FINAL REPORT

A HIGH SEASONAL PERFORMANCE FACTOR GAS HEAT PUMP FOR THE NORTH CENTRAL UNITED STATES

Prepared By: Consolidated Natural Gas Service Company Inc.
Prepared For: The U.S. Department of Energy
Prepared Under: DOE Prime Contract EY-76-C-02-2883

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Consolidated Natural Gas Service Company Inc.

Mechanical Technology Inc.

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EXECUTIVE SUMMARY

ES 1.0 BACKGROUND

The High Seasonal Performance Factor (HSPF) gas heat pump development and demonstration program was initiated by Consolidated Natural Gas Service Company (CNG) and the U.S. Department of Energy (DOE) in February, 1976. Mechanical Technology Incorporated (MTI) was selected as the principal subcontractor. A Program Advisory Board was immediately established for the purpose of reviewing program objectives on a regular basis and providing recommendations for each major program decision.

The HSPF System technology upon which the program was based had a direct anticedent in a residential steam turbine-driven gas heat pump development program conducted between 1965 and 1969 under the sponsorship of CNG, Lone Star Gas Company, Northern Illinois Gas Company, Southern California Gas Company, Texas Eastern Gas Transmission Company and Washington Gas Light Company. The residential heat pump development program demonstrated the energy conservation effectiveness of certain system arrangements, in particular the two-step heating of conditioned air via heat exchangers arranged in series. This program also revealed the great difficulty in achieving an acceptably low cost for a residential sized steam turbine-driven gas heat pump.

Between 1973 and 1975, CNG sponsored a related project, aimed at the development of a superior-efficiency steam turbine for use as a gas heat pump driver. Analysis of project results neither confirmed nor denied the feasibility of such a turbine. CNG terminated this effort in order to direct available resources to the development and demonstration of a complete system.

The residential program established that the HSPF System had an excellent potential for energy conservation but would not be competitive as a residential sized product. Through economies of scale, a more competitive product should have been possible in larger size systems. Accordingly, the DOE/CNG/MTI HSPF program was focused initially on a 7½-ton, single-zone, roof-top package system. This capacity rating and this system configuration were selected for initial demonstration because they represented

the largest market volume portion of the commercial HVAC equipment market. Also, the single-zone package system configuration would preserve the HSPF features demonstrated previously. Finally, 7½-tons would represent the smallest practical capacity for the HSPF turbomachinery elements and would thereby provide a limiting case demonstration of turbomachinery performance. A schematic of the 7½-ton HSPF system is shown on Figure ES-1.

ES 2.0 7½-TON PROGRAM

The 7½-ton program consisted of these principal elements:

- Technical Effort
- Market Potential Study
- Comparative Cost of Ownership Study
- System Cost Analysis

The Technical Effort, initiated ahead of the other program elements, included system specifications, component and system design, component and subsystem bench test, and a full 7½-ton breadboard system demonstration test.

The Market Potential Study, the Comparative Cost of Ownership Study and the System Cost Analysis for a 7½-ton HSPF product were all completed by July, 1977. These efforts produced the following information:

1. 7½-ton HSPF product cost estimates indicated that the system would have an equipment first-cost premium of several hundred percent relative to competitive conventional package equipment.

2. The first-cost premium tended to reduce as unit size was increased. The premium could become manageably small at 75 to 125 tons.

3. An installation cost advantage was shown to exist for the HSPF system. The cost advantage represented 30% of the cost of competitive equipment at 7½ tons and 25% at 30 tons. The advantage was due to elimination of the requirement for a heavy electric power connection.

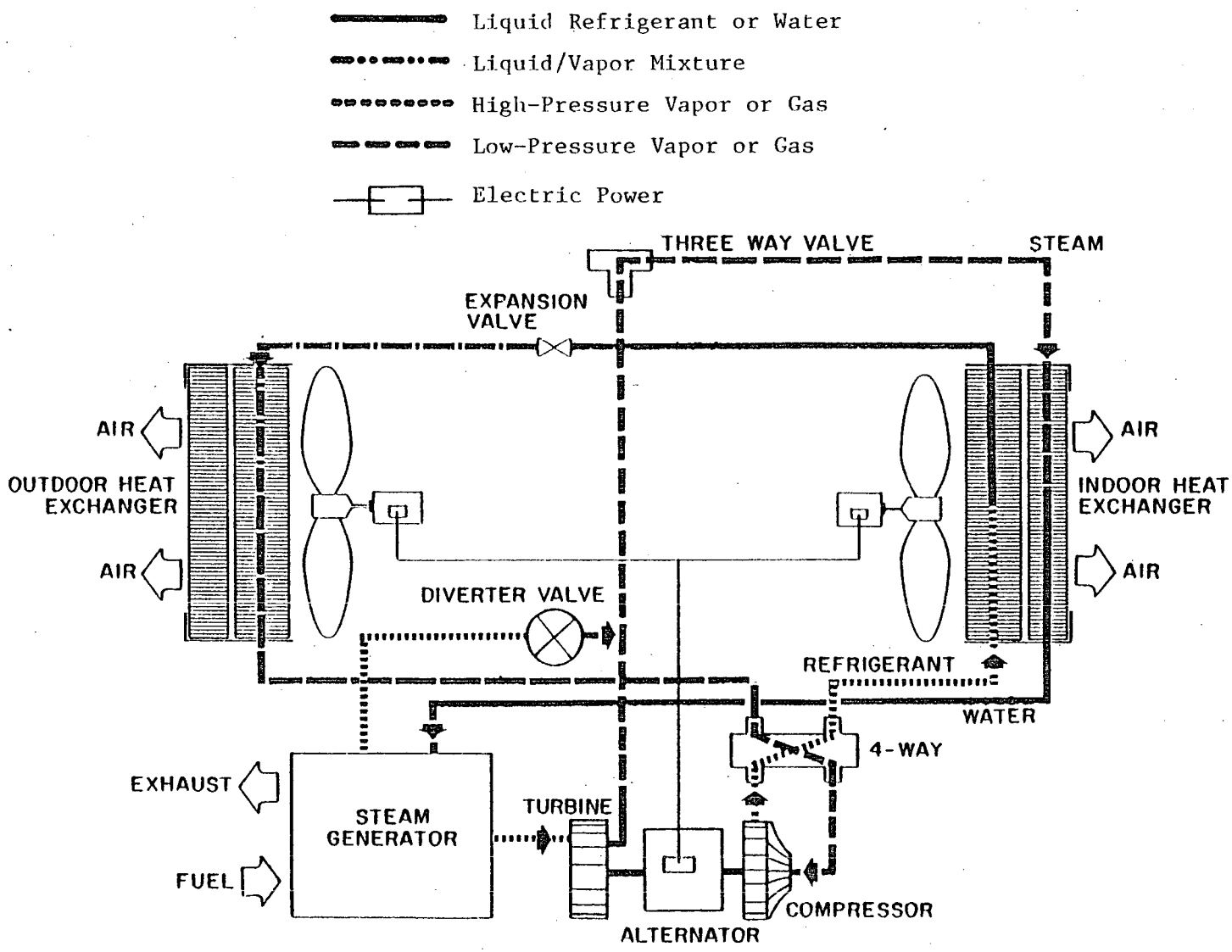

Fig. ES-1 7½-Ton Unit Heating Service Schematic

4. The Market Potential Study, conducted by William E. Hill & Co., indicated that the estimated first-cost premium would result in an unacceptably small market at the 7½-ton size level.

5. In spite of the very large first-cost premium of several hundred percent, the 7½-ton HSPF system appeared to offer payback periods of 10 to 15 years depending on the application and, especially, on local utility rates.

The Program Advisory Board reviewed this information and recommended that the 7½-ton program be terminated with the exception of the technical demonstration effort. The Board also recommended that DOE and CNG consider redirecting the balance of the program to larger sized systems. The basis for this recommendation was the perceived trend toward a more competitive system cost as system size increased, and an indication of greater acceptance of the principals of life-cycle cost analysis in the large built-up equipment market.

In consideration of the above information and the Board's recommendations, CNG and DOE terminated the 7½-ton program in September, 1977, with the exception of the technical demonstration effort. The 7½-ton breadboard system demonstration was successfully concluded in July, 1978. The breadboard system's performance was equivalent to seasonal performance factors of 1.18 and 0.60 for the heating and cooling seasons, respectively, when operated as required for a retail building load and Cleveland weather. The demonstrated performance corresponded to a 50% reduction in HVAC-related energy consumption relative to conventional equipment. The breadboard system demonstration indicated that the design performance objectives of 1.21 SPF heating and 0.62 SPF cooling were achieveable in a production unit.

ES 3.0 LSM PROGRAM

DOE and CNG considered the Board's further recommendation and in September 1977, initiated the first phase of a planned multiphase program to develop a large size model (LSM) of the HSPF. The principal LSM program objectives were first to demonstrate the total potential of the LSM HSPF gas heat pump as a profitable product line and then to secure a definite arrangement for

commercialization. With the successful demonstration of HSPF System technology with the 7½-ton demonstrator, no full hardware development or further technical demonstration were needed in the first phase of the LSM program. Therefore, the first phase of the LSM program was planned around the following major elements:

- Supporting Technical Effort
- Market Potential
- System Cost Analysis
- Comparative Cost of Ownership Study (CCOS)
- Business Potential Study

A second phase of the LSM program, that would have resulted in an initial full-scale field demonstration, was also planned. DOE, CNG and MTI commitment to the second and subsequent phases was predicated on positive results from the Comparative Cost of Ownership Study and Business Potential Study, and on securing arrangements for commercialization.

The first-phase technical effort was limited to a system optimization analysis and modeling task and a design-for-cost analysis task. The most cost-effective LSM configuration identified through system optimization for meeting most HVAC loads of 100 tons and up did not incorporate a heat pumping function. The optimized LSM system used the steam turbine prime mover to drive a chiller and also to generate electric power for use within the building. Heating loads were to be met entirely through heat recovery from the steam prime mover.

The LSM system modeling effort demonstrated that the value of the energy savings possible through heat pumping would be far too small to justify the added components. The heat pumping function was, therefore, ruled out on economic grounds. Failure to economically justify the heat pumping function was due to the relatively small heating load in the modern office building upon which the analysis was based. Also, the heating load that did exist occurred generally at very low outdoor temperatures when outdoor air is an undesirable heat source.

Nominal nameplate cooling capacity for the initial LSM configuration for the first phase study was set at 120 tons. However, the LSM product configuration was planned for application to loads ranging from 60 to 400 tons.

Results from all of the first phase LSM program elements were available by April, 1979. These results are listed below.

1. The equipment first-cost premium for the LSM has been estimated at about 70% over the best conventional HVAC system. This amount would pose a significant barrier to market penetration.

2. HSPF system efficiency advantages were found to be largely eliminated in the context of a hot and chilled water distribution system and multizone loads. System seasonal efficiency was further reduced for HVAC loads dominated by the cooling load.

3. The LSM was shown to offer little or no return on investment or source energy savings in applications representing the majority of the 60- to 400-ton equipment market.

4. A relatively small market was identified where the LSM system should represent an attractive investment or means for conserving source energy. The market is characterized by locations where gas and electric energy costs are, or are expected to be, relatively high and by applications where the heating load predominates.

5. The limited market potential projected for the LSM, combined with the traditional difficulty in financing new products in the HVAC industry, led to a very low estimate for the probability that a business venture based on an LSM product could succeed.

Following a review of the above results, the CNG, DOE and MTI project management team determined that all technical effort on the LSM program should be terminated. The basis for this determination was the very low

potential indicated for the LSM HSPF gas heat pump to become the basis for a profitable product line. The Program Advisory Board met in May 1979, and after reviewing the results, confirmed the project management team's findings. CNG, DOE, MTI and the Board recommended no further effort on a steam turbine-based HSPF System.

ES 4.0 TECHNOLOGY SUMMARY

A summary comparison of the technical characteristics of the 7½-ton and LSM HSPF product concepts is presented in Table ES-I.

ES 5.0 CONCLUSIONS

The conclusions drawn from the results achieved through the 7½-ton and LSM HSPF programs are reviewed here. For the purpose of review, these conclusions are organized according to the following categories:

1. HSPF technology
2. Prospects for a 7½-ton HSPF unit product
3. Prospects for an LSM HSPF product.
4. Prospects for a heat actuated heat pump product.
5. Prospects for advanced HVAC products.

ES 5.1 Conclusions Related to HSPF Technology

The term "HSPF technology" refers here to the several heat pump system features described in detail in Sections 2.0 and 5.0. These features are:

- Pairs of indoor and outdoor heat exchangers arranged in series.
- Modulated operation of the steam turbine driver.
- Supplemental space heating capacity via direct injection of steam from the steam generator.
- Modulated self-propulsion of system accessories (fans and blower).

These features were fully incorporated into the 7½-ton unit. The 7½-ton unit was demonstrated to be a technically effective approach to energy con-

TABLE ES-1

SUMMARY OF TECHNICAL CHARACTERISTICS OF THE 7½-TON AND LSM HSPF GAS HEAT PUMP PRODUCT CONCEPTS EVALUATED IN THE PROGRAM

Characteristics	7½-Ton HSPF System	LSM HSPF System
Steam Description	Rooftop Air-to-Air, Single Zone, Unitary Heat Pump with Self-Powered Auxiliaries	Central, Built-Up, Steam-Turbine-Assisted Chiller with Power Generation and Heat Recovery
Operational Features	Speed and Load-Modulated Capacity	Constant Speed, Constant Steam Turbine Output; Electricity Generated or Consumed Depending on HVAC Load
Steam Cycle Entry	1100°F, 300 psia [1 stage]	1050°F, 815 psia [1 - Two-Row Stage]
Reheat	1100°F, 30 psia [1 stage]	1050°F, 64 psia [1 - Two-Row Stage, 1 - Single-Row stage]
Regenerator	635°F, 3 psia	483°F, 2.4 psia
Condenser	150°F, 3 psia	140°F, 2.4 psia
Turbomachinery Package Configuration	Single Shaft Turbo-Alternator-Compressor, with Variable Speed from 40,000 to 60,000 rpm. Single Disk Partial Admission Turbine with Reentry, High-Speed Homopolar Alternator, Single-Stage Centrifugal Compressor	Multi-shaft, Geared, Turbo-motor/Generator - Compressor Fixed Speed. Two-Wheel Turbine with one Two-Row Partial Admission Wheel and One One-Row Full Admission Stage at 40,000 rpm, Induction Motor/Generator at 3600 rpm, Single-Stage Centrifugal Compressor with Guide Vanes at 24,000 rpm
Overall Power Cycle Efficiency (1)	21.8% To 26.2%	21.8%
System Design Performance (2)	7½ Tons Rated Cooling and a 0.62 SPF, 120,000 Btu/hr Rated Heating and a 1.21 SPF	120 Tons Cooling with a 1.056 COP, 56,500 Btu/hr Heating and 50 kW of Electricity with an 84.6% Efficiency
Estimated Selling Price	$12,383	$90,331
Selling Price of Standard High-Performance Alternative	$2,440	$54,486

(1) Power cycle efficiency based upon higher heating value of fuel into shaft power output of turbine.

(2) Seasonal performance factors, coefficients of performance, and efficiencies based upon higher heating value of fuel consumed at point of consumption.

servation for single zone heating and cooling loads, especially where the heating load was dominant.

The LSM program, however, demonstrated that the HSPF System suffered significant performance degradation when adapted to the needs of larger HVAC systems, which included hot and chilled water distribution and multizone loads. Degradation was pronounced where the application was dominated by the cooling load. This result was found to generally apply to analog HSPF LSM systems employing alternative prime mover drivers to the high performance steam turbine system. The steam turbine driven system, however, suffered more than systems driven by prime movers with superior design-point efficiency. This degradation was attributed to the nature of the larger HVAC system loads, which depended more upon design-point operation, were more likely to be dominated by cooling requirements and which were less able to utilize the HSPF features than smaller, single-zone, air-to-air systems.

For either the 7½-ton or the 120-ton (LSM) application the high-performance steam prime mover was too costly to manufacture to be employed in a competitive heat pump system. This result would apply for the steam turbine driven heat pumps under several hundred tons nameplate capacity. As a consequence, it was shown that there was no reasonably profitable scenerio within which a business venture based upon manufacturing and/or marketing a steam driven HSPF System could expect to realize an attractive rate of return.

The excessive cost premium for the high-performance steam turbine prime mover was found to be attributable in large part to:

- Lack of production-oriented system design experience;
- Lack of production technology;
- Lack of a production capacity base.

Development of production experience and technology was not addressed within the scope of the HSPF programs. The principal cost elements in the high-performance steam turbine prime mover were found to be the following, listed here in order of importance:

- Controls, valves and plumbing
- Heat exchangers

- Steam generator subsystem
- Turbine expander subsystem
- Water treatment subsystem

Despite the severe cost obstacles posed by the high-performance steam turbine prime mover, it was shown to offer acceptable operating efficiency in the context of the 7½-ton application. The steam prime mover also demonstrated the ability to provide these other desirable operating characteristics:

- Quiet operation
- Long service life, with infrequent scheduled maintenance
- Low emissions when fired with natural gas or light oils.

ES 5.2 Prospects for a 7½-Ton Unit Product

The 7½-ton, steam-driven HSPF System was demonstrated to be a technically effective approach to energy conservation for single zone heating and cooling loads, especially where the heating load was dominant. Technical effectiveness was demonstrated in terms of:

- Significantly reduced energy consumption relative to conventional equipment
- Favorable system operating characteristics for HVAC application
- Total technical compatibility of system components with HVAC application.

The 7½-ton HSPF steam-driven system was also shown, however, to have an equipment cost far too high to be competitive as an HVAC product. This conclusion is valid in spite of the fact that acceptable equipment payback times were indicated for some application scenarios through 1985. The excessive system cost was primarily attributable to the steam turbine prime mover system.

ES 5.3 Prospects for an LSM HSPF Product

The LSM version of the HSPF System suffered performance degradation when adapted to the needs of hot and chilled water distribution systems and multi-

zone loads. Also, energy utilization efficiency was shown to decline anywhere the application was dominated by the cooling load.

The equipment first-cost premium over competing equipment was shown to be far less severe for the LSM system than for the 7½-ton system. Because of the reduced performance margin, however, the market applications for which an LSM HSPF System would be competitive were found to be very limited. The more favorable applications were characterized by relatively high heating loads and by market areas where energy costs were expected to be very high, with the projected rate of cost increase for natural gas not being significantly more than electricity.

The most competitive version of the LSM system did not incorporate a heat pumping function. Within the constraints posed by large building HVAC load profiles and hot- and chilled-water multizone distribution systems, cogeneration of thermal energy and electricity for in-building use was demonstrated to be an economically more attractive application than heat pumping.

The technical effort under this program provides some basis upon which to expect more encouraging results with an organic Rankine system. This result is far from conclusive, however. This same effort failed to identify any novel system approach or component for a steam-Rankine system that would have resulted in a viable heat pump system.

ES 5.4 Prospects for a Heat Actuated Heat Pump Product

The results reviewed above were developed from studies of gas heat pump systems incorporating steam-Rankine power cycles. In this subsection, the applicability of these results to gas heat pumps incorporating other power cycles, in general, is considered.

The results show that in addition to external factors such as utility rate schedules, it is the overall system performance in an actual building application and the overall system installed cost that defines commercial viability. The performance potential of a new Rankine fluid or even a whole new prime mover with a high potential for performance will in and of them-

selves not make a commercially viable gas heat pump system. The innovation offered must address overall system performance and/or system cost.

Drawing upon the results developed through the HSPF programs, criteria have been developed that should apply to any successful heat-actuated heat pump ventures. These criteria are as follows:

1. Commercial availability of a prime mover than could be produced at less than $200/kW when integrated with a heat-actuated heat pump system. This prime mover should share the favorable operating and service life characteristics of the steam turbine.

2. Applicability to a major heating-cooling equipment market characterized by annualized heating-to-cooling load ratios greater than unity (1.0) and by energy costs significantly higher than those prevailing in most utility service areas in 1979.

   The residential and small (or light) commercial equipment market would qualify in terms of heating-to-cooling load ratio and did not pose the disadvantages of multizone loads or hot- and chilled-water distribution.

3. Public confidence in the sustained availability of natural gas for commercial and residential HVAC application.

4. Prospects for the commercial success of any technically adequate heat actuated heat pump will depend more strongly on local utility rate levels and rate schedules than on the local climate or building characteristics. Therefore, preservation of a favorable ratio of the cost of gas versus electricity in utility service areas considered as a potential market is a critical criteria.

ES 5.5 Prospects for Advanced HVAC Products

The HSPF programs focused upon commercial size HSPF heat-actuated heat pump systems. Some of the results that were obtained, however, are applicable to the development and commercialization of any advanced commercial HVAC system. Some of these results are as follows:

1. Equipment first-cost consideration continues to represent the most important single purchase criterion. While an increasing acceptance of life-cycle cost analysis exists, the life-cycle basis continues to be secondary to equipment first cost as a purchase criterion. It was found that the average life-cycle cost-sensitive buyer will consider a payback of three years on equipment investment as acceptable only if the first-cost premium is not more than 25%. The super-position of equipment first-cost considerations on life-cycle cost evaluation may totally preclude commercial acceptance of some types of advanced energy conserving HVAC equipment that would otherwise represent an attractive investment.

2. The profitability of virtually all HVAC system manufacturers and assemblers is so low that revolutionary new product development and its commercial introduction cannot be justified solely by an economic venture analysis.

3. The capital formation required to support manufacturing and sales in the HVAC industry is very high relative to the capital generated from sales. This characteristic makes it difficult to economically justify the introduction of a new product even if the development costs are totally underwritten by another party (e.g. the government).

ES 6.0 RECOMMENDATIONS

1. Any further technical development of heat-actuated heat pump systems based upon a high performance steam turbine prime mover should be predicted on a major breakthrough in terms of system cost reduction. Innovative breakthroughs are especially required to reduce the cost of producing controls, valves, plumbing and heat exchangers.

2. For most large building HVAC applications, co-generation was found to represent an economically more attractive option than a heat actuated heat pump. In-building co-genera-

tion should, therefore be considered as a large building alternative to the heat actuated heat pump.

3. Screening for candidate heat-actuated heat pump systems should focus on these prime mover attributes:

   - Production in appropriate volume at under $200/kW, or for even less for residential application
   - Favorable operating and service life characteristics of the steam turbine
   - System features that represent a favorable synergism of HVAC heat pump application and a heat-actuated prime mover.

4. Regular re-evaluations of the acceptance of true life-cycle costing as the primary purchasing criterion should be conducted. This can be accomplished by replicating the surveys made by William E. Hill and Company for CNG and DOE every two or three years.

5. To minimize total product development cost, a recommendation was made that the first phase of any associated technical effort be limited to support of an analysis of overall economic and venture prospects. The likelihood that this approach will result in lost time to ultimate product introduction is very small. A review of past efforts for gas heat pump product development suggests that a thorough early economic and venture analysis is likely to save time overall, as well as money.

6. The government must take a broader view of its role in the commercialization of energy conservative products. In addition to the technology development and consumers' needs, the economic needs of the commercializing agent must also be considered and supported through incentives or regulation.

TABLE OF CONTENTS

Section		Page
	EXECUTIVE SUMMARY	ii
ES 1.0	BACKGROUND	ii
ES 2.0	7½-TON PROGRAM	iii
ES 3.0	LSM PROGRAM	v
ES 4.0	TECHNOLOGY SUMMARY	viii
ES 5.0	CONCLUSIONS	viii
	ES 5.1 Conclusions Related to HSPF Technology	viii
	ES 5.2 Prospects for a 7½-Ton Unit Product	xi
	ES 5.3 Prospects for an LSM HSPF Product	xi
	ES 5.4 Prospects for a Heat Actuated Heap Pump Product	xii
	ES 5.5 Prospects for Advanced HVAC Products	xiii
ES 6.0	RECOMMENDATIONS	xiv
	LIST OF FIGURES	xvix
	LIST OF TABLES	xxii
	IDENTIFICATION OF TERMS AND CONVENTIONS	xxiv
1.0	INTRODUCTION	1-1
	1.1 7½-Ton Program Overview	1-7
2.0	7½-TON TECHNICAL DEMONSTRATION	2-1
	2.1 HSPF 7½-Ton System Description	2-1
	2.2 Size Selection	2-6
	2.3 System Design	2-12
	2.4 Component Design	2-14
	2.4.1 Steam-Turbine Cycle	2-14
	2.4.2 Steam Generator	2-18
	2.4.3 Steam Turbine	2-18
	2.4.4 Alternator	2-21
	2.4.5 Refrigerant Compressor	2-21
	2.4.6 Regenerator	2-21
	2.4.7 High-Speed Unit	2-21
	2.5 Component Test	2-25
	2.6 Proof-of-Concept Test	2-25

TABLE OF CONTENTS (Cont'd)

Section		Page
3.0	7½-TON ECONOMIC ANALYSIS	3-1
	3.1 Overview	3-1
	3.2 7½-Ton Comparative Cost of Ownership Study	3-1
	3.3 7½-Ton Unit Venture Prospects	3-10
	3.4 Program Redirection	3-14
4.0	LARGE SIZE MODEL PROGRAM OVERVIEW	4-1
5.0	MARKET POTENTIAL STUDIES	5-1
	5.1 Market Characteristics	5-4
	5.2 Price Sensitivity	5-7
	5.3 Market Potential	5-14
	5.4 Key Observations	5-19
6.0	LARGE SIZE MODEL TECHNICAL EFFORT	6-1
	6.1 Introduction to the Large Size Model	6-1
	6.2 System Design Analysis	6-3
	6.3 Component Design Effort	6-10
	6.3.1 General Approach	6-10
	6.3.2 Component Design Details	6-12
	6.4 Steam Turbine Prime Mover System Reoptimization for a Motel Application	6-16
	6.5 Technology-Based Potential Market Limitations for the LSM	6-21
	6.6 Alternate Driver Study	6-26
7.0	LARGE SIZE MODEL COMPARATIVE COST OF OWNERSHIP STUDY	7-1
	7.1 Introduction and Overview	7-1
	7.2 Scope	7-3
	7.3 Results	7-6
	7.4 Results - Extension Studies	7-11
	7.4.1 Cooling Load to Heating Load Ratio Sensitivity Study	7-11
	7.4.2 LSM HSPF Gas Heat Pump Heat Exchanger Area Sensitivity Study	7-11
	7.4.3 Comparison of Electric and Gas Steam Generators	7-11
	7.5 Summary of Results	7-17

TABLE OF CONTENTS (Cont'd)

Section		Page
8.0	HSPF GAS HEAT PUMP EQUIPMENT COST STUDIES	8-1
	8.1 Cost Study Methodology	8-1
	8.2 The LSM Cost Study	8-5
	8.3 Summary of Results	8-7
9.0	BUSINESS POTENTIAL ANALYSIS	9-1
	9.1 Requirements of the Business Potential Analysis . . .	9-5
	9.2 Business Potential Methodology	9-6
	9.3 Business Potential Analysis	9-12
	9.4 Business Potential Conclusions	9-25
	ATTACHMENT A - LIST OF SUBCONTRACTORS, CONSULTANTS, AND PRINCIPAL VENDORS	A-1
	ATTACHMENT B - TASK ORGANIZATION AND PROGRAM MILESTONES . .	B-1

APPENDICES (Separate Volumes) . . . . . .

Volume	Title
I	7½-Ton Technical Demonstration
II	7½-Ton Market Potential Study
III	7½-Ton Comparative Cost of Ownership Study
IV	System Cost Analysis
V	Large Size Model Technical Effort
VI	Large Size Model Market Potential Study
VII	Large Size Model Comparative Cost of Ownership Study
VIII	Business Potential Study

LIST OF FIGURES

Number		Page
ES-1	7½-Ton Unit Heating Service Schematic	iv
1-1	3.4 kW_e Steam Turboalternator for Residential-Size Gas Heat Pump	1-3
1-2	Three-Ton R-11 Heat Pump Compressor for Residential-Size Gas Heat Pump	1-4
2-1	Heating Service Schematic	2-4
2-2	Cooling Service Schematic	2-5
2-3	Rotating Unit Schematic	2-7
2-4	Top View of Rooftop Unit Mock-up	2-8
2-5	Air Flow Paths	2-9
2-6	Elevation View of Rooftop Unit Showing Major Equipment	2-10
2-7	7½-Ton HSPF Program Tasks	2-11
2-8	Heat Flow Versus Ambient Temperature	2-15
2-9	System Performance Versus Ambient Temperature	2-16
2-10	Steam-Turbine Power Cycle T-S Diagram for 7½-Ton Unit	2-17
2-11	The SES Steam Generator Developed for the Environmental Protection Agency	2-19
2-12	Steam Turbine Wheel	2-20
2-13	Refrigerant Compressor Wheel	2-22
2-14	Regenerator Core	2-23
2-15	High-Speed Turbomachinery Components	2-24
2-16	Component Test Rig	2-26
2-17	System Tests Being Conducted at MTI's Malta, New York Facility	2-27
3-1	System Performance - Heating Service	3-4
3-2	Simple Payback Time of HSPF for a Retail Store (7½-Tons) HSPF Gas Heat Pump - Maximum Cooling Versus Combination Heat/Cool Unit (YAC)	3-9
3-3	Venture Success Study for the 7½-Ton HSPF Gas Heat Pump Product	3-12
5-1	General Methodology of Market Potential Study	5-2
5-2	Simplified Structure of the Air Conditioning Industry	5-5
5-3	Simplified Structure of the Non-Residential Heating and Air Conditioning Market - 1976	5-6
5-4	Importance of First and Operating Costs by Customer Type - 1976	5-9

LIST OF FIGURES (cont'd)

Number		Page
5-5	Importance of First and Operating Costs by Customer Type-1977	5-10
5-6	Estimated Cost-Orientation Among Commercial Air-Conditioning Customers (1970, 1976 Estimated; 1981 Projected)	5-11
5-7	Market Acceptance of Operating Cost Sensitive Customer to Price Premium and Payback - 1977 Survey	5-12
5-8	Market Potential Among Premium/Payback Customers - Large Tonnage Equipment 1977 Survey	5-13
5-9	Comparison of New Nonresidential Construction and Commercial Unitary and Chiller Shipments, 1960 - 1976	5-15
5-10	Analysis of the Potential Market for the HSPF, 50-400 Tons, 1981	5-17
5-11	Distribution of Central Built-up Equipment by Size, Total U.S. (thousands of tons)	5-20
5-12	Distribution of Commercial Unitary Equipment Shipments by Nominal Size Range, Total U.S. (thousands of tons)	5-21
6-1	LSM Built-up System Concept Schematic	6-2
6-2	Large Size Model 125-ton System Schematic	6-4
6-3	Illustration of Power Availability and Power Requirements for the Proposed LSM Product Design	6-5
6-4	Economic/Marketability Constraints on the HSPF Heat Pump System along with the Distribution of Annual Energy Costs for HVAC Related Needs Using Alternative Systems in an Office Building Application	6-8
6-5	Cross Section of Turbomachinery Unit	6-13
6-6	Steam-Turbine Power Cycle T-S Diagram for LSM Unit	6-14
6-7	Cross Section of Three-Stage Turbine	6-15
6-8	Compressor Module No. 3 Layout	6-18
6-9	Thermal Efficiency Comparison of Alternate 90 hp Drivers	6-27
6-10	Recoverable Waste Heat Comparison of Alternate Drivers for LSM Gas Heat Pumps	6-28
7-1	LSM HSPF and Best Conventional Alternative System Comparison for Motel Application	7-7
7-2	LSM HSPF and Best Conventional Alternative System Comparison for Office Application	7-10
7-3	Sensitivity to Change of Cool/Heat Ratio in Motel Application. LSM HSPF Gas Heat Pump (A·6) Versus Electric-Motor Driven Reciprocating Compressor and Gas Steam Generator (A·1)	7-13

LIST OF FIGURES (cont'd)

Number		Page
7-4	Sensitivity to Change of Heat Exchanger Area in Motel Application. LSM Gas Heat Pump (A·6) (A·6-1) (A·6-2) Versus Electric-Motor-Driven Reciprocating Compressor and Gas Steam Generator (A·1).	7-15
8-1	Product Costing Plan - Preliminary Estimate	8-2
8-2	Product Costing Plan - Interim Iteration.	8-3
8-3	Product Costing Plan - Final Iteration.	8-4
9-1	New Product Planning and Development Process.	9-3
9-2	Business Analysis Task Flow and Relationships	9-4
9-3	Schematic Flow of Simplified Model of the Firm.	9-8
9-4	Illustration of the Effect of a 1990 Valuation of the Future Income Upon DCF	9-11
9-5	Cumulative Cash Flow and DCF for Company A with Baseline Business Scenario, Typical for all Companies	9-14
9-6	Impact of Reduced Fixed Investment on Cumulative Cash Flow and DCF for Company A with Baseline Business Scenario. .	9-17
9-7	Impact of Improved Profit Margins on Cumulative Cash Flow and DCF for Company A with Baseline Business Scenario. .	9-18
9-8	Impact of Combined Changes on Cumulative Cash Flow and DCF for Company A with Baseline Business Scenario.	9-19
9-9	Variation of 1990 Cumulative Cash Flow and DCF with Sales Growth Rate.	9-23
9-10	Impact of Variations of Growth Scenarios on Cumulative Cash Flow and DCF without Valuation in 1990.	9-24

LIST OF TABLES

Number		Page
ES-1	Summary of Technical Characteristics of the 7½-Ton and LSM HSPF Gas Heat Pump Product Concepts Evaluated in the Program.	ix
1-1	Members of the HSPF Gas Heat Pump Program Advisory Board.	1-6
1-2	7½-Ton Program Task Organization.	1-9
2-1	Major System Components	2-13
2-2	7½-Ton HSPF System Performance.	2-28
3-1	Seasonal Performance Factors for a Retail Store Based Upon Energy Consumed at the Site.	3-7
3-2	Unitized Building Total Utility Costs for a Retail Store.	3-8
3-3	Interrelationships Between 7½-Ton Venture Success Study Elements	3-13
5-1	Assumptions - Key Parameters Air Conditioning Market Potential for the HSPF 50-400 Tons	5-18
5-2	Projected National Market for 50 to 400 Ton Equipment in 1981	5-22
5-3	Projected North Central Region Market for 7½- to 50-Ton Unitary Equipment in 1981.	5-23
6-1	Turbine Stage Data.	6-17
6-2	Compressor Design Parameters.	6-19
6-3	Effect of Process Load on Steam-Turbine Power Cycle Performance	6-20
6-4	Economy Motel Application Energy Cost and Consumption	6-22
6-5	Economy Motel Application - Steam Turbine Power System Optimization	6-23
7-1	Evaluation Parameters, LSM HSPF Gas Heat Pump Comparative Cost of Ownership Study.	7-2
7-2	Case Identification Numbers	7-4
7-3	Summary of Economic Comparisons	7-5
7-4	Summary of Results/Motel.	7-8
7-5	Summary of Results/Office Building.	7-9
7-6	Extension Study Summary of Results for Motel - Sensitivity to Change of Cool/Heat Ratio	7-12
7-7	Extension Study Summary of Results for Motel - Sensitivity to Change of LSM HSPF GHP Heat Exchanger Area.	7-14
7-8	Extension Study Summary of Results for Office - Revised Reference System	7-16

LIST OF TABLES (cont'd)

Number		Page
8-1	Preliminary 7½-Ton System Manufacturing Cost Basic Assumptions	8-8
8-2	7½-Ton Manufacturing "Final" Cost - Alternate Assumptions	8-10
8-3	Interim Product Costing for LSM Turbochiller Package	8-11
8-4	Product Selling Price Comparison	8-12
9-1	Company Financial Data used for Modelling	9-10
9-2	Business Potential Analysis Paths to Commercialization Considered in Base Analysis	9-13
9-3	Summary of Parameter Variation for Sensitivity Analysis for Baseline Business Scenario	9-16
9-4	Comparison of Financial Ratios Used for Hypothical Firm	9-21
9-5	Sale Growth Rate Scenarios Considered	9-22
9-6	Business Potential Conclusions	9-26
9-7	Factors Needing Change	9-28
A-1	HSPF Gas Heat Pump Program Subcontractors	A-2
A-2	HSPF Gas Heat Pump Program Consultants	A-3
A-3	HSPF Gas Heat Pump Program Principal Vendors	A-5
B-1	Large Size Model Program Task Organization	B-2
B-2	Large Size Model Principal Program Milestones	B-3

IDENTIFICATION OF TERMS AND CONVENTIONS

A number of terms and conventions are employed through the balance of this report. Those that are not in common use or are special with respect to this report are listed and defined below.

Term	Definition
CNG	Consolidated Natural Gas Service Company, Inc.
DOE	The U.S. Department of Energy, Conservation and Solar Applications Wing, Office of Buildings and Community Systems.
ERDA	U.S. Energy Research and Development Administration Conservation Wing, Buildings and Industry Division.
HSPF System	HSPF refers to High Seasonal Performance Factor. HSPF System refers to the special system features embodied in the 7½-ton HSPF gas heat pump demonstrator.
LSM	Proposed Large Size Model of the HSPF gas heat pump; the object of the LSM program.
LSM Program	That portion of the HSPF gas heat pump program containing the tasks associated with the LSM; Tasks VI through VIII.
MTI	Mechanical Technology Incorporated
7½-Ton Breadboard Demonstration	The 7½-ton demonstration unit produced and tested under the 7½-ton program.
7½-Ton Program	That portion of the HSPF gas heat pump program containing the tasks associated with the 7½-ton unit; Tasks I through V.
7½-Ton Unit	The proposed product embodiment of the 7½-ton HSPF gas heat pump; the object of the 7½-ton program.
HVAC	Heating, Ventilating and Air Conditioning.

1.0 INTRODUCTION

The Consolidated Natural Gas Company (CNG) includes a number of distribution companies which retail gas energy in much of Ohio, Pennsylvania and West Virginia. CNG also wholesales most of the gas energy utilized in upper New York State. Because of climate and population density, these areas have an inherently heavy space heating load. The company has penetrated 82% of the space heating market within its service areas. As a result of the company's interest in and commitment to the space heating market, requirements for a gas heat pump for CNG's service area were first developed in 1965. These requirements were:

1. All rotating machinery. Only rotating machinery had, by 1965, proven itself able to operate for the required tens of thousands of hours without replacement, repair or maintenance.
2. Minimal maintenance. This requirement was known to apply especially to the light commercial and residential equipment markets where scheduled maintenance needs are frequently ignored.
3. Sealed, closed cycle system employing a nontoxic working fluid.
4. Good heating season performance in the North-Central region.
5. Competitive first cost.
6. Quiet, reliable operation. Owners of gas heating equipment have come to expect trouble-free, innocuous operation from their equipment for decades.

A technical development program was initiated in 1965 for a system that appeared to satisfy the above requirements. This program was directed toward a residential steam turbine-driven gas heat pump demonstration. The program was conducted between 1965 and 1969, under the sponsorship of Lone Star Gas, Northern Illinois Gas, Southern California Gas, Texas Eastern Gas Transmission Co., and the Washington Gas Light Company, in addition to CNG.

This system was based upon a very simple regenerated steam turbine operating in a moderate-pressure, high-temperature steam cycle, driving a reverse

cycle heat pump. The system was designed to serve a single residence. Because of the relatively crude state-of-the-art of shaft seals in 1965, first magnetic, and then electric alternator-motor driven couplings were adapted for transmitting power from the turbine to the compressor and the air-moving components. The system, in its final embodiment in 1969, incorporated the following features:

1. All rotating machinery with process lubricated bearings.
2. Hermetically sealed subsystems.
3. Self-propelled auxiliary equipment; 100% gas energized.
4. Modulated output in response to demand.
5. Phased heating of the conditioned air through indoor heat exchangers arranged in series.
6. Direct injection of steam for supplemental heat in very cold weather.

These features represent most of the elements that were later to define the High Seasonal Performance Factor (HSPF) Gas Heat Pump System. Figures 1-1 and 1-2 show the turboalternator and compressor wheel, respectively, that were developed under the residential program.

The residential heat pump development program succeeded in demonstrating the energy conservation effectiveness of the cited system features, especially the principal of two-step heating of conditioned air via heat exchangers arranged in series. This program also revealed the great difficulty in achieving an acceptably low cost for a single residential-size steam turbine-driven gas heat pump. A conclusion was reached that this technology could probably never be competitive for residential-size systems.

During 1973-74, CNG sponsored a related project, aimed at the development of a superior-efficiency steam turbine for use as a driver for larger-size gas heat pumps. This program was based upon a radial outflow steam turbine development initiated under the Environmental Protection Agency's steam car program. The radial-flow turbine project provided indications that a steam turbine with superior efficiency might be achievable. However, analysis of

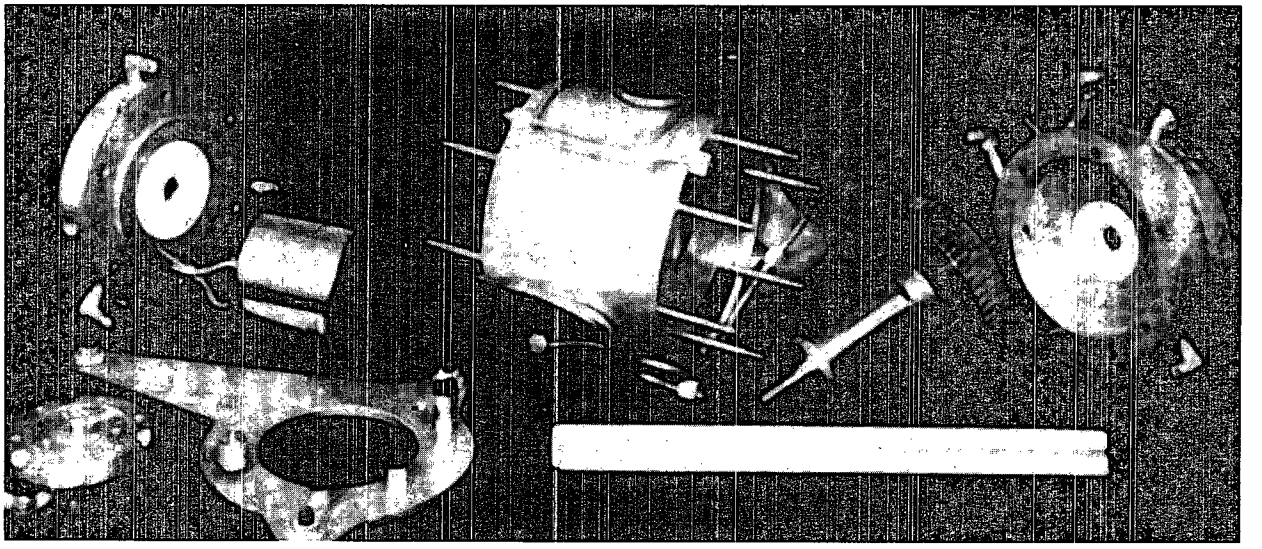

Fig. 1-1 3.4 kWe Steam Turboalternator for Residential-Size Gas Heat Pump

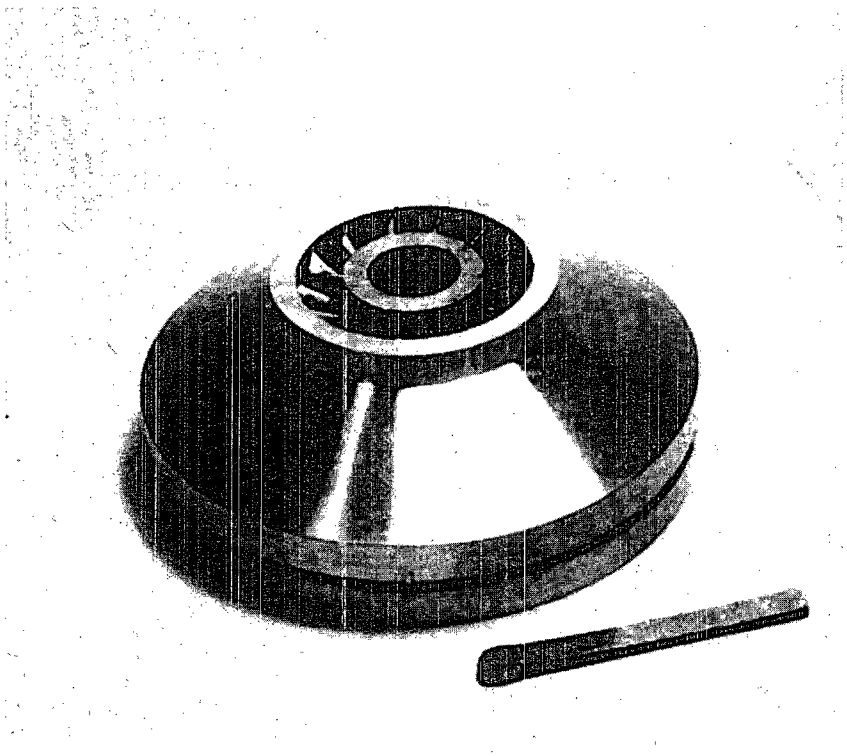

Fig. 1-2 Three-Ton R-11 Heat Pump Compressor for Residential-Size Gas Heat Pump

the program neither confirmed nor denied the feasibility of such a turbine. CNG terminated this program in January, 1976, in order to direct available resources to the development and demonstration of a complete heat pump system.

Late in 1975, an important additional system feature was introduced; outdoor heat exchangers arranged in series. When the outdoor exchangers were combined with the previously used indoor heat exchangers arranged in series, phased heat exchange both indoors and outdoors during both cooling and heating service was provided. Important Second Law of Thermodynamics advantages of these features are reveiwed in Section 2.0. This last feature completes the list of principal features that defined the HSPF Gas Heat Pump technology package.

By late 1975, it had been established that the HSPF System offered excellent potential for energy conservation but would not be competitive in the marketplace as a product sized for single dwellings. CNG approached Mechanical Technolgy Incorporated (MTI), a firm that had become well established as a leading developer of turbomachinery systems, for assistance in pursuing the development of a commercial-size HSPF system. MTI agreed to assume responsibility for system integration as well as for a major portion of component development.

Subsequently, in response to a CNG-MTI proposal, the U.S. Energy Research and Development Administration (ERDA) agreed to provide cosponsorship (a 50% cost share) for the commercial-size system development program. MTI was proposed as the principal subcontractor. The CNG/ERDA/MTI HSPF commercial-size gas heat pump program was entitled Demonstration of a High Seasonal Performance Factor Gas Heat Pump for the North-Central United States and was formally initiated on February 1, 1976. With ERDA encouragement, a Program Advisory Board was established to review program objectives and progress on a regular basis and to provide recommendations for each major program decision. The Board was composed of executives from MTI, Masco Corporation, Carrier Corporation, CNG and four other gas utilities. With two exceptions, each Board member was responsible for research management in his parent organization. Table 1-1 comprises a list of Program Advisory Board members.

TABLE 1-1

MEMBERS OF THE HSPF GAS HEAT PUMP
PROGRAM ADVISORY BOARD

Name	Affiliation
R.C. Weast, Chairman	Vice President, Research Consolidated Natural Gas Service Co.
S.J. Cunningham	Manager of Research Southern California Gas Company
C.V. Fenn	President (retired) Carrier Corporation
C. Janjikian	Group Vice President Masco Corporation
J.R. Kelso	Vice President, Marketing (Appointed 1978) Consolidated Natural Gas Service Co.
R.L. Leusch	Vice President, Marketing (Retired 1978) Consolidated Natural Gas Service Co.
G. Long	Director of Research & Development Northern Illinois Gas Company
E.C. Moran	Director of Research Washington Gas Light Company
D. Sandell	Vice President, Energy Systems Carrier Corporation
B. Sternlicht, Vice Chairman	Chairman of the Board & Technical Directory, Mechanical Technology Inc.
L.H. Sutherland	Vice President, Development & Research Lone Star Gas Company

The prime contract between ERDA and CNG was executed in May, 1976. ERDA subsequently became a part of the U.S. Department of Energy, under whose sponsorship the program continued. During contract negotiations, an agreement was reached between ERDA and CNG that provided for a 6-year waiver of the patent rights normally reserved for the Government, in favor of Consolidated. CNG subsequently reached an agreement with MTI, providing for MTI a right of first refusal for an exclusive license under Consolidated's waiver. The Government agreed to extend Consolidated's waiver in the event CNG and MTI achieved product commercialization within the 6-year period. In recognition of the favored position granted by the right of first refusal, MTI agreed to provide a supportive business development effort.

1.1 7½-Ton Program Overview

The HSPF commercial-size program quickly focused on the development of a 7½-ton unitary rooftop air-to-air embodiment of the HSPF system. This effort is the subject of the next three sections of this report. For the purpose of providing an overview, the major 7½-ton program tasks will be reviewed here briefly.

The first task was begun on January 1, 1976, which was prior to the execution of the contract with ERDA. The purpose of this task was to select the HSPF system configuration and the system capacity that were to be incorporated in the first program technical demonstration. The 7½-ton unitary rooftop air-to-air embodiment was selected because 7½ tons represented the largest unit market volume portion of the commercial HVAC equipment market and because the single-zone package system configuration would preserve and fully utilize the HSPF features demonstrated previously. Also, 7½ tons was expected to represent the smallest practical capacity for the HSPF turbomachinery elements and would, thereby, provide a limiting case demonstration of turbomachinery performance. The selection process was conducted over a one-month period without the aid of a market survey, a product cost assessment, or an owning and operating cost analysis.

The remainder of the program was initiated on February 1, 1976 and was organized at ERDA's request into five tasks. Each task was subdivided

into several phases. The task comprising the greatest effort, Task IV, corresponded to the technical demonstration. Tasks II and III corresponded to the 7½-Ton Unit Comparative Cost of Ownership and the Market Potential studies, respectively. Tasks I and V represented administrative functions. Each of these tasks are described on Table 1-2. Also shown on Table 1-2 is the overall period of performance for each task, and a reference to the report section containing a full description of each task.

The technical demonstration effort was conducted largely by MTI. This effort also included an equipment cost estimate for the 7½-ton unit. All major technical decisions were reviewed by both MTI and CNG.

A number of demonstration effort components and consulting services were contracted separately by CNG. A comprehensive listing of all subcontractors, consultants and principal vendors comprises Exhibit A to this volume. The 7½-Ton Unit Comparative Cost of Ownership Study was performed by CNG's Service Company Marketing and Computer Services Departments. The equipment performance data needed for this study were provided by MTI. Consultants were employed to gather utility rate data and appropriate weather data. The Market Potential Study was performed under the CNG Service Company Marketing Department's direction by William E. Hill and Company.

The 7½-ton program tasks were developed concurrently with no iterations. The results from each task were reviewed periodically by the Program Advisory Board. Three Board meetings were held throughout the 7½-ton program: two at intermediate points in the program when interim results were reviewed and the third at the point when the final results from the Equipment Cost, Market Potential and Comparative Cost of Ownership Studies were presented. The recommendations developed at the third meeting led to the termination of the 7½-ton program. The 7½-ton program termination and the program redirection to larger systems is the subject of Section 4.0 of this report.

TABLE 1-2

7½-TON PROGRAM TASK ORGANIZATION

ERDA Assigned Program Task Number	Report Section Containing Full Task Description	Task Description	Period for Performance
I	1.0	Program Management, 7½-Ton Program	2-76/7-78
II	3.0	7½-Ton Comparative Cost of Ownership Study	2-77/7-77
III	7.0	7½-Ton Market Potential Study	9-76/4-77
IV	2.0	7½-Ton Technical Demonstration	2-76/7-78
	8.0	7½-Ton Product Cost Study	12-76/7-77
V	2.0	7½-Ton Demonstrator Delivery	12-79

2.0 7½-TON TECHNICAL DEMONSTRATION

The HSPF gas heat pump system configuration and capacity selected for initial demonstration was a unitary air-to-air rooftop system rated at 7½ tons cooling capacity.

The goals for technical performance were defined in terms of a seasonal performance factor of 1.3 for heating service and 0.6 for cooling service for an application in Cleveland, Ohio. Such levels of performance would have provided superior energy conservation capability compared to contemporary HVAC concepts. These overall goals provided the basis for subsequent performance criteria established for each system and component.

The principal objective of the 7½-ton program technical effort was to demonstrate the feasibility of achieving the performance goals. Specific technical effort tasks were to design, fabricate and test the critical components and to assemble a laboratory breadboard model of the HSPF system concept. The breadboard model was to incorporate all the design features of the HSPF system concept but not necessarily in a product configuration. All the system components would be identical or functionally equivalent to the product concept; however, their physical location to each other would not be typical of the product configuration.

2.1 HSPF 7½-Ton System Description

The HSPF system concept was defined in terms of a number of system features as applied to a gas-fueled, steam turbine-driven, unitary air-to-air heat pump. The 7½-ton technical effort addressed a number of technically advanced components including a compact high-efficiency steam generator and a single-shaft, gas bearing turbomachine incorporating the steam turbine, alternator and refrigerant compressor. In addition to these advanced components, the HSPF system concept included the following novel operational and design features:

1) Phased cooling of system condensers. During the cooling operation, heat being rejected to the surroundings from the heat pump (air conditioning) circuit was given first access to cooling air. The outdoor air first accepted rejected heat from the refrigeration circuit and second, from the steam turbine's exhaust.

2) Phased heating of the conditioned air. First, the pumped heat was transferred to the return conditioned air, and second, the steam turbine's exhaust was used to heat the air.

   Phased cooling of system condensers (Item 1) and phased heating of conditioned air significantly reduced the head pressure requirement of the heat pump compressor. Second Law of Thermodynamics considerations showed that reducing the work per unit heat being pumped was a favorable trade-off for the resulting elevation in the turbine's back pressure.

3) Direct heating of conditioned air with steam, bypassing the turbine at very low temperatures. The bypassed steam augmented the turbine's exhaust, both of which then heated the conditioned air after it was heated by pumped heat. By this method, useful heat pumping could coexist with heating by means of bypassed steam. This technique, coupled with the technique of item 2 above, led to useful heat pumping, down to about 7°F with single-stage centrifugal compression.

4) Fully modulated steam cycle operation. Full utilization of this technique could only be accomplished when both the heat pump compressor and the air-moving fans were speed-coupled to the prime mover. An advantage inherent in this approach was that at low capacity operation, air-moving power requirements did not destroy the basic energy-saving advantages

of modulation. Control inputs were to the burner and to the turbine throttle.

5) Self-powered auxiliaries. The use of the steam-turbine prime mover to provide electricity for the auxiliaries such as the air handling fans and the feed pump removed the need for utility electric service for these functions. This use of the high-quality energy in the steam to first provide electricity for the auxiliaries and then for heating service was an example of energy cascading and improved the overall fuel efficiency of the system.

Figures 2-1 and 2-2 are schematics of the HSPF heat pump system operating in the heating and cooling modes, respectively. These schematics illustrate the phased heating of air in both the indoor and outdoor heat exchanger assemblies, the direct use of steam for supplemental heating and the self-powering of the auxiliaries. Phased heating indoors and phased cooling outdoors required the four-heat-exchanger system configuration shown in Figures 2-1 and 2-2. This configuration is the "trademark" of the HSPF system.

The operation of the steam-turbine prime mover system or driver can be best described as follows. Driver operation began at the feed pump and the steam generator. The feed pump supplied condensate water to the steam generator at a pressure somewhat above 300 psia. The steam generator then produced 300 psia, 1100°F steam. This steam passed through a steam turbine to produce shaft power. The steam system also provided for reheat and regeneration. However, these details are not necessary to understand the basic principles of operation of the HSPF system. The steam exiting from the steam turbine was condensed in either the outdoor steam heat exchanger (for cooling) or the indoor steam heat exchanger (for heating). The power from the steam turbine drove an alternator and a refrigerant compressor. The alternator provided power for the auxiliaries. The refrigerant compressor compressed the heat pump refrigerant in a standard heat pump refrigeration circuit.

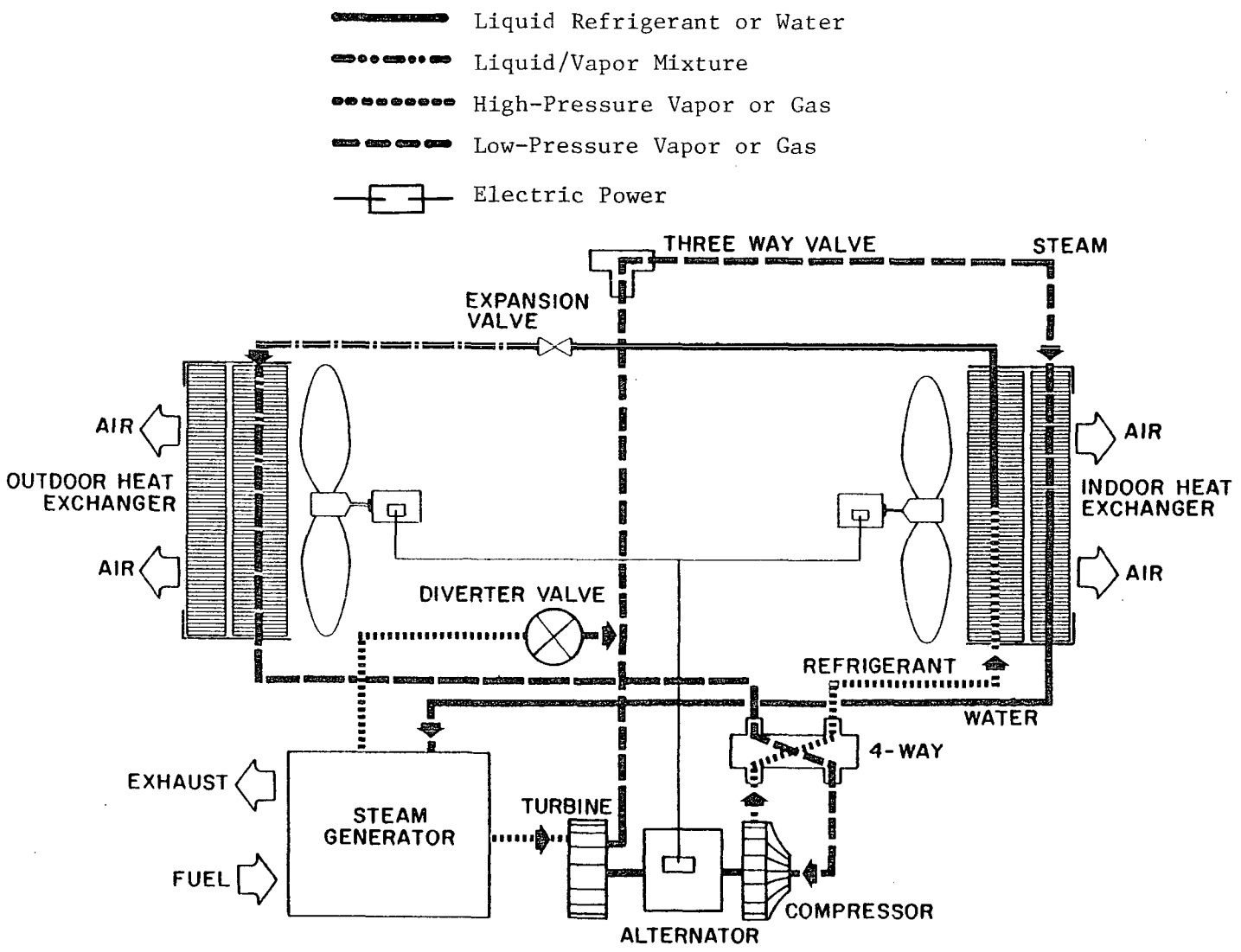

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Fig. 2-1 Heating Service Schematic

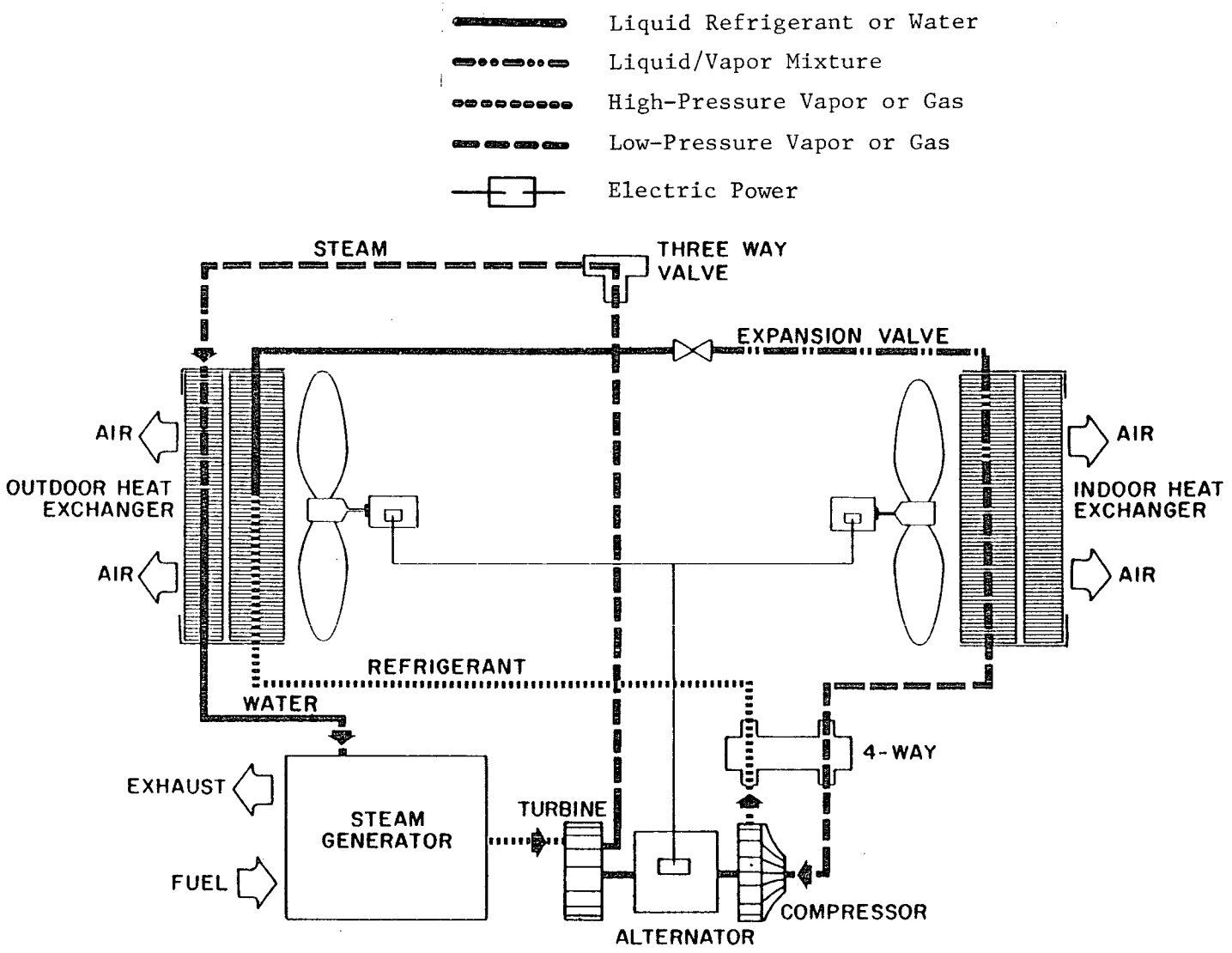

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Fig. 2-2 Cooling Service Schematic

The rotating machinery package is depicted in Figure 2-3 in simplified form. The turbomachinery package design consisted of the axial-flow steam turbine, the alternator rotor, and the refrigerant compressor, all of which were mounted on a single shaft. The shaft was supported by refrigerant-lubricated gas bearings. The shaft seal between the steam and refrigerant systems was at the steam turbine. Thus, the alternator and bearings operated in an environment of superheated refrigerant. The system was designed for use with refrigerant 11.

The major auxiliary devices were driven by electric motors which were powered by the high-frequency alternator. The breadboard demonstrator employed a rectifier and direct-current drive motors. The product concept, however, favored synchronous high-frequency ac transmission and drive motors. Either way, the major auxiliary devices were speed-locked to the prime mover.

A mock-up of the HSPF in a rooftop product configuration was fabricated to illustrate the features noted above and the packaging of the novel components. This mock-up is pictured in Figure 2-4. Figure 2-5 shows the air path through the refrigerant and steam heat exchangers and illustrates the novel combination of phased heat exchangers noted above. Figure 2-6 illustrates the general arrangement of the mechanical components. These components represent the principal difference between the HSPF heat pump and conventional units using electric motor-driven compressors.

The 7½-ton HSPF technical effort program plan as implemented is shown in Figure 2-7. The program effort was terminated at the end of proof-of-concept testing of the 7½-ton breadboard demonstrator in July, 1978.

2.2 Size Selection

The selection of an appropriate size for the initial HSPF unit involved considerations of several interrelated factors, including energy conservation potential, total potential market size, technical performance objectives and system cost goals. The total energy conservation potential was a function of both total potential market size and of technical performance. The actual energy conservation realized per unit was a function of the system cost and

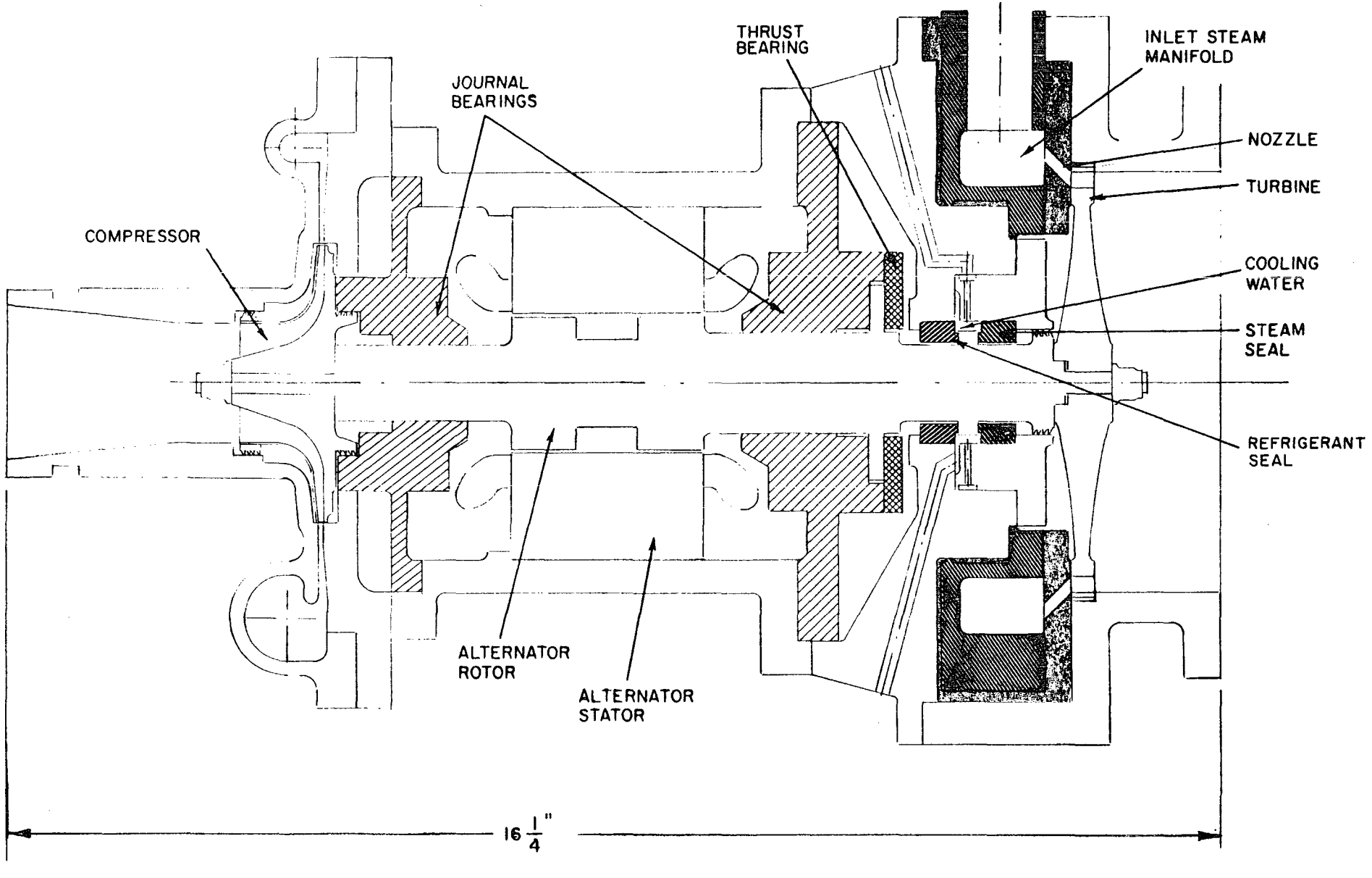

Fig. 2-3 Rotating Unit Schematic

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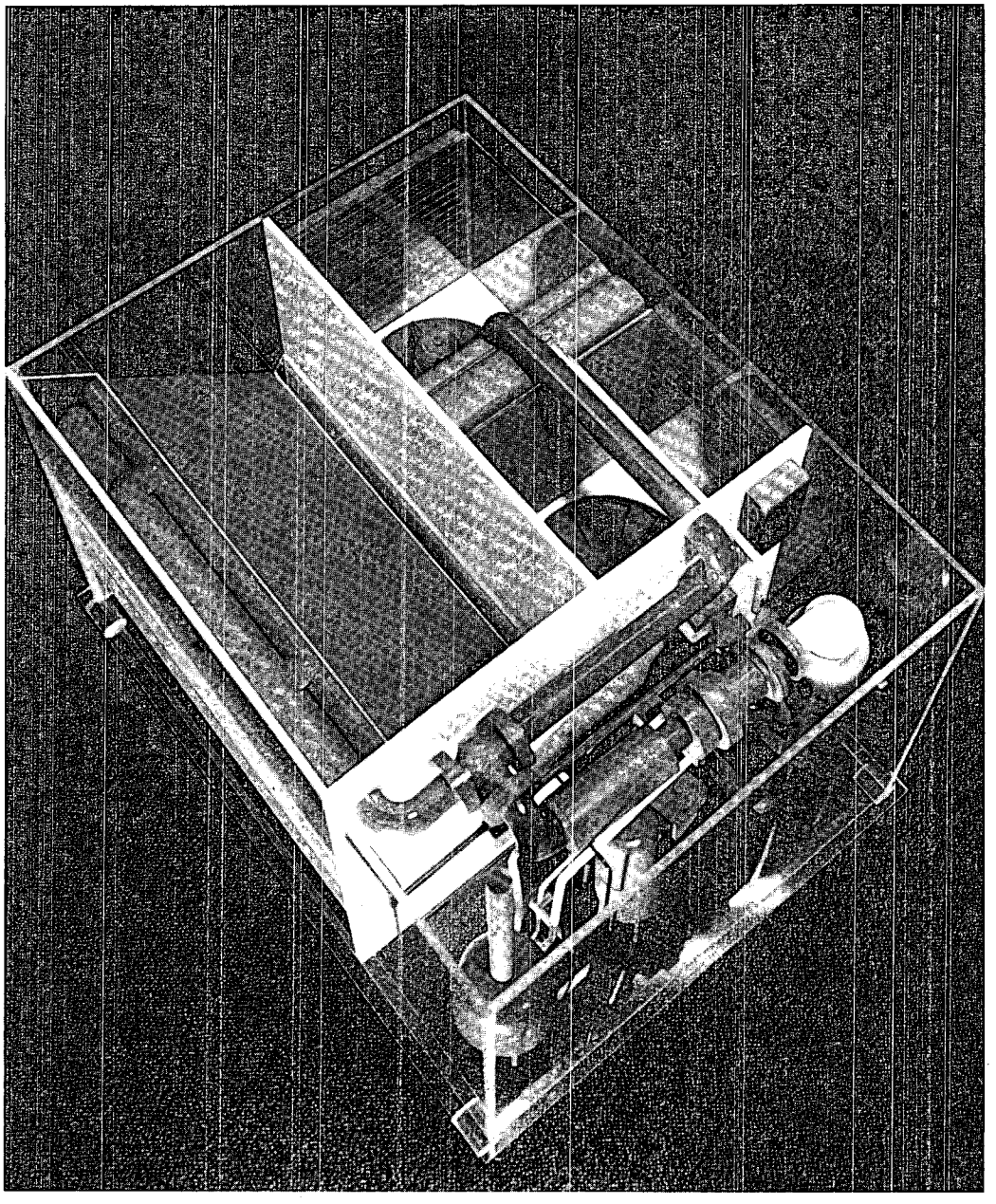
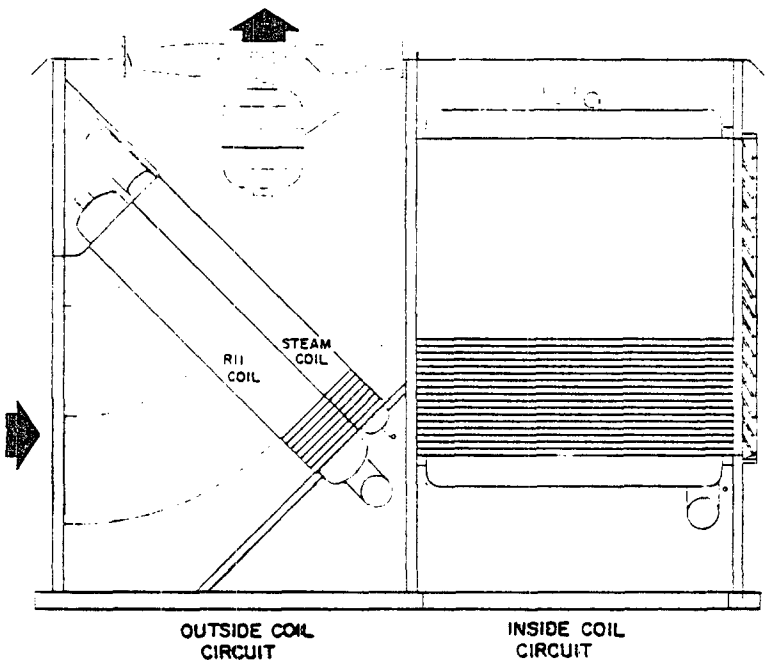


Fig. 2-4 Top View of Rooftop Unit Mock-up



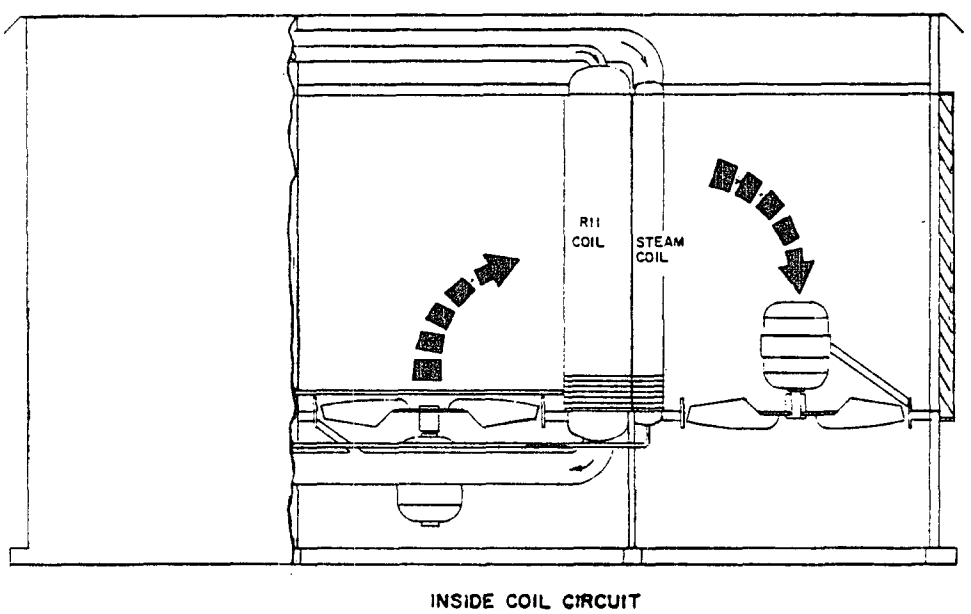


Fig. 2-5 Air Flow Paths

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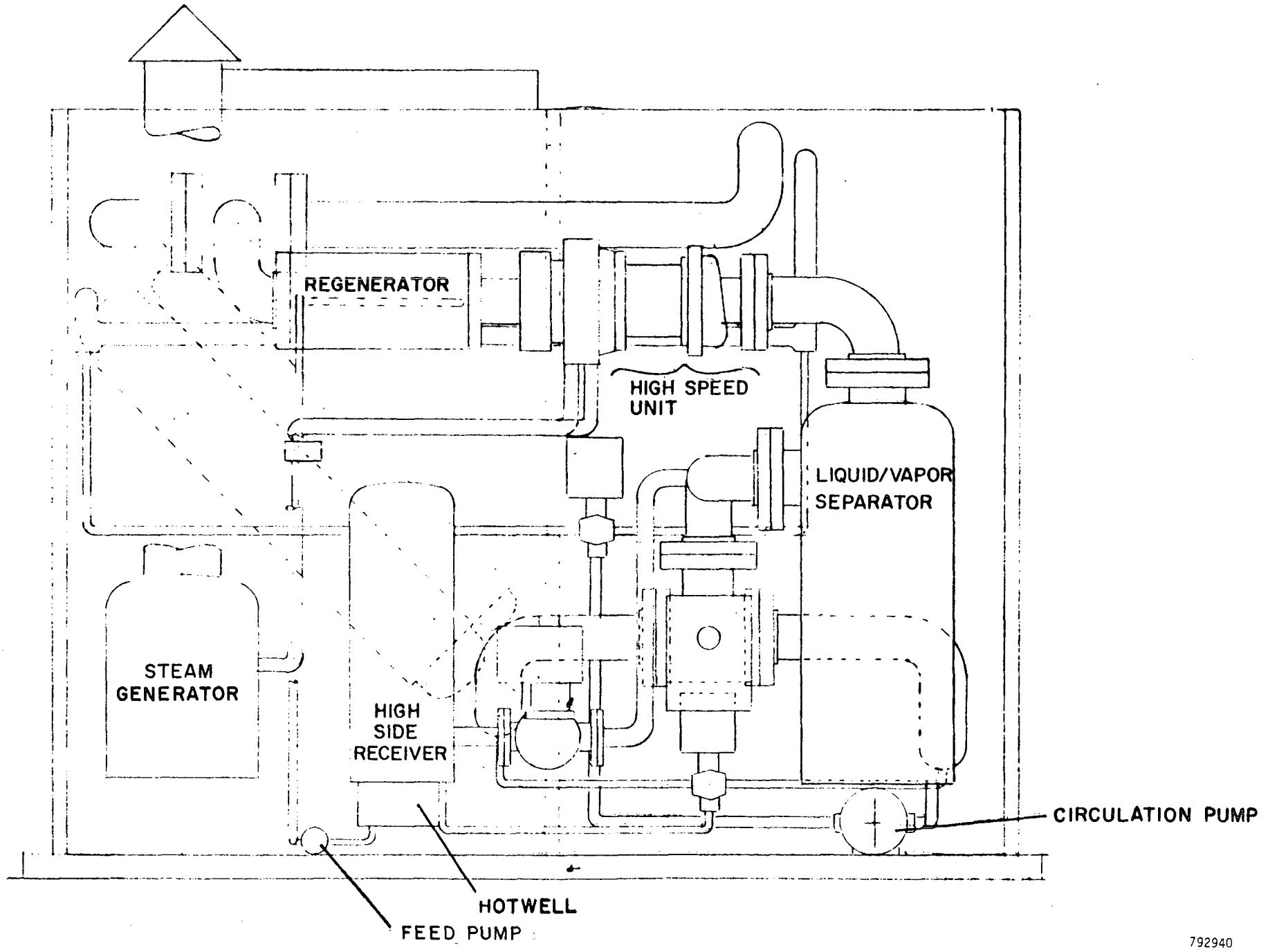

Fig. 2-6 Elevation View of Rooftop Unit Showing Major Equipment

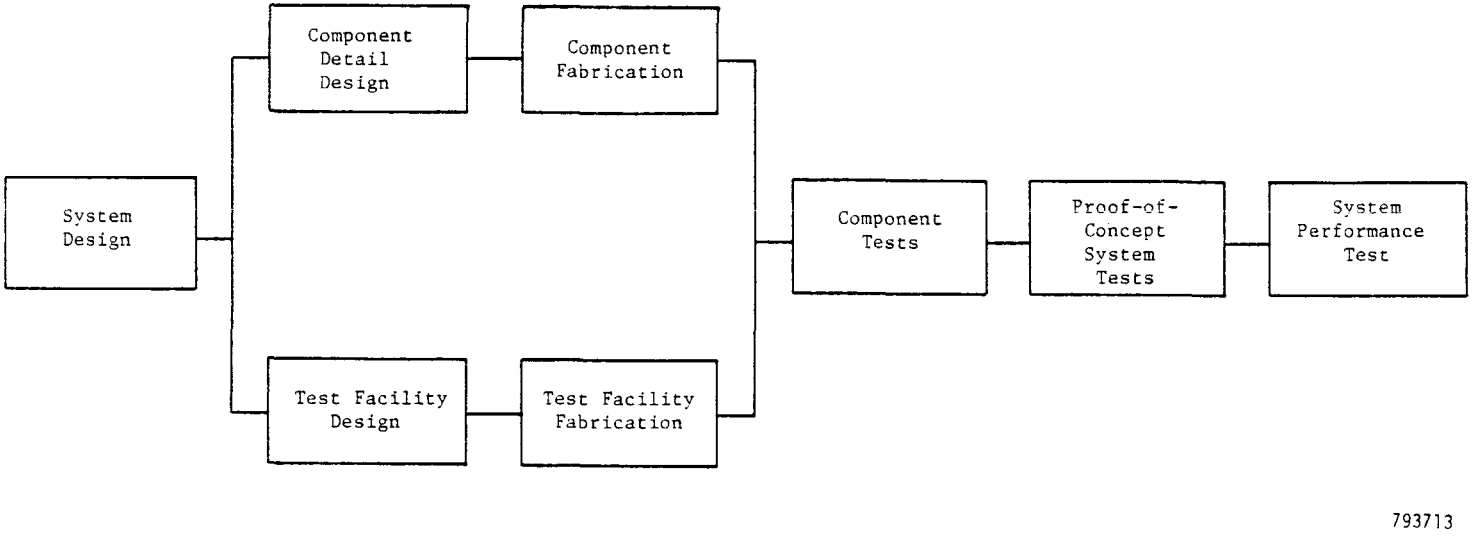

Fig. 2-7 7-½-Ton HSPF Program Tasks

performance. The selection of size was determined primarily by qualitative market potential considerations.

At the time the initial HSPF unit size was selected, specific information was not available regarding the cost of the system or the market reaction to system cost.

A design goal of 7½ tons nameplate cooling capacity was established because this was the size with the largest unit sales volume in the commercial equipment market. The heat capacity of 120,000 Btu/hr at 31°F resulted from the characteristics of the system, as sized for the 7½-ton cooling nameplate rating. For this unit size, the physical size of the turbomachinery was near the lower limit for practical turbomachinery with acceptable performance. Therefore, the 7½-ton unit size for the breadboard demonstrator would demonstrate HSPF concept performance within the context of a large potential market coincident with technology limits.

2.3 System Design

The overall system design goal was to achieve design point heating and cooling loads with optimized system performance. Optimization of system performance required matching the various components which, in turn, necessitated trade-off studies. Table 2-1 lists the major components in the system. The difference between the components used in the breadboard demonstrator and those expected to be used in the final product are noted.

Of the components listed in Table 2-1, MTI was responsible for the design of those components in the high-speed rotating package which included the turbine, alternator, seal, bearings and compressor. The steam generator was designed by Scientific Energy Systems, Watertown, Massachusetts, under a subcontract to CNG. The heat exchangers and drive motors were commercially available equipment. MTI was responsible for specifying and purchasing these components. The feed pump was designed and fabricated by Phillips Engineering Company, St. Joseph, Michigan.

The design of the turbomachinery components required especially careful matching among the components in the high-speed rotating package. The

TABLE 2-1

MAJOR SYSTEM COMPONENTS

Steam Generator	Monotube, power burner, with integrated superheater, reheater and combustion air preheater.
Turbine	Single stage, axial flow with reentry.
Alternator	3-Phase inductor homopolar.
Bearings	Refrigerant-gas-lubricated.
Breadboard	Pivoted shoe.
Product	Hydresil® foil type.
Shaft Seal	
Breadboard	Combination floating labyrinth and carbon-ring face seal.
Product	Hermetic.
Compressor	Single-stage centrifugal, fixed geometry.
Air/Steam and Air/Refrigerant Heat Exchangers	Flat tube and plate-fin, all parallel circuits. Series air-flow
Feed Pump	Hermetic diaphragm-type with gear-type booster.
Drive Motors	
Breadboard	dc
Product	ac squirrel-cage.

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design effort culminated with the optimization of overall system performance. The design process, as a whole, required numerous iterations, starting with a basic design for each component and focusing finally on the overall system output/performance goal. A computer simulation was used to generate a family of operating characteristics as a function of outdoor ambient temperature. Based on fixed heat exchanger surface areas, compressor, turbine and alternator maps and air flows specified for the refrigerant condenser and evaporator, the computer simulation executed the necessary component matches and calculated system performance. A large number of simulation runs were carried out to aid in specifying the wide range component performance required (in particular, the compressor/turbine combination) and to generate system performance data for the cost-of-ownership studies.

Figures 2-8 and 2-9 show representative summary data generated by the simulation. Figure 2-8 shows the heating capacity of the system as supplied by the combination of the refrigerant system (as a heat pump) and the steam condenser. At the 35°F design point, pumped heat and steam turbine power cycle rejected heat are about equal in magnitude. The upper portion of the figure shows the turbine speed established by the controller as a function of ambient temperature to satisfy a surge avoidance criterion. Figure 2-9 presents the overall heating COP for the system (ratio of heat delivered to the load to heating value of gas consumed). Integrating that performance against weather profiles typical of Pittsburgh or Cleveland, a seasonal performance factor of about 1.21 was predicted for the breadboard unit.

2.4 Component Design

2.4.1 Steam-Turbine Cycle

An idealized representation of the steam-turbine power cycle selected is shown in Figure 2-10. This cycle provided for reheat and regeneration to maximize thermodynamic efficiency. The state points for the first stage expansion were a maximum of 300 psia and 1100°F. After the first expansion, the steam was reheated to 1100°F at state point 3. The pressure ratio across both stages of expansion was 10 to 1. The steam exiting the second stage of expansion had considerable superheat. This superheated steam was cooled in a regenerator from state points 4 to 5 by heating the feedwater from state points 8

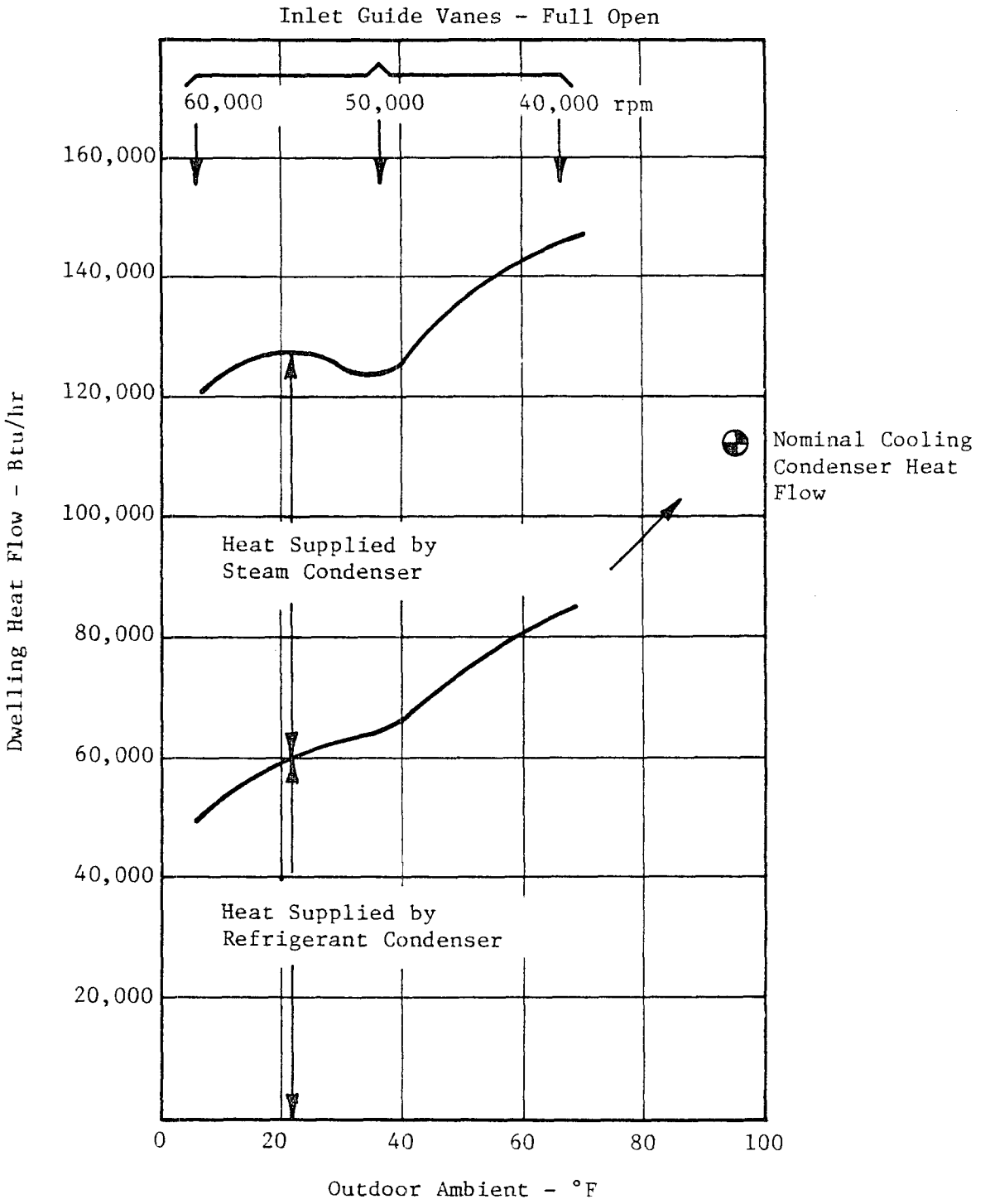

Fig. 2-8 Heat Flow Versus Ambient Temperature

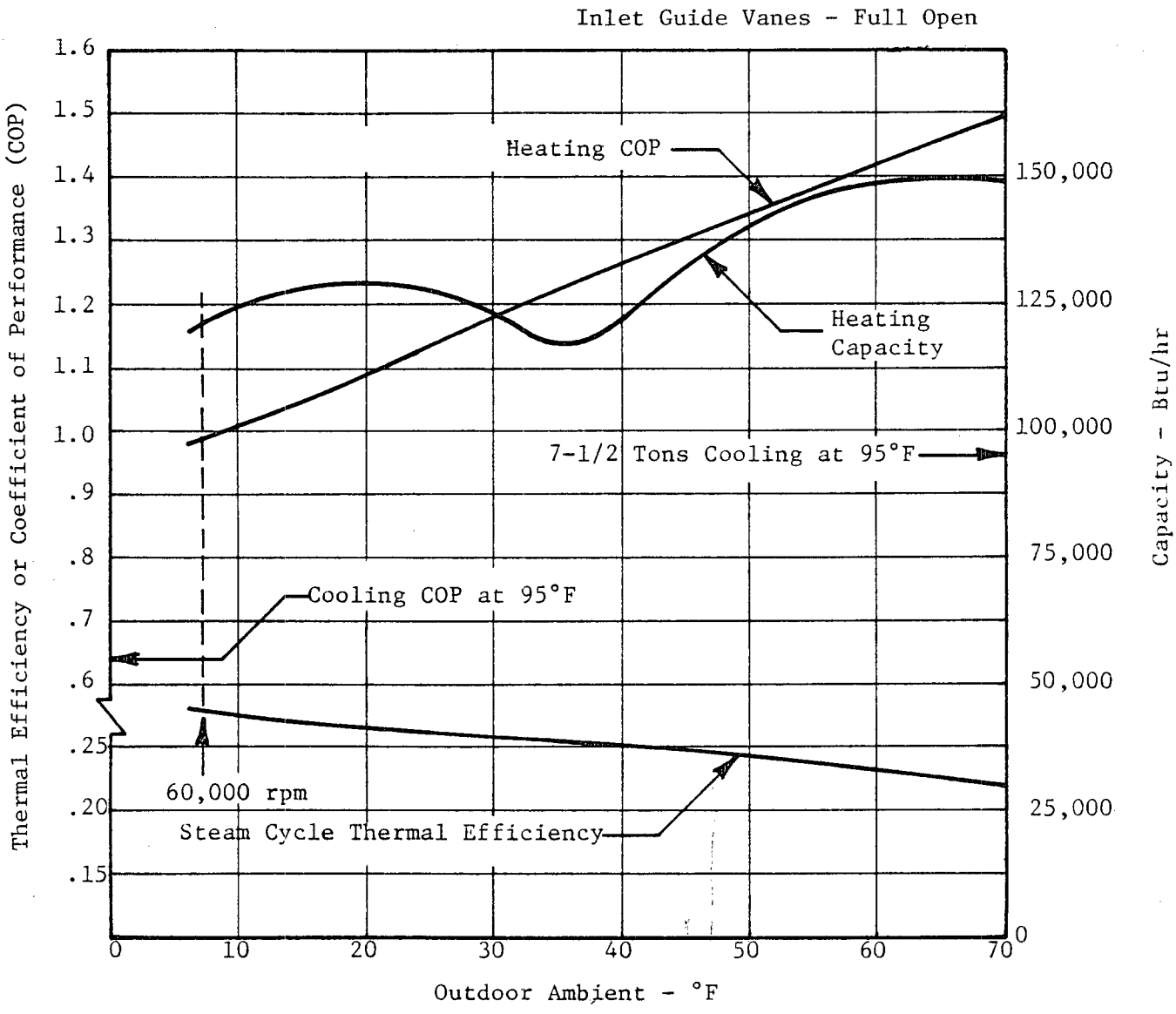

Fig. 2-9 System Performance versus Ambient Temperature

State Point 1 - 300 psia, 1100°F
State Point 3 - 30 psia, 1100°F
4 - 3 psia, 635°F
6 - 3 psia, 150°F

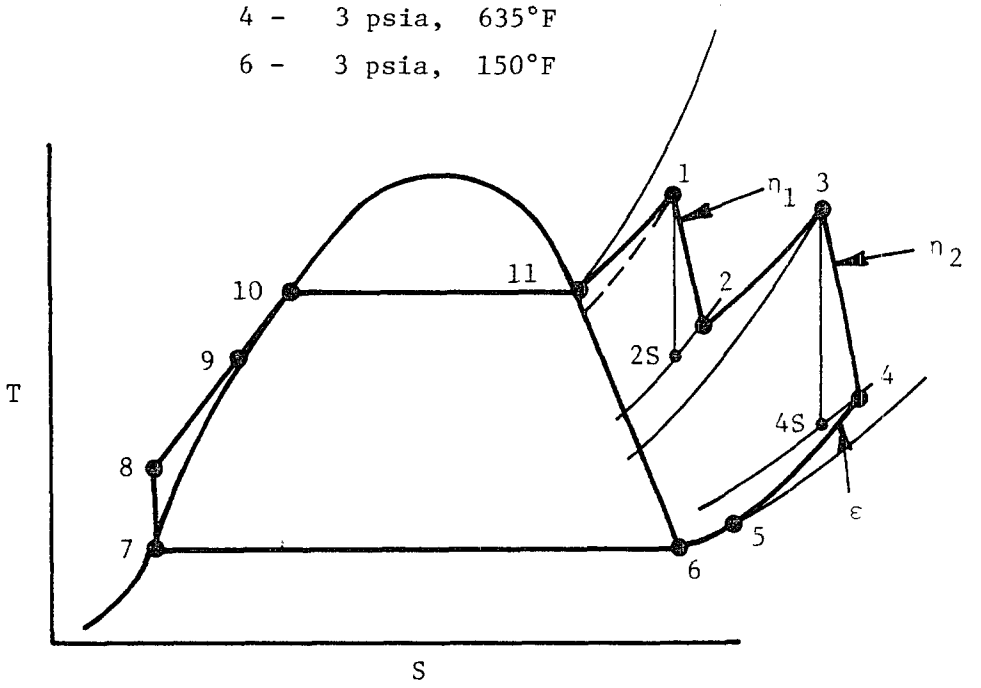

Fig. 2-10 Steam-Turbine Power Cycle T-S Diagram for 7½-Ton Unit

to 9. Even with the low aerodynamic expansion efficiencies characteristic of small-scale turbomachinery operating with high-energy fluids, the overall thermal efficiency of this turbine cycle was high.

The selection of the peak temperature was made on the basis of the established state-of-the-art for water chemistry and material compatibility. The selection of pressure was made on the basis of optimizing turbine and steam generator efficiency.

2.4.2 Steam Generator

The steam generator was designed and fabricated by Scientific Energy Systems (SES), Watertown, Massachusetts. The design was based upon a compact steam generator developed by SES for the Environmental Protection Agency's Rankine automotive program. The specifications for temperature, pressure and flow rates were provided by MTI. The SES steam generator shown in Figure 2-11 was developed for the Environmental Protection Agency.

2.4.3 Steam Turbine

The steam turbine design effort included consideration of numerous steam turbine design alternatives such as Ljungstrom, radial inflow, Curtis, Terry and reentry. The combination of the selection of a reheat steam cycle plus the efficiency characteristics of the different turbines resulted in the selection of a mechanically simple, single-row, axial-flow turbine wheel employing reentry. The single-row, impulse-bladed wheel accommodated two stages of expansion by use of a set of initial entry nozzles and a set of reentry nozzles.

Each stage of expansion used only a part of the total arc of the wheel. The combined first- and second-stage expansions used less than 90° of arc. The turbine wheel is shown in Figure 2-12. The efficiency of the first stage of expansion at 50,000 rpm was estimated at 56%; the second stage at 64%. The maximum design output of the turbine was 11 horsepower. Since the turbine geometry was fixed, variations in efficiency occurred with speed modulation.

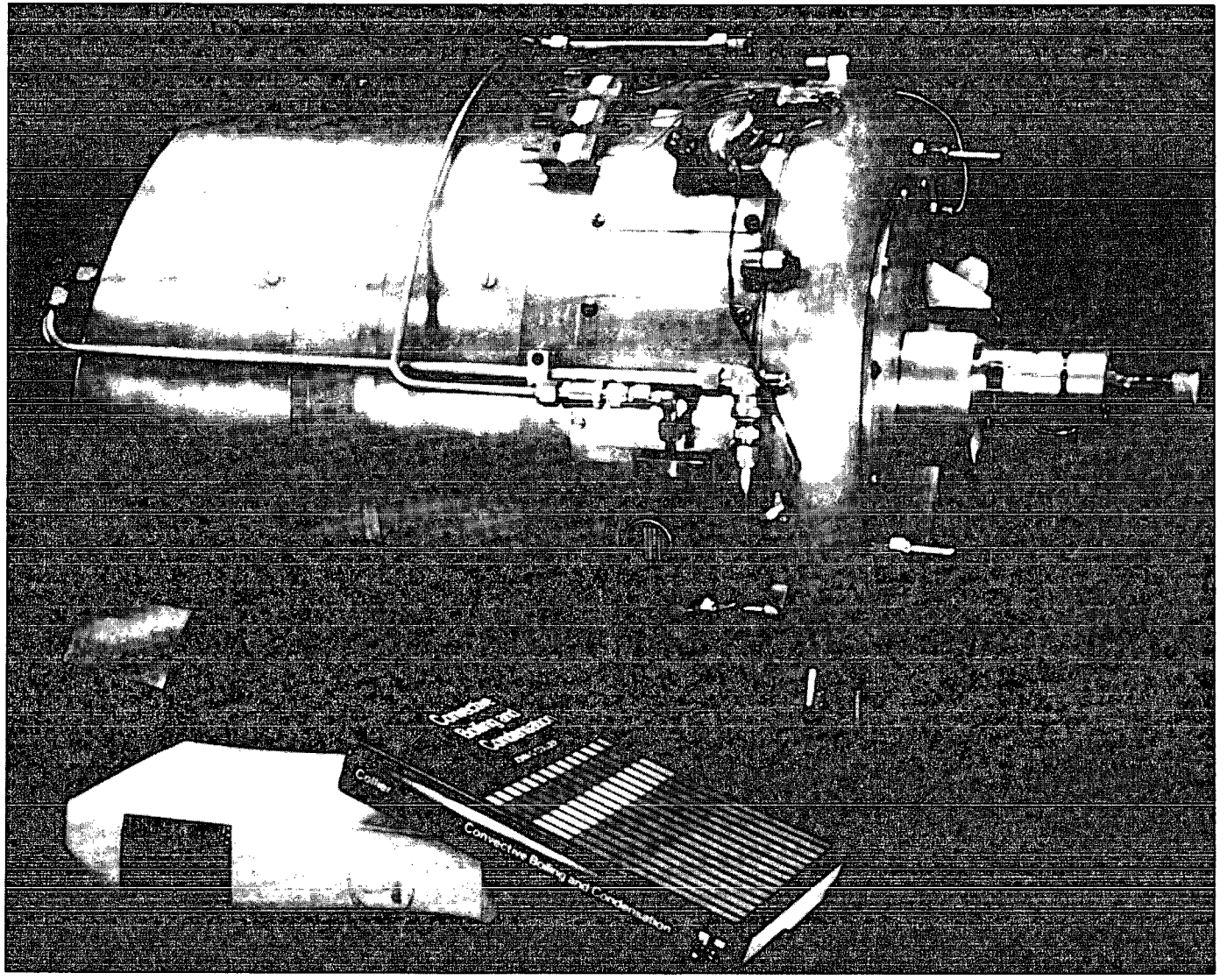

Fig. 2-11 The SES Steam Generator Developed for the Environmental Protection Agency

MTI-19276

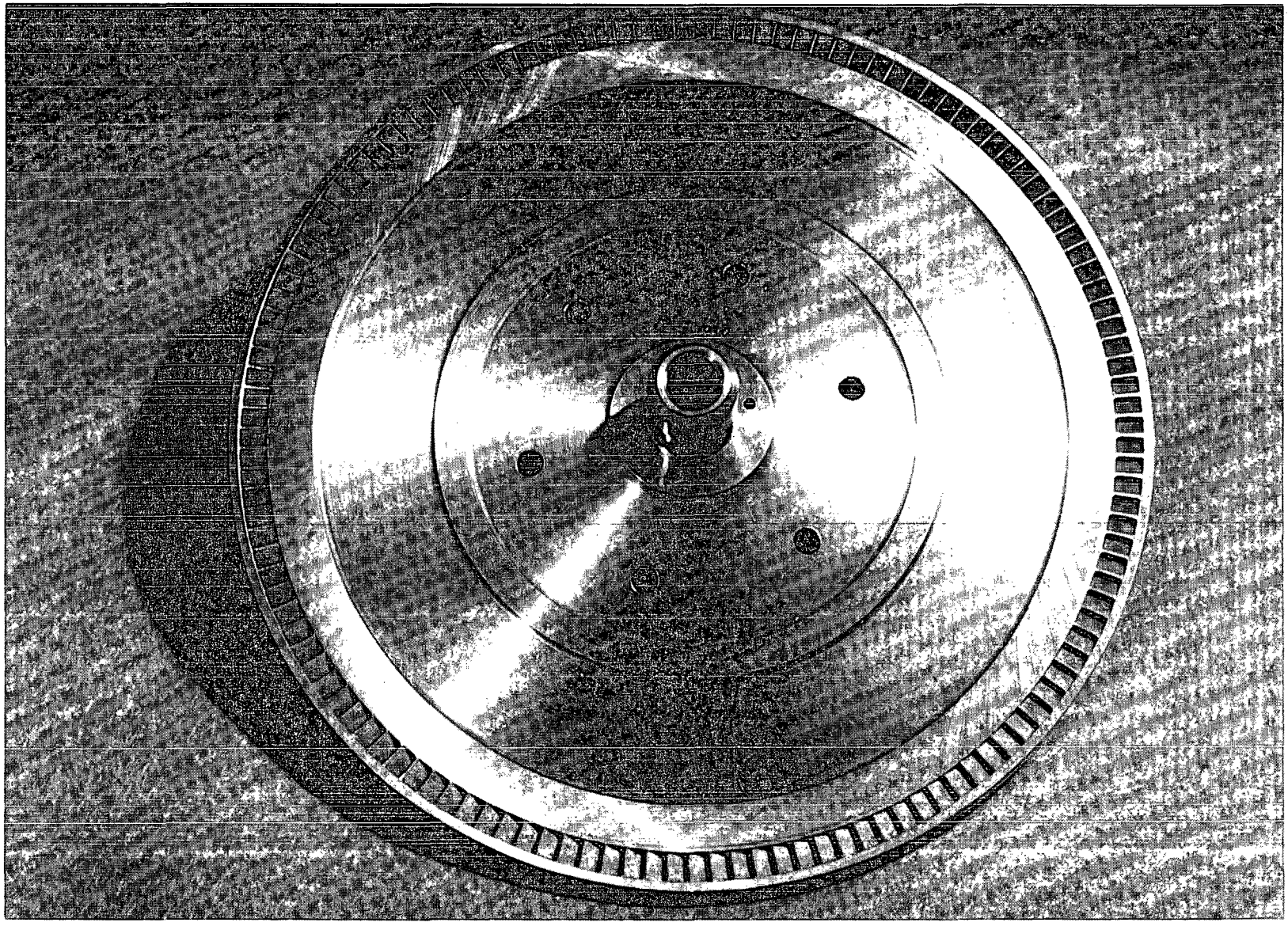

Fig. 2-12 Steam Turbine Wheel

MTI-18450

2.4.4 Alternator

The alternator was a "homopolar inductor" design. The output of the alternator at 50,000 rpm was 20 kW at 240 Vdc. The efficiency goal of the alternator was 90%. This goal was exceeded with actual performance at 93%.

2.4.5 Refrigerant Compressor

The refrigerant compressor employed a shrouded centrifugal compressor wheel. The compressor wheel is shown in Figure 2-13. This compressor was designed to provide 7½ tons of cooling capacity at the standard rating conditions with R-11. Capacity modulation was obtained through changes in speed and movable inlet guide vanes. The design has a target efficiency of 75% with a pressure ratio of 3.92 and a flow rate of 0.236 lb/sec. Test data showed that the compressor capacity was achieved but that overall efficiency and pressure rise were not achieved. These deficiencies were determined to be due to a nonoptimal diffuser design. No effort was made to optimize the diffuser design. The lower than design efficiency and pressure ratio of the compressor was largely responsible for the failure to meet the design performance of the breadboard system.

2.4.6 Regenerator

The regenerator in a steam-turbine prime mover handled a small portion of the thermal energy budget. In this system, between 8% and 13% of the heat input to the steam evaporator was handled by the regenerator. Recovery of this energy nonetheless improved the efficiency of the cycle. The regenerator should have transferred the energy from the turbine exhaust to the evaporator feedwater effectively and with a low pressure drop. The regenerator shown in Figure 2-14 accomplished these functions.

2.4.7 High-Speed Unit

The high-speed unit assembly consisted of the turbine, alternator rotor, compressor bearings and seals. Use of a mechanical shaft seal in this application, rather than the magnetic and electrical couplings employed in the residential unit developed previously, was made feasible by subsequent advances in high-speed mechanical shaft seal technology. The high-speed unit assembly is shown in Figure 2-15.

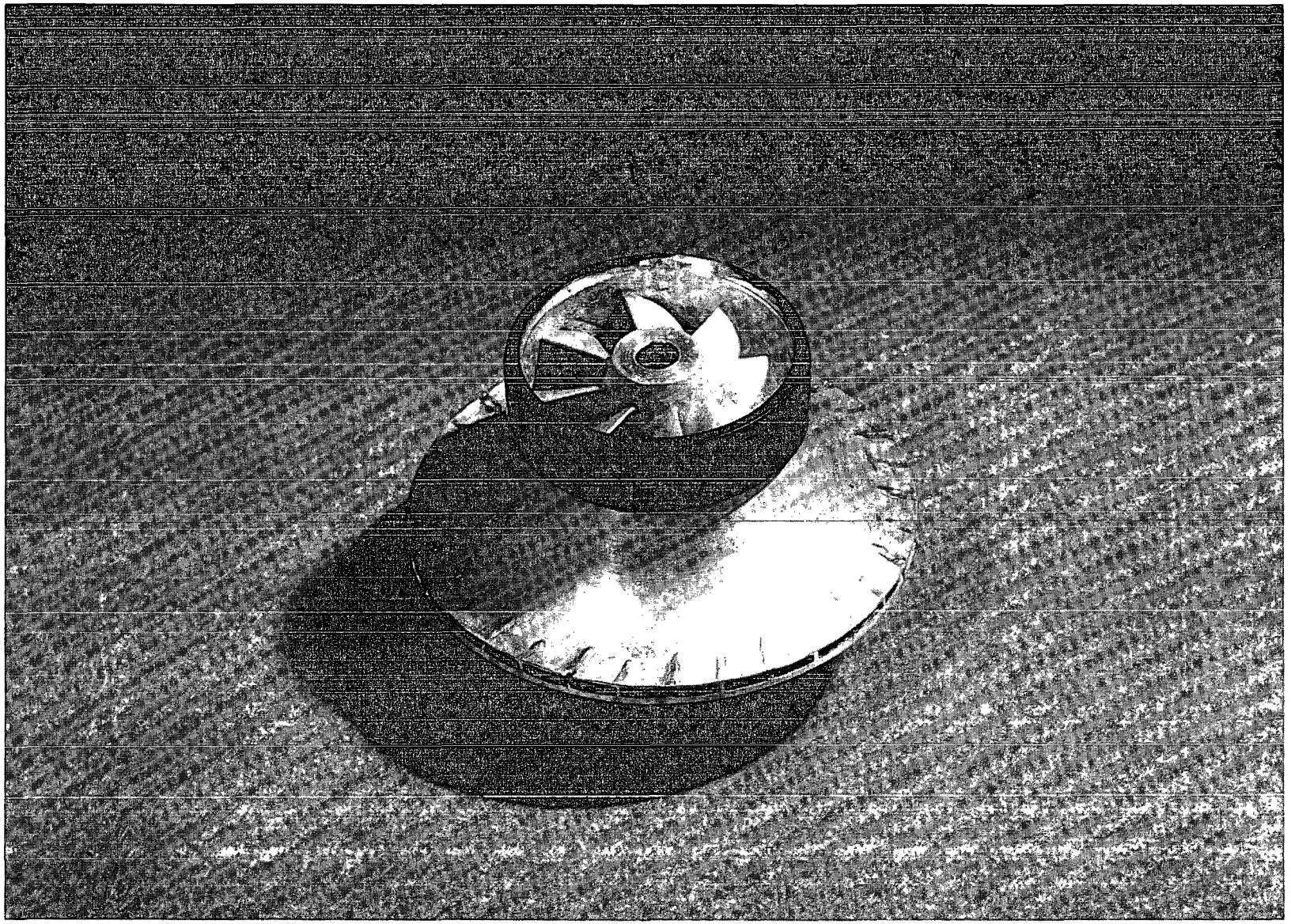

Fig. 2-13 Refrigerant Compressor Wheel

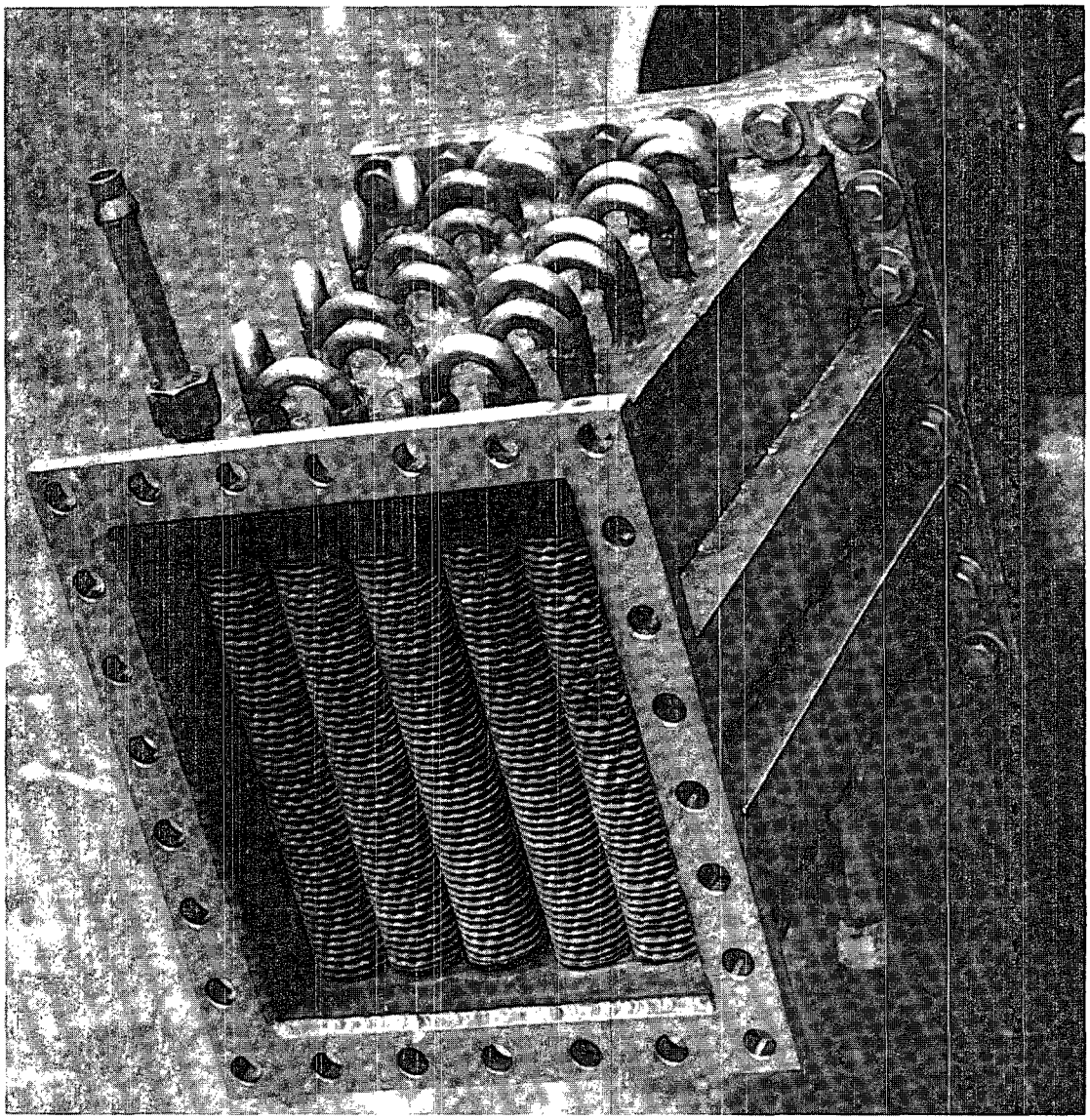

Fig. 2-14 Regenerator Core

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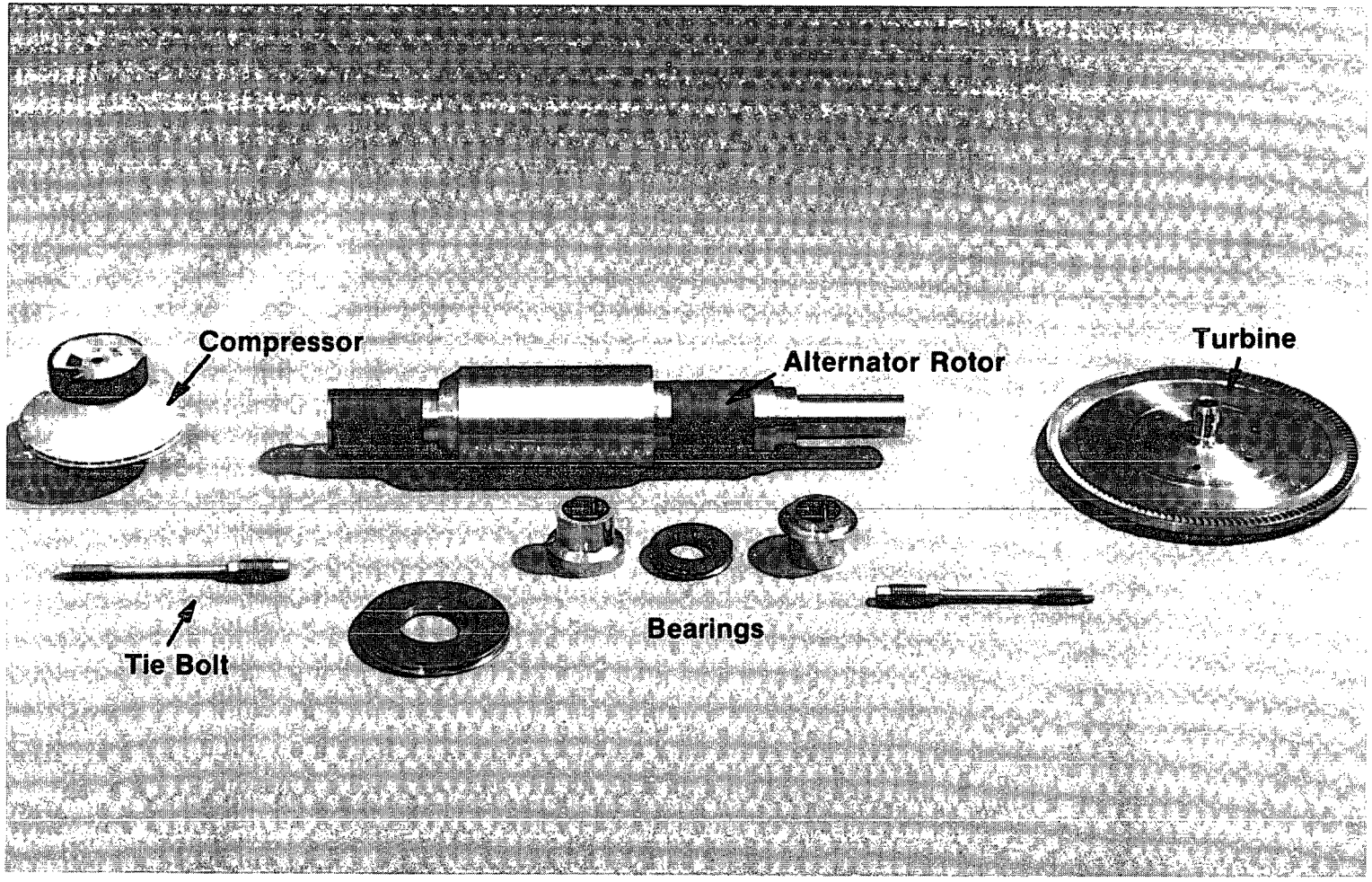

Fig. 2-15 High-Speed Turbomachinery Components

2.5 Component Test

All the individual components were tested to determine their performance. The test rig used to test the turbine, alternator and compressor is shown in Figure 2-16. In general, all components met their design goals closely except the compressor whose head rise capability and efficiency were less than predicted.

2.6 Proof-of-Concept Test

The components were assembled and tested as a system at MTI (Figure 2-17). In making the initial assembly, a number of clearances within the rotating package were set larger than design. There was a concern that the combined effects of thermal expansion and mechanical stress might result in more movement than predicted. The setting of larger clearances was dictated by prudence for protection of system integrity during the initial test. It was anticipated that some of these larger clearances, particularly on the turbine nozzle and recovery scoop, would result in degraded system performance.

The system tests for the proof-of-concept were successful. The equipment operated at speed, and repeated starts and stops did not affect the equipment. The performance was less than predicted, as expected. The differences were due to the larger clearances used, below-design performance of the compressor (as was determined during the component tests) and larger-than-expected parasitic losses.

The test results are presented in Table 2-2 in terms of the simulated seasonal performance of the demonstrator with heating and cooling loads corresponding to Cleveland, Ohio. The results are expressed as heating and cooling seasonal performance factors (SPFs). The test result SPFs were computed directly from coefficient of performance data generated by the demonstrator during proof-of-concept testing at MTI. Also shown in Table 2-2, for reference purposes, are four sets of analytically-derived performance data. The analytically-derived data and the test results are listed together in Table 2-2 in the chronological order of their dates of origin, as follows:

1. The original design performance predictions, made 7-76, at the time the 7½-ton unit design was first completed

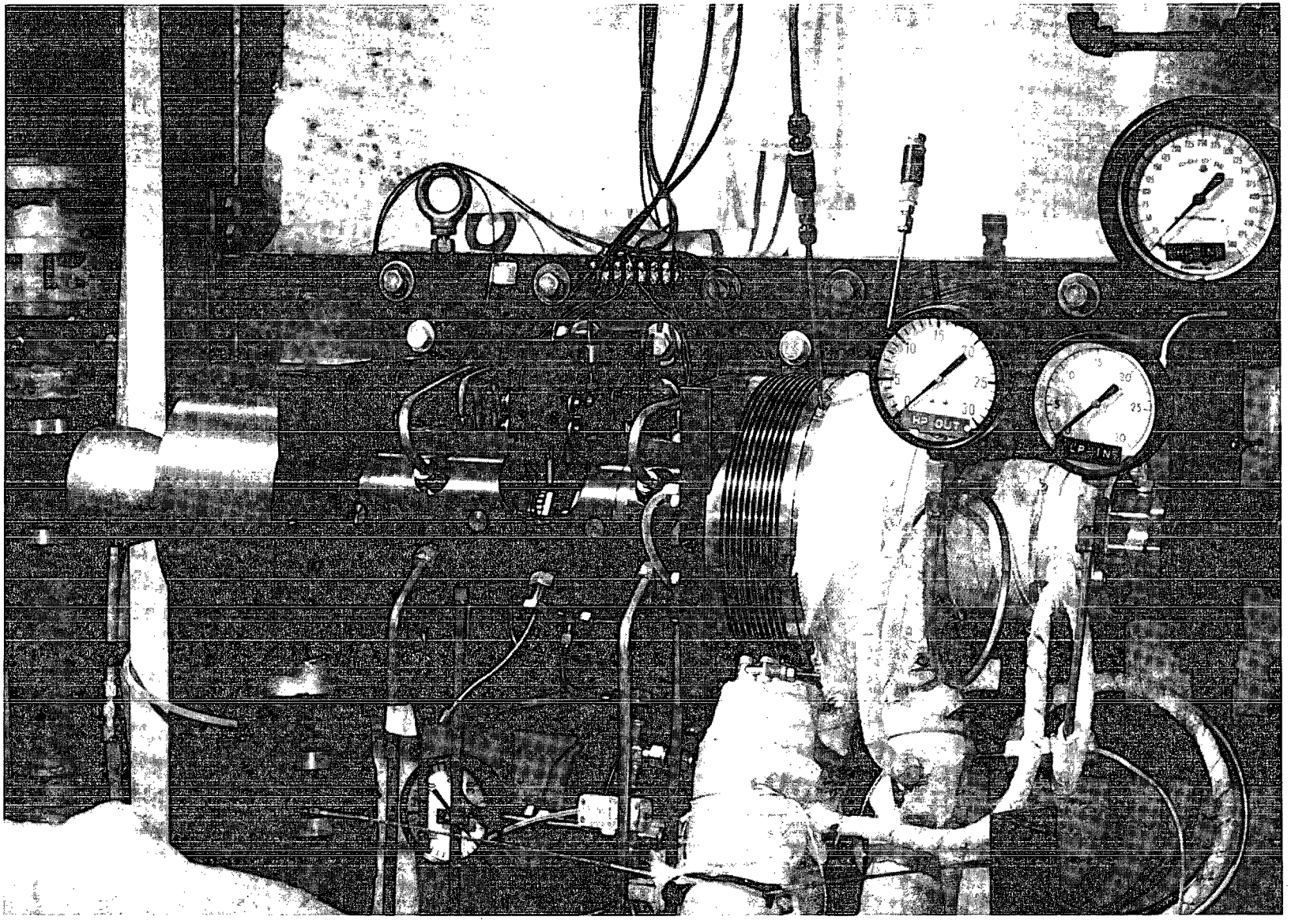

Fig. 2-16 Component Test Rig

MTI-17862

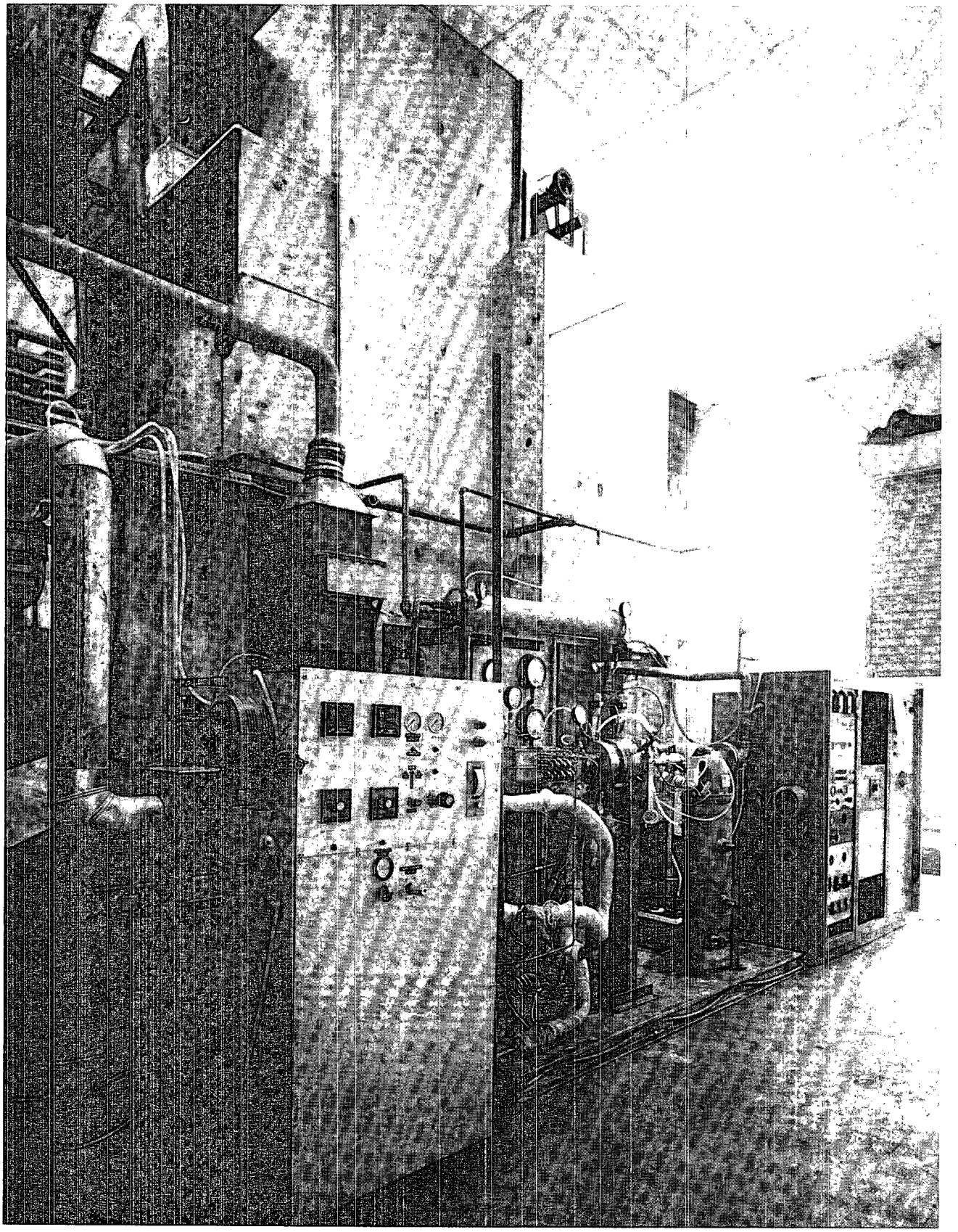

Fig. 2-17 System Tests Being Conducted at MTI's Malta, New York Facility

TABLE 2-2

7½-TON HSPF SYSTEM PERFORMANCE

	Seasonal Performance Factor[1] (Cleveland Weather)	
	Heating	Cooling
Original Design Performance Prediction (7-76)	1.21	.93
Generated by Comparative Cost of Ownership Study, Retail Store, Cleveland	1.31	.91
Derived from Observed Proof-of-Concept Test Data, 7-1/2-Ton Breadboard Demonstrator	1.08	.705
Predicted for Performance Demonstration	1.18	.90
Expected Product Performance		
a) Noncondensing Steam Generator	1.24	.91
b) Condensing Steam Generator	1.31	.96

NOTE: (1) Includes allowance for energy to drive self-powered auxiliary components, and based on higher heating value for natural gas.

2. Performance data generated in early 1977 by the Comparative Cost of Ownership Study (CCOS) run for Cleveland weather, the Retail Store (see Table 3-1, page 3-7). The performance specification input for the CCOS was from the same analysis that supported the original design performance predictions. The small differences between the design performance and CCOS results (+8%, heating; -2% cooling) are due to the much finer-scale simulation provided by the CCOS. The CCOS data should be most definitive.

3. The proof-of-concept test results generated at MTI in the summer of 1978

4. Performance believed to be achievable had the demonstrator been modified to meet original design specifications. These modifications would have incorporated system clearances set at design points, modifications to the compressor wheel, and reduction of system thermal and mechanical parasitics.

5. Performance to be expected from a refined product. These data are expressed in terms of both noncondensing and condensing-type steam generators. The data under 1 through 4 (above) corresponds to a noncondensing steam generator. A condensing type steam generator would require either additional heat transfer surface, or incorporation of pulse combustion or jet impingement technology.

Prior to the time the initial system tests were completed, the market and venture prospects for the 7½-ton model had been shown to be unfavorable, and the program had been redirected to the Large Size Model. Further system performance testing with the modifications identified under 4 (above) was, therefore, not carried out.

3.0 7½-TON ECONOMIC ANALYSIS

3.1 Overview

The 7½-ton economic analysis included an analysis of each of the following parameters.

- Product owning and operating cost compared to available alternatives
- Potential market size for a single product family
- Product first cost, compared to available alternatives
- Prospects for a successful product manufacuturing and marketing venture.

The analysis of potential market size is reviewed in Section 5.0, and the analysis of equipment cost is reviewed in Section 8.0. The analysis of the other two parameters is reviewed in this section, along with the market potential and equipment cost results obtained for the 7½-ton unit.

3.2 7½-Ton Comparative Cost of Ownership Study

The objective of the 7½-ton comparative cost of ownership study (CCOS) was to complete a preliminary evaluation of the comparative owning and operating costs of the proposed roof-top air-to-air 7½-ton HSPF gas heat pump product versus appropriate competitive equipment using the best data currently available. The CCOS was conducted by the Consolidated Natural Gas Service Company, with the help of the following organizations and consultants:

- Tom Parrish, Houston Natural Gas Co., assisted in loading the E-Cube-II program and, in particular, helped in adapting HVAC equipment instantaneous performance data, stated as functions of load factor and ambient temperature, for input to E-Cube-II.
- George Kelley, National Bureau of Standards, assisted in breaking out HVAC equipment instantaneous performance as functions of load factor and ambient temperature.

- HVAC equipment distributors and installers, in Cleveland and in Pittsburgh, and consultant Paul B. Moore, provided installed eost data on selected competitive HVAC equipment.

- Utility rate specialists, from gas and electric utilities in ten metropolitan areas in the U.S., provided current utility rate schedules.

- MTI supplied projected performance data for the 7½-ton HSPF gas heat pump.

Assembling the instantaneous performance data for the HSPF gas heat pump and for the competitive HVAC equipment versus load and temperature represented the greatest effort in preparing for the CCOS. Adding to the magnitude of the effort, two versions of the 7½-ton HSPF gas heat pump were defined for the study. One version was designated "nominal"; and the other as "max." "Max" referred to a set of technically consistent modifications designed to maximize the unit's hottest-day cooling output, at the expense of design-point (hottest day) operating efficiency.

Evaluation parameters included four HVAC equipment system types; four building types; ten weather areas (cities); ten actual commercial gas and electric rate schedules; and consultant-furnished equipment initial installed cost estimates. Descriptions of the data elements needed for each of the evaluation parameters follow.

Equipment

The four types of HVAC equipment were:

1. Year-round air conditioner (combination package gas heating and electric cooling units, referred to as a YAC)

2. Single package electric heat pump

3. Single package HSPF heat pump, max version

4. Single package HSPF heat pump, nominal version.

The conventional equipment types specified in 1 and 2 above were characterized by performance criteria for advanced gas direct-fired heating and advanced electric compression cooling equipment expected to be on the market in the mid-1980s. For example, the gas heating unit was presumed to have a power burner and IID. The performance estimates employed for the two HSPF gas heat pump options considered were prepared from design analysis data provided by MTI. An example of a map developed for modelling instantaneous equipment performance is shown in Figure 3-1. This map corresponds to the electric heat pump; heating mode.

Equipment cost estimates were provided from the Product First Cost Analysis (see Subsection 3.3). The first-cost premium estimate for the 7½-ton HSPF gas heat pump compared to the YAC, ranged from 400% to 550%. Prior to the completion of the cost estimates, a working assumption for the first cost of the 7½-ton HSPF gas heat pump was made that was substantially low. When the definitive estimate became available, the results from the CCOS were corrected accordingly. The correct results are shown herein. However, all of the data presented in Appendix Volume III, 7½-ton Comparative Cost of Ownership Study is based upon the initial, low equipment cost estimate. The return on investment and payback time data contained therein are affected.

Utility Rates

Commercial gas and electric utility rates which were in effect during 1976 for each city in the study were obtained and applied to the energy demand. All surcharges and taxes were included.

Utility rates were applied to the total building utility energy consumptions. This procedure permitted accurate determinations of energy costs by applying actual utility rates, including demand charges and promotional rates, to the total monthly consumptions of the respective equipment building combinations. General usages of utility energy for consumptions other than equipment usage were thereby included in the monthly and annual utility charge calculations.

The initial equipment costs and annual utility costs were escalated for inflation to January 1, 1984, the beginning of the study period. An assumed

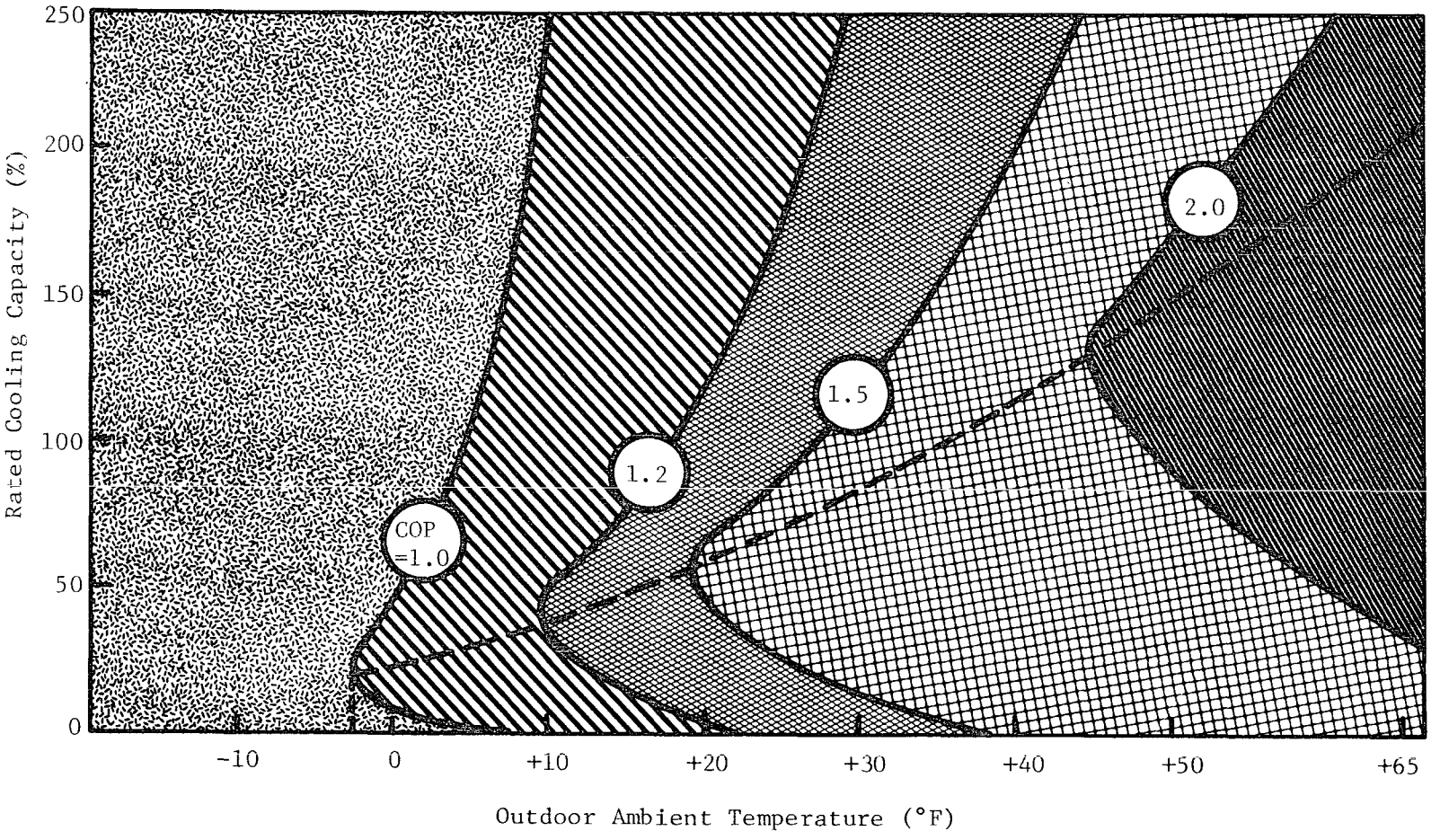

Fig. 3-1 System Performance - Heating Service

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national inflation factor was applied to annual utility costs during each year of the 20-year study period from 1984 through 2003.

Building Types and Weather Areas

The four building types consisted of a rest home, an office building, a retail store and a light manufacturing plant. The building specifications were modified to hold a constant 7½-ton cooling load in all ten cities (weather areas). These cities were: Cleveland, Pittsburgh, Syracuse, Detroit, Chicago, Los Angeles, Dallas, St. Louis, Atlanta and Washington, D.C. Heating loads were allowed to float. Representative year weather tapes for each city were used to obtain hourly weather data. Advanced building design standards (e.g., ASHRAE 90-75) were assumed for the load calculations. The building internal loads were simulated by 24-hour profiles typical for the usage patterns of the building type.

Building peak heating and cooling load determinations were made using ASHRAE methods. The building energy analysis models simulated the following:

- Hourly thermal requirements for the building
- Hourly, monthly and annual energy requirements of the two gas heat pumps and alternate equipment systems
- Monthly and annual utility costs using the actual present day (1976) and projected rates
- Economic comparisions via the E-Cube II format of the various equipment options, including the HSPF nominal and max systems versus the reference (lowest initial cost) system, which in all cases was the year-round gas-electric air conditioner (YAC).

In comparing the merit of the two gas heat pump versions to the two options for conventional equipment, the following product acceptance criteria were employed:

a. Three investment subcriteria: discounted cash flow rate of return, discounted cash flow payback and simple payback

b. The seasonal performance factor, based on source energy consumption

c. Annual utility costs.

Criteria (a) was regarded as dominant, in terms of market acceptance and product success.

The results of the equipment operating and ownership comparisons for the retail store building are presented in Tables 3-1 and 3-2 and Figure 3-2. These results may be summarized as follows.

1. Based upon best estimates for the actual first-cost premium for the 7½-ton HSPF gas heat pump, the gas heat pump would not represent a good investment for any of the cases studied. Simple payback times were 8 years and up in all cities. The worst case was Los Angeles, where annual HVAC loads are lowest; payback times there exceeded 20 years. In other parts of the country, payback times were largely a function of local utility rate schedule.

2. The heating season energy conservation performance or seasonal performance factor of the gas heat pump approached twice that of advanced direct-fired gas heating equipment.

3. Based on consumption of primary energy resources, the cooling season seasonal performance factor of the gas heat pump was shown to be approximately twice that of present-day, direct-fired absorption cooling units, and 25% better than the best electric compression units.

4. Only a small differential was found in annual operation or utility costs between the maximum and nominal versions of the gas heat pump. The maximum version was physically smaller by 30% and could offer some reduction in cost. (This potential cost reduction would not be adequate to affect the cost premium induced conclusions, however.)

TABLE 3-1

SEASONAL PERFORMANCE FACTORS FOR A RETAIL STORE BASED UPON ENERGY CONSUMED AT THE SITE

LOCATION	SERVICE	ON-SITE SEASONAL PERFORMANCE FACTORS			
		YAC	EHP	GHPM	GHPN
CLEVELAND	Heating	0.711	1.860	1.328	1.313
	Cooling	2.219*	2.557*	0.866	0.911
PITTSBURGH	Heating	0.712	1.856*	1.328	1.313
	Cooling	2.245*	2.591*	0.947	0.961
DETROIT	Heating	0.709	1.868*	1.330	1.314
	Cooling	2.176*	2.493*	0.985	0.981
CHICAGO	Heating	0.710	1.844*	1.323	1.309
	Cooling	2.183*	2.509*	0.962	0.968
SYRACUSE	Heating	0.712	1.842*	1.326	1.312
	Cooling	2.245*	2.580*	0.983	0.985
WASHINGTON	Heating	0.680	1.937*	1.310	1.289
	Cooling	2.234*	2.578*	0.939	0.951
ATLANTA	Heating	0.674	2.062*	1.327	1.304
	Cooling	2.235*	2.583*	0.949	0.960
ST. LOUIS	Heating	0.683	1.818*	1.284	1.264
	Cooling	2.184*	2.520*	0.925	0.940
DALLAS	Heating	0.673	2.077*	1.332	1.310
	Cooling	2.131*	2.455*	0.857	0.877
LOS ANGELES	Heating	0.677	2.335*	1.386	1.359
	Cooling	2.377*	2.716*	1.016	0.995

Equipment Seasonal Performance Factors are ratios of the cooling or heating effects (BTU outputs) divided by the respective equipment energy amounts (BTU inputs) at the point-of-use.

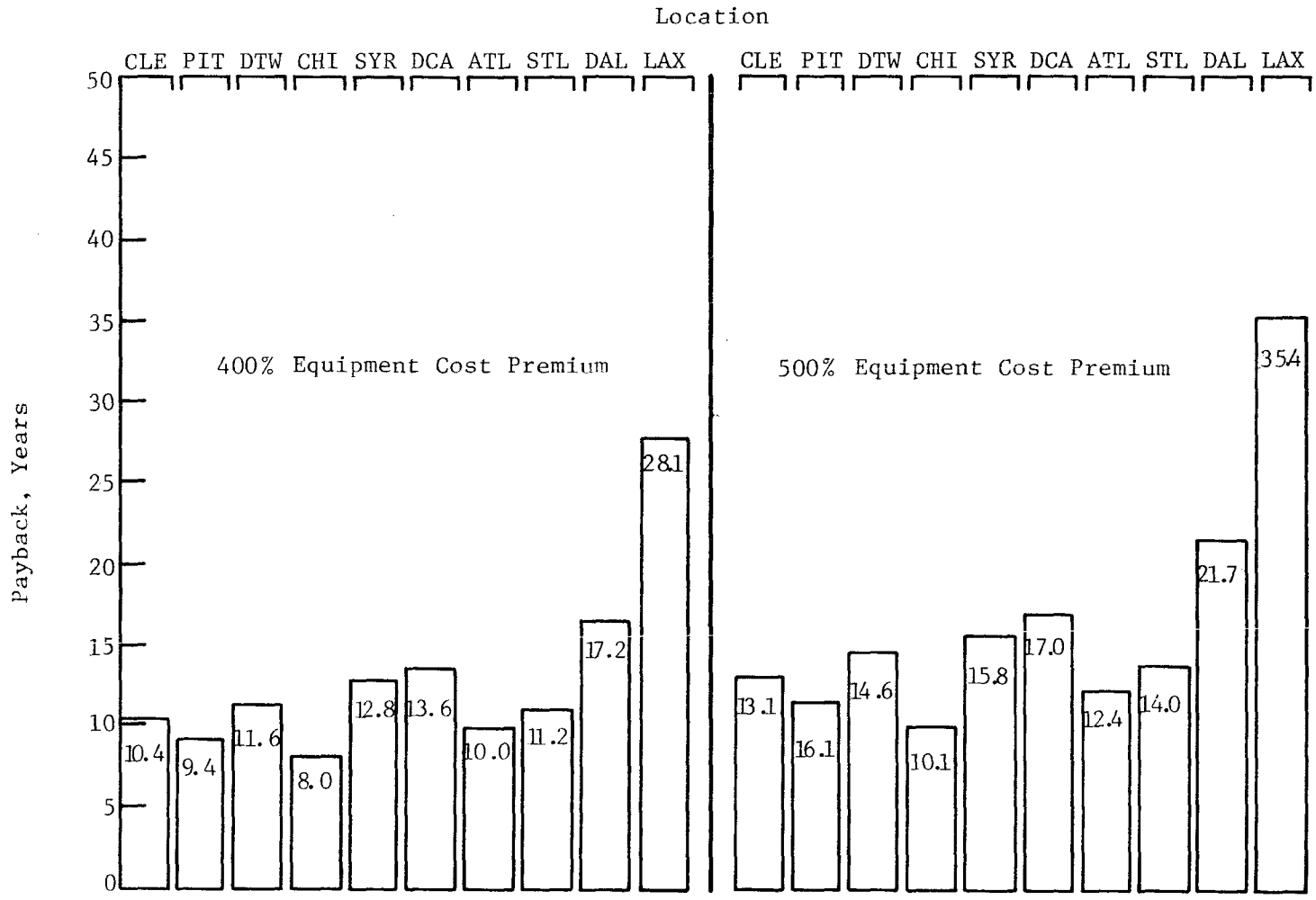
*SPF based upon on-site valuation of electricity at 3413 Btu/kW-hr. To make the comparible-to-gas SPF values, multiply the electric values by 0.32.

TABLE 3-2

UNITIZED BUILDING TOTAL UTILITY COSTS FOR A RETAIL STORE

LOCATION	UTILITY	UNIT COST ¢/ft²/year			
		YAC	EHP	GHPM	GHPN
CLEVELAND**	Elec.	63.19	90.77	45.34	45.34
	Gas	10.81	-----	11.20	11.02
	Total	74.00	90.77	56.54	56.36
PITTSBURGH	Elec.	55.51	60.74*	35.47	35.47
	Gas	9.89	-----	10.25	10.24
	Total	65.40	60.74	45.72	45.71
DETROIT	Elec.	50.48	68.82*	37.62	37.62
	Gas	13.56	-----	10.83	10.93
	Total	64.04	68.82	48.45	48.55
CHICAGO**	Elec.	62.12	47.77	43.57	43.58
	Gas	14.76	-----	11.46	11.52
	Total	76.88	47.77	55.04	55.10
SYRACUSE	Elec.	49.02	74.04	34.27	34.28
	Gas	13.64	-----	13.95	14.02
	Total	66.66	74.04	48.22	48.30
WASHINGTON	Elec.	67.12	71.59*	47.00	47.00
	Gas	4.94	-----	10.52	10.45
	Total	77.06	71.59	57.52	57.45
ATLANTA	Elec.	61.48	61.65	38.46	38.46
	Gas	2.23	-----	6.10	6.06
	Total	63.71	61.65	44.56	44.52
ST. LOUIS	Elec.	62.59	36.31*	44.55	44.56
	Gas	6.84	-----	8.36	8.32
	Total	69.43	36.31	52.91	52.88
DALLAS	Elec.	52.43	42.31*	32.24	32.24
	Gas	2.56	-----	10.60	10.42
	Total	54.99	42.31	42.84	42.66
LOS ANGELES	Elec.	114.91	115.89	103.05	103.05
	Gas	1.82	------	6.29	6.42
	Total	116.73	115.89	109.34	109.47

*All Electric or Heating Rate
**Winter/Summer Rate



$$\text{Payback} = \frac{\Delta\ \text{Investment}}{\Delta\ \text{Operation Cost}}$$

$$\text{Actual Simple Payback} \cong (\text{Base Payback}) \times \frac{(\text{Actual Premium})}{(\text{Base Premium})}$$

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Fig. 3-2 Simple Payback Time of HSPF for a Retail Store (7½-Tons) HSPF Gas Heat Pump - Maximum Cooling Versus Combination Heat/Cool Unit (YAC)

5. The 7½-ton HSPF gas heat pump would be the most resource-efficient HVAC option, on an annual basis, in any of the metropolitan areas studied.

6. Annual utility costs for the gas heat pump were shown to be significantly lower than the year-round air conditioning units (YACs) in all study cities.

7. Annual operating or utility costs for the gas heat pump were shown to be less than those of the electric heat pump in those study cities without a special promotional all-electric utility rate.

8. The effect of promotional all-electric rates on the relative competitive position of the gas and electric heat pumps were substantial.

9. An installation cost advantage was shown to exist for the HSPF system. This advantage represented 30% of the cost of competitive equipment at 7½-tons and 25% at 30 tons. This advantage was due to eliminating the requirment for a heavy electric power connection.

10. On economic criteria, the projected advanced design electric heat pump appeared to be a better investment than a YAC in seven of the ten cities studied with the cost premiums assumed.

The programs used in the CCOS study are described in Appendix Volume III to this report, Addendum Subsections 6.9, 6.10 and 6.11. Flow charts are included with Subsection 6.9. A complete set of printouts is shown in Subsection 6.13. The sample payback time results shown in Appendix Volume III have not been corrected for the low initial equipment cost premium assumption.

3.3 7½-Ton Unit Venture Prospects

Technical feasibility, along with the venture prospects for a business producing and/or marketing 7½-ton HSPF gas heat pump product, represented the bottom line for the entire 7½-ton program. The venture prospects were evaluated with the aid of the 7½-Ton Comparative Cost of Ownership Study, the 7½-Ton Market Potential Study and the Equipment Cost Study. The interrelationships between the Market Potential, the Equipment Cost and the

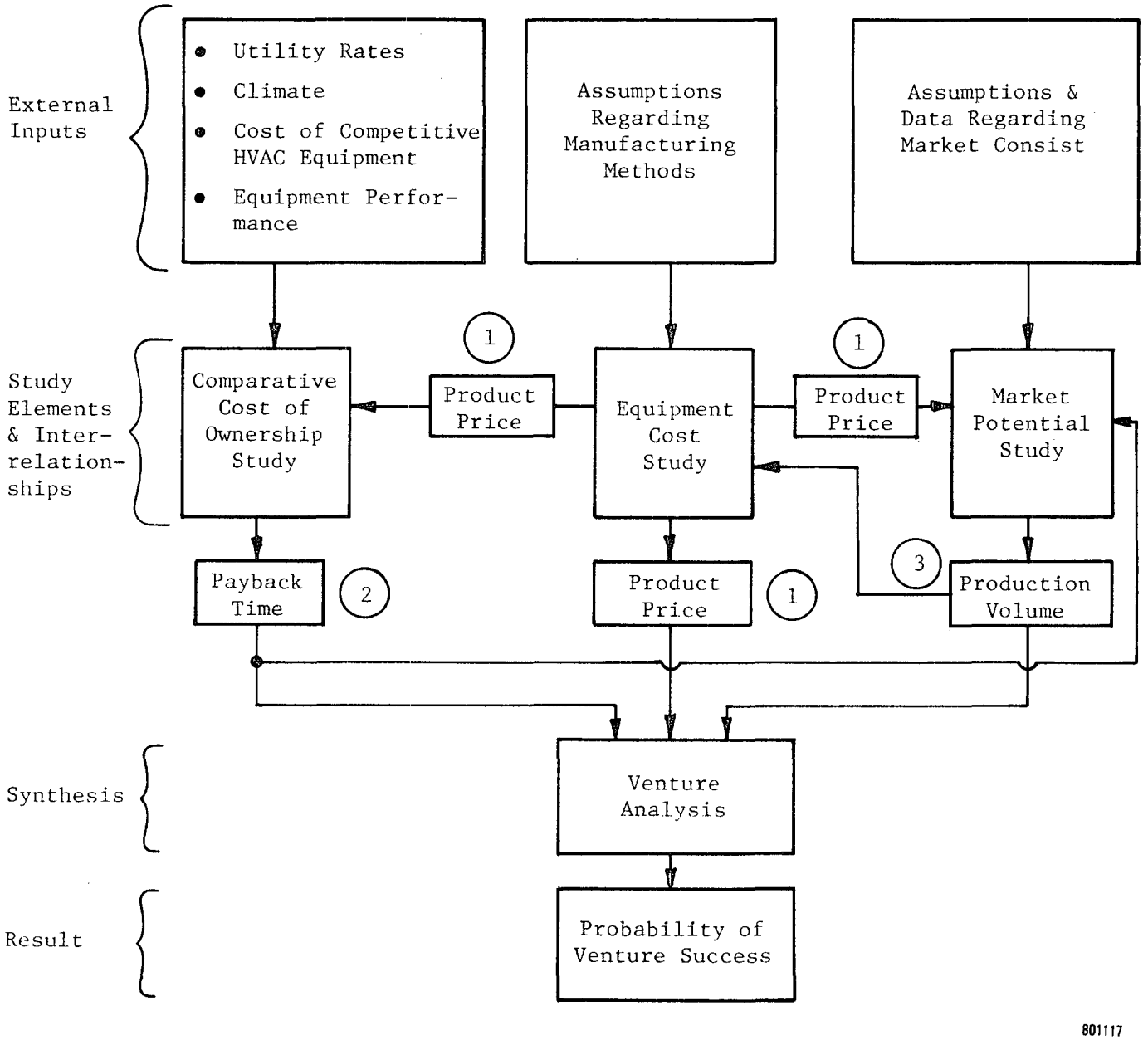
Ownership Studies, which were the principal inputs to the Venture Analysis, are shown in Figure 3-3 and Table 3-3.

The Comparative Cost of Ownership Study, Equipment Cost Study and the Market Potential Study were performed concurrently, and each was performed once. Ideally, these studies should have been performed twice; the first result from each would be employed to adjust the assumptions incorporated in the second iteration. The results from the second iteration would then produce more accurate values for each of the parameters that form the interrelationships between the three studies.

The coincidental one-time performance of these studies meant that assumed values for several important parameters had to be used. Assumed values employed were as follows:

1. An assumed range for system equipment cost was employed in the Comparative Cost of Ownership and Market Potential Studies (see Subsection 3.2).
2. An assumed range for market volume (and corresponding production volume) was employed in the Equipment Cost Study.
3. An assumed range for product ownership costs was employed in the Market Potential Study.

The initial assumption regarding the projected cost for a 7½-ton HSPF gas heat pump product was substantially in error. The range of assumed unit costs was defined as a 30% to 70% equipment cost premium over the baseline conventional equipment, a gas furnace/electric air conditioner package (YAC). This percentage translated to an assumed cost range of $2900 to $3800 for the HSPF heat pump, based upon a quoted price of $2200 for the conventional equipment. The HSPF heat pump product cost, as later determined by the Equipment Cost Study, approached $12,000, representing a 400% to 550% first-cost premium. The magnitude of this difference created doubts regarding interpretation of the results producted by the Market Potential Study. The results from the Comparative Cost of Ownership Study, however, were adjusted.



NOTE: Circled numbers refer to notes listed on Table 3-3.

Fig. 3-3 Venture Success Study for the 7½-Ton HSPF Gas Heat Pump Product

TABLE 3-3

INTERRELATIONSHIPS BETWEEN 7½-TON VENTURE SUCCESS STUDY ELEMENTS

1. Product Price refers to the selling price of the 7½-ton HSPF unit as determined from the equipment cost study based upon a specified production volume. This volume is used to determine the first cost premium and utility the payback times. This volume would also be used to define revenues, profits and investments in the venture analysis.

2. Payback Time refers to the number of years required by the user to recover the gas heat pump's first-cost premium from operational (energy) cost savings.

3. Production Volume refers to the number of units sold as defined from the market potential and penetration into the market. The market potential is a function of both first-cost premium and user payback time.

The equipment cost result became the dominant input to the Venture Analysis. Based on any interpretation of the Market Study, the projected market became very small. The Market Study clearly showed a rapid increase in buyer resistance as first-cost premium increased beyond 50%. The final cost result, corresponding to a several hundred percent first-cost premium was well off-scale. Extrapolation based on Market Study results, however, indicated that the market volume would be virtually nil. This, in turn, would further aggravate production costs, as the Cost Study assumed a production rate of several thousand units per year. As the production rate decreased, production costs would increase.

The Venture Analysis recommendation was thus based solely upon implication of the projected gas heat pump product cost. Several other results became available from the three studies that contributed to the Venture Analysis. The significant results are summarized below:

1. The 7½-ton HSPF product cost estimates indicated that the 7½-ton system would have an equipment first-cost premium of several hundred percent relative to competitive conventional package equipment.
2. The first-cost premium tended to reduce as unit size was increased. It appeared that the premium might become manageable at the 75- to 125-ton unit size.
3. A negligible market potential was indicated at the 7½-ton size level, principally as a result of the high first-cost premium.

The Program Advisory Board reviewed this information and recommended that the 7½-ton program be terminated, except for the technical demonstration effort.

3.4 Program Redirection

Following the full review of the results of the 7½-ton program, the Board recommended that, if the HSPF program was to continue, the program should be redirected to larger size systems. This recommendation was based on the

indications that system cost would become more price competitive as system size increased and that a greater acceptance of the principals of lifetime cost analysis existed in the large equipment market than in the small equipment market.

4.0 LARGE SIZE MODEL PROGRAM OVERVIEW

The large size model (LSM) program was initiated in September, 1977. The LSM system was planned as a built-up central HVAC system consisting of a steam turbine-driven chiller/cogeneration unit. The electric generator also served as a supplemental motor drive for the chiller during peak load periods. Internal and external air-source heat pumping were options. Overall, the system was planned to provide heating, ventilating and airconditioning functions for large commercial applications. The objectives established for the LSM program were to demonstrate the total potential of the LSM HSPF Gas Heat Pump as a profitable product line and to secure a definite arrangement for commercialization.

The program tasks were grouped into three phases. The task organization is shown in tabular form as Attachment B. The report section containing a full description of each task is cited. The dates shown for the three major phases correspond to the proposed periods during which the effort for these phases was to be performed. Attachment B also shows the principal milestones associated with the LSM program, and the degree to which each was reached. For those milestones corresponding to program decision points, the decision rendered is indicated.

Generally, the first phase was to demonstrate the viability of a business venture based upon LSM HSPF technology. Hardware development was limited to that effort needed to support the program objectives. Accordingly, with the successful demonstration of HSPF System technology with the 7½-ton breadboard demonstrator, further technical demonstration in the first phase of the LSM program was not needed. The first phase of the LSM program was, therefore, planned around the following major tasks:

1) Technical Effort (support for the system cost analysis, only)
2) Market Potential Study
3) System Cost Analysis
4) Comparative Cost of Ownership Study
5) Business Potential Study (Venture Analysis)

Unlike the experience of the 7½-ton program, each of these tasks was phased to maximize the accuracy of the information for the Business Potential Study. Except for slippage experienced in a few areas, phasing maximized the effectiveness of the Business Potential Study.

The results of each task were integrated and scheduled to meet a major program review milestone in April 1979. The results that were available at this review are as follows:

1. The equipment first-cost premium for the LSM was estimated at about 70% over the best conventional HVAC system. This would pose a significant barrier to market penetration.
2. HSPF system efficiency advantages were found to be largely eliminated in the context of a hot and chilled water distribution system and multizone loads. System seasonal efficiency was further reduced for HVAC loads dominated by the cooling load.
3. The LSM was shown to offer little or no return on investment or source energy savings in applications representing the majority of the 60- to 400-ton equipment market.
4. A relatively small market was identified where the LSM system should represent an attractive investment or means for conserving source energy. That market is characterized by locations where gas and electric energy costs are, or are expected to be, relatively high and by applications where the heating load predominates.
5. The limited market potential project for the LSM, combined with the traditional difficulty in financing new products in the HVAC industry, led to a very low estimate for the probability that a business venture based on an LSM product could succeed.

Following review of the above results, the CNG, DOE and MTI project management team determined that all technical effort on the LSM program should be terminated. The basis for this determination was the very low potential for

the LSM HSPF gas heat pump to become the basis for a profitable product line. The Program Advisory Board met in May, 1979, and after reviewing the results, confirmed the project management team's findings. CNG, DOE, MTI and the Board recommended that further efforts on a steam turbine-based HSPF system cease at this time.

The first phase also included initial planning and negotiation for a first full-scale LSM field test. Negotiations and arrangements were initiated with the Wood County, West Virginia, Park Board to use the proposed new lodge at Mountwood Park as a field test site. The proposed lodge was a very favorable site. Arrangements were nearly complete when the LSM program was terminated. An effort to design, build and test an engineering test unit was also planned as a part of the first phase. This task (No. IX) also was not initiated, because of the LSM program termination.

The second phase of the LSM program would have carried the project through to the completion of the first full-scale field demonstration as well as an extensive product development effort. DOE, CNG and MTI commitment to the second phase would have been predicated on a positive result from the Business Potential Study. The third phase would have been to conduct a second and a third field test. The second and third field tests were to differ from each other in terms of the nature of the HVAC load and climate. The second and third field test units would have incorporated the results of the second-phase product development effort. Continued development beyond FY 79 was also predicated on a commitment from a venture partner capable of marketing the LSM.

5.0 MARKET POTENTIAL STUDIES

A comprehensive evaluation of the market potential for the nonresidential heating and cooling market was conducted by William E. Hill and Company under contract to CNG. The market potential was segmented by equipment class (unitary and built-up) and by unit size. The market potential study evaluated a number of important market influences, including: market structure, distribution chain and customer attitudes.

The market potential estimation effort was carried out in two phases:

- The first phase centered on the 7½-ton HSPF gas heat pump concept. The starting point for this phase was a specific product concept (rooftop unitary heat pump) and a specific market area (North Central United States). Thus, the focus of the first phase was the projection of market potential for unitary heat pumps in the 7½- to 50-ton range for the North Central states.
- The second phase centered on the Large Size Model (LSM) gas heat pump concept. The starting point for this phase was a technological concept. The focus of the second phase was to specify a product that maximized the market potential.

The results of these market studies were used to evaluate the business potential of the two heat pump concepts, as covered in Sections 3.0 and 9.0 of this report for the 7½-ton unit and LSM, respectively.

The methodology and the basic results of the two market studies, which are described in detail in Appendix Volume II and Appendix Volume VI, are similar. The market evaluation of the LSM concept was an extension and expansion of the study evaluating the 7½-ton concept; therefore, the discussion comprising this section of this report will focus primarily on the LSM market potential study.

The approach of the market studies is illustrated in Figure 5-1. Historical HVAC statistics and data were analyzed first. General commercial construction activity data were obtained from F.W. Dodge; HVAC shipment volumes and

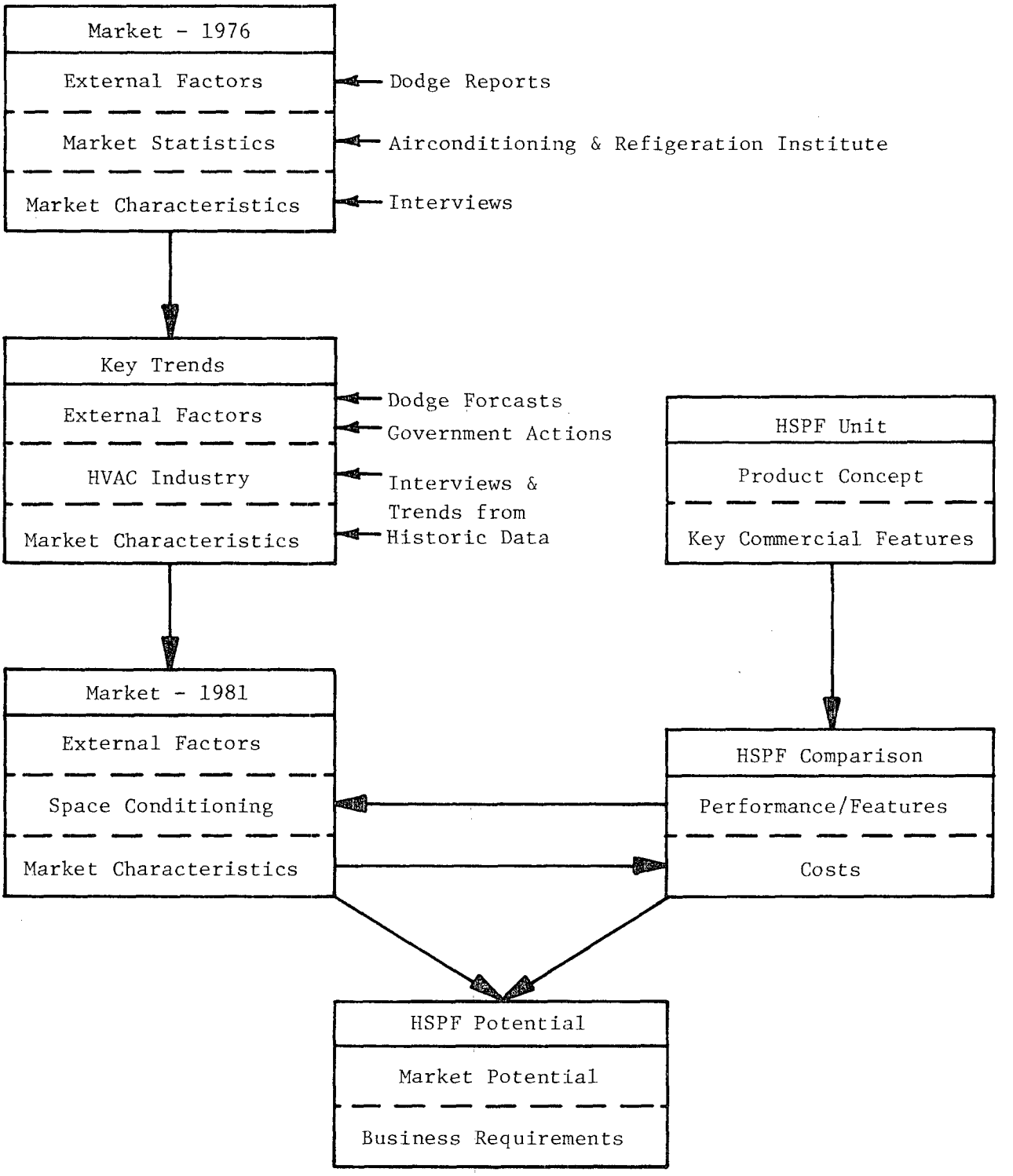


Fig. 5-1 General Methodology of Market Potential Study

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values were obtained from the Air Conditioning and Refrigeration Institute, the American Gas Association, the U.S. Department of Commerce, and other trade and industry sources. These data were supplemented with field research designed to enrich the understanding provided by the quantitative data. The field research also provided insight into the HVAC industry, its characteristics and its trends. The field research consisted of the following:

- Members of the William E. Hill & Co. professional staff visited sixteen major cities for the 7½-ton unit study and seven major cities for the LSM study. In each of these cities, the staff conducted in-depth interviews with consulting engineers, owners, builders, utilities, distributors and contractors. This research investigated the major elements in the market structure and qualitative influences on the market in question. Over 100 in-person field interviews were conducted.

- Telephone interviews were conducted among additional mechanical contractors and consulting engineers who were involved in large-tonnage equipment applications. These interviews provided a broader statistical base and additional quantitative data not available from other sources. Almost 200 telephone interviews were conducted.

- In-depth personal interviews were also conducted with top-level managers with several equipment manufacturers active in large-tonnage HVAC equipment. Numerous manufacturer field sales managers and technical sales representatives were also interviewed.

The quantitative data and interviews provided the base point and identified the key trends from which the market potential and market characteristics of the HVAC industry could be projected for 1981. For the 7½-ton unit, this projection was combined with the product specification assumptions to yield the market and business potential. For the LSM, the projection defined the product concept that would maximize the market potential.

A key output of these market potential studies was a quantitative statement of the commercial HVAC market's sensitivity to price premiums and life-cycle costing. Energy-conserving HVAC equipment would generally be more expensive; therefore, it usually had to be justified on the basis of life-cycle costs. The acceptance of life-cycle costing in purchase decision-making was an important criterion in assessing the market potential.

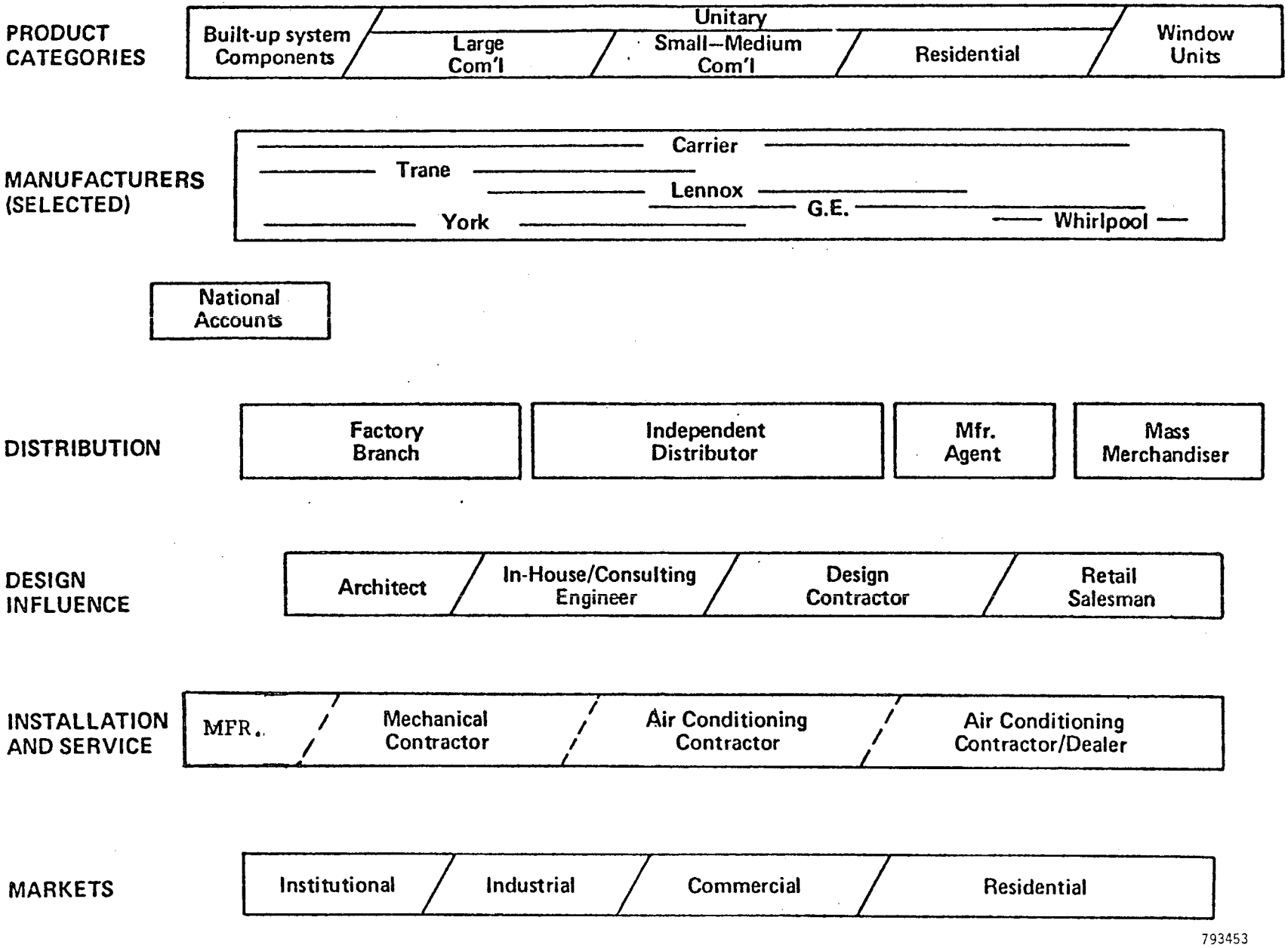
5.1 Market Characteristics

The heating, ventilating and air conditioning (HVAC) industry is large, diverse, and highly segmented. While the study was concerned with only one product segment of the HVAC industry, it was necessary to understand the entire industry including product manufacturing, distribution, selection and use.

Figure 5-2 presents a schematic of the HVAC industry from product to manufacturer to end use. The flow of goods from manufacturer to end use, as shown in Figure 5-2, tends to follow a vertical path. However, virtually any combination of manufacturer, distribution, service, specification, purchase and end use is possible and does occur. This is illustrated by the "simplified" structure in Figure 5-3.

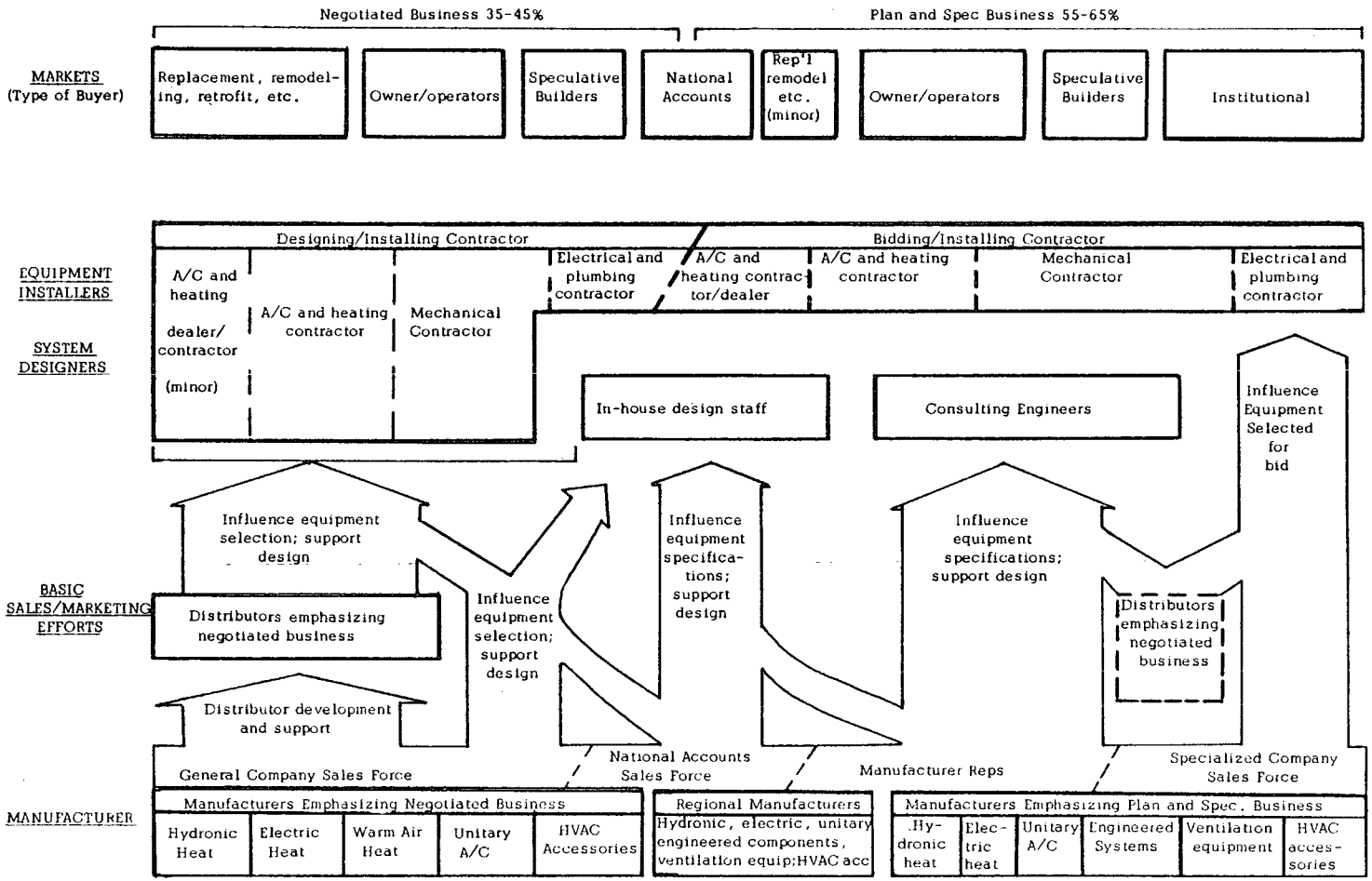
The manufacturing of HVAC equipment tends to fall into one of two approaches. The first is an integrated manufacturing approach which is the more familiar and recognizable. In this case, the manufacturer makes virtually all the components that go into the product. General Electric and Carrier are examples of integrated manufacturers. The second approach is the assembly approach. An extreme example of this approach would be an assembler who does no manufacturing but who simply assembles components which have been purchased from original equipment manufacturers (OEM) and subcontractors. Most companies in the HVAC industry are assemblers.

The integrated manufacturers have, in general, the largest market shares. They have a more extensive product line within any product classification (such as central chillers); they have national coverage; they have extensive supporting products (such as terminal units); and they frequently have



Source: William E. Hill & Co., Inc.

Fig. 5-2 Simplified Structure of the Air Conditioning Industry



Source: William E. Hill & Co., Inc.

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Fig. 5-3 Simplified Structure of the Non-Residential Heating and Air Conditioning Market - 1976

a better reputation with the specifiers. The integrated manufacturers account for a large proportion of the shipments of any given product category. In general, three or four integrated manufacturers account for over 60% of shipments of a given product classification. In the large-tonnage built-up product line, there is an even greater concentration with three integrated manufacturers accounting for over 80% of shipments. To optimize their production capacity, the integrated manufacturers often buy some components from OEM suppliers.

Production economics limit the number of OEM suppliers and, thus, there is greater concentration in some HVAC component lines than in the HVAC product business. None of the assemblers has adequate market share to attain economic production on key components. The component suppliers provide these components to the assemblers who assembly the product with limited capital investment and with production operations which are relatively insensitive to the volume of production.

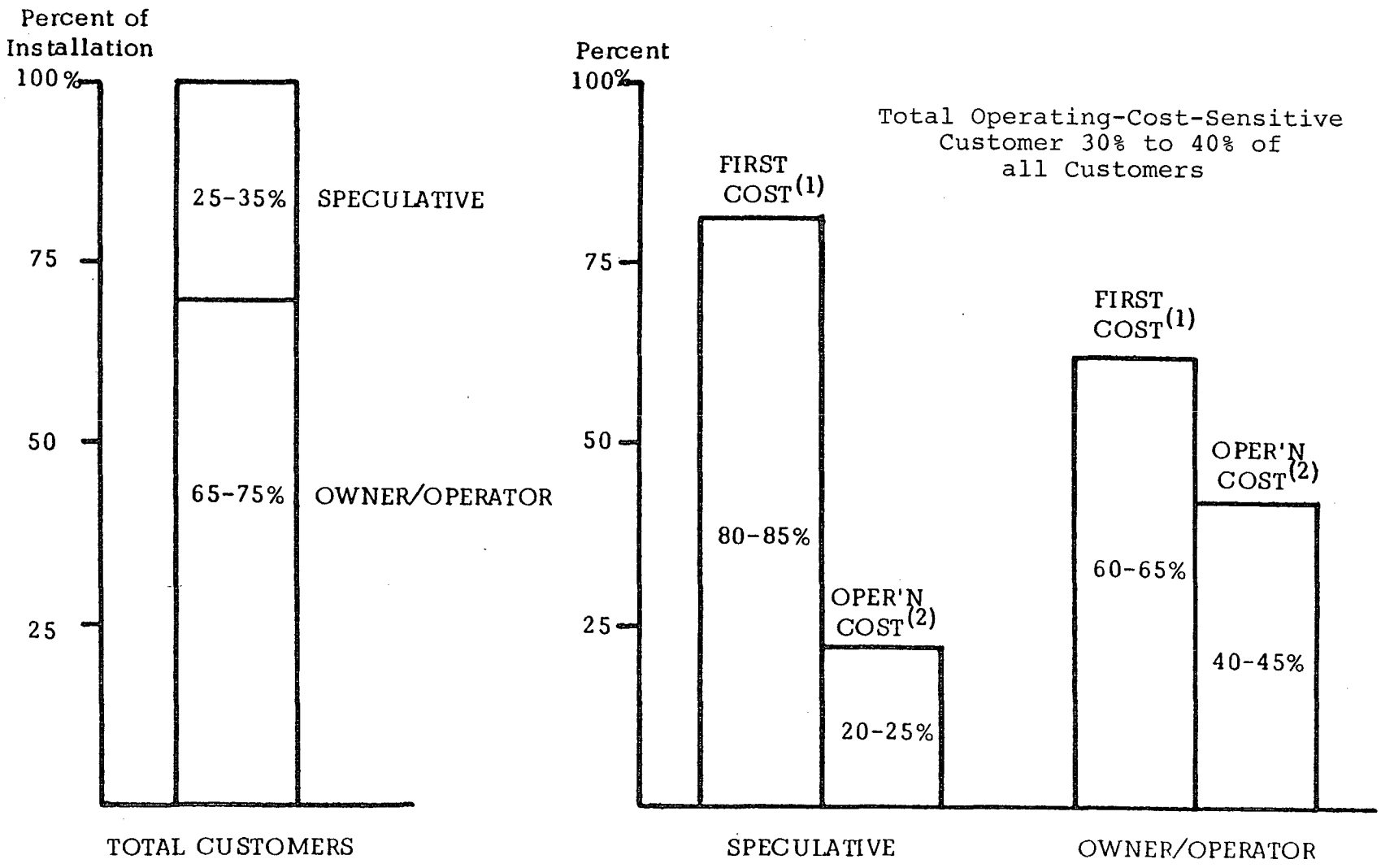
Because they have lower capital investment, lower administrative and sell burdens, and limited market objectives, the assemblers can meet or even beat the prices offered by the integrated manufacturers. The existence of the component suppliers enables the assemblers to easily enter the market and exert price pressure on the integrated manufacturers. Competition between the component suppliers and the internal manufacturing capability of the integrated manufacturers keeps a price and profit lid on the component suppliers.

5.2 Price Sensitivity

The HVAC Market is characterized as highly first-cost sensitive. From the viewpoint of the end user or purchaser, HVAC products are virtually undifferentiated. A 100-ton chiller provides the required 100 tons of cooling at the rating point. Since, historically, operating costs were usually not a significant factor, the selection of equipment was made on first cost, reputation for equipment reliability, and service. First cost as the principal purchasing criterion tends to put severe pressure on HVAC profitability. However, quantitative data as to the degree of price sensitivity in commercial HVAC customers were not publicly available until the Hill studies were completed under the HSPF programs.

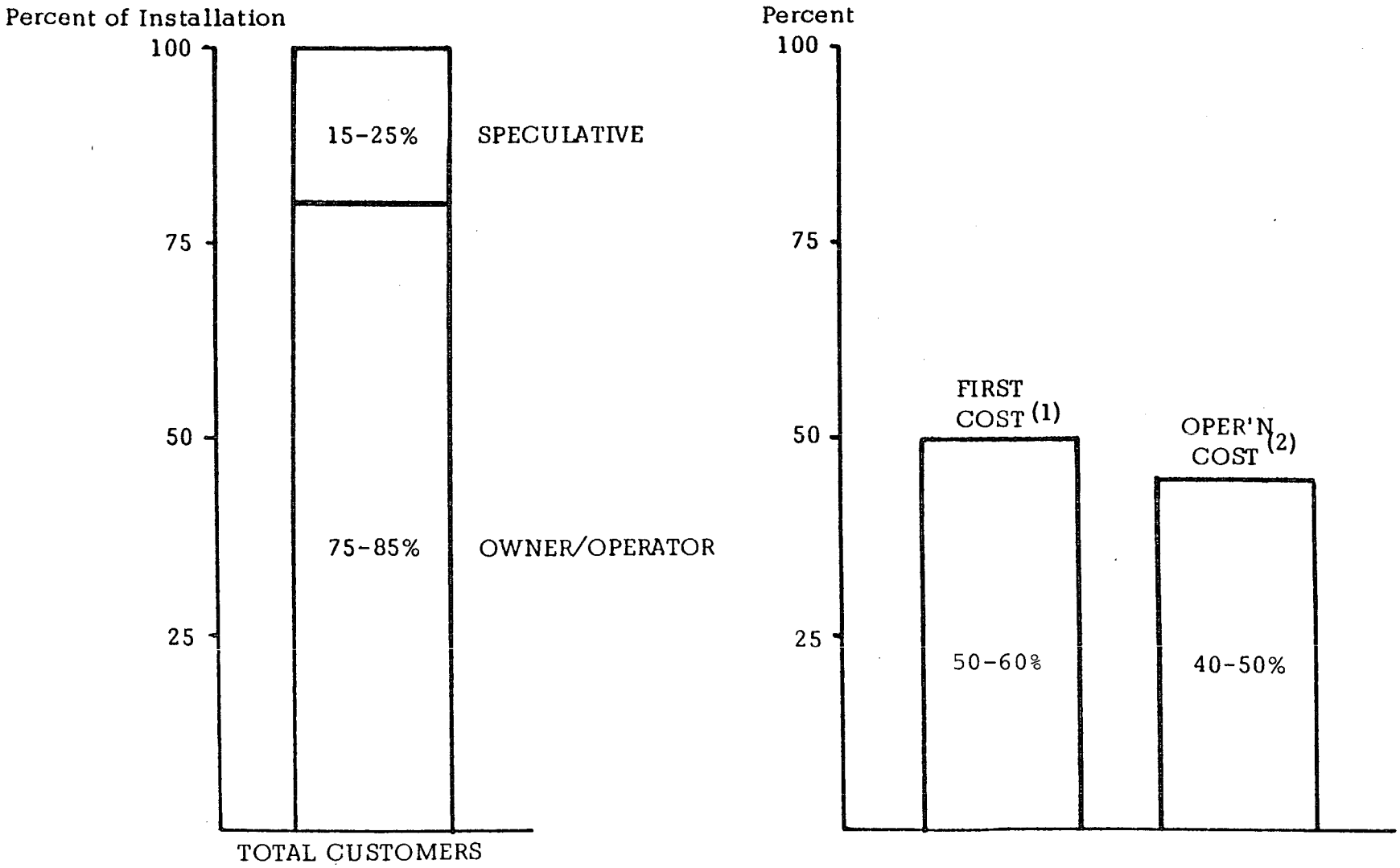
During the surveys with users, distributors and specifiers, two separate measures were taken regarding the first-cost price sensitivity of the market. The observations were conducted about 18 months apart (early 1976 and late 1977); first for the 7½-ton unitary market and later for the 50-ton and over market. Figures 5-4 and 5-5 summarize these data from the two surveys. Figure 5-4 shows that the attitudes toward operating cost (or life-cycle costing) as the principal criterion for purchasing differ between speculative builders and owner/operators. This difference was expected. In the 1976 survey, some 30% to 40% of all customers would consider operating costs. The data from 1977 (Figure 5-5) did not differentiate between customer type but showed that 40% to 50% of the customers would consider operating costs. According to projections developed by William E. Hill & Co., operating cost sensitivity increased through the 1970s and is expected to continue to grow. (See Figure 5-6).

In addition to knowing the proportion of the market sensitive to operating cost, it is necessary to know the acceptability of a given price premium and operating cost advantage. This information was also obtained by Hill and is shown in Figure 5-7. The market acceptance among those buyers who are sensitive to operating cost is a function of both the price premium and the operating cost advantage, as indicated by payback time. The payback time is simply the number of years of operating cost savings needed to equal the first-cost premium. Both parameters are used to specify market acceptability. Classical investment theory would suggest that only payback (ROI or DCF) should be adequate to specify market acceptability. In practice, however, there appears to be a psychological or practical first-cost barrier. The explicit limitation of first-cost premium indicates the importance of first cost as a purchasing criterion. The data in Figure 5-7 indicate that a piece of equipment that has a 30% price premium and a two-year payback would be attractive to only 32% of those who would consider operating costs. As noted previously, only about 50% of the market would consider operating costs. The impact of Figures 5-6 and 5-7 are combined and presented in Figure 5-8. The data in this last figure are applicable to any HVAC device.



(1) First cost sole or primary criterion.
(2) Operating cost an important criterion - not necessarily prime.
(3) Based on 1976 telephone survey data for the 7 1/2-50 ton unitary equipment market.
Source: William E. Hill & Co., Inc. estimates.

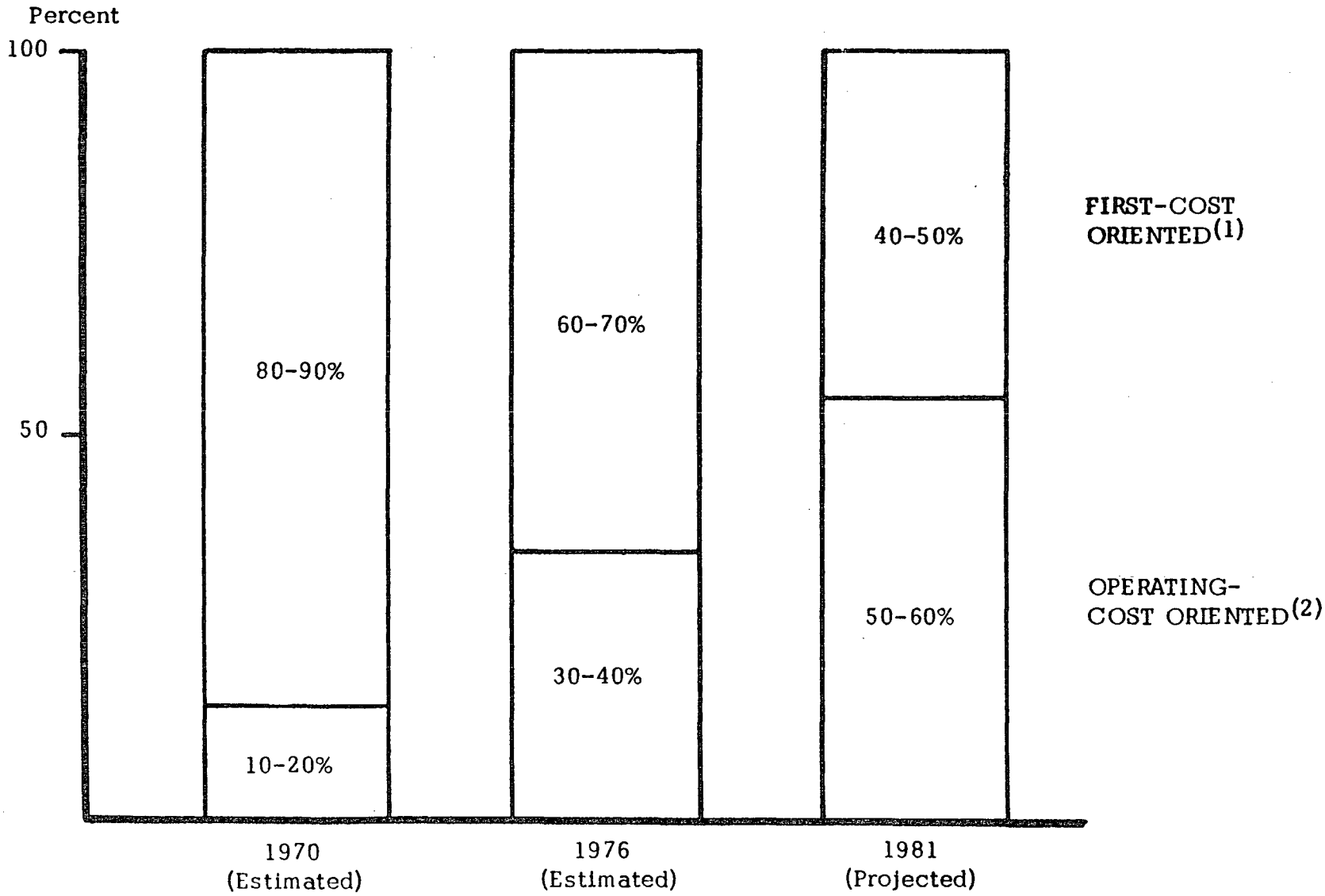
Fig. 5-4 Importance of First and Operating Costs by Customer Type(3)



(1) First cost sole or primary criterion.
(2) Operating cost are important criterion - not necessarily prime.
(3) Based on 1977 telephone survey data for the 50+ ton unitary and central built up equipment market.

Source: William E. Hill & Co., Inc. estimates.

Fig. 5-5 Importance of First and Operating Costs by Customer Type[3]



(1) First-costs are the sole or primary consideration.
(2) Operating costs are an important, not necessarily prime consideration.

Source: William E. Hill & Co. Inc.

Fig. 5-6 Estimated Cost-Orientation Among Commercial Air-Conditioning Customers (1970, 1976 Estimated; 1981 Projected)

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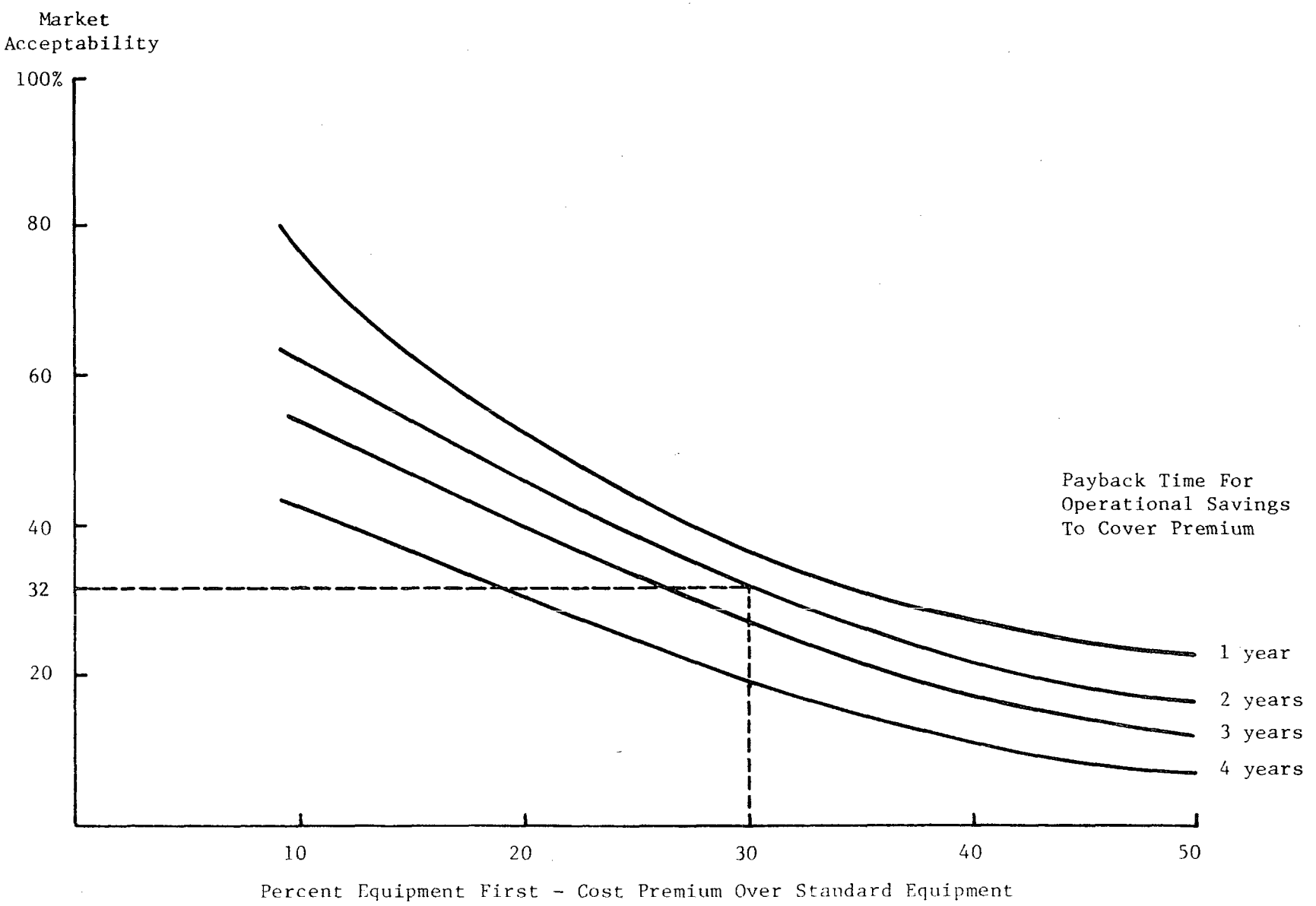

(1) Of total large tonnage equipment (unitary; built up) customers.
Source: W.E. Hill

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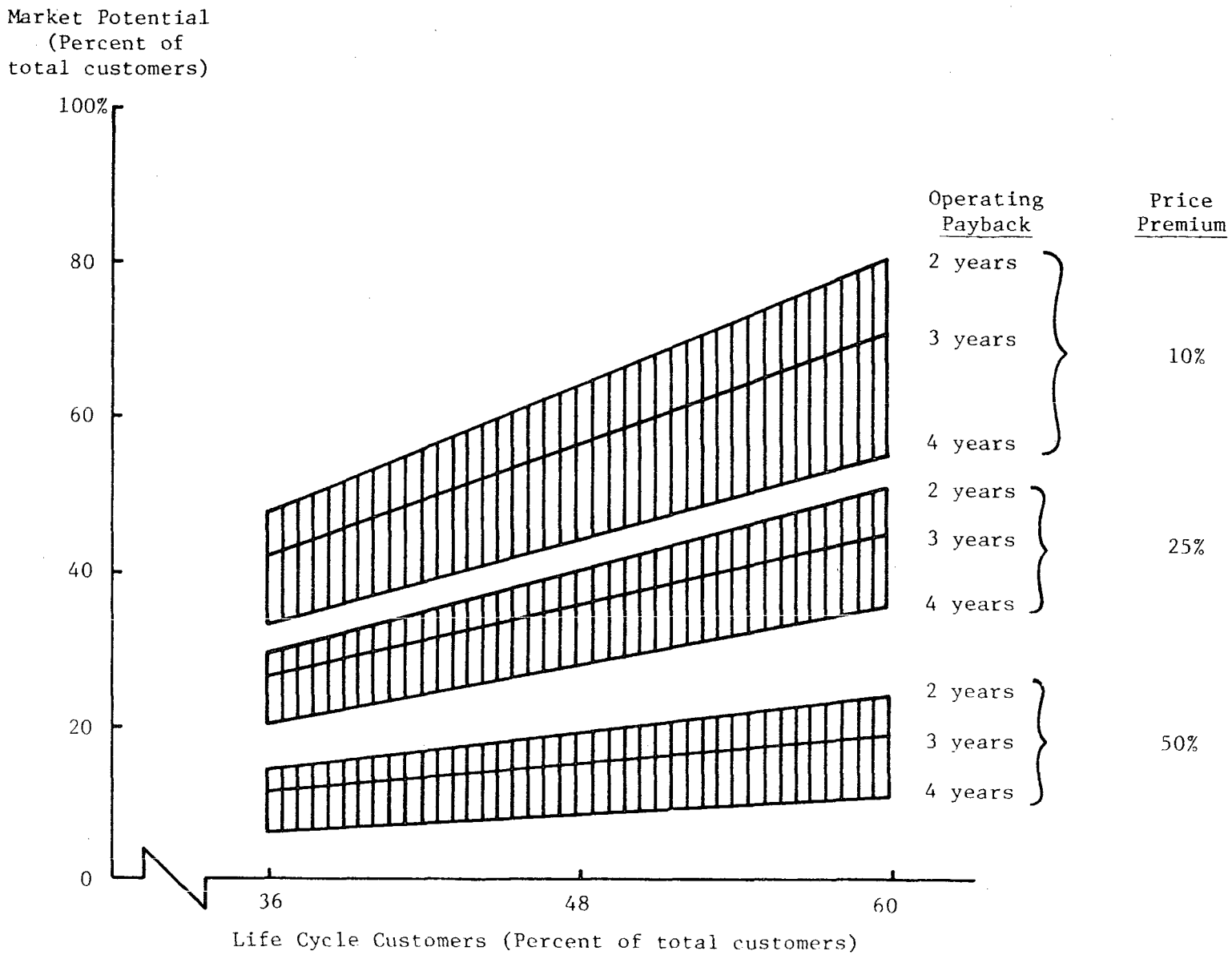
Fig. 5-7 Market Acceptance of Operating Cost Sensitive Customer to Price Premium and Payback - 1977 Survey(1)

CONSOLIDATED NATURAL GAS SERVICE COMPANY



Source: W.E. Hill

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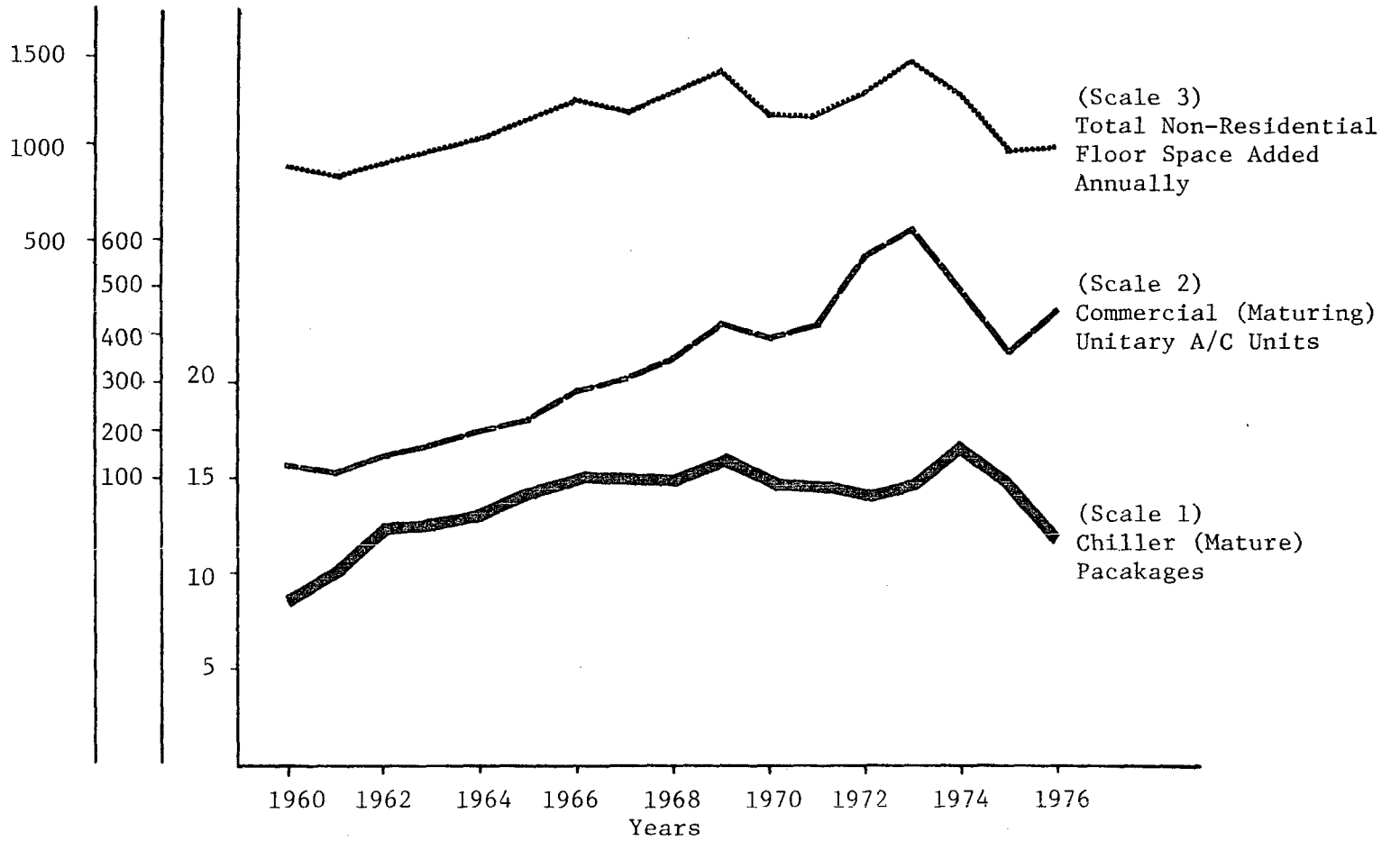
Fig. 5-8 Market Potential Among Premium/Payback Customers - Large Tonnage Equipment 1977 Survey

5.3 Market Potential

Projecting the market potential of any new product is a complicated task requiring a thorough evaluation of both the market and the product. Actual sales penetration into the projected market potential will be affected by the commercialization path and the marketing approach. Additional factors influencing HVAC equipment are the general business environment, market attitudes towards life-cycle costing, fuel costs, fuel availability and public policy.

The market potential projection started with the historic sales data for HVAC products. To facilitate the market potential projections, the market data were segmented by geographic regions, unit size, unit type, type of construction and building type. The four basic geographic regions considered were too broad to have a climatic factor for equipment performance. Unit sizes were considered from 5 tons and up. The two basic equipment types were unitary and built-up. The type of construction was segmented to consider the demand factors imposed by new construction, major remodeling, replacement and energy retrofits. Two broad classes of buildings existed, residential and nonresidential. Since the product concept was directed toward the large tonnage market, only the nonresidential building type was considered.

Four basic types of construction impacted upon the market potential analysis by effecting the basic demand for HVAC equipment. New construction is very sensitive to the general economy and is very cyclic. For example, as shown in Figure 5-9, a building boom in the early 1970s created a peak demand for HVAC equipment. Only now, six to seven years later, is that demand again equaled by current shipments. Large buildings have a long planning and construction cycle and, therefore, fairly accurate projections of new construction for one to two years into the future are available. These estimates of new construction by building type and geographic area were used to define the market potential.



Source: ARI; U.S. Department of Commerce

Fig. 5-9 Comparison of New Nonresidential Construction and Commercial Unitary and Chiller Shipments, 1960 - 1976

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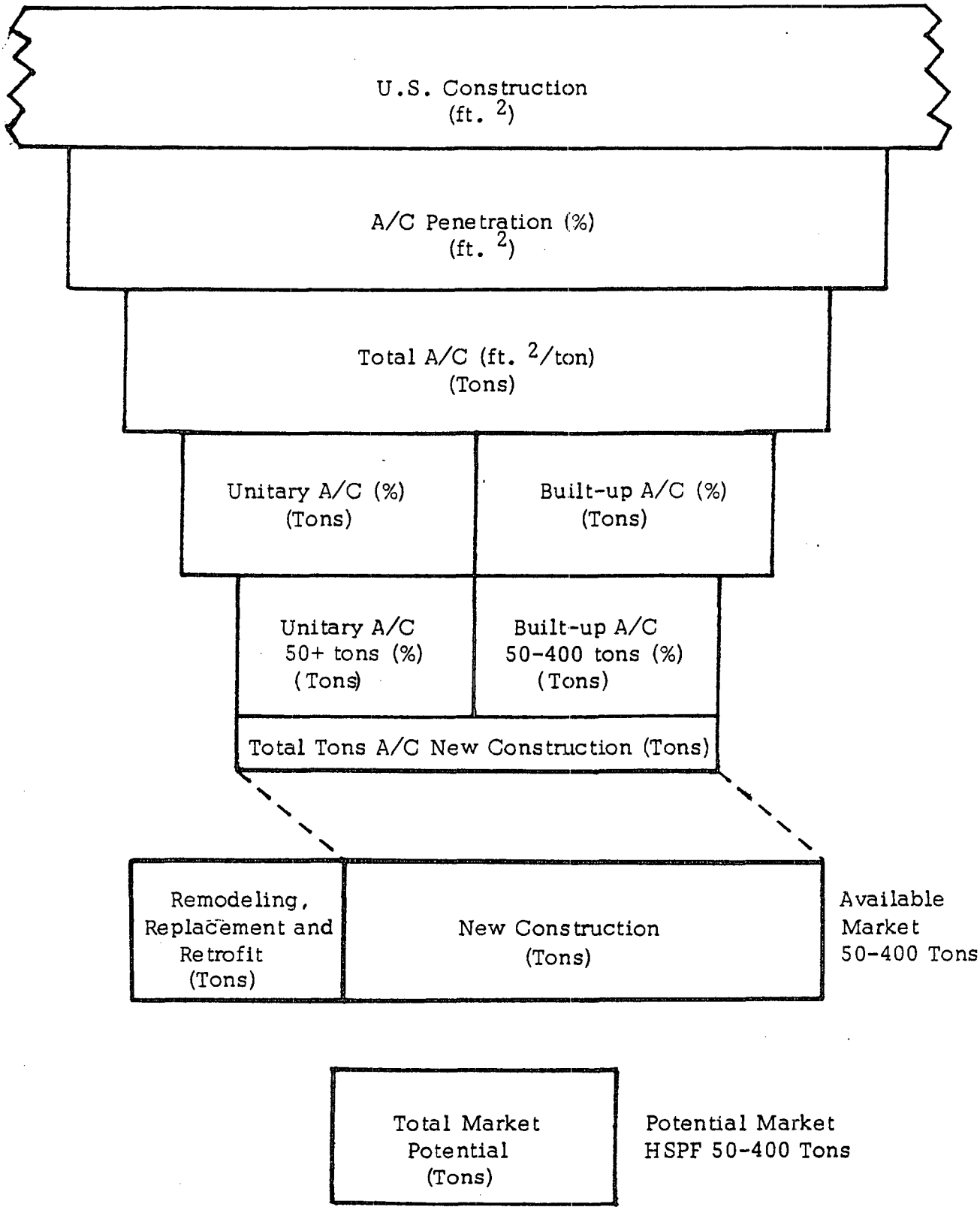
The nonresidential building type classification was further segmented into five major subtypes as follows:

- Commercial - includes retail stores, shopping centers, restaurants and offices, commercial warehouses, garages and other miscellaneous commercial buildings.
- Industrial - includes plants, manufacturer warehouses and labs, and other industrial buildings.
- Institutional - includes schools, colleges, hospital medical clinics and various types of public buildings.
- Apartments - includes low-rise and high-rise apartment buildings.
- Hotels/Motels - includes high- and low-rise hotels, attached restaurants, meeting rooms, etc.

The commercial classification was further segmented to consider high-rise and low-rise banks and offices and all other commercial buildings. The institutional classification was segmented to include education and science buildings, hospitals and health treatment centers, public buildings, religious buildings, and all others.

The methodology for combining the historic data by the various market segments to yield a market potential forecast for the LSM is illustrated in Figure 5-10. The approach for the 7½-ton unit was similar. The starting point for the forecast was the projection of commercial construction activity. Although reasonably firm projections of new construction activity for two to three years are available, long-term projections of annual activity are subject to change due to general economic conditions. The wide swings in construction activity are shown in Figure 5-9. Over a period of 5 to 10 years, construction activity would show a growth trend ranging from a slight decline to a modest growth; the growth rate selected by Hill is 6.9%, as shown in Table 5-1. The other factors affecting the projections are also shown in Table 5-1. The segmentation by unit size, unitary and built-up,



Source: William E. Hill & Co. estimates

Fig. 5-10 Analysis of the Potential Market for the HSPF, 50-400 Tons, 1981

TABLE 5-1

ASSUMPTIONS - KEY PARAMETERS AIR CONDITIONING
MARKET POTENTIAL FOR THE HSPF 50-400 TONS
United States

Parameter	1976 (E)	1981 (P)
U.S. Construction (1) (MM Sq. Ft.)	1,353	1,889
A/C Penetrations (2) (%)	72.2%	76.2%
A/C Ft^2/Ton (2)	338	388
Unitary A/C (2) (%)	46.1 (4)	44.5 (4)
Built-up A/C (2) (%)	35.3 (4)	35.7 (4)
Unitary A/C 50+ Tons (2) (%)	7.3%	7.65%
Built-up A/C 50-400 Tons (2) (%)	48.0%	48.0
Replacement, Remodeling & Retrofit Unitary (2) (% of total market unitary)	30%	30%
Replacement, Remodeling & Retrofit Built-up (2) (% of total market built-up)	20%	20%

(1) William E. Hill & Company estimates based on F. W. Dodge Data.
(2) William E. Hill & Company estimates.
(3) Thousands of tons.
(4) Percentage unitary and built-up doesn't equal 100 percent since unitary percentage is reduced for the apartment and nonhousekeeping residential segments by 80 percent to remove the tons of equipment that are through the wall units and small unitary.

(E) Estimated.
(P) Projected.

and new construction and remodeling were taken from historic data. Figures 5-11 and 5-12 present the unit size distribution for central built-up equipment and commercial unitary equipment, respectively.

The culmination of the projections is shown in Tables 5-2 and 5-3. The market projection in Table 5-2 is for both unitary and built-up equipment. Built-up potential was over four times that of the unitary. A recommendation was therefore made that the LSM be designed as a built-up system. The data in Table 5-3 are for unitary equipment in the North Central region only.

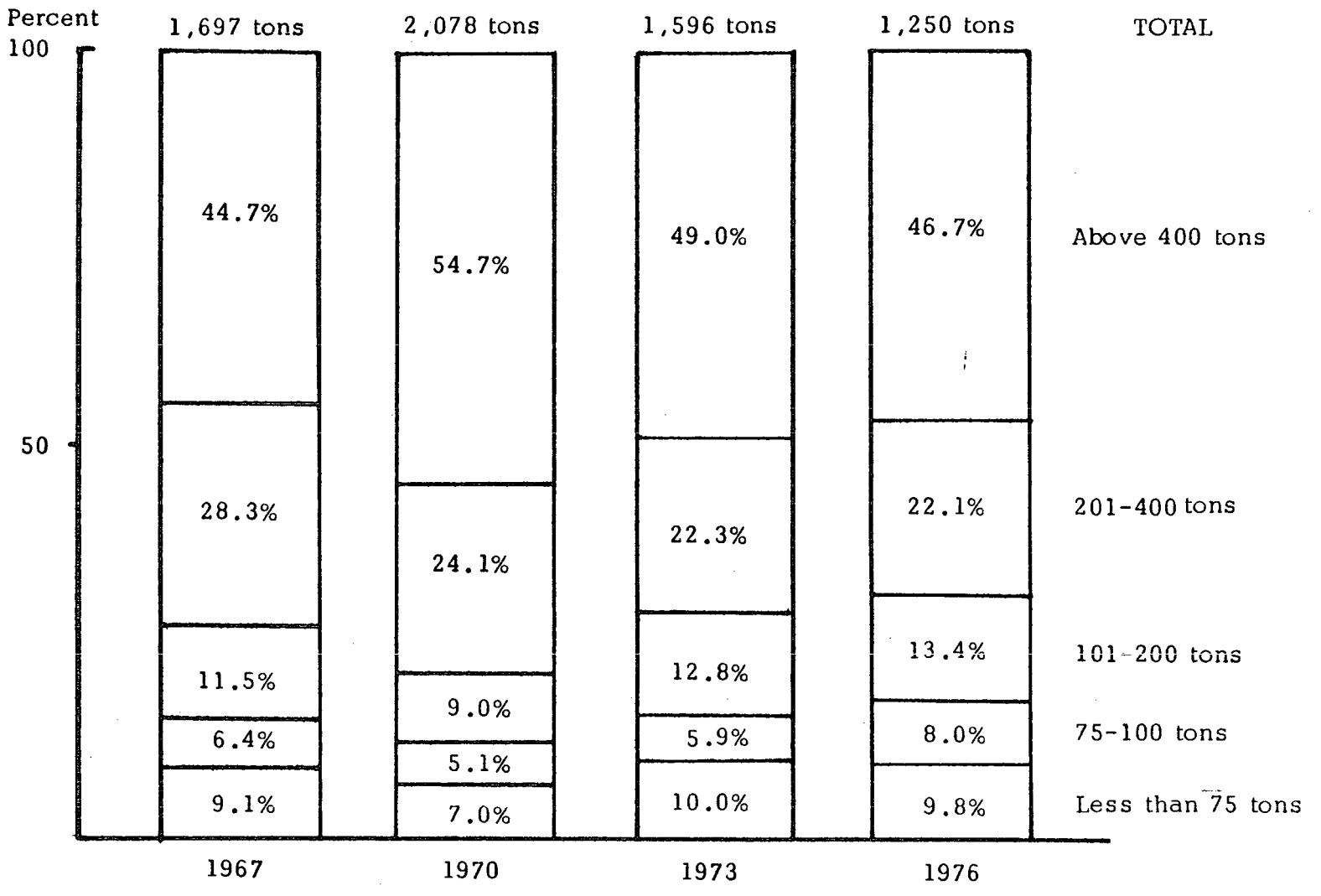
5.4 Key Observations

During the course of the market potential studies, a number of qualitative observations were made. The key observations made by Hill are presented below.

A. Equipment selection in the nonresidential air conditioning market was heavily influenced by the complex interaction between customers, system designers, installers, servicers, distributors and manufacturers.

- The time span of building ownership and whether the system purchaser would also be the user strongly affected the selection of equipment. Speculative customers, about 30% of total customers, tended to be very sensitive to equipment first costs and generally purchased only on a first-cost basis. Nonspeculative customers, particularly those who would also be system users, were more concerned with system operating costs and features.

- System designers strongly influenced or were totally responsible for equipment selection. In-house and consulting engineers were most often involved in large or complex projects, particularly for large national companies and institutional applications. These engineers designed almost all of the built-up systems and much of the large custom unitary systems. Mechanical and air conditioning contractors designed a high percentage of unitary applications using 7½-ton to 50-ton units, particularly in the



Source: William E. Hill & Company estimates based on ARI data.

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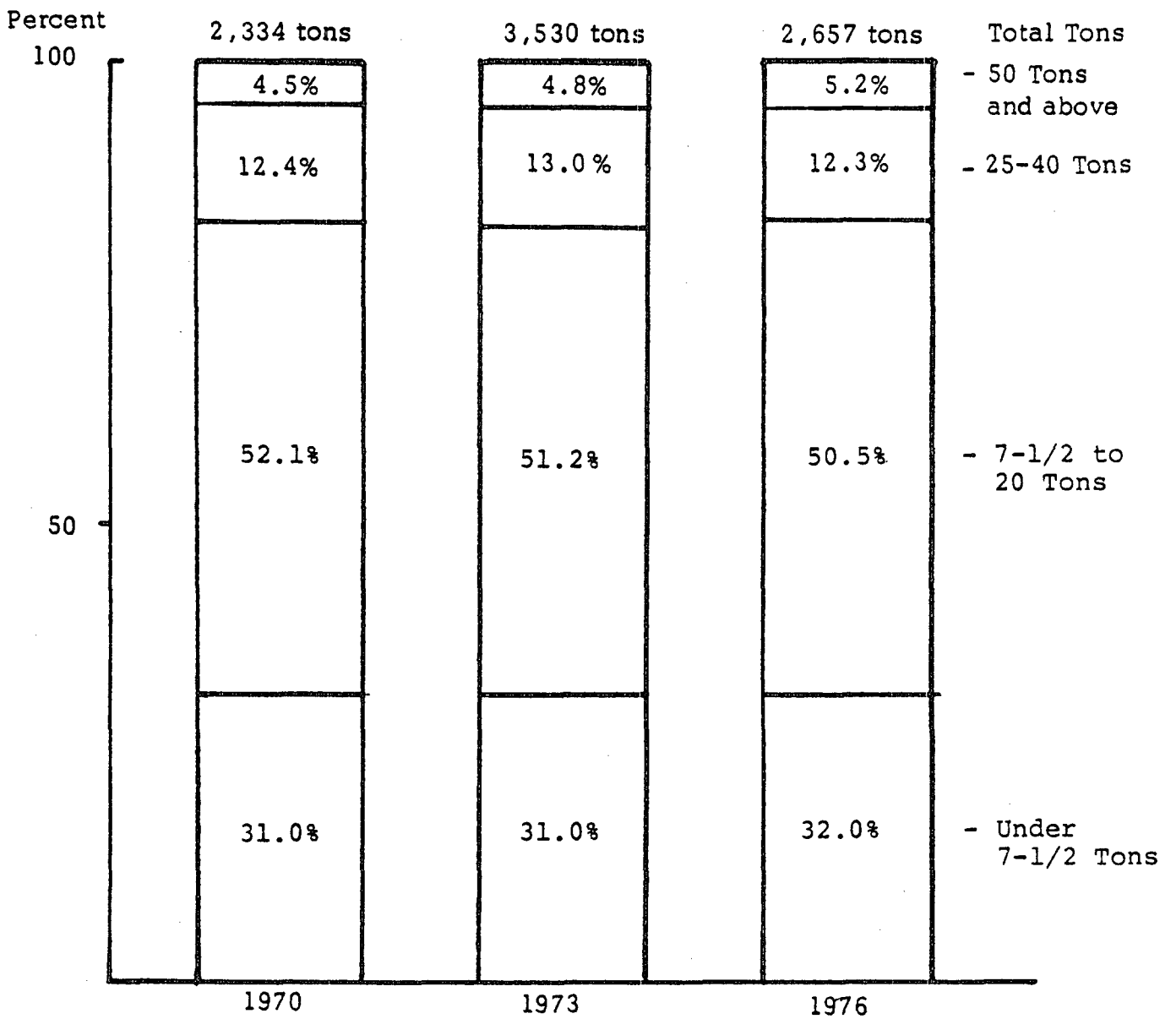
Fig. 5-11 Distribution of Central Built-up Equipment by Size
Total U.S. (thousands of tons)



Source: William E. Hill & Company estimates based on ARI data.

Fig. 5-12 Distribution of Commercial Unitary Equipment Shipments by Nominal Size Range, Total U.S. (thousands of tons)

TABLE 5-2

PROJECTED NATIONAL MARKET FOR 50 TO 400 TON EQUIPMENT IN 1981

Building Type Market Segment	1981 Market (thousands of tons)		
	Unitary	Built-up	Total
Commercial			
Banks and Offices			
High Rise	0.0	84.0	84.0
Low Rise	24.1	113.3	142.4
Stores and Other	60.4	102.0	162.4
Industrial	28.5	90.6	119.1
Institutional			
Education and Science	29.9	65.9	105.8
Hospital and Healt Treatment	11.2	110.1	121.3
Public Buildings	7.2	35.8	43.0
Religious	1.9	9.0	10.9
Other	8.4	42.6	51.0
Apartments	2.1	108.4	110.5
Non-Housekeeping Residential	1.0	32.1	42.1
Total Available Market	179.9	793.8	973.7

Source: W. E. Hill

TABLE 5-3

PROJECTED NORTH CENTRAL REGION MARKET FOR 7½- TO 50-TON UNITARY EQUIPMENT IN 1981

Building Type Market Segment	1981 Market (thousands of tons)
Commercial	
Banks and Offices	
High Rise	0.0
Low Rise	27.4
Stores and Other	73.0
Industrial	42.5
Institutional	
Education and Science	22.3
Hospital and Health Treatment	11.8
Public Buildings	6.9
Religious	1.1
Other	16.3
Apartments	0.0
Non-Housekeeping Residential	5.9
New Construction	207.2
Remodeling, Replacement and Retrofit	105
Total Market Available	312.2

Source: W. E. Hill

commercial market and replacements. Contractor/dealers generally did little designing, other than sizing units for residential and small commercial applications.

- Installing contractors also influenced equipment selection, and tended to specialize in certain markets or specific applications. Mechanical contractors did most of the large plan and specifications work, almost all of the built-up systems, and generally installed plumbing and process piping in addition to HVAC equipment. Air conditioning contractors did little or no piping work and tended to concentrate on commercial and industrial negotiated work. Both would install and service several brands of equipment. Contractor/dealers generally handled one brand of equipment and tended to emphasize residential and small commercial applications.

- The availability and quality of service/maintenance was an important selection factor. Often different contractors were involved in the installation and service of a system. While all installing contractors provided service during warranty, service generally represented a small proportion of mechanical contractors' business. Service was very important to the contractor/dealers since the installing contractor and service contractor were often different parties.

B. Market characteristics, functional requirements and equipment evaluation processes changed in response to key external and industry trends. These key trends impacted the need, use and selection of space conditioning equipment.

- Functional requirements were impacted by dramatic shifts in the energy situation and changes in building design and construction practices. The energy situation significantly affected space conditioning requirements and the methods employed. Energy-efficient buildings could provide energy-use savings of 15% to 20% in most cases. System efficiencies

and flexiblity in terms of control and capabilities became increasingly important. It was likely that less capacity would be required for a given conditioned space which would affect equipment size and other features.

- Many large-tonnage building applications had minimal heating requirements and/or required simultaneous heating and cooling for different zones. The importance of such applications was expected to increase with improvements in energy-efficient building design and construction practices. Thus, in many cases, heating became a secondary consideration which could often be handled by small auxiliary systems used primarily for morning building start-up and for very cold weather. Use of condenser heat exhaust was one such auxiliary approach which was gaining rapid acceptance.

- Although first costs were important, rising energy and service costs increased the importance of operating costs and system efficiencies. To be acceptable, a premium-priced unit had to provide significant compensating advantages in terms of operating costs, maintenance or other characteristics. Approximately 50% of the large-tonnage market considered operating costs to be extremely important, but resistance existed to very large first-cost premiums or long payback periods.

- Total owning and operating costs were becoming more widely used in equipment selection decisions. As operating costs increased, greater sophistication was being applied to the analysis of functional requirements, total energy used and equipment selection to insure that all relevant cost and use factors were given proper consideration.

- Energy retrofit activity was increasing rapidly to replace older, energy-inefficient systems. In addition to new, more energy-efficiency equipment, a great deal of the retrofit activity

involved peripheral systems such as automated controls and variable air-volume terminals.

C. Rising energy and service costs recently began to increase the importance of operating costs, and the analysis of total owning and operating costs in the equipment selection process. First-cost premiums were becoming more acceptable, however, if the energy-use efficiencies of the unit resulted in significantly reduced operating costs.

D. Equipment reliability and the availability of service and parts were extremely important equipment selection considerations. In many applications, particularly in the commercial segment, air conditioning was considered to be a necessity for conducting business. System designers often built redundancy into the system by using several smaller units rather than a single larger unit.

E. Historically, the air conditioning market was slow to accept new products and technologies, unless they presented significant advantages and satisfied the prevailing market characteristics and requirements. Most of the innovations in the last five to ten years involved controls and peripheral equipment (i.e., variable-air-volume), rather than radically new components or technologies in the basic refrigeration equipment. New products only achieved rapid acceptance when they satisfied the prevailing market characteristics and requirements. Examples were the lower first-costs of combination heat/cool units in the 1960s; variable-air-volume and economizer controls in the early-to-mid 1970s, and electric heat pumps in areas where natural gas was not available.

F. The nature of the large-tonnage equipment markets suggested that the marketer(s) should have had a strong current position in the market. An established position (and good technical reputation) in the marketplace was desirable, perhaps necessary, due to the uniqueness of the HSPF. Starting from scratch probably would have been prohibitively expensive and unlikely to succeed in the face of strongly entrenched competition. Most major suppliers of large-tonnage equipment did not produce steam generators, however, so that marketing the HSPF could have increased their total revenues per installation.

G. The market perception with regard to the future availability of natural gas was perhaps the most important, basic issue affecting the market potential for the HSPF concept. Tremendous confusion and skepticism about the energy situation (particularly natural gas) existed in the marketplace. The extent of this problem had to be recognized and effectively dealt with in the HSPF commercialization program.

6.0 LARGE SIZE MODEL TECHNICAL EFFORT

6.1 Introduction to the Large Size Model

The Large Size Model (LSM) product has been identified in Section 4.0 as a heating, ventilating and air-conditioning (HVAC) system for large commercial applications. The limiting definition of commercial applications is the size of the unit. Typically, the commercial market is defined as being served with units in excess of 7½ tons of capacity. Although equipment in the 3- to 5-ton class is frequently used in commercial applications, it is usually considered as residential equipment employed in a commercial application. HVAC products for commercial applications are segmented further as either unitary or built-up systems. The unitary systems are systems in which all the product functions of heating, ventilating, air conditioning and by-product heat rejection are fully assembled and packaged in the factory. The 7½-ton HSPF unit, for example, was designed as a unitary system. Built-up systems are systems in which each of the functions is fabricated separately at the factory, often by different manufacturers, and assembled at the site.

The specific market and corresponding product area that is addressed in this technical effort is that defined as large built-up commercial. This area encompasses built-up equipment of 50- to 400-tons cooling capacity. This market segment had been identified for further exploration as a part of the program redirection decision. The selection of the specific market segment of 50- to 400-ton built-up systems was the result of the LSM Market Potential Study (see Section 7.0 and Appendix Volume VI). The very large tonnage (over 400 tons) built-up systems were dismissed for the time being due to an apparent lack of adequate market potential.

The LSM product concept examined in detail was a nominal 120-ton cooling capacity HSPF built-up system. This system was to be technically representative of an HSPF product line serving the built-up system market in the 50- to 400-ton range.

The basic technology concept developed for the LSM is shown in schematic form in Figure 6-1. The technology concept for the LSM was intended to be based as far as possible on all of the HSPF features that had been incorporated in

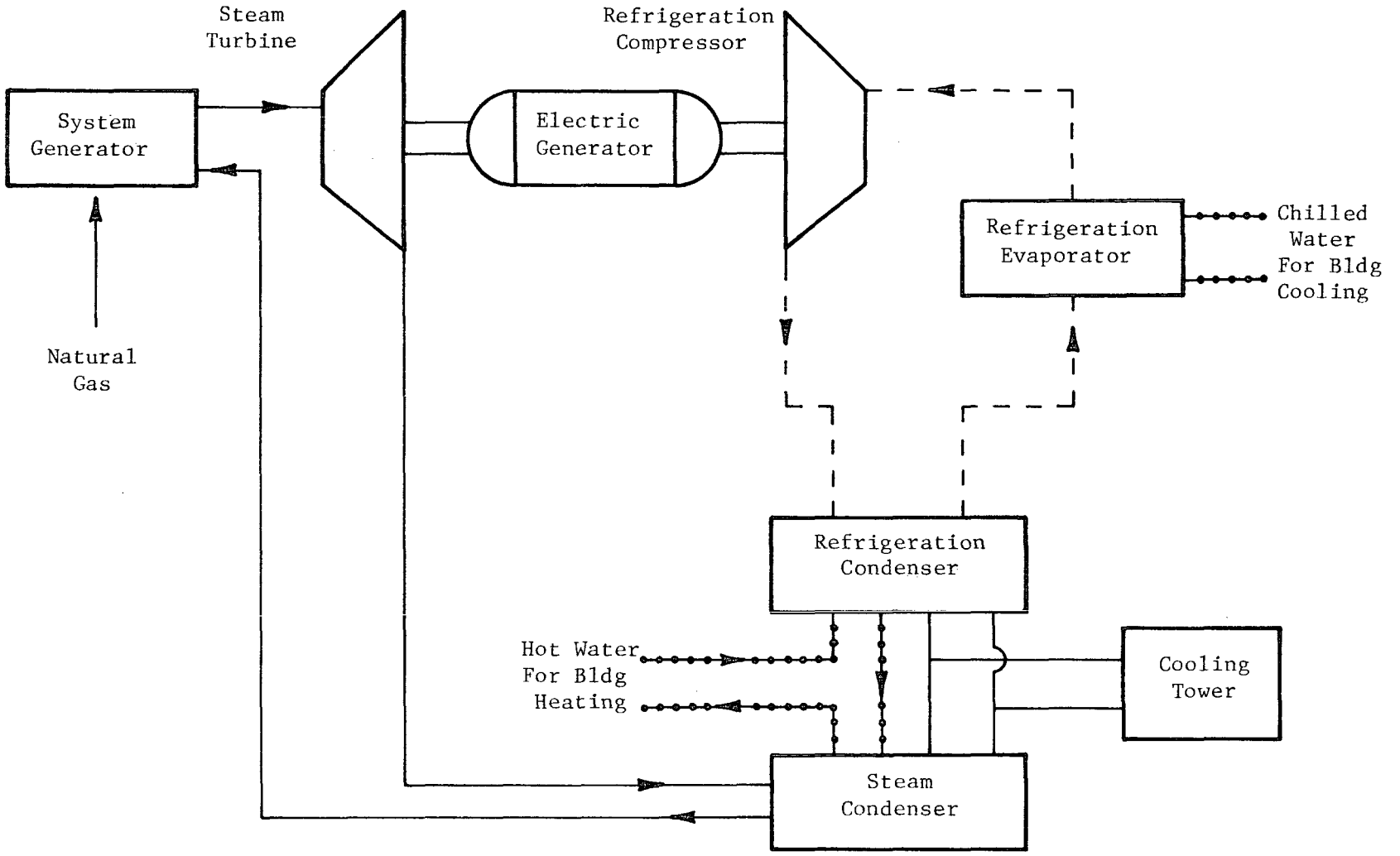


Fig. 6-1 LSM Built-Up System Concept Schematic

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the 7½-ton unit. Meeting this objective was found to be impractical. Specifically, all forms of heat pumping and speed modulation were eliminated entirely in order to control system cost. The LSM HSPF technology concept continued to incorporate the use of a high-performance, steam-turbine prime mover system driving a standard-refrigerant, heat pump system plus the auxiliary power functions. The steam-turbine prime mover system would be fueled by natural gas as in the basic concept, but could also be fueld by other fossil fuels such as oil, coal and coal-derived fuels or by other combustables. The LSM design that emerged, the corresponding performance, technical features and cost objectives are fully described in Appendix Volume V.

The principal result of the technical effort was a design that provided heating, cooling and electricity to a building as shown schematically in Figure 6-2. An operating-cost/first-cost trade-off study indicated that the steam turbine prime mover should be sized to be able to supply half of the power required to provide the maximum cooling load and that the steam-turbine prime mover should be base-loaded. Thus, at low cooling loads, the LSM would be able to provide electricity to offset the normal building electrical load. No provision was made for export of electricity beyond the building. Conversely, at high cooling loads, the LSM would require electric power from the local electric utility. The power balancing aspect is shown in Figure 6-3. During the heating season, the generation of electric power for the building would offer a significant operating economic advantage. During the cooling season, the generation of electric power also would offer an economic advantage, although not nearly as great a one as during the heating season.

These findings may be somewhat specific to the simulated application employed in the modeling effort.

6.2 System Design Analysis

The System Design Analysis evolved in three phases, occurring chronologically in sequence. The initial system model consisted of a simple steam turbine prime-mover-driven internal source heat pump, without either an economizer mode for the building or electric power generation capability. The steam turbine prime mover's rejected heat was utilized, along with the pumped heat, by the building, when a demand existed for space heat. When providing cooling

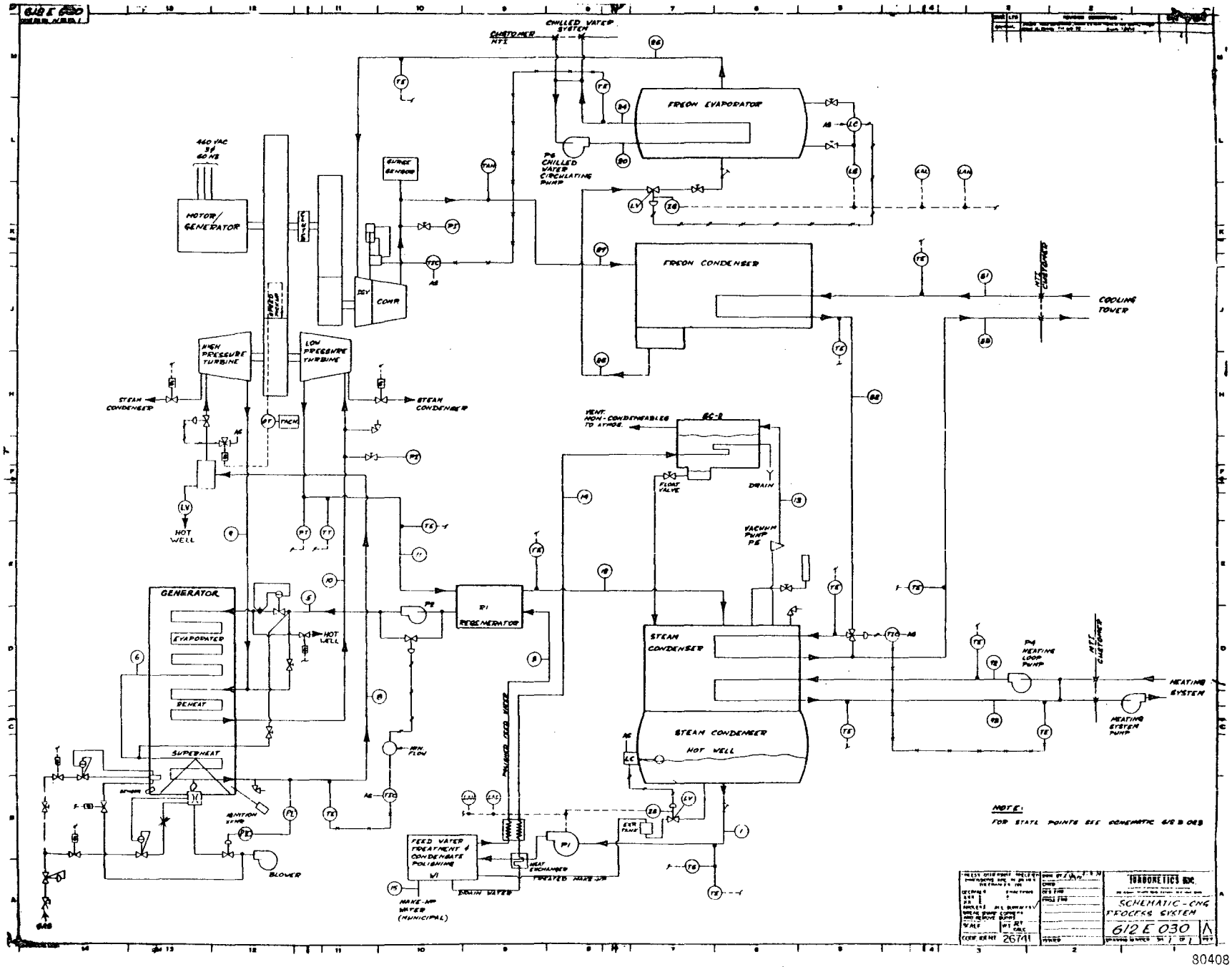
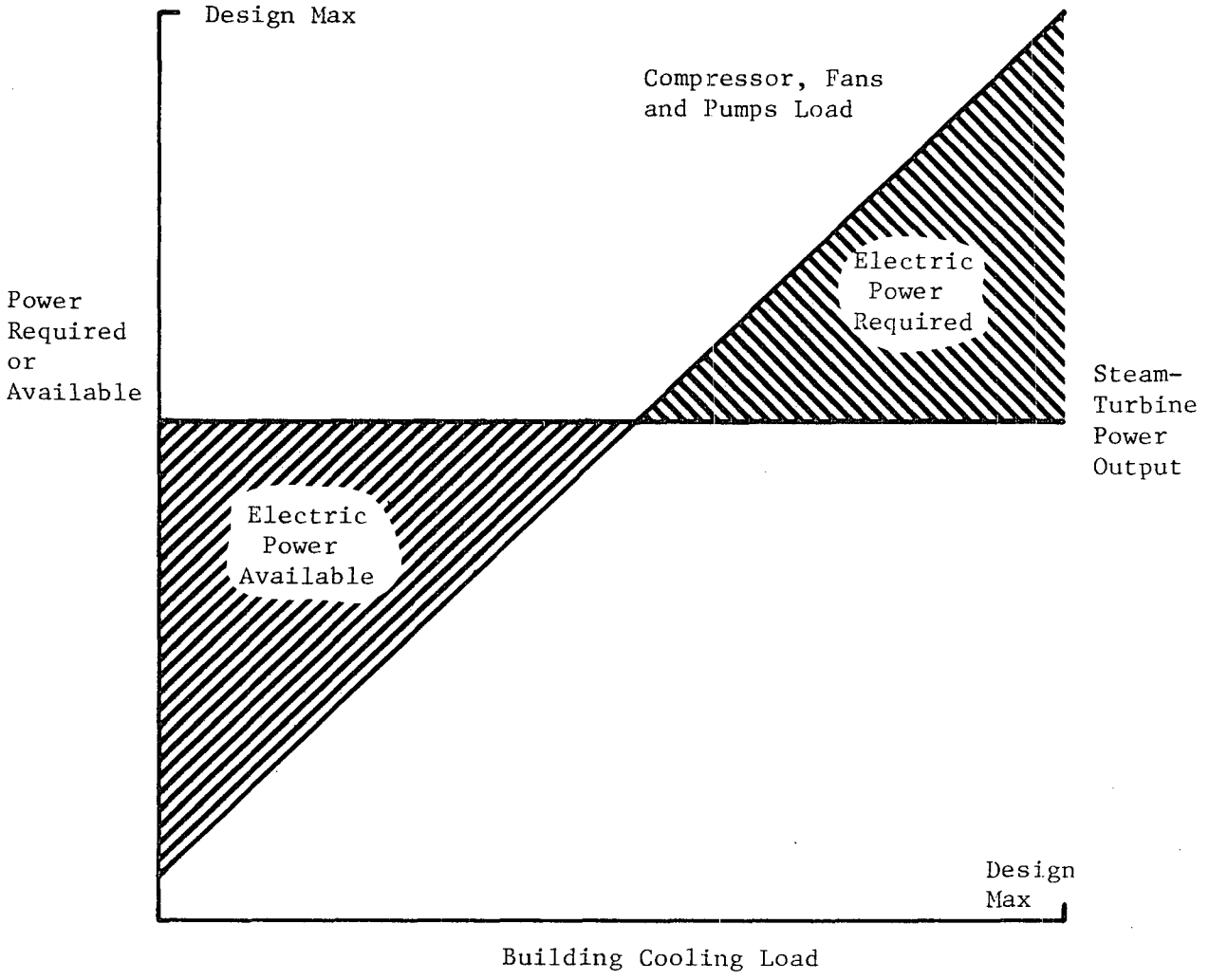


Fig. 6-2 Large Size Model 125-ton System Schematic



Fig, 6-3 Illustration of Power Availability and Power Requirements for the Proposed LSM Product Design

service, the heat rejected from the building's cooling load was combined with the heat rejected from the steam turbine prime mover system (driver) and dissipated through a cooling tower. To meet space heating loads, the model assumed a demand for 115 to 180°F hot water for baseboard radiation so that the system would be compatible with standard distribution system practice. The model included all aspects of the HVAC system including the electrical power requirements of the air and water distribution equipment. The model was used to generate the annual energy usage and cost in an office application.

The modeling results quickly provided the following information and insights. 1) Large amounts of electrical power were required for the air and water distribution equipment. 2) A very small percentage of the total heating requirement occurred coincidentally with a cooling load. 3) When coincident heating and cooling loads did occur, the rejected heat from the steam-turbine power cycle at idling conditions could supply virtually the entire heating load.

The recommendations from the first design analysis model were as follows:

- Power generation should be included in the system concept in order to broaden the base for developing significant cost and energy savings
- Other heat pumping sources should be investigated for inclusion in the system in order to increase the amount of heat available for heat pumping when the internal (cooling load) source is small or nonexistent.

The objective of the system design analysis' second phase was to improve the LSM's overall performance by incorporating the above recommendations plus any others that might have developed through this phase of study. The system improvement effort focused upon system marketability as influenced by operating (energy) cost savings and capital costs of the system. According to the W.E. Hill LSM market study (Appendix Volume VI), conducted early in the LSM program, the capital cost of the system should not be more than 50% greater than the cost of conventional primary heating/cooling equipment.

In addition, the operating cost savings should offer the user a two- to three-year return on the incremental investment (cost premium). Thus, the objective was to maximize the potential marketability of the LSM by maximizing energy cost savings while minimizing LSM system cost using the two- to three-year payback period as the economic criterion. Figure 6-4 illustrates the marketability constraints on the system.

The system model was restructured to include all the internal and external heat sources. The system also included electric power generation capability. Since the system was still tailored to include retrofit markets using baseboard radiation with scheduled water temperatures reaching a maximum of 180°F at 0°F ambient, external source heat pumping required a two-stage heat pump. This condition was later relaxed with the realization that the retrofit market would be more difficult to serve.

The model permitted an assessment of the cost effectiveness of each heat pumping source as well as the effectiveness of different sized steam turbine drivers operating in two different control modes - base-loaded and modulated. The system could be analyzed with and without electrical power generation capabilities. In the base-loaded control mode, the driver was maintained at full load and generated electricity whenever the compressor could not use the driver's total output. If the steam turbine driver had inadequate power to meet the compressor's needs, the electric motor/generator assisted in driving the compressor. In the modulated control mode, the total heat rejected from the refrigerant condenser and steam condenser was matched to the corresponding heating load of the building by adjusting the driver output and the heat pump's evaporator load.

A parametric study was performed using the new system model. The results showed that the system could produce energy cost savings compared to a conventional system consisting of a gas-fired steam generator for heating and a 150-horsepower, electric-motor-driven chiller for cooling. The greatest operating cost savings were obtained from the base-loaded mode of control where, at times, excess electrical power was generated for use elsewhere in the office building. The excess power offset electrical power which would otherwise be purchased from the electric grid.

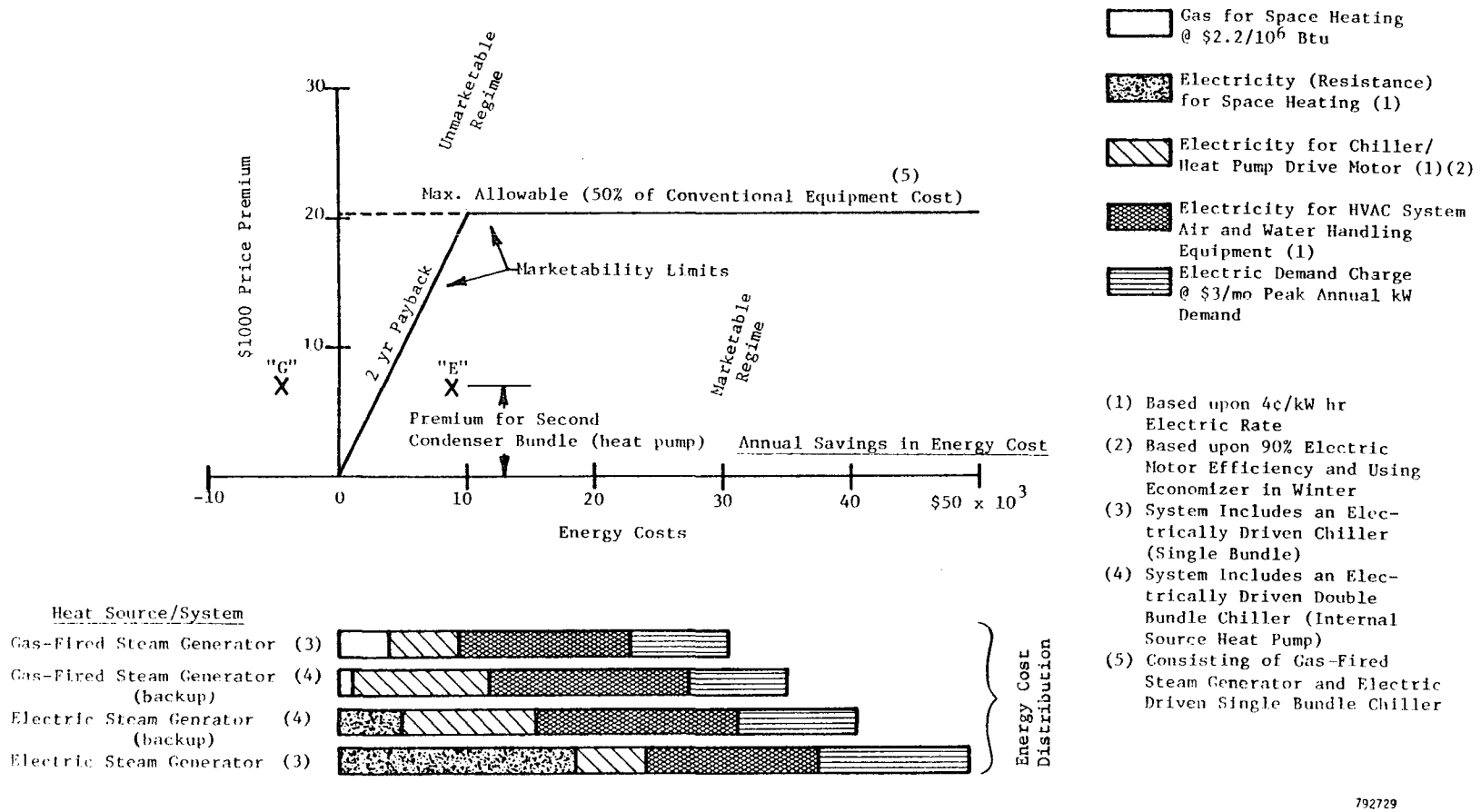


Fig. 6-4 Economic/Marketability Constraints on the HSPF Heat Pump System along with the Distribution of Annual Energy Costs for HVAC Related Needs Using Alternative Systems in an Office Building Application

The office building application was analyzed using driver sizes ranging from 90 hp to 330 hp. This analysis indicated that the 90-hp size offered the highest energy cost saving per driver horsepower. In the base-loaded control mode, the 90-hp driver size also has an equipment cost advantage over larger sizes, thus offering the largest probability of meeting the first-cost premium marketability constraint.

Annual energy cost sensitivities to the various heat pump sources were studied for the 90-hp, gas-fired, steam turbine system and for the conventional electric-motor-driven system. For an office building application where gas is available, the results of this study showed that heat pumping was not cost effective, even if the system used the most effective heat pumping source. With the steam-turbine-driven system, the energy costs were insensitive to heat pumping sources since nearly all the building's heating requirement was supplied from the driver's rejected heat. The energy costs were the same without heat pumping as they were when heat pumping was included. Therefore, no financial return could be generated on the $7000 extra investment for the heat pump. Accordingly, heat pumping with a base-loaded, gas-fired, steam turbine driver was not economically justifiable.

Energy costs are very sensitive to the heat pump source when an electric motor drives the heat pump. Figure 6-4 illustrates the distribution of energy costs for four types of electric-motor-driven systems. All four combinations, heat pumping or not heating pumping and heating electrically or with gas, are shown. The cost of electricity far exceeds the cost of gas in each case. The conventional system, an electrically driven chiller with a gas-fired steam generator, incurred a cost for electricity, both metered and demand, that was about 6½ times the cost of the gas used for space heating. The cost for gas heat is only $4000/yr, which is too small a base to derive enough savings to justify heat pumping. To heat pump electrically rather than burn gas directly would cost the user about $4500 more per year - definitely unmarketable as represented by point "G" in Figure 6-4. However, if gas was not available and the fuel source alternative was electric resistance heating, then a very attractive energy cost savings (approximately $9000/yr) would develop in favor of heat pumping. This savings could easily provide the user with a less than one-year return on his incremental investment for

the heat pumping functions and thus represented a marketable system as denoted by point "E" in Figure 6-4.

The system concept was fixed at the 90-hp steam turbine size, operating in a base-loaded control mode. A 75-hp induction motor was used to augment the output of the turbine when the chiller compressor was heavily loaded. Otherwise, the induction motor functioned as a generator, converting the turbine's excess power output into electrical energy. The initial analysis was performed using the load of an office building. This class of buildings has a large cooling load relative to the heating load. This system showed an energy cost savings when compared with the conventional system, but not a savings in source energy usage. Since the large office building application is not favorable to gas heat pumps, the lack of energy savings was not unexpected.

The third phase of the system modeling occurred as a parallel effort to the component design task. One objective of this phase was to optimize the system for appropriate applications. Another objective was to expand or transform the model into a design tool capable of analyzing the changes in energy cost and consumption as influenced by refrigerant, compressor performance, heat exchange areas, water flow rates, heating water supply temperatures and other design variables. Thus, the advanced system model became a decision-making tool assisting in the component design effort by producing the component cost/operating cost trade-off information. Ultimately, the model was used to optimized the heat exchanger areas using a two-year payback criterion.

6.3 Component Design Effort

6.3.1 General Approach

The objective of the component design effort was to produce component specifications and drawings sufficiently detailed so that a good manufacturing cost estimate could be made for the system. A significant pattern of this effort focused on alternative methods for transmitting the power from the turbine to the electric motor generator and to the compressor. A system loss analysis was conducted to minimize bearing and windage losses. Each alternative power transmission concept was modeled and simulated in the

office building application to determine its annual energy cost. Differences in operating costs, relative technical risks and engineering judgment on relative equipment cost were used to make a selection of the two most promising systems - one, a mechanical power transmission system and the other, an electrical power transmission system. The electrical power transmission system was relegated to lower priority status since it encompassed much higher technical risks. Cost estimates were made for both systems and found to be nearly the same.

The compressor, turbine and steam generator designs were carried out to a higher level of detail than other parts of the system. These components were unique components in this system, which required extra engineering details and layouts for accurate cost estimates. The steam turbine buckets and nozzles required several iterations of manufacturability/performance compromises before a reasonable cost/performance balance was achieved. Other purchasable components like controls, valves, heat exchangers, etc. were specified for their respective duties. Other specially manufactured items like gears, shafts, machine housings, etc. were specified and their costs estimated on the basis of previous manufacturing experience.

Whenever possible, component or subsystem functions were integrated to minimize the overall cost. For example, the steam regenerator was incorporated into the duct between the turbine exhaust and the steam condenser, eliminating piping. The lube oil was cooled in the condenser's condensate well while at the same time supplying heat to deaerate the condensate, eliminat- a shell and tube heat exchanger. The start-up and shutdown sequencing control was designed to eliminate costly electrical frequency synchronizing controls to bring the turbine-generator on-line at start-up. The control sequence also eliminated the normally required governor control and throttle valve on the turbine and all the special related controls for the steam generator.

Some design improvements were made to the system which resulted in reduced energy consumption. The first improvement was a reduction in the heating system hot water delivery temperature. Originally, this temperature was scheduled to match the temperatures required for baseboard hot water systems.

The water delivery temperature was fixed at 110°F which is suitable for fan coil distribution systems. The 110°F temperature was much closer to the normal delivery temperature of an electrically driven heat pump system, but precluded the system from the retrofit market segment. The second change was the result of an improved expander design. The original model was based upon a single-row, axial-flow turbine. The design was revised to incorporate a more efficient two-row Curtis design.

When the interim system cost analysis and cost-of-ownership studies were completed, it was evident that the system could not meet the marketability criterion previously established in the W.E. Hill study. Therefore, the component design effort was truncated and the system analysis (modeling) effort was reinitiated to look at alternate heat engine drivers and reoptimize the steam-turbine system for a motel application. The motel application required more heat and less cooling than the office building application. This application should have improved the propsects for the LSM steam turbine system.

6.3.2 Component Design Details

The selection of the steam cycle state points and the turbomachinery package were intensively examined in order to maximize performance and minimize cost. The numerous iterations of performance, first cost, and operating cost resulted in a unique design. The turbomachinery package, including the turbine, the generator, the compressor, the step-up/step-down gearing, the mechanical clutch, and the steam regenerator were packaged in a single integrated unit as shown in Figure 6-5. The compressor and turbine were designed as plug-in components for ease of manufacture, assembly, and maintenance. The final steam cycle selected included reheat and regeneration.

The cycle state points are illustrated in Figure 6-6. The selection of these state points was made by iteration on the cost and performance of the steam generator and the turbine itself. The final turbine design is shown in Figure 6-7. The turbine had two wheels and three stages of expansion. The first wheel was a Curtis wheel with two rows of turbine blades. This wheel provided for two stages of expansion as in the 7½-ton system by using partial admission with recovery, reheat and reentry. The second wheel provided for the third stage of expansion with full admission to the turbine.

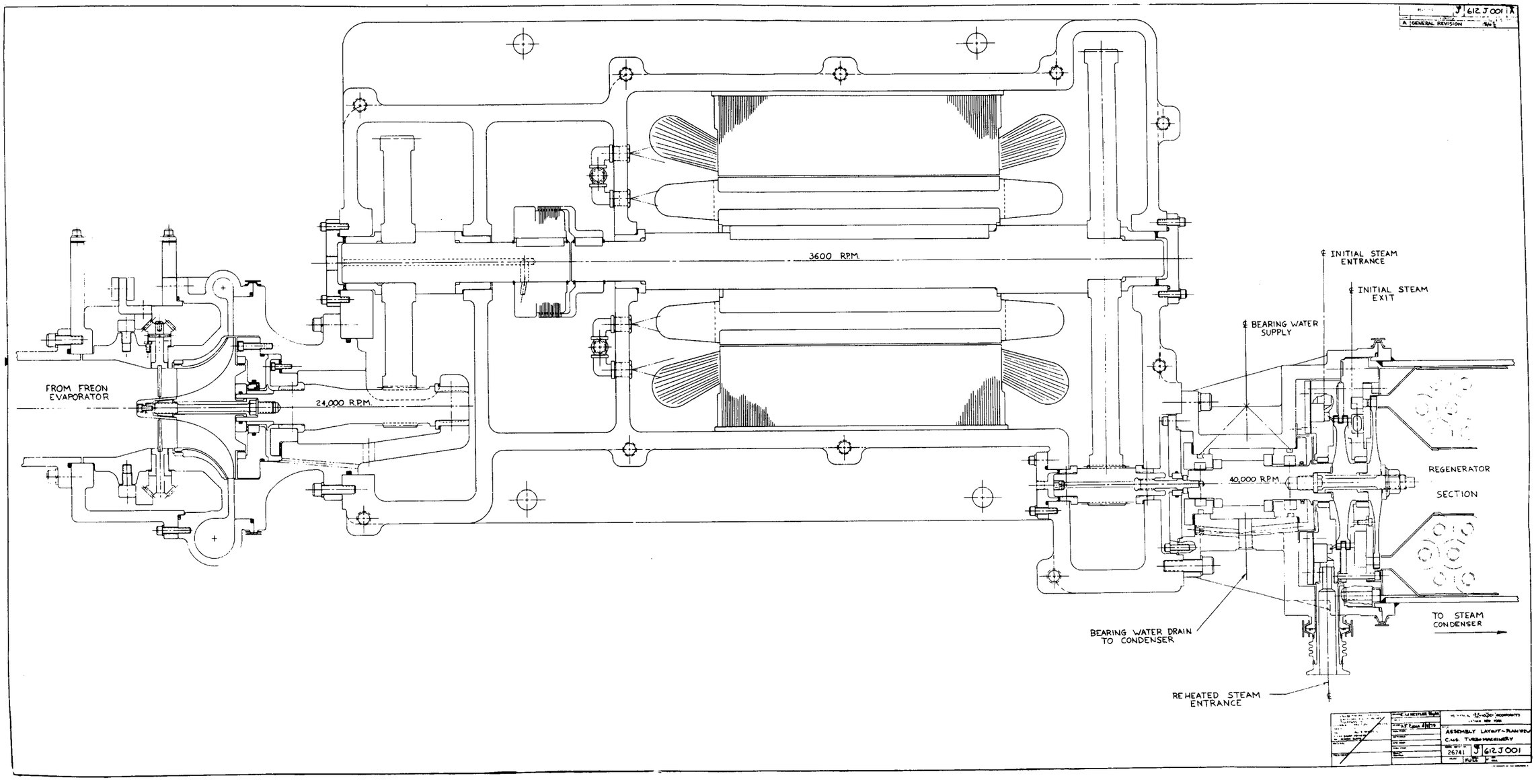


Fig. 6-5 Cross Section of Turbo-machinery Unit

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State Point 1 - 815 psia, 1050°F

State Point 3 - 64 psia, 1050°F

State Point 4 - 2.4 psia, 483°F

State Point 6 - 2.4 psia, 140°F

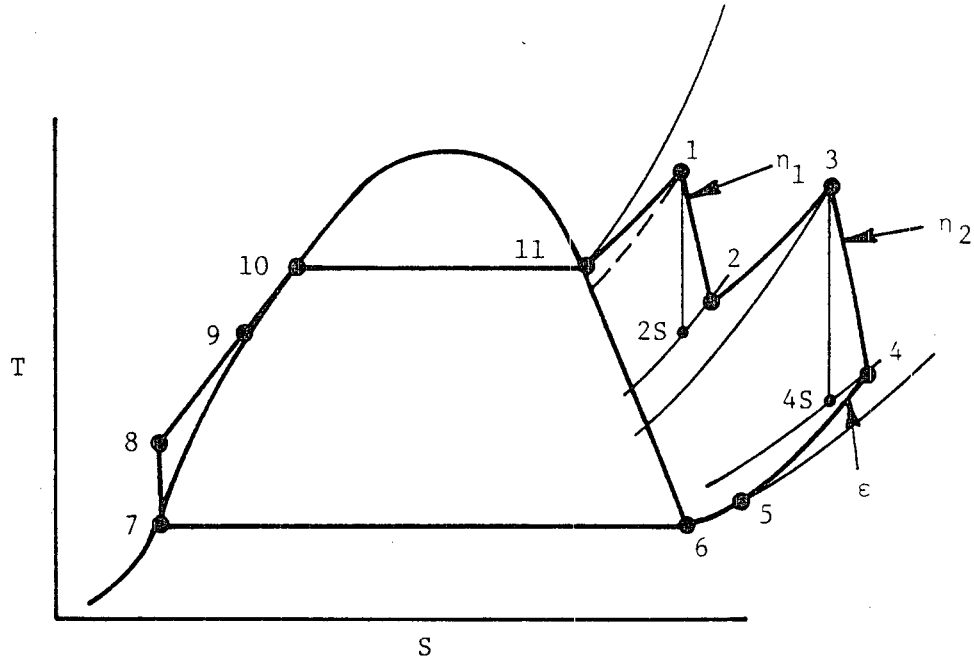

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Fig. 6-6 Steam-Turbine Power Cycle T-S Diagram for LSM Unit

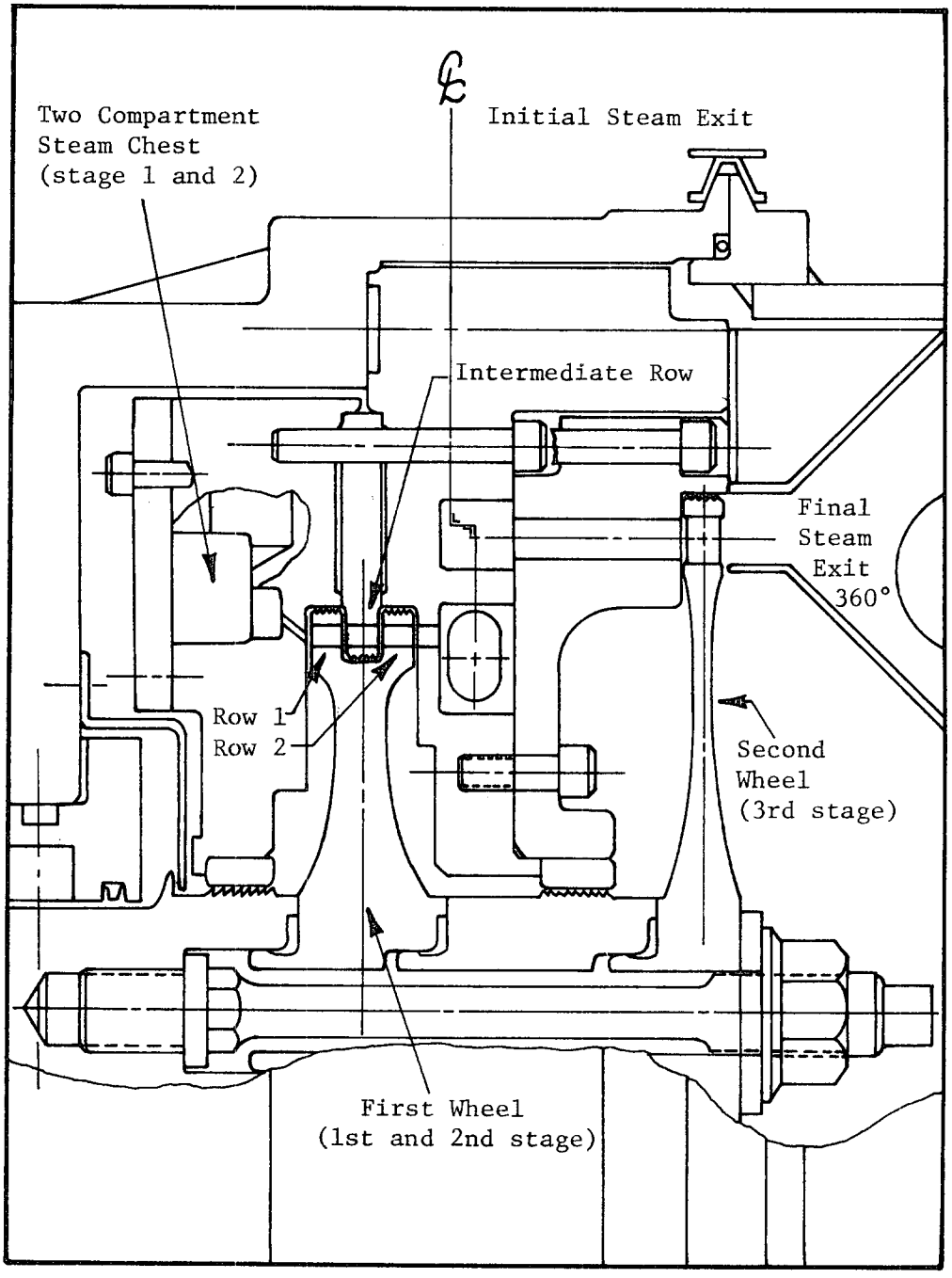

Fig. 6-7 Cross Section of Three-Stage Turbine

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Table 6-1 presents some technical details on performance and size of the turbine. The turbine speed is constant at 40,000 rpm and output is 90 hp.

The compressor is a single-stage centrifugal compressor as shown in Figure 6-8. This compressor is similar to the geometry used in the 7½-ton system. However, the speed of the compressor is constant at 24,049 rpm. Capacity modulation is achieved through variable inlet guide vanes. Some performance information on the compressor is given in Table 6-2.

The motor/generator was a 3600-rpm, 75-hp induction unit. This unit was directly coupled to the steam turbine through a 11.11:1 ratio gear set. The motor generator was also coupled to the compressor through a 6.67:1 ratio gear set. On the compressor side, a mechanical clutch was provided on the 3600 rpm shaft. Additional details of the component design are provided in Appendix Volume V.

6.4 Steam Turbine Prime Mover System Reoptimization for a Motel Application

In general, the steam-turbine system's performance, relative to conventional HVAC equipment, improved as the annual heating load increased. The motel application required about twice as much heating and one-half as much cooling as the office building. The same peak cooling loads (120 tons) existed in both of these buildings. The motel was expected to be a more advantageous application for the steam turbine. Since the original system optimization was based upon the office building load, reoptimizing for the motel load was expected to produce further performance gains.

The motel study included the effects of process loads, which were much larger than for the office building. Also, the viability of heat pumping was reexamined. In this case, outside air was the only heat pump heat source available.

The results of the process load analysis showed that these loads should not be supplied by the steam turbine's steam condenser. Table 6-3 displays these results. Using a separate auxiliary steam generator for the process load reduced both energy cost and energy consumption as compared to supplying the loads via the steam condenser.

TABLE 6-1

TURBINE STAGE DATA

	Stage 1 (2-Row)			Stage 2 (2-Row)		Stage 3 (1-Row)
Turbine Speed (rpm)	40,000			40,000		40,000
Admission (%)	8.3			52.1		100.1
Turbine Efficiency (%)	50.8			68.8		73.3
	Row-1	Int.	Row-2			
Rotor Blade Height (in.)	0.112	0.112	0.126			0.340
Blade Pitch Dia. (in.)	5.50	5.50	5.514			7.00

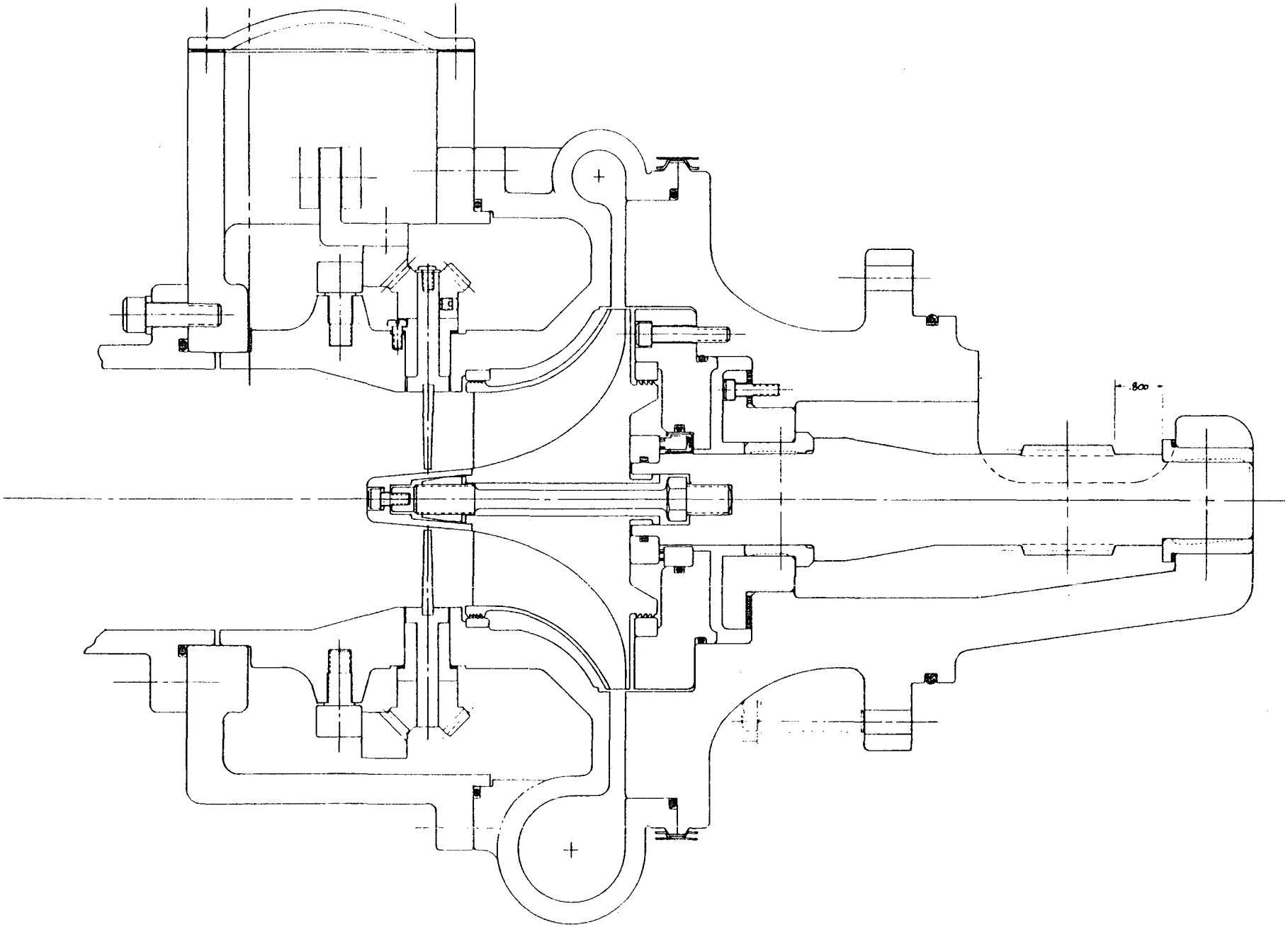

Fig. 6-8 Compressor Module No. 3 Layout

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TABLE 6-2

COMPRESSOR DESIGN PARAMETERS

Working Fluid	R-12
Evaporator Temperature (saturation)	30°F
Condenser Temperature (saturation)	105°F
Power Required	148.6 hp
Rotor Speed	24,049 rpm
Estimated Compressor Efficiency	0.785
Pressure Ratio, R	3.2736
Rotor Tip Diameter	6.2 in.
Inducer Tip Diameter	3.5 in.
Inducer Hub Diameter	1.0 in.
Diffuser Inlet Diameter	7.0 in.
Diffuser Exit Diameter	11.4 in.

TABLE 6-3

EFFECT OF PROCESS LOAD ON STEAM-TURBINE POWER CYCLE PERFORMANCE(1)

	HEAT SOURCE FOR PROCESS LOAD(2)	
	Steam Condenser(3)	Auxiliary Steam Generator(4)
Energy Cost $1000/yr(5)	21.3	21.0
Energy Consumption, 10^9 Btu/yr	8.36	8.10

(1) Base-loaded, 90-hp steam turbine without heat pumping

(2) 0.46×10^9 Btu/yr

(3) Delivering 110°F water

(4) 85% thermal efficiency

(5) $2.2/$10^6$ Btu gas price, $0.04/kW-hr electric rate plus $3/mo/peak annual kW demand

Heat pumping was modeled as a two-stage system as an expedient for this study. Single-stage heat pumping was technically possible, providing a positive displacement-type compressor was used. Single-stage heat pumping would produce a slight improvement in performance over the two-stage system. However, the performance differences between the single stage and two stages were not large enough to reverse the conclusions which are evident from Table 6-4 for the steam turbine driver from the alternative driver study described below. They are:

- Heat pumping required more energy consumption and had higher energy costs than if the system did not pump heat
- Modulating the turbine's output resulted in higher energy costs than if the driver was base-loaded. A slight reduction in energy consumption occurred with the modulated control and no heat pumping.

Thus, the base-loaded no heat pumping concept remained the best configuration selection for the motel application as it was for the office building application. However, in the motel application, the decision could be based upon differences in energy cost and consumption, whereas the office building heat pumping cases resulted in insignificant changes in operating cost because the heating loads were too low.

The base-loaded no heat pumping configuration was then optimized for the motel application. The results are shown in Table 6-5. Small reductions in both heat exchanger costs and energy costs were achieved.

6.5 Technology-Based Potential Market Limitations for the LSM

The potential market for the LSM concept, or for any large gas or heat actuated heat pump (HAHP) was primarily in the three sectors listed below in decending order of size or market potential:

- Office buildings
- Retail space, education and other public buildings
- Hotel and apartment.

The office building market sector, however, was least favorable, and the

TABLE 6-4

ECONOMY MOTEL APPLICATION
ENERGY COST AND CONSUMPTION(1)

Control/Configuration		Alternative LSM Gas Heat Pump Drivers					
Load Control	Configuration	Steam Turbine	S/E (Adv)(2)	S/E(Exist)(3)	P/I Diesel(4)	S/I(5)	Elect.(6) Motor
Base Loaded	Heat(7) Pumping	$23.3(8) (7.98)(9)	21.5 (6.62)	22.2 (7.04)	22.0 (6.78)	22.9 (7.37)	--
Base Loaded	No Heat Pumping	19.8 (7.55)	17.7 (6.39)	18.6 (6.77)	17.4 (6.46)	17.9 (6.87)	--
Modulated	Heat Pumping	27.6 (8.01)	26.6 (7.50)	27.8 (8.00)	27.0 (7.51)	27.6 (7.95)	37.6 (8.92)
Modulated	No Heat Pumping	24.2 (6.85)	25.0 (7.06)	25.4 (7.18)	23.2 (6.73)	23.8 (6.98)	32.7 (8.68)

(1) Excluding process loads

(2) Stirling engine advanced design

(3) Stirling engine existing P75 design

(4) Pilot-ignited, gas-fired Diesel (existing)

(5) Spark-ignited, gas-fired engine

(6) Electric-motor-driven chiller/heat pump with gas-fired steam generator

(7) External source two-stage heat pumping

(8) Energy cost $1000/yr for gas @ $2.2/$10^6$ Btu, and electric @ 4¢/kWhr and $3/kW/mo demand

(9) Total energy consumed by system 10^9 Btu/yr

TABLE 6-5

ECONOMY MOTEL APPLICATION(*)
STEAM TURBINE POWER SYSTEM OPTIMIZATION

Design Parameter	Optimum for Office Building	Optimum for Motel
Steam Condenser Areas, ft^2:		
Heating System Bundle	153	140
Cooling Tower Bundle	99	118
Refrigerant Condenser Areas, ft^2:		
Cooling Tower Bundle	813	700
Refrigerant Evaporator Area, ft^2:	577	525
Heating Loop Water Flow, GPM	229	233
COST TRADE-OFFS		
Heat Exchanger Capital Cost	$26,688	$25,898
System Energy Cost/yr	$19,849	$19,248

*Does not include process load. Mechanical drive concept.

hotel-apartment market was most favorable for heat actuated heat pump application. This division occurred because the office building application usually presented a high ratio of annual cooling load to annual heating load, typically 1.5-3 (for Pittsburgh weather), and had an associated low balance-point temperature during heating service. For this application, an HAHP would have to satisfy an annual HVAC demand that was dominated by the cooling load. HAHPs would be less energy-efficient when operated as a chiller than when operated as a heating device, and they would usually be less competitive with alternative systems when operated as a chiller than when operated for space heating.

A low balance-point temperature during heating service removed that portion of the annual heating load that would otherwise be associated with mild ambient temperatures for exterior source heat pumping. Based on Pittsburgh weather, typically less than 20% of the annual heating load in an office building would correspond to ambient temperatures above 32°F (0°C). For a heating load characteristic of a single-family residence, about 50% of the annual heating load would occur at 32°F (0°C) and above. The efficiency advantage of air-source heat pumping was considerably reduced at low ambient temperatures. Also, internal sources for heating pumping would be reduced or eliminated at low ambient temperature due to increased heat loss and economizer operations.

The hotel-apartment market was characterized by relatively low ratios of annual cooling load to heating load. In some cases, the ratio would approach that of a single-family residence, typically 0.2 to 0.3 for Pittsburgh weather. These applications tended to have a significant portion of the annual heating load occurring at mild ambient temperatures. Because of the nature of the load associated with the hotel-apartment market sector, the HAHP could often show a significant annual operating cost advantage over competing systems in many applications. This market sector, however, was small, representing less than 20% of the total potential market for large HAHPs.

The total potential market for large (60 tons and up) HAHPs is defined here as the HVAC equipment market presently served by chillers and combination heating/cooling systems with unit nameplate ratings of 60 tons cooling

capacity or greater. More than half of hotel-apartment-institutional construction with a total building requirement for cooling in excess of 60 tons was excluded from the potential market because it was served by multiple, small-sized, unitary equipment intallations. Multiple, small-sized, unitary installations represented a lower initial equipment cost than did large central unit installations. The central unit installation would, however, usually provide better energy efficiency and a quieter living environment. Since a commercial HAHP was expected to have a cost premium relative to conventional central equipment and since central equipment had a cost premium relative to multiple unitary installations, it was unlikely that the HAHP could have penetrated the multiple-unit market. Therefore, this market did not offer a promising opportunity for the HAHP.

The parameters characteristic of normal large building HVAC distribution systems would pose constraints on the operating efficiency of any heat pump system, especially an HAHP. Hot and chilled water distribution temperatures were significantly higher and lower, respectively, than the warm and cool air temperatures in forced-air systems. Generally, heat pumps serving most large buildings had to operate over correspondingly greater temperature differences, source to sink, than heat pumps serving smaller buildings. Greater temperature differences meant a reduced coefficient of performance and a greater strain on the heat pump compressor. The phased heat exchange system employed in the CNG/DOE HSPF reduced source-sink temperature differences, but the percentage reduction effected for hot and chilled water systems was less than half that effected for forced air systems.

Temperature and volume modulation of the heating/cooling distribution medium were means that could be employed to reduce the seasonal energy consumption of a heat pump. These means had been frequently proposed for application to HAHPs because of the relative ease with which the heat-actuated prime mover's speed (or frequency) could be modulated. The controlling zone demand that had to be satisfied in a multizone system at any point in time, however, would tend to limit the extent to which modulation could be employed.

The constraints imposed by multizone hot and chilled water systems on HAHP performance could be largely overcome through distribution system engineering

modifications. Advantageous modifications would include increased fan-coil or passive surface heat exchange capacity, increased water circulation rates and split or zoned supply circuits. Each of these modifications would add cost to the distribution system. The installed cost of the distribution systems in large buildings typically would be two to three times as much as that for the central equipment. Therefore, any incremental cost increases in the distribution system would add significant costs relative to the central equipment.

If the needs of a central HAHP necessitated increased distribution system costs, these costs would be added to any central equipment first-cost premium attributable to the heat pump. These costs would represent a further burden with respect to market penetration.

6.6 Alternate Driver Study

The scope of this study was limited to replacing the 90-hp steam turbine driver with an equivalent-size, gas-energized, alternative prime mover. No other engine sizes were considered. Both internal and external combustion engines were included in the list of four alternative drivers. The internal combustion engines included one operating on a spark-ignited Otto cycle and one on a liquid, pilot-ignited, gas-fired diesel cycle. The latter was included to take advantage of the inherent higher efficiencies that are available with the diesel cycle. Commercially available engine performance was modeled although a specific engine was not modeled. Two Stirling engine options were modeled - an existing unit and a proposed advanced design. The United Stirling Model P75 engine performance was used as the basis for what is designated as the "Existing" design in Table 6-4. The engine designated as the "Advanced" Stirling engine was based upon performance of an advanced model P40 design. In both cases, the engine's efficiency was adjusted for natural gas fuel and their rated sizes were scaled to 90 hp. Figure 6-9 presents the part-load efficiencies of the alternate drivers as well as the steam turbine.

The waste heat recovery for each of these drivers was also modeled. For the internal combustion engines, heat was recovered from the water jacket, lube oil and the exhaust gas. The exhaust gas was cooled only to 325°F to avoid condensation. Figure 6-10 presents the maximum recoverable heat for

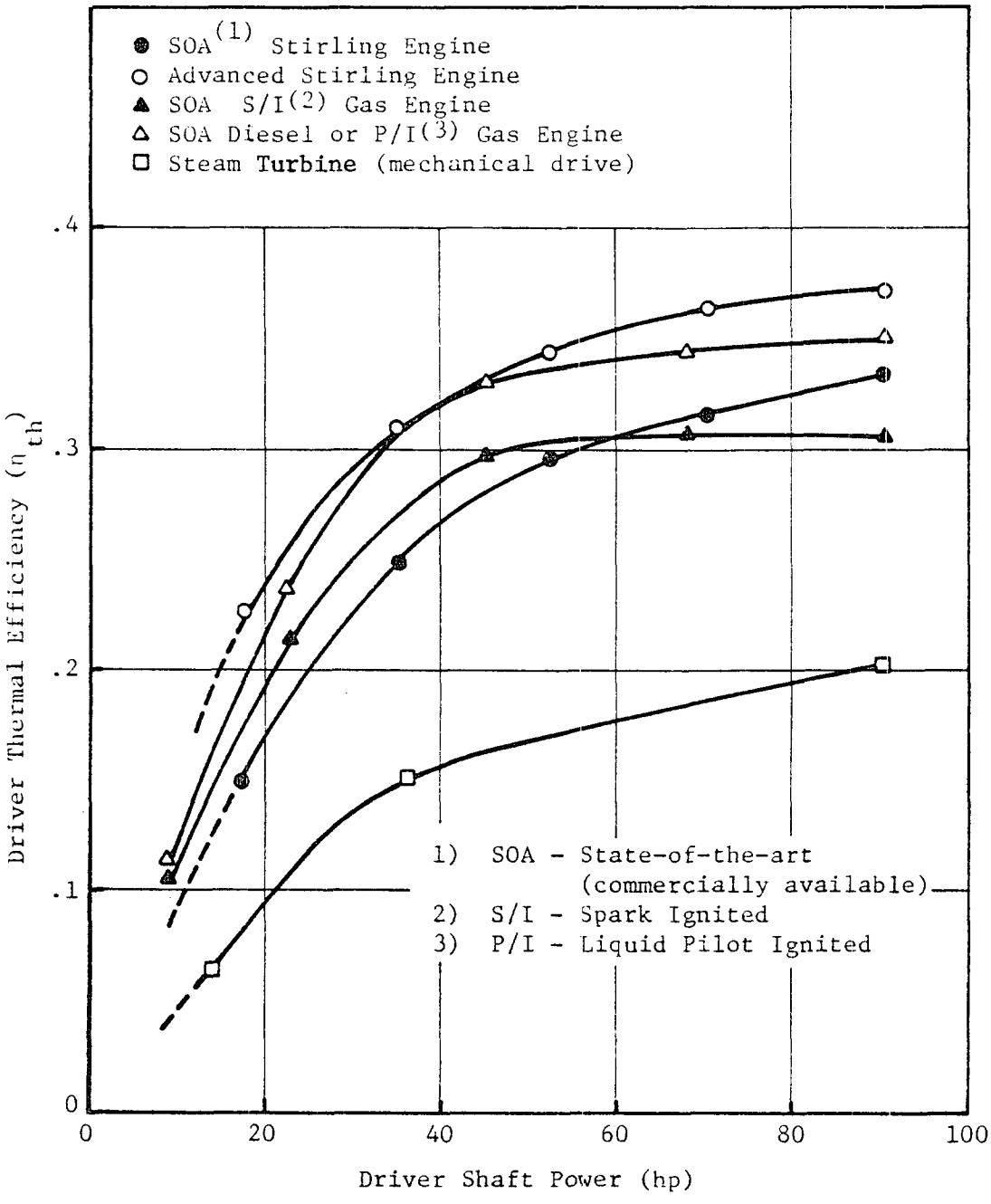

Fig. 6-9 Thermal Efficiency Comparison of Alternate 90 hp Drivers (including all required power for accessories) for LSM Gas Heat Pumps

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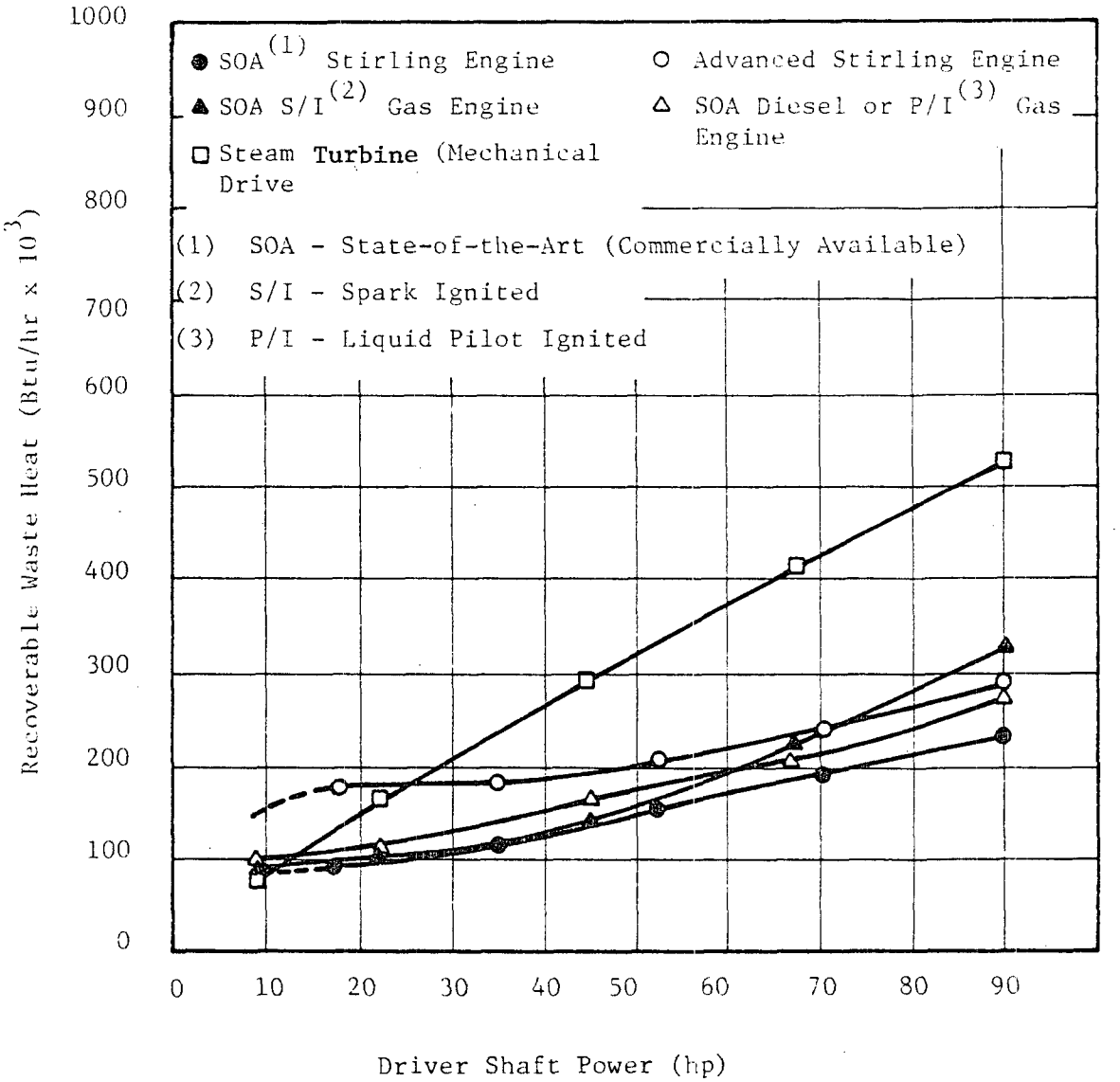

Fig. 6-10 Recoverable Waste Heat Comparison of Alternate Drivers for LSM Gas Heat Pumps

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each of the drivers as operated over a wide load range. The steam turbine driver is shown for comparison purposes.

All four alternate drivers were modeled to perform in the modulated and base-loaded control modes with and without heat pumping. The results of this analysis are also shown in Table 6-4. Like the steam-turbine driver, the base-loaded configurations without heat pumping produced the most energy-efficient and cost-effective system designs. A comparison of energy costs and energy usage for different drivers showed small differences when compared to the rather significant differences in engine efficiencies. For example, the full-load thermal efficiency of the advanced Stirling engine was about 85% higher than the steam turbine, but the Stirling engine only produced a savings of 12% in energy cost and 18% in energy usage. These lower than expected savings for the more efficienct driver were the result of the high heating loads and low cooling loads representative of the motel application. Large amounts of gas were required for the auxiliary steam generator to augment the small amounts of rejected heat from the higher efficiency drivers. The large amounts of gas for auxiliary heat diluted the potential energy savings for these alternate drivers in high heating loads. The more efficiency drivers would produce larger savings in an application that had a larger cooling load and less heating, such as the office building.

7.0 LARGE SIZE MODEL COMPARATIVE COST OF OWNERSHIP STUDY

7.1 Introduction and Overview

The purpose of the study was to determine the comparative ownership costs of the Large Size Model (LSM) HSPF gas heat pump with respect to appropriate competitive equipment. The five principal elements of the study can be summarized as follows:

1. Computer models were employed to calculate the building loads and simulate the corresponding hourly thermal energy requirement, to simulate the alternate equipment system operations, to compute the precise applicable utility rates and to compute the final economic comparisons of the various equipment systems against a reference system, which was the lowest initial cost system.

2. Evaluation parameters included two building types, five HVAC equipment system types, five weather areas, five utility rate schedules, and three scenarios for future escalation and schedule changes for utility rates. The specific evaluation parameters employed are listed in Table 7-1.

3. Two comparison criteria were employed in evaluating the LSM versus competitive equipment; source energy consumption and discounted cash flow rates of return for an equipment owner/operator.

4. Three related issues were evaluated in extensions of the base study:

    a. Sensitivity of LSM HSPF operating economics to the cooling-to-heating load ratio.

    b. Sensitivity of LSM HSPF operating economics to heat transfer surface area.

    c. One-to-one comparison of the all-electric system to the gas/electric reference system.

TABLE 7-1

EVALUATION PARAMETERS, LSM HSPF GAS HEAT PUMP
COMPARATIVE COST OF OWNERSHIP STUDY

Evaluation Parameters	Selected Options for Study	
1. Building Type	Office Building - Current Construction Specifications	
	Motel or Hotel - Current Construction Specifications	
2. HVAC Equipment System Type	Heating Service	Cooling Service
	(1) Electric Resistance Coil Heater	Package Thru-Wall Terminal Unit
	(2) Gas Steam Generator	Electric Reciprocating Chiller
	(3) Electric Steam Generator	Electric Reciprocating Double-Bundle Condenser
	(4) Electric Steam Generator	Electric Heat Pump/Heat Recovery Type
	(5) Individual Gas Furnaces	Electric Split-System Air Conditioner with A-Coil
	(6) HSPF LSM, Mechanical Drive	Same as heating
	(7) HSPF LSM, Electric Drive	Same as heating
3. Weather Areas	Chicago Cleveland Dallas Los Angeles New York City	
4. Utility Rate Schedules	Chicago Cleveland Dallas Los Angeles New York City	
5. Scenarios for Future Utility Rates	AGA Projection for Future Rate Levels DOE Projection for Future Rate Levels Utility Plans for Future Rate Forms	

5. A summary was prepared with respect to the results from both building types for the two LSM HSPF gas heat pump options studied.

Utility rates were computed on the basis of the total building utility energy consumptions. This computation permitted accurate determination of energy costs by applying actual utility rates to the total consumptions of the respective equipment/building combinations. General usages of utility energy for consumptions other than equipment usage were included in the monthly and annual utility charge calculations.

7.2 Scope

The base study of the office and motel buildings comprised 44 cases which were evenly divided between building types. The case identification numbers with their respective categories are shown in Table 7-2. In addition to the base category (present rates and actual weather), additional categories examined the effects of escalating rate-levels and future schedules. Rate-level projections, according to the data furnished by the Department of Energy (DOE) and the American Gas Association (A.G.A.) were used. These studies were performed for the five cities indicated. The effect of future rate schedules was examined in three cities. The rate sensitivity examinations all used constant weather (Cleveland, Ohio). A sample Case Data Sheet (Case No. 1 - Category 1a) is shown as Table 7-3. All 44 case data sheets are included in Appendix Volume VII.

Three extensions of the base study were performed. One concerned sensitivity of the LSM to the cooling to heating load ratio. The office building was found to represent a nonoptimum application for the LSM HSPF gas heat pump due to having a high ratio of annual cooling load to heating load. The ratio for Pittsburgh weather was about 2.2. A survey of office space cooling load to heating load ratios determined that this value is typical of contemporary office space construction. In addition to cool/heat ratio, heat exchanger area was recognized as a potential important influence on LSM HSPF heat pump performance. Since the motel showed a lower cool/heat ratio than the office building, the motel was chosen as the basis for examination of the sensitivity of both the cool/heat ratio and the system heat exchanger area for the LSM HSPF gas heat pump electrical drive version.

TABLE 7-2

CASE IDENTIFICATION NUMBERS

Location City	Base: Present Rates/ Actual Weather Motel 1a	Base: Present Rates/ Actual Weather Office 1b	Present Rates/ Constant Weather Motel 2a	Present Rates/ Constant Weather Office 2b	Rate Level & Rate Form Sensitivity: DOE Adj. Rates. Constant Weather Motel 4a	Rate Level & Rate Form Sensitivity: DOE Adj. Rates. Constant Weather Office 3a	Rate Level & Rate Form Sensitivity: DOE Adj. Rates Constant Weather Motel 4b	Rate Level & Rate Form Sensitivity: DOE Adj. Rates Constant Weather Office 3b	Future Rates/ Constant Weather Motel 4c	Future Rates/ Constant Weather Office 3c
Chicago	1	12	2	4	32	19	37	24	42	29
Cleveland	3	12	3	12	33	20	38	25	--	--
New York	4	13	5	14	34	21	34	26	43	30
Dallas	6	15	7	16	35	22	40	27	--	--
Los Angeles	8	17	9	18	36	23	41	28	44	31

TABLE 7-3

CHICAGO

SUMMARY OF ECONOMIC COMPARISONS

ECONOMY MOTEL

Category 1a Case No. 1

SYSTEM TITLES:

A-1 Electric motor reciprocating compressor chiller with gas-fired hot water steam generator

A-2 Electric motor reciprocating compressor chiller with a double-bundle (heat reclaim) condenser and electric steam generator

A-5 LSM HSPF Gas Heat Pump - Mechanically Coupled Version

A-6 LSM HSPF Gas Heat Pump - Electrically Coupled Version

Annual Operating Costs	Alternative Systems			
	A-1	A-2	A-5	A-6
Initial Investment	$315,000	$326,000	$345,868	$345,868
Natural Gas - Annual	$ 13,146	0	$ 21,530	$ 21,472
Electricity - Annual	$ 37,364	$ 81,749	$ 21,013	$ 19,073
Maintenance - Annual	$ 6,500	$ 7,300	$ 7,300	$ 7,300
Insurance - Annual (1.0%)	$ 3,150	$ 3,260	$ 3,459	$ 3,459
Ad Valorem Tax - Annual	$ 3,150	$ 3,260	$ 3,459	$ 3,459
First-Year Interest (10.0%)	$ 23,625	$ 24,450	$ 25,940	$ 25,940
First-Year Total Operating Cost	$ 86,935	$120,019	$ 82,701	$ 80,703
Present Worth of Net Cash Flow	$339,383	$452,799	$329,532	$322,779
Rate of Return	BASE	N/A	9.05	13.32
Total Source Energy Consumption (Btu x 10^6)	$ 11,450	$ 17,784	$ 11,068	$ 10,539

The base study used the popular, low-cost electric compressor and gas steam generator system as the reference. However, in doing so the study failed to gain insight into comparison of the gas steam generator system with a similar system equipped with an electric steam generator. The office building was evaluated using constant weather and varying rates in the various cities to make this gas - electric comparison. This comparison constituted the third extension study.

Professional consultants' recommendations for appropriate competitive systems were accepted. These systems comprise the first five listed under Item 2, Figure 7-1. HVAC system initial costs and installation costs were furnished by the same consultants for the motel and office buildings. The gas heat pump costs were assigned as a 50% premium over an alternate system that consisted of an electric motor reciprocating compressor with heat reclaim and electric steam generator.

7.3 Results

The Summary of Results/Motel - Base Study (see Table 7-4) and Summary of Results/Office - Base Study (see Table 7-5), summarize the comparisons performed in the study. The bar charts on Figures 7-1 and 7-2 are referenced to the Base System (reciprocating compressor and gas steam generator) and show the corresponding relative source energy consumptions percentages and rates of return, if any, also expressed in percentages. The best gas heat pump and the best alternate systems were chosen to be shown on these bar charts. The best gas heat pump was the electrical-driven version (designated A-6 and B-5 for the motel and office, respectively) and the best alternate systems were the electric motor reciprocating compressor with heat reclaim condenser and electric steam generator (designated A-2 and B-2 for the motel and office, respectively). The office building's system B-4, the electric heat pumps connected with an enclosed water loop, was inferior with respect to investment criteria and source energy efficiency to the alternate systems and the base system in all cases and was not shown on the bar charts.

The data in the bar charts shown on Figures 7-1 and 7-2 are for the base category which used present rates and actual weather for each city.

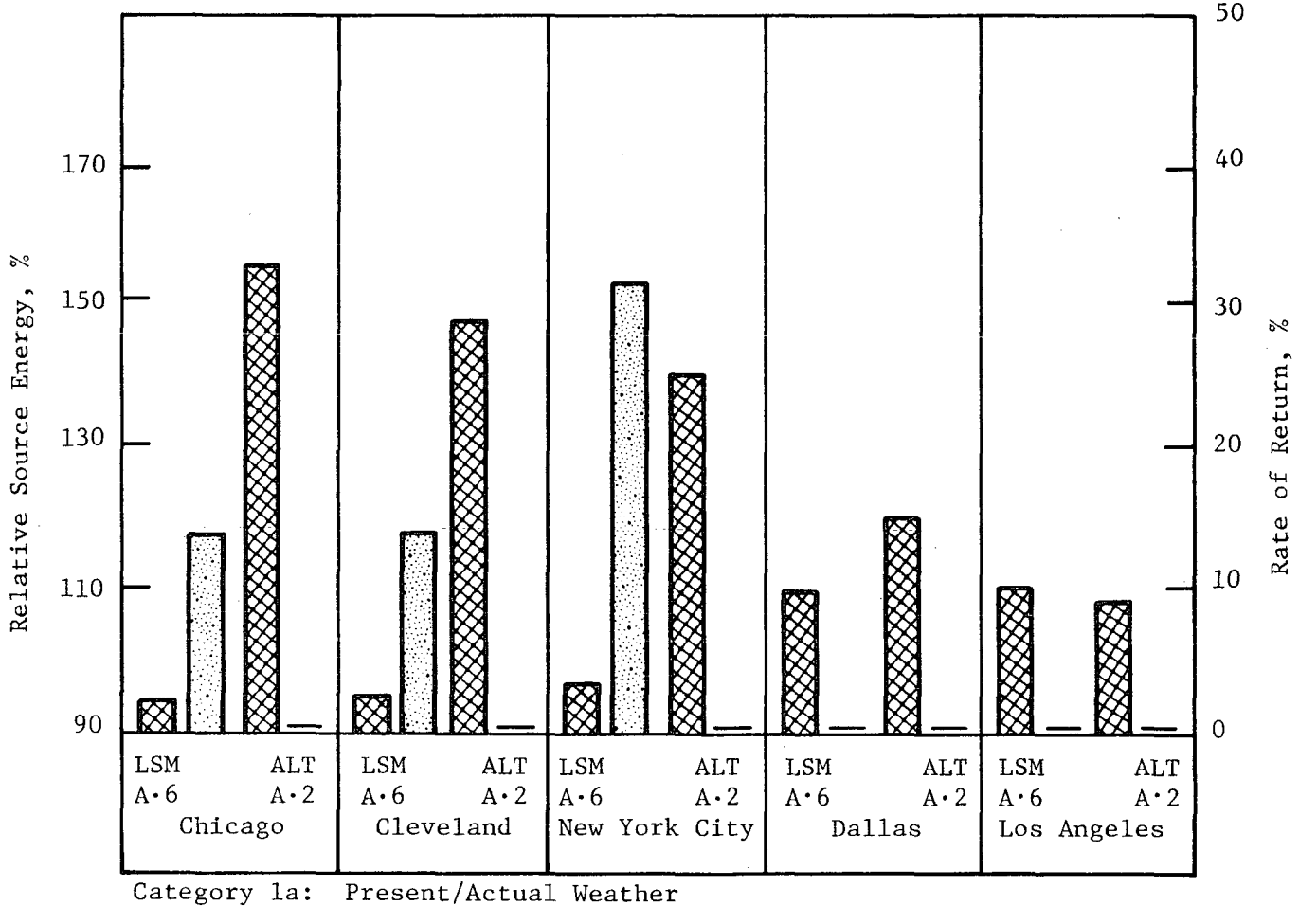

Fig. 7-1 LSM HSPF and Best Conventional Alternative System Comparison for Motel Application

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TABLE 7-4

BASE STUDY

SUMMARY OF RESULTS/MOTEL

Best LSM HSPF (A-6) versus Best Competitive System
(Reciprocating Chiller and Gas Steam Generator)(A-1)

Location (Utility Rates & Weather)	Basis For Utility Rates	Source Energy Savings (%)	Annual Utility Cost Svgs. (1000's $)	Equip. Cost Premium (1000's $)	Rate of Return (%)
Chicago	Present	+ 8	+20	+31	13.32
	DOE Project.	+ 8	+20	+31	3.48
	AGA Project.	+ 8	+20	+31	14.17
	Future Schd.	+ 8	+20	+31	13.17
Cleveland	Present	+ 7 +(27)*	+22	+31	13.75
	DOE Project.	+ 7	+26	+31	N/C**
	AGA Project.	+ 7	+21	+31	13.78
New York	Present	+ 4	+22	+31	31.27
	DOE Project.	+ 4	+19	+31	26.61
	AGA Project.	+ 4	+23	+31	33.51
	Future Schd.	+ 4	+29	+31	46.97
Dallas	Present	-10	+10	+31	N/C
	DOE Project.	-10	+11	+31	6.77
	AGA Project.	-10	+16	+31	10.43
Los Angeles	Present	-10	+ 7	+31	N/C
	DOE Project.	-10	+ 5	+31	N/C
	AGA Project.	-10	+14	+31	4.66
	Future Schd.	-10	+20	+31	9.20

Note: MTI Cost Study Equipment Cost Premium = $54,000
* With External Source Heat Pumping
**N/C = Not Computable Because of Negligible, Negative or Zero Savings

TABLE 7-5

BASE STUDY

SUMMARY OF RESULTS/OFFICE BUILDING

Best LSM HSPF (B-5) versus Best Competitive System
(Reciprocating Chiller and Gas Steam Generator)(B-1)

Location (Utility Rates & Weather)	Basis For Utility Rates	Source Energy Savings (%)	Annual Utility Cost Svgs. (1000's $)	Equip. Cost Premium (1000's $)	Rate of Return (%)
Chicago	Present	- 5	+ 5	+95	N/C*
	DOE Project.	- 5	+ 2	+95	N/C
	AGA Project.	- 5	+ 9	+95	N/C
	Future Schd.	- 5	+ 6	+95	N/C
Cleveland	Present	- 9	+ 5	+95	N/C
	DOE Project.	- 9	+ 1	+95	N/C
	AGA Project.	- 9	+ 9	+95	N/C
New York	Present	-17	+11	+95	N/C
	DOE Project.	-17	+ 8	+95	N/C
	AGA Project.	-17	+13	+95	0.72
	Future Schd.	-17	+17	+95	5.80
Dallas	Present	-35	+ 3	+95	N/C
	DOE Project.	-35	+ 3	+95	N/C
	AGA Project.	-35	+ 4	+95	N/C
Los Angeles	Present	-37	+ 3	+95	N/C
	DOE Project.	-37	+ 1	+95	N/C
	AGA Project.	-37	+ 2	+95	N/C
	Future Schd.	-37	+ 5	+95	N/C

Note: MTI Cost Study Equipment Cost Premium = $118,000
* N/C = Not Computable Because of Negligible, Negative or Zero Savings.

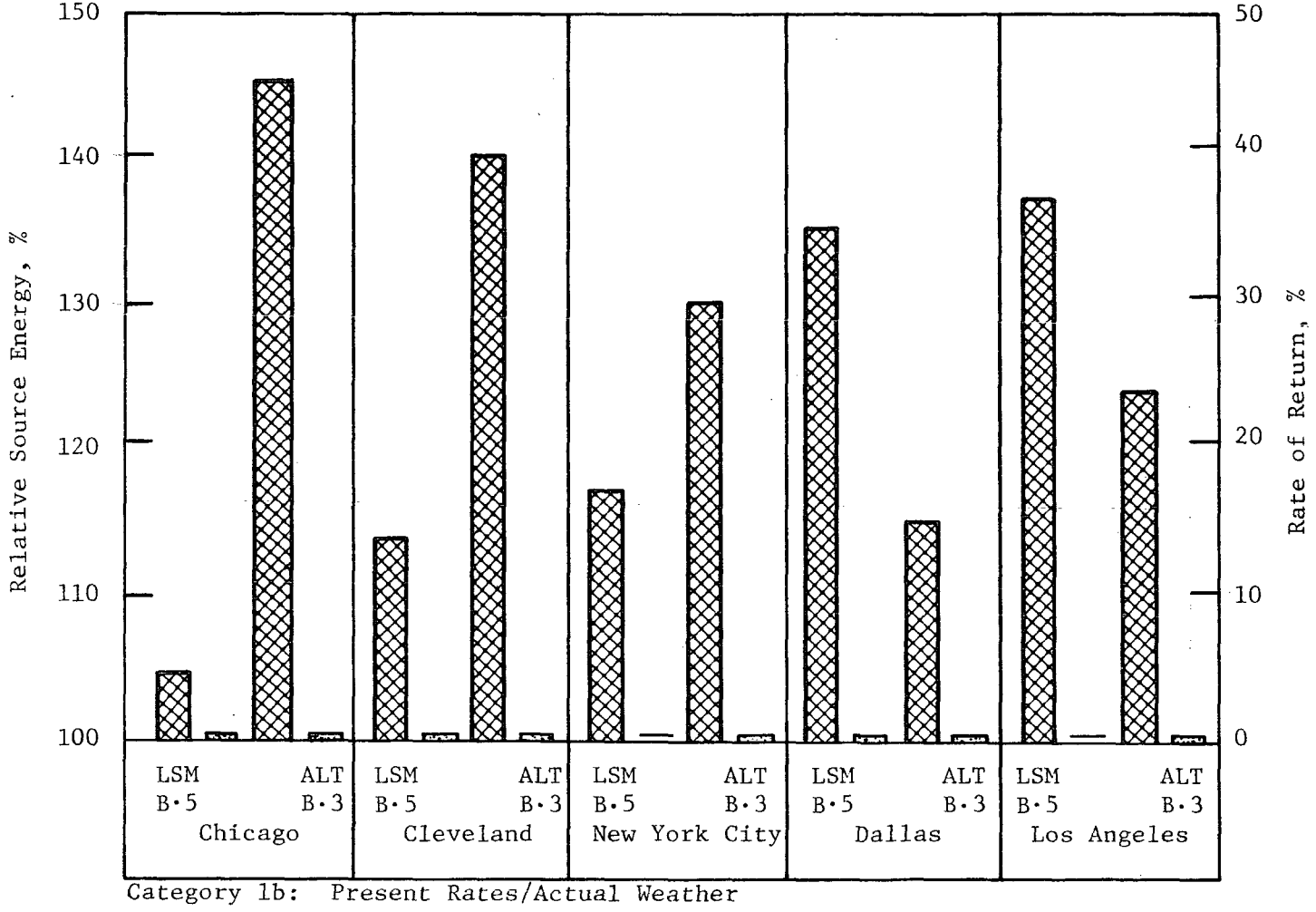

Fig. 7-2 LSM HSPF and Best Conventional Alternative System Comparison for Office Application

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7.4 Results - Extension Studies

7.4.1 Cooling Load to Heating Load Ratio Sensitivity Study

Table 7-6 shows the variation of the rates of return for the motel with cool/heat load ratio variations of 0.59, 0.45 and 0.36. The respective rates of return were 11.48%, 13.06% and 14.18%, showing a direct relationship between cool/heat load ratio and rate of return. For the extension study, the motel was modeled with Cleveland's weather and present utility rates. As anticipated, the lower the cool/heat ratio, the better the rate of return for the LSM HSPF gas heat pump compared to the base system, the reciprocating compressor and gas steam generator. A plot of these results is shown as Figure 7-3.

7.4.2 LSM HSPF Gas Heat Pump Heat Exchanger Area Sensitivity Study

Within the scope of this study, the electrically coupled version of the LSM HSPF gas heat pump was modeled with three sets of heat exchanger areas. These sets were designated as A-6 (Large), A-6.1 (Medium) and A-6.2 (Small). Table 7-7 shows the response of rates of return for the motel. For this study, Cleveland weather and present utility rates were employed. The respective rates of return were: for the large heat exchanger - 13.06%; for the medium size heat exchanger - 5.00%; and for the smallest heat exchanger - 4.49%.

Variations in LSM HSPF heat exchanger area produced relatively large changes in the rate of return, but relatively small changes in source energy savings. The effect of varying heat exchanger area on heat pump cost was not considered. Within the range of the variations employed, the dollar value of the incremental heat exchanger areas was small compared to the cost of the gas heat pump system. A curve of the heat exchanger sensitivity results is shown as Figure 7-4.

7.4.3 Comparison of Electric and Gas Steam Generators

The third extension study was performed primarily to compare space heating via an electric steam generator to space heating via a gas steam generator. System B-2, Electric Reciprocating Compressor with Heat Reclaim Condenser, was also included in the study. The results of this study are shown on Table 7-8.

TABLE 7-6

EXTENSION STUDY SUMMARY OF RESULTS FOR MOTEL - SENSITIVITY TO CHANGE OF COOL/HEAT RATIO

Electric Motor Reciprocating Compressor and Gas Steam Generator (A-1) (Reference System) versus Electrically Coupled LSM HSPF Gas Heat Pump (A-6)

Cool/Heat Ratio	Location (Utility Rates and Weather)	Basis For Utility Rates	Source Energy Savings (%)	Annual Utility Cost Svgs. (1000's $)	Equip. Cost Premium (1000's $)	Rate of Return (%)
0.59	Cleveland and Cleveland	Present	4	9.08	+31	11.48
0.45	Cleveland and Cleveland	Present	6.4	9.84	+31	13.06
0.36	Cleveland and Cleveland	Present	7.7	10.4	+31	14.18

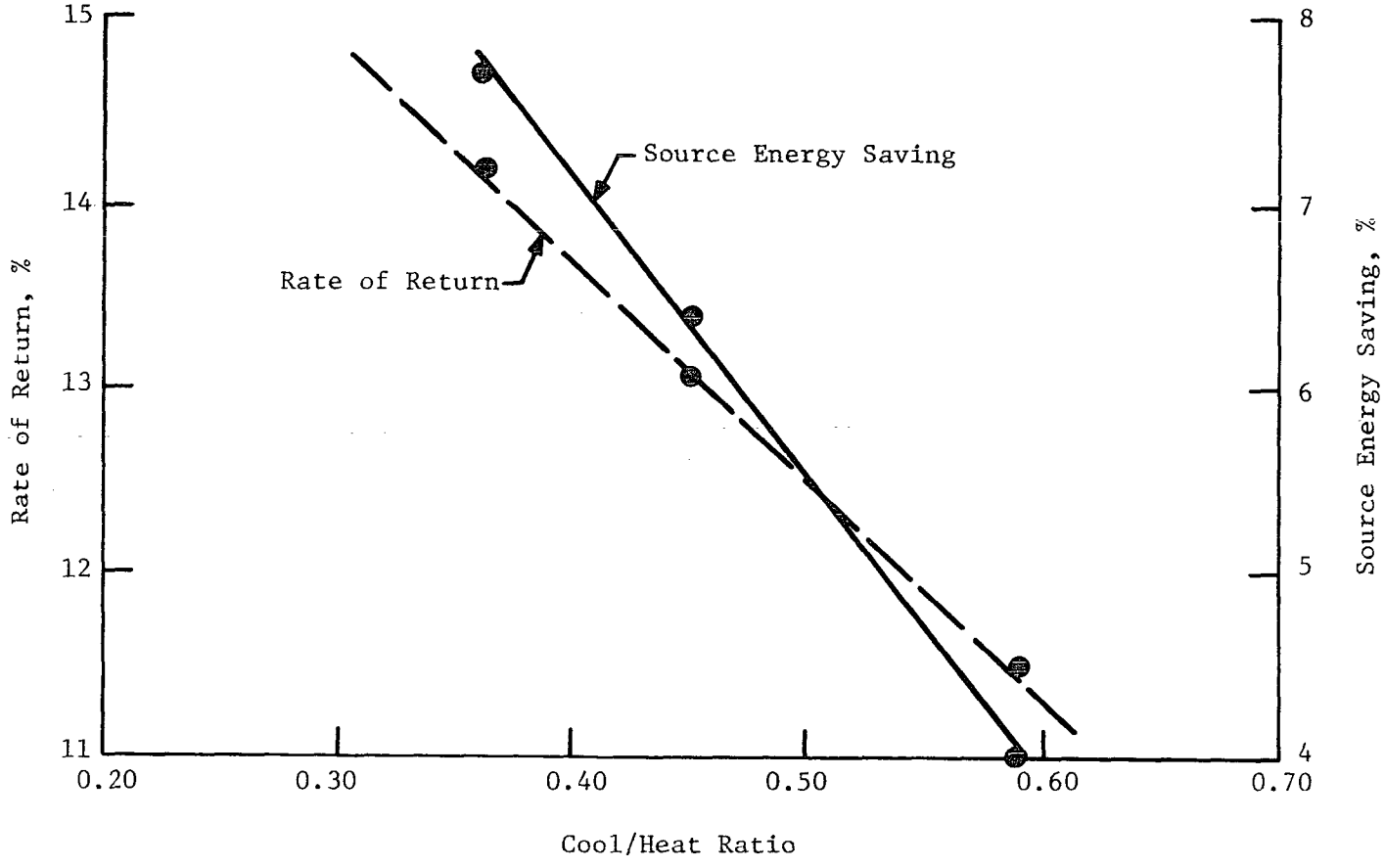

Fig. 7-3 Sensitivity to Change of Cool/Heat Ratio in Motel Application. LSM HSPF Gas Heat Pump (A·6) Versus Electric-Motor Driven Reciprocating Compressor and Gas Steam Generator (A·1).

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TABLE 7-7

EXTENSION STUDY SUMMARY OF RESULTS FOR MOTEL - SENSITIVTY TO CHANGE OF LSM HSPF GHP HEAT EXCHANGER AREA

Electric Motor Reciprocating Compressor and Gas-Fired Steam Generator (A-1) (Reference System)

versus

Electrically Coupled LSM HSPF GHP (A-6) (Large HX), and
Electrically Coupled LSM HSPF GHP (A-6.1) (Medium HX), and
Electrically Coupled LSM HSPF GHP (A-6.2) (Small HX).

Heat Exchanger Size	Location (Utility Rate and Weather)	Basis For Utility Rates	Source Energy Savings (%)	Annual Utility Cost Svgs. (1000's $)	Equip. Cost Premium (1000's $)	Rate of Return (%)
Large (A-6)	Cleveland and Cleveland	Present	6.44	9.84	+31	13.06
Medium (A-6.1)	Cleveland and Cleveland	Present	6.52	6.28	+31	5.00
Small (A-6.2)	Cleveland and Cleveland	Present	6.06	6.08	+31	4.49

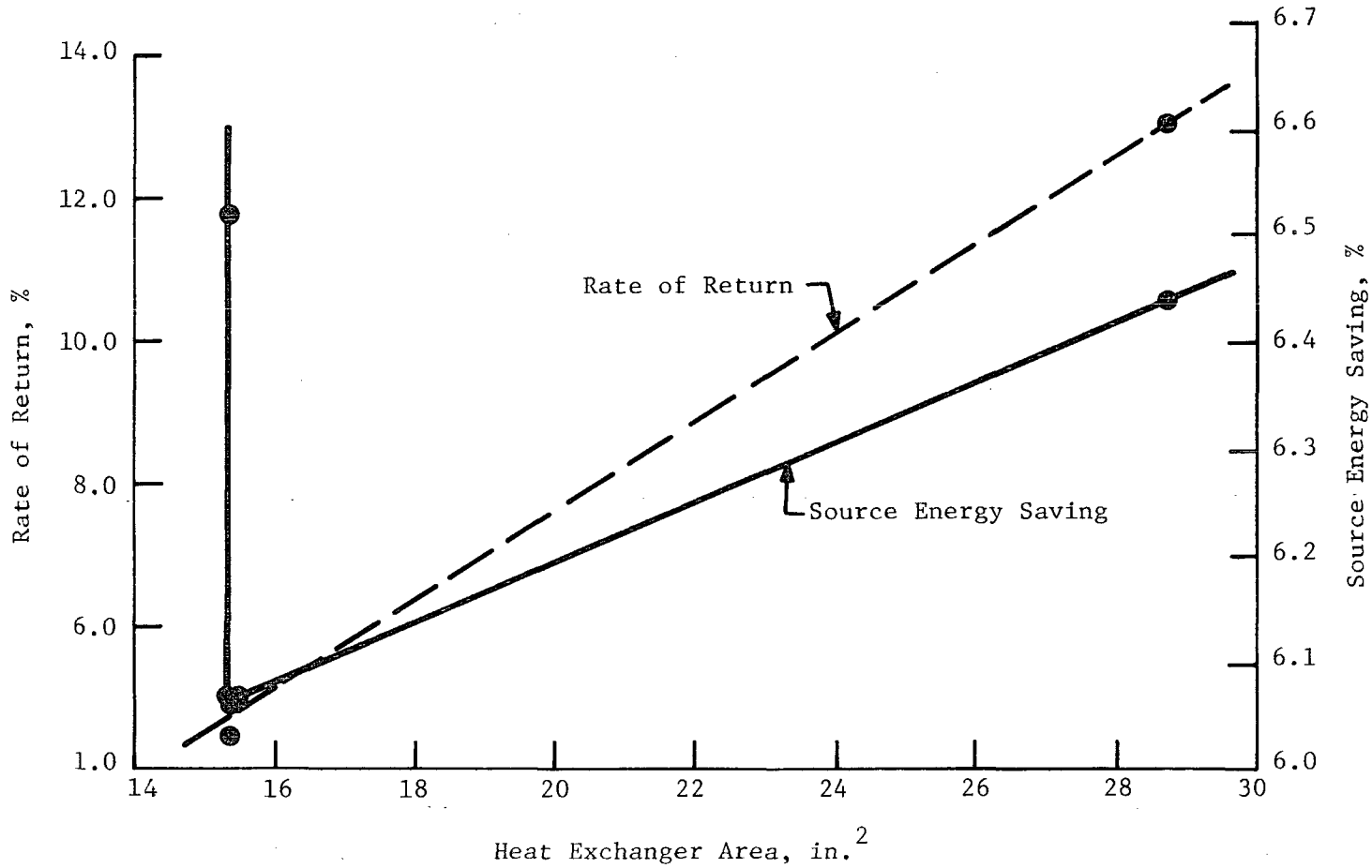

Fig. 7-4 Sensitivity to Change of Heat Exchanger Area in Motel Application. LSM Gas Heat Pump (A·6) (A·6-1) (A·6-2) Versus Electric-Motor-Driven Reciprocating Compressor and Gas Steam Generator (A·1)

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TABLE 7-8

EXTENSION STUDY SUMMARY OF RESULTS FOR
OFFICE - REVISED REFERENCE SYSTEM

Electric Motor Reciprocating Compressor and Electric
Steam Generator (B-1.1) (Reference System)

Versus

Electric Motor Reciprocating Compressor and Gas Steam Generator (B-1)

Location Utility Rates and Weather	Basis For Utility Rates	Source Energy Savings (%)	Annual Utility Cost Savings (1000's $)	Equipment Cost Premium (1000's $)	Rate of Return (%)
Chicago and Cleveland	Present	17	14.91	2.165	370.51
	DOE Adj.	17	13.11	2.165	327.54
	AGA Adj.	17	16.14	2.165	402.73
	Future	17	14.46	2.165	359.77
Cleveland and Cleveland	Present	17	13.65	2.165	338.28
	DOE Adj.	17	11.21	2.165	278.13
	AGA Adj.	17	14.78	2.165	366.21
New York and Cleveland	Present	17	30.43	2.165	742.19
	DOE Adj.	17	25.22	2.165	626.17
	AGA Adj.	17	30.27	2.165	750.18
	Future	17	28.40	2.165	707.81
Dallas and Cleveland	Present	17	13.16	2.165	327.54
	DOE Adj.	17	14.25	2.165	355.47
	AGA Adj.	17	13.90	2.165	344.73
Los Angeles and Cleveland	Present	17	14.64	2.165	364.06
	DOE Adj.	17	13.15	2.165	327.54
	AGA Adj.	17	15.76	2.165	391.99
	Future	17	12.24	2.165	303.91

The office building was the basis for this study. Cleveland location (weather) and present utility rates, DOE adjusted rates, and A.G.A. adjusted rates corresponding to five study cities (Chicago, Cleveland, New York, Dallas and Los Angeles) were applied. Future rate schedules were also applied to three cities: Chicago, New York and Los Angeles. As shown in Table 7-8, very high rates of return are available for using the gas steam generator system rather than the electric steam generator system. These systems are identical except for the steam generators. The source energy savings for the gas steam generator system are also substantial. The consultant who supplied the estimated total installed costs for all systems predicted that the electric steam generator system would cost more than the gas steam generator system. An assumption was made that the electric steam generator system would cost less in order to permit the electric steam generator system to become the base system for the economic runs. The results were consistently favorable for the gas steam generator system.

Results of each of the substudies comprising the Comparative Cost of Ownership Study are contained in Appendix Volume VII.

7.5 Summary of Results

The following results or key observations were made from the motel building portion of the base study:

- The motel is a more favorable application than the office building.
- New York City is a good application for either version of the LSM HSPF gas heat pump
  - Lower source energy consumption than base system
  - Substantial operating cost savings
  - Rate of return up to 31% with present rates and actual weather.
- The LSM HSPF gas heat pump looked best with future rate schedules and A.G.A. rate-level projections.
- No economic advantage for the LSM HSPF gas heat pump existed through the use of inflated rate levels under the DOE projections as compared to current rate levels.

Other future rate-level projections result in an economic advantage for the LSM HSPF.

- The best energy cost savings for the gas heat pump was in New York City, followed by Cleveland, Chicago and Dallas. Los Angeles showed the lowest energy cost savings.
- Source energy consumption of the LSM HSPF gas heat pump was less than that for reciprocating compressor and gas steam generator system in Chicago, Cleveland and New York City.

The following key observations were made from the office building portion of the base study:

- The office building was generally not a favorable application for the LSM HSPF heat pump.
- The electrically-driven reciprocating chiller and gas steam generator (System B-1) showed the lowest source energy consumption of all systems. The LSM HSPF gas heat pump system was the second lowest source energy system in northern locations; third lowest in southern locations.
- The LSM HSPF gas heat pump with electric coupling drive was economically superior to the mechanical coupling version in all cities. The electrically coupled LSM HSPF gas heat pump showed utility energy cost savings over all other systems in all cities and for all rate projections.
- The energy cost savings of the electrically coupled LSM HSPF heat pump were insufficient to yield a positive rate of return except in New York City with future rate schedules.
- The best energy cost savings was in New York City; then Chicago, followed by Cleveland. Dallas and Los Angeles showed very small savings.

The following key observations were made from the extension study analyses:

- A lower cool/heat ratio showed an improved (higher) rate of return for the LSM HSPF gas heat pump when compared with the reciprocating compressor and gas steam generator system.
- The rate of return and source energy savings increased when the LSM HSPF gas heat pump was equipped with larger heat exchanger area.
- The application of much lower (promotional or special) electric rates than now exist would be necessary to enable the electric steam generator system to be competitive with the gas steam generator.

8.0 HSPF GAS HEAT PUMP EQUIPMENT COST STUDIES

The objective of the equipment cost studies was to develop careful estimates for component and system costs for products based on the DOE/CNG/MTI 7-1/2-ton and LSM HSPF gas-fired heat pump systems. This section describes first the individual tasks that constituted the cost studies and, their timing and interaction with the parallel design, market study and business venture analysis efforts. Second, the results of the cost studies for both the 7-1/2-ton and LSM HSPF systems are reviewed.

8.1 Cost Study Methodology

In concept, the equipment cost studies were to produce three cost estimates each for both 7-1/2- and LSM HSPF systems: preliminary, interim, and final system costs. The cost study methodology is illustrated in Figures 8-1, 8-2 and 8-3. As a result of program management decisions, however, both the 7-1/2-ton and LSM system development programs were either redirected or terminated before the final cost figures were generated. The redirection/termination decisions were, in fact, largely a result of the respective preliminary and interim cost study results. Even though the final system cost study was not performed for either of the two heat pump systems, the description of the methodology that would have been used in performing the studies is included below. The final study methodology is presented in order to clearly point out how the entire study program was integrated with the overall hardware product development program.

The cost studies for the 7-1/2-ton and LSM heat pump system were conducted by two different teams. The 7-1/2-ton system costs were produced by the Masco Corporation under contract to CNG and the LSM costs were developed primarily by MTI personnel, with some assistance from Charles J. R. McClure and Associates Inc., Consulting Engineers, St. Louis. In general, the cost study methodologies used by the Masco and MTI teams were identical. In the discussion which follows, illustrations and examples from the LSM study will be cited in order to clarify procedures used in both studies.

Prior to conducting either the preliminary, interim or final cost study efforts, a means of tabulating the system's costs, component by component, was devised. The entire LSM heat pump system was subdivided into 15

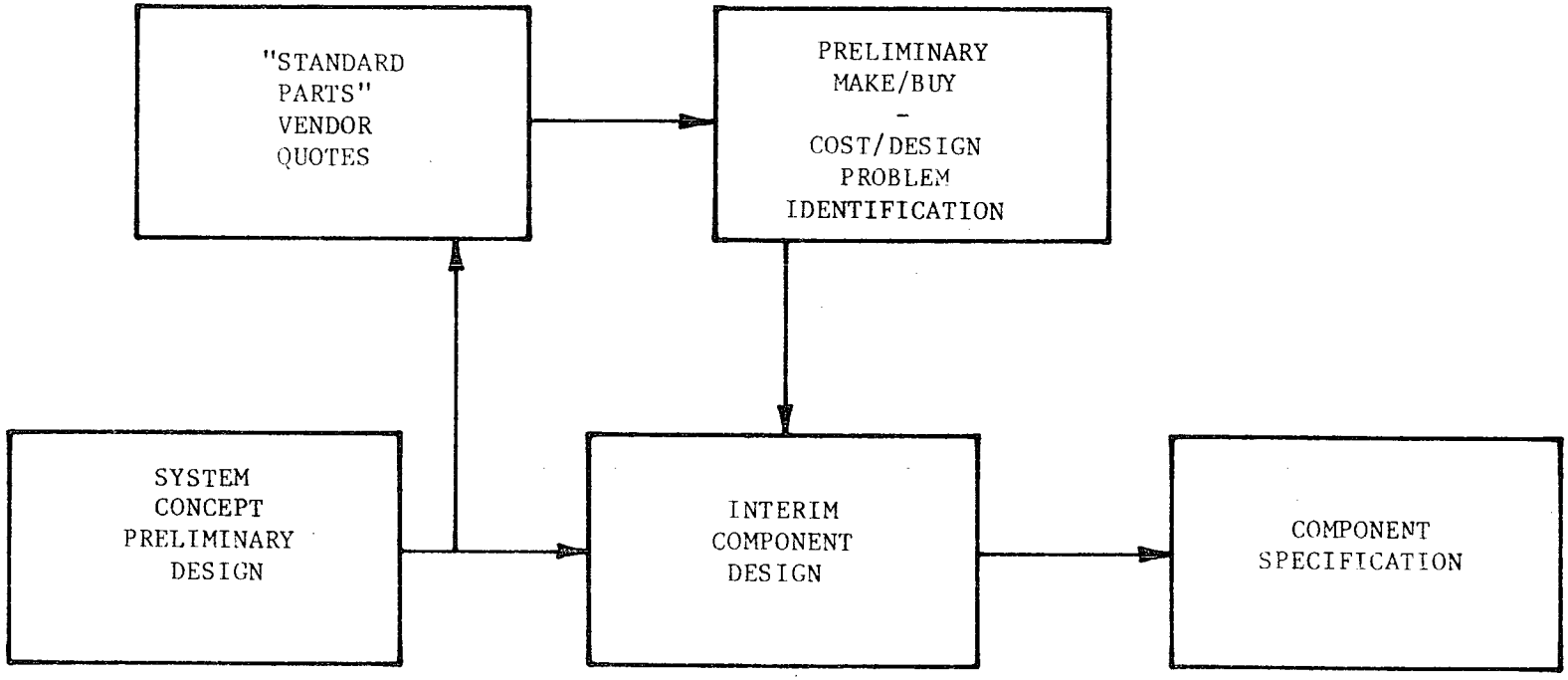

Fig. 8-1 Product Costing Plan - Preliminary Estimate

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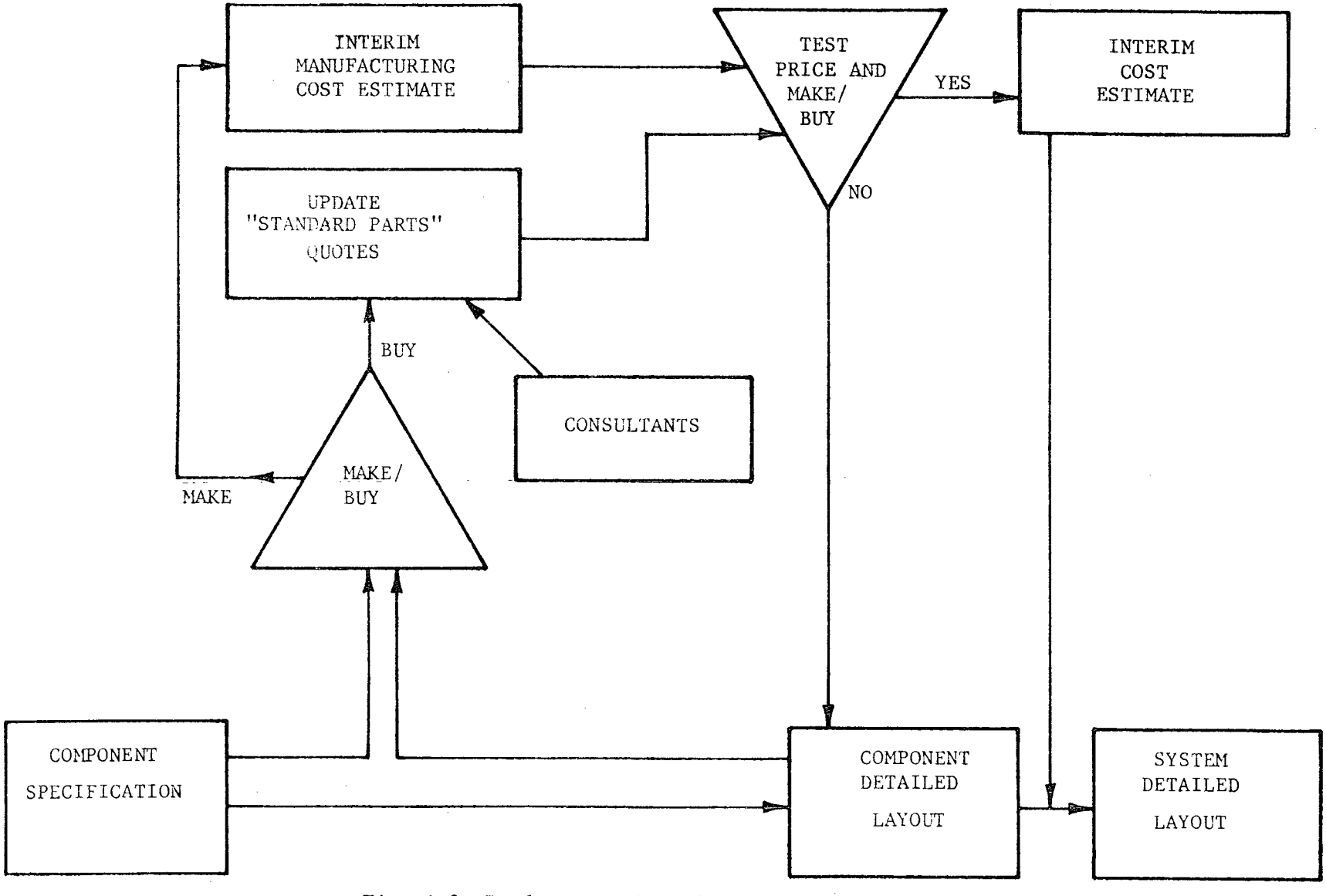

Fig. 8-2 Product Costing Plan - Interim Iteration

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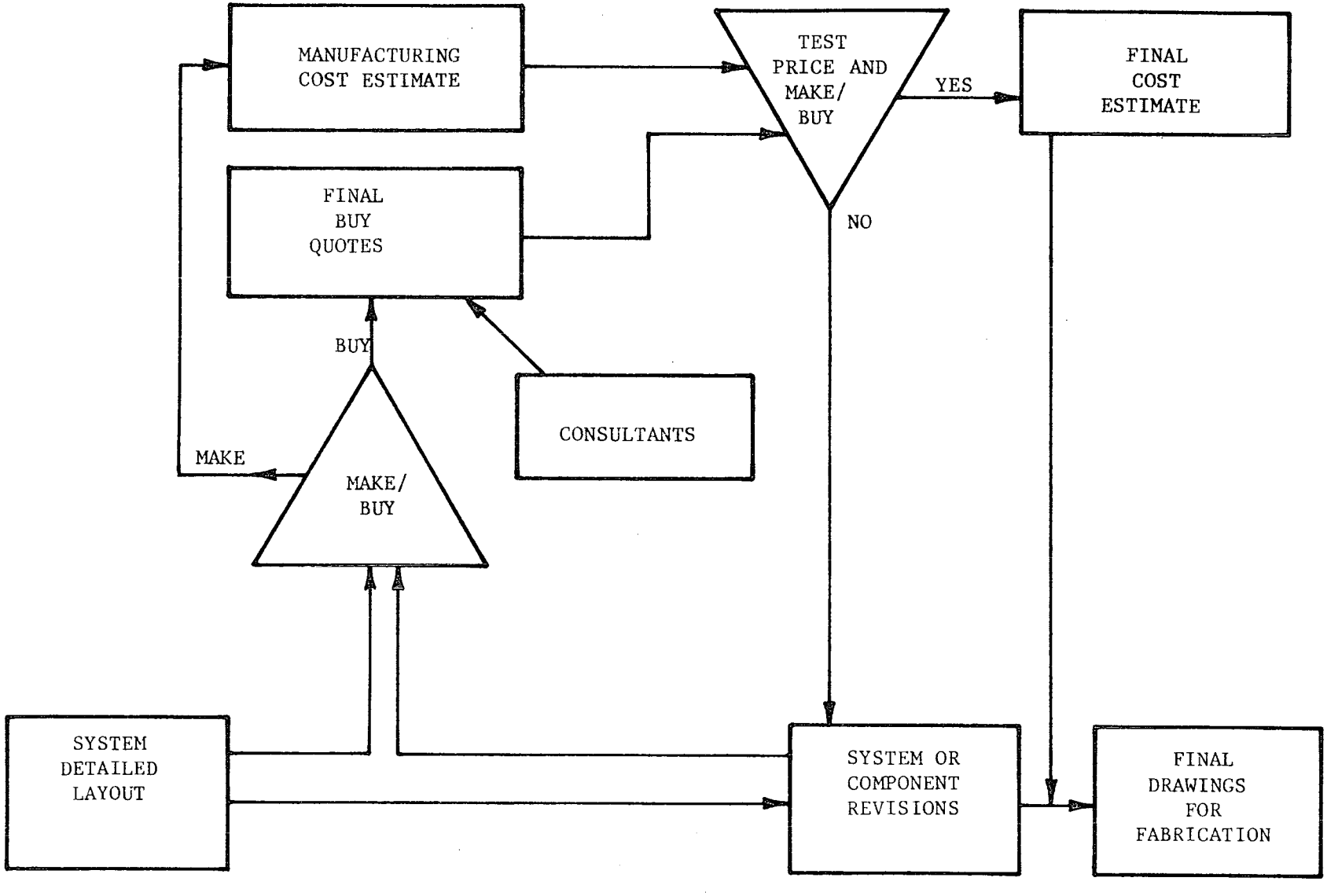

Fig. 8-3 Product Costing Plan - Final Iteration

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subassemblies; and each subassembly was subdivided into components, parts and/or operations (104 in all). Each of these items was assigned a unique "tracking" number. All of these latter items were listed on a tabulations sheet so that their costs could be recorded.

8.2 The LSM Cost Study

The LSM study was initiated when a system concept had been agreed upon, and a preliminary list of system components had been generated. For the preliminary system cost, a vendor quote was obtained for every commercially available component on the list.

The design engineers integrated the results of the preliminary cost study effort with their continuing design work. With this preliminary cost information, an interim system design and set of component specifications were developed. This interim set of designs and specifications was the basis of an interim costing effort much more intense than the preliminary system cost effort. At this point, the focus for the design effort continues to be support for the cost study.

The design information available at the start of the interim cost work was reviewed to determine whether individual components should be purchased or fabricated. In the case of some components, it was not known at the time of the review to which list they should be assigned. In these cases, two estimates were obtained: a purchase estimate and a make estimate.

Components that were assigned to the manufacturing list were generally costed using either Masco or Turbonetics Inc. (a subsidiary of MTI) manufacturing methods. Several approaches were used to determine these "make" costs at MTI. If Turbonetics had previously manufactured a part similar to the one(s) required in the heat pump, then a rough cost estimate was developed by extrapolating the costing data from the earlier made component. If the component did not resemble any components previously made by Turbonetics, then an estimate was made of the material, machine costs and manufacturing time involved. Masco followed this latter approach for all components. This information was reduced to component costs by multiplying the three

quantities by the appropriate cost factors (e.g., dollars per pound, dollar per hour, etc).

Several methods were used to determine the cost of purchased components. The first method utilized the experience of the costing team members. If any team member had purchased or received a quote for a component similar to the one required for the heat pump, then that cost was submitted for the preliminary estimate. Each cost was appreciated for the effects of inflation, recently announced cost changes and other factors. When a component's cost could not be determined using this method, then the component's specification was compared with that of commercially available hardware in equipment catalogs. Equipment part and model numbers were recorded; and vendors were requested to submit a verbal or written quote. Informal quotations were considered adequate for the preliminary and interim system costing. The lowest cost component with acceptable performance was the cost included.

The final product costing work was to follow a procedure similar to that used in the interim costing phase. Had the 7½-ton and LSM heat pump system effort not been terminated, the final system costing would have proceeded as follows.

In estimating the cost of manufactured components, detailed manufacturing drawings would be constructed. Decisions would be made as to whether parts could be cast and machined, or completely machined. Depending on the manufacturing process chosen, additional appropriate drawings would be made; material and labor estimates developed; and suppliers of materials and services located and asked to offer quotes for their portion of the required work. All of this information would then be catalogued. Ultimately, a firm cost for each manufactured component would be produced. This cost would have incorporated cost reduction considerations resulting from mass production.

For each of the purchased components, as many as three suppliers would be identified. A firm quote would be obtained from each supplier for the subject component in the quantities needed to satisfy the projected market requirements. For each component, its performance specifications, potential

suppliers, their product model numbers and nomenclature, and cost information would be catalogued.

Simultaneously with the hardware cost study effort, a product assembly and testing cost assessment would be performed. These direct costs would be added to the product cost estimate.

To determine the expected selling price, direct overhead and a gross margin (general administrative, selling and profit) were applied to the direct costs. The direct overheat rate selected was based upon the experience and expectations of the team members for the type of manufacturing operation and process used. The gross margin rate was taken from an anlaysis of companies currently producing HVAC equipment.

In order to determine the competitiveness of the gas-fired heat pump, it was necessary to determine the cost of commercially available heating and cooling systems that could provide the same services as the gas-fired heat pumps. Sections 3.0 and 6.0, the Comparative Cost of Ownership Studies for the 7½-ton and LSM systems respectively, describe the methodology used in determining the purchase and operating costs for several competitive systems.

8.3 Summary of Results

The preliminary 7½-ton heat pump design was costed. Since the costing efforts for the interim and final designs were not conducted, it was necessary to make a number of assumptions to extrapolate the preliminary results into a "final" selling price. The preliminary projection of the selling price of the 7½-ton gas-fired heat pump was $22,489 ($2998 per ton). Cost breakdown by components is shown in Table 8-1. A surprising result of the preliminary costing analysis was the extremely high cost for plumbing and control valves. The approach was to specify these items as catalogue purchases. However, due to the pressure, temperature, and flow rates for the steam and the flow rates for the refrigerant, the piping and valves applicable for the system were not standard components. The applicable components were specialty items which were typically very much overdesigned for this application. In developing the "final" cost estimate, an assumption

TABLE 8-1

PRELIMINARY 7½-TON SYSTEM MANUFACTURING COST

BASIC ASSUMPTIONS

Components	Cost
Turbomachinery	$ 1,931
Controls	1,800
Frame and Enclosure	1,358
Heat Exchanger	1,090
Fan and Motors	850
Valves and Piping	4,651
System Assembly and Test	2,560
Steam Generator and Regenerator	2,071
Cost of Components and Assembly	16,311
Mark Up (27.5%)	6,187
Total	$22,498

was made that these components would be made at significantly less cost for the application. This and other cost reduction assumptions were made to yield a "final" selling price projection of $12,383 ($1651 per ton) as shown in Table 8-2. Both prices were shown by the 7-1/2-ton Market Potential Study to be too high for a marketable product.

The conventional alternative to the 7-1/2-ton gas-fired heat pump was the year-round air conditioner (YAC). The YAC had a gas-fired forced hot air heating system and an electrically driven chiller. The YAC could be purchased for approximately $2200 ($300 per ton).

In developing the preliminary and interim costs for the LSM heat pump an assumption was made that the system would be manufactured in quantities of 200 per year. The interim projection of the selling price for the heat pump element (turbochiller package) of the heating and cooling system was $77,331. When the costs of system components, such as the cooling tower and gas steam generator were added to this figure, the grand total was $90,331. The individual components and their costs for the turbochiller package are displayed in Table 8-3.

As explained previously, equipment costs for a number of commercially available alternatives to the LSM system were determined. Two systems were selected as being the most likely alternative that a contractor would consider buying. These were: 1) a standard heating and cooling system; and 2) a standard heat recovery (heat pumping) system. The selling prices for these systems and the LSM as complete central built-up packages are presented in Table 8-4. This table shows that the LSM package had a first cost premium of 65 to 120% of the energy conservation option and the standard option, respectively.

TABLE 8-2

7½-TON MANUFACTURING "FINAL" COST - ALTERNATE ASSUMPTIONS

Components	Cost
Turbomachinery	$1448
Controls	1000
Frame and Enclosure	1000
Heat Exchanger	1000
Fans and Motors	750
Valves and Piping	1500
System Assembly and Test	1280
Steam Generator and Regenerator	1000
Cost of Goods	8978
Mark Up (27.5%)	3405
	$12,383

TABLE 8-3

INTERIM PRODUCT COSTING FOR LSM TURBOCHILLER PACKAGE

Major Subassemblies	Cost
High Speed turbine	$ 5,449
Turbine Lube System	335
Motor (Generator) and Electrical Control (75 hp)	2,027
Mechanical Drive (Gears)	3,333
Steam Condenser/Condensate System	8,336
Water Treatment	3,085
Steam Generator	3,020
Refrigeration System	15,886
Refrigeration Compressor	1,413
Water Circulation	950
Steam Controls	5,465
Refrigerant Controls	2,034
Water Controls	497
System Control Panel	1,984
System Assembly and Test	2,250
Cost of Goods	$56,064
Markup (27.5%)	21,267
Total	$77,331

TABLE 8-4

PRODUCT SELLING PRICE COMPARISON

System Description	Selling Price
Standard	
Single-Bundle Recip	$27,200
Circulating Pumps	1,176
Cooling Tower and Pump	6,510
Gas Steam Generator	5,900
Total	$40,786
Energy Conservative	
Double-Bundle Recip	$34,700
Circulating Pumps	1,176
Cooling Tower and Pump	6,510
Gas Steam Generator	5,900
Hot Water Storage Tank	6,200
Total	$54,486
LSM HSPF	
Turbochiller Package	$77,331
Cooling Tower and Pump	7,100
Gas Steam Generator	5,900
Total	$90,331

9.0 BUSINESS POTENTIAL ANALYSIS

The study and evaluation of the business potential for the introduction of an HSPF gas heat pump product into the HVAC equipment market was conducted by Mechanical Technology Incorporated (MTI). The evaluation described in this section was conducted for the Large Size Model (LSM) HSPF gas heat pump. The methods of analysis and evaluation described could have also been generally applied to the 7½-ton program, but they were not developed for use until after than program had been terminated.

Among the questions addressed by the Business Potential Analysis, the following were probably the most significant.

- How do firms presently engaged in manufacturing and/or marketing HVAC products compare, in a business sense, with firms in other sectors of the economy?
- Why has the HVAC business historically lagged in terms of introducing technically innovative products?
- What kind of firm would be most likely to find a gas heat pump an attractive (profitable, overall) addition to its product line?
- Why is no U.S. firm presently committed to manufacture or market a gas heat pump?
- What kind of added incentive may be needed to generate interest or enthusiasm on the part of manufacturers in a gas heat pump product?

The objective of the study was to identify a business rationale under which a company currently in the HVAC equipment market would invest in and introduce a new product based on the LSM HSPF system. Selection of a company already in this business area was based upon an assessment of the characteristics of the HVAC business and also upon the objective of achieving significant and rapid market penetration. Significant and rapid market penetration was required in order to satisfy DOE's requirement for a significant near-term energy conservation impact.

As the system designer and developer under contract to CNG, MTI had potential business interests in the commercialization of the HSPF gas heat pump. Therefore, MTI conducted the business potential study and evaluation on its own part and on "ex-scope" to the program. While the study did not consittute a deliverable item under CNG's prime contract or MTI's subcontract, CNG and MTI viewed the results as being significant to all parties involved in the development and commercialization of advanced energy-efficient HVAC equipment, and therefore provided them as a part of this report. The study results are presented in summary form in this section and in complete form as Appendix Volume VIII.

The business potential study and evaluation (analysis) was conducted as a part of the overall program to design, develop, demonstrate and commercialize a gas heat pump. The business potential analysis, therefore, interacted with other programmatic activities. Figure 9-1 illustrates the new product planning and development process which was followed in the overall LSM program. The point in the process during which this work was performed is the step termed "Business Analysis". This step had three interrelated tasks, the details of which are shown in Figure 9-2. The LSM market potential study had the objective of defining the product requirements as viewed by the customer, specifications of the market structure and distribution, and market potential. The product requirements provided inputs such as product features, performance, and cost to the product definition task. The product features from the market study combined with the basic technology concept resulted in product requirement such as size of unit, control, modes, distribution, efficiency etc.

These product requirements provided inputs to system and component design efforts and the cost-of-ownership study. In the product definition task, the product design was carried out only to the degree necessary for reasonable performance and manufacturing cost estimates. The cost-of-ownership study had the objective of defining the economic value of a gas heat pump product to the customer. The relationships between these tasks and the balance of the program were explained in Section 4.0 of this report.

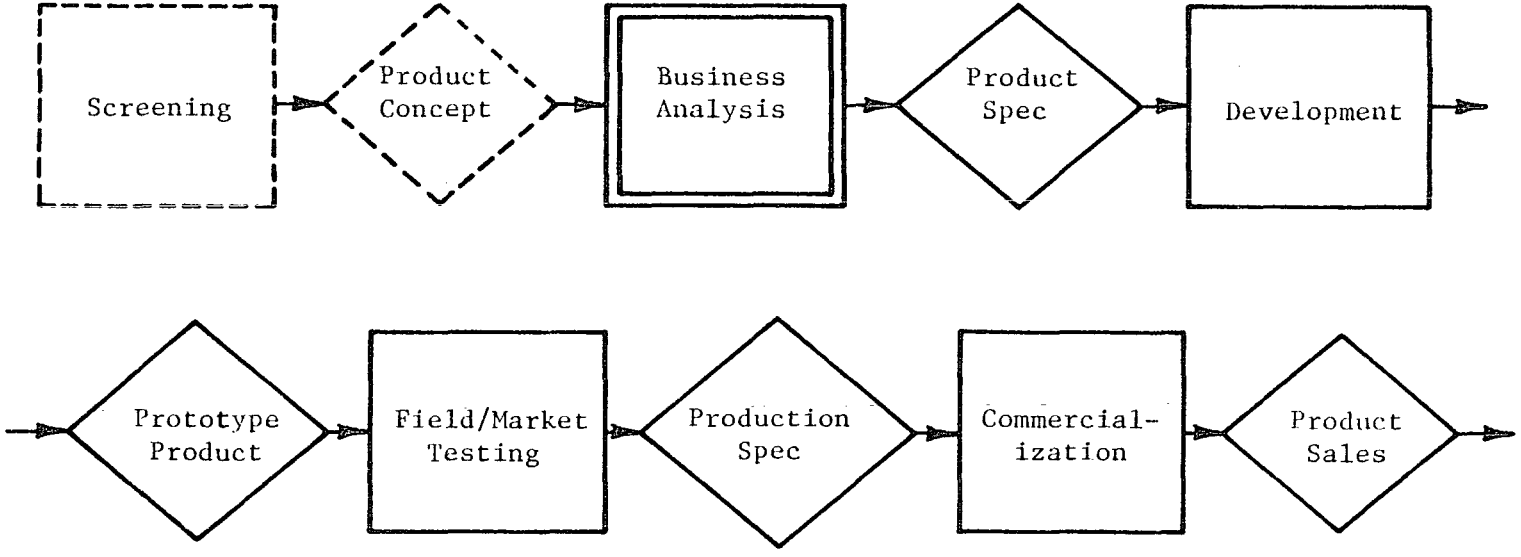

Fig. 9-1 New Product Planning and Development Process

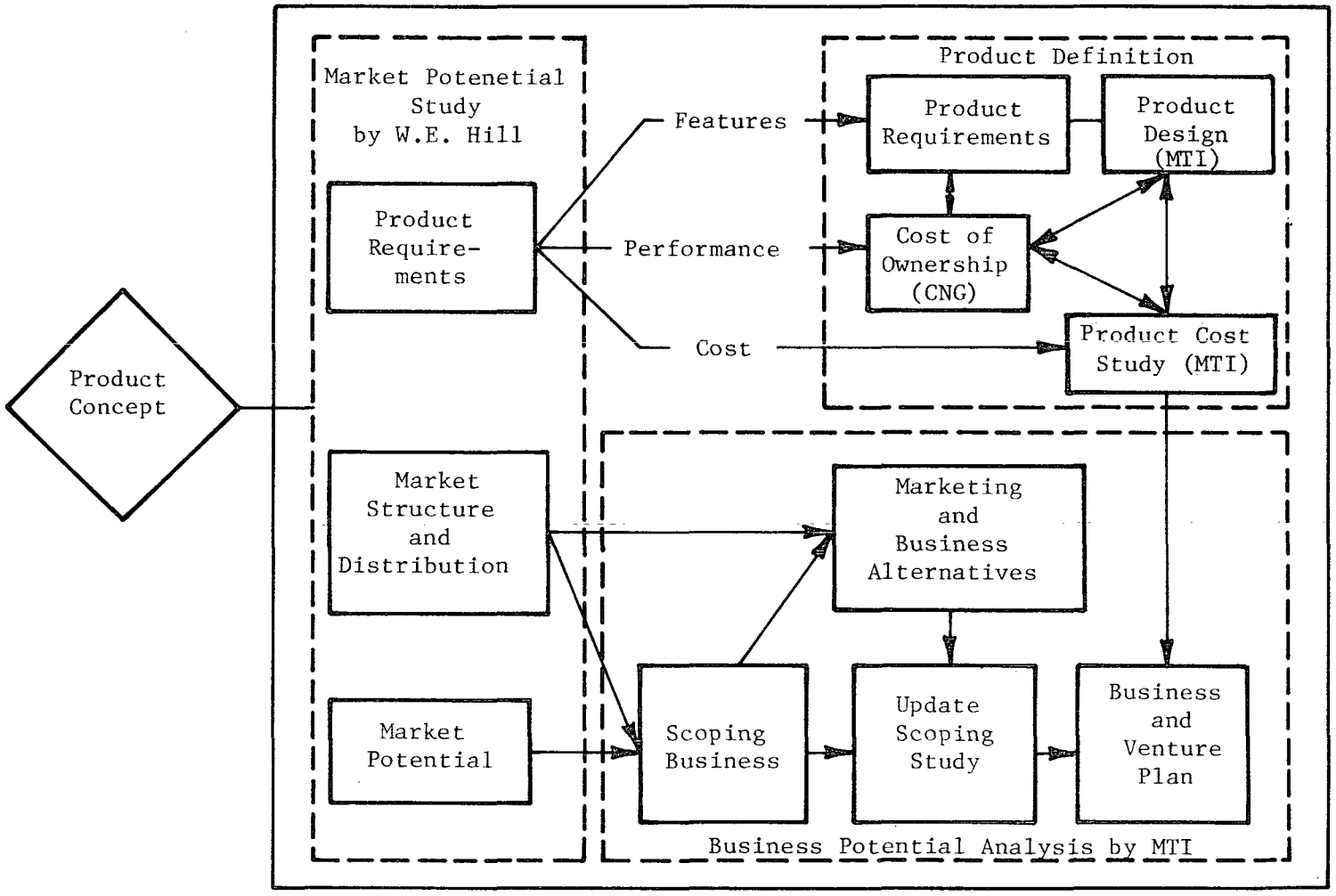

Fig. 9-2 Business Analysis Task Flow and Relationships

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The gas heat pump was viewed as being sold on the basis of energy cost savings. The product was expected to cost more than standard, more energy consumptive alternatives. A cost objective was also defined, as many potential heat pump customers specified operating savings and capital costs as independent requirements. The market potential and product definition tasks provided inputs to the business potential task. The market structure placed constraints on how the product could be sold and, in this case, the type of companies which could best market the product. The market potential combined with the product definition, cost-of-ownership, and business structure defined a sales projection. This projection, combined with required investments and expected profitability, formed the basis for the financial returns to a company marketing this product.

9.1 Requirements of the Business Potential Analysis

The principal objective of the business potential analysis was to define a business scenario based upon an exclusive licensing agreement with a large national firm already serving the HVAC equipment. The objective to maximizing the market potential and need for a firm already in the climate conditioning market were explained previously. In addition, an exclusive agreement was felt to be necessary to protect the firm's investment and to permit an acceptable return on that investment. Further requriements resulted from the hope that this study would be used as a tool to convince businesses of the wisdom of investing in the gas heat pump product. For this reason, the financial conditions typical of firms in the HVAC business, especially the firms who would be the most desirable licensees, were studied in the analysis. The applicable financial conditions were derived from an annual report and Securities and Exchange Commisions (SEC) Form 10-K of the candidate companies.

Because the business potential anlaysis and the product definition task were conducted concurrently, information that could be uniquely associated with the gas heat pump product was not concurrently available. Therefore, a generalized advanced climate conditioning product was assumed that would serve the same market as the LSM HSPF gas heat pump. This generalized product was assumed to have costs, profits and investments in the same ratio to sales revenues as the average product of each of the companies

investigated. In this respect, the results and conclusions of the market potential analysis were not unique to a gas heat pump, but were typical of any new product in the HVAC or more generally the climate conditioning business.

The financial characteristics of the companies considered together with the market potential. These data were then used in a simplified financial model of the firm to evaluate the economic returns of developing and introducing the gas heat pump product. The evaluation effort included an analysis of the impact of variations in profit margins, fixed plant investment requirements and sales growth. These sensitivity analyses were based on the basic business potential situation of companies currently in the HVAC business. As an alternate scenario, an entreprenuerial start-up venture was evaluated. The following sections discuss the methodology used, the results and conclusions.

9.2 Business Potential Methodology

The basic approach to the business potential analysis was to use a simplified model of the firm to project profitability, cash flows, and investment levels over an evaluation period through 1990 using the baseline market penetration projection developed in Appendix Volume VIII. This model was applied using financial data from four firms in the HVAC industry: two integrated manufacturers (referred to as companies A and B), a component supplier (company C), and an assembler (company D). The financial data on these companies were derived from their 1978 annual reports and SEC 10-K forms. In applying the financial data, two principal assumptions were made: 1) a simple linear relationship between gross sales, profitability and investments exists, and 2) the historic financial data for a whole line-of-business for a company, as reported under SEC rules, is characteristic for a single new product innovation. The first assumption was a standard business planning analysis assumption asn was usually very acceptable. For the four companies selected, their ten-year financial data shoed a high degree of correlation with sales, as will be shown later. The second assumption may not have been as accurate. The desired approach would have been either to have used a specific existing product that was identical in function or to have built up a product financial picture for the innovation. The former

approach was not feasible since specific product line financial data are highly proprietary. The latter approach was not feasible due to the lack of specificity of the product innovation at the point in time this analysis was performed. The only approach that would be accurate to any degree and that would give an indication of the impact of different approaches to the business was to define an "average HVAC product" as characterized by the overall financial position for the complete HVAC line-of-business. Use of these average HVAC product financial characteristics made the results of this analysis applicable to the HVAC industry in general.

Figure 9-3 presents the flow of information in the model of the firm used in the anlaysis. The market potential, market penetration, baseline sales projection, sales price consideration, and the impact on market acceptability of the price premium have all been discussed in previous sections. Three factors were derived from the financial reports of the four companies considered.

For the bulk of the analysis, the nonreoccurring costs were considered as zero. The implications of this assumption are that there are no development costs, no introductory advertising or promotional costs, no manufacturing dislocations, no royalty expense, or not other costs normally experienced in the development and introduction of a new product. These implications made the analysis very optimistic in that if a business decision could not be justified on this basis, it could never be justified.

The financial data for the four companies was correlated to sales through the use of a linear regression or least-squares fit. Those financial data which were normally present in the ten-year financial summary (i.e. cost of goods, net operating margin, working capital, and net plant) correlated very well with gross sales revenues. Typical regression correlation values (i.e. r^2) were around 0.99 and always over 0.97. A perfect linear correlation would have a correlation value of 1.0. Therefore, a great deal of confidence was placed in the assumption of a linear relation with sales. For those values of interest which had only a two-year data base, (i.e. gross investment in land, buildings and equipment), an average ratio to sales was determined. The specific quantities that were selected to be inputted in

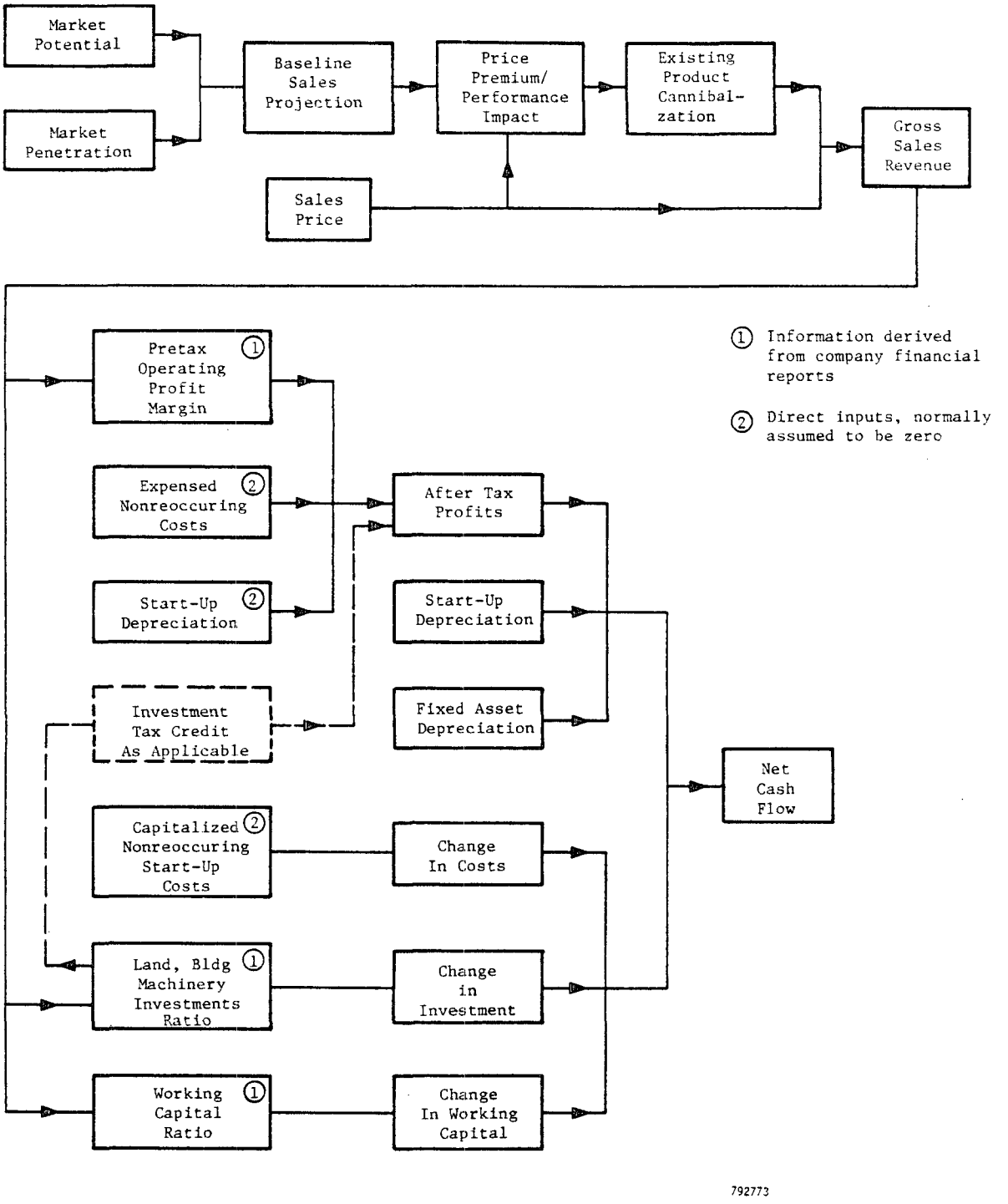

Fig. 9-3 Schematic Flow of Simplified Model of the Firm

the model were the gross operating margin, the working capital, the gross investment in land, the gross investment in buildings and the gross investment in equipment. The results of the data correlation for those quantities for the four companies are shown in Table 9-1.

The overall financial data were adjusted to reflect the "air conditioning" line-of-business for companies A, B, and D. The line-of-business data presented in the annual reports and 10-K forms permitted a direct adjustment to gross margins and an implied adjustment to the fixed asset ratios. The line-of-business information presented only the total assets applied to a line-of-business. It was assumed that each of the asset items of interest would vary from the total company data to the line-of-business, directly with the total assets applied.

Since the period of evaluation covered only 1979 through 1990, a method to determine the residual value of the business beyond 1990 was needed. One approach would be to determine a market value for the business in 1990. Theoretically the market value would be identical to the present value of future earnings of the company. The typical mode expressing a market evaluation would be the ratio of stock to earnings per share. Under less depressed conditions, the stock market would normally value a company at a price-earnings ratio of ten. Therefore, the after-tax earnings of the new product business in 1990 were multiplied by a factor of ten to define a proxy for the present value of the future income. The impact of such a valuation procedure is shown in Figure 9-4. A more conservative valuation would be liquidation value of the working capital and fixed assets. The former approach was chosen since it would be more reflective of an on-going business and give a more favorable financial result.

There are two financial criteria that were considered in the evaluation of the business potential of the LSM product. The first criterion was cumulative cash flow. The second criterion was discounted cash flow (DCF) or internal rate of return (IRR). Since the models used were based upon operating profits, the method of financing did not impact the analysis, thus obviating return on equity as an evaluation criterion. Also, debt leveraging could give attractive returns on equity for a poor investment. Other

TABLE 9-1

COMPANY FINANCIAL DATA USED FOR MODELLING

	Integrated Manufacturers		Component Supplier	Assembler
	Company A	Company B	Company C	Company D
Operating Profit	Sales * 0.1001 -25726	Sales * 0.0527 +18923	Sales * 0.1289 -7197	Sales * 0.1054 -2453
Working Capital	Sales * 0.3652 +26230	Sales * 0.2936 +25534	Sales * 0.1515 +4285	Sales * 0.2189 +2727
Fixed Assets				
Land	Sales * 0.0091	Sales * 0.0071	Sales * 0.0069	Sales * 0.0057
Buildings	Sales * 0.0976	Sales * 0.1563	Sales * 0.0620	Sales * 0.0802
Equipment	Sales * 0.2362	Sales * 0.1853	Sales * 0.1920	Sales * 0.1897
Operating Profit Margin (average of two year's of data)	0.0906	0.1053	0.1161	.0893

Note: Sales are expressed in thousands of dollars

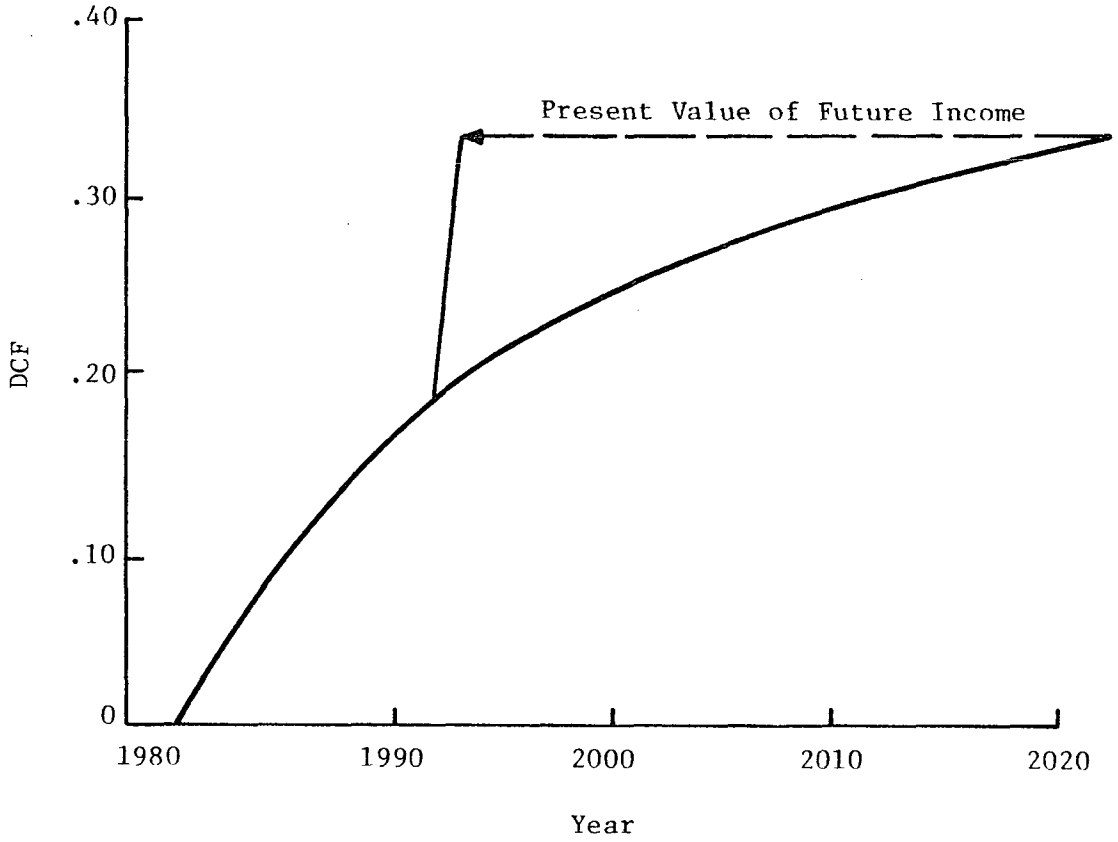

Fig. 9-4 Illustration of the Effect of a 1990 Valuation of the Future Income Upon DCF

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investment criteria were felt to be of interest only if the cash flow and DCF criteria were attractive. The evaluation did not consider nonquantitative financial factors such as overall market perspective of the company or factors such as technical risk of current product obsolescence, goverment regulation or earnings per share criteria.

9.3 Business Potential Analysis

The financial data from four companies in the HVAC industry were used in a simple model of the firm to investigate the business potential for these firms in order to introduce an innovative LSM HSPF heat pump as an add-on product. The financial data used were based on historic performance of each of the four companies. Nonreoccurring costs normally associate with new products such as R&D expenses, product introduction marketing expenses, and production costs variances, were not included, except as noted later. No royalty or licensing fees were considered. The analysis was conducted using constant 1979 dollars. No assumption was made as to the capital structure to support the financing of the new product. The specific approaches and the relation to the companies selected are shown in Table 9-2.

The results of this baseline calculation showing the cumulative cash flow and the product line business DCF with a continuing business valuation in 1990 were determined. Figure 9-5 is typical of the results for all four companies. The results showed that the cumulative cash flow generation through the business became more and more negative in all cases. The only reason for the existence of a DCF was the valuation of the product line as an ongoing business at a price of ten times after-tax operating income. Considering the business results, it was highly unlikely that such a valuation would actually be obtained.

The scenario assumed did not yield a business potential that would be attractive under any reasonable circumstances.

The sensitivity analyses to the baseline cases were justified partially upon the following logic. A reduction in the investment requirements in fixed assets was justified since excess plant capacity in the HVAC industry appeared to exist. This evaluation was purely subjective and may not have

TABLE 9-2

BUSINESS POTENTIAL ANALYSIS PATHS TO COMMERCIALIZATION CONSIDERED IN BASE ANALYSIS

Path No.[1]	Description	Companies Considered
2	Existing Integrated HVAC Manufacturer	A Large Integrated Broad-Based HVAC Manufacturer B Large Integrated Commercial HVAC Manufacturer
3	Existing HVAC Component Manufacturer extending to be an integrated manufacturer	C Large Component Supplier to full-size range of air-conditioning equipment
4	Existing HVAC Assembler extending to be an integrated manufacturer	D Large Assembler of commercial sized HVAC equipment

[1] See also Table 3-1, Appendix Volume VIII.

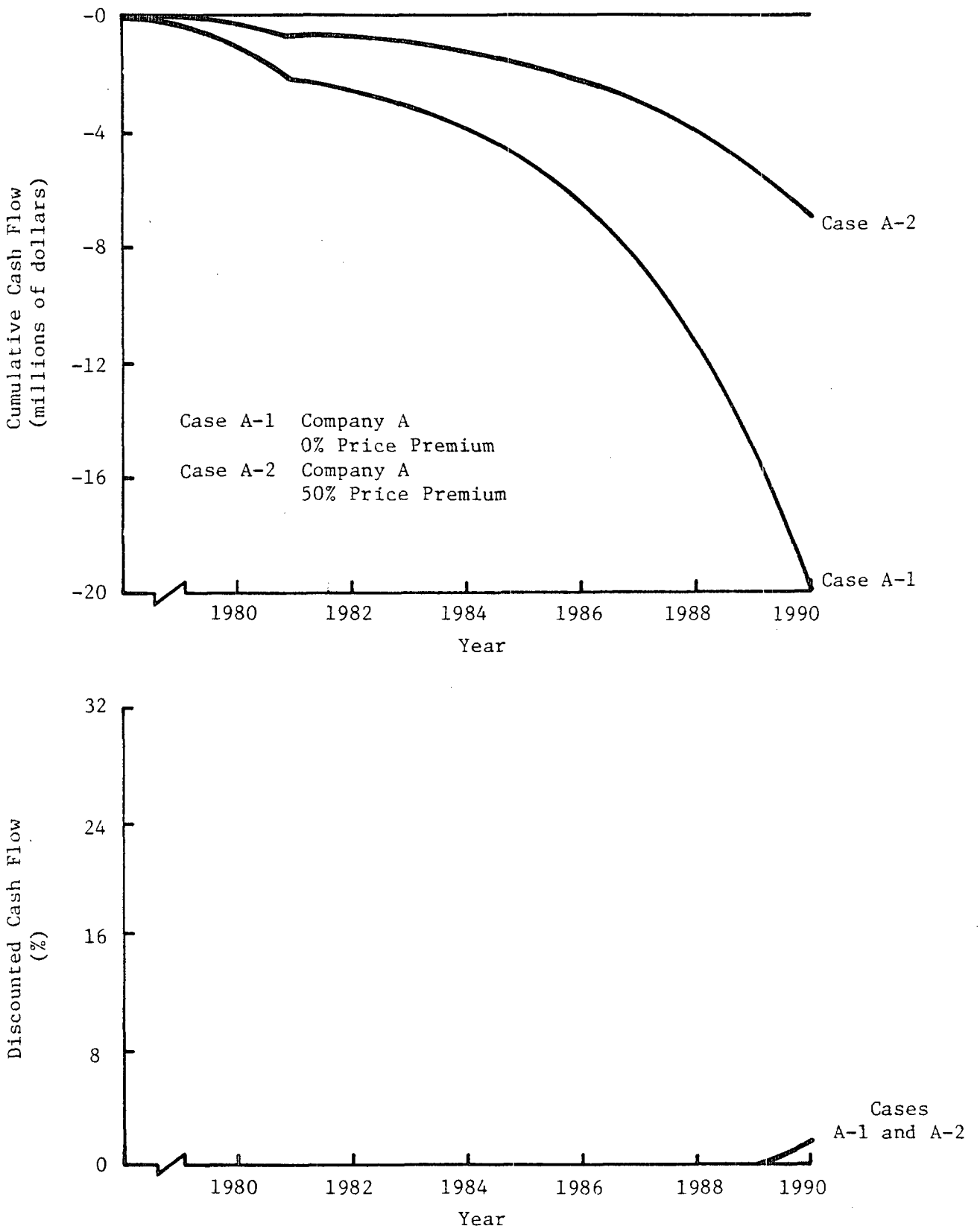


Fig. 9-5 Cumulative Cash Flow and DCF for Company A with Baseline Business Scenario, Typical for All Companies

been true for any of the four firms selected. However, if the supposition was true, significant increases in production could have been realized with very little increase in fixed assets. Therefore, the fixed asset requriements were reduced by 50% and 100% to evaluate the impact on profitability.

Profit margins were observed to be low in the HVAC industry due to competition and undifferentiated nature of the product. However, if a company exclusively had a differentiated product that offered a performance advantage, that company should be able to command a higher profit margin. Therefore, profit margins of 15%, 20%, and 25% were evaluated. This evaluation was made under the assumption that the cost of manufacture of the new product allowed increased profit margin within the constraints of the customer's sensitivity to premiums and operating advantages. From the results of the equipment cost study of Section 8.0 and the LSM cost-of-ownership study of Section 6.0, the possibility for profit margin increases would seem to be nonexistent for a product based upon the LSM.

A final sensitivity was a combination of the above, wherein the investment was reduced by 50% and the profit margin was increased to 15%. Table 9-3 summarizes the sensitivity analyses parameter variations. The resultant cumulative cash flows and DCF's for Company A are presented in Figures 9-6 through 9-8. These results were typical for all companies.

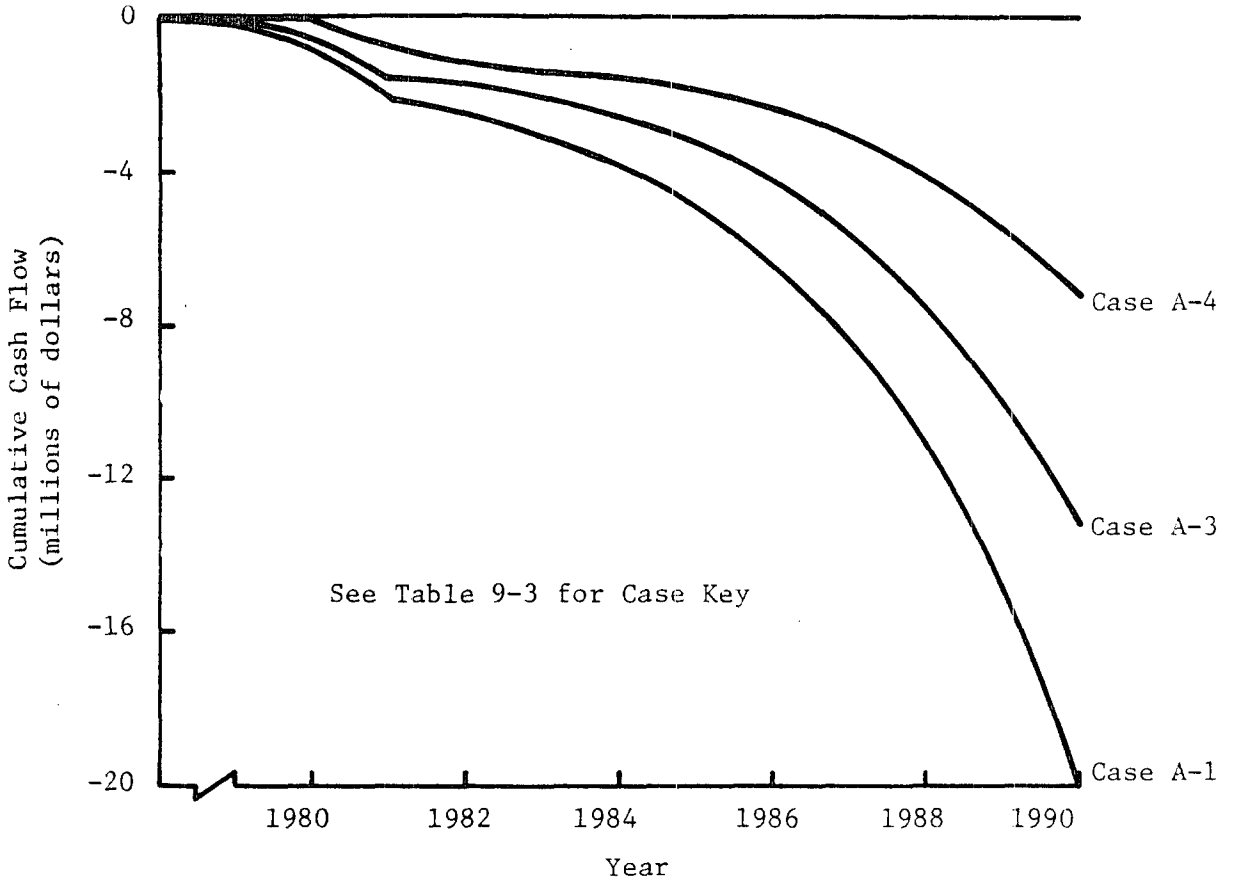
The basic conclusion of this analysis was that companies in the HVAC industry could not justify the increased sales as projected for the LSC product with reasonable values of capital investment requirements and profit margins. This conclusion was based upon a quantitative financial analysis including the cash flows due to operations and the residual value of the business at the end of the evaluation period. The influence of changes in investment requriements and profit margins did not change the conclusion. The only variable which was not changed in these anlayses was the sales projection. Other analyses suggested that the sales growth rate and sales objectives had a significant impact on the outcome of the quantitative analyses. The sensitivity to sales growth was examined next.

TABLE 9-3

SUMMARY OF PARAMETER VARIATION FOR SENSITIVITY ANALYSIS FOR BASELINE BUSINESS SCENARIO

Variable	Run Number							
	1	2	3	4	5	6	7	8
Price Premium	0%	50%	0%	0%	0%	0%	0%	0%
Profit Margin	(1)	(1)	(1)	(1)	.15	.20	.30	.15
Land Ratio to Sales	(1)	(1)	0	0	(1)	(1)	(1)	0
Building Ratio to Sales	(1)	(1)	.5x(1)	0	(1)	(1)	(1)	.5x(1)
Equipment Ratio to Sales	(1)	(1)	.5x(1)	0	(1)	(1)	(1)	.5x(1)
Working Capital Ratio to Sales	Historic Company Data (see Table 5-1)							
Sales Volume	Baseline Projection (see Table 6-2)							

(1) Historic company data (see Table 9-1)



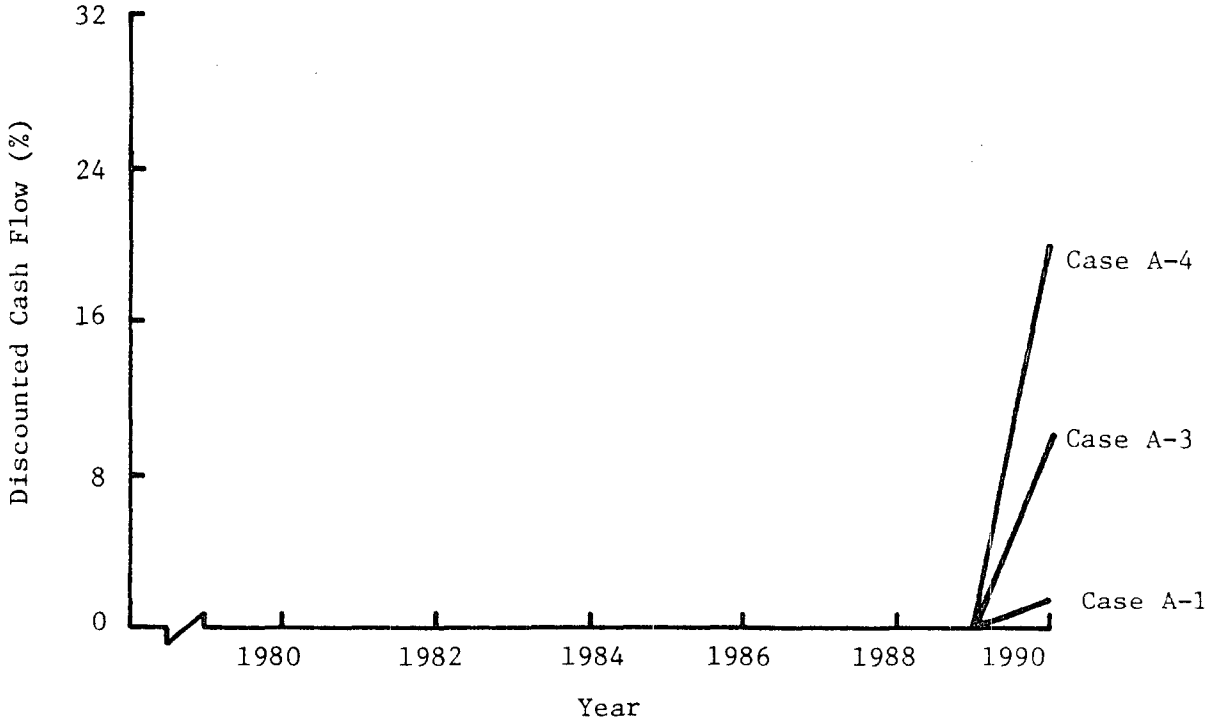


Fig. 9-6 Impact of Reduced Fixed Investment on Cumulative Cash Flow and DCF for Company A with Baseline Business Scenario

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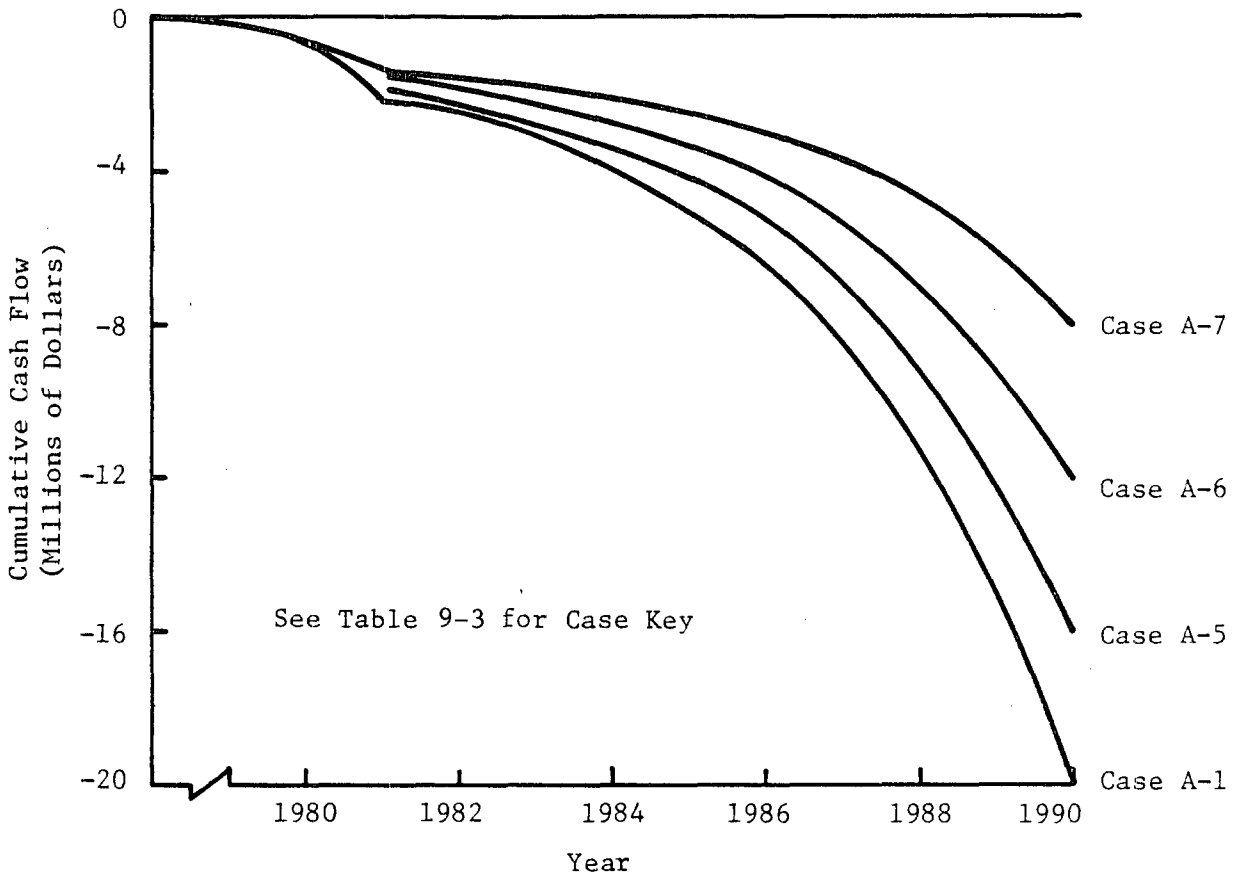

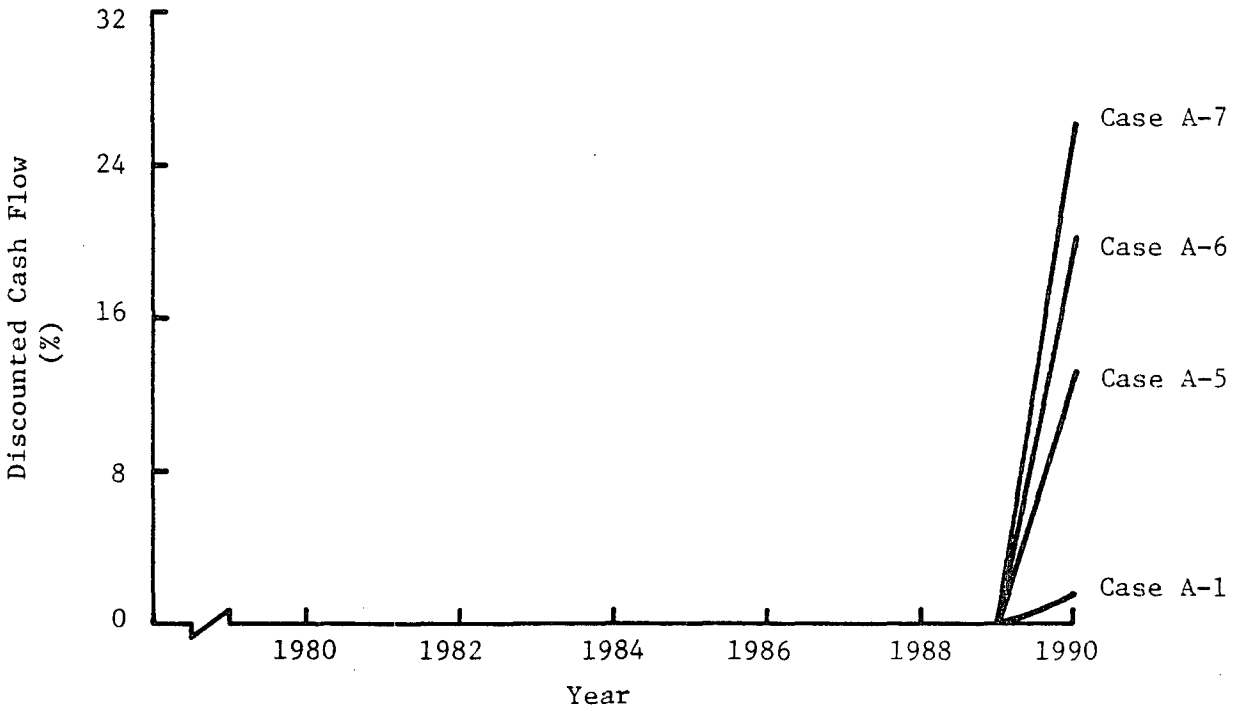

Fig. 9-7 Impact of Improved Profit Margins on Cumulative Cash Flow and DCF for Company A with Baseline Business Scenario

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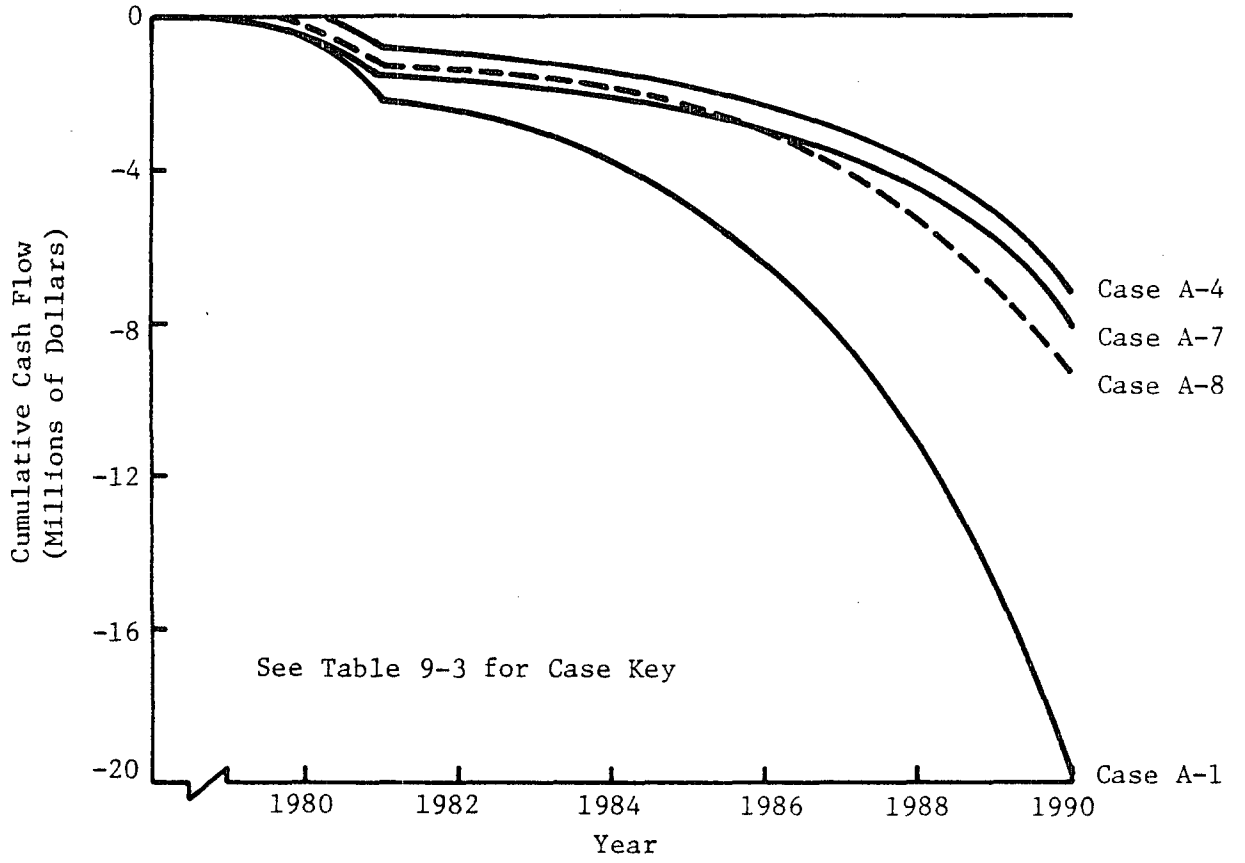


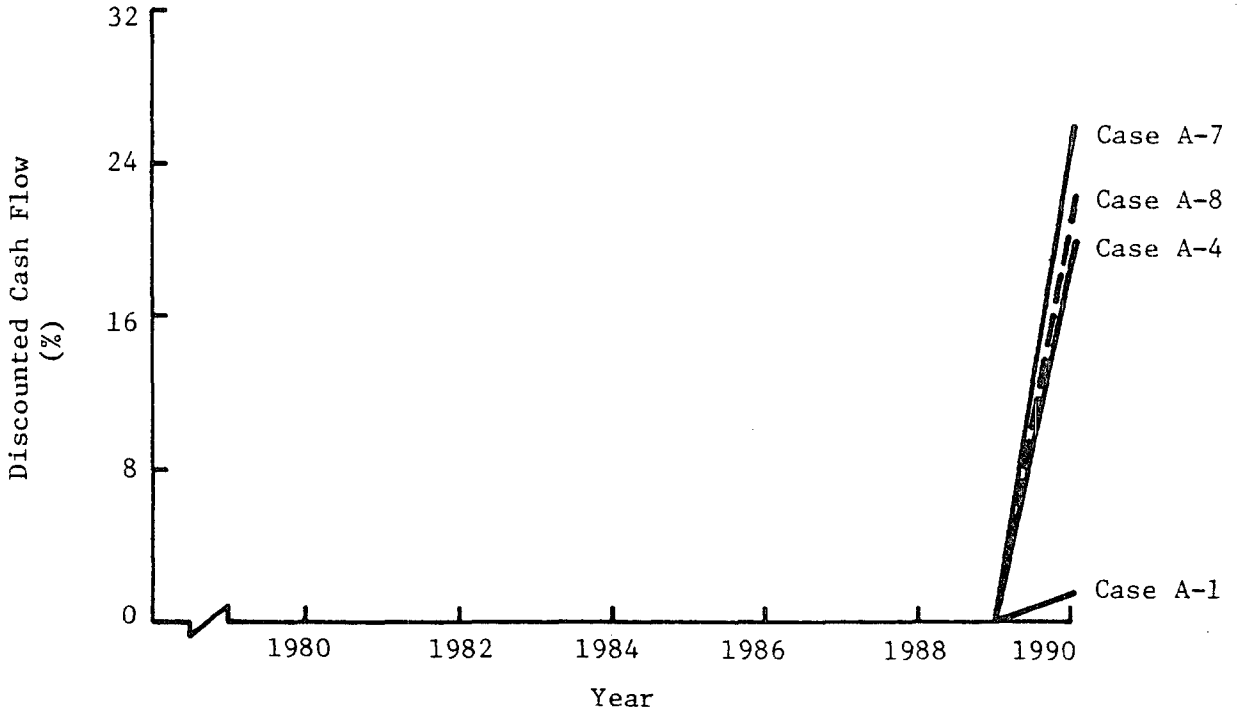


Fig. 9-8 Impact of Combined Changes on Cumulative Cash Flow and DCF for Company A with Baseline Business Scenario

The impact of various sales growth scenarios was tested with a hypothetical HVAC company. The values of profitability, working capital, and fixed asset accounts were selected to approximate the best that might be expected for such a company as presented in Table 9-4. An improved profitability was assumed for the hypothetical company. Since no start-up costs were assumed, the actual level of sales or cash flows was not important. However, the growth patterns and trends were important. The base sales were grown at a number of different constant rates of growth over the entire evaluation period until 1993. Three ramp type growth rates were also evaluated. In the rapm sales scenario, the sales were grown over five years at the rate necessary to equal values achieved by a constant growth rate through 1993. After the five-year accelerated growth, sales were then kept constant. The cases considered are presented in Table 9-5. Also noted in Table 9-4 is the value of growth that would show a break-even point in the cumulative cash flow as predicted by the relation developed in Appendix Volume VIII.

Figure 9-9 presents the values of DCF with and without valuation and the cumulative cash flow in 1990 as a function of sales growth rate. This figure clearly shows the impact of sales growth rate. Both the cumulative cash flow and the DCF without valuation decreased with increasing sales growth.

The DCF with the end point valuation, however, increased slightly with growth rate. This apparent anomaly was the result of the fact that the end valuation depended on the profit in 1990. At higher growth rates, the profits were higher and the valuation was higher in dollar terms. This increase offset the increased negative cumulative cash flow in such a way that the DCF remined about constant.

Figure 9-10 presents the cumulative cash flow and DCF without valuation for the constant growth and the ramp growth cases. The ramp growth was considered equivalent to the average annual growth rate in sales from 1981 to 1993. The data in Figure 9-10 are extremely interesting. While, as seen in Figure 9-9, the constant growth rate scenario shoed that a growth rate beyond about 21.8% resulted in no return, the ramp growth scenario did not show any maximum growth rate restrictions. The conclusion reached was that from a DCF basis, a ramp growth was the preferred strategy. However, the

TABLE 9-4

COMPARISON OF FINANCIAL RATIOS
USED FOR HYPOTHICAL FIRM

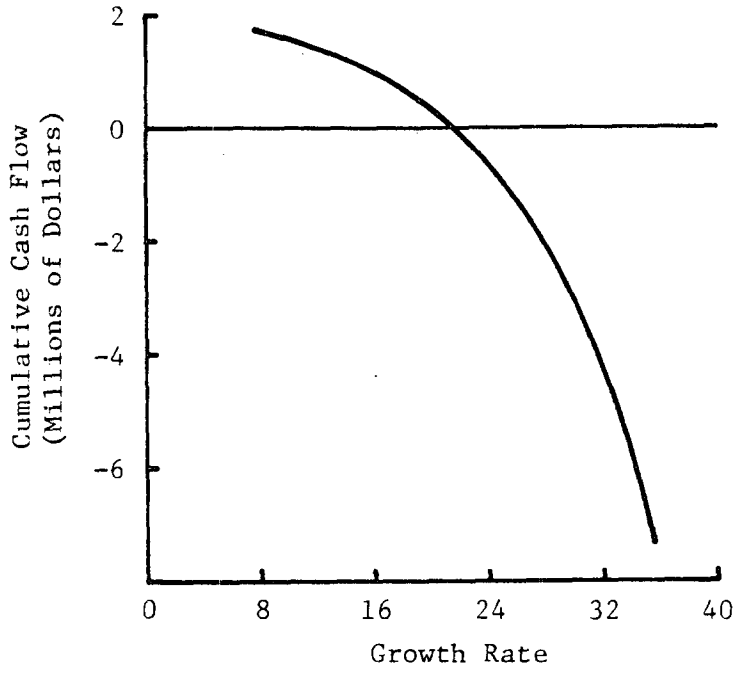
Financial Ratio	Company				
	A	B	C	D	Hypothical E
Pretax Profit Margin	10.01%	10.53%	12.89%	10.54%	15.0%
Working Capital Ratio	36.52	29.36	15.15	21.89	20.0
Fixed Asset Ratios					
Land	0.91	0.71	0.69	0.57	0.7
Buildings	9.76	15.63	6.20	8.02	8.0
Equipment	23.62	18.53	19.20	18.97	20.0
Maximum Theoretical Allowable Growth Rate *	7.93	9.32	19.41	12.46	19.07

* Depreciation cash flow is ignored in these values. See Appendix Volume VIII for derivation.

TABLE 9-5

SALE GROWTH RATE SCENARIOS CONSIDERED

Type	Values	Comment
Constant	7.5%, 10%, 12.5%, 15% 20%, 35%	Simple Growth Rate for Break Even Cumulative Cash Flow is 19%
Ramp	55% for Five Years, Then Zero	Equivalent Sales in 1993 to 20% Constant Growth
	40% for Five Years, Then Zero	Equivalent Sales in 1993 to 15% Constant Growth
	25% for Five Years, Then Zero	Equivalent Sales in 1993 to 10% Constant Growth



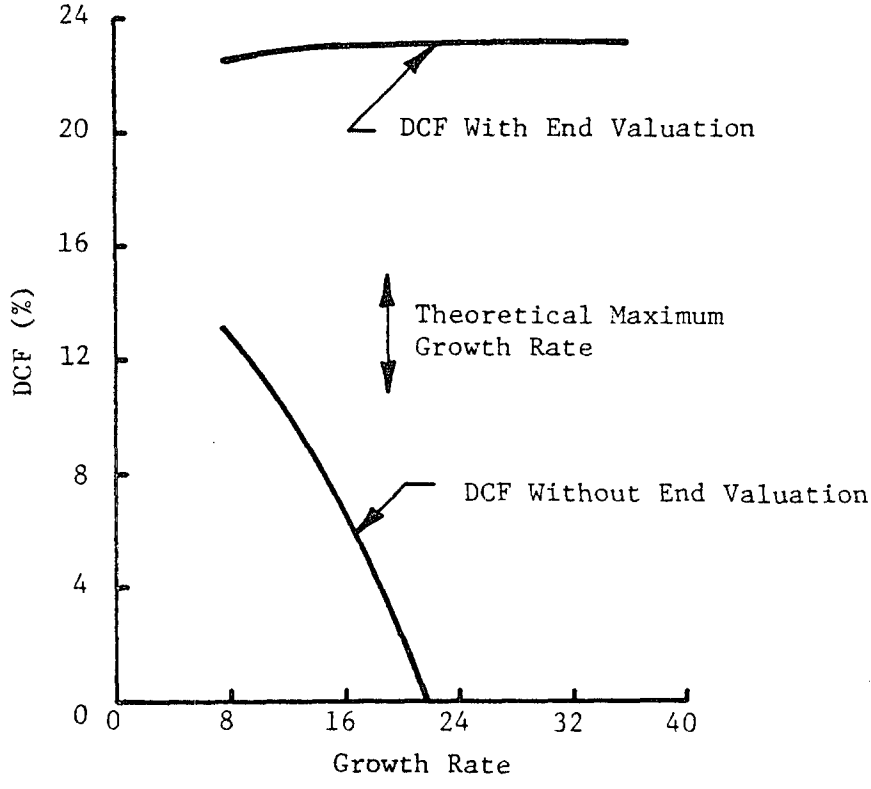


Fig. 9-9 Variation of 1990 Cumulative Cash Flow and DCF with Sales Growth Rate

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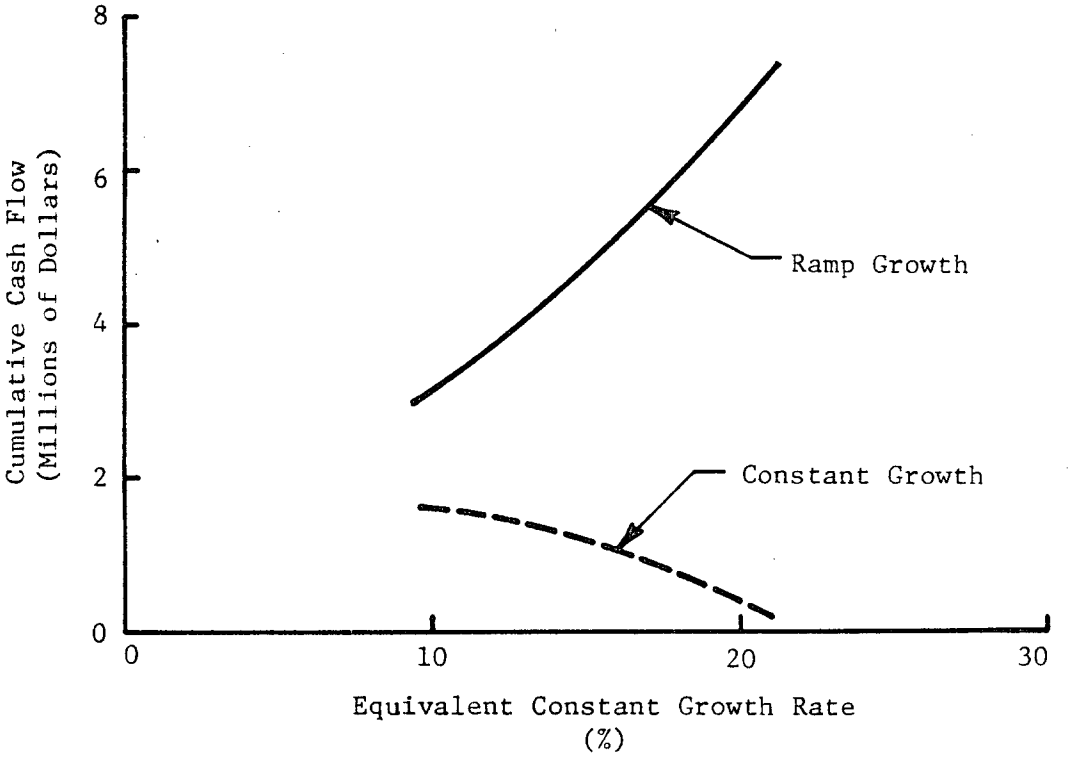

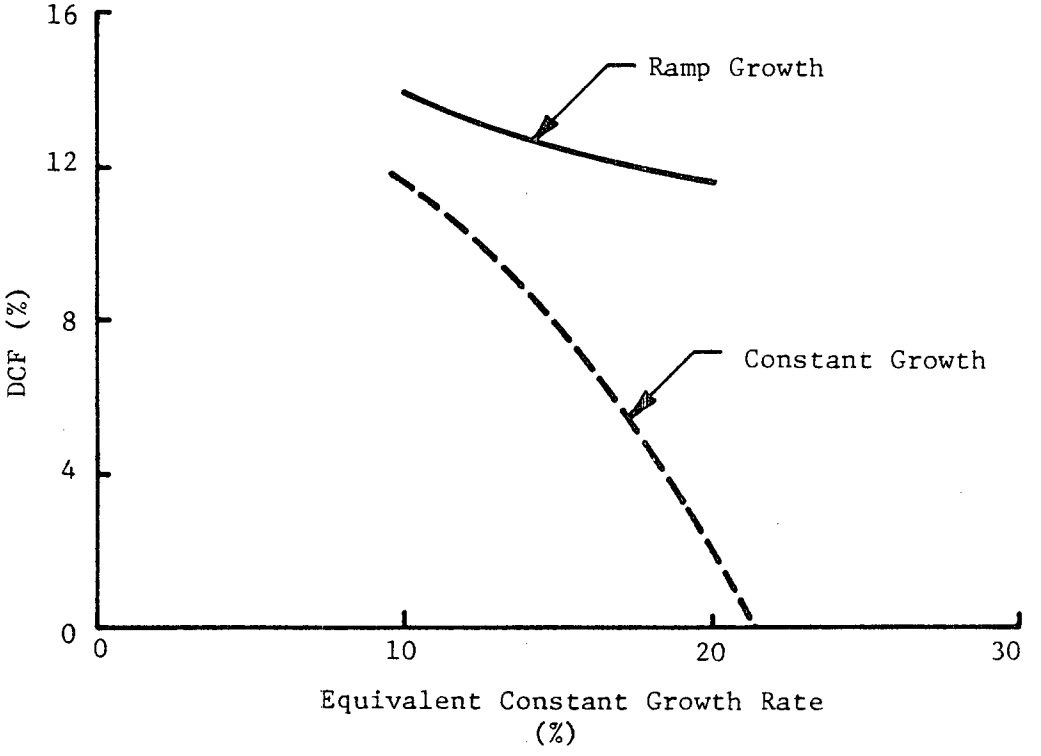

Fig. 9-10 Impact of Variations of Growth Scenarios on Cumulative Cash Flow and DCF without Valuation in 1990

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risks associated with this strategy were great since the maximum negative cumulative cash flow was so much greater.

The sales growth strategy may have been the most dominant variable in the determination of the business potential of a new product. In the case of the LSM, there was no indication that the market would permit a ramp growth or that any company would accept the financial risk. Therefore, expansive growth variations were not considered further.

An entrepreneurial venture, as part of an existing corporation, was evaluated on a start-up basis. This venture had extremely limited sales objectives (100 units per year) and used a "boot strap" approach. The analysis showed that this entrepreneurial approach could be economically attractive if there were no development costs. This approach was not considered feasible.

9.4 Business Potential Conclusions

A quantitative business potential analysis showed that under historic financial conditions, the HVAC industry could not make large investments in revolutionary product innovation and/or significant market share changes resulting in a high and sustained growth rate for one company. (See Table 9-6). A number of very important qualifiers existed in this statement.

First, only the HVAC industry was addressed. Second, the historic financial ratios of four specific firms in the HVAC industry were used. These firms were selected because they met the qualifications specified for the exclusive licensee for the LSM technology. While ther may have been some HVAC component manufacturers that may have had financial characteristics more typical of a different industry (such as instrumentation or control manufacturers), the four companies selected appeared to be typical of the HVAC industry as a whole. If there was any bias, it would have appeared that as the four companies had better financial characteristics than the average HVAC company.

Third, the evaluation criteria used were based a quantitative financial analysis. Specifically, the project DCF was the principal determination of financial viability.

TABLE 9-6

BUSINESS POTENTIAL CONCLUSIONS

Quantitative Financial Business Analysis(1) shows that under Historic Conditions(1), a company in the HVAC Industry(1) cannot afford:

- Investment in Revolutionary Product Innovation
- Significant increase in market share from a high and sustained growth.

(1) Important qualifiers to the conclusion - see text

Fourth, while the analyses basically ignored the nonreoccurring developments and start-up costs, the present value of cumulative cash flows that would have been available to offset start-up costs were small with reasonable discount rates. From MTI's experience, the development of a revolutionary gas or electric heat pump required a minimum of $10 million and three to five years. None of the scenarios could accommodate the development costs of a revolutionary new product. The final qualification related to growth rates. Many smaller growth-oriented companies have had corporate sales growth objectives of 20% to 30% per year. Many new companies or product lines have had even higher growth rates. In fact, a recent study by the Harvard Business School(1) suggested that if a new business venture did not aim for a rapid movement into the market, its chances for ultimate success were very much reduced.

The four companies studied simply could not support even modest overall growth rates. These qualifications arose from the specifications felt necessary for the ultimate cussess of the LSM introduction. However, all the qualifications that appeared necessary for the LSM combined to produce a business scenario that was an investment disaster.

The business potential analysis showed that by relaxing some of the constraints upon the baseline business scenario and/or by assuming some different financial characteristics, it was possible to develop scenarios that were marginally attractive. If, for example, the vast bulk of product development were paid by somone else, an entrepreneurial firm could bring out a new product. Also, the HVAC companies considered could achieve reasonable financial results of their sales objectives were managed. However, in this event, even they would need a benevolent investor to support the development costs. Table 9-7 lists those factors which would have to be changed if a quantitative business analysis was to show a positive investment criteria.

An alternative was that the decision could be made on a basis other than a quantitative financial analysis, as for example, the threat of product

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(1) Biggadike, R., "The Risky Business of Diversification" Harvard Business Review, May-June 1979, pp. 103-111.

TABLE 9-7

FACTORS NEEDING CHANGE

Changes needed to make an HVAC innovation financially attractive under a quantitative financial analysis:

- Higher profit margins
- Lower fixed plant investments
- Lower working capital requirements
- Minimization of development costs
- Managed sales growth

obsolescence. Also, goverment regulation of product performance, fuel use policy, and energy price policy could significantly impact the decision process.

This analysis clearly pointed out the reasons for the lack of product innovation in the HVAC industry. It was not economical for manufacturers to innovate. The analysis also indicated that there could be a shakeout in the industry. A number of the smaller integrated firms and assemblers could be badly hurt by any business slowdown or a forced shift in their product design. Recent sales of HVAC divisions by companies whose principal business was not in the HVAC business (e.g. Chrysler's Airtemp, General Motor's Frigidaire and Sindstrand's compressor division) suggested a confirmation of these observations. Whether a product innovation came through government action or noneconomic business decisions, it would seem highly probable that a product innovation would occur. If such an innovation should occur, the impact on the HVAC industry would not necessarily be positive.

The ultimate commercialization of any new innovation in the HVAC industry would be difficult. The baseline business scenario which was characteristic of the conventional approach would not be successful. As much innovation would be needed in the commercialization approach as in the product itself. Therefore, due to the nature of this development, the government would seem to be the only entity capable of initiating an innovative commercialization approach.

ATTACHMENT A

LISTS OF SUBCONTRACTORS, CONSULTANTS, AND PRINCIPAL VENDORS

TABLE A-1

HSPF GAS HEAT PUMP PROGRAM SUBCONTRACTORS

Subcontractor	Address	Program Role
Advanced Mechanical Technology, Inc.	141 California St. Newton, MA 02158	LSM design-for-cost effort for steam circuit components and burner
Masco Corporation	21001 Van Born Rd. Taylor, MI 48180	Production cost estimates for 7½-ton unit components and subassemblies.
Mechanical Technology Incorporated	968 Albany-Shaker Rd. Latham, New York 12110	Primary responsibility for technical demonstrations, venture analysis (business potential study); supporting data for Market Potential and Cost of Ownership Studies. Preferred position as technology licensee.
Philips Engineering	721 Pleasant Street St. Joseph, MI 49085	7½-ton condensate return pump.
Scientific Energy Systems Corporation	570 Pleasant Street Water Town, MA 02172	7½-ton steam generation system and regenerator.
William E. Hill & Company	640 Fifth Avenue New York, NY 10019	7½-ton and LSM Market Potential Studies.

TABLE A-2

HSPF GAS HEAT PUMP PROGRAM CONSULTANTS

Consultants	Address	Program Role
W.B. Bryan	Case Western Reserve University, Cleveland, OH	Heat exchanger designs, 7½-ton unit.
Byers, Urban Klug, White & Partners	300 Keith Bldg. Cleveland, OH	Building HVAC load and HVAC equipment simulation, LSM Comparative Cost of Ownership Study; HVAC plans for the Mountwood Park Lodge.
Controlled Environmental Systems, Inc.	16712 Westfield Ave. Cleveland, OH	Cost for electrical service connections for competitive equipment, 7½-ton Comparative Cost of Ownership Study
R.G. Fleischman	1030 Indiana Road Verona, PA 15147	Cost of competitive HVAC equipment, and building HVAC distribution systems, LSM Comparative Cost of Ownership Study.
Charles J.R. McClure and Associates, Inc.	7616 S. Big Bend Blvd. St. Louis, MO 63119	LSM System Component Cost
R.F. Merriwether & Associates	Northwood Executive Building San Antonio, TX	7½-ton and LSM refrigeration loop design and component selection. Unit layout and enclosure design options.
T.H. Parrish	6914 Triola Houston, TX	HVAC equipment modelling; E-Lube program modifications, 7½-ton Comparative Cost of Ownership Study.

TABLE A-2 (Cont'd)

Consultants	Address	Program Role
D. Schwartz	Brayton Road Esmond, RI 02917	Steam generation system trouble-shooting, 7½-ton unit, steam generation system design alternative, LSM.
A.J. Stromquist	421 Madison Avenue Cuyahoga Falls, OH 422	Critique of 7½-ton turbine and nozzle designs.
R.P. Welden	1472 E. Broad Street Westfield, NH 07090	Steam circuit chemistry and water treatment.

TABLE A-3

HSPF GAS HEAT PUMP PROGRAM PRINCIPAL VENDORS

	Address	Program Role
Dyna Technology	Sioux City, Iowa	7½-ton unit motors.
Modine Manufacturing Company	1500 Dekoven Avenue Racine, Wisconsin 53401	7½-ton unit heat exchangers.
Robinson Industries	Box 100 Zelienople, Pennsylvania 16063	7½-ton unit fans
Truton Corporation	Madison Heights, Michigan	7½-ton unit turbine wheels and nozzle

ATTACHMENT B

TASK ORGANIZATION AND PROGRAM MILESTONES

TABLE B-1

LARGE SIZE MODEL PROGRAM TASK ORGANIZATION

DOE-Assigned Program Task Number	Report Section Containing Full Task Description	Task Description
First Phase, 10-77/4-79		
II-3	6.0	LSM Comparative Cost of Ownership Study
III-8	7.0	LSM Market Potential Study
VI	5.0	LSM System Integration
VII-1	4.0	Field Test Planning
VIII	8.0 & 9.0	Business Potential Study and Product Cost Estimate
IX	-	Engineering Test Unit
Second Phase, 4-79/10-81		
VII-2,3,4	-	No. 1 Field Test
X & IX	-	LSM Product Development
Third Phase, 10-79/10-81		
XII	-	No. 2 Field Test
VIII	-	No. 3 Field Test

TABLE B-2

LARGE SIZE MODEL PRINCIPAL PROGRAM MILESTONES

Program Milestone Number	Proposed Milestone Date	Milestone Description	Status as of 12-79/ Decisions Rendered
1*	11-1-77	Begin Large Size Model Market Potential Study	Completed per milestone date.
3B*	3-30-78	Project Advisory Board Meeting No. 4. • Review Large Size Model Market Potential Study. • Review 7½-Ton Demonstrator System Test Results.	Completed per milestone date.
4A*	4-1-78	Decide on Initial Field Test Site	Completed per milestone date/Mountwood Lodge selected.
6A*	5-1-78	Fix LSM Initial Market Prototype frame size.	Completed per milestone date/120 tons selected.
15*	4-25-79	Project Advisory Board Meeting No. 5. • Review LSM Business Potential Study. • Review LSM Comparative Cost of Ownership Analysis. • Review LSM System Cost. • Review Recommended Specifications and Scheduling for first two Field Test Sites.	Completed 5-4-79/Recommendation to terminate LSM Program.
16	8-79	Begin Final Screening for Venture Partner.	Initial inquiries only.
17	9-1-79	Identify Venture Partner Candidate.	Not initiated.
19	10-79	Project Advisory Board Meeting No. 6. • Introduce Venture Partner Candidate (VPC). • Review Preparations for Initial Field Test. • Recommend re. Proceeding with Second and Third Field Test Sites • MTI/VPC	Not initiated.
20	10-79	Select Second Field Test Site	Not initiated.
21	3-80	MTI/VPC Letter of Intent.	Not initiated.
22	3-80	Select Third Field Test Site.	Not initiated.
28	12-31-80	Open Initial Field Test Site.	Not initiated.
31	4-81	Start Up Second Field Test Unit.	Not initiated.
34	9-81	Start Up Third Field Test Unit.	Not initiated.
35	10-81	Terminate DOE Support of First Three Field Test Sites	Not initiated.

NOTE

*These milestones were achieved prior to program termination.