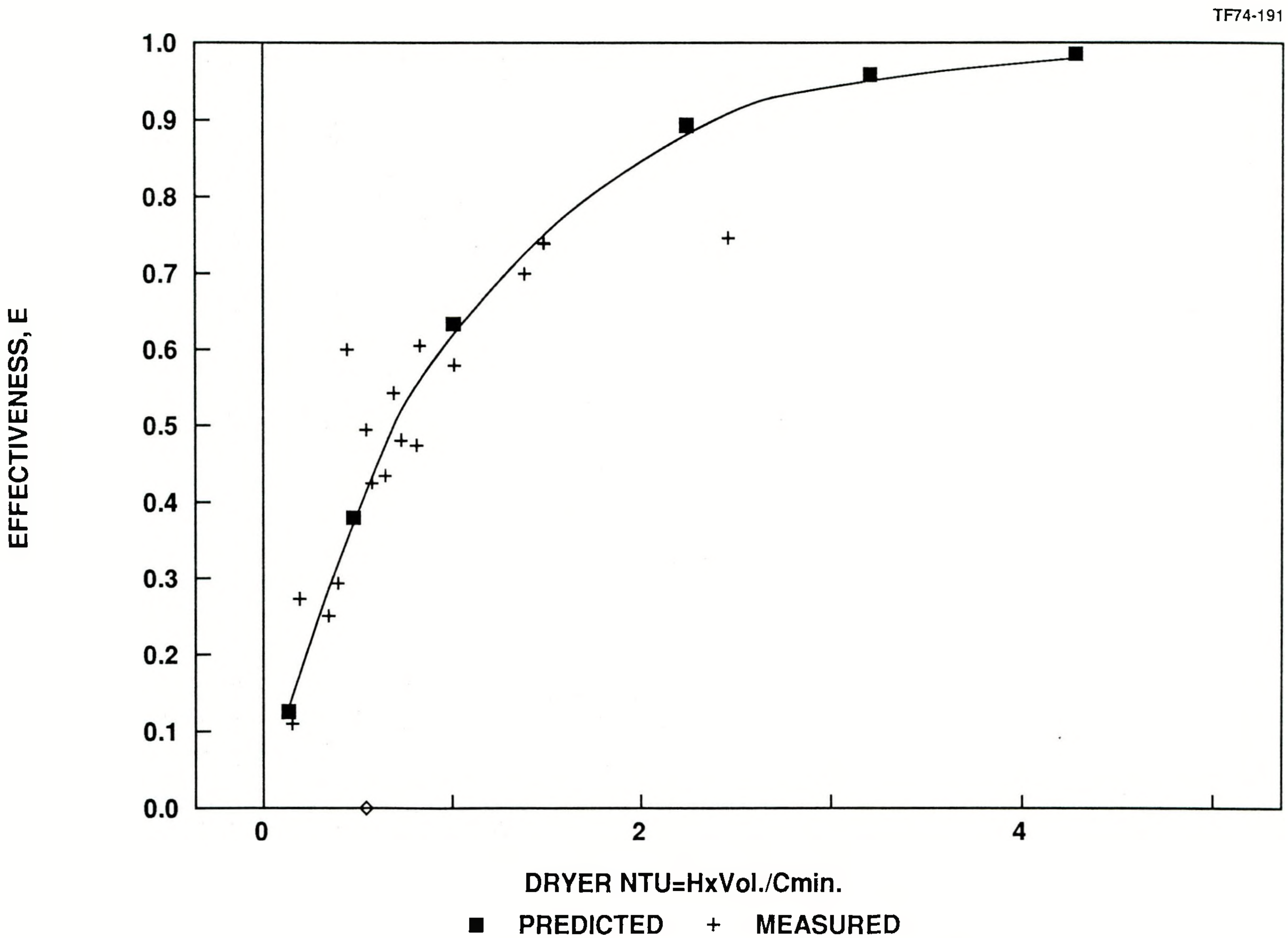

Figure 6.51 Steam Hot Test Data: Dryer Effectiveness Measured vs. Theoretical; Using Corrected Msteam

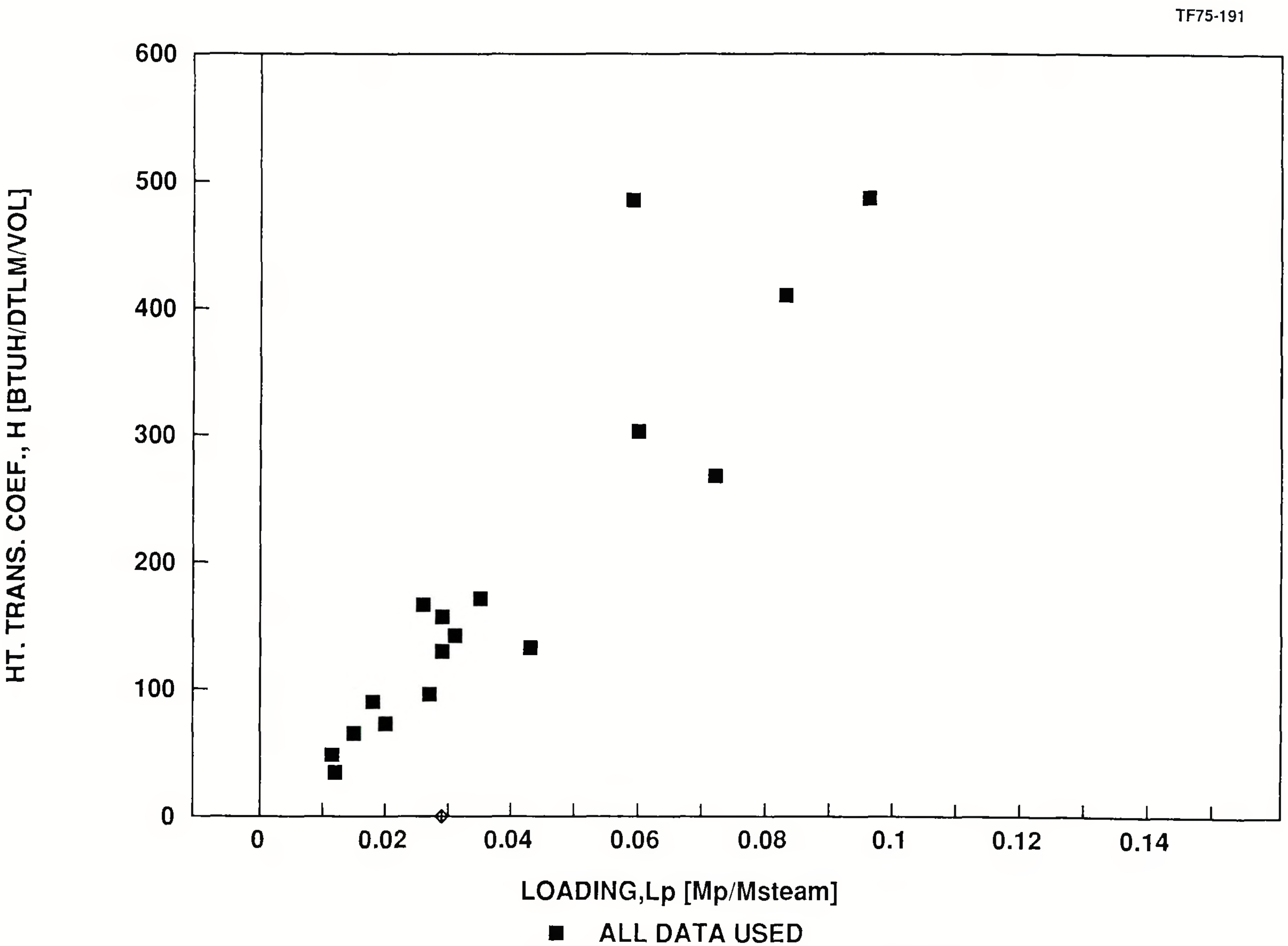

Figure 6.52 Steam Hot Test Data with Corrected Lp

It is of interest to calculate the equivalent dryer heat transfer coefficient from published data of conventional air (spray and flash) dryer systems in an attempt to discern the effects of dryer size on heat transfer coefficients, etc. This comparison has been done and is given in Table 6.5.

Apparently, the local particle heat transfer coefficient (h_p) (item c) is consistently lower for actual pneumatic dryer systems than the predicted values. The normalized heat transfer coefficient (H_n) (item a) is clearly much lower in actual dryers than in Tecogen's experiment. This is due to the need for very large volumes in present day dryers. However, the manufacturer's conventional heat transfer coefficients (items d and e) are more comparable to Tecogen's IRIS dryer. Perhaps another conclusion that can be drawn from Table 6.5's comparisons is that the conventional wisdom of not depending upon dryer size, scaling laws to size larger dryers using results from small dryer tests seems to be confirmed.

6.4.7 <u>Discussion of Test Results</u>

The steam hot tests conducted during Phase I demonstrate the ability of the IRIS dryer to dry a slurry of clay and water. The clay powder slurries, mixed to a typical water-feedstock mass ratio of 50 percent, were dried to an average value of 4 percent across the tested flow rate range – a range that would normally be encountered by a dryer of this size. This average outlet dryness is lower than the 5-percent dryness that was used consistently in all of the previous systems analysis and energy cost savings studies. Thus, the system energy and cost savings identified in these earlier analyses are still applicable. The dried powder did not suffer any physical distress, and according to the project partner APV Crepaco, Inc., the dried clay powder is of acceptable quality.

Testing is continuing in order to dry several temperature-sensitive materials (coffee creamer, a sugar substitute, and brewery yeast). At this time, there is confidence that the steam atmosphere dryer is conducive to drying these feedstocks if the correct particle-to-water mixture ratio were used and if the proper operating temperatures could be maintained. This confidence is due in part to the confirmed successes achieved by other independent researchers who have attempted steam atmosphere drying. The steam drying successes identified in Table 3.7, for example, offer compelling evidence of past successes with temperature-sensitive feedstocks and are a source of encouragement for believing Tecogen's future testing will produce similar results. A continuation of testing with these temperature-sensitive products with a larger IRIS dryer that could operate in a range of temperatures and pressures, with slurries of various mixture ratios and with different atomizers, is suggested for the next phase of the steam atmosphere dryer project.

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TABLE 6.5

COMPARISON OF STATE-OF-THE-ART PNEUMATIC DRYERS AND COMPUTER MODELS WITH TECOGEN'S HOT TEST RESULTS

REFERENCE:	APV CREPACO,INC. HEAT-SENSITIVE FEEDSTOCK		APV CREPACO, INC. NON-HEAT SENSITIVE		TECOGEN INC. TEST NO. 29H NON-HEAT-SENSITIVE	TECH. PAPER BY W.H. GAUVIN AND M.H. COSTIN (Tecogen Ref. No. 2)		p. 218, "INDUSTRIAL DRYING" by Williams and Gardner
	Spray/Air	Spray/Air	Spray/Air	Spray/Air	IRIS/Steam	Spray/Air	Spray/Steam	Spray/Air
Dryer Type/Medium								
Dryer Feedstock Cap. (lb/hr)	10,000	40,000	10,000	40,000	14.44	2,200	2,200	500
Initial Wetness (LBw/LBp)	0.5	0.50	0.5	0.50	0.5	2.333	2.333	3.75
Final Wetness (LBw/LBp)	0.05	0.05	0.05	0.05	0.124	0.1	0.1	0.0526
Dryer Inlet Temp. (F)	482	482	932	932	290	711	640.4	230
Dryer Outlet Temp. (F)	212	212	257	257	253	320	356	100
Wet Bulb Temp. (F)	160	160	160	160	212	140	212	95
Dryer Residence Time (Sec)	24	24	24	24				
Log Mean Temp. Diff. (F)	148	148	325	325	58	339	261	39
Dryer Heat Trans. (Btuh)	5,437,250	21,749,000	5,535,867.5	22,143,470	8,434	6,887,353	6,344,713	318504
Dryer Size								
Diameter (ft)	28.2	38.7	22.3	32.8	0.5	31.2	32.8	15
Height (ft)	57.4	78.7	44.3	66.3	4.17	46.75	49.2	31.8
Dryer Volume (cu. ft)	10,739	27,730	5,183	16,781	0.859	15,890.35	18,462.53	1683
Dryer Mean Cross-Section Area (sq. ft)	312	588	195	422	0.2	764	845	88
Dryer Heat Transfer Coef.s								
a. Norm. Ht. Trans. Coef. (Hn) (Btuh/Vol/DTLM/Mt)	0.00023	0.000088	0.00022	0.000067	7.87923	0.00017	0.00018	0.002019
b. Sizing Ht. Trans. Coeff. (H) (Btuh/Vol/DTLM)	3.4	5.3	3.3	4.1	170.7	1.3	1.3	4.8
c. Local Particle Ht. Trans. (hp, meas'd)(Btuh/part. Ap/DTLM) (hp, theory = 2 x Kg/Dp)	3.7 104.1	3.7 104.1	1.7 138.8	1.7 138.8	9.5 (with R=100) 190.2 (with R=5) 74.9 hp, theory	2 179.7	2.4 96.9	2.2 138.8
d. Mfg.'s Conventional Ht. Trans. Coef. (Btuh/Across.)	17,419.7	36,998	28,362	52,439.5	42,975.8	9,013.1	7,512.7	3606.6
e. Mfg.'s Conventional Dryer Duty (LBm/hr/Across.)	16.0	34.0	25.6	47.4	36.8	6.7	6.1	21.2

The steam hot testing provided invaluable data that was used to quantify the heat transfer performance of the IRIS steam dryer. The heat transfer analysis provided in this section demonstrates that the heat transfer coefficients measured for the IRIS dryer are comparable to what was predicted by Tecogen's IRIS dryer heat transfer models and thus also conform to theoretical formulations of the local particle heat transfer coefficient. A comparison of these measured results with actual spray dryer performances as quoted in published articles does indicate that the measured local particle heat transfer coefficient ($h_{p,meas'd}$) and the normalized heat transfer sizing coefficient (Hn) for the IRIS bench-scale dryer are higher than what is apparently available from full-scale size air dryers. Actual heat transfer and drying tests with full-scale dryer systems must be performed if a high level of confidence in the results is to be realized. For example, during the testing of this bench-scale size IRIS the clay powder was observed to clog the dryer at its inlet where the slurry injector nozzle was installed. This clogging occurred because of the relatively small diameter of the IRIS dryer in proportion to the large plume of the atomizing steam jetting from the steam nozzle. If a larger bench-scale IRIS model were to have been used this clogging could have been either eliminated completely or certainly greatly reduced. Thus, given the industry's rule of thumb, coupled with the demonstrated drying success of the IRIS dryer, a larger scale IRIS dryer with recompression needs to be designed, built, and tested in the next project phases.

The hot tests that were conducted did confirm that the IRIS dryer was certainly able to meet and/or exceed the particle retention times needed to completely dry the particle. For example, the testing indicated that if the IRIS dryer were to be able to sustain at least 10 to 14 external recirculations (i.e., have a recirculation ratio of (R) = 10 or more) then the theoretical local heat transfer coefficient ($h_{p,theor.}$) would still be matched. Moreover, even if the IRIS dryer's external recirculation were greater than 10, the particle exposure time in the superheated steam would be extended, thus compensating for low local particle heat transfer coefficients, and WITHOUT the need to compromise the dryer's size (i.e., larger diameters or lengths). Thus, one of the major benefits of the IRIS dryer over the conventional air dryer has been given more credibility: the IRIS dryer can be much smaller than the conventional dryer and STILL dry the particle by enabling the particle to be exposed to the superheated steam for long enough periods of time via external and internal recirculation and not by needing to resort to increased dryer lengths, as is done in conventional pneumatic dryers.

Although the hot test could not, at this time, measure the amount of external recirculation, evidence of this recirculation was made available through the cold testing that was also conducted during this task. During these cold (air) tests both the visualization of the vortex flow field and the fluid dynamics of the IRIS dryer were quantified. This included the measurement of the external recirculation and the recirculation ratio (R). A critical observation was not only that ample external recirculation was available but that this recirculation is sustained despite an increase in the dryer's particle loading (L_p) and the effect this increased particle loading has on the vortex field pressure drop. This observation was necessary in order to be able to recommend the indirectly heated steam atmosphere dryer system (as modeled in Tasks 2, 3, and 6 of this project) as the system of choice for the steam atmosphere dryer project.

The hot testing also provided evidence that wall conduction heat transfer may support a dominant part of the heat exchange process between the hot steam and the slurry feedstock, thus also supporting the use of the indirect steam atmosphere dryer system. A plot of the measured data indicates that even as little as 600 watts of wall heating (used to simulate the heat transfer from a jacketed vessel) was sufficient to see the effects of improved heating of the feedstock and the evaporation of the entrained water. Certainly, a true steam-jacketed IRIS dryer needs to be built and tested in order to quantify the magnitude of this heat transfer in improving the feedstock heating.

The hot testing provided an opportunity to design and test the first steam nozzle slurry atomizer. A steam slurry atomizer was designed and then modified during the testing program to finally perform satisfactorily. The design for this working steam atomizer model will be the starting point for making more permanent design improvements when a larger size IRIS dryer is constructed in Phase II. The successful testing of the steam atomizer is thus another major developmental success that can be attributed to the Phase I hot test program.

The cold testing also provided an opportunity to test the validity of the fluid dynamic IRIS computer models. The cold testing clearly demonstrated that the fluid dynamic constraints imposed on the dryer's operation, namely, the dryer particle collection efficiency, vortex (and hence dryer) pressure drop, and particle terminal velocity effects were either adequately modeled or at least conservatively modeled by Tecogen's computer codes. Thus, the testing verified that the IRIS operational zone (i.e., dryer inlet velocity vs. particle size) was properly identified by the computer models. Therefore the IRIS computer model developed exclusively for this steam dryer project becomes an even better tool for sizing the next larger scale IRIS dryers particularly when the measured heat transfer and fluid dynamic data have been incorporated into this software.

As a result of these cold (fluid dynamic) and hot (heat transfer) tests, the following accomplishments are cited. These results provided considerable new information to the field of steam atmosphere drying with respect to the IRIS dryer performance.

From the Cold Testing:

1. Measured recirculation collection efficiency as a function of particle diameters inlet gas steam velocity and dryer vortex flow field pressure drop. Recirculation ratios as high as 100:1 were recorded.

2. Visually observed dryer vortex flow field (up to full saturation) as a function of particle loading (L_p). Particle loading is defined as the ratio of particle mass flow rate to gas stream mass flow rate.

3. Observed and measured vortex field pressure drop as a function of particle loading and enabled a verification of the predicted overall dryer pressure drop. Pressure drops were found to be 60 to 70 percent of the predicted values.

4. Determined a dimensionless relationship for the vortex (dryer) pressure drop as a function of four (4) dimensionless groups. These groups were developed from a dimensional analysis of the measured cold test data.

5. Verified an increase in the dryer's recirculation ratio (R) as the dryer's particle loading increased. Test demonstrates that there will be adequate particle exposure time in the drying medium.

6. Determined a theoretical fluid dynamic and heat transfer model for the IRIS dryer and identified the optimum zone of operation with respect to particle size and gas velocity. The fluid dynamic and heat transfer limitations identified by this model have been tested and demonstrated to be conservative in this application to IRIS modeling.

7. Tested effects of increased lengths of the recirculation spout tube on the dryer collector efficiency and verified this method as a means of controlling dryer collector efficiency. A patent disclosure has been filed with DOE.

From the Hot Testing:

8. Measured actual local particle heat transfer coefficient h_p and dryer sizing coefficient H ($Q/Vol/\Delta T_{LM}$). Confirmed that the local heat transfer coefficient (h_p) is not dependent on particle flow rate or dryer loading.

9. Successfully dried slurry (liquid) feedstock with a wetness of 0.5 lb water/lb particle. Average drynesses of 90 percent were recorded. The dryer's drying performance of 0.04 (D.B.) exceeded the predicted values of 0.05 (D.B.).

10. Succeeded in developing a preliminary design for a steam atomizer for slurry feedstock.

11. Succeeded in testing the steam atomizer nozzle in two different locations in the IRIS steam dryer and was thus able to compare the drying ability as a result of testing in these two locations. Recommended location is at the dryer inlet.

12. Tested several slurry feedstocks:
    Temperature insensitive (clay) as well as temperature sensitive (non-dairy coffee creamer and maltodextrin-100, a food sweetener).
    The steam dried clay powder exhibited no damage as a result of drying.

13. Verified the contributions of conduction heat transfer via vessel wall heating to the overall heat transfer mechanism in support of the indirect drying design.

These accomplishments are all positive in that they have contributed to the development of the IRIS-type steam dryer.

Based on these observations, it is suggested that the laboratory testing conducted in Task 4 has been successful in meeting or surpassing many of the original goals identified for it in the Work Statement for the Phase I project work.

7. TASK 5 – ENERGY SAVINGS AND SYSTEM COST ANALYSIS

A steam dryer energy savings and system cost analysis was also conducted as part of the Phase I Feasibility Study. This analysis combined the thermodynamic analysis computer programs of the steam atmosphere dryer and the standard air dryer systems with a component materials cost and labor analysis program to determine the following comparative characteristics for both the standard air and steam atmosphere with recompression systems:

- System Materials and Engineering Labor Costs
- Operation and Maintenance Costs
- Energy Cost Savings Per Year
- Overall System Cost Savings Per Year
- Simple Payback
- Cost Per Pound of Evaporated Water

The validity of the thermodynamic computer programs for analyzing the standard air and steam dryer cycles had been demonstrated in work conducted for Tasks 2 and 4. For example, Figures 4.2 and 6.50 indicate that these computer programs are reasonable in predicting dryer performance.

In order to similarly predict each dryer's material and labor costs, a computer program was written that utilized previously published and/or new component price quotations for the standard air and steam atmosphere dryer systems. The computer model's results were then compared against quotations received for complete air dryer systems. These comparisons between the quoted price and the computer predictions also revealed excellent agreement.

7.1 STANDARD AIR SYSTEM COST MODEL

The standard air system used in the computer models is shown in Figure 7.1. A spray dryer was used as the specific dryer component. Based on individual component vendor costs and complete spray dryer system quotations the following equations were used to determine the cost of the "typical" (spray) air dryer system.

Curve fits of the cost of the spray dryer as a function of dry product flow rate were made using the relationships shown in Figure 7.2. These cost functions were obtained from published data on air dryer costs (see Reference 39 in Appendix A). Modifications were made to these curves to best "fit" the price quotations obtained for complete air dryer systems. The final equations used for the air dryer cost model were:

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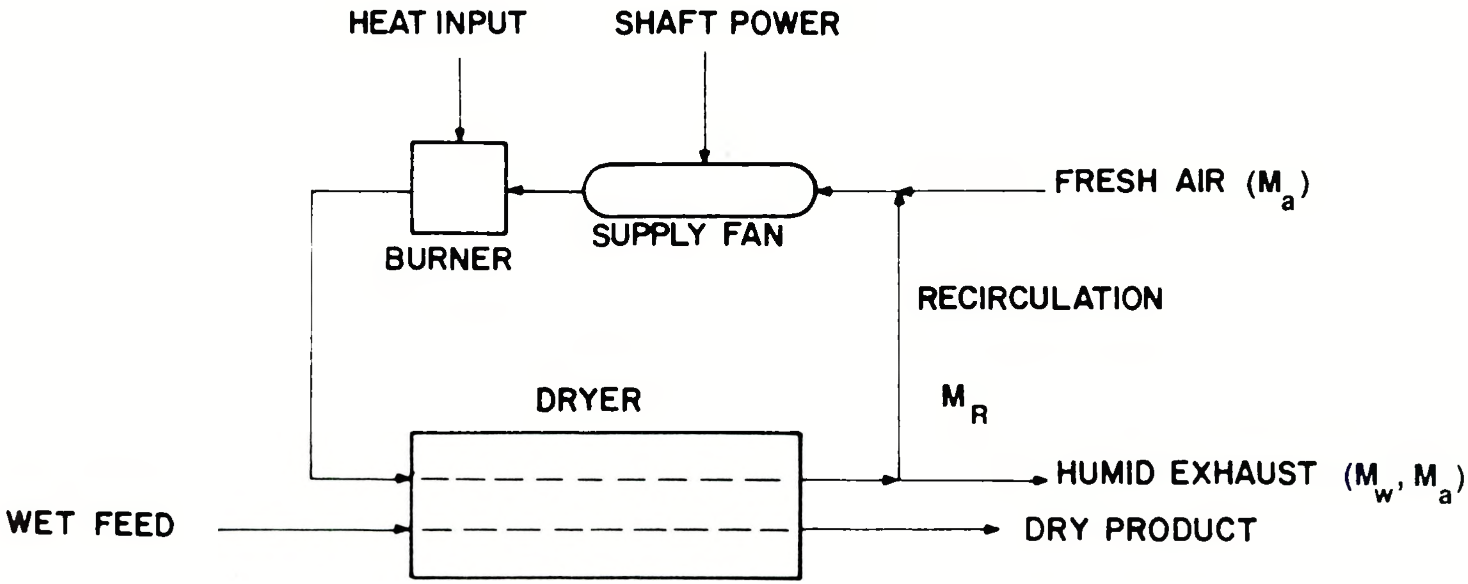

DEFINE: RECIRC. RATIO $(R_f) = \dfrac{M_R}{M_a}$

HUMIDITY RATIO $(R_w) = \dfrac{M_w}{M_a}$

Figure 7.1 Schematic of Air-Drying System

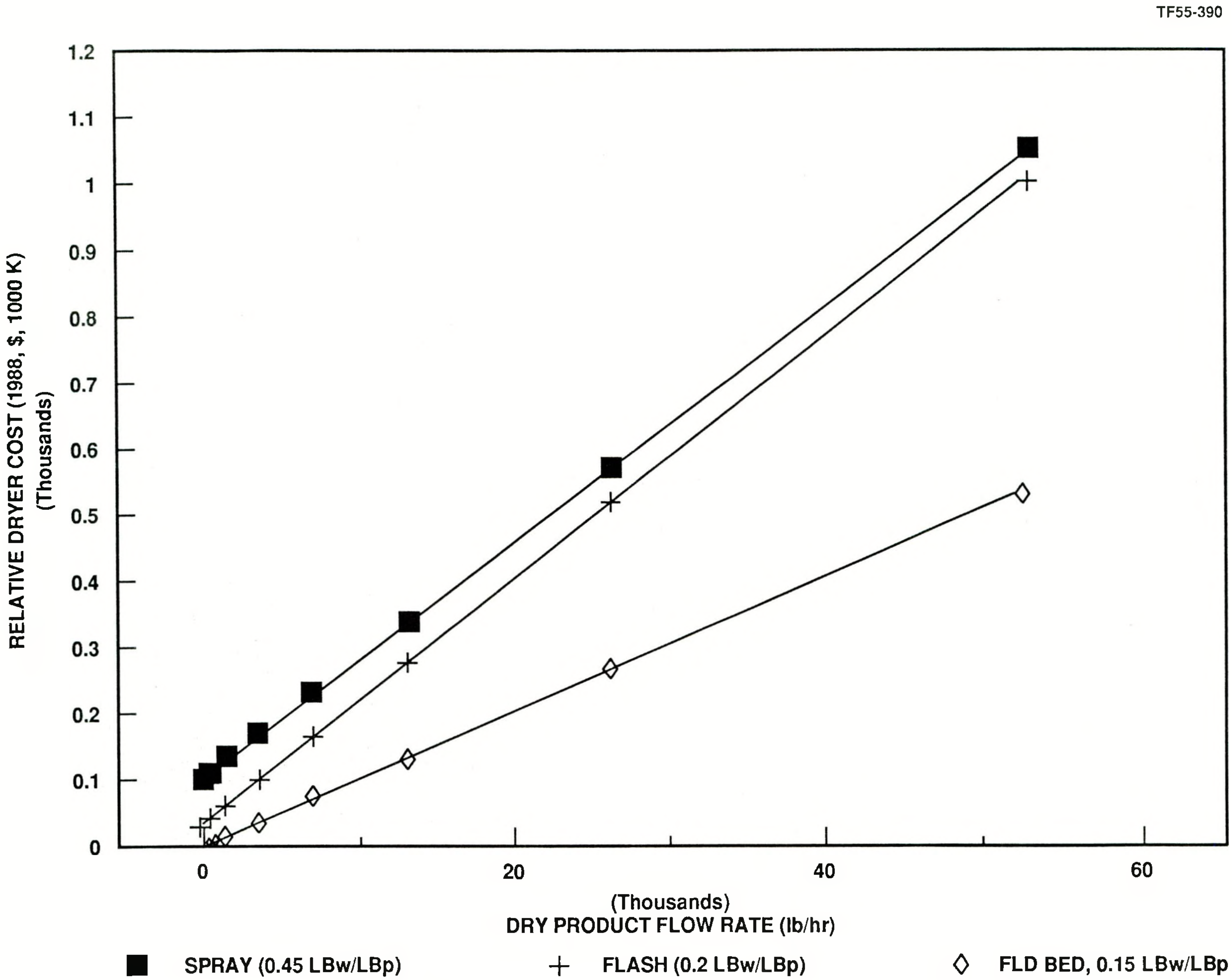

Figure 7.2 Relative Dryer Costs
Spray, Flash, and Fluidized Bed Dryers

1. Spray Dryer Vessel Cost ($):

$$(\text{Curve Fit of Figure 7.2}) \times 1000 \times 1.2 \times \left(\frac{UA_{dryer}}{6.61}\right)^{\frac{1}{4}} \times \left[\frac{5000\ \text{lb/hr}_{evap}}{(\text{lb/hr})_{evap}}\right]^{\frac{1}{4}}$$

2. Air Heater Cost ($):

$$\$50{,}000 \times \left[\frac{\text{Heater Requirements, Btuh}}{8.5 \times 10^6\ \text{Btuh}}\right]^{0.6}$$

3. Air Blower Cost ($):

$$\$27{,}202 \times 0.7 \times \left[\frac{\text{Air Blower, kWe}}{186}\right]^{0.456}$$

4. Air Cyclone and Stack Scrubber ($):

$$\$13{,}560 \times \left(\frac{\text{Air Dryer Volume Flow, cfm}}{9436}\right)^{0.6}$$

5. Air Ducting ($):

0.1 x (Sum of 1, 2, 3, and 4 Above)

6. Controls ($):

Steam Dryer Controls x 0.85

7. Miscellaneous Costs ($):

0.05 x All of Above (1 through 6)

8. Engineering Labor:

15% x All of Above (1 through 7)

9. Manufacturing Product Margin:

$$15\% \left(\text{or a Multiplier of } 1.1765 = \frac{1}{1 - \text{Margin}} \right)$$

10. O&M Costs ($):

$$[\text{lb/hr}_{evap}] \times 0.0015 \times \left[\frac{20{,}000}{\text{lb/hr}} \right]^{0.4} \times \text{hrs}$$

The 10 functions just listed for standard air dryer component costs were used to determine the total cost for a spray-dryer pneumatic system. Using this computer model the system costs were determined for four different pneumatic dryer systems, and these results were compared with four budgetary quotations received from a leading dryer manufacturer for those systems. The results are shown in Table 7.1 (Rows 6 and 13). This table indicates a very good agreement between the computer cost model for the pneumatic dryer system and the manufacturer's budgetary costs for these systems. The non-heat-sensitive pneumatic dryer system was used to directly compare the pneumatic system with the new steam atmosphere system. For this air system, Tecogen's cost model is within 2 to 9 percent of the budgetary quotations and in both instances Tecogen's cost model results are higher than the budgetary quotes. Thus, the comparative cost analysis presented in the following analysis should be considered conservative; i.e., the system cost differences for the air and steam system and the simple paybacks will be higher than what might be realized if these systems were to be built.

7.2 STEAM ATMOSPHERE DRYER (WITH RECOMPRESSION) COST MODEL

The indirect steam atmosphere dryer system (shown in Figure 7.3) has been used for the energy savings and cost analysis comparisons with the standard air dryer system. A direct dryer system (as previously shown in Figure 5.1) is also a viable steam atmosphere dryer system. Its thermodynamic performance (i.e., energy requirements) is comparable to the indirect steam dryer system if a multiple stage system is used (as previously diagrammed in Figure 5.3). However, the indirect atmosphere dryer system appears to be a simpler system, complicated only by the need for extended heat transfer surface within the steam dryer vessel rather than the external (steam reboiler) surface required in the direct steam dryer system. When it is installed, therefore, the indirect steam atmosphere dryer system is expected to require a "footprint" that is smaller than that required by the direct steam dryer system. However, its total materials and engineering and technical

TABLE 7.1

STATE-OF-THE-ART SPRAY DRYER SYSTEM
(Ref. Leading Dryer Manufacturer)

	Heat-Sensitive Feedstock		Non-Heat-Sensitive	
Dryer Feedstock Capacity (lb_{evap}/hr)	5,000	20,000	5,000	20,000
Dryer Inlet Temp. (F)	482	482	932	932
Dryer Outlet Temp. (F)	212	212	257	257
Dryer Residence Time (sec)	24	24	24	24
Components: Air Spray Dryer and Atomizer (304 SST) Air Dryer Ducting Air Heater (Direct, Nat. Gas Air Heating) Air Blower System Cyclone Separator and Stack Scrubber Controls and Instrumentation and Alarms Miscellaneous Components				
Non-Installed Costs	$1,168,000	$2,000,000	$793,000	$1,500,000
Air Heat Input (MMBtu/hr)	11,100,000	43,490,000	8,530,000	31,350,000
System Power Reqs. (kWe)	200	850	105	375
Total Heat In Q1 (Btu/lb_{evap}) Q1, w/o Elec. Power Para.	 2,675 2,220	 2,658 2,174.5	 1,945 1,706	 1,781 1,567.5
Spray Dryer Size Dia. x Height (ft)	28.2 x 57.4	38.7 x 78.7	22.3 x 44.3	32.8 x 66.3
Weight (tons)	19.8	30.8	11.6	24.2
Installed Floor Space Reqs. (ft)	39.5 x 59 x 72 (Ht.)	49.2 x 85.3 x 93.5 (Ht.)	32 x 44.3 x 59.4 (Ht.)	42.6 x 67.2 x 81 (Ht.)
Tecogen's Cost Model Results (Non-Installed)	$949,000	$1.96 M	$805,000	$1.63 M

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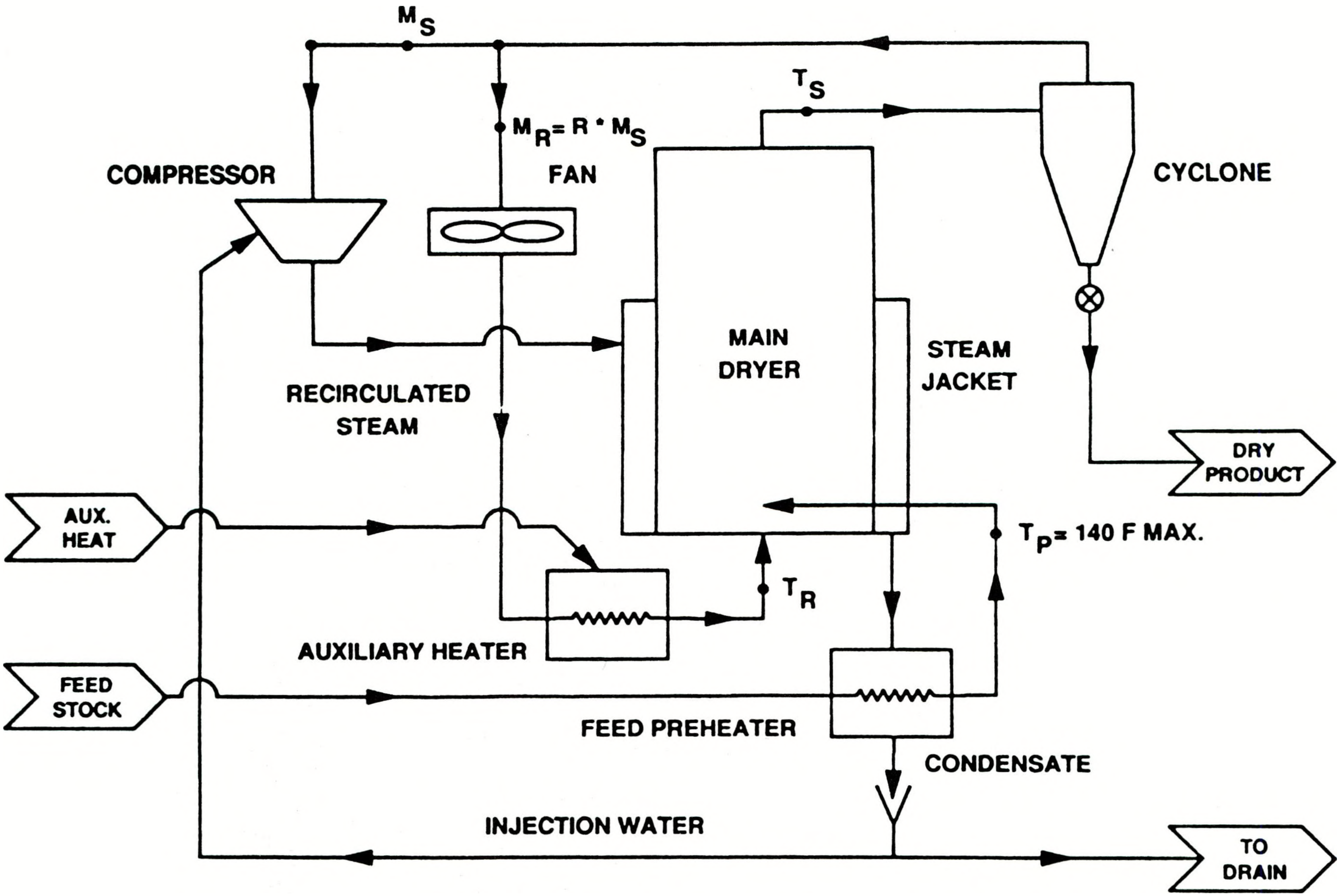

Figure 7.3 Indirectly Heated Steam Dryer

labor costs are expected to be slightly more than the costs of comparable items in the direct steam dryer system. Thus, the use of the indirect steam atmosphere dryer system once again provides a conservative cost estimate for the steam dryer system when compared to the standard air dryer system costs while representing the better of the two steam atmosphere dryer systems.

The component cost functions used for the indirect steam atmosphere cost model were:

1. Steam, Helical Screw Compressor:

For: Steam Volume Flow Rate (cfm) less than:

2300: $\$ = 11 \times 2200 + 75{,}000 \times 2.5 \times (kW_{comp}/225)^{0.4}$

3700: $\$ = 11 \times 3600 + 75{,}000 \times 2.5 \times (kW_{comp}/225)^{0.4}$

5700: $\$ = 11 \times 5600 + 75{,}000 \times 2.5 \times (kW_{comp}/225)^{0.4}$

8900: $\$ = 11 \times 8800 + 75{,}000 \times 2.5 \times (kW_{comp}/225)^{0.4}$

13,100: $\$ = 11 \times 13{,}000 + 75{,}000 \times 2.5 \times (kW_{comp}/225)^{0.4}$

23,800: $\$ = 11 \times 21{,}500 + 75{,}000 \times 2.5 \times (kW_{comp}/225)^{0.4}$

The steam compressor costs are incremented upward based on the actual volume of steam flow rate used with the compressor as measured by the compressor's inlet temperature and pressure. The compressor costs are also a function of compressor power requirements.

These relationships are based on Tecogen's cost model for its product line of steam compressors and are adjusted to consider the quotations received from four compressor manufacturers who quoted on Tecogen's compressor specifications.

2. Steam Blower and Motor ($):

$$\left[\frac{\text{Steam Blower, kWe}}{180}\right]^{0.456} \times 28{,}000$$

3. Steam Compressor Electric Motor and Coupling ($):

$$\left(\frac{\text{Compressor, kWe}}{150}\right)^{1.06} \times 11{,}000 + (4 \times \text{Compressor, kWe})$$

4. Compressor Gearbox ($):

$$\left(\frac{\text{Compressor, kWe}}{50}\right)^{0.6} \times 12{,}000$$

5. Lube System ($):

$$\text{Compressor Steam Volume (cfm)} \times 0.05 + 10{,}000$$

6. Base Frame:

$$0.05 \times (\text{Sum of Items 1, 3, 4, and 5 Above})$$

7. Jacketed IRIS Dryer (With Extended Heat Transfer Surface Area):

$$(\text{Cost of Pneumatic System; see Item 1 of Section 7.1})$$

$$\times\ (\text{Ratio of Steam to Air Volume Flow Rates})^{0.1}$$

$$\times\ (\text{Compressor Pressure Ratio})^{0.125} \times \left(\frac{2 \times (\text{Diameter} \times \text{Height})_{\text{IRIS}}}{(\text{Diameter} \times \text{Height})_{\text{standard air dryer}}}\right)$$

$$+ \left[\frac{UA_{\text{steam dryer}}}{194{,}000}\right]^{0.6} \times 0.8 \times 220{,}000$$

The IRIS dryer cost model is based on the cost functions of a pneumatic spray dryer and is adjusted for the design characteristics of the IRIS vessel. For example, the steam to air volume ratio term adjusts the steam dryer's cost to consider the effects of lower steam flow rates on the sizes of piping or ducting that would be connected to the dryer vessel. The compressor ratio term adjusts the steam dryer's cost to consider the fact that the steam dryer's vessel will be jacketed and this must be structurally designed to withstand a higher than atmosphere pressure. The ratio of IRIS diameter and height product to the standard air diameter and height product adjusts the steam dryer's cost to consider the steam vessel's smaller surface area. The diameters and heights used in this equation are: D = 8.5 ft, h = 17 ft; and are based on the heat transfer coefficients measured in the laboratory testing and presented in Figure 6.52. The last term models the cost of the extended surface area that may be necessary to provide adequate surface heat transfer between the vessel's jacketed wall and the wet feedstock particles. This cost is based on a manufacturer's surface area cost of $220,000 for a heat exchanger (UA) size of 194,000 Btu/hr-°F. Once again, a conservative pricing philosophy prevails. The IRIS cost estimate given by this function is thought to be conservative (i.e., higher than expected) due to the fact that the extended surface area may not be needed if it is determined from the work performed in the next project phase that the heat transfer coefficients between the jacketed dryer walls and the wet feedstock are high enough to promote the necessary heat transfer without extended surface.

8. Steam Dryer Controls ($):

$$\$40{,}000$$

9. Steam Cyclone Separator ($):

$$\left[\frac{\text{Steam Volume Flow, cfm}}{9436}\right]^{0.6} \text{x } 13{,}560 \text{ x } 1.3$$

10. Auxiliary Steam Heater:

$$32{,}000 \text{ x } \left(\frac{\text{Auxiliary Boiler, Btu/hr}}{280{,}000}\right)^{0.6}$$

11. Feedstock Preheater ($):

$$\left[\frac{\text{UA}}{11{,}000}\right]^{0.6} \text{x } 8{,}000$$

12. Steam Piping and Ducting ($):

$$\$_{\text{Air Ducting}} \times 1.5 \times \left[\frac{\text{Steam Volume}}{\text{Air Volume Flow}}\right]^{0.5}$$

13. Miscellaneous Costs ($):

$$0.075 \times (\text{Sum of All of the Above, Items 1 through 12})$$

14. Fixed Cost Engineering ($):

$$15\% \times \text{All of the Above (Items 1 through 13)}$$

15. Manufacturer's Product Margin:

$$15\% \text{ or a Multiplier of } \left(1.1765 = \frac{1}{1 - \text{Margin}}\right) \text{ Times All of the Above}$$

16. Steam O&M Costs ($):

$$\left(\frac{\text{lb}}{\text{hr}}\right)_{\text{evap}} \times 0.0015 \times \left[\frac{20{,}000}{\text{lb/hr}}\right]^{0.4} \times \text{hrs} + [\text{kW}_{\text{compressor}} \times 0.01 \times \text{hrs}]$$

The individual component costs for the steam and air dryer systems were determined by means of the cost functions just listed. A summary of the percentage cost comparisons by component is given in Table 7.2 for the cases studied in this Task 5 for the Phase I Feasibility Study. From Table 7.2 it is clear that the steam components that have the most influence on the cost of the steam dryer system are the steam IRIS dryer (23 percent of total system cost), steam compressor and electric drive (22.8 percent), the steam ducting and piping (8.2 percent), and miscellaneous costs (5.2 percent). The cost functions for these components have been identified as conservative; that is, they are more expensive than what may be actually realized in a fully developed system. Thus, it is thought to be appropriate to use a range for the simple payback. This confidence range will identify a simple payback for the system that falls between +5 to −10 percent of the value calculated using the above component cost functions.

TABLE 7.2

DRYER COMPONENT COST COMPARISONS AND ECONOMIC ANALYSIS PARAMETERS

Steam Cycle Materials and Assembly Labor	Percent of Total Cost	Air Standard Cycle Materials and Assembly Labor	Percent of Total Cost
Steam Compressor	20		
Steam Blower and Motor	2.2	Air Circulator and Motor	1.4
Electric Motor and Coupling	2.8		
Gearbox	3.1		
Compressor Lube System and Base	2.2		
IRIS Steam Dryer	23	Spray Dryer	51.8
Steam Cyclone	1.4	Air Cyclone	2.9
Auxiliary Steam Heater	3.1	Air Heater	6
Feedstock Preheater	0.6		
Steam Ducting and Piping	8.2	Air Ducting	6.2
Controls and Insts.	2.1	Controls and Insts.	2.2
Misc. Costs (7.5 Percent Total)	5.2	Misc. Costs (5.0 Percent Total)	3.5
SUBTOTAL:			
Engineering Labor	11.1	Engineering Labor	11
15-Percent Margin	15	15-Percent Margin	15

7.3 STEAM DRYER SYSTEM DESIGN POINT SELECTION

Prior to proceeding with an extensive parametric cost analysis it was necessary to determine a thermodynamic design point for the steam dryer system. Once identified, this design point would serve as the base system for determining how various economic parameters (such as gas and electric costs, operating hours, etc.) affect the system's economic performance. The principal thermodynamic design point parameters as identified in Task 2 were:

- Compressor pressure ratio,
- System pressure drop, and
- Dryer inlet temperature.

Applying the thermodynamic and cost model computer programs, it was determined that the minimum simple payback occurs at a steam dryer inlet temperature of 270°F corresponding to a compressor pressure ratio of 3.25, as shown in Figure 7.4.

Figure 7.4 assumes a dryer discharge temperature of 220°F and a system pressure drop of 40 inches of water. An analysis was also conducted to determine if the economic performance (i.e., simple payback) benefitted from a 10°F increase in the inlet and discharge temperatures. The benefit would result from a <u>decrease</u> in IRIS steam dryer size and thus a decrease in its cost, a cost that represents 23 percent of the total steam system cost (see Table 7.2). Using the measured heat transfer coefficients from Task 4, the IRIS dryer's volume decreased by 33 percent (from 971 ft^3 to 654 ft^3) by increasing the dryer steam inlet temperature to 280°F and the dryer steam discharge temperature to 230°F. However, in order to accommodate these higher dryer temperatures the compressor's pressure ratio increased from 3.25 to 3.75 with a corresponding increase in compressor power, size, and cost. The net consequence did <u>not</u> reduce the steam systems' simple payback. In fact, the decrease in the IRIS dryer cost was comparable to the increase in the compressor cost, resulting in no significant change in the simple payback.

A similar economic analysis determined the effect of the system's overall pressure drop on the steam dryer's simple payback. A system pressure drop of 10, 20, and 40 inches of water was used to determine the changes in the system's simple payback. In this analysis, the standard air dryer's thermodynamic performance was also determined by means of a 10, 20, and 40 in. wc pressure drop. The results of this analysis indicated a decrease in simple payback of only 2 percent for the systems operating with either the 40-in. or 10-in. overall pressure drop. For this analysis it was assumed that the pressure drop did not affect the

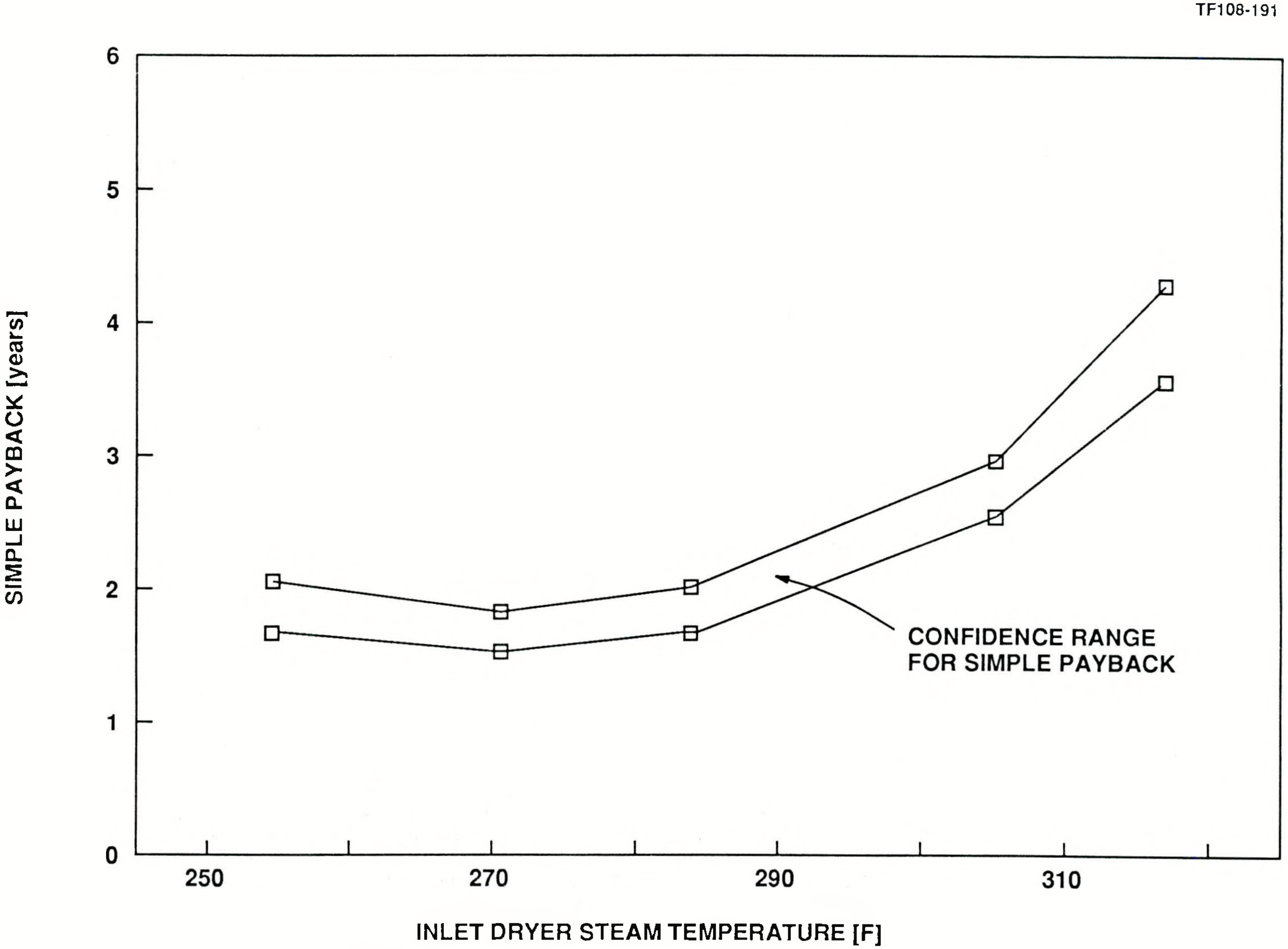

Figure 7.4 Selecting Thermodynamic Design Point for Steam System

relative costs for the steam and air components. That is, if a 10-in. pressure drop implies larger steam atmosphere piping and thus a more expensive system, then likewise the air dryer piping system would also be larger and more expensive, but the relative costs for these piping systems was assumed not to change. The system's net pressure drop was assumed to affect only the thermodynamic performance of the systems as was evidenced by increases in the dryer's energy requirements (Btu/lb_{evap}).

The result of this analysis was to select the following steam atmosphere dryer operating characteristics as the base system:

- Compressor Pressure Ratio = 3.25

- Compressor Overall Efficiency ($\eta_{th} \times \eta_{GB} \times \eta_{mech}$) =
  0.6 at 5,000 lb/hr
  0.62 at 10,000 lb/hr
  0.64 at 15,000 lb/hr
  0.66 at 20,000 lb/hr
  0.67 at 25,000 lb/hr

- IRIS Dryer Diameter = 8.52 ft
  IRIS Dryer Height = 17 ft

- Steam Dryer Inlet Temperature = 270°F
  Steam Dryer Outlet Temperature = 220°F

- Steam Dryer Operating Pressure = 14.7 psia

- Feedstock (D.B.) Moisture Inlet Content = 0.5 lb_w/lb_{prod}
  and Moisture Outlet Content = 0.05 lb_w/lb_{prod}

- Dryer System Pressure Drop = 40 in.

The standard air dryer's design and operating conditions were chosen from a manufacturer's selection (see Table 7.1) for a spray dryer that would meet Tecogen's request for quotation. A comparison of the differences in the drying energy requirements between the air and steam atmosphere dryers is shown in Figure 7.5. Thus, the steam atmosphere system is shown to have an energy requirement that is 53 to 57 percent less than the comparable air dryer system.

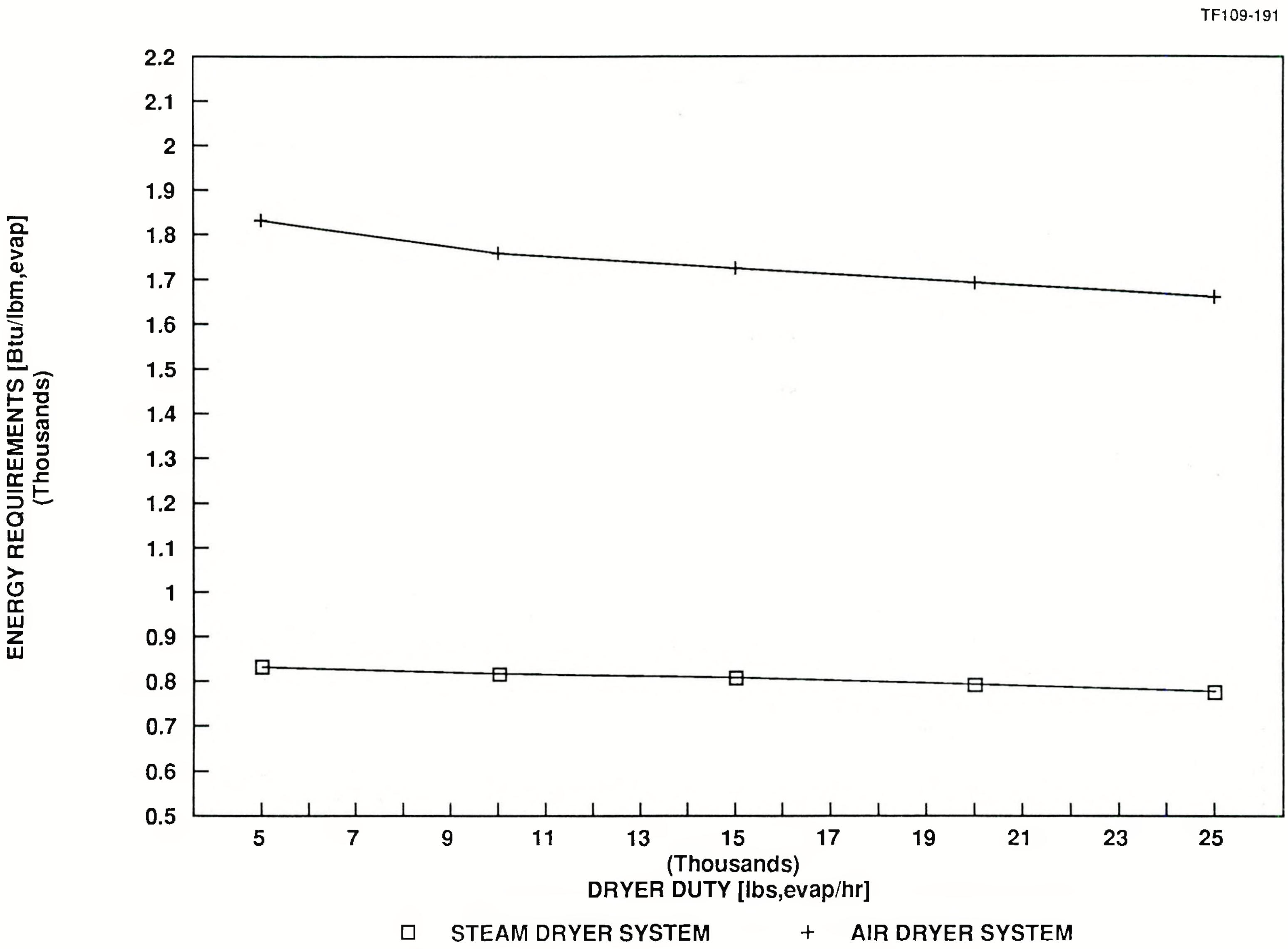

Figure 7.5 Energy Requirements for Air and Steam Dryers

7.4 STEAM DRYER VS. AIR DRYER COST ANALYSIS

Once the thermodynamic baseline system for the steam atmosphere dryer system was determined, a parametric analysis was conducted to determine the economic characteristics of the steam dryer system relative to the standard air dryer system. The principal economic characteristics is considered were:

- Simple payback
- Cost per pound of evaporated water
- Total cost savings per year
- Estimated cost for the steam dryer system

The simple payback was determined based on the net difference in the cost to operate both types of systems per year divided into the cost of the steam atmosphere system as shown in the following equation.

$$\text{Simple Payback (S.P.)} = \frac{\$_{\text{steam dryer}}}{\$_{\text{O\&M, air}} - \$_{\text{O\&M, steam}}}$$

where:

$\$_{\text{steam dryer}}$ = cost of the steam atmosphere dryer system

$\$_{\text{O\&M, air}}$ = yearly cost to operate the air dryer using gaseous fuel and operation and maintenance costs (see cost functions for air system)

$\$_{\text{O\&M, steam}}$ = yearly cost to operate the steam dryer using electric power and operation and maintenance costs (see cost functions for steam system)

In this analysis it was assumed that the prime mover for the steam compressor was an electric motor and that any auxiliary steam heating requirements were provided by burning natural gas. All heat input for the air atmosphere dryer was in the form of natural gas. The analysis did not consider the effect of depreciating hardware costs, tax deductions, the inflation costs for the natural gas or electric utilities, or the present value of investment monies. This simple payback analysis provides a first order analysis that is consistent with the degree of accuracy available from the cost model.

The results of the economic cost analysis are presented in Figures 7.6 through 7.9.

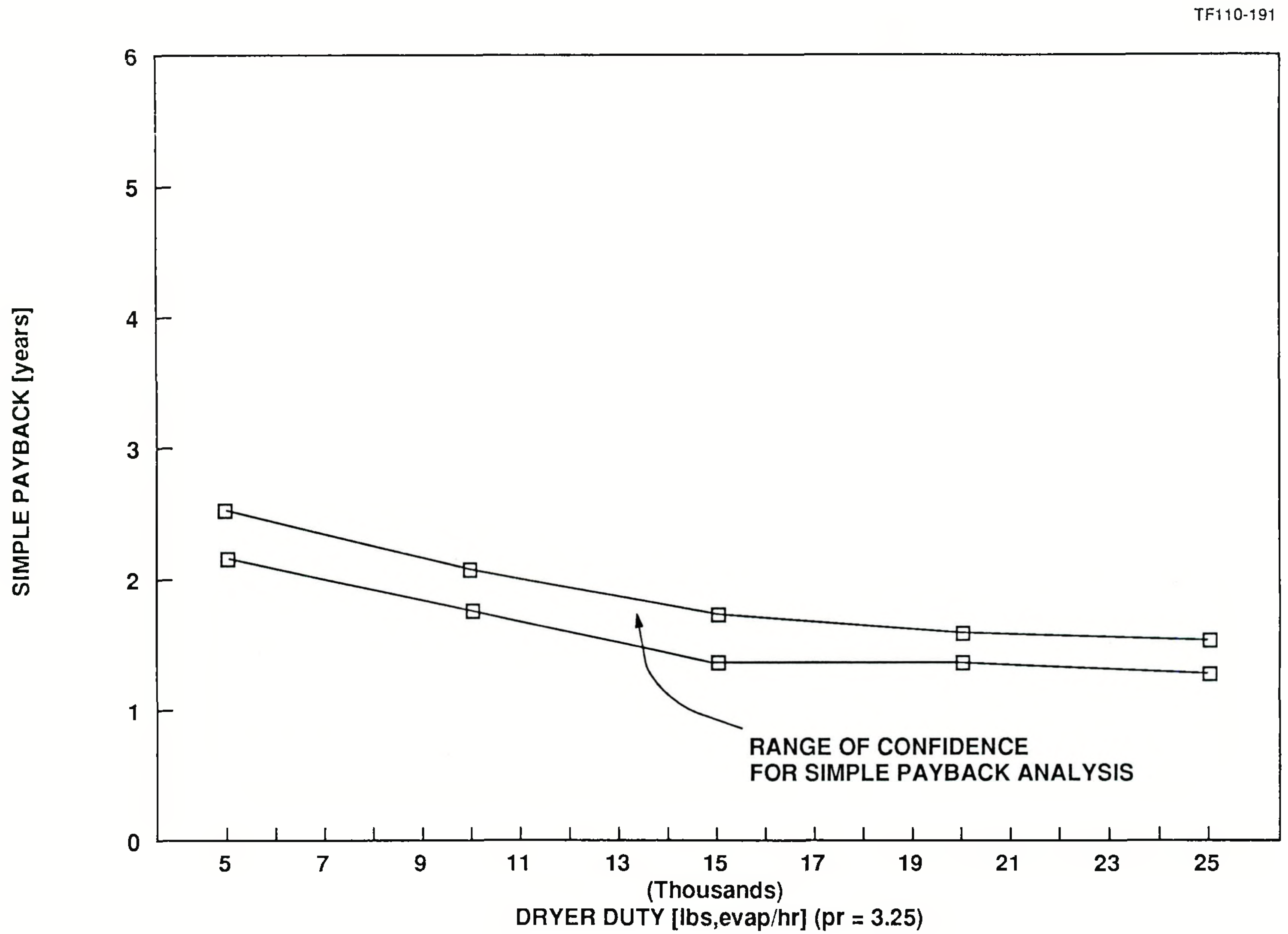

Figure 7.6 Simple Payback for Steam Dryer System
Hours = 6525, Elec. = \$0.05/kWh, Gas = \$3.5/MMBtu

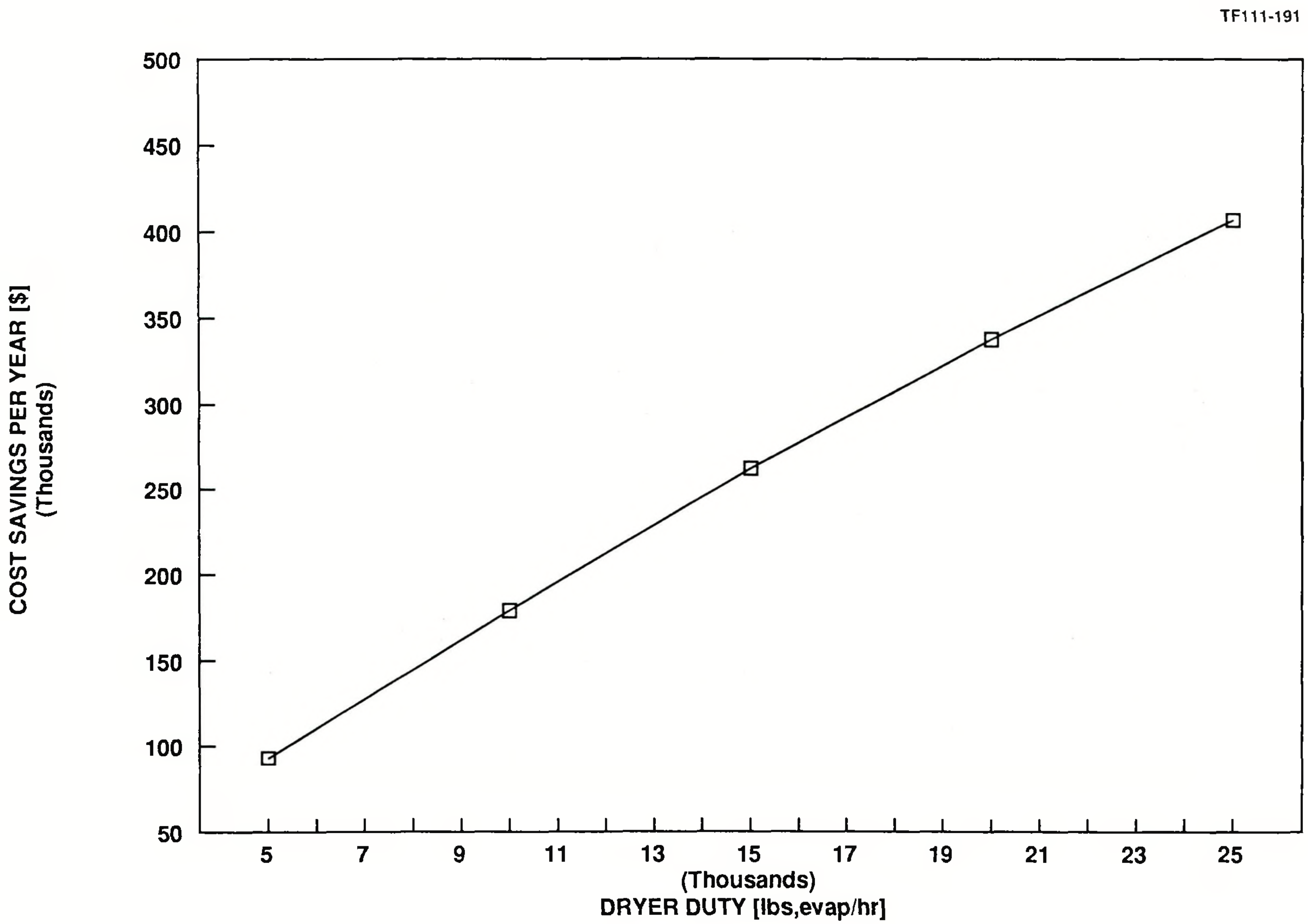

Figure 7.7 Cost Savings/yr for Steam Dryer System
Hours = 6525, Elec. = $0.05/kWh, Gas = $3.5/MMBtu

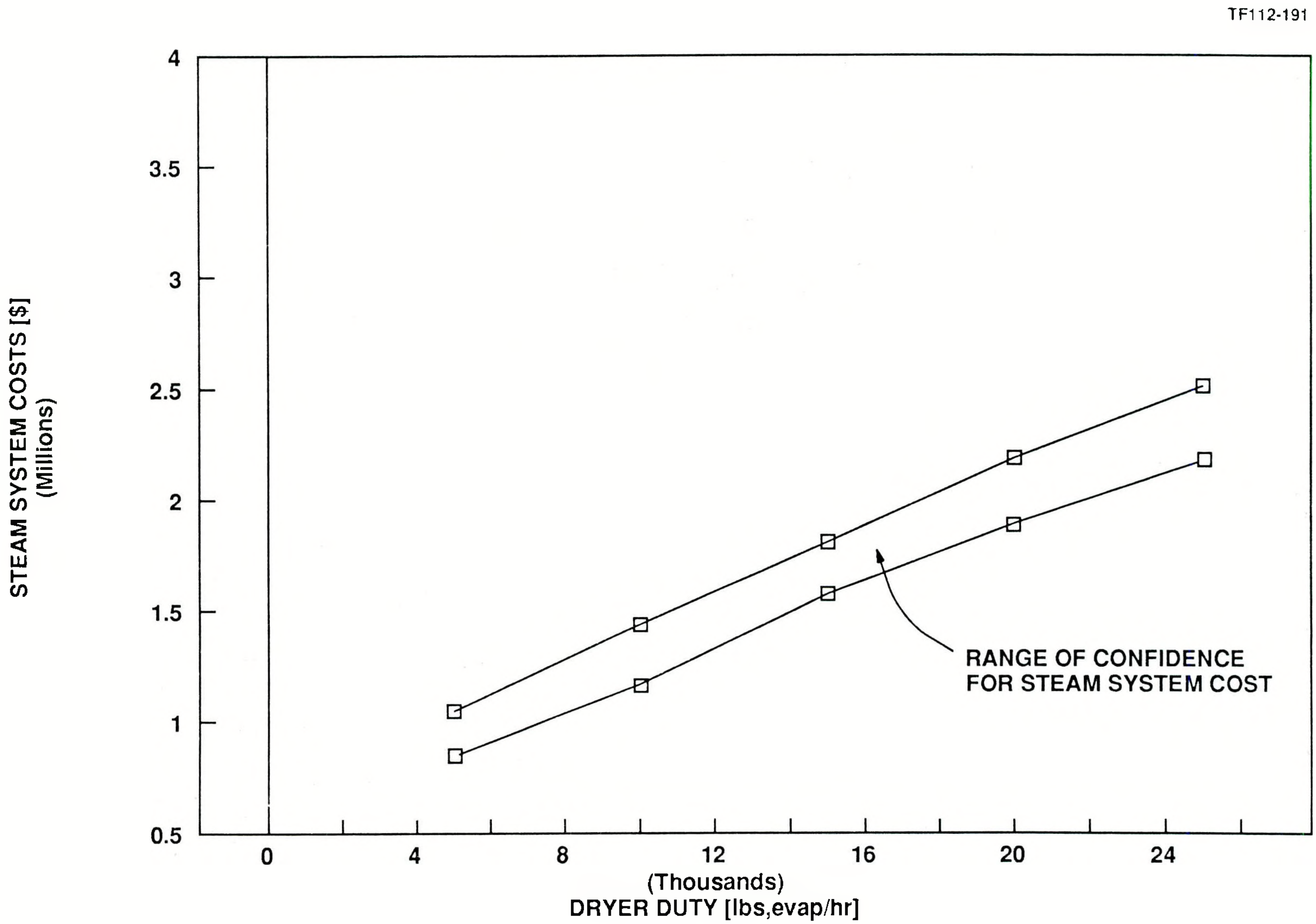

Figure 7.8 Steam Dryer System Costs

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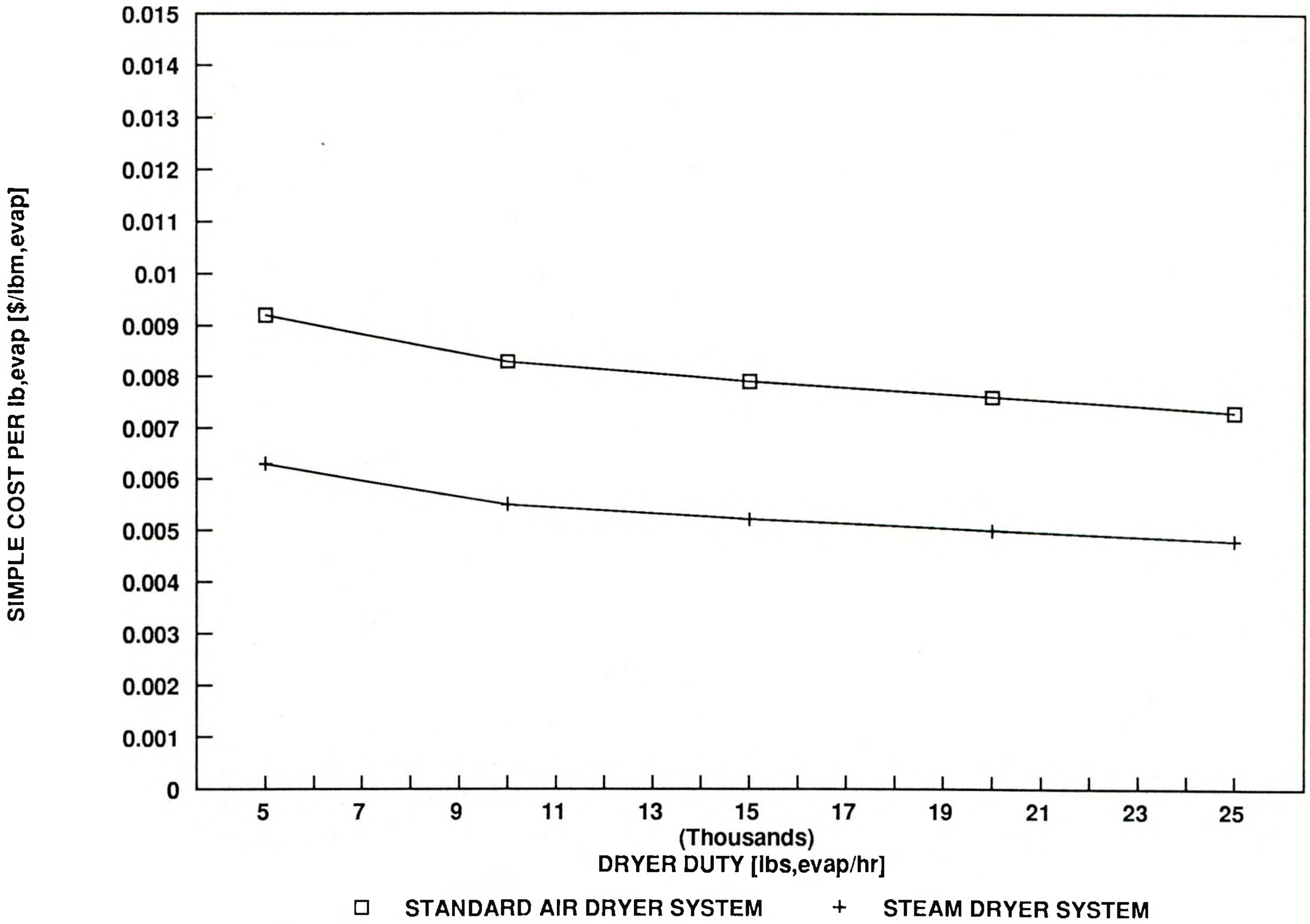

Figure 7.9 Steam Dryer System Cost to Dry Feedstock
Elec. = $0.05/kWh, Gas = $3.5/MMBtu

Figure 7.6 reveals that a simple payback of 2.5 to 1.5 years is available for the steam atmosphere dryer system. This is considered very acceptable to the drying industry, which considers simple paybacks of 3 years or less to be good. The higher simple payback for a drying system of 5000 to 7000 lb/hr is due to the need for a steam compressor whose size is incrementally relatively large for the steam volume flow rate associated with 5000 to 7000 lb/hr.

The cost savings calculated for the steam system relative to the air dryer system is shown in Figure 7.7. Savings of $100,000 to $400,000 are possible depending on the size of the drying system. The cost for a steam atmosphere dryer system is shown in Figure 7.8 and indicates a cost range from $1M to $2.5M depending on the dryer size.

An interesting cost comparison between the steam and air dryer systems is given in Figure 7.9. Figure 7.9 displays the estimated cost to evaporate a pound of water in either the steam or air dryer system. Using Figure 7.9, a 32- to 35-percent reduction in cost to evaporate one pound of water has been discerned using the steam atmosphere dryer rather than the air dryer system. It has also been determined that this difference in the cost to evaporate one pound of water from the feedstock is still high; 20 to 27 percent if present amortization and tax deduction schedules are enforced while calculating these costs.

In preparing Figures 7.6 through 7.9, assumptions for the operating hours per year (6525 hrs), electric power cost (5¢/kWh), and natural gas costs ($3.5/MMBtu) were made. These costs are typical of the prevalent cost for utilities and the operating hours (i.e., two work shifts) for the U.S. drying industry. A parametric analysis using the electric and natural gas utilities and the system operating hours is interesting and useful. The results of Tecogen's parametric analysis for these variables is given in Figures 7.10 through 7.13. These results indicate, for example, that the cost of electricity and gas can be as high as 6¢/kWhr and as low as $2.5 to $3/MMBtu, respectively, before the steam dryer system's simple payback exceeds 3 years. It is also interesting to note (from Figure 7.12) that a 15-percent increase in gas utility cost will result in a 25-percent increase in total cost savings when the steam dryer system is used. Similarly, it may be observed from Figure 7.13 that even a 45- to 50-percent utilization factor (i.e., 4000 to 4380 hours per year) would still keep the steam atmosphere drying system from exceeding the 3-year simple payback criterion.

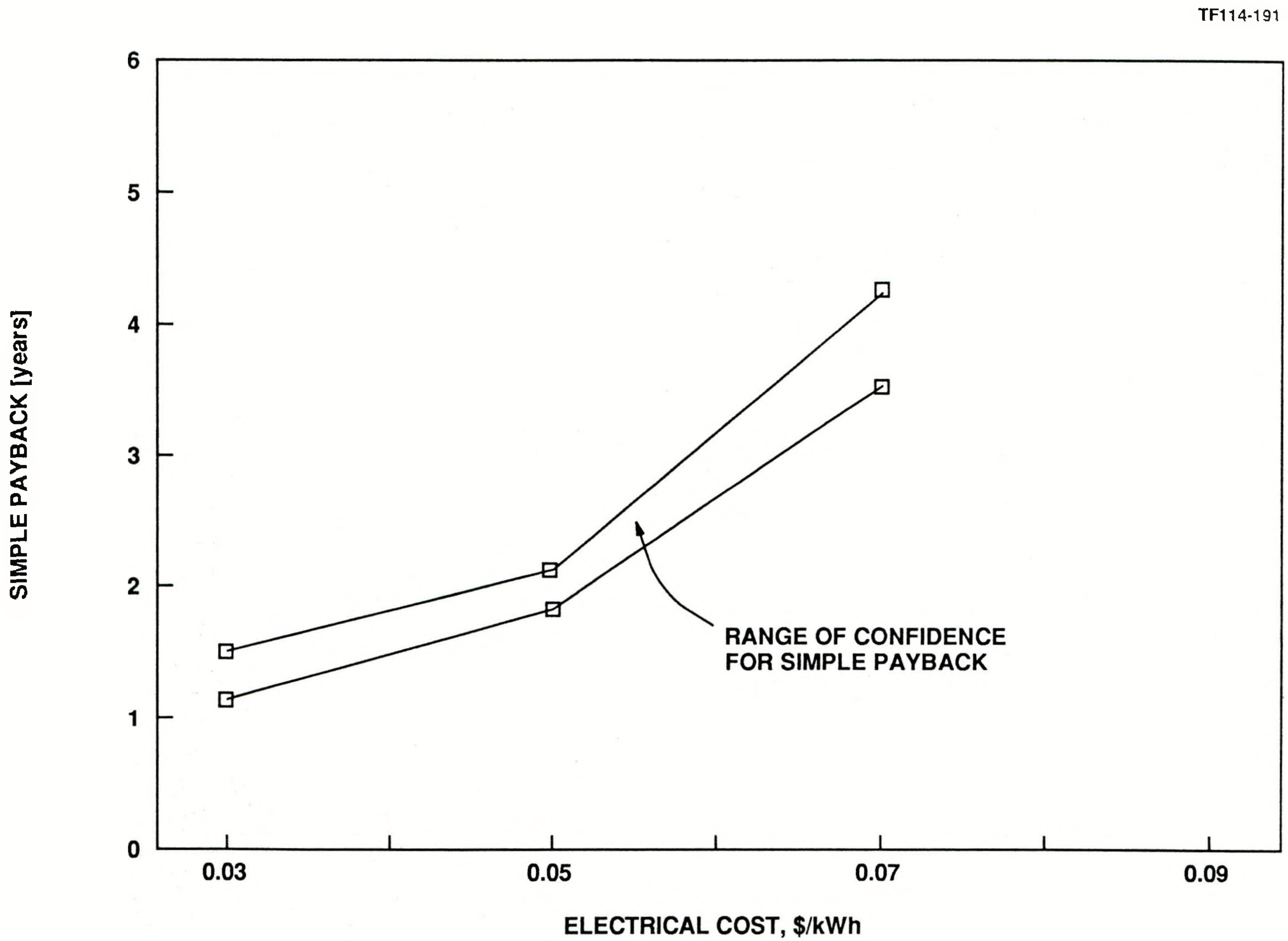

Figure 7.10 Simple Payback for Steam Dryer System
Hours = 6525, Gas = $3.5/MMBtu

TF115-191

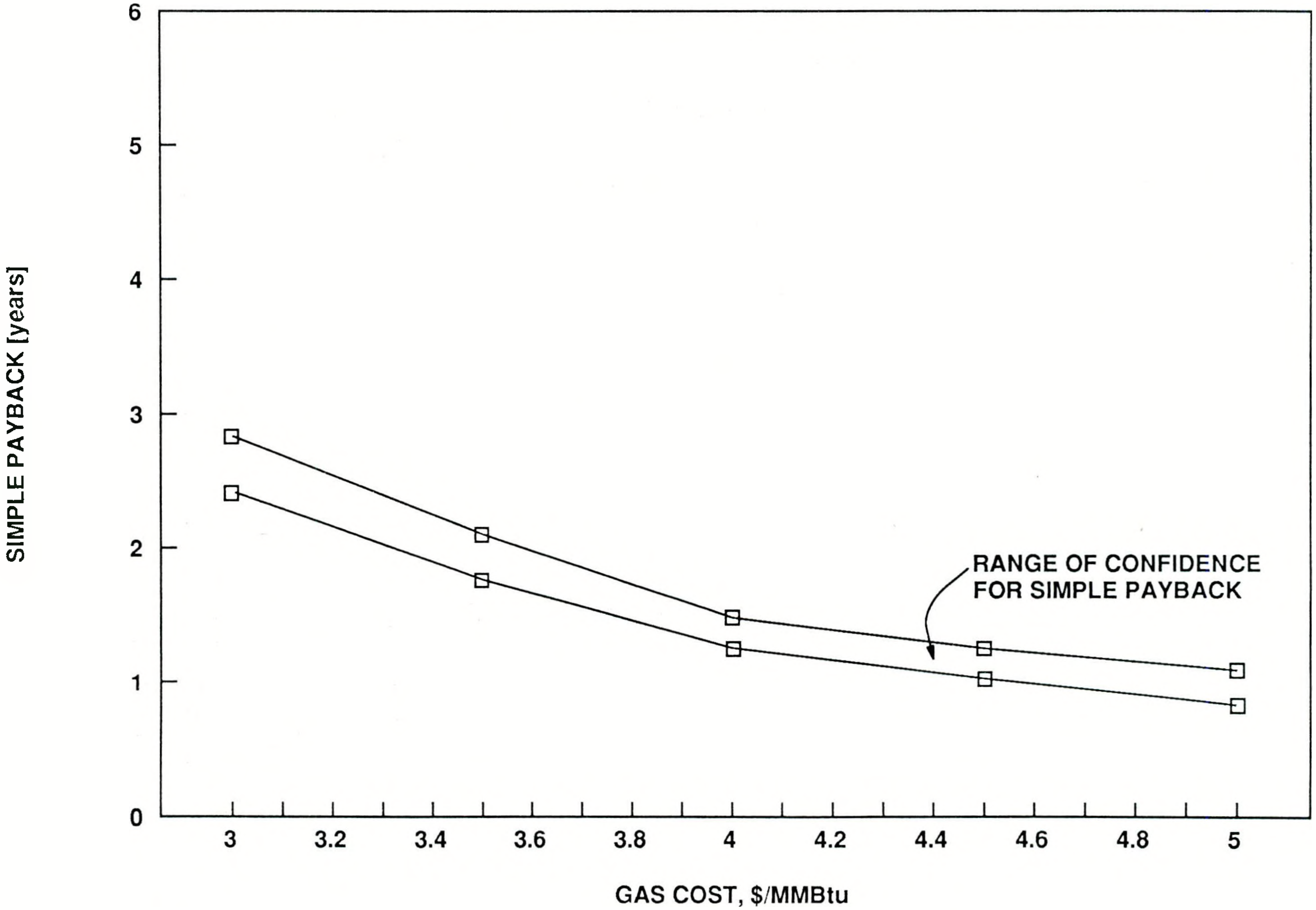

Figure 7.11 Simple Payback for Steam Dryer System
Hours = 6525, Elec. = \$0.05/kWh, 10,000 lb_{evap}/hr

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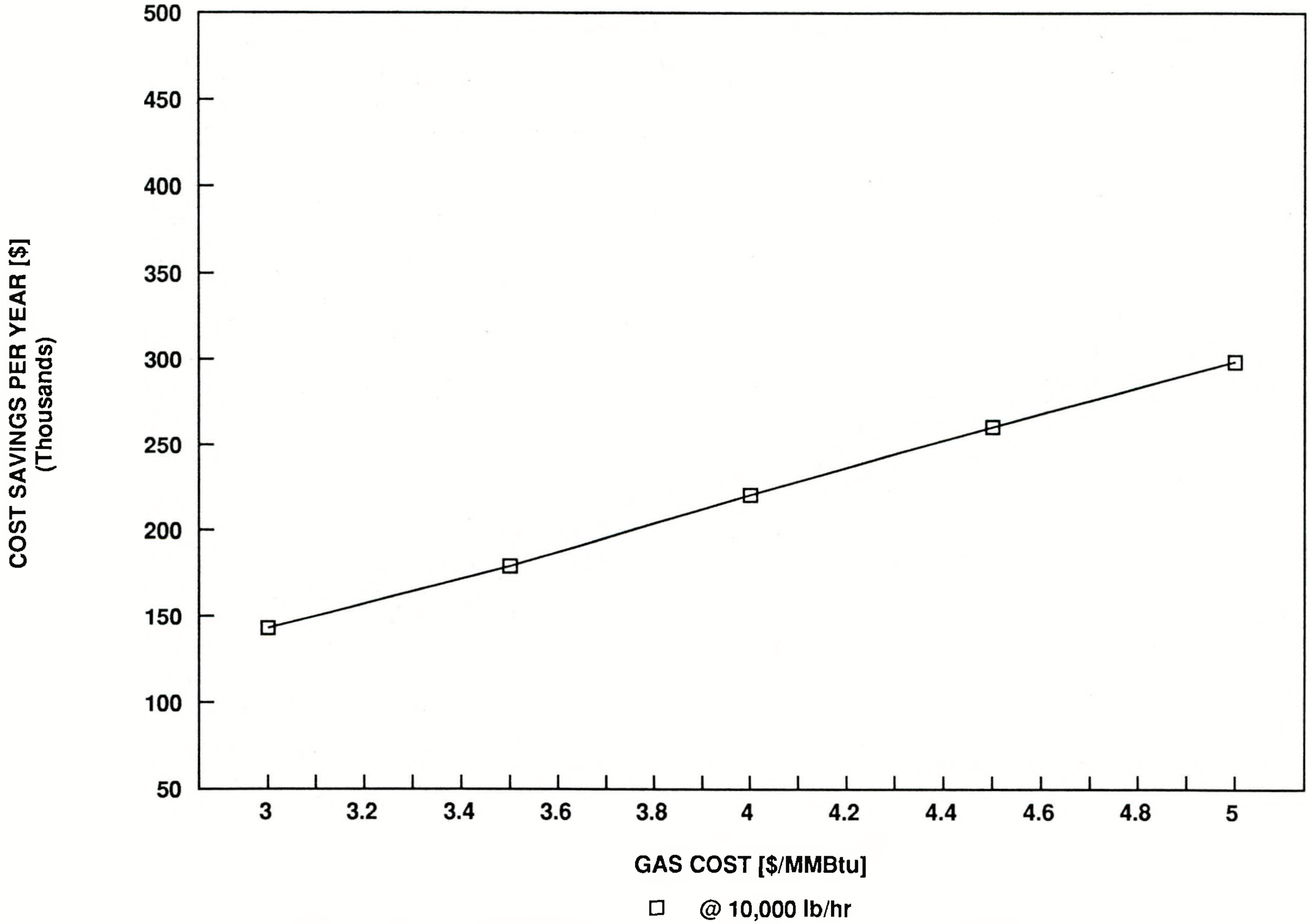

Figure 7.12 Cost Savings/yr for Steam Dryer System
Hours = 6525, Elec. = $0.05/kWh

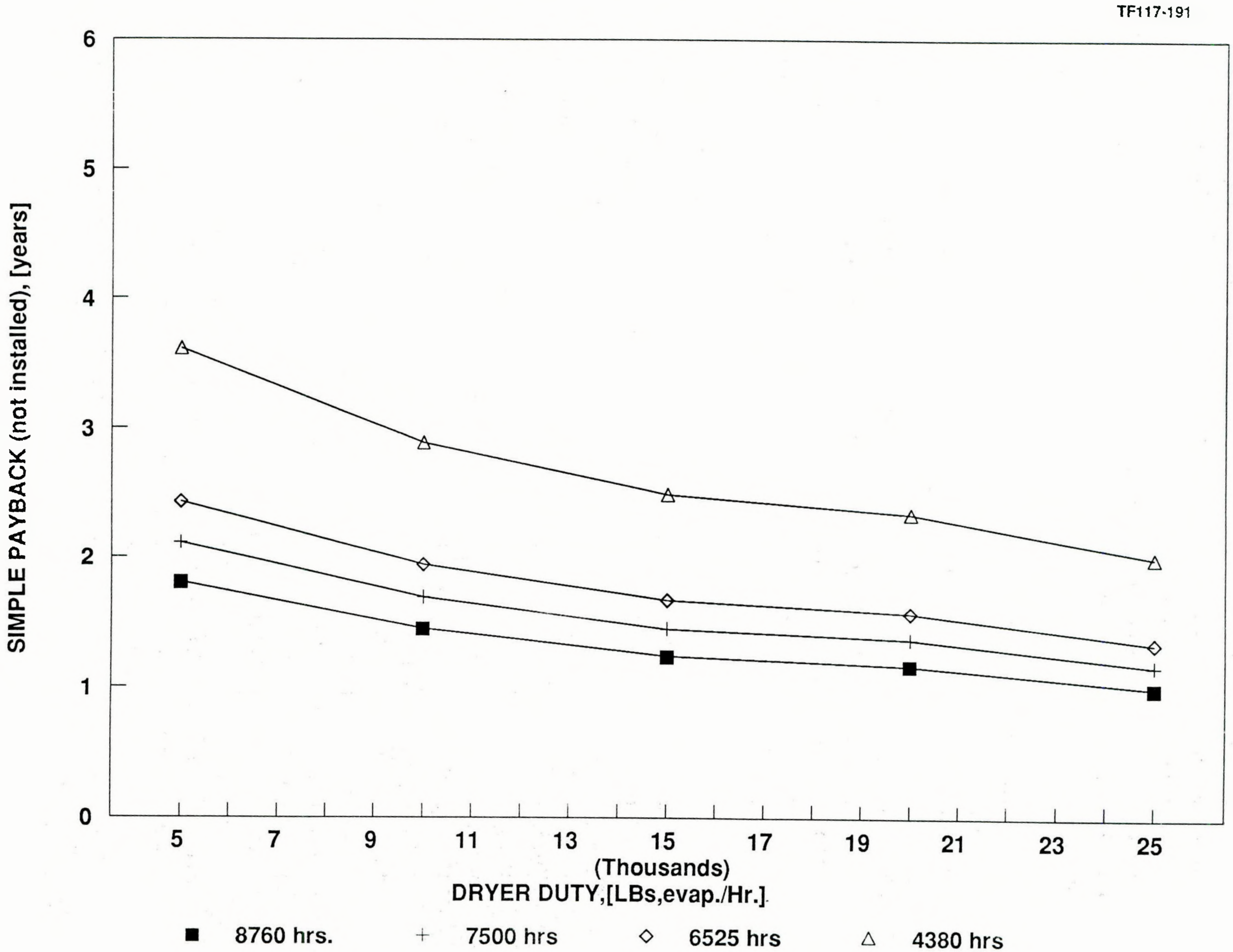

Figure 7.13 Simple Payback for Steam Dryer System
Elec. = $0.05/kWh, Gas = $3.5/MMBtu

7.5 CONCLUSION

The economic cost savings analysis conducted during this task indicates that the steam atmosphere drying system will enable a 30-percent reduction in the cost to dry a wet feedstock and thus provides a cost savings of more than $100,000 per year. These savings can be provided while requiring a simple payback of less than 2 years. These results are clearly very favorable to the marketing of the steam atmosphere dryer.

From this study it was also determined that the typical "smaller" dryer of 5000 to 7500 lb_{evap}/hr capacity will cost $1 million to $1.25 million. Larger dryers (typically in the range of 15,000 to 20,000 lb_{evap}/hr) will cost $1.7 million to $2 million. These costs are high enough to attract current manufacturers of air dryer systems whose average unit sale is in the $1 million to $1.5 million range. Tecogen's research of present U.S. industrial drying requirements (see Task 1 summary) indicates that a 10-percent market penetration represents the sale of 336 "small" units or 93 "large" units and thus an annual steam atmosphere dryer manufacturing business of $19 million to $35 million per year for 10 years.

The energy savings incentive for the U.S. drying industry is no less compelling. Given a 53- to 57-percent decrease in drying energy requirements (as was determined previously in this section), the energy saved in the United States could reach 107 to 115 x 10^{12} Btu/hr if 100 percent of the dryers were displaced by steam dryers. A 1-percent displacement would still save a significant amount of energy: 1.2 x 10^{12} Btu/hr. It is also interesting to note that the atmospheric effluent of uncollected feedstock in an air dryer system would be reduced to zero, thus saving an estimated 75,000 tons/yr. Similarly, the exhaust gas emissions of NO_x and CO would be reduced as a consequence of both decreasing the energy requirements of a typical dryer system as well as changing the heating fuel source from gas to electric. For example, a 55-percent reduction in energy requirements represents a NO_x and CO savings of 2200 and 1300 tons/yr, respectively, across the United States. A change in these same pollutants as a result of changing the fuel source from natural gas to electric would be a function of the fuel of choice used in the power generating stations and the difference in the emissions cleanup and/or combustion efficiency for the dryer combustion systems. However, a low estimate of savings in NO_x and CO of 3600 and 2200 tons/yr, respectively, can be discerned from the published data of the energy heat input required for these systems. Clearly, the displacement of existing air dryer systems can contribute to energy saving as well as reduce product effluent and combustion emissions for the U.S. drying industry.

The potential U.S. energy savings of more than 1 x 10^{12} Btu/yr in addition to the reduction of stack effluent (75,000 tons/yr) and combustion exhaust products (over 2,000 tons/yr) are based on Tecogen's expectation to successfully dry temperature-sensitive as well as temperature-insensitive feedstocks; feedstocks identified in Table 3.6, for example. Tecogen's laboratory testing to date has clearly established the ability of the IRIS-type steam dryer to successfully dry clay; i.e., a non-temperature-sensitive material; a material similar to what would be dried in the chemical, stone, clay and glass, mining, and pulp and paper industrial sectors. Tecogen is continuing with attempts to a dry temperature-sensitive material and is encouraged by the success reported by other independent researchers in steam atmosphere drying of temperature-sensitive material. Tecogen expects to be able to successfully dry these products and requires only more experimental testing time with a wider range of food-type products. This additional testing time was anticipated by DOE and was intended to occupy much of the first task (i.e., "Extended Laboratory Testing") of the second phase of the project. This opportunity is still thought to be ideal for continuing to test the ability of the IRIS dryer to dry the more temperature-sensitive products.

8. TASK 6 – PROGRAM MANAGEMENT

In addition to coordinating the current Phase I work effort and ensuring the timely completion of the scheduled work, Tecogen Inc. was also responsible for planning the second phase of the project. This effort would assist in providing a smooth transition into the project's next phase with a minimum of work interruption. A preliminary work breakdown structure for the project's Phase II is given in Figure 8.1. In order to continue with the Phase II work, Tecogen Inc. has received commitments for technical and financial support from APV Crepaco, Inc. and New York State Energy Research and Development Authority (NYSERDA).

APV Crepaco, Inc. is a world leader in the manufacturing of industrial dryers and evaporator equipment. Two divisions of APV Crepaco, Inc., located conveniently in Attleboro Falls, Massachusetts (1 hour from Tecogen's engineering offices) and in Tonawanda, New York, will be available to provide the project the engineering assistance required for implementing Phase II's Scope of Work. Paul Miller, General Manager of APV's Dryer Division, and Peter Worrall, Vice President of Engineering of APV's Evaporator Division, will be the managers of APV's efforts during Phase II. Mr. Miller and Mr. Worrall will communicate with Mr. DiBella to provide the technical and marketing assistance required to complete the work tasks. A summary of these work responsibilities is presented in Table 8.1.

A Tecogen/APV Crepaco preliminary product development plan for the steam atmosphere dryer program has already been prepared and is identified in Figure 8.2. Thus, the proposed Phase II efforts as described in this proposal would eventually lead to a combined engineering project team effort in the subsequent Phase III of the program, culminating in the joint manufacturing and marketing of a steam atmosphere dryer system with steam recompression. Letters of Interest in the project from APV Crepaco are given in Appendix C for reference.

The New York State Energy Research and Development Authority has agreed to participate as a co-sponsor to the program. Citing the advantages that the steam atmosphere dryer system would offer to New York State dryer manufacturers of equipment (dryers, steam fan, steam compressors, heat exchangers and evaporators, for example) as well as users [chemical, food (breweries, dairies, etc.)] NYSERDA will provide co-funding to assist DOE, Tecogen, and APV Crepaco in Phase II. It is also possible that an existing NYSERDA program, E.D.G.E. (Economic Development Through Greater Energy Efficiency) can subsequently be developed to help co-fund an industrial field testing of the system in New York State.

TF112-190

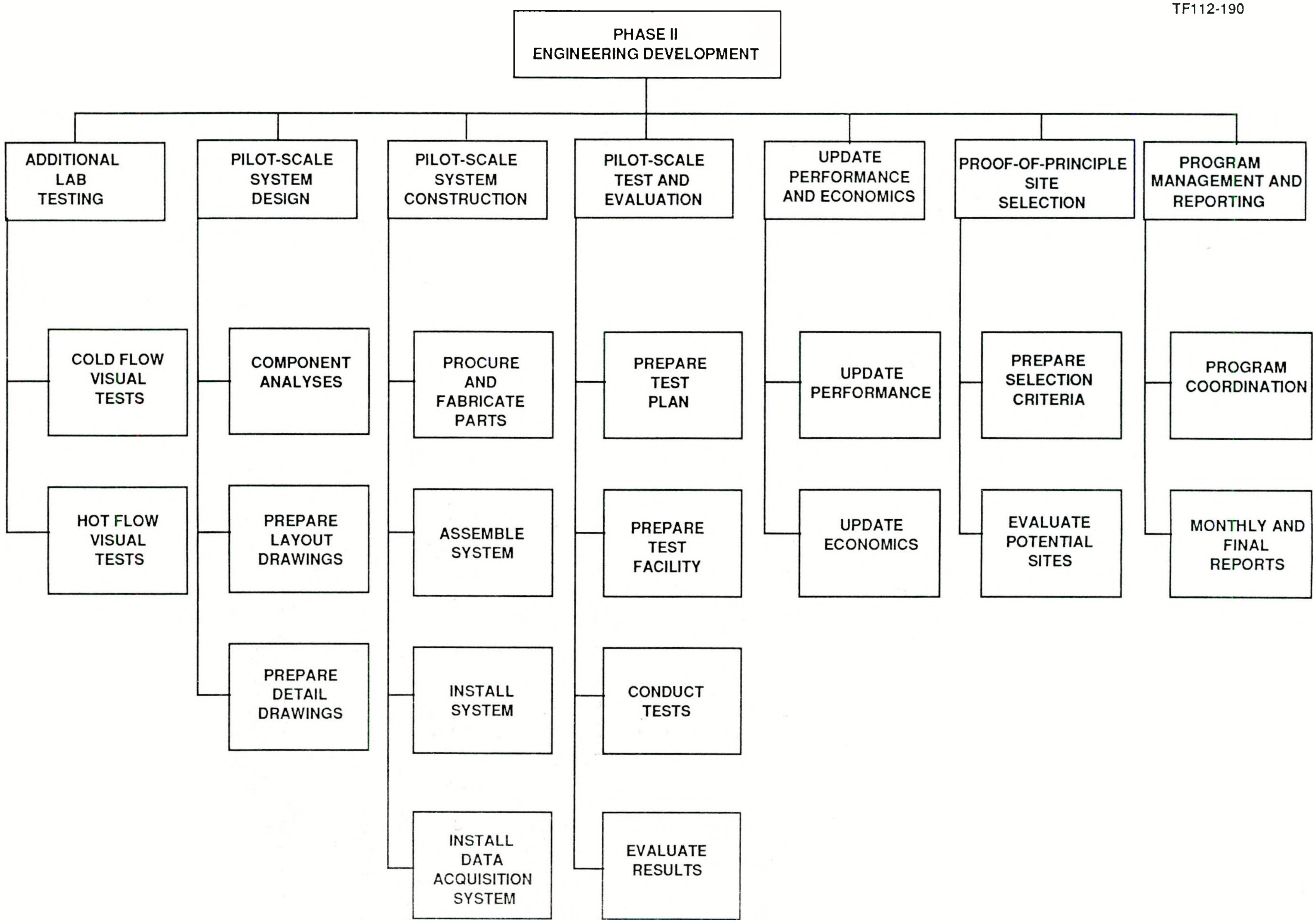

Figure 8.1 Work Breakdown Structure for Phase II

TABLE 8.1

APV TECHNICAL RESPONSIBILITIES
DURING PROJECT

1. Review Tecogen dryer designs and recommend design changes.

2. Review Tecogen laboratory test data and suggest additional laboratory tests.

3. Review concepts for feedstock material handling before and after steam dryer.

4. Participate in non-proprietary review meetings with DOE and Tecogen.

5. Further define market potentials for steam dryer applications.

6. Assist in identifying prototype field site.

Later: 7. Market steam dryer design with Tecogen partnership. (Tecogen provides steam recompression expertise; APV provides steam dryer manufacturing.)

TF126-990

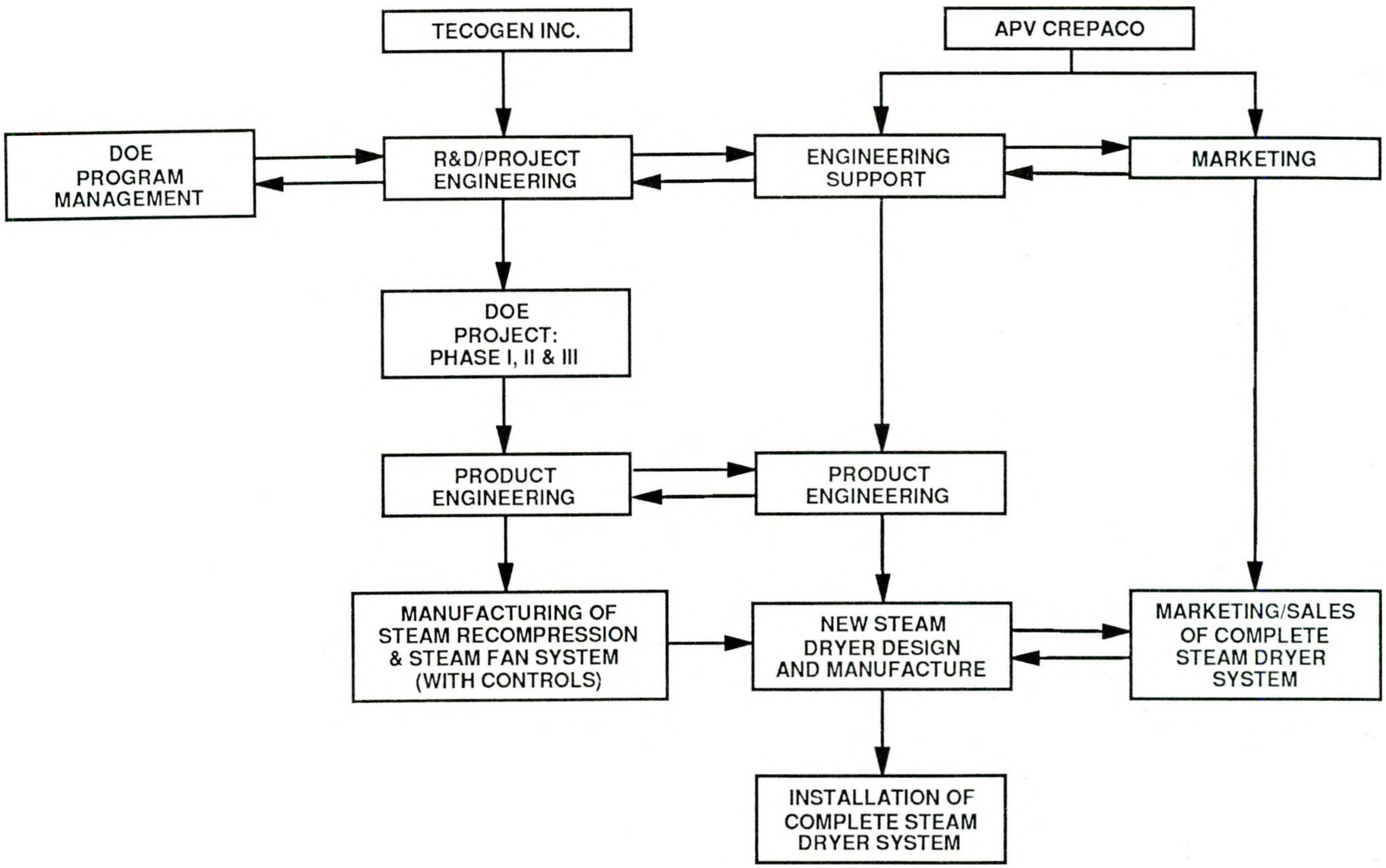

Figure 8.2 Tecogen-APV Crepaco Development Plan for Steam Atmosphere Dryer Development

At this time a Letter of Interest has been received from NYSERDA testifying to their financial interest in co-funding Phase II. Negotiations between Tecogen and NYSERDA for the awarding of these Phase II funds are in progress at this time. The NYSERDA Letter of Interest is reproduced for reference in Appendix C.

The expanded schedule for Phase II of the program: Engineering Development, is shown in Figure 8.3. This phase of the work will begin immediately after Phase I and take 12 months to complete. While the pilot-scale system is being designed and constructed, 6 months of additional development testing will be performed on the Phase I facilities as needed to support the design effort. Pilot-scale system testing and evaluation will begin in the 8th month and continue for 4 months. During the 11th month, the performance and economic predictions will be updated. The Proof-of Principle Site Selection task will be completed by the 10th month. The Program Management and Reporting task extends throughout Phase II.

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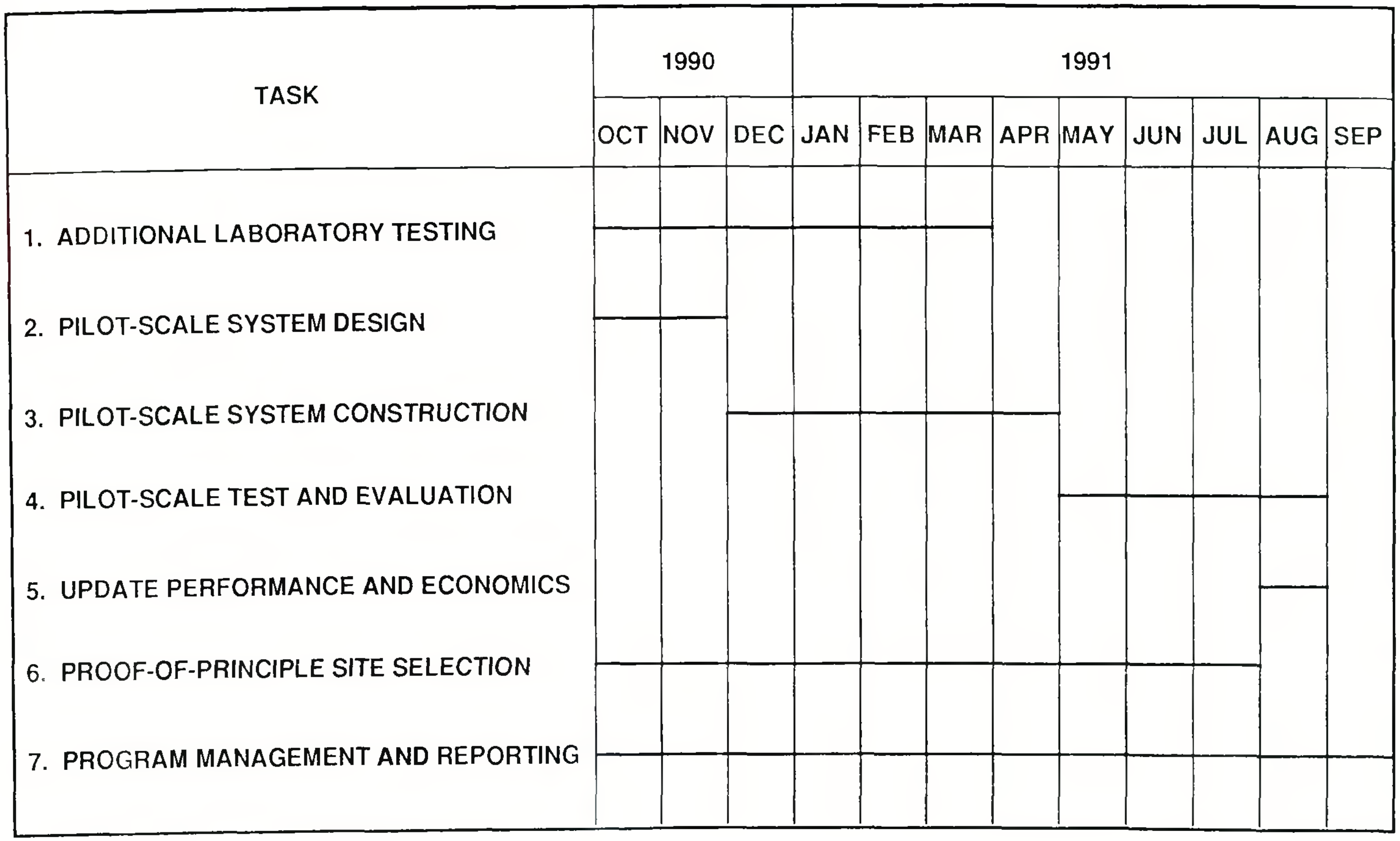

Figure 8.3 Phase II Schedule

APPENDIX A

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APPENDIX B

SELECTIVE LIST OF INDUSTRIAL DRYER MANUFACTURERS

No.	Manufacturer	Ref No.	Specialty Dryer Type
1.	Stork-Bowen Engineering, Inc.	25	Spray; Atomizer
2.	Mizusawa Industrial Chemicals, Ltd.	18	Fluidized Bed
3.	MoDo Chemectics AB S-891 01 Ornskolosvik Sweden	14	Steam Dryer for Pulp
4.	Swedish Energy Technology Inc. Gothenburg, Sweden	19	
5.	Chemical Engineering Research Group Council for Scientific and Industrial Research, Pretoria, South Africa		Steam Spray Drying
6.	THERMEX (PTY) LTD. P.O. Box 2628 Alberton, South Africa 1450	10	
7.	NIRO Atomizer, Inc. 9165 Rumsey Road Columbia, MD 21045 (301) 997-8700	34 and T	Spray Dryer
8.	Wyssmont Co., Inc. Fort Lee, NJ	13	Rotating Dryers
9.	APV Anhydro Division of APV Crepaco Inc. 120 John L. Dietsch Sq. Attleboro, MA (508) 695-7014	T	Spray and Fluidized Bed
10.	APV Crepaco, Inc. 395 Fillmore Ave. Tonawanda, NY 14150 (716) 692-3000	T	Process Equipment
11.	Barr & Murphy, Ltd. 92 Boulevard Prevost, Suite 300 Quebec, Boisbriand Canada J7G2S2 (514) 437-5252	T	Specialists in Design of Spray Dryers

T: Thomas Registry
W: Web Systems

No.	Manufacturer	Ref No.	Specialty Dryer Type
12.	Dimat, Inc. P.O. Box 233 Cedarburg, WI 53012 (415/4) 377-3050	T	Custom Spray Drying
13.	Enders-Process Equipment Corp. P.O. Box 308 Glen Ellyn, IL 60137 (708) 469-3787	T	Multiple Effect Evaporators
14.	Henningsen Foods, Inc. 2-T Corporate Park Dr. White Plains, NY 10604 (914) 694-1000	T	Spray Drying and Power Removal Systems
15.	Mohr Industrial Corp. P.O. Box 1148-T Dearborn, MI 48121 (313) 846-3000	T	Industrial Processing Equipment
16.	PSP Industries 300-T Montague Expressway Milpitas, CA 95035 (408) 942-1155	T	Custom Spray Dryers
17.	Pyro Processing Co., Inc. Dept. D Balligomingo Road W. Conschocken, PA 19428 (215) 825-1166	T.	Custom Drying of Solids
18.	Rogers, C.E., Co. Box 118 Mora, MN 55051 (612) 679-2172	T	Manufacturers P.O. of Spray Dryers
19.	Swenson Process Equipment, Inc. 15700 Lathrop Ave. Harvey, IL 60426 (708) 331-5500	T	Machinery for Process Industry (Including Dryers)
20.	Custom Products Louisville Drying Machinery Division 1100 Industrial Blvd. Louisville, KY 40219 (502) 969-3163	T	Manufacturers of Rotary Dryers

T: Thomas Registry
W: Web Systems

No.	Manufacturer	Ref No.	Specialty Dryer Type
21.	Industrial Kiln & Dryer (An ACL Co.) 4 N. 12th St. Council Bluffs, IA 51501 (712) 328-3030	T	Manufacturers of Flash Dryers
22.	Heil Manufacturing Co. Milwaukee, WI	W	
23.	A-C	W	
24.	Taco Corp.	29	
25.	Red Ray Manufacturing Co., Inc.	29	
26.	Western Precipitation Corp.	29	
27.	J.W. Greer Co.	29	
28.	Edward, Renneburg & Sons	29	
29.	F.J. Stokes Machine Co.	29	
30.	Hardinge Co.	29	
31.	Traylor Engineering and Manufacturing Co.	29	
32.	Wyssmont Co.	29	
33.	The A.P.V. Company Limited P.O. Box 4, Manor Royal Crawley, West Sussex c/o P.W. Dickinson	27	Manufacturers of Spray Dryers

T: Thomas Registry
W: Web Systems

APPENDIX C

LETTERS OF INTEREST

COPY

APV

APV Crepaco Inc
165 John L. Dietsch Square
Attleboro Falls, MA 02763
Tel.: (508) 695-7014
Fax: (508) 695-7018
Telex: 92-7634

July 30, 1990

Mr. Frederick E. Becker
Director, Energy Technology
TECOGEN INC.
45 First Avenue
P. O. Box 9046
Waltham, MA 02254-9046

Re: Drying with Super-Heated Steam
DOE / Tecogen Project

Dear Mr. Becker:

This letter is to formally state that APV Crepaco, Inc. is interested in participating in the research project described in Tecogen Inc. proposal TP 017-90.

APV Crepaco actively participates in the field of industrial drying, evaporation and distillation, as well as heat transfer, mixing and fluid flow, and believes that this technology, coupled with Tecogen's can be joined together to bring about more effective drying operations.

We feel that the areas of food and dairy drying, pollution and environmental projects, and the drying of chemicals and minerals have enormous potential for energy enhancement, and we would also include drying or stripping of materials containing volatile organic solvents.

As we understand, our participation initially includes for the review and critique of various technical documents, but then would move into the engineering and supply of your prototype dryer design.

For the initial work, I would be your contact in the APV organization and later your contact would be one of our project engineers who would become intimately involved in the project.

Mr. Frederick E. Becker
Director, Energy Technology
TECOGEN INC.
July 30, 1990
Page 2

APV

It is recognized by both Tecogen and APV that no definitive agreement has been reached for the commercialization of the process. However, by its participation in the development of the technology, it is agreed by both parties that APV will have a priority status with a right of first refusal.

In closing, let me say that APV looks forward to this technical relationship with Tecogen and we feel we can be a meaningful contributor to the program.

Very truly yours,

Paul H. Miller
General Manager
Dryer Division

nmf

cc: Derek Dinnage , Lake Mills, Wi. ; Vice President, Technical Director
Peter Worrall , Tonawanda, N.Y.; Vice-President, Chem. Div.

APV

APV Crepaco Inc
165 John L. Dietsch Square
Attleboro Falls, MA 02763
Tel.: (508) 695-7014
Fax: (508) 695-7018
Telex: 92-7634

September 6, 1990

Mr. Francis A. DiBella
TECOGEN INC.
45 First Avenue
P.O. Box 9046
Waltham, MA 02254-9046

Re: Super-Heated Steam Research Project

Dear Frank:

APV acknowledges receipt of your August 10 letter and the attached work project statement.

We have reviewed the statement and your estimated hour input that is expected from APV, and wish to indicate that your proposed hours to be expended by APV are reasonable and are within the time that we felt could be allotted the project.

We look forward to your obtaining approval from DOE to proceed with phase two at which point we can become involved in the project.

Very truly yours,

Paul H. Miller
General Manager
Dryer Division

nmf

cc: Peter Worrall
APV Crepaco - Tonawanda, NY

Derek Dinnage
APV Crepaco - Lake Mills, WI

APV

APV Crepaco Inc
165 John L. Dietsch Square
Attleboro Falls, MA 02763
Tel. (508) 695-7014
Fax. (508) 695-7018
Telex: 92-7634

September 11, 1990

Mr. Francis A. DiBella
TECOGEN INC.
45 First Avenue
P. O. Box 9046
Waltham, MA 02254-9046

Re: Super-Heated Steam Research Project

Dear Frank:

In a phone conversation this morning you requested we quantify our letter of September 6 and assign a dollar value of the proposed APV contribution to the Super-Heated Steam Research Project.

We estimate that APV will expend 500 hours of engineering and other professional time which would be billed at $75.00 per hour, if this were a typical commercial project, or a total of $37,500.00. Inasmuch as APV has its equipment fabricated by others, the limit of our contribution in this phase we estimate to be $10,000.00, the engineering and inspection portion of the fabrication phase.

In summation, the APV contribution would be $47,500.00.

Please call if you have any questions on this estimate.

Very truly yours,

Paul H. Miller
General Manager
Dryer Division

nmf

cc: Peter Worrall
APV Crepaco - Tonawanda, NY

Derek Dinnage
APV Crepaco - Lake Mills, WI

New York State
Energy Research and Development Authority
Two Rockefeller Plaza • Albany, New York 12223-9998
(518) 465-6251

WILLIAM D. COTTER
Chairman

IRVIN L. WHITE
President

September 27, 1990

Mr. Frank Di Bella, P.E.
Tecogen Inc.
45 First Avenue
P.O. Box 9046
Waltham, MA 02254-9046

Subject: Proposed Agreement No. 1606-EEED-IE-91, Steam Atmosphere Drying of Industrial Solids with Dryer Exhaust Steam Recompression

Dear Mr. Di Bella:

This is to advise you that the Tecogen proposal for conducting the above project (Phase II, "Engineering Development", of the DOE Project) has been reviewed by the Energy Authority's Research and Development Management Committee (RDMC) and has been recommended by the RDMC for proceeding to the Technical Review Committee (TRC) for review and evaluation.

Please note on the attached Project Planning Request that we contemplate our cofunding will be $165,000for this project.

The TRC will meet to review the project (including Mr. Reinhardt's verbal presentation) and to make recommendation to the RDMC. Assuming a positive TRC review and subsequent RDMC positive action, the RDMC would recommend this project for contract negotiation. This includes:

1. Reaching agreement with the Energy Authority on mutually acceptable terms and conditions.

2. Reaching agreement with the Energy Authority on a reasonable cost and time schedule for performing the work.

The agreement is then subject to final executive approval before execution by both parties of the contract.

This is to advise you that the Energy Authority will not be responsible for any costs incurred by you for this project should a contract not be consummated.

Very truly yours,

Robert Carr