DOE/NASA CONTRACTOR REPORT

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PRELIMINARY DESIGN PACKAGE FOR SOLAR HEATING AND COOLING SYSTEMS

Prepared by

AiResearch Manufacturing Company
2525 West 190th Street
Torrance, California 90509

Under Contract NAS8-32091 with

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16. ABSTRACT

This report contains summarized preliminary design information on AiResearch Manufacturing Company activities associated with the development, delivery and support of solar heating and cooling systems. These systems are for single family dwellings and commercial applications.

AiResearch approach to the heating/cooling systems is to use a reversible vapor compression heat pump that is driven in the cooling mode by a Rankine power loop, and in the heating mode by a variable speed electric motor. The heating/cooling systems differ from the heating-only systems in the arrangement of the heat pump subsystem and the addition of a cooling tower to provide the heat sink for cooling mode operation.

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1. INTRODUCTION

This document summarizes preliminary design activities conducted by AiResearch Manufacturing Company of California under Contract NAS8-32091. It was prepared in accordance with the requirements of Appendix A, Data Requirement 500-7 of the Statement of Work, and contains data pertinent to the heating/cooling systems preliminary design review.

The AiResearch approach to the heating/cooling systems is to use a reversible vapor compression heat pump that is driven in the cooling mode by a Rankine power loop, and in the heating mode by a variable speed electric motor. The heating/cooling systems differ from the heating-only systems in (1) the arrangement of the heat pump subsystem and (2) the addition of a cooling tower to provide the heat sink for cooling mode operation. The remainder of the subsystems are essentially the same for the heating/cooling systems as for the heating-only systems; consequently, most of the data presented in AiResearch Report 76-12994, PDR Data Package (Heating Systems), are pertinent to the heating/cooling systems. Data from 76-12994 are referenced in this package.

The information presented in this document covers differences that are specific to the heating/cooling systems and new or updated data that were generated since the publication of AiResearch Report 76-12994.

Since the installation sites have not been selected, only limited work was done at the overall system level. This work was aimed primarily at concept refinement for the various subsystems. The major part of the activities was concerned with the design of the heat pump subsystem.

1.1 OVERALL SYSTEM APPROACHES

Overall system and subsystem approaches have been defined in AiResearch Report 76-12994. Figure 1-1 is a schematic of the 3-ton/80 KBTUH heating/cooling system.

Functionally, the collector, collector loop, energy storage tank, auxiliary heater, and domestic hot water supply are the same as for the single-family residence 80,000-Btu/hr heating system. A cooling tower is added to the heat pump system to provide a suitable heat sink in the cooling mode of operation. In the heating mode, the cooling tower is inactive. In the cooling mode, the heat pump utilizes thermal energy from the storage tank as the primary source of power. The Rankine air conditioner described in more detail later cools and dehumidifies recirculated air from the conditioned space directly as shown in the schematic of Figure 1-1.

Schematically, the 10-ton/250 KBTUH system is identical to the one depicted in Figure 1-1.

A schematic of the 25-ton/800 KBTUH system is presented in Figure 1-2. The collector, energy storage, auxiliary heater, and domestic hot water subsystems are functionally the same as for the 800,000-Btu/hr heating system. The auxiliary heater and domestic hot water tanks are the same equipment. Collector and storage tank sizes are different.

A cooling tower is added to serve as a heat sink for the Rankine-cycle air conditioner in the cooling mode of operation. In the heating mode, the cooling tower is inactive and is drained to prevent freezing.

The water heat transport loop from the heat pump to the twelve terminal units serves to transmit the cooling effect to the residence. Control of the cooling capacity of each of the twelve terminal units is by cycling the ventilation fans on and off using thermostats suitably located at each unit.

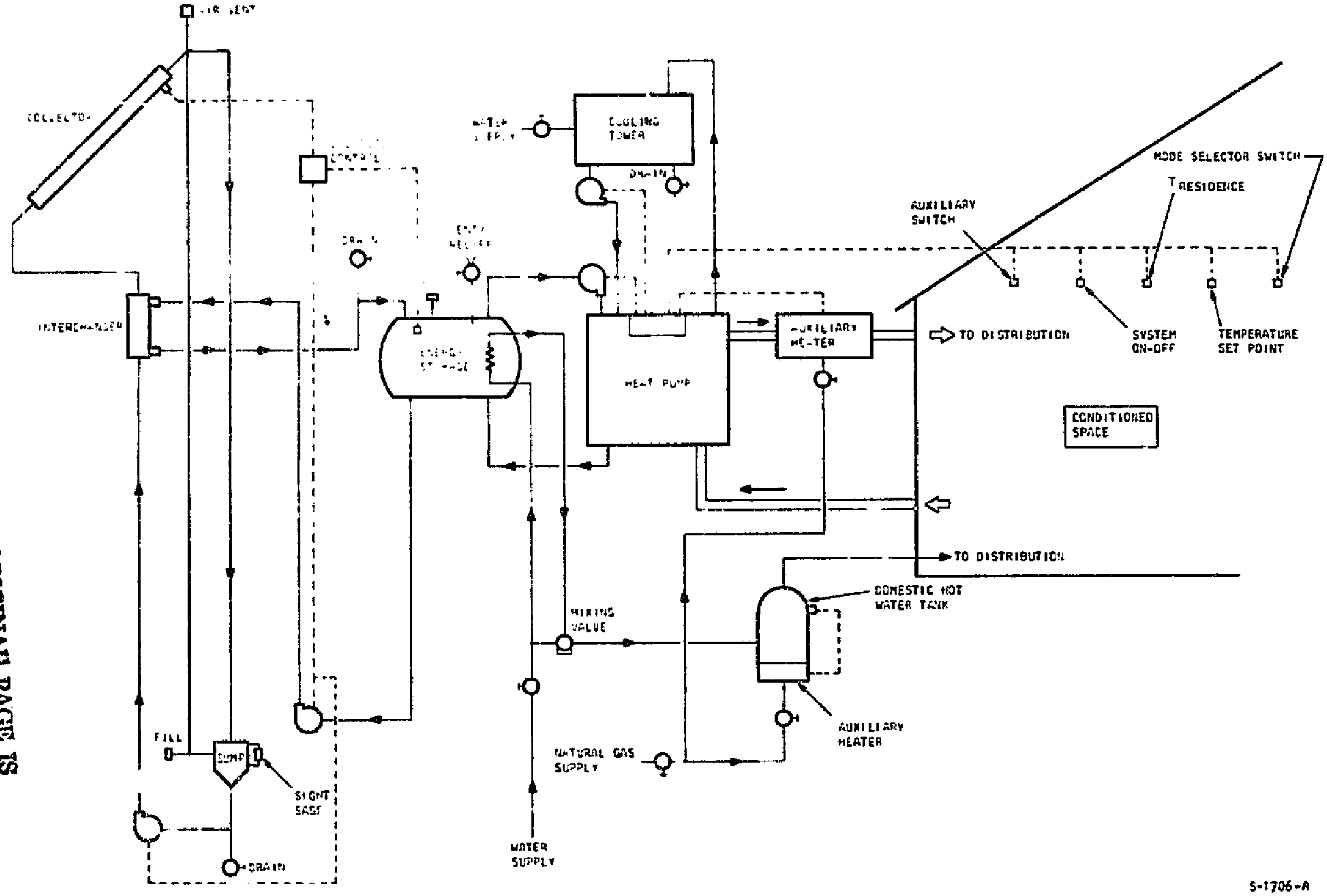

Figure 1-1. Single-Family Residence Heating and Cooling System

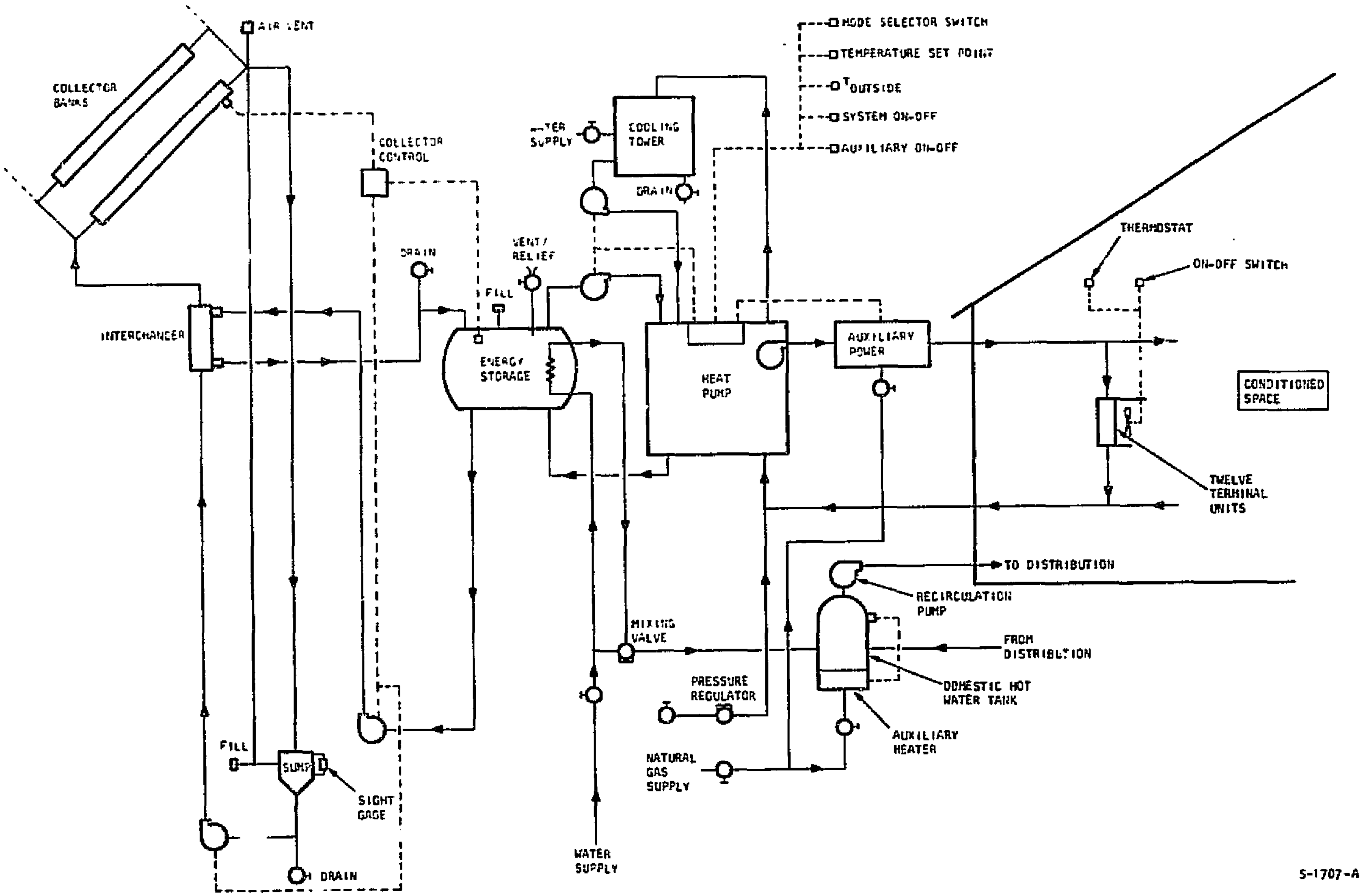

Figure 1-[illegible]. Multifamily Residence Heating and Cooling System

1.2 OVERALL SYSTEM PERFORMANCE

System performance for the single-family and multifamily residence systems is presented in Table 1-1. These data were taken from the AiResearch proposal and were derived using Nashville weather data and the residence models supplied by NASA. They are included here for reference purposes only. It is believed that these data are representative of the system designs that will be generated following site selection.

1.3 DOCUMENT ORGANIZATION

A review of the work performed at the subsystem level is presented in this package, followed by the detail work performed on the heat pump and its major components. Topics covered include:

- Heat pump subsystem
- Turbomachine/motor
- Motor control
- System control
- Heat pump heat exchangers
- Cooling tower
- R-11 pump

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TABLE 1-1

HEATING/COOLING SYSTEMS PERFORMANCE SUMMARY

	Single-Family Residence	Multifamily Residence
System Features		
Collector area, sq ft	1000	10,000
Energy storage tank, gal	1800	18,000
Heat pump capacity		
Heating, Btu/hr	80,000	800,000
Cooling, tons	3	25
Auxiliary heater capacity, Btu/hr	80,000	800,000
Domestic hot water		
Storage, gal	50	650
Recovery rate, Btu/hr	60,000	750,000
Yearly Performance		
Residence loads		
Heating, 10^6 Btu	184	1840
Cooling, 10^6 Btu	34.7	294
Hot water load, 10^6 Btu	30.7	384
Solar Contribution		
Heating, percent	67	67
Cooling, percent	75	65
Auxiliary thermal energy contribution		
Heating, percent	25.1	24.3
Auxiliary electrical power contribution		
Heating, percent	7.9	8.7
Cooling, percent	25	35
Total energy expenditure		
Thermal (fuel oil or gas), 10^6 Btu	83.2	772
Electrical, kw-hr	8670	94,185
Ultimate energy saving, 10^6 Btu	208	2080
Present value benefit, dollars (20-year, residential; 25-year, commercial)	3000	71,000

2. SUMMARY

Pending selection of the installation sites and final definition of the overall system capacities, most of the effort has been concentrated on the design of the heat pump. Work involving other subsystems was pursued only to the level of detail necessary to select approaches and to refine the overall system schematic. A summary of this work is given in this section for each of the subsystems considered.

2.1 SOLAR COLLECTOR

An RFP for collector procurement was issued on August 30, 1976. Thirty-two manufacturers were solicited, and 16 different designs were proposed, ranging from simple flat-plate collectors with black paint absorbers to concentrating type collectors. All proposals received have been evaluated using the methodology shown in Figure 2-1.

A number of designs were subjected to detailed evaluation using the present value cost technique as a basis for cost effectiveness comparison. Tentative selections were made as a result of these investigations. The performance data submitted by the manufacturers whose collectors are the most attractive, however, are either calculated data or questionable in comparison with similar designs. AiResearch has therefore requested that these potential subcontractors supply test performance data (efficiency vs $\Delta T/I$) certified by an independent test agency. These data are due at AiResearch prior to December 13, 1976. Upon receipt of these data, final evaluation will be completed, and the results of these collector investigations will be presented to NASA prior

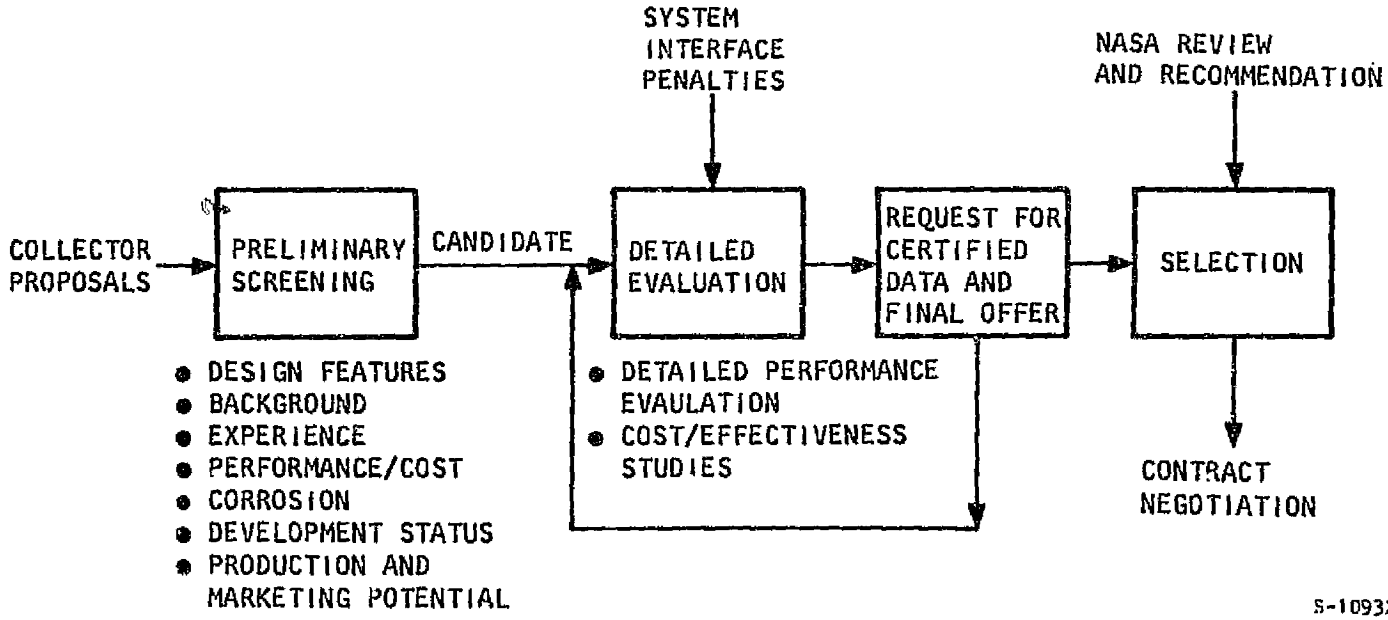

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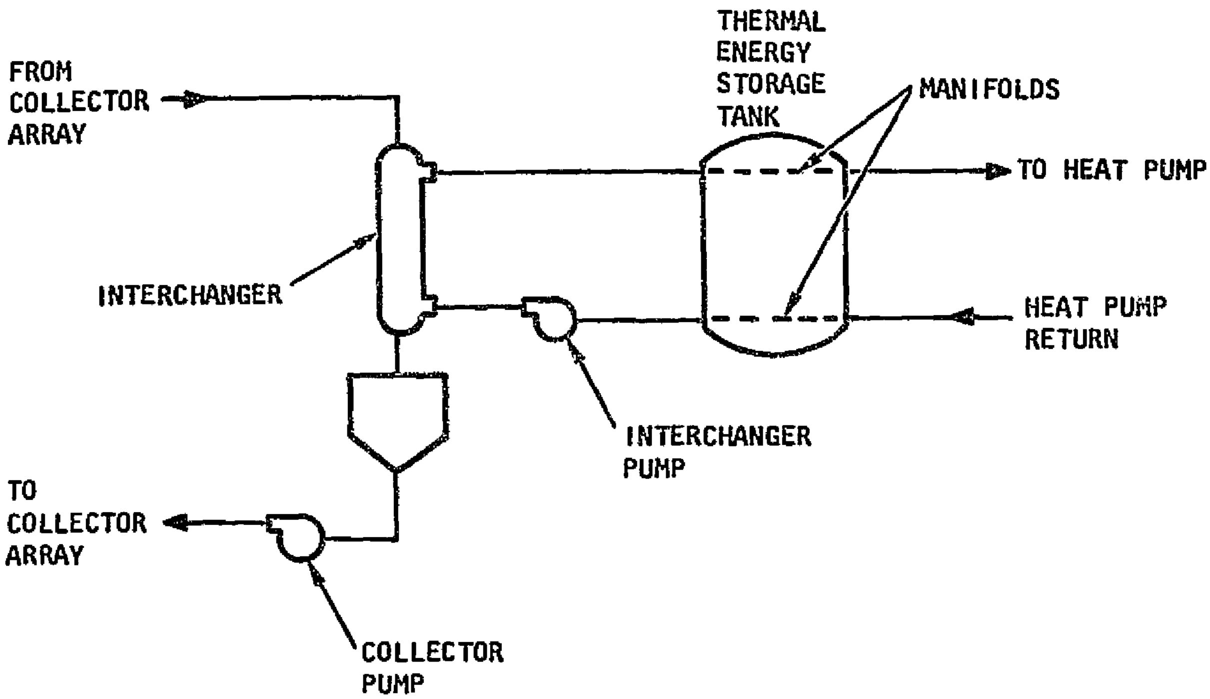
Figure 2-1. Solar Collector Selection Rationale to selection of one or two subcontractors. (Different collectors may be optimum for the heating-only systems and for the heating/cooling systems.)

2.2 THERMAL ENERGY STORAGE TANK

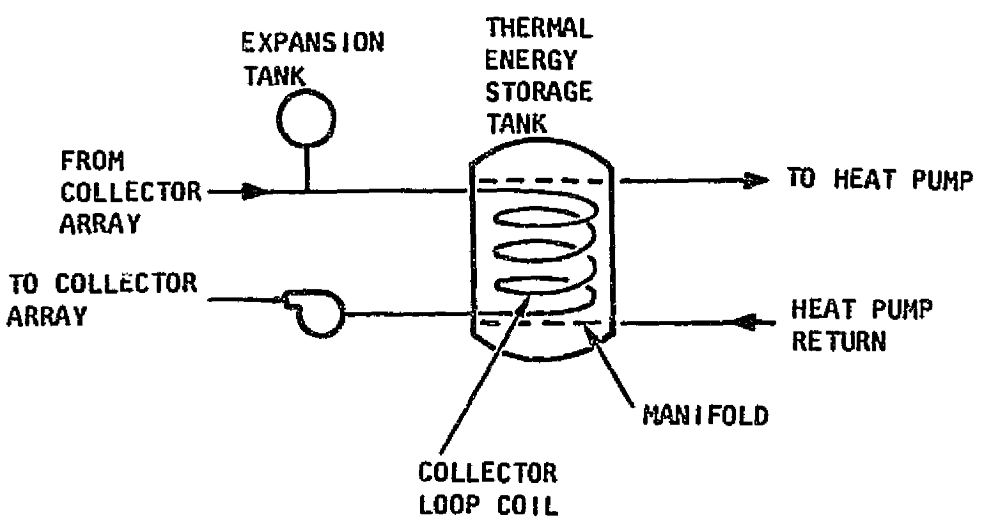
Tests on a manifold configuration designed to promote thermal stratification were conducted with limited success (see AiReserch Report 76-12994, pages 4-1 through 4-6). The data show that stratification could be maintained with circulation in the collector loop (water inlet at the top of the tank and water outlet at the bottom). However, when the water flow is reversed (water inlet at the bottom) simulating heat pump operating conditions, mixing occurred rapidly, thus destroying the plug-flow situation in the tank.

A new manifold design has been developed using high solidity screens (50 percent) to straighten and distribute the flow from both the top and bottom manifolds. Testing of this new manifold configuration has been delayed pending detailed investigations of the interchanger-pump-tank circuit. Figure 2-2 shows the two approaches under consideration.

A cost effectiveness study is being conducted to determine the relative cost of a tank with an internal coil to transfer the collector heat to the thermal energy storage tank. Preliminary sizing indicates that about 500 ft of steel coil would be required in the single-family residence tank to achieve the same thermal performance as the interchanger of the baseline approach. The advantages of the coil-in-tank approach are (1) limited fluid volume in the collector loop, thus permitting the use of antifreeze at a reasonable cost and eliminating the requirement for tank draining to prevent freezing; and (2) lower pumping power.



a. BASELINE APPROACH



b. COIL-IN-TANK APPROACH

Figure 2-2. Candidate Storage Tank Approaches

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2.3 AUXILIARY THERMAL ENERGY

Review of the geographic locations selected by NASA for system installation indicates that natural gas may not be available at all sites for use as an auxiliary heat source. The packages for the 80 and 250 KBTUH heating-only systems were developed on the assumption that natural gas would be available; they feature a gas-fired furnace integrated in the air-handling portion of the package. If natural gas is unavailable, the package could be modified as follows, depending on whether fuel-oil or electricity is used as the auxiliary heat source.

(a) Fuel-oil--Integration of an oil-fired furnace in the baseline package would require major redesign. A small oil-fired boiler could be used with a hot water coil incorporated in the baseline package in lieu of the gas-fired furnace. The cost of the boiler alone for the 80 KBTUH unit is about $750, compared to $140 for the gas-fired furnace. This cost is prohibitive.

(b) Electrical heat--In this case, an electrical heater could be fitted in the space occupied by the baseline gas-fired furnace. Electrical thermal energy, however, is two to three times more expensive than fuel-oil and is extremely wasteful of natural resources. This approach is not cost effective and further is in direct conflict with the ultimate purpose of the present program.

In order to promote modularity while providing flexibility, AiResearch/Dunham-Bush have revised the auxiliary thermal energy approach for the single-family residence (3-ton/80 KBTUH) and small commercial (10-ton/250 KBTUH) systems. The updated approach is to divorce the auxiliary thermal energy subsystem from the heat pump package. In this manner, existing off-the-shelf gas, fuel-oil, or electric furnaces could be used. This will entail additional ducting between the heat pump package and the furnace; however, the heat pump package will be the same in all cases. Further, detail heat pump packages can be developed without waiting for final site selection.

2.4 HEATING/COOLING HEAT PUMP

The baseline reversible (heating and cooling) heat pump schematics originally proposed used reversing water valves rather than refrigerant valves. A schematic of the multifamily unit is shown in Figure 2-3. This approach was initially selected to reduce the pressure drop in the R-11 circuit and also to reduce the overall cost of the heat pump. Examination of the water loop arrangement around the condenser reveals that cooling tower water will be mixed with the other water loops every time the system is switched from the cooling to the heating mode. This will result in the introduction of oxygen and undesirable contaminants into the water loops. To obviate this problem a number of approaches were considered, including:

(a) The use of parallel condensers, one each for heating and cooling mode operation. The added cost of this approach is estimated to be about $2000 for the 25-ton system excluding any valving that may be necessary to isolate the two units.

(b) The use of a closed-loop cooling tower. This will result in a cost increase of about $4000 for the 25-ton system. In addition, a water pumping penalty of about 5 kw will be incurred to overcome heat exchanger pressure drop and to provide for water recirculation within the tower.

(c) The use of an intermediate water loop between the condenser and the open-loop cooling tower. This approach has the same thermodynamic disadvantages as (b) above and none of the advantages.

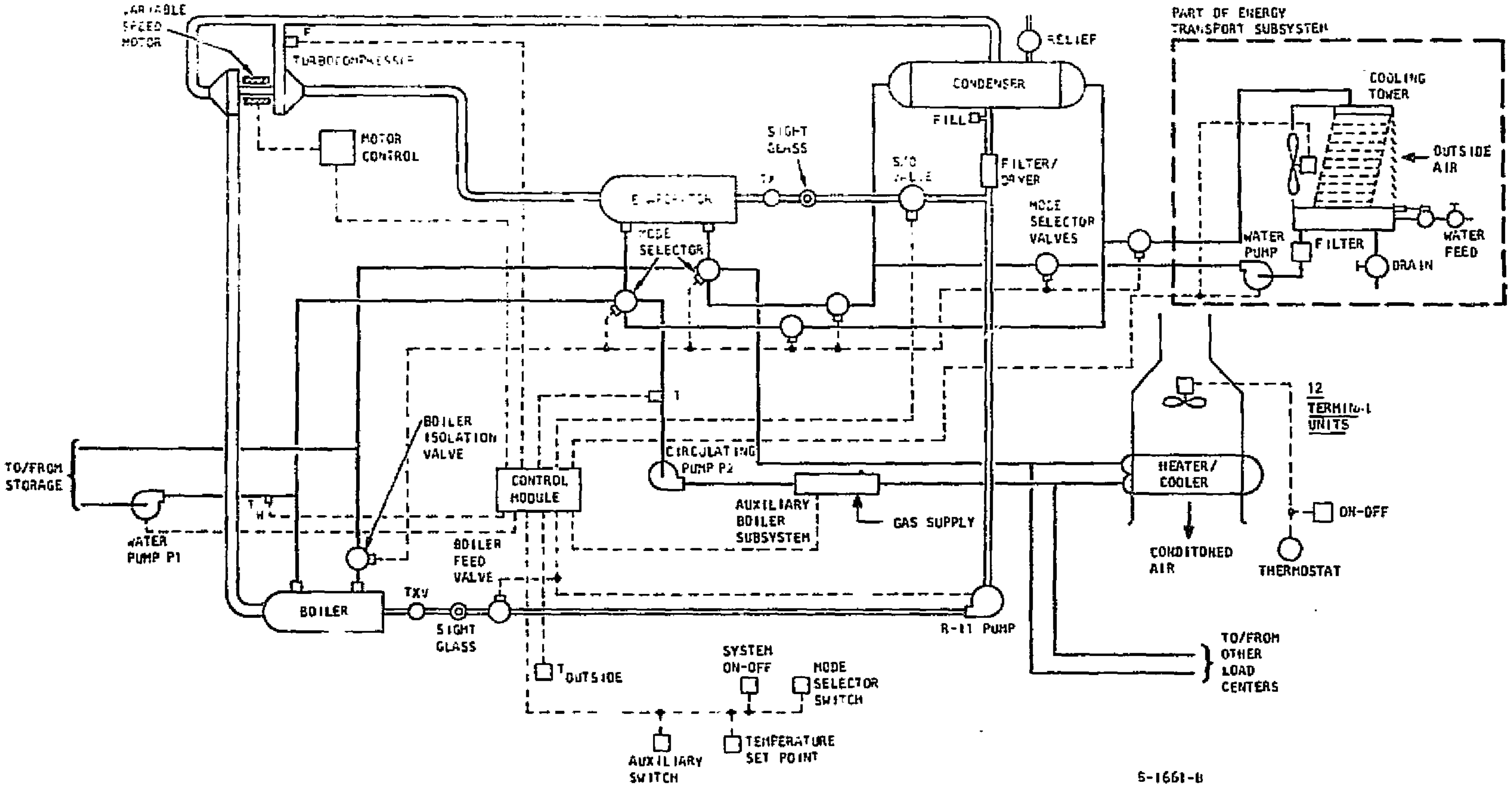

Figure 2-3. Baseline 25-Ton/600 KBTUH Cooling and Heating Subsystem

(d) The use of a system arrangement whereby the cooling mode condenser is used for this purpose only, so that the cooling tower-condenser circuit is closed and does not interface with the remainder of the system water circuits. This approach was selected.

A schematic of the updated multifamily heating/cooling system is shown in Figure 2-4.

The major differences between the system arrangement originally proposed and the updated version are as follows:

(a) Elimination of the water reversing valves and replacement of these valves by an R-11 reversing valve and an R-11 shutoff valve.

(b) Changing the functions of the system heat exchangers:

(1) The cooling mode condenser is used only in the cooling mode so that this condenser and the cooling tower constitute a dedicated water loop.

(2) The water recirculation loop is isolated from the other water loops and interfaces only with one of the heat pump heat exchangers. This heat exchanger functions as an evaporator in the cooling mode and as a condenser in the heating mode.

(3) The cooling mode boiler is designed to function as the evaporator in the heating mode. In this manner the water loop to the storage tank is also isolated.

As shown in the schematic of Figure 2-4, a water shutoff valve is used to control on and off a small flow of water to a coil around the cooling mode condenser shell. In the heating mode, hot water from the storage tank maintains the condenser at a temperature higher than that at compressor inlet, thus preventing R-11 condensation and accumulation in this heat exchanger.

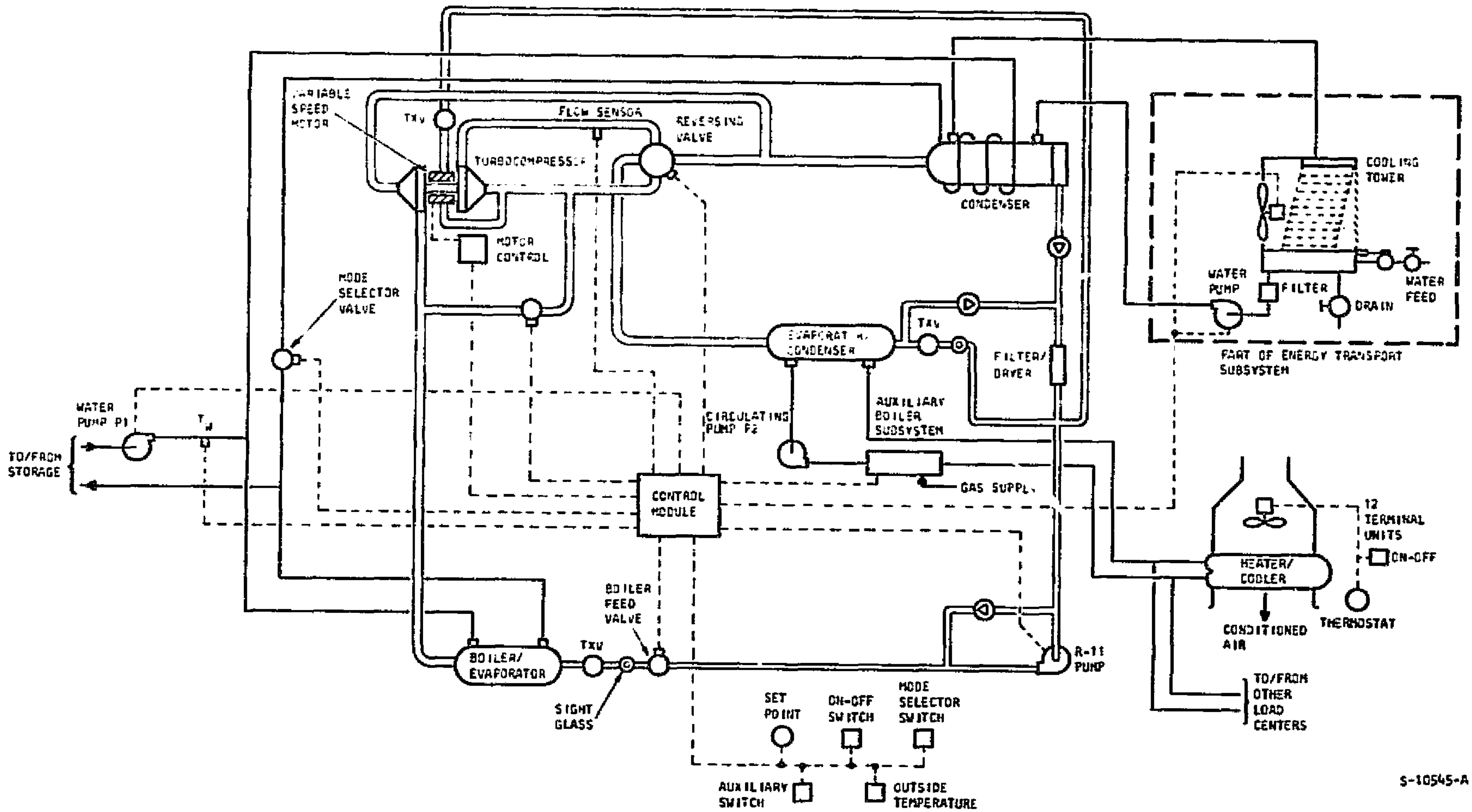

Figure 2-4. Updated 25-Ton/600 KBTUH Cooling and Heating Subsystem

This flow of water will only be sufficient to make up the heat leaks from the heat exchanger.

Detail design of the heat pump heat exchangers revealed that serious flow distribution problems may be encountered in the evaporating/boiling heat exchangers. The distributors necessary to assure proper flow through all heat exchanger passages would be large and would incorporate a large number of tubes. Further single-tube evaporator tests reported later indicated that at the low pressure drops specified, slugging problems could occur. For these reasons, the decision was made to change the configuration of these heat exchangers and to use vertical tube evaporators/boilers throughout. Single-tube heat transfer and pressure drop tests were conducted to generate the basic data for design. These tests demonstrated stability of operation under these conditions. As a result, all heat exchangers were resized and new package drawings will be developed.

3. SPACE HEATING AND COOLING SUBSYSTEMS

3.1 GENERAL

The characteristics of the heat pumps with combined capabilities for heating and cooling are presented below. To maximize modularity and minimize cost, much of the equipment contained in the heating/cooling heat pumps is identical to that of the heating-only versions described in the previous PDR, AiResearch Report 76-12994. In general, the heating/cooling subsystem heat exchangers are designed so that performance requirements are balanced for the heating and cooling modes. The approach used for the motor-driven compressor was to design a turbocompressor with the integral motor meeting the requirements of the heating/cooling heat pump. The same machine is used without the turbine for the heating-only heat pump. Here again, the decision was made to promote modularity.

System and motor controls in the cooling mode are relatively simple compared with those in the heating mode. The approach taken was to add to the heating mode control logic the simple circuitry necessary for operation in the cooling mode.

The preliminary design of the heat pump in both the heating and cooling modes of operation has been completed. Subsystem performance characteristics were determined, and problem statements for all heat pump components were released. These problem statements cover the heating and cooling cases.

The heat pump rated capacities listed below were selected for the applications shown.

	Heating	Cooling
Single-family residence	60 KBTUH	3 tons
Multifamily residence	600 KBTUH	25 tons
Commercial application	200 KBTUH	10 tons

3.2 SINGLE-FAMILY RESIDENCE SPACE HEATING AND COOLING SUBSYSTEM 3-TON/ 60 KBTUH HEAT PUMP

3.2.1 Subsystem Arrangement

Figure 3-1 is a schematic of the 3-ton/60 KBTUH subsystem. Figure 3-2 is the same schematic, showing the flow paths of the water and R-11 working fluid in the heating and cooling modes of operation.

The functions of two system heat exchangers are reversed from the heating to the cooling mode. The heat pump effect heat exchanger always uses the conditioned space air as a heat exchange fluid, operating as an evaporator in the cooling mode and as a condenser in the heating mode. The boiler/evaporator uses water from the thermal energy storage tank as a source of turbine power in the cooling mode and as a heat energy source in the heating mode. The third heat exchanger is dedicated as a condenser and is used only in the cooling mode with cooling tower water as a heat sink. This arrangement isolates the exposed tower water and precludes contamination of the storage water system.

Four selector valves (three R-11 and one water), actuated from a mode selector switch within the residence, permit operation in the heating or the cooling mode. In the heating mode, operation is identical to that for the single-family residence heating subsystem. In this mode, the compressor is motor-driven at variable speed and the turbine is inactive. The entire Rankine power loop is disabled by interruption of the heat input to the boiler and isolation of the cooling tower.

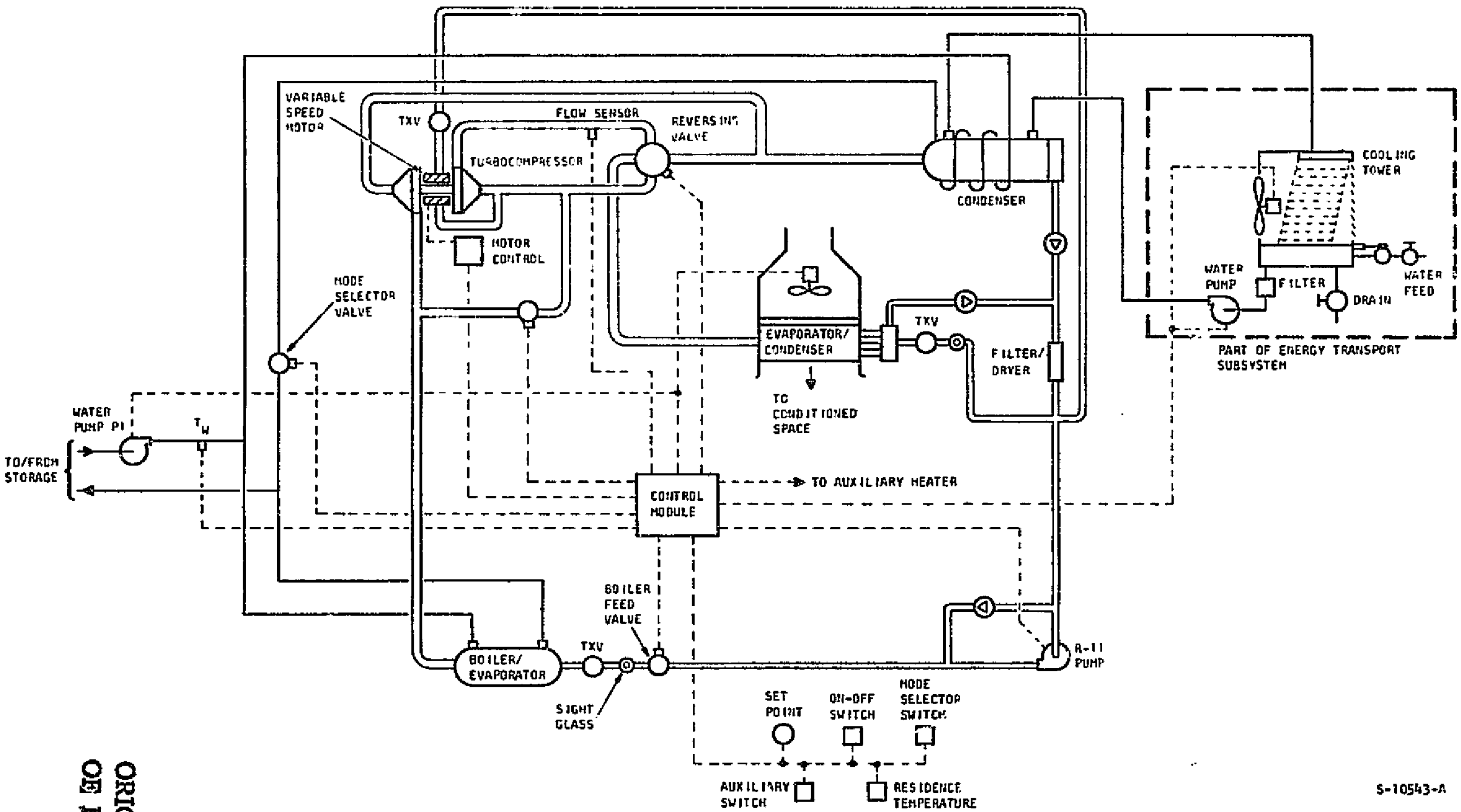

Figure 3-1. Single-Family Residence Space Heating and Cooling Subsystem

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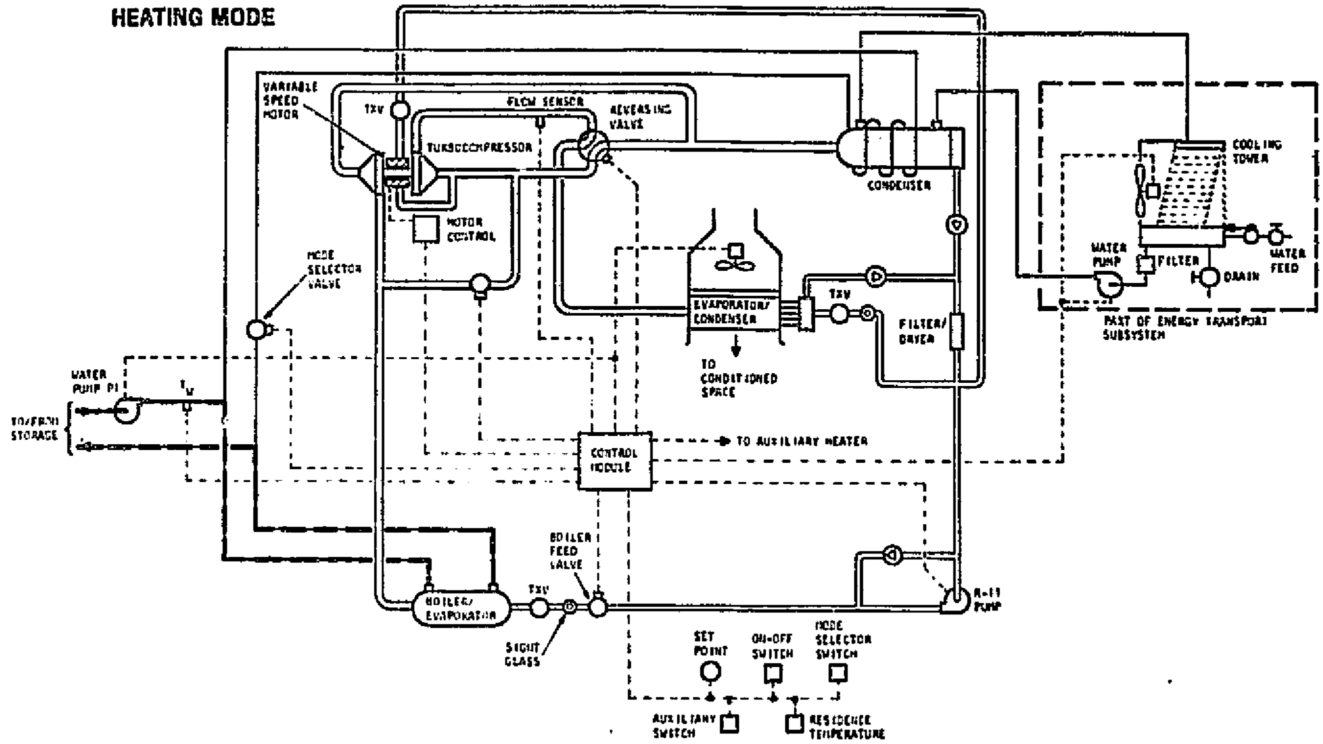
HEATING MODE



COOLING MODE

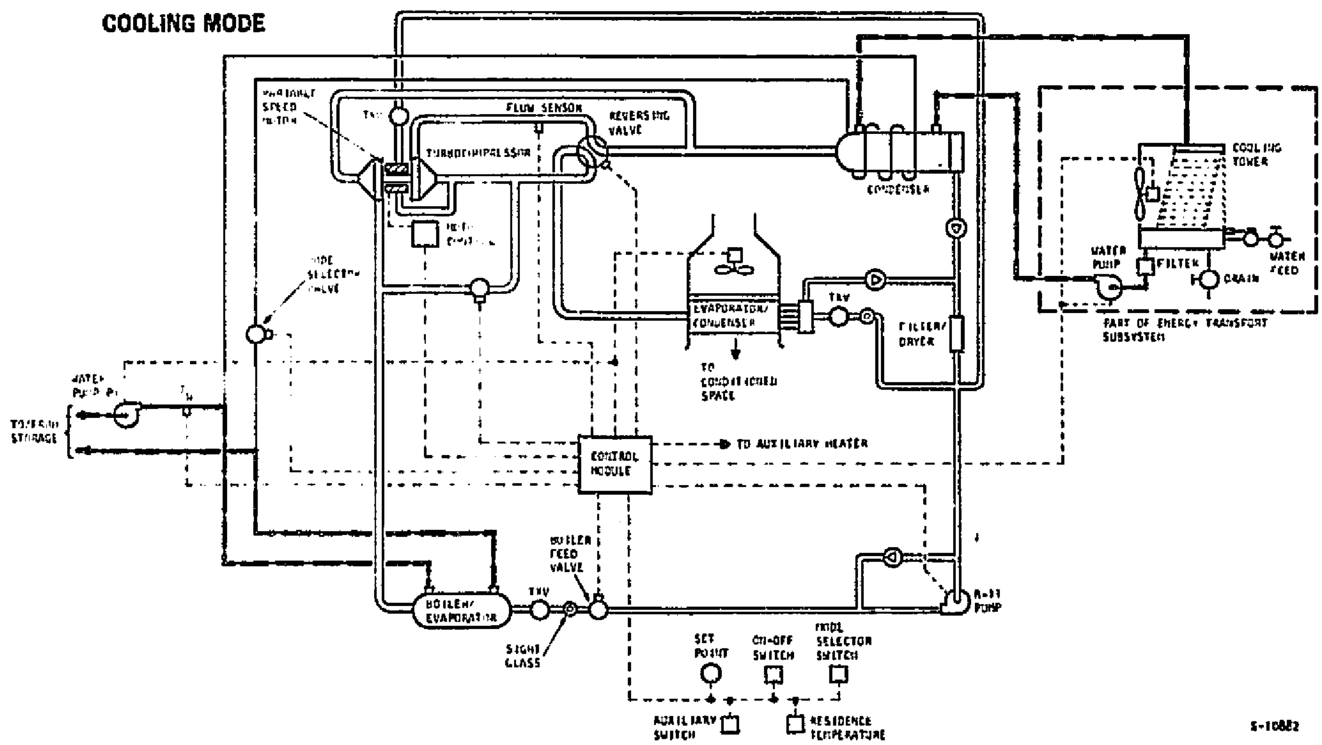


Figure 3-2. 3-Ton Subsystem Flow Paths In Heating and Cooling Modes

In the cooling mode of operation, heat energy from the hot water storage tank is used as the Rankine power loop heat source. A cooling tower serves as the ultimate heat sink. When thermal energy available from the storage tank is inadequate, the motor integral with the turbomachine is activated. In this augmented mode of operation, turbocompressor speed is controlled at a constant value by the motor, but the turbine still supplies a large portion of the power to drive the compressor. As the temperature of the water heat source to the boiler drops further to 155°F, the boiler is isolated and all power necessary to drive the compressor is developed by the electric motor.

The only residence control necessary for operation of the subsystem in the cooling mode, in addition to the heating mode sensors and controls, is the mode selector switch. This switch will actuate (1) the mode selector valve in the hot water supply lines, (2) the refrigerant reversing valve, (3) the turbine bypass valve, and (4) the boiler feed valve. In addition, the switch will control power to the R-11 pump and the cooling tower fan and water pump.

In the cooling mode, the system is essentially controlled to the residence temperature set point. The system is activated when the temperature exceeds 1°F above the set point and deactivated when the temperature drops 1°F below the set point.

When the air conditioner is activated from the temperature error signal, the control module will energize (1) the cooling tower fan and water pump, (2) the hot water pump, (3) the R-11 pump, (4) the ventilation fan, and (5) the turbocompressor motor.

In normal operation, turbine power is sufficient to drive the compressor; and the motor will be deactivated when the system has stabilized after the starting sequence. As the water temperature from the storage tank drops, compressor speed is reduced, resulting in a loss of capacity. If the water temperature continues to drop, the power developed by the turbine will become inadequate and compressor surge will result. Surge characteristics for the compressor are shown in Figure 3-3, which indicates that outlet volumetric flow is a constant at any point on the surge line. This parameter was selected to determine when the motor is to be energized to prevent surge.

With the motor on, the turbine continues to provide a significant portion of the total power required to drive the compressor. When the water temperature decreases to 155°F, the control module closes the power loop and the motor carries the entire load. A temperature sensor at the water pump outlet provides the signal. Power loop shutdown involves (1) R-11 pump deactivation, (2) water pump deactivation, and (3) closing the boiler feed valve.

System shutdown from the normal operating mode (without motor augmentation) involves steps exactly the reverse of the starting sequence listed above. With the motor on, system shutdown also involves deactivating the motor. The control module then is reset for operation in the normal mode. Shutdown occurs when the residence temperature drops 1°F below the residence temperature set point.

3.2.2 Subsystem Performance

Cooling mode performance is essentially dependent on the outside air and the residence wetbulb temperatures. To a lesser degree, the drybulb temperatures also will affect performance.

The system will deliver a nominal three tons of air conditioning under the design point conditions. These design conditions represent a worst-case situation (2-1/2 percent time); normally the cooling subsystem will operate

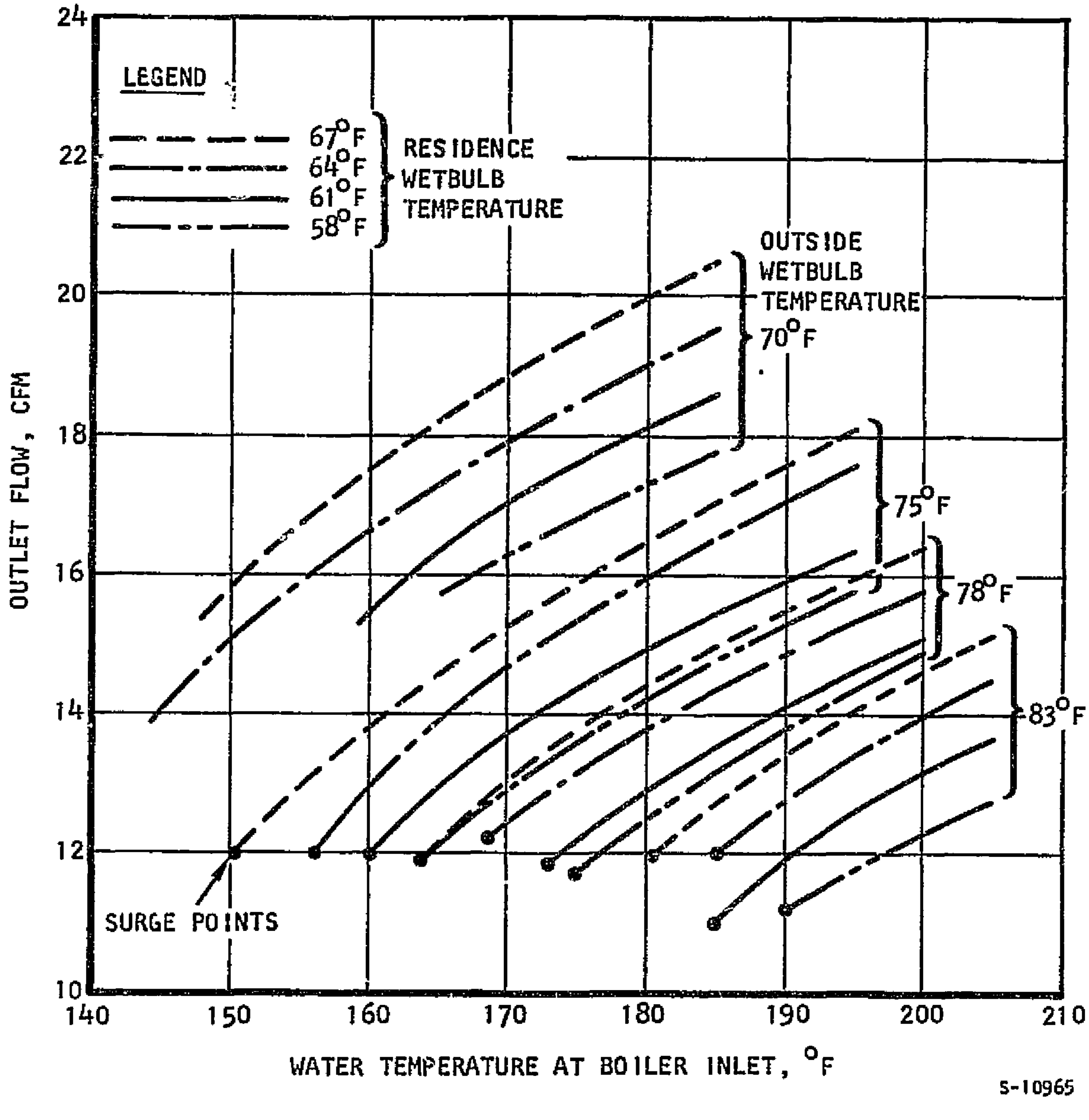


Figure 3-3. 3-Ton Compressor Surge Characteristics

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under much more favorable conditions, particularly with regard to the outside air wetbulb temperature.

Figure 3-4 is a plot of subsystem coefficient of performance (COP) and capacity as a function of heat source water temperature. COP is defined as the ratio of the evaporator load (cooling effect) to the boiler heat input. The data are shown for the residence conditions noted and for a range of ambient wetbulb temperatures.

Examination of the data shows a marked increase in COP and capacity as the ambient wetbulb temperature drops. The capacity of the system drops as the boiler inlet water temperature decreases. The minimum capacity of the system without motor augmentation is about 2.1 tons. If the capacity drops below this value, motor augmentation will be necessary.

Figure 3-5 shows subsystem performance over its entire range of operation: (1) normal mode with turbine drive only, (2) augmented mode with motor-assisted turbine drive, and (3) auxiliary mode with motor drive only. The data are shown for the design point conditions specified, and for a range of outside humidity operating situations on either side of the 78°F wetbulb design condition.

Turbine performance in the cooling mode is presented in Figure 3-6. The percentage of energy supplied by the turbine from heat storage and turbine efficiency values in the motor augmented mode are shown up to the storage temperature, which permits 100 percent turbine power. As the ambient wetbulb temperature increases, more heat energy is required to attain the minimum speed for motor cutout due to increased compressor horsepower per ton and less turbine power per pound of flow. In the figure, as the wetbulb increases from the 78°F design point to an extreme of 83°F, the required storage water temperature for 100 percent use of heat storage energy increases from 168°F

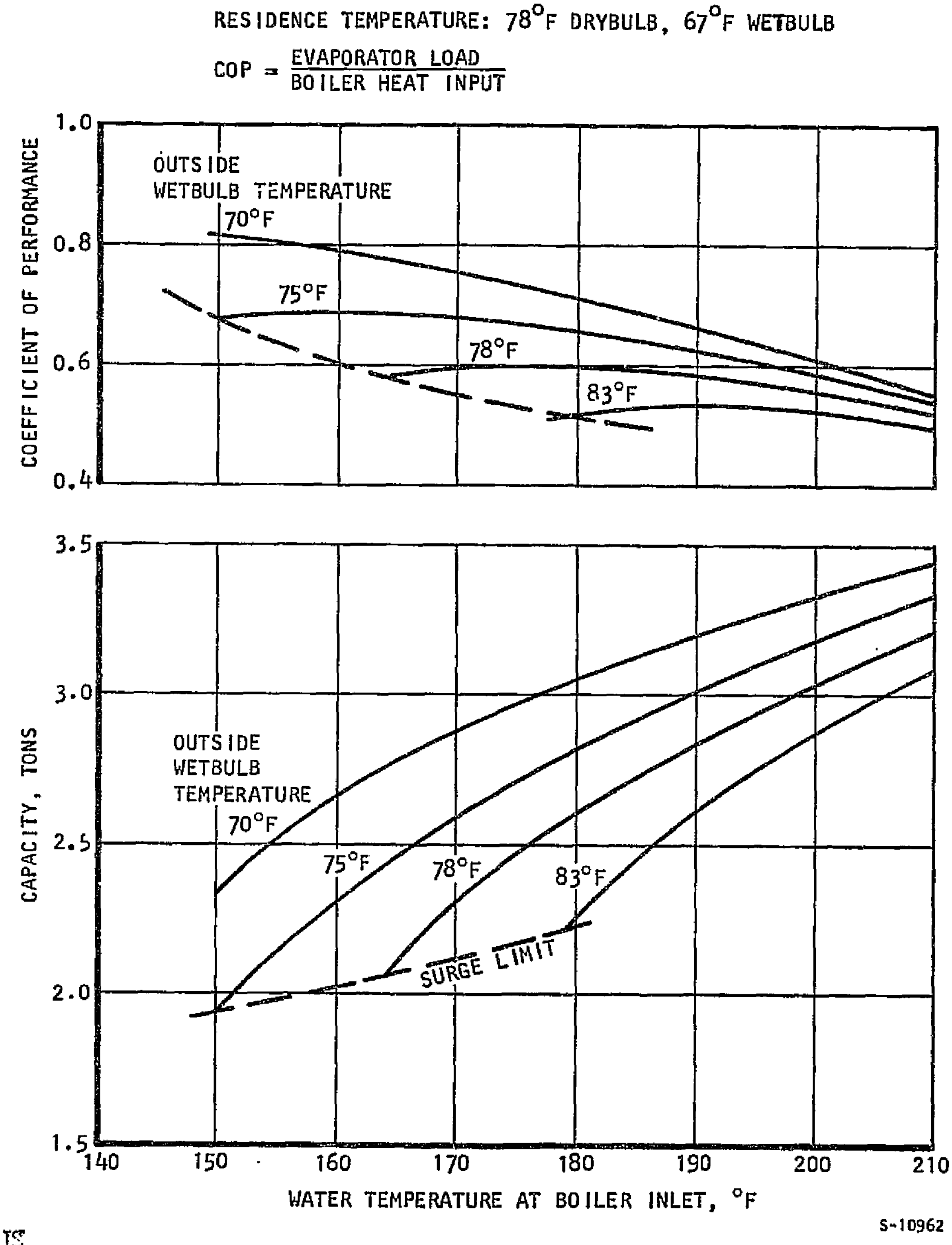

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Figure 3-4. 3-Ton Cooling Subsystem Performance With Turbine Drive

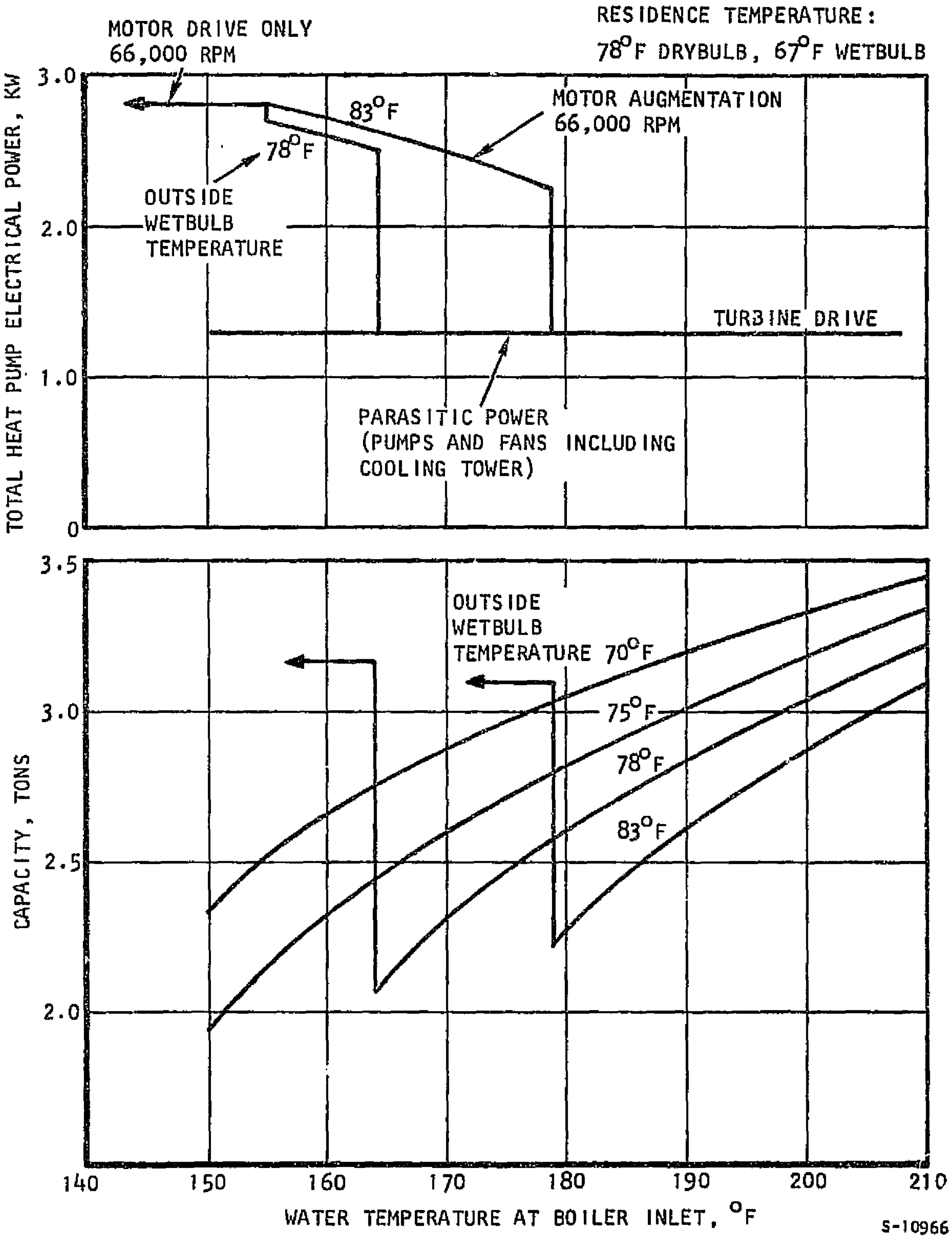

Figure 3-5. 3-Ton Heat Pump Performance

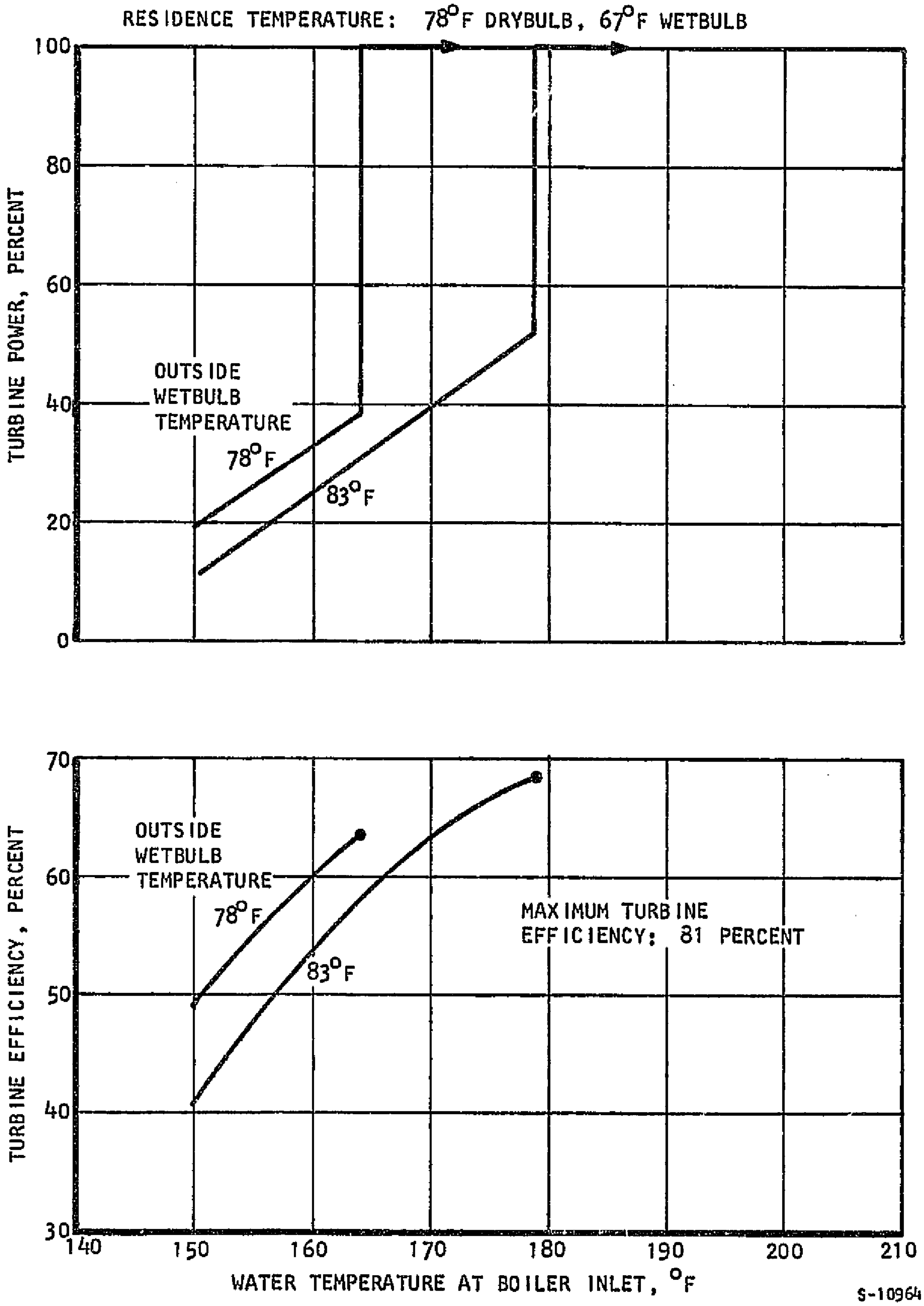

Figure 3-6. Turbine Performance in Augmented Mode, 3-Ton System

to 178.5°F. Above the range of motor augmentation the turbine efficiency increases, reaching a maximum of 81 percent.

Figure 3-7 is a compressor performance map with an overlay of water storage temperatures and ambient wetbulb temperatures. A single point on the compressor map is defined for any set of controlling conditions. When the compressor flowrate drops to a value near the surge line, the motor is energized, returning the compressor to the normal operational range. Compressor design has been optimized to provide peak efficiencies in the high outside wetbulb range and in the entire range of storage water temperature where the turbine is energized (above 155°F).

Subsystem performance is summarized in Table 3-1 for several storage water temperatures in the range of turbine operation. The significant factor is electrical COP. With 200°F water, heat energy is used to increase the electrical COP to a high value of 8.0. As the water temperature drops, less energy is available and the COP drops accordingly. When the water temperature falls below 155°F, the compressor is motor driven only, and a more typical COP of 4.2 results.

3.3 COMMERCIAL APPLICATION SPACE HEATING AND COOLING SUBSYSTEM 10-TON/ 200 KBTUH HEAT PUMP

3.3.1 Subsystem Arrangement and Control

Figure 3-8 is a schematic of the 10-ton/200 KBTUH subsystem. The same schematic is used in Figure 3-9 to show flowpaths in the heating and cooling modes of operation. The schematic arrangement and control of this hardware are identical to those of the 3-ton/60 KBTUH unit described previously. Compressor surge characteristics for the 10-ton unit are shown in Figure 3-10.

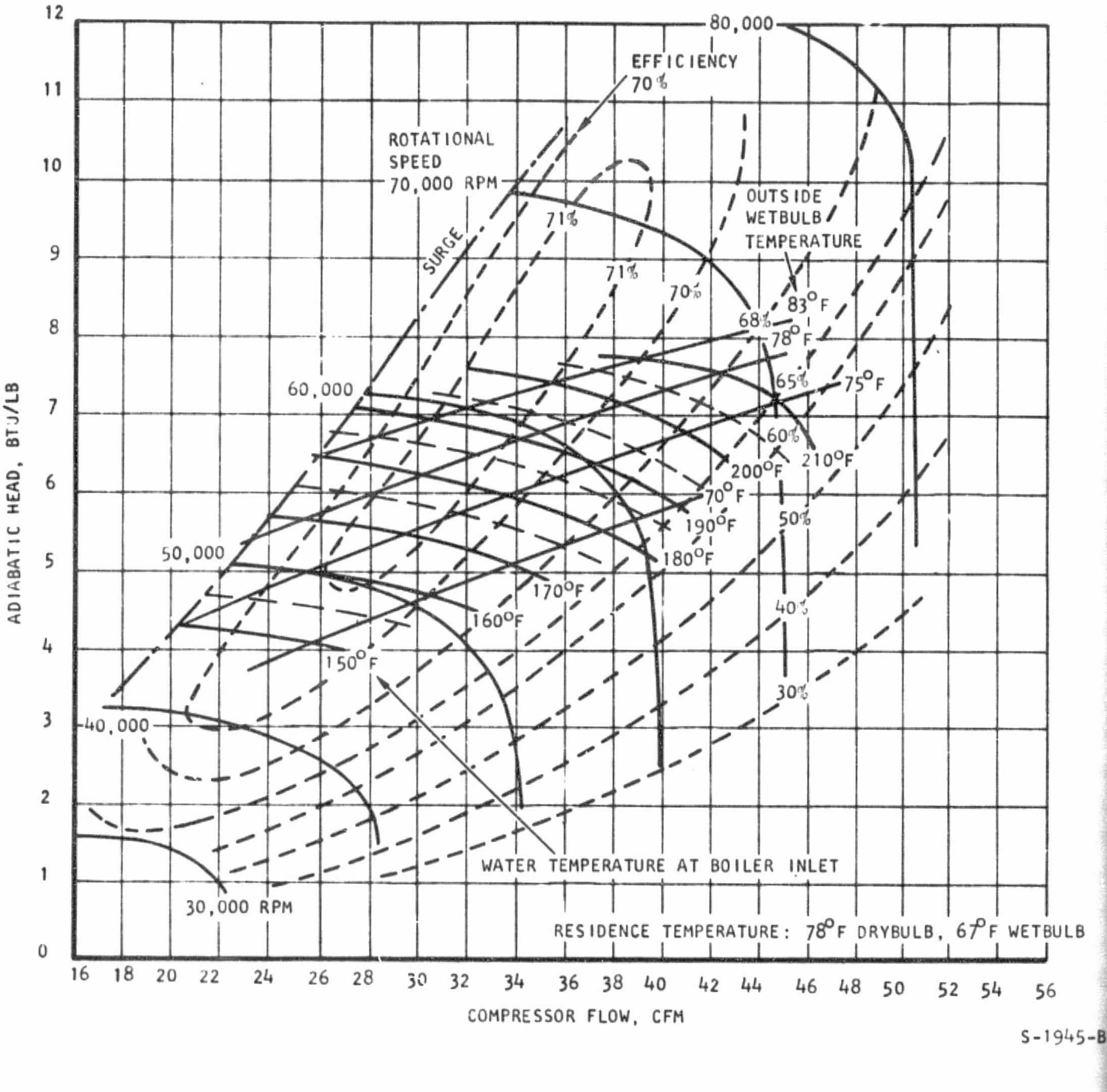

Figure 3-7. Compressor Map, 3-Ton/80,000-Btu/hr Unit

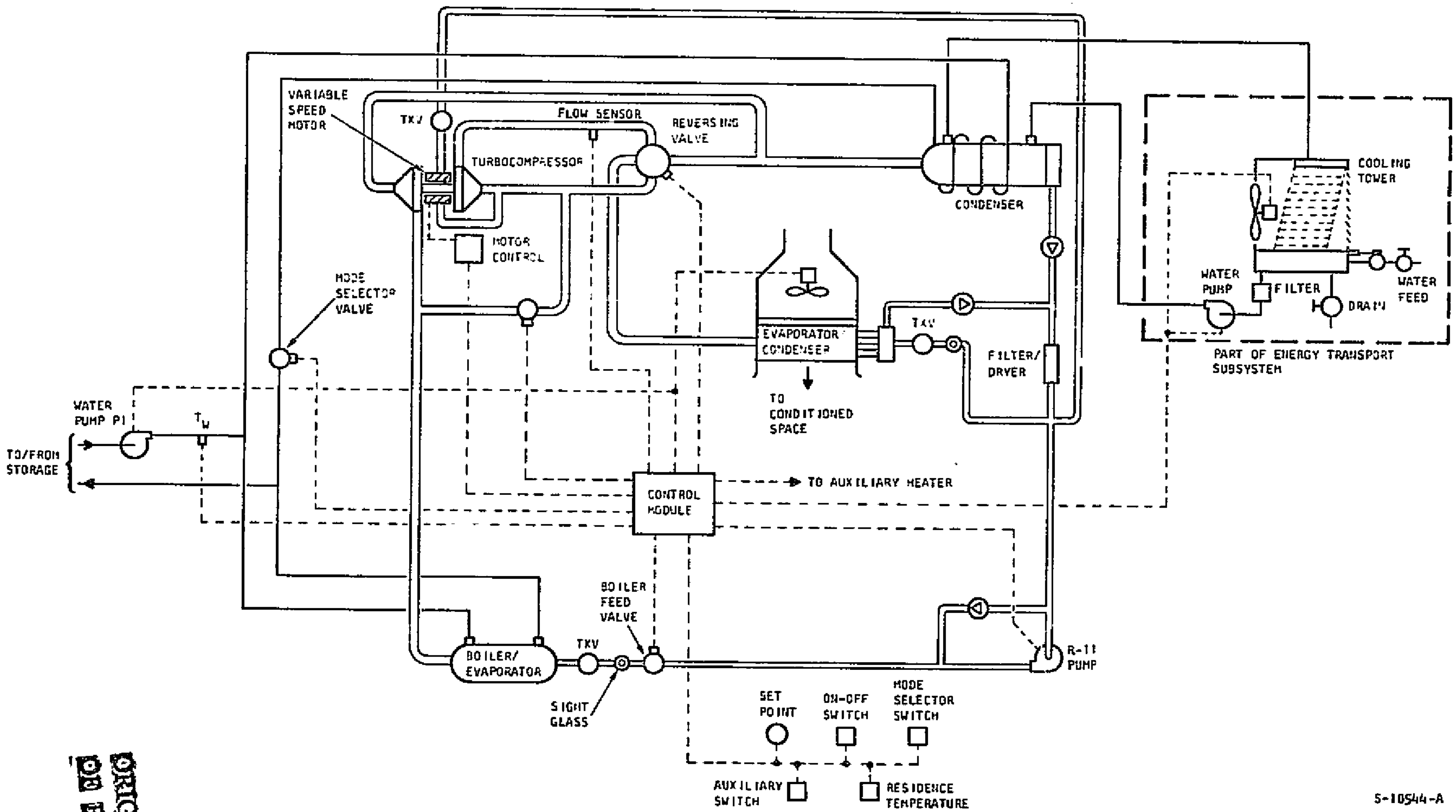

Figure 3-8. Commercial Application Space Heating and Cooling Subsystem

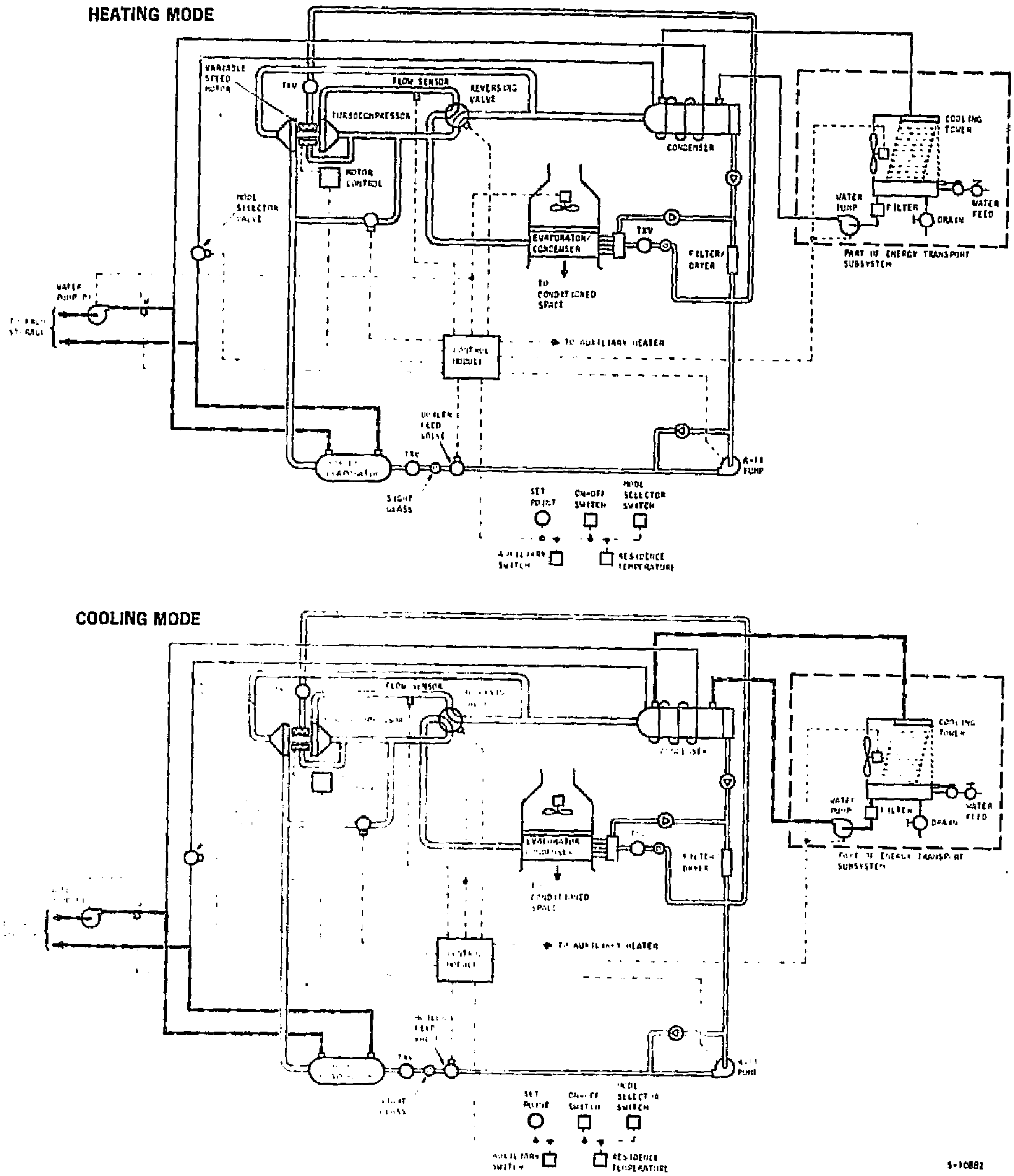

Figure 3-9. 10-Ton Subsystem Flow Paths In Heating and Cooling Modes

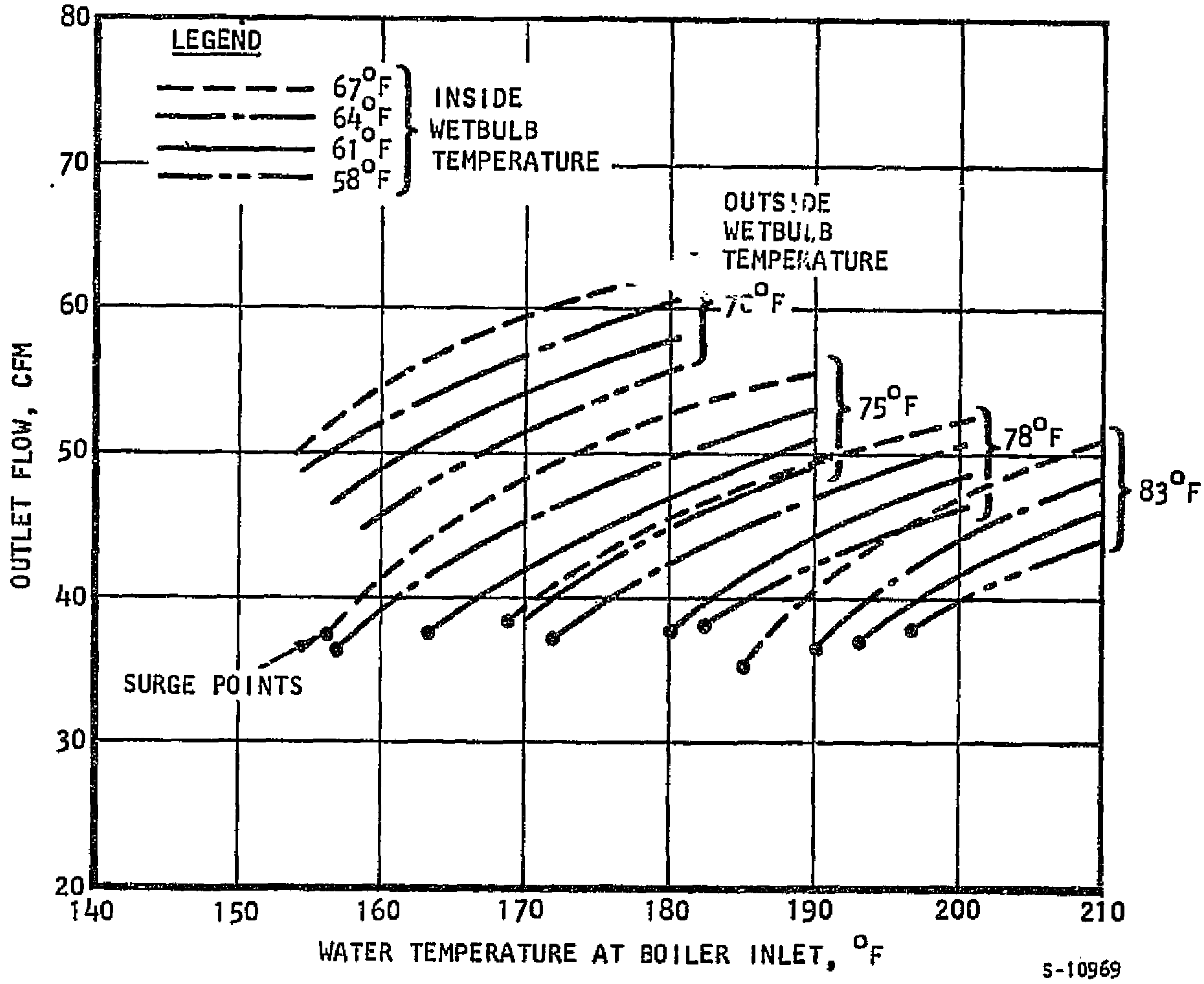

Figure 3-10. 10-Ton Compressor Surge Characteristics

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TABLE 3-1

3-TON COOLING SUBSYSTEM PERFORMANCE SUMMARY

AMBIENT TEMPERATURE: 95°F Drybulb, 78°F Wetbulb

RESIDENCE TEMPERATURE: 78°F Drybulb, 67°F Wetbulb

Water temperature at boiler inlet	200°F	170°F	160°F	150°F
Operating mode	Turbine Drive	Turbine Drive	Motor Augmented	Motor Drive
Capacity, tons	3.1	2.3	3.17	3.17
Thermal COP = $\frac{\text{Evaporator Load}}{\text{Boiler Heat Input}}$	0.55	0.58	1.10	--
Electrical COP = $\frac{\text{Evaporator Load}}{\text{Total Electrical Power Input}}$	8.0	6.0	4.5	4.2

3.3.2 Subsystem Performance

The system is a nominal 10-ton rated air conditioner at the design point conditions. These design conditions represent a worst-case situation (2-1/2 percent time). Normally the cooling subsystem will operate under much more favorable conditions, particularly with regard to the outside wetbulb temperature.

Figure 3-11 is a plot of subsystem COP and capacity as a function of heat source water temperature. COP is defined as the ratio of the evaporator load (cooling effect) to the boiler heat input. The data are shown for the residence conditions noted and for a range of ambient wetbulb temperatures.

Examination of the data shows a marked increase in COP and capacity as the ambient wetbulb temperature drops. The capacity of the system drops as the boiler inlet water temperature decreases. The minimum capacity of the system without motor augmentation is about 6.2 tons. If the capacity drops below this value, motor augmentation will be necessary.

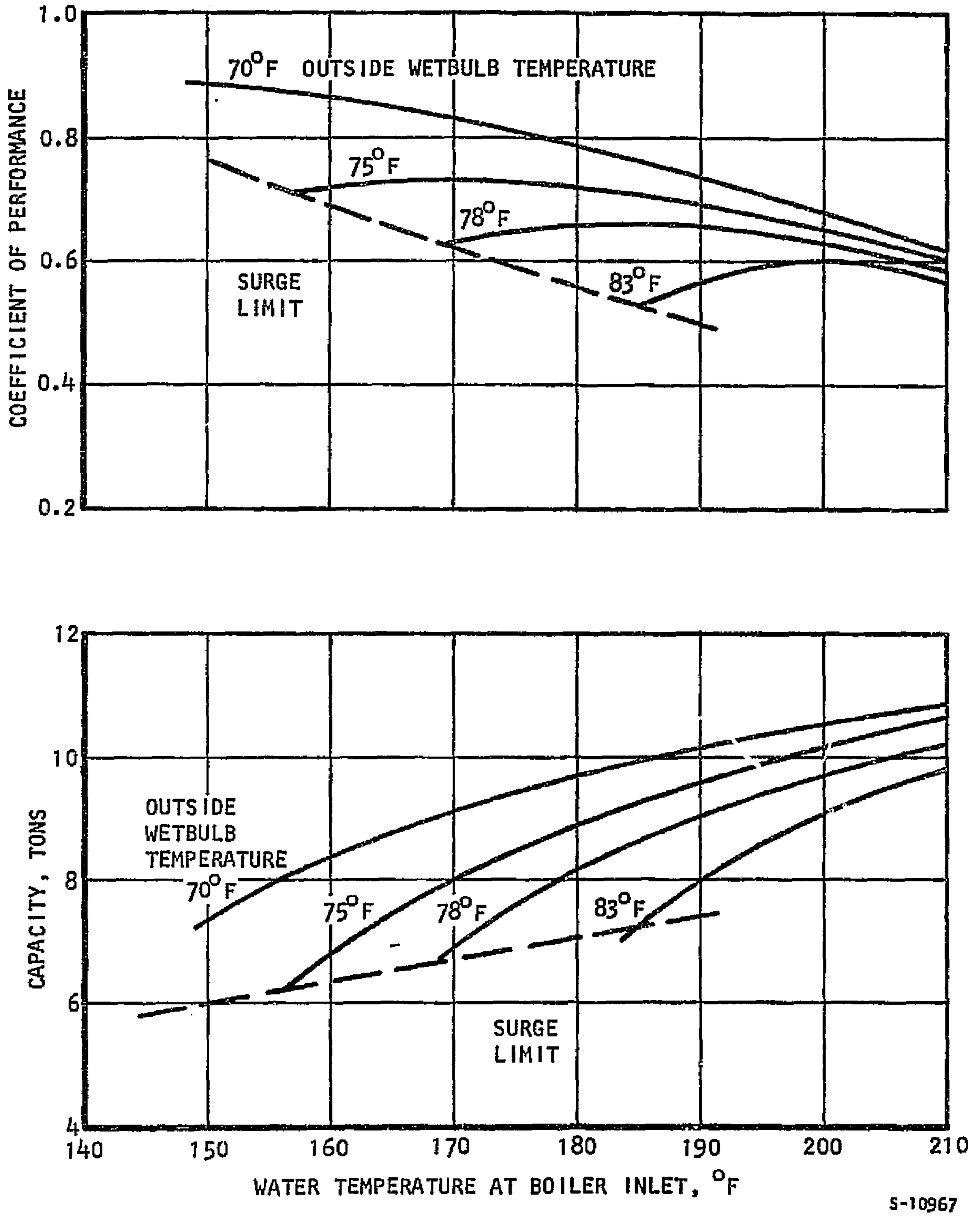

Figure 3-11. 10-Ton Cooling Subsystem Performance With Turbine Drive

Figure 3-12 shows subsystem performance over its entire range of operation: (1) normal mode with turbine drive only, (2) augmented mode with motor-assisted turbine drive, and (3) auxiliary mode with motor drive only. The data are shown for the design point conditions specified, and for off-design operating situations (including a very humid climate condition).

Turbine performance in the cooling mode is presented in Figure 3-13. The percentage of energy supplied by the turbine from heat storage and turbine efficiency values in the motor augmented mode are shown up to the storage temperature, which permits 100 percent turbine power. As the ambient wetbulb temperature increases, more heat energy is required to attain the minimum speed for motor cutout due to increased compressor horsepower per ton and less turbine power per pound of flow. In the figure, as the wetbulb increases from the 78°F design point to an extreme of 83°F, the required storage water temperature for 100 percent use of heat storage energy increases from 168°F to 185°F. Above the range of motor augmentation, the turbine efficiency increases, reaching a maximum of 81 percent.

Figure 3-14 is a compressor performance map with an overlay of water storage temperature and ambient wetbulb temperature. A single point on the compressor map is defined for any set of controlling conditions. When the compressor flowrate drops to a value near the surge line, the motor is energized, returning the compressor to the normal operational range. Compressor design has been optimized to provide peak efficiencies in the high outside wetbulb range and in the entire range of storage water temperature where the turbine is energized (above 155°F).

Subsystem performance is summarized in Table 3-2 for several storage water temperatures in the range of turbine operation. The significant factor

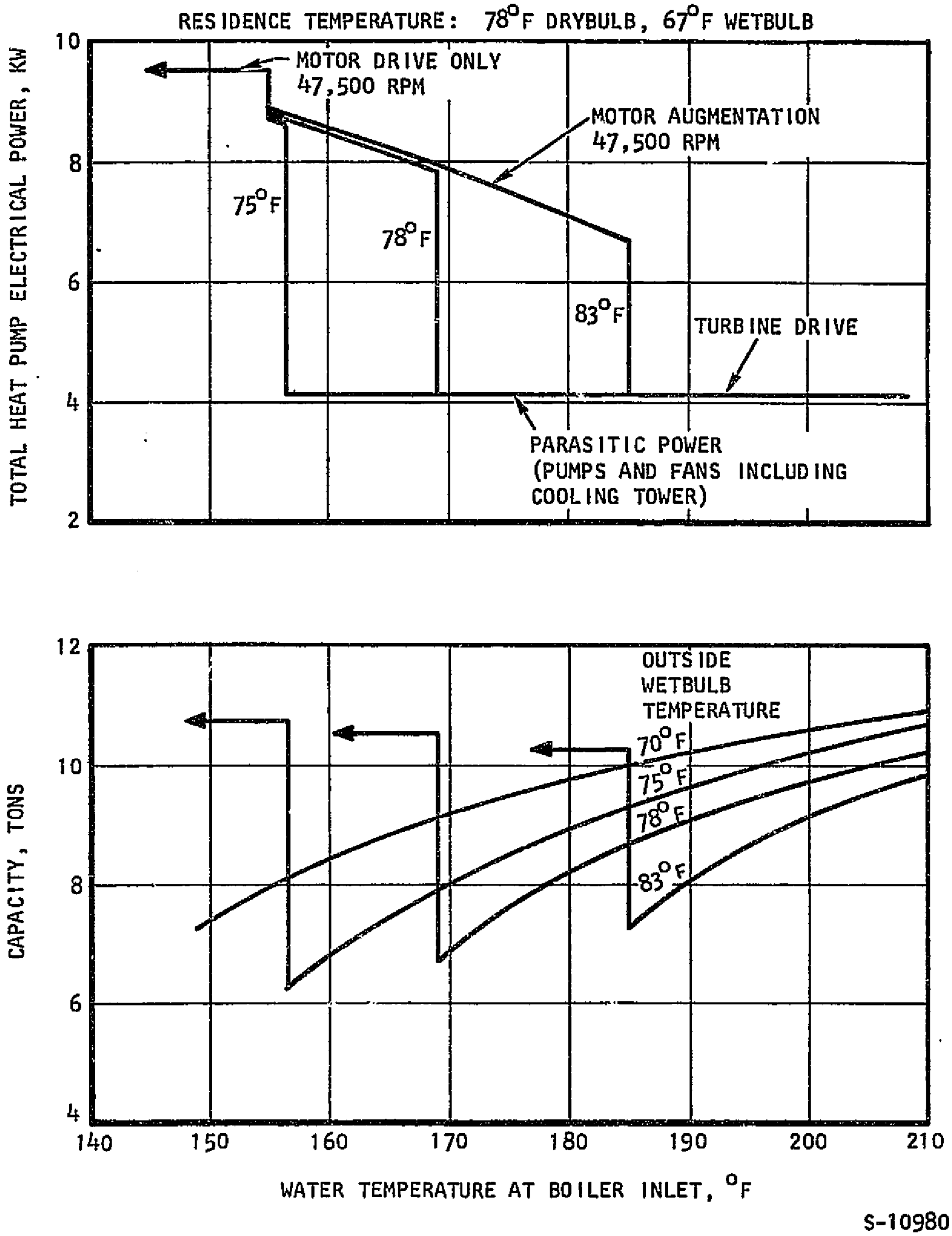

Figure 3-12. 10-Ton Heat Pump Performance

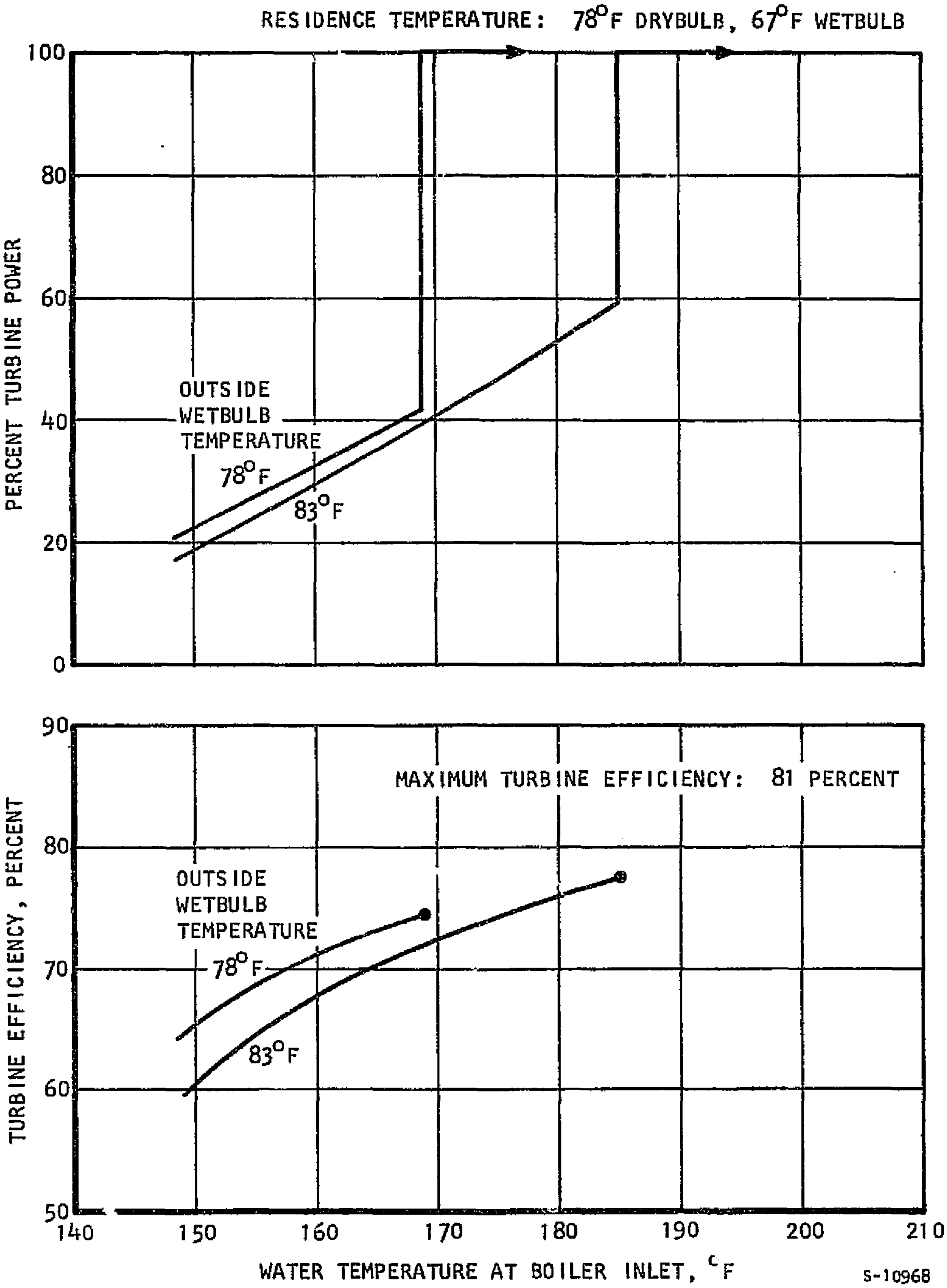

Figure 3-13. Turbine Performance in Augmented Mode, 10-Ton System

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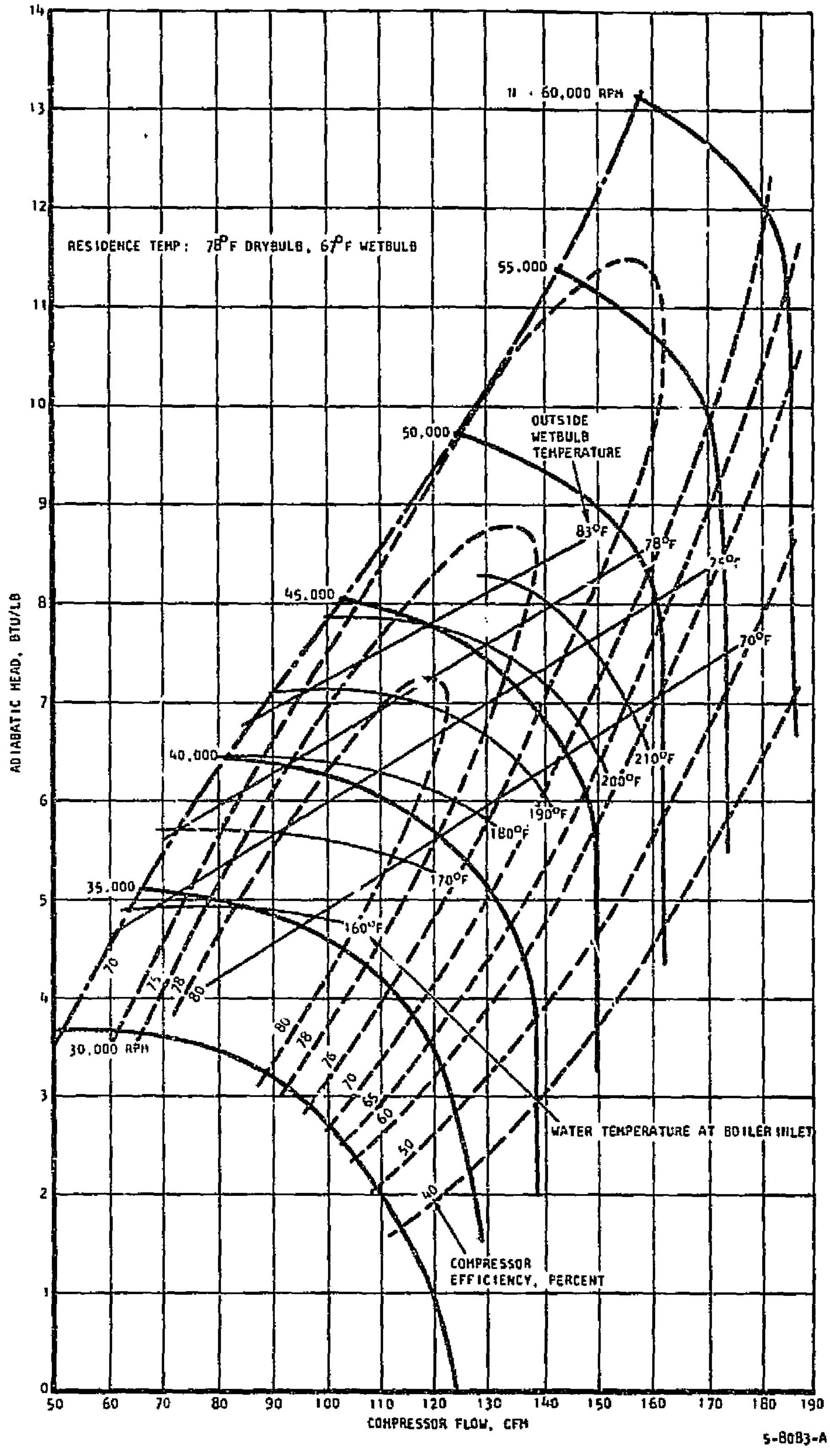

Figure 3-14. 10-Ton Air Conditioner Operating Range

is electrical COP. With 200°F water, heat energy is used to increase the electrical COP to a high value of 8.2. As the water temperature drops, less energy is available, and the COP drops accordingly. When the water temperature falls below 155°F, the compressor is motor driven only and a more typical COP of 3.9 results.

TABLE 3-2

10-TON COOLING SUBSYSTEM PERFORMANCE SUMMARY

OUTSIDE TEMPERATURE: 95°F Drybulb, 78°F Wetbulb

RESIDENCE TEMPERATURE: 78°F Drybulb, 67°F Wetbulb

Water temperature at boiler inlet, °F	200	180	160	150
Operating mode	Turbine Drive	Turbine Drive	Motor Augmented	Motor Drive
Capacity, tons	9.6	8.2	10.5	10.5
Thermal COP = $\frac{\text{Evaporator Load}}{\text{Boiler Heat Input}}$	0.63	0.66	1.21	--
Electrical COP = $\frac{\text{Evaporator Load}}{\text{Total Electrical Power Input}}$	8.2	7.0	4.4	3.9

3.4 MULTIFAMILY RESIDENCE SPACE HEATING AND COOLING SUBSYSTEM 25-TON/ 600 KBTUH HEAT PUMP

3.4.1 Subsystem Arrangement and Controls

Figure 3-15 is a schematic of the 25-ton/600 KBTUH subsystem. Figure 3-16 describes the flowpaths for the heating and cooling modes. This subsystem is schematically identical to the single-family residence and commercial application units, except that a water chiller heat exchanger is used instead of the air to refrigerant evaporator/condenser. A water circulation loop distributes chilled or heated water to terminal units located in each residence area.

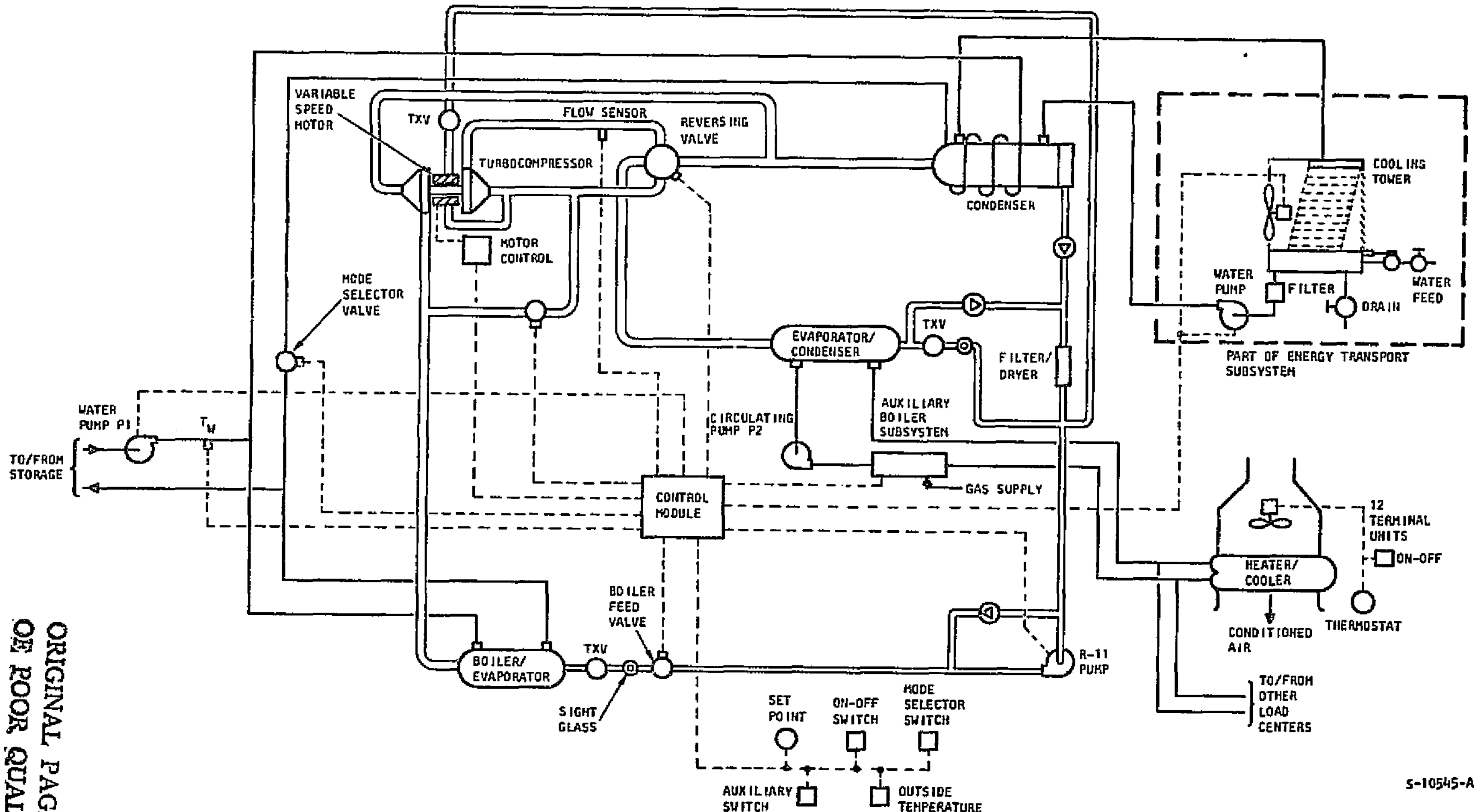

Figure 3-15. Multifamily Residence Space Heating and Cooling Subsystem

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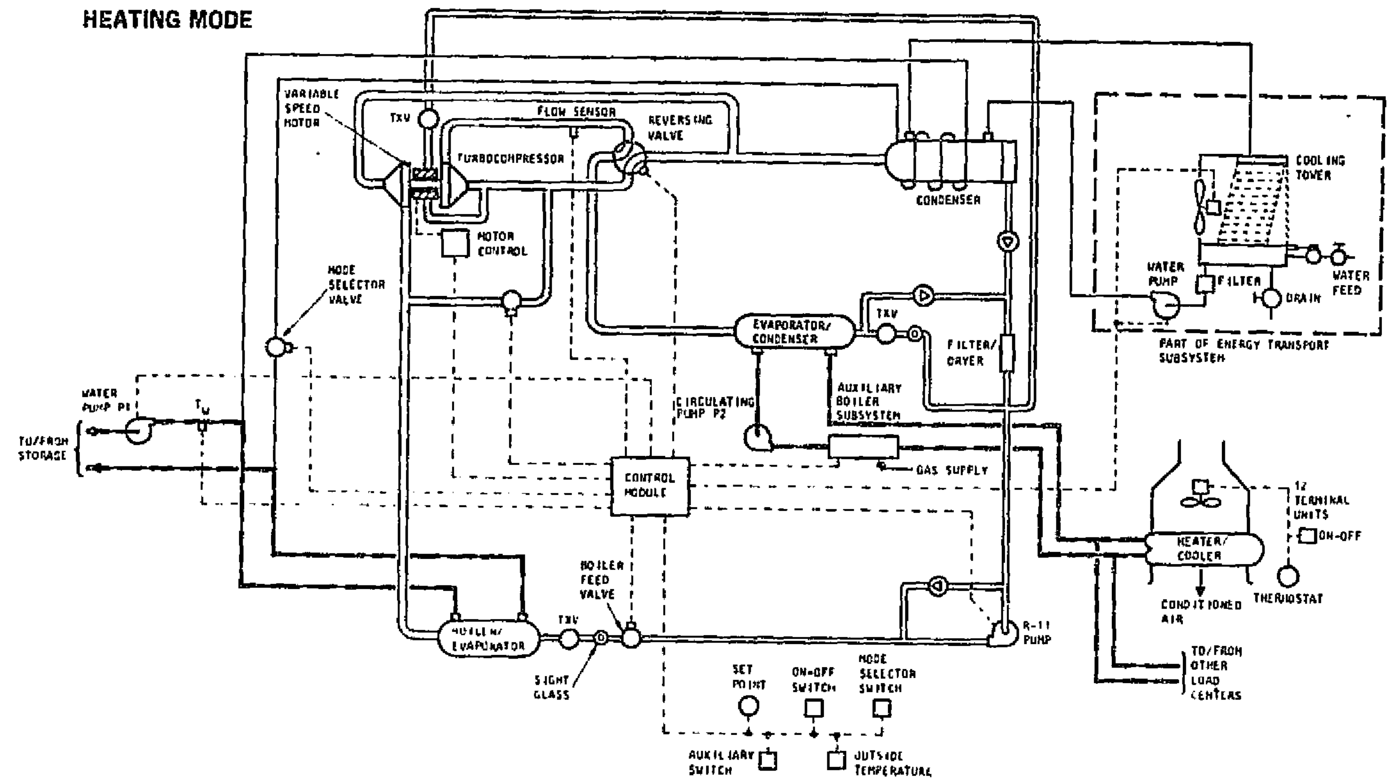
HEATING MODE



COOLING MODE

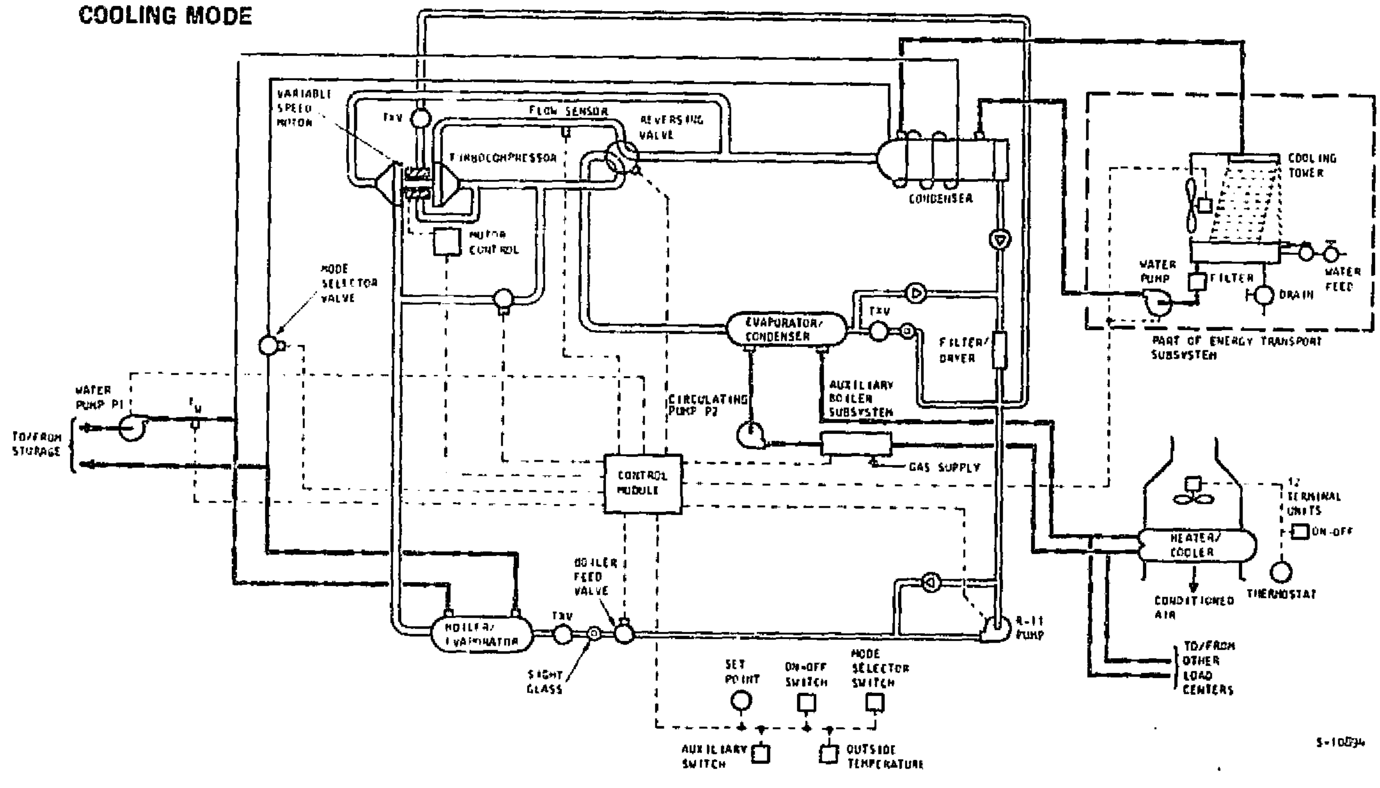


Figure 3-16. 25-Ton Subsystem Flow Paths in Heating and Cooling Modes

Control of the heat pump subsystem differs from the 3- and 10-ton systems in that conditioned water temperature from the heat exchanger is monitored and controlled rather than residence temperature. The heat pump is cycled to maintain a chilled water temperature of 46° to 50°F. Each terminal unit is provided with a separate residence sensor and control that maintains the selected conditions within that space.

Figure 3-17 describes the surge characteristics of the 25-ton turbo-compressor. A near-constant 105 cfm at the outlet during surge is indicated.

3.4.2 Subsystem Performance

Figure 3-18 is a plot of subsystem COP and capacity as a function of heat source water temperature. COP is defined as the ratio of the evaporator load (cooling effect) to the boiler heat input. The data are shown for the residence conditions noted and for a range of ambient wetbulb temperatures.

Examination of the data shows a marked increase in COP and capacity as the ambient wetbulb temperature drops. The capacity of the system drops as the boiler inlet water temperature decreases. The minimum capacity of the system without motor augmentation is about 19.5 tons. If the capacity drops below this value, motor augmentation will be necessary.

Figure 3-19 shows subsystem performance over its entire range of operation: (1) normal mode with turbine drive only, (2) augmented mode with motor-assisted turbine drive, and (3) auxiliary mode with motor drive only. The data are shown for the design point conditions specified, and for off-design operating situations (including a very humid climate condition).

Turbine performance in the cooling mode is presented in Figure 3-20. The percentage of energy supplied by the turbine from heat storage and turbine efficiency values in the motor augmented mode are shown up to the storage

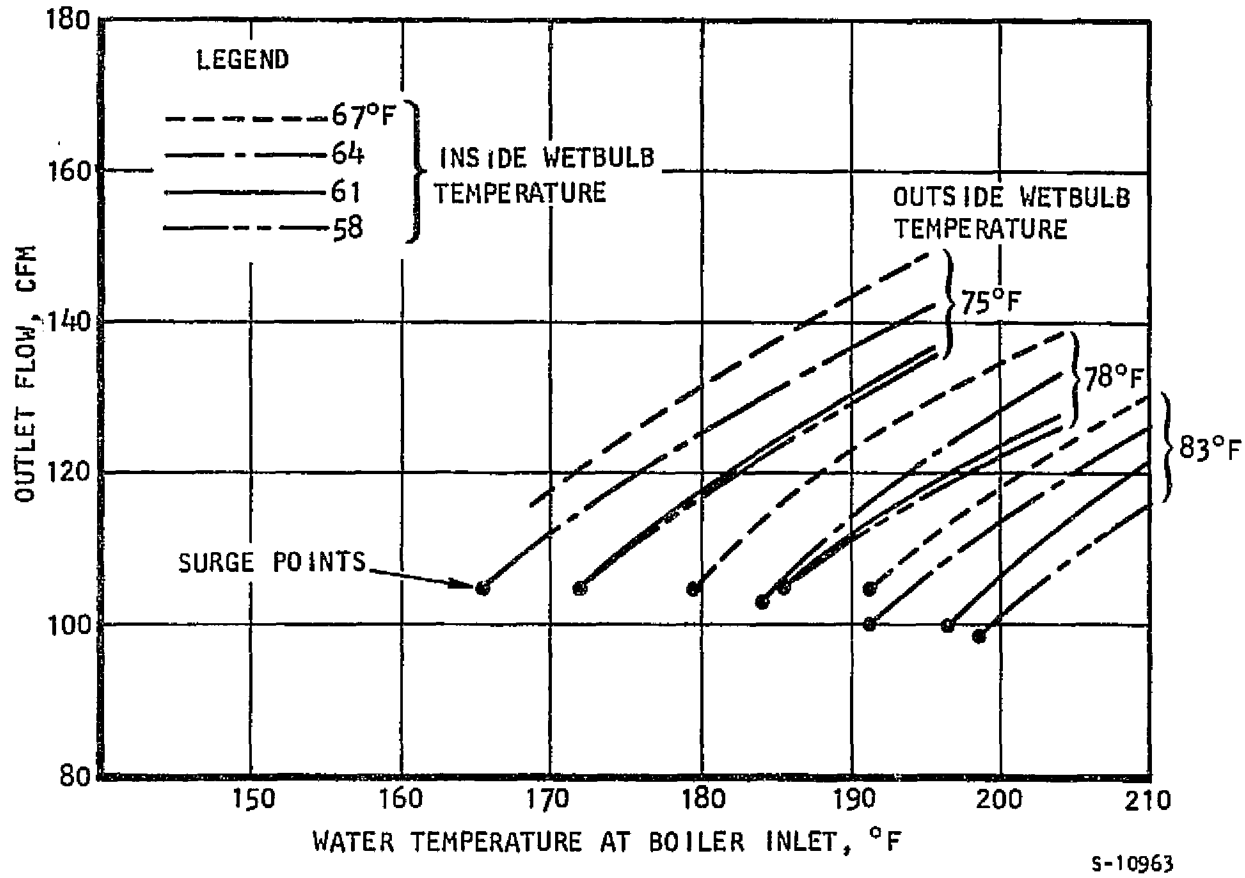


Figure 3-17. 25-Ton Compressor Surge Characteristics

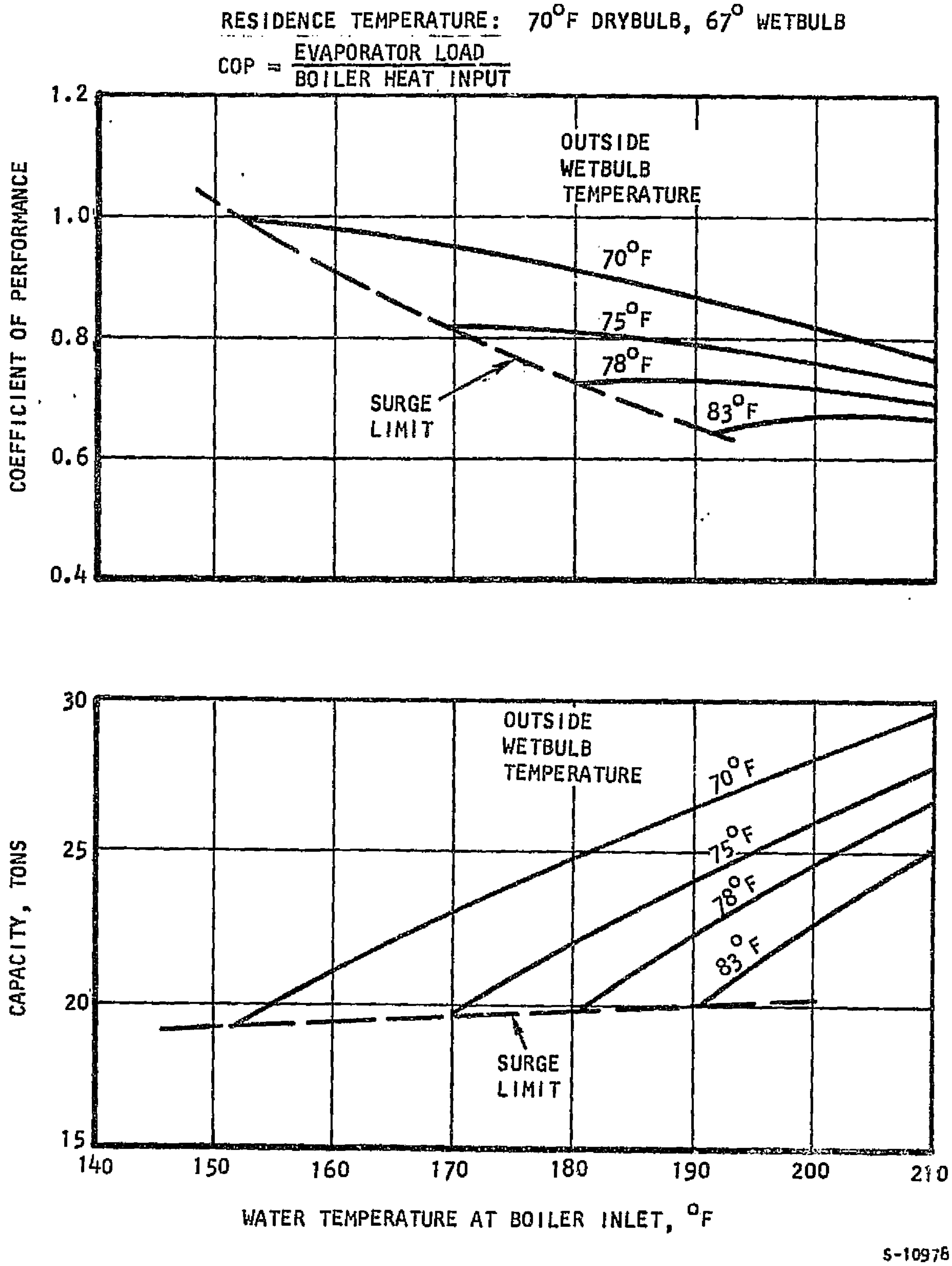


Figure 3-18. 25-Ton Cooling Subsystem Performance With Turbine Drive

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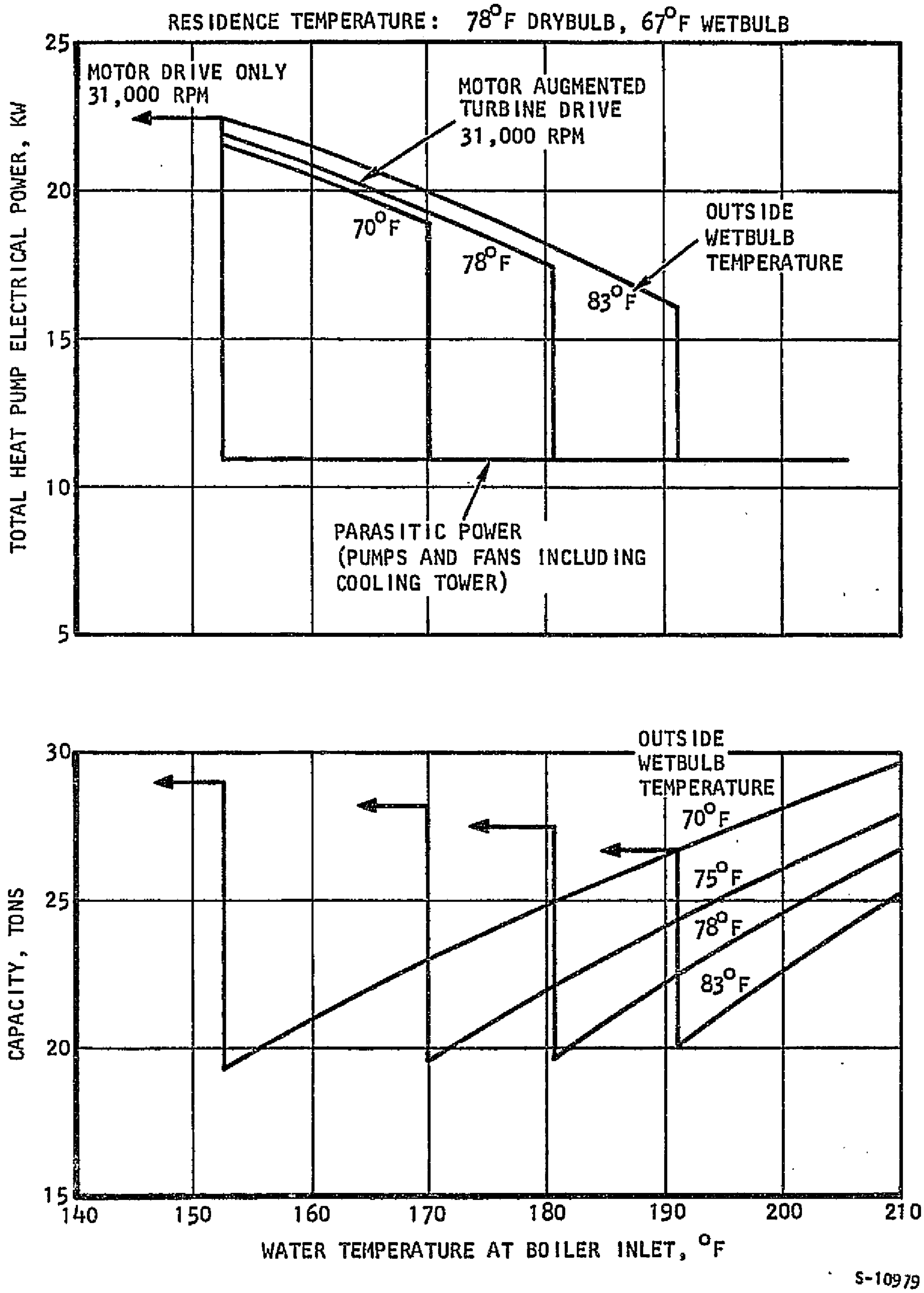

Figure 3-19. 25-Ton Heat Pump Performance

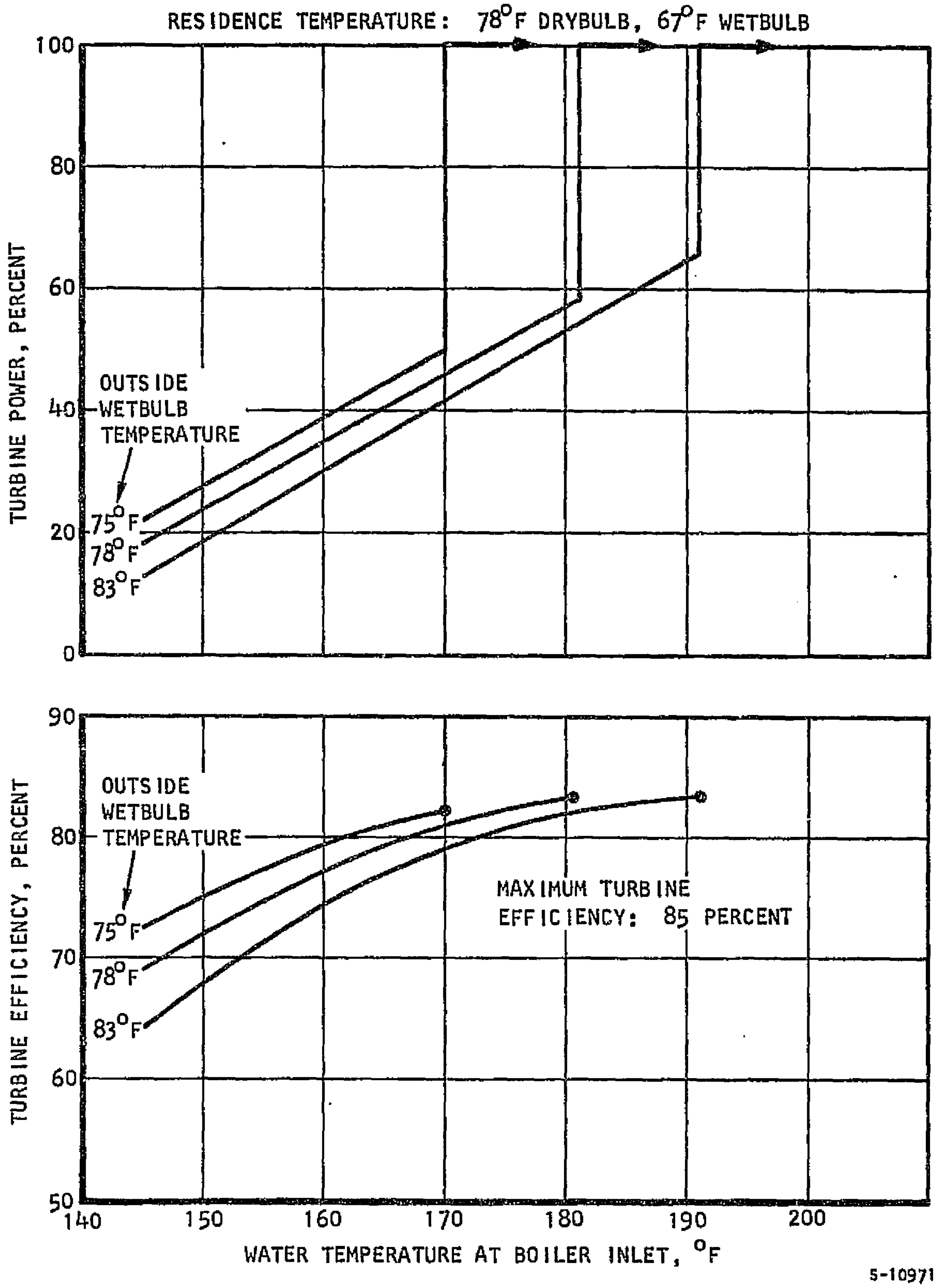

Figure 3-20. Turbine Performance In Augmented Mode, 25-Ton System

temperature, which permits 100 percent turbine power. As the ambient wetbulb temperature increases, more heat energy is required to attain the miminum speed for motor cutout due to increased compressor horsepower per ton and less turbine power per pound of flow. In the figure, as the wetbulb increases from the 78°F design point to an extreme of 83°F, the required storage water temperature for 100 percent use of heat storage energy increases from 182°F to 192°F. Above the range of motor augmentation the turbine efficiency increases, reaching a maximum of 81 percent.

Figure 3-21 is a compressor performance map with an overlay of water storage temperature and ambient wetbulb temperature. A single point on the compressor map is defined for any set of controlling conditions. When the compressor flowrate drops to a value near the surge line, the motor is energized, returning the compressor to the normal operational range. Compressor design has been optimized to provide peak efficiencies in the high outside wetbulb range and in the entire range of storage water temperature where the turbine is energized (above 155°F).

Subsystem performance is summarized in Table 3-3 for several storage water temperatures in the range of turbine operation. The significant factor is electrical COP. With 200°F water, heat energy is used to increase the electrical COP to a high value of 7.9. As the water temperature drops, less energy is available and the COP drops accordingly. When the water temperature falls below 155°F, the compressor is motor driven only, and a more typical COP of 4.3 results.

3.5 DESIGN POINT PERFORMANCE

Design point performance in the cooling mode for the 3-, 10-, and 25-ton systems is shown in Figures 3-22, 3-23, and 3-24, respectively.

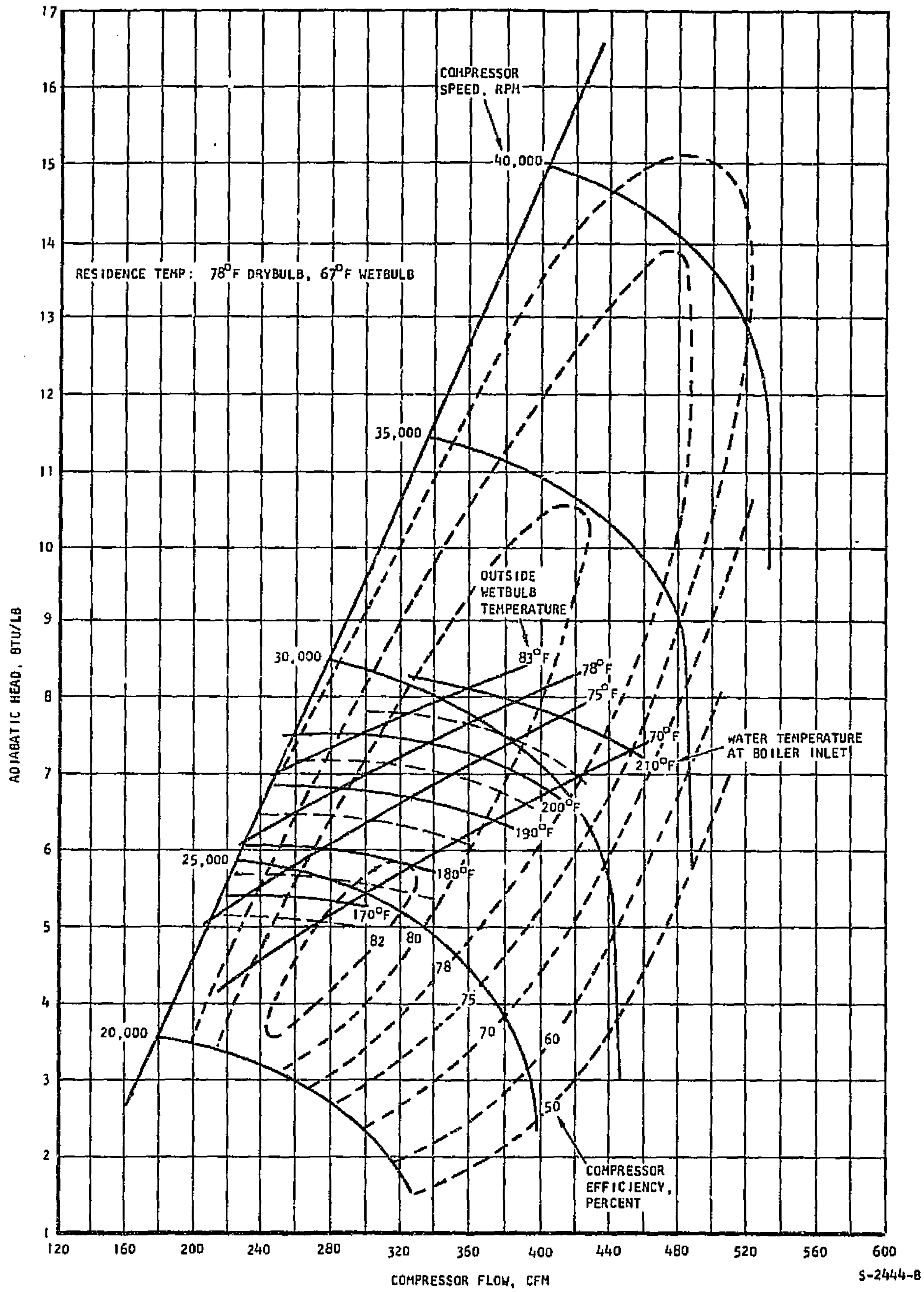

Figure 3-21. Compressor Map, 25-Ton/800,000-Btu/hr Unit

TABLE 3-3

25-TON COOLING SUBSYSTEM PERFORMANCE SUMMARY

AMBIENT TEMPERATURE: 95°F Drybulb, 78°F Wetbulb

RESIDENCE TEMPERATURE: 78°F Drybulb, 67°F Wetbulb

Water temperature at boiler inlet	200°F	175°F	150°F
Operating mode	Turbine Drive	Motor Augmented	Motor Drive
Capacity, tons	24.6	27.5	27.5
Thermal COP = $\frac{\text{Evaporator Load}}{\text{Boiler Heat Input}}$	0.72	1.13	--
Electrical COP = $\frac{\text{Evaporator Load}}{\text{Total Electrical Power Input}}$	7.9	5.5	4.3

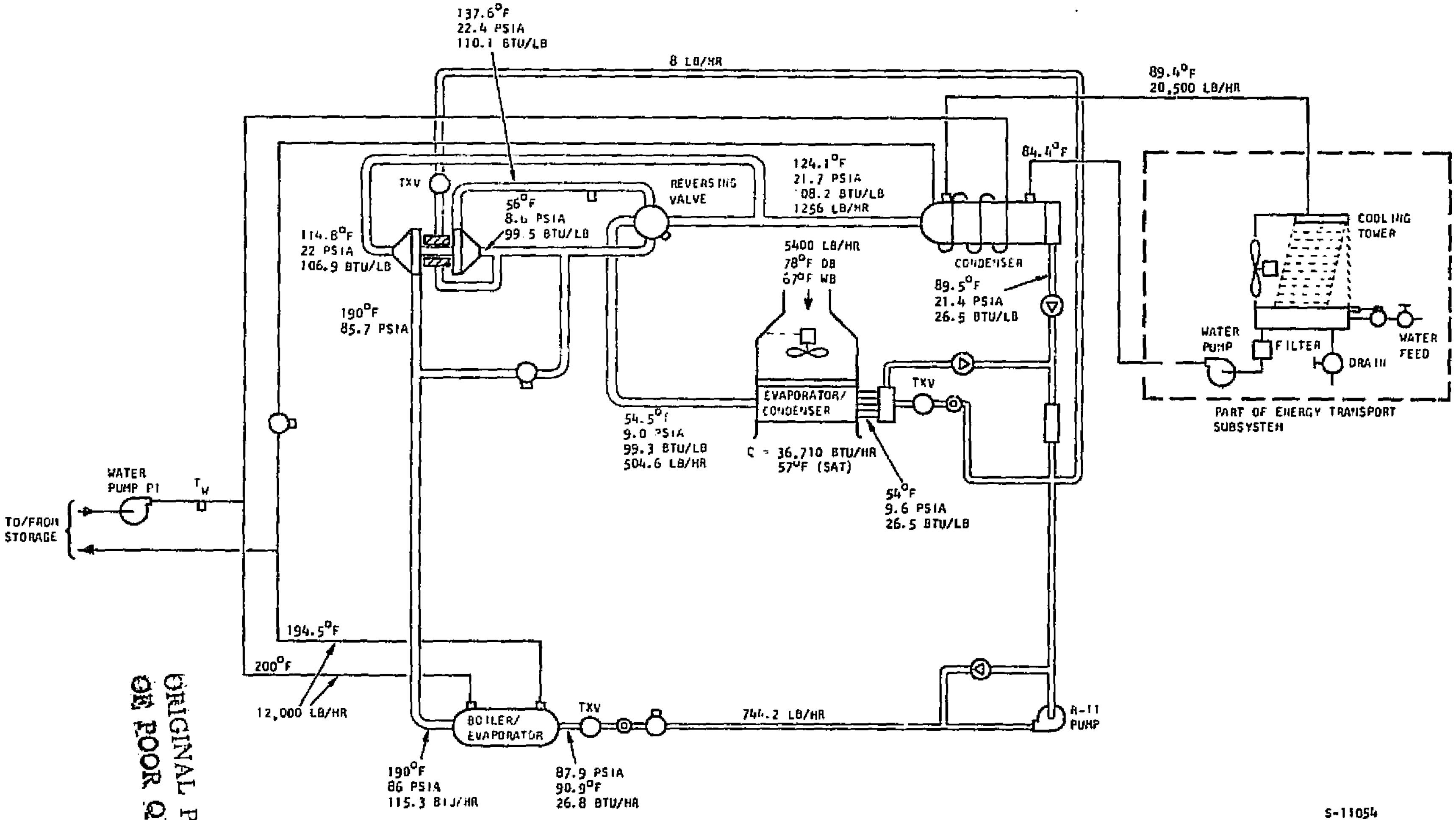

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Figure 3-22. 3-Ton Cooling System Design Point Performance

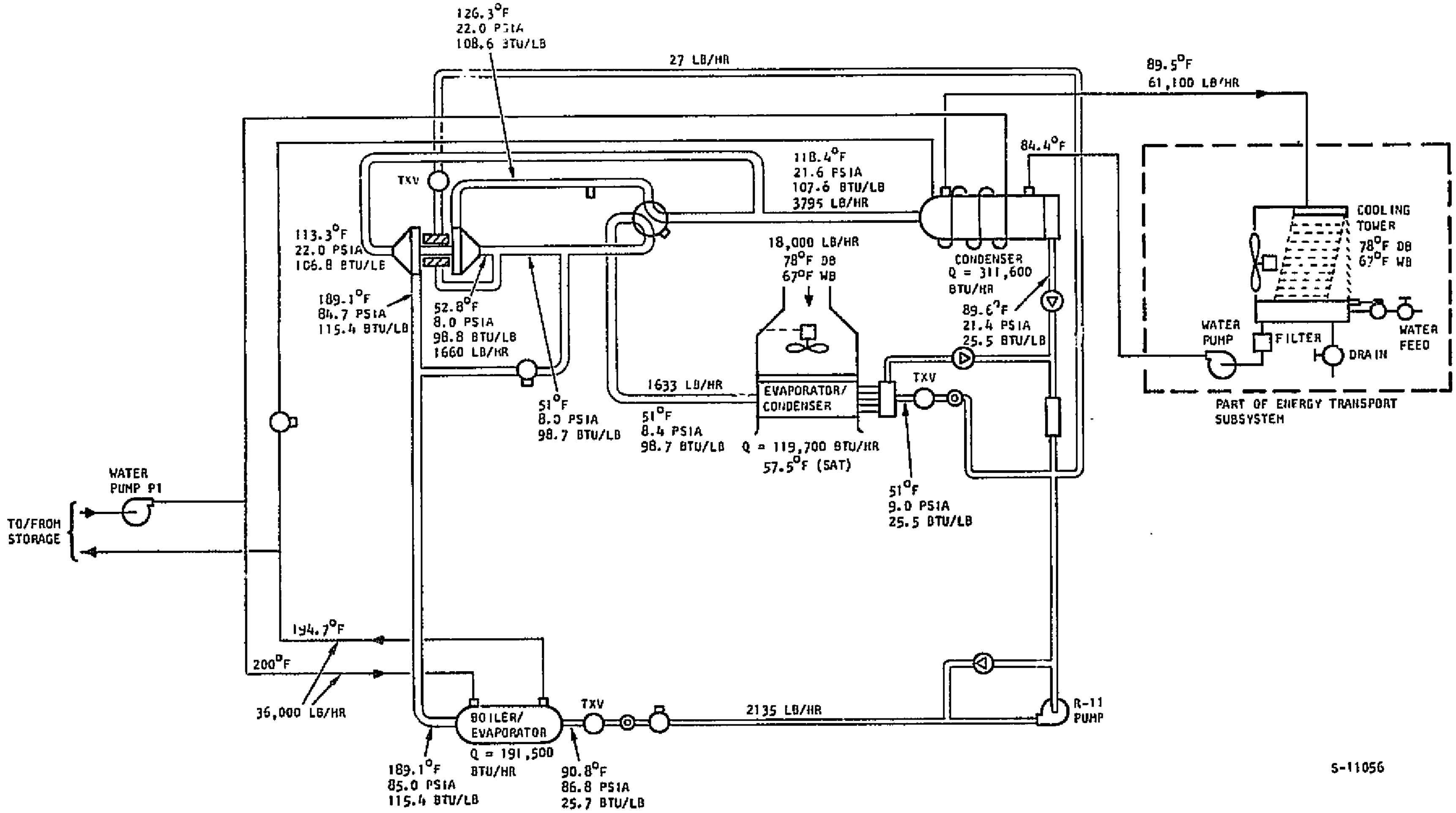


Figure 3-23. 10-Ton Cooling System Design Point Performance

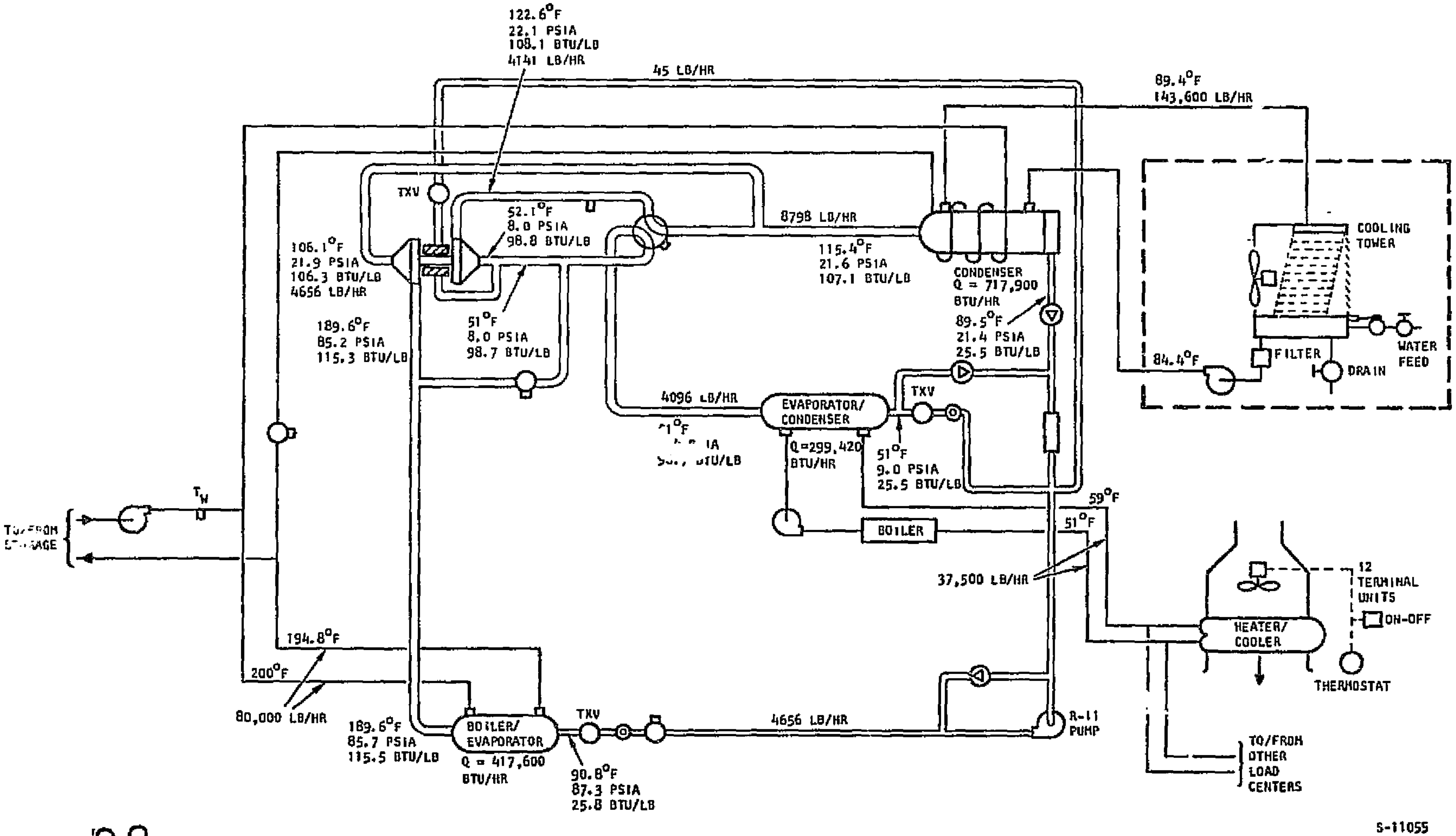


Figure 3-24. 25-Ton Cooling System Design Point Performance

4. TURBOMACHINE/MOTOR DETAIL DESIGN

4.1 GENERAL

This section presents the results of the design effort expended to date on the heat pump turbomachines and motors. The basic design of all units is similar; similarity will minimize assembly tooling and procedures.

Detail design efforts have been expended in the following major areas for all machines.

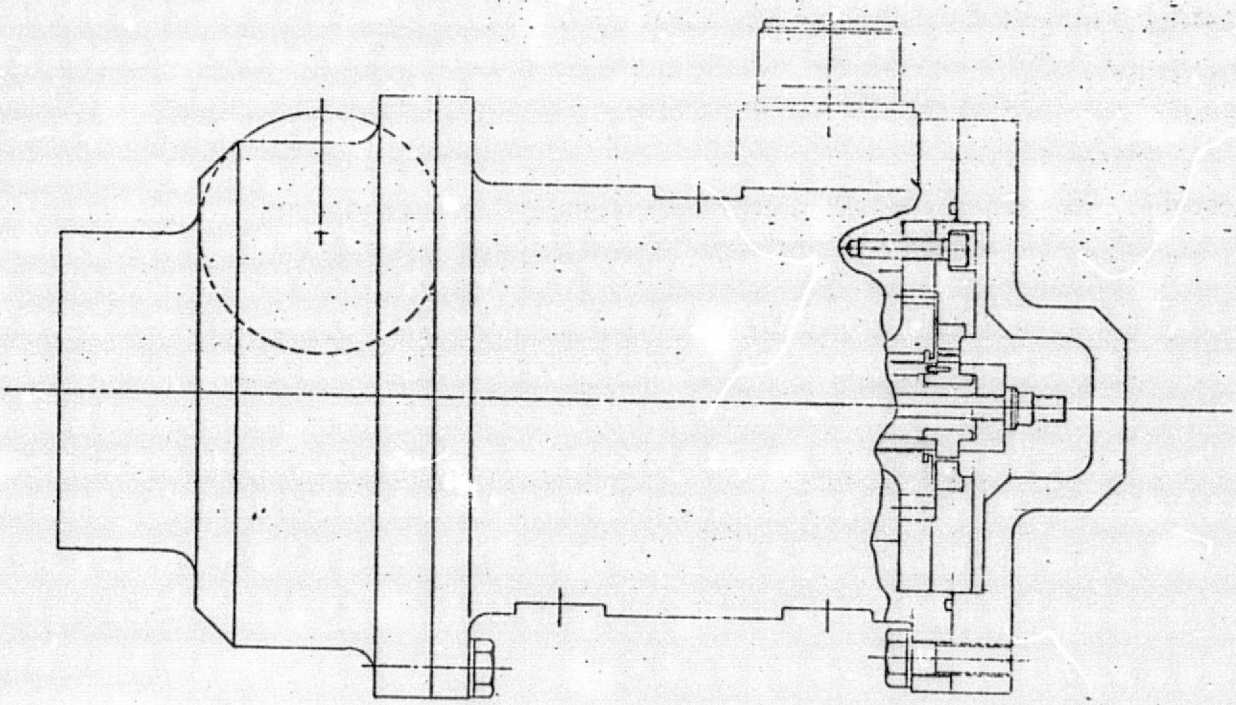
- Motor
  - Magnet cylinder
  - Position/speed sensor
  - Stator connector
- Compressor and diffuser
- Turbine and nozzle
- Rotor assembly
  - Bearings
  - Critical speeds
  - Seals
- Cooling and lubricant flow

The layout drawings presented reflect current design status.

4.2 SINGLE-FAMILY RESIDENCE UNIT (3-TON/60 KBTUH)

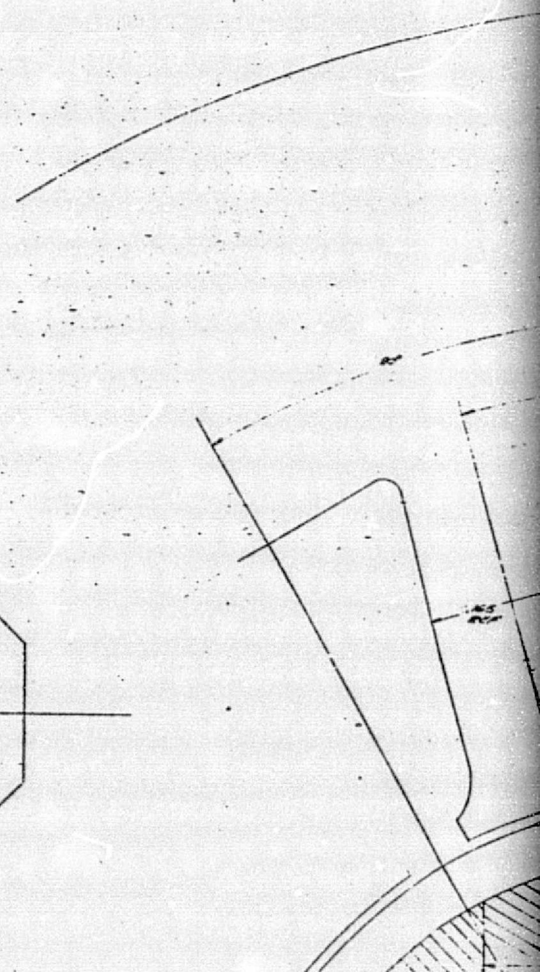
4.2.1 Description

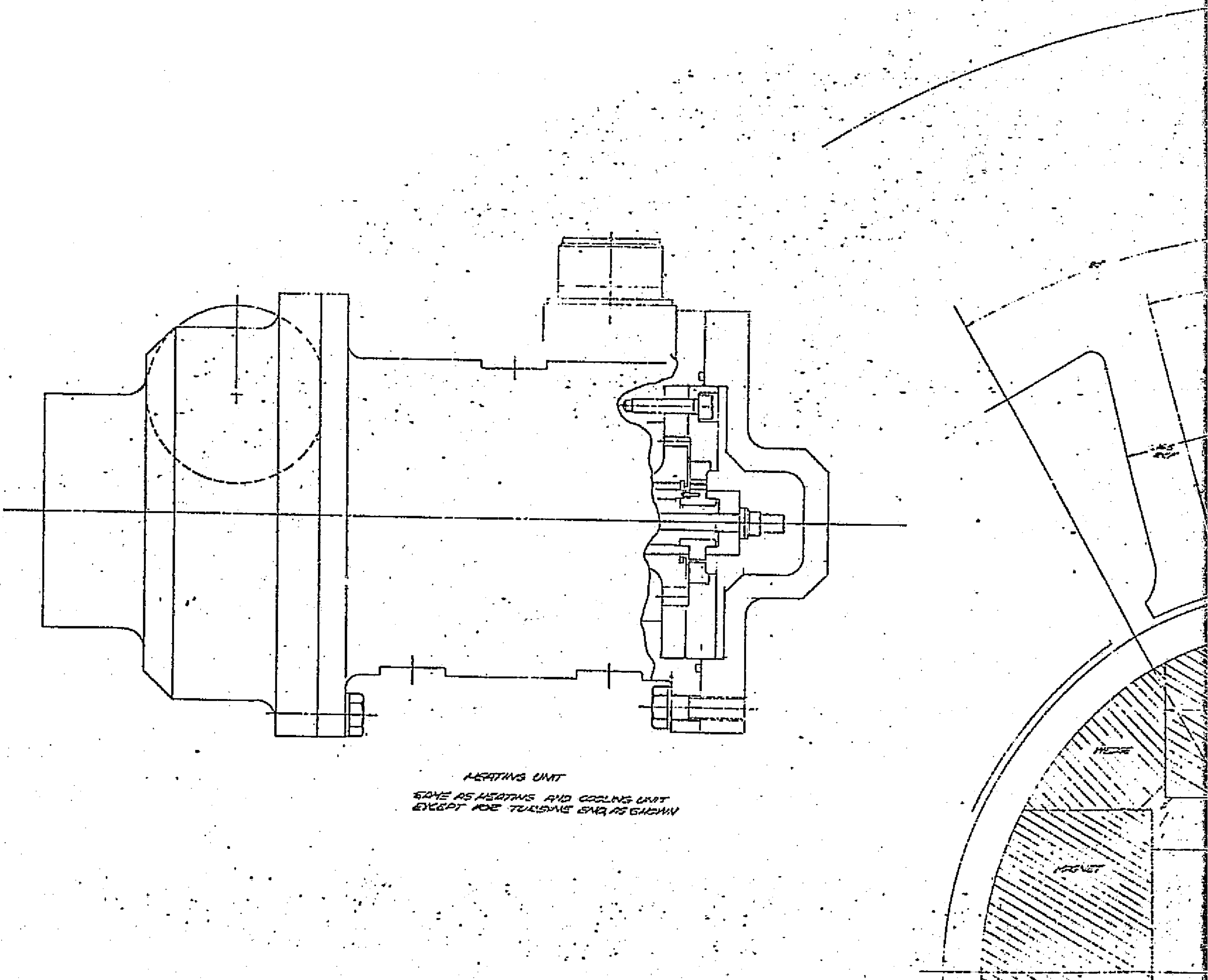
The layout of this machine is shown in Drawing SK71622, Rev. 1; a layout of the motor is shown in Drawing L-208679.



HEATING UNIT

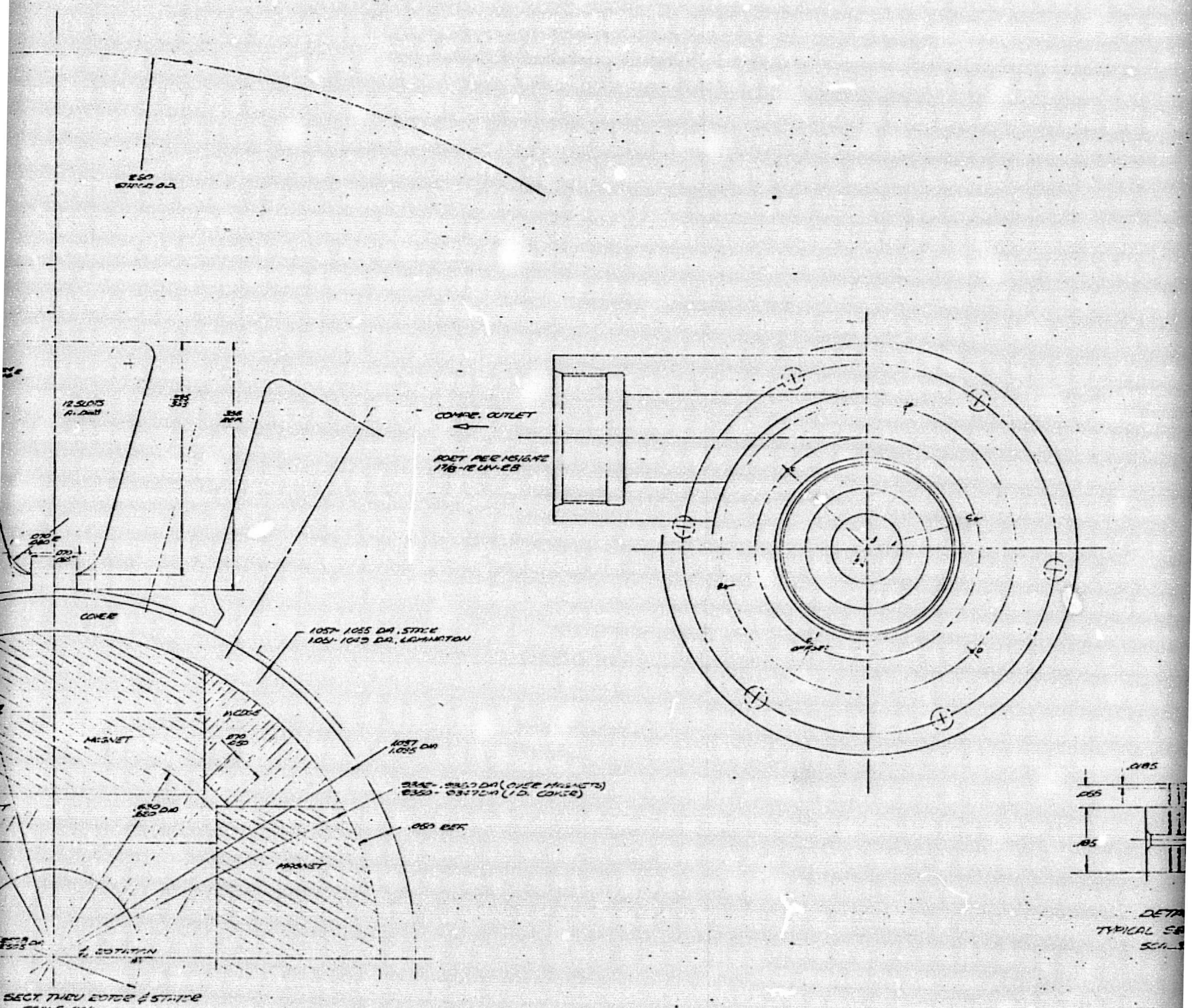
SAME AS HEATING AND COOLING UNIT
EXCEPT FOR TURBINE END, AS SHOWN



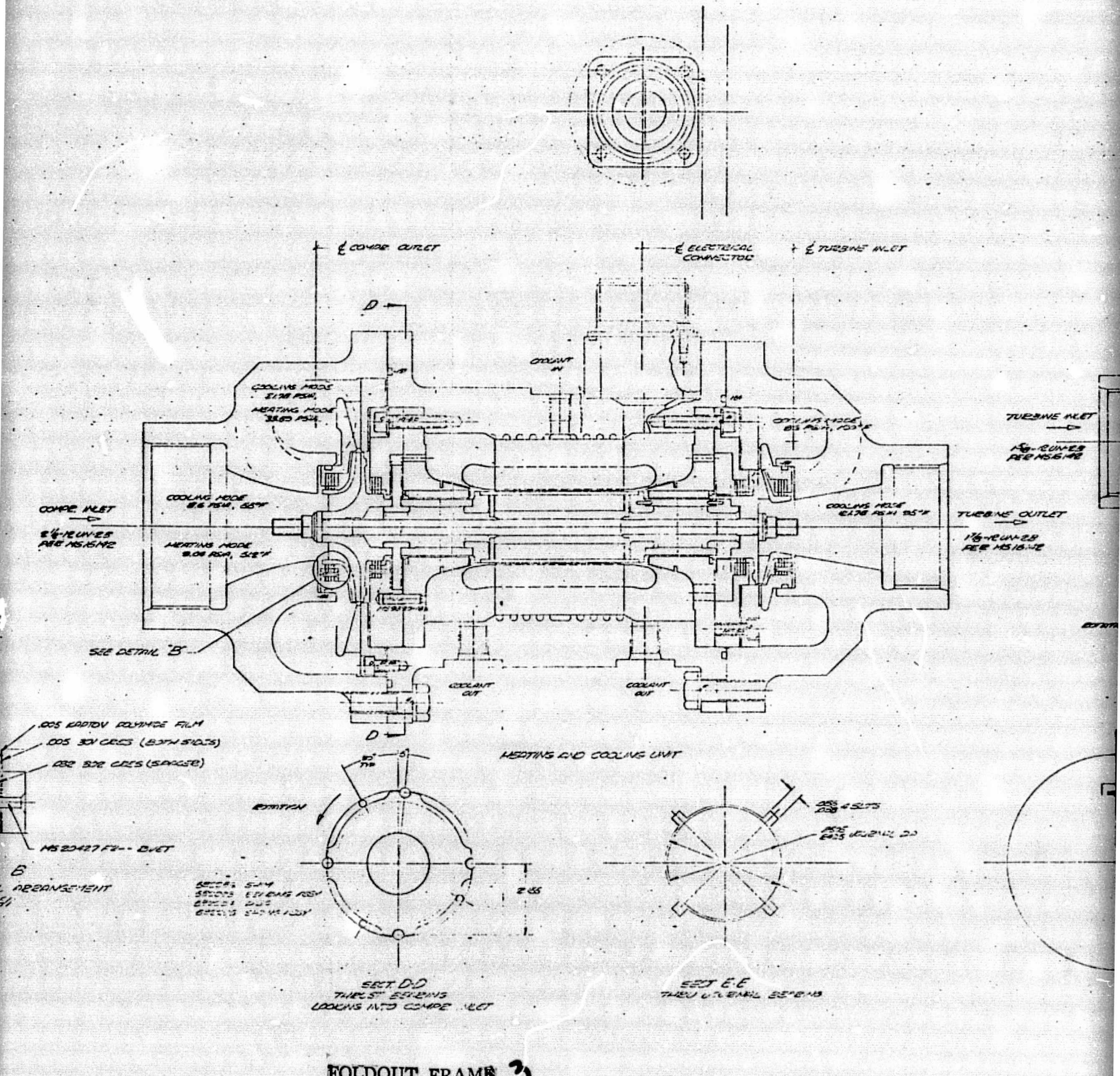


HEATING UNIT

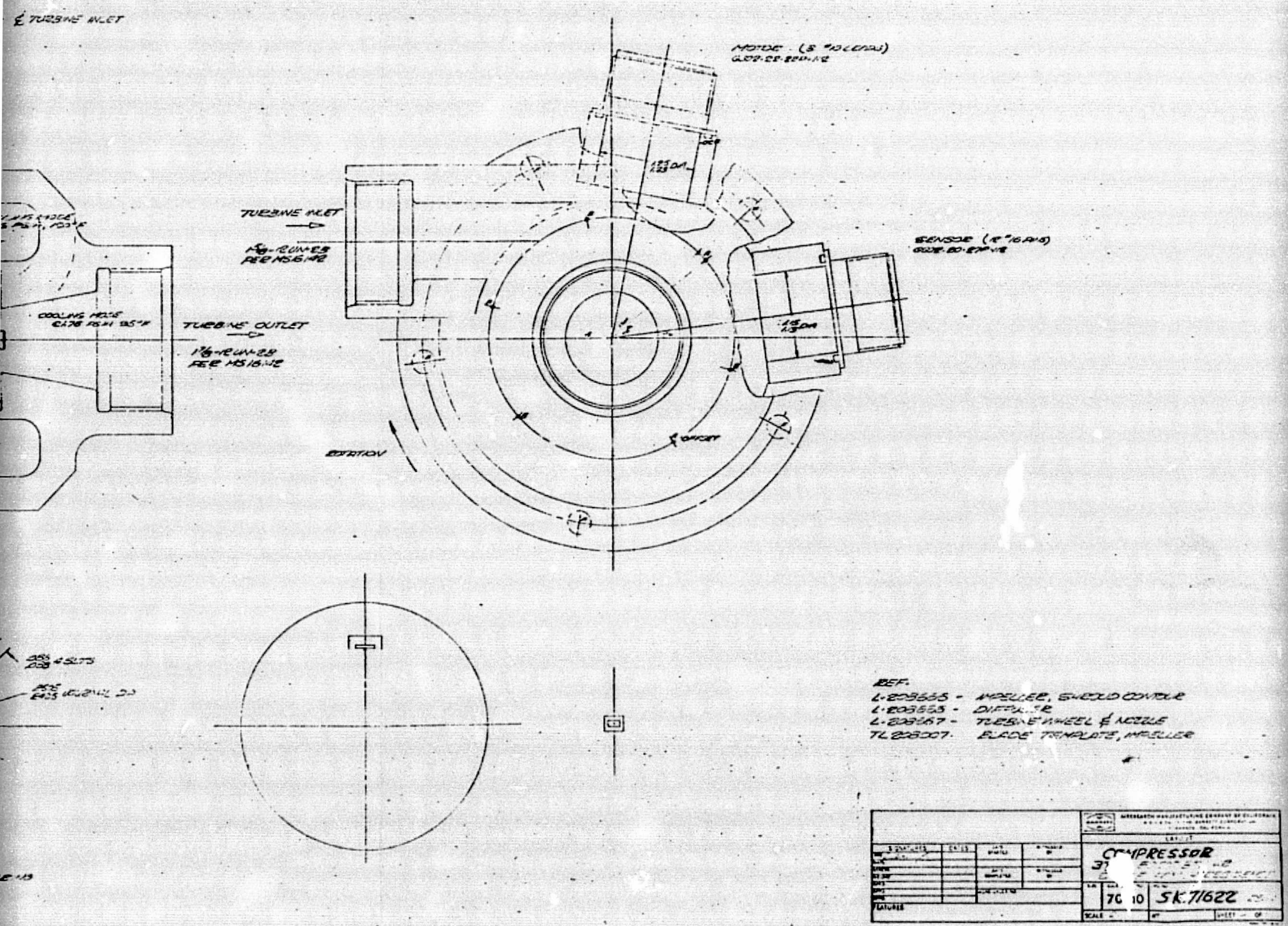
SAME AS HEATING AND COOLING UNIT
EXCEPT FOR TURBINE END AS SHOWN



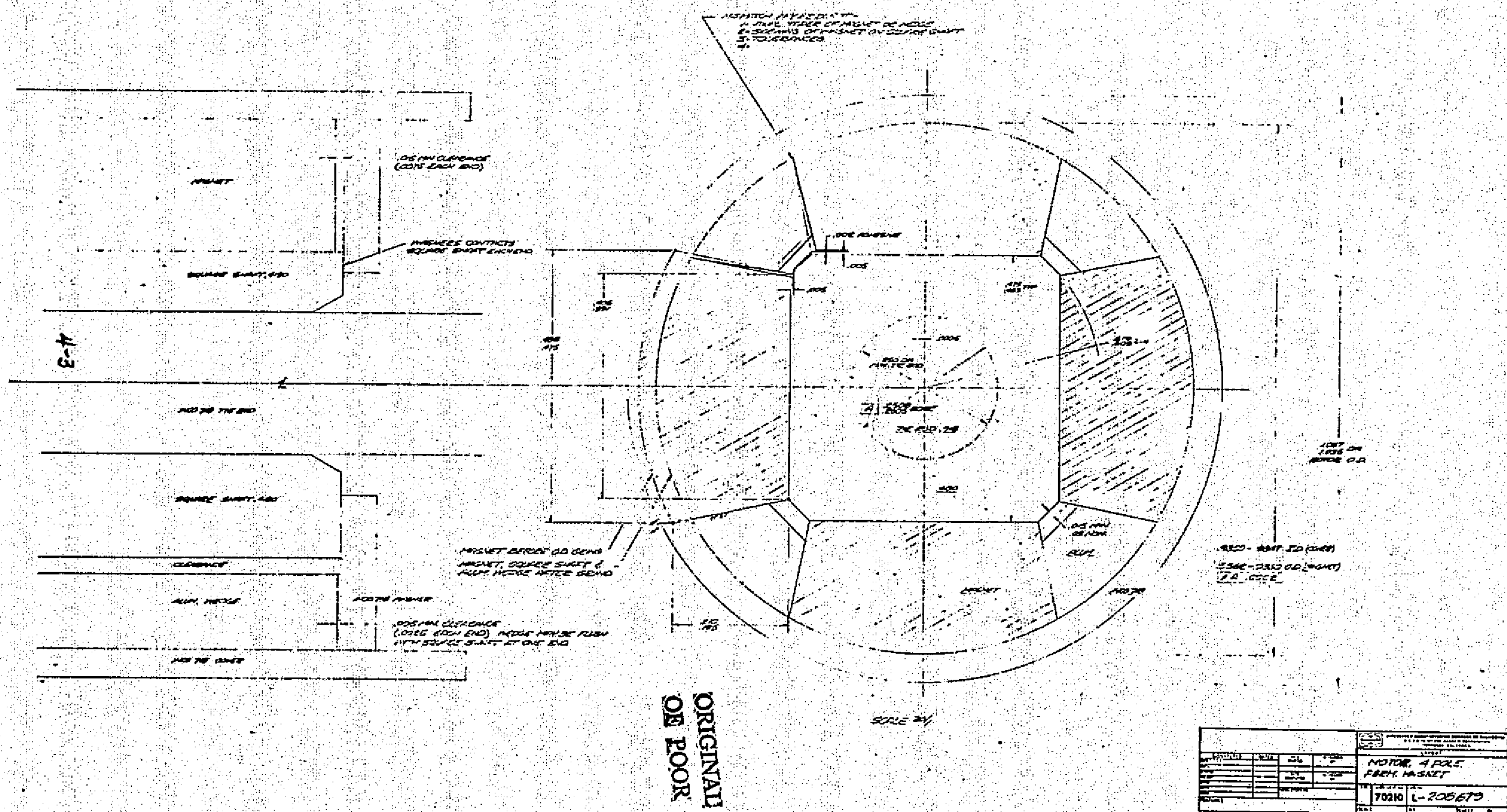
FOLDOUT FRAME
2



FOLDOUT FRAME 3



FOLDOUT FRAME 4


MOTOR, 4 POLE
PERM. MAGNET
70210 L-205679

The major element of the machine is the rotating assembly, which comprises the permanent magnet (PM) motor rotor, the compressor wheel, the thrust bearing disk, two journal bearings, two retaining rings for balancing, the speed and position sensing disk, and the turbine wheel. A solid shaft made from Inconel 718 extends the full length of the rotating assembly. The center section of the shaft contains the motor rotor, which is shown in Drawing L-208579. The complete assembly is clamped together with some individual components brazed or shrunk onto the shaft. Nuts on either end of the shaft lock the total assembly together.

The center housing is made of aluminum and contains the motor stator. The compressor scroll and turbine torus are aluminum; they bolt to the center housing. Sandwiched between torus, scroll, and center housings are the turbine nozzle assembly, the compressor diffuser assembly, and the labyrinth seals to prevent leakage around the wheels. Shims are provided at both ends of the machine for proper positioning of the nozzle and diffuser assemblies in relation to the rotating assembly. O-ring seals are provided to prevent any external leakage. Because of the small mass of the rotating assembly, no special provision is included in the design for containment.

The speed/position sensing ring is located near the turbine end of the unit and surrounds a slotted ferrite rotating disk. The sensing ring consists of two Hall effect sensors, which are located 180 deg apart around the ring. As the disk rotates, signals are generated in the Hall effect sensors and fed to the control system for proper motor control. Self-acting foil segment journal and thrust bearings are contained in the housing and are held in position by rings and pins respectively.

A small flow of liquid refrigerant for motor and bearing cooling is metered to the center housing. The refrigerant is bled into the motor cavity from the condenser outlet and is returned to the evaporator inlet as a superheated vapor. A thermal expansion valve controls the cooling flow to the motor cavity. The pressure in the cavity is maintained slightly above compressor inlet pressure.

The construction and assembly of the heating-only machine are identical to those of the heating/cooling unit, except that the turbine wheel is replaced with an equivalent weight disk and the torus and nozzle assembly are replaced with a close-off plate.

4.2.2 Design Analysis

All detail design analyses for this unit were based on the problem statement of Table 4-1. The results of the detailed analyses show the following:

(a) The critical speed of this unit is in excess of 150,000 rpm with a maximum operating speed of approximately 82,000 rpm.

(b) The unit rotor stresses and wheel stresses at 20 percent overspeed are well below the design limits for the materials used.

(c) The bearing loads, including magnetic unbalance and motor cogging, are well below the limits demonstrated in the laboratory on bearings identical to those used in this machine.

(d) The maximum temperature expected in the unit is less than 200°F, which is well below any design temperature limitations. The critical parts from a temperature standpoint are the magnets and the Hall effect sensors; their limitations are 300°F.

TABLE 4-1

3-TON TURBOCOMPRESSOR

	Cooling Mode	Heating Mode
Compressor		
Fluid type	R-11	R-11
Inlet pressure, psia	8.6	9.04
Inlet temperature, °F	55	51.2
Inlet H, Btu/lb	99.31	98.79
Outlet pressure, psia	21.78	35.59
Outlet H, Actual, Btu/lb	109.05	115.10
ΔH adiabatic, Btu/lb	7.01	10.93
ΔH actual, Btu/lb	9.74	16.31
Volume flow, inlet, cfs	0.55	0.86
Weight flow, lb/sec	0.122	0.20
Speed, rpm	60,000(constant)	82,000
Theoretical efficiency, percent	72	67
Power at wheel, kw	1.25	3.42
Turbine		
Inlet pressure, psia	86	NA
Inlet temperature, °F	190	
Inlet H, Btu/lb	115.38	
Outlet pressure, psia	21.78	
Outlet H, Btu/lb	106.64	
ΔH adiabatic, Btu/lb	10.93	
ΔH actual, Btu/lb	8.74	
Volume flow, inlet, cfs	0.3833	
Weight flow, inlet, lb/sec	0.150	
Speed, rpm	60,000(constant)	
Theoretical efficiency, percent	80	
Power at wheel, kw	1.316	
Motor		
Maximum speed, rpm	60,000(constant)	82,600
Minimum speed, rpm	NA	36,500
Power at maximum speed, kw	1.4*	3.42**
Power at minimum speed, kw	1.4*	NA
Growth potential, power, percent	NA	20
Growth potential, speed, percent	10	10
Maximum overspeed, rpm	NA	90,860
Pressure, cavity, psia	9.6	9.6
Temperature, cavity, °F	100	150
Viscosity, μ, Reynolds	1.64×10^{-9}	1.75×10^{-9}
Motor efficiency, percent	NA	91.18
Bearings/lubricant, types	Foil/R-11/vapor	Foil/R-11/vapor
Cooling flow, lb/sec (approx)	0.0024	0.0063
Type motor	PM Samco	PM Samco
No-load power	TBD	NA
Maximum design speed, rpm	NA	100,000
General		
Design life, yr	20	20
No. of starts, lifetime	60,000	60,000
Service hours, lifetime	75,000	50,000
Type control	on/off	Modulating

* Power delivered to the compressor assuming 5 percent mechanical losses.

** Power delivered to the compressor assuming no mechanical losses.

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4.2.3 Performance

Estimated motor performance is presented in Table 4-2. The aerodynamic performance of the compressor and turbine is shown in Figures 4-1 through 4-3. These data are presented here for completeness.

4.2.4 Design Status

All detailed drawings have been completed and released. All parts are on order or being fabricated, and assembly of the first unit is scheduled for February 1977.

4.3 MULTIFAMILY RESIDENCE UNIT (25-TON/600 KBTUH)

4.3.1 Description

The layout of this machine is shown in Drawing SK71624, and a layout of the motor is presented in Drawing L-208688. The arrangement of this unit is essentially the same as that of the 3-ton/60 KBTUH machine.

4.3.2 Design Analysis

All detail design analyses for this unit were based on the problem statement shown in Table 4-3.

The results of the detailed analyses show that the critical speed is approximately 60,000 rpm, and the maximum operating speed of the machine is 41,000 rpm. All other criteria--stress, bearing loads, and thermal characteristics--are well within the allowable design limits. The internal temperature profile is essentially the same as for the 3-ton unit.

4.3.3 Performance

The performance of the motor is shown in Table 4-4, and the aerodynamic performance of the compressor and turbine are shown in Figures 4-4 through 4-6.

TABLE 4-2

COMPRESSOR MOTOR CHARACTERISTICS,
3-TON/60 KBTUH HEAT PUMP

Parameter	Units
Rpm	82,600
Kw (shaft)	4.22
Kva (input)	4.80
Pf (lagging)	0.95
Poles, no.	4.0
Phases	3.0
Frequency, Hz	2753
Tip speed, ft/sec	392
Stator current density, amp/sq in.	12,000
Slots, no.	12
Pole embrace	0.58
Stator material	0.007-in. Trancor-T
Magnet material	Samarium cobalt
Dimensions, in.	
Rotor dia	1.076
Rotor length	1.953
Stack ID	1.096
Stack OD	2.660
Stack length	1.953
End turn extension	0.49
Total stator length	2.933
Slot height	0.363
Tooth width	0.174
Slot opening	0.072
Weight, lb	
Stator	2.048
Rotor	0.464
Total weight	2.512
Losses, w	
Windage	45
Copper	132
Stray and pole head	45
Stator teeth	130
Stator core	71.5
Efficiency	0.92
No-load voltage, per unit	1.20
Impedance data, per unit	
Synchronous reactances	0.383
Commutation reactance	0.383
Stator resistance (hot)	0.110
Stator leakage reactance (XL)	0.287

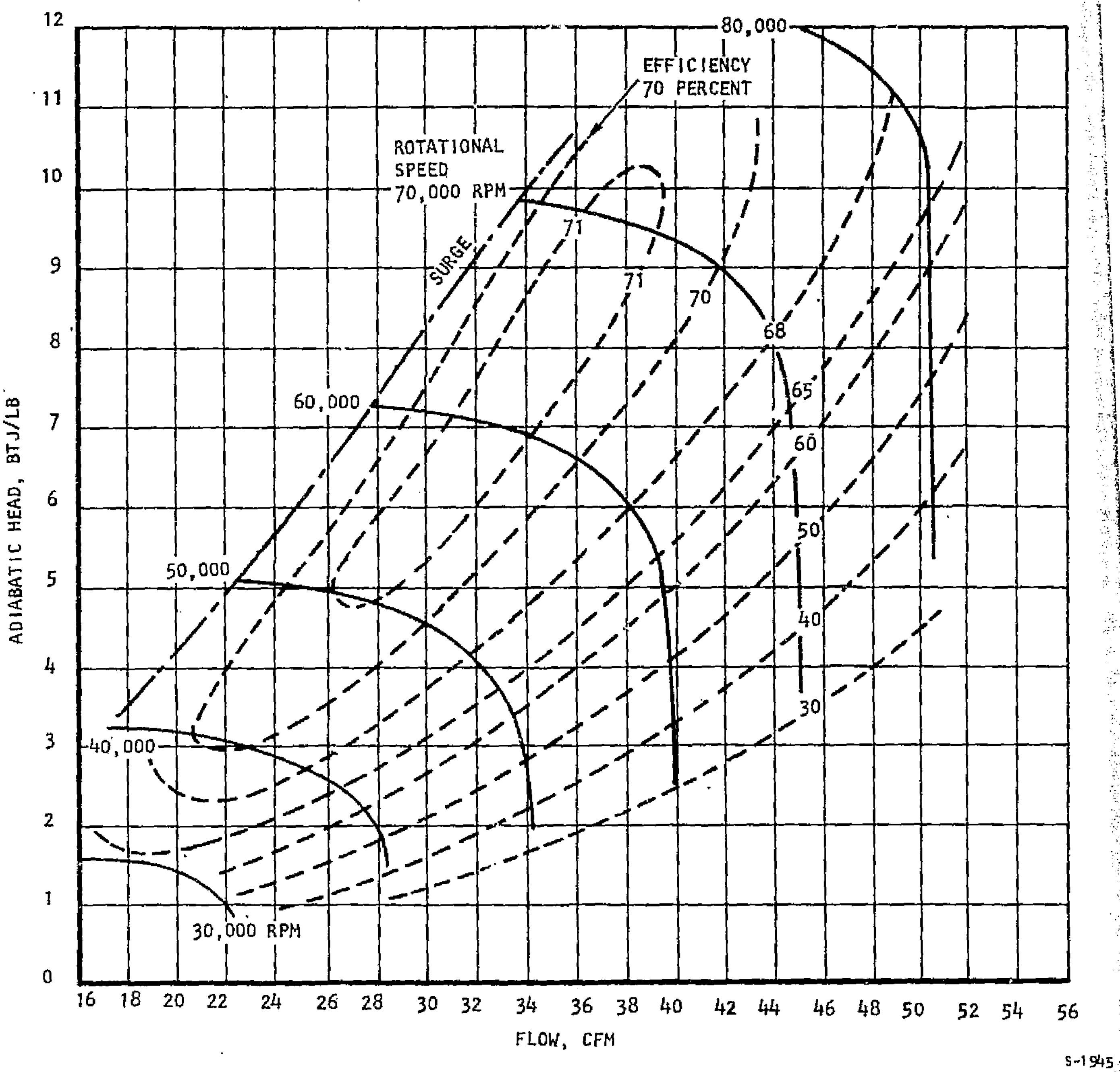

Figure 4-1. Compressor Map (3-Ton/80,000-Btu/hr Unit)

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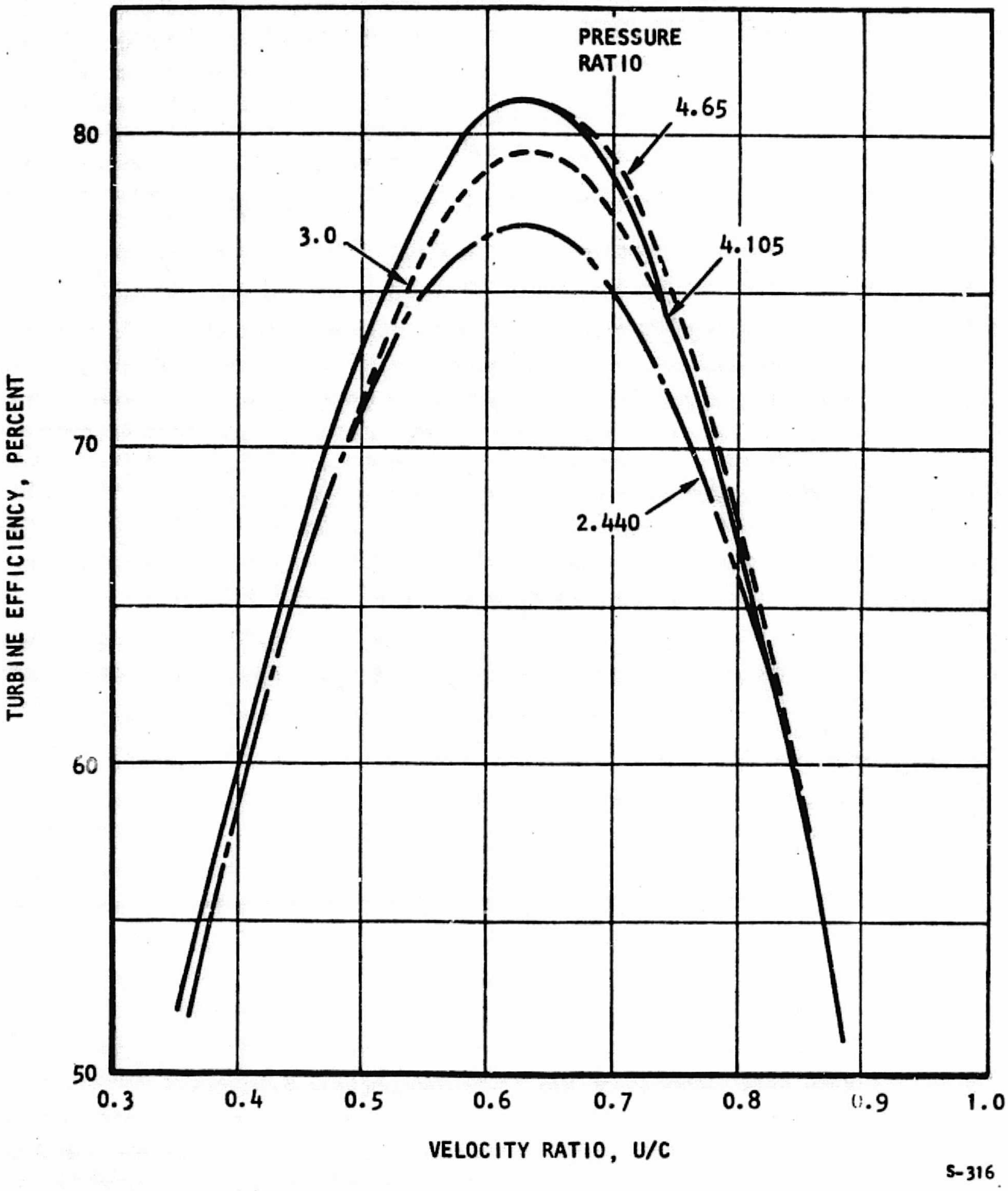


Figure 4-2. Turbine Efficiency of 3-Ton Unit

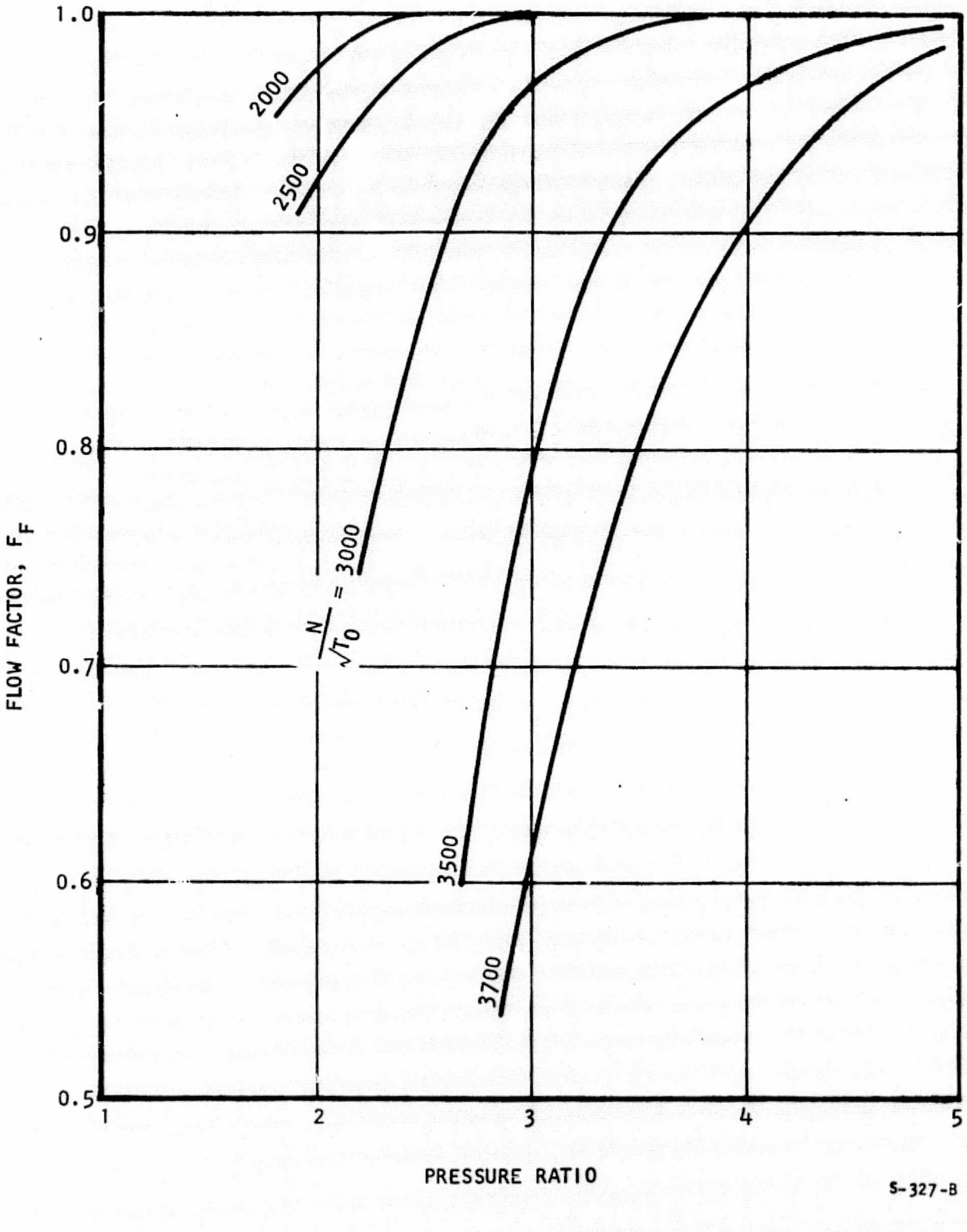
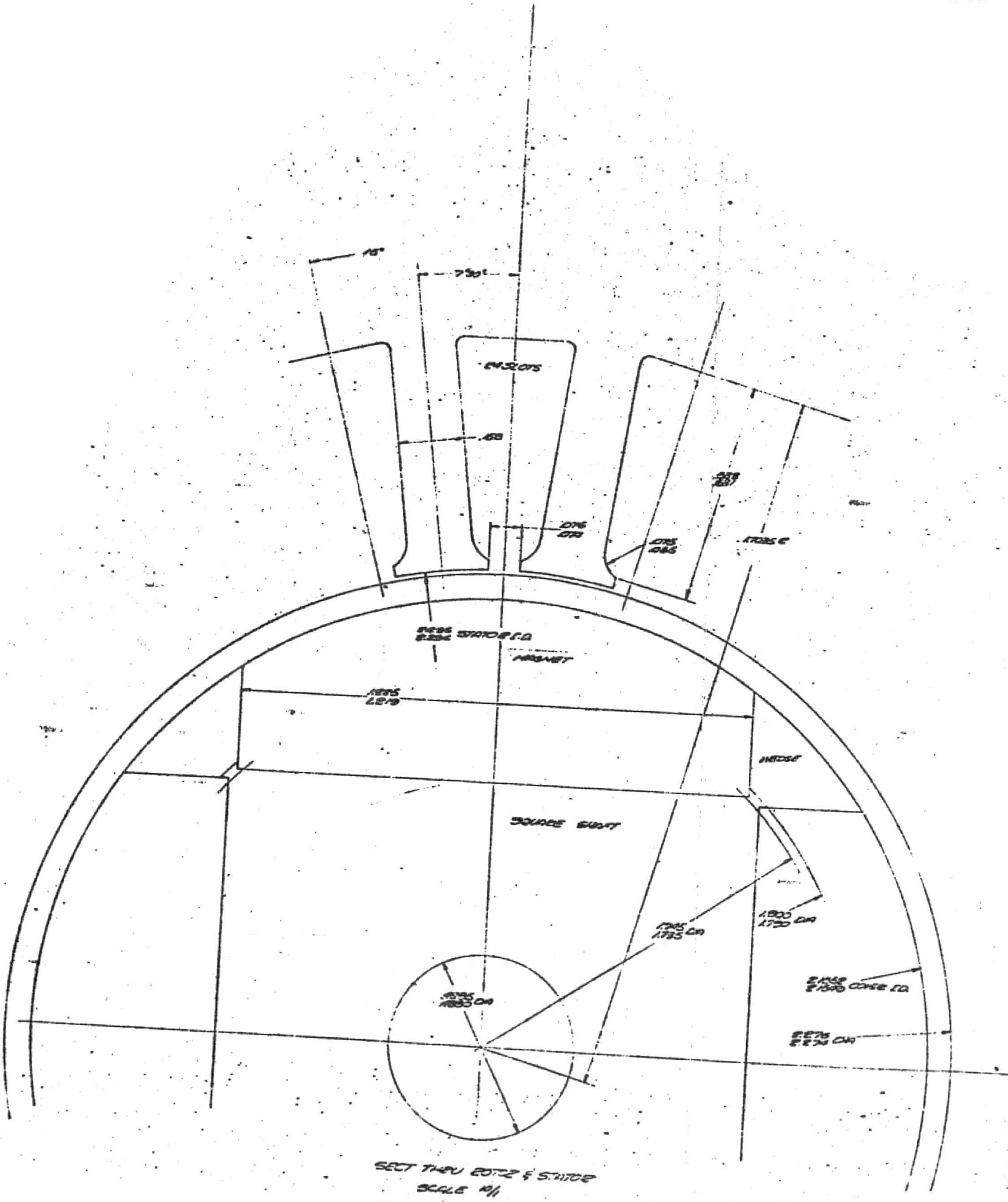


Figure 4-3. Turbine Flow Coefficient (3-Ton Unit)



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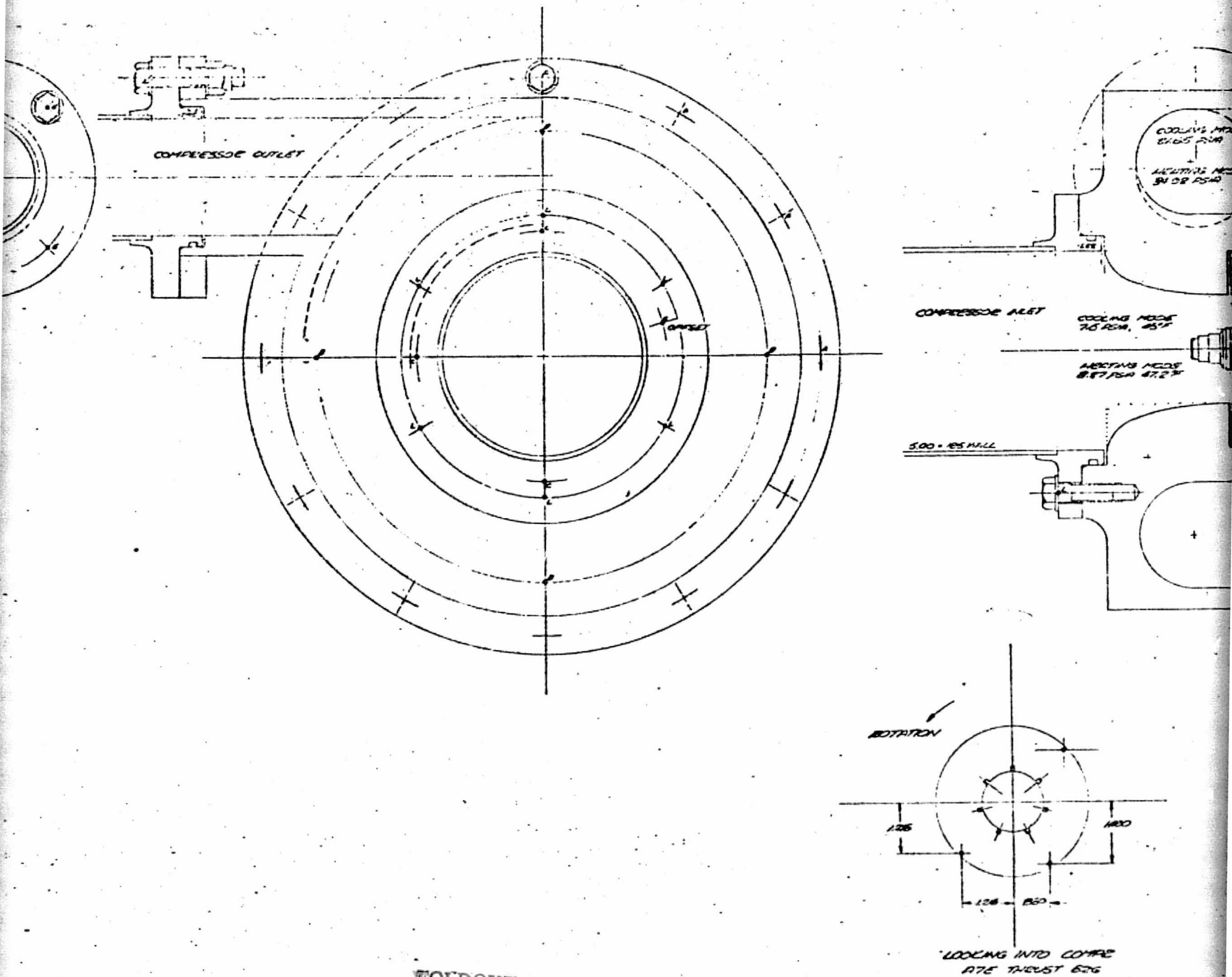

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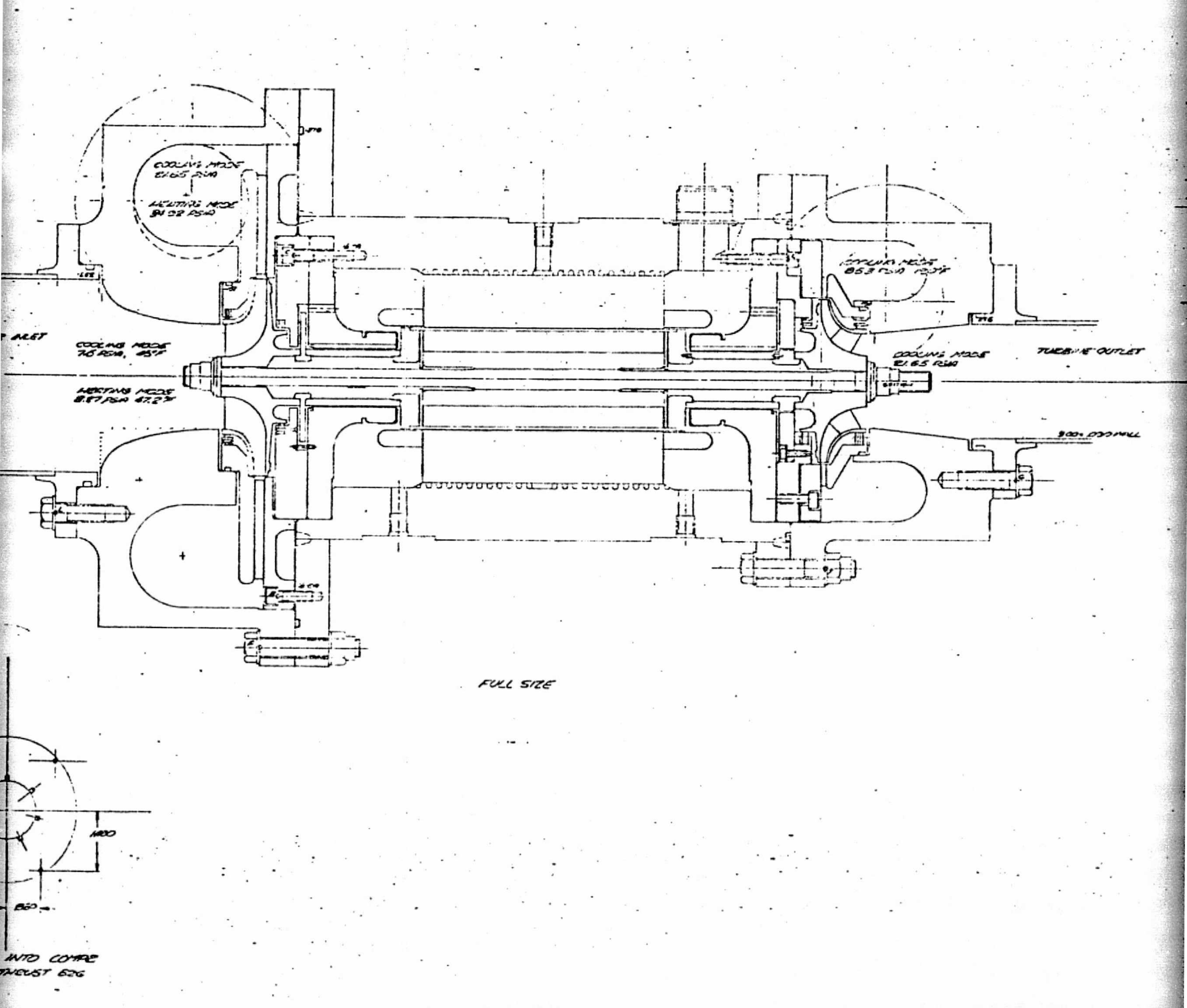

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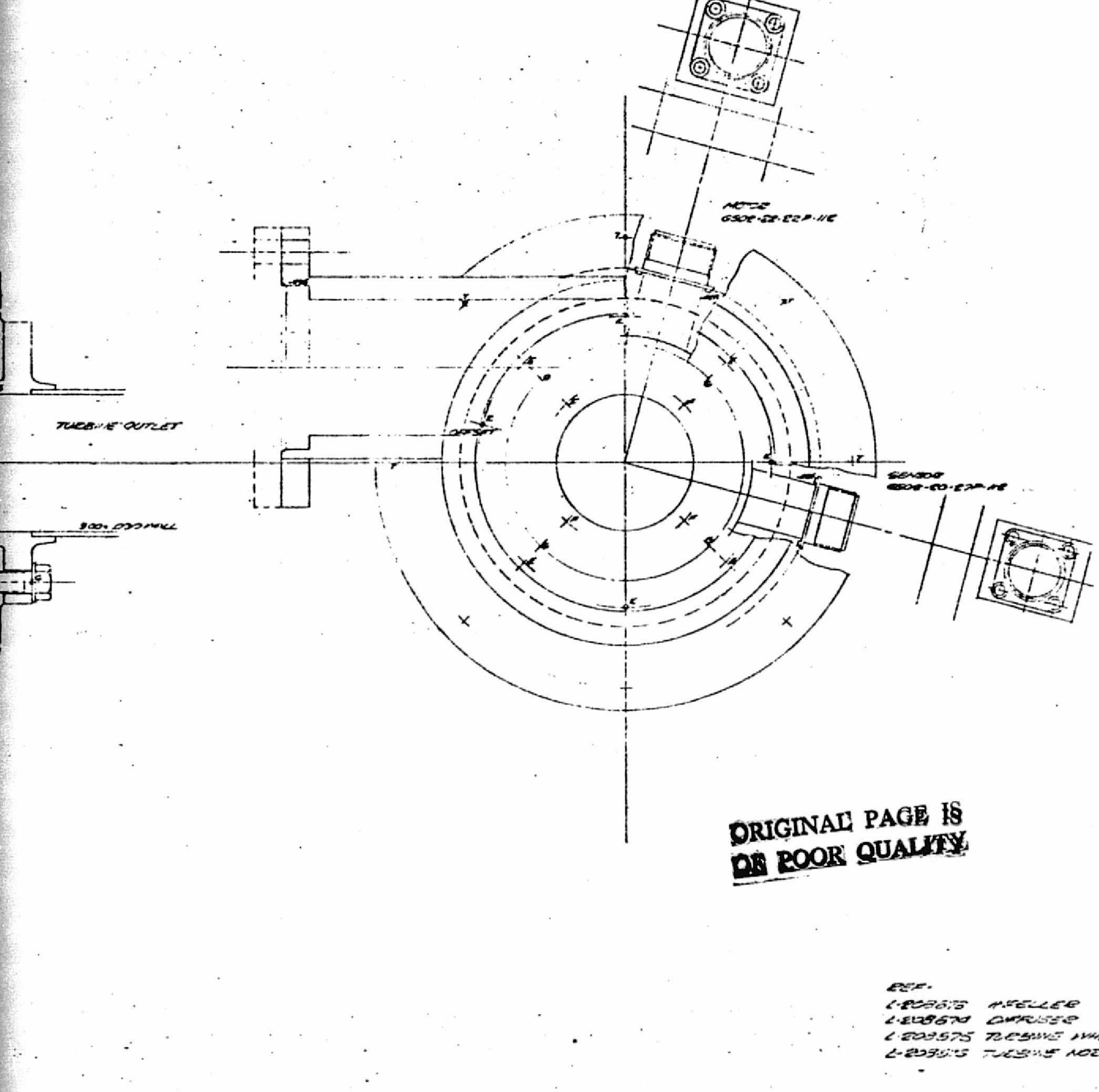
3



REF.
[illegible] IMPELLER
[illegible] DIFFUSER
L-209575 TURBINE WHEEL
[illegible] TURBINE NOZZLE

AIRESEARCH MANUFACTURING COMPANY OF CALIFORNIA	
LAYOUT	
COMPRESSOR 25 TON, 300 KBTUH COMMERCIAL BUILDINGS	
70210	SK71624
SCALE 1/1	SHEET OF

FOLDOUT FRAME 4

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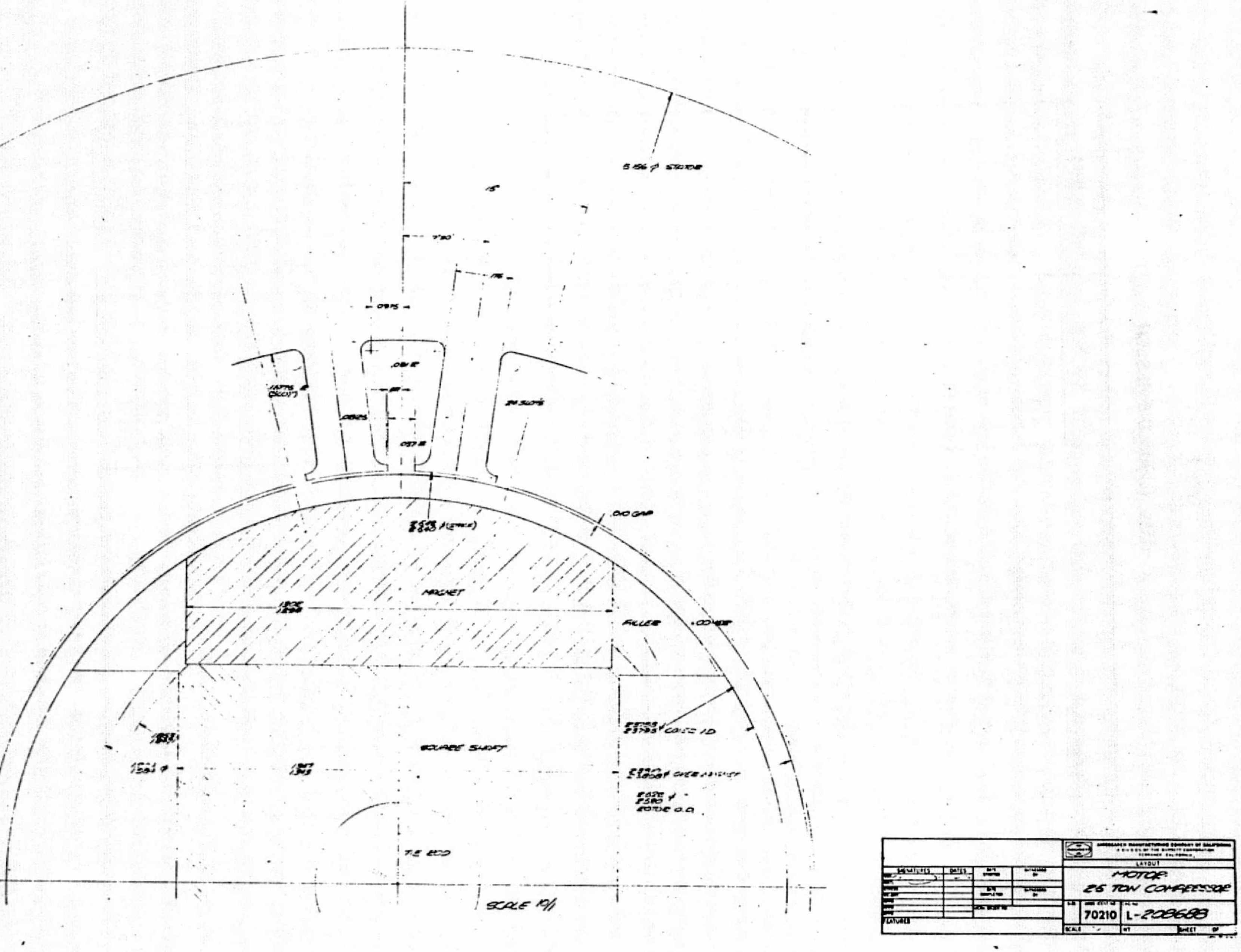


TABLE 4-3

25-TON TURBOCOMPRESSOR

	Cooling Mode	Heating Mode
Compressor		
Fluid type	R-11	R-11
Inlet pressure, psia	7.50	8.27
Inlet temperature, °F	45	47.2
Inlet H, Btu/lb	98.01	98.28
Outlet pressure, psia	21.65	34.92
Outlet H, actual, Btu/lb	107.82	117.01
ΔH adiabatic, Btu/lb	8.02	11.05
ΔH actual, Btu/lb	9.81	18.73
Volume flow, inlet, cfs	5.67	9.03
Weight flow, lb/sec	1.1	1.93
Speed, rpm	30,000(constant)	41,000
Theoretical efficiency, percent	81.8	59
Power at wheel, kw	11.38	38.0
Turbine		
Inlet pressure, psia	85.3	NA
Inlet temperature, °F	190	
Inlet H, Btu/lb	115.4	
Outlet pressure, psia	21.65	
Outlet H, Btu/lb	106.14	
ΔH adiabatic, Btu/lb	10.94	
ΔH actual, Btu/lb	9.26	
Volume flow, inlet, cfs	2.55	
Weight flow, inlet, lb/sec	1.326	
Speed, rpm	30,000(constant)	
Theoretical efficiency, percent	84.7	
Power at wheel, kw	12.08	
Motor		
Maximum speed, rpm	30,000(constant)	41,000
Minimum speed, rpm	NA	18,500
Power at maximum speed, kw	14.5*	38.0**
Power at minimum speed, kw	14.5*	NA
Growth potential, power, percent	NA	20
Growth potential, speed, percent	10	10
Maximum overspeed, rpm	NA	45,100
Pressure, cavity, psia	9.5	9.5
Temperature, cavity, °F	100	150
Viscosity, μ, Reynolds	1.64×10^{-9}	1.75×10^{-9}
Motor, efficiency, percent	NA	92.91
Bearings/lubricant, types	Foil/R-11/vapor	Foil/R-11/vapor
Type motor	PM Samco	PM Samco
No-load power	TBD	NA
Maximum design speed, rpm	NA	50,000
General		
Design life, yr	15	15
No. of starts, lifetime	60,000	60,000
Service hours, lifetime	75,000	50,000
Type control	on/off	Modulating

* Power delivered to the compressor assuming 5 percent mechanical losses.

** Power delivered to the compressor assuming no mechanical losses.

TABLE 4-4

COMPRESSOR MOTOR CHARACTERISTICS, 25-TON/600 KBTUH HEAT PUMP

Parameter	Units
Rpm	41,000
Kw (shaft)	44.68
Kva (input)	54.55
Pf (lagging)	0.819
Poles, no.	4
Phases	3
Frequency, Hz	1367
Tip speed, ft/sec	474.4
Stator current density, amp/sq in.	12,000
Slots, no.	24
Pole embrace	0.728
Stator material	0.007-in. Trancor-T
Magnet material	Samarium cobalt
Dimensions, in.	
Rotor dia	2.501
Rotor length	5.951
Stack ID	2.521
Stack OD	5.156
Stack length	5.951
End turn extnesion	1.075
Total stator length	8.101
Slot height	0.41
Tooth width	0.178
Slot opening	0.082
Weight, lb	
Stator	21.97
Rotor	7.47
Total weight	29.44
Losses, w	
Windage	543
Copper	775
Stray and pole head	447
Stator teeth	473
Stator core	653
Efficiency	0.947
No-load voltage, per unit	1.20
Impedance data, per unit	
Synchronous reactances	0.29
Commutation reactance	0.29
Stator resistance (hot)	0.0636
Stator leakage reactance (XL)	0.153

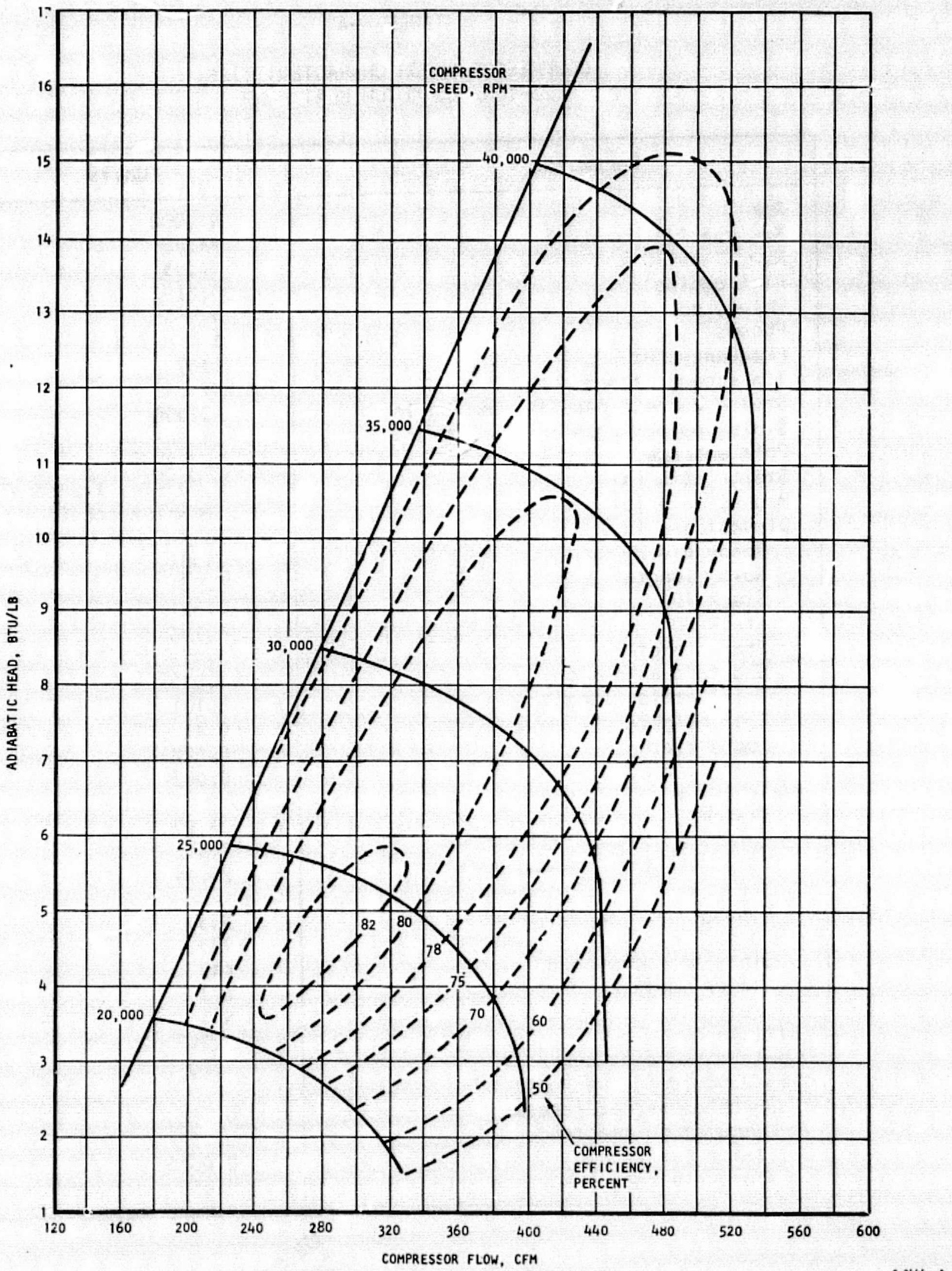


Figure 4-4. Compressor Map (25-Ton/800,000-Btu/hr Unit)

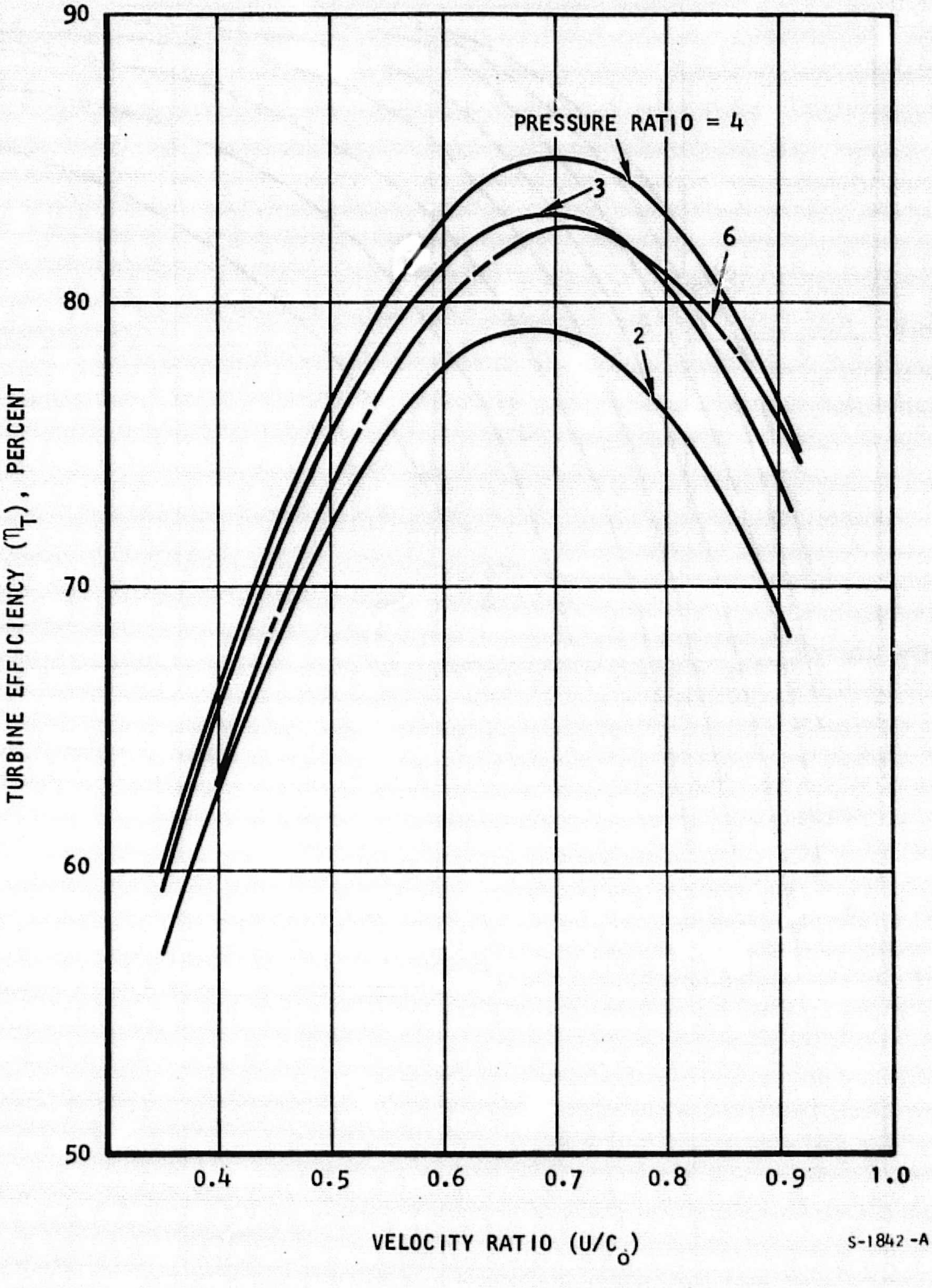


Figure 4-5. Turbine Efficiencies (25-Ton Unit)

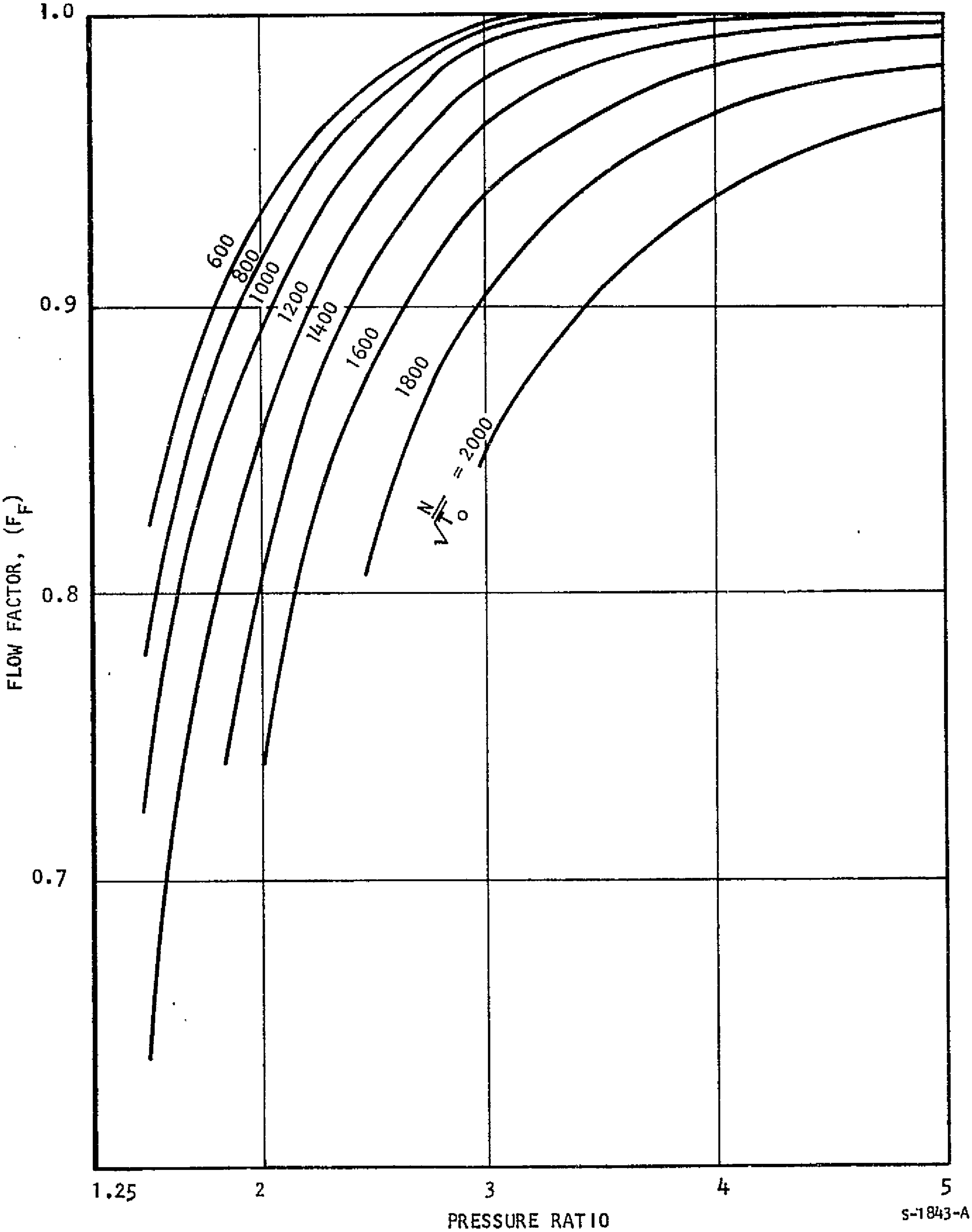


Figure 4-6. Turbine Flow Coefficients (25-Ton Unit)

4.3.4 Design Status

Detail drawings are approximately 50 percent complete; release of all drawings is scheduled for the first week in December.

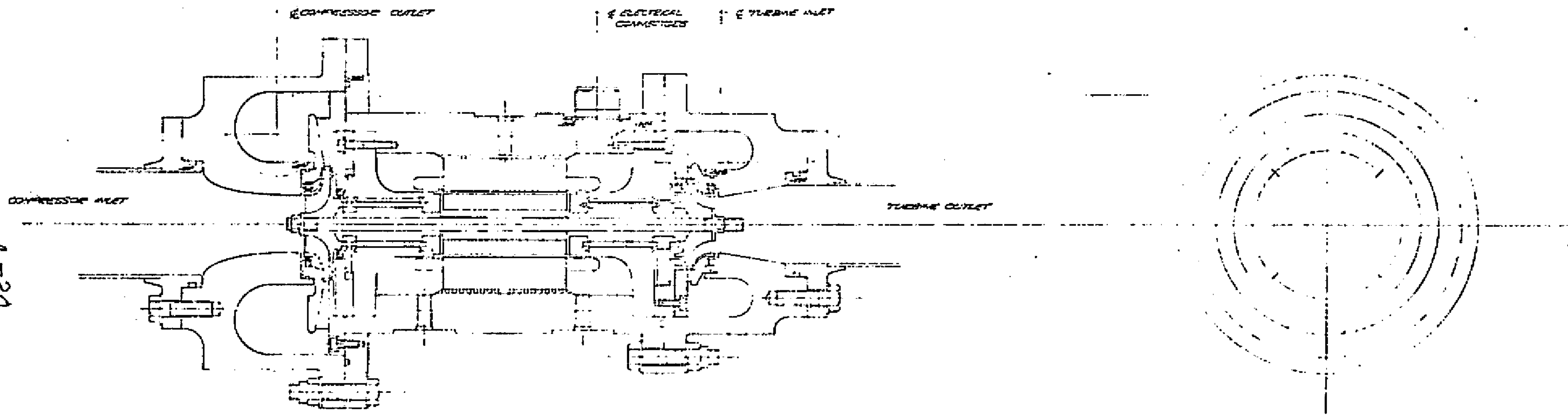
4.4 COMMERCIAL UNIT (10-TON/200 KBTUH)

The layout of this machine is shown in Drawing SK71623. The detailed analyses of this unit are based on the problem statement shown in Table 4-5.

The detailed stress, bearing, and thermal analyses have not been completed, but no problems are anticipated.

The performance of the motor is shown in Table 4-6, and the aerodynamic performance of the compressor and turbine is shown in Figures 4-7 through 4-9.

Detailed design analyses have been initiated. Scheduled completion of all detailed drawings is late December.



SK71623-1 10 TON TURBOCOMPRESSOR
47500 RPM COOLING MODE
57,440 RPM HEATING MODE

70210 SK 71623

TABLE 4-5

1?-TON TURBOCOMPRESSOR

	Cooling Mode	Heating Mode
Compressor		
Fluid type	R-11	R-11
Inlet pressure, psia	8.04	8.74
Inlet temperature, °F	50.80	58.0
Inlet H, Btu/lb	98.57	99.7
Outlet pressure, psia	22.40	31.8
Outlet H, actual, Btu/lb	108.57	114.00
ΔH adiabatic, Btu/lb	7.8	10.20
ΔH actual, Btu/lb	10.0	15.0
Volume flow, inlet, cfs	2.19	2.93
Weight flow, lb/sec	0.453	0.657
Speed, rpm	45,430(constant)	57,440
Theoretical efficiency, percent	78	68
Power at wheel, kw	4.76	10.4
Turbine		
Inlet pressure, psia	85.6	NA
Inlet temperature, °F	190.0	
Inlet H, Btu/lb	115.3	
Outlet pressure, psia	22.0	
Outlet H, Btu/lb	104.7	
ΔH adiabatic, Btu/lb	10.7	
ΔH actual, Btu/lb	8.56	
Volume flow, inlet, cfs	3.1	
Weight flow, inlet, lb/sec	0.595	
Speed, rpm	45,430	
Theoretical efficiency, percent	0.80	
Power at wheel, kw	4.90	
Motor		
Maximum speed, rpm	47,500(constant)	37,440
Minimum speed, rpm	NA	27,500
Power at maximum speed, kw	5.5*	10.4**
Power at minimum speed, kw	5.5*	NA
Growth potential, power, percent	NA	20
Growth potential, speed, percent	10	10
Maximum overspeed, rpm	NA	63,140
Pressure, cavity, psia	9.6	9.6
Viscosity, μ, Reynolds	1.64×10^{-9}	1.75×10^{-9}
Temperature, cavity, °F	100	150
Motor efficiency, percent	NA	90
Bearings/lubricant, types	Foil/R-11/vapor	Foil/R-11/vapor
Cooling flow, lb/sec (approx)	0.0022	0.0060
Type motor	PM Samco	PM Samco
No-load power	TBD	NA
Maximum design speed, rpm	NA	69,500
General		
Design life, yr	20	20
No. of starts, lifetime	60,000	60,000
Service hours, lifetime	75,000	50,000
Type control	on/off	Modulating

* Power delivered to the compressor assuming 5 percent mechanical losses.

** Power delivered to the compressor assuming no mechanical losses.

TABLE 4-6

COMPRESSOR MOTOR CHARACTERISTICS,
10-TON/200 KBTUH HEAT PUMP

Parameter	Units
Rpm	57,440
Kw (shaft)	15.2
Kva (input)	19.3
Pf (lagging)	0.786
Poles, no.	4
Phases	3
Frequency, Hz	1915
Tip speed, ft/sec	487
Stator current density, amp/sq in.	12,000
Slots, no.	24
Pole embrace	0.62
Stator material	0.007-in. Trancor-T
Magnet material	Samarium cobalt
Dimensions, in.	
Rotor diameter	1.864
Rotor length	3.488
Stack ID	1.884
Stack OD	3.729
Stack length	3.488
End turn extension	0.85
Total stator length	5.188
Slot height	0.361
Tooth width	0.1295
Slot opening	0.062
Weight, lb	
Stator	6.48
Rotor	2.45
Total weight	8.93
Losses, w	
Windage	361
Copper	355
Stray and pole head	152
Stator teeth	301
Stator core	295
Efficiency	0.928
No-load voltage, per unit	1.2
Impedance data, per unit	
Synchronous reactances	0.27
Commutation reactance	0.27
Stator resistance (hot)	0.018
Stator leakage reactance (XL)	0.173

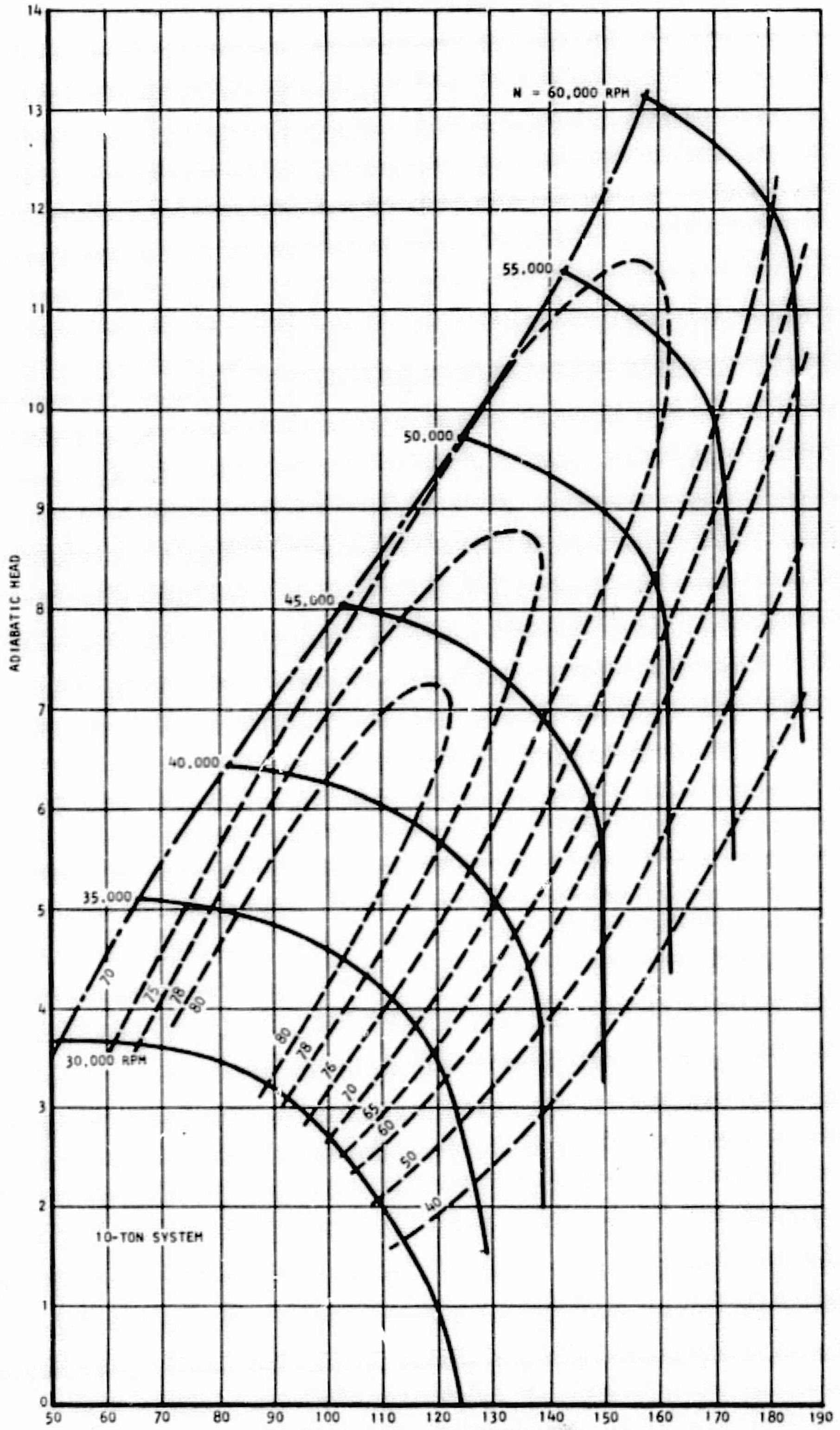


Figure 4-7. Compressor Map (10-Ton Unit)

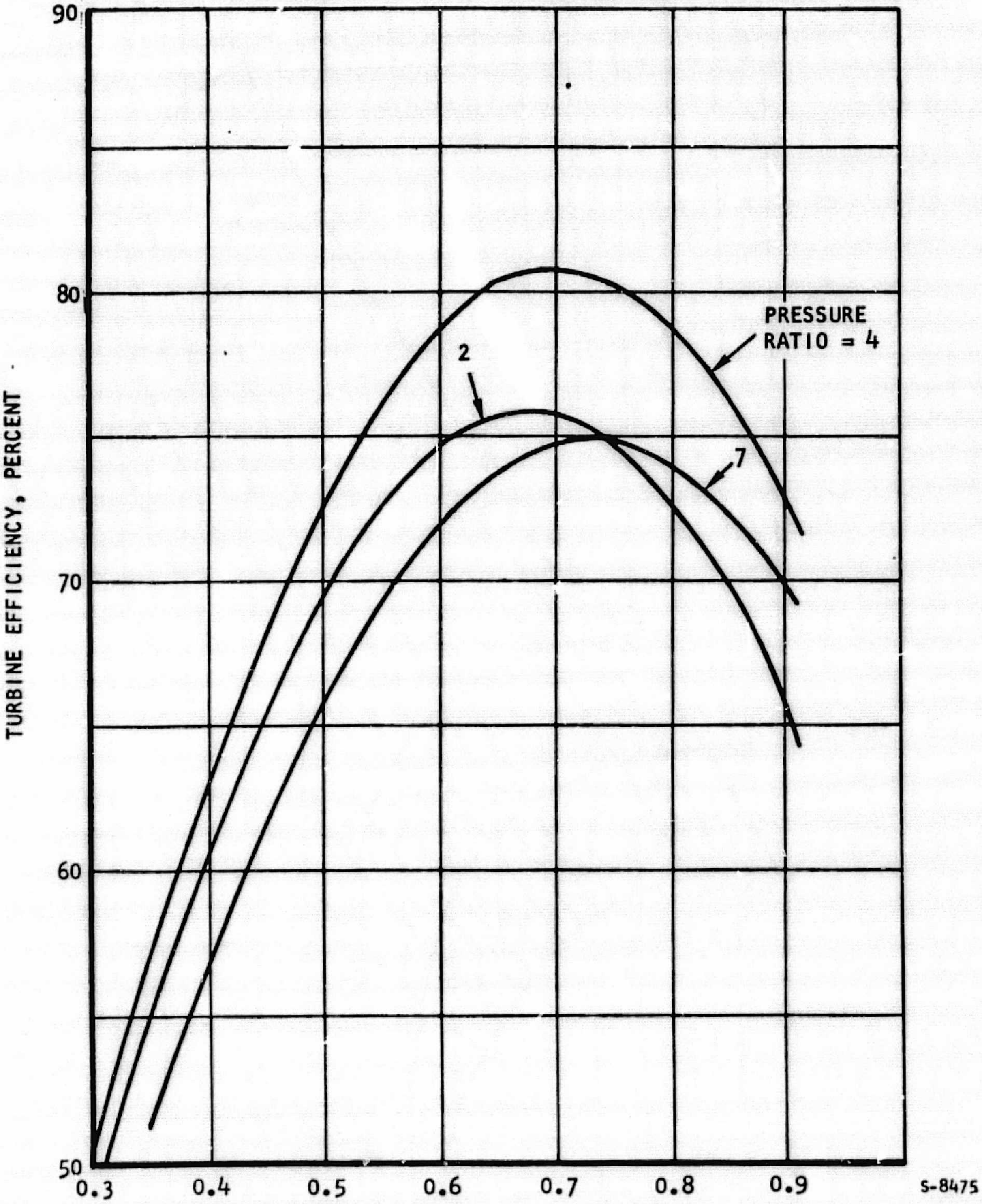


Figure 4-8. Turbine Efficiencies (10-Ton Unit)

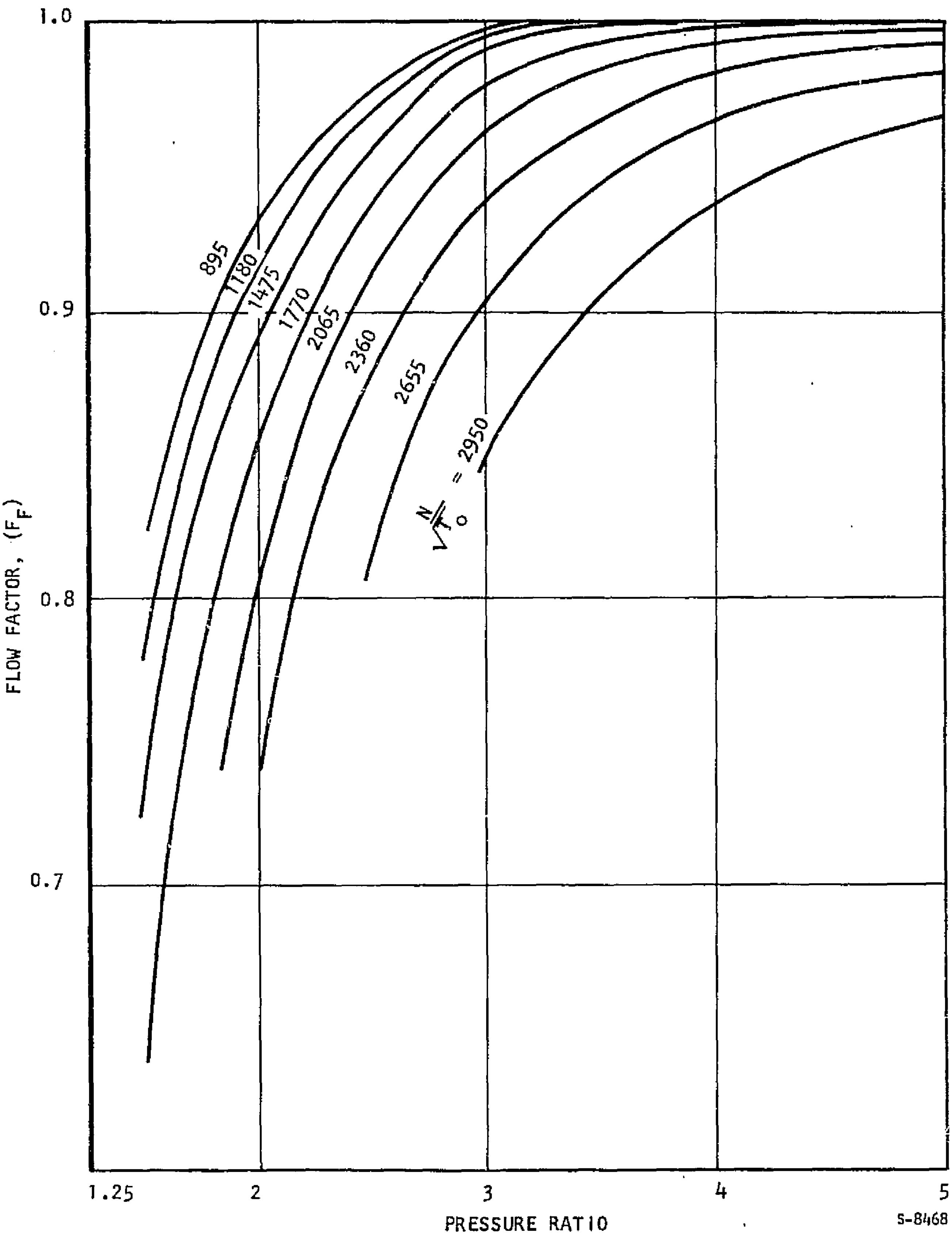


Figure 4-9. Turbine Flow Coefficients (10-Ton Unit)

5. MOTOR CONTROL

5.1 GENERAL

Design activities involved selection of an approach for (1) control mechanization of the three sizes of machines and (2) circuit design for the 3-ton/60 KBTUH and 25-ton/200 KBTUH machines.

5.2 CONCEPT SELECTION

Two system concepts have been studied in detail: (1) SCR system, and (2) transistor system.

The development of the two motor control techniques, SCR for high power and transistors for low power, is now in the breadboard stage. Conceptual designs for both motor controls are complete, and the detail circuit design is under way. As the motor designs evolve, trades between SCR and transistor drive techniques become more clear. Studies of the optimum number of poles for the motor required consideration of both the inverter switching frequency and the mechanical stresses in the machine rotor. A trade of SCR vs transistors for motor controls is given in Table 5-1.

5.3 25-TON MOTOR CONTROL

The 25-ton machine is controlled by a line commutated inverter driven by a 3-phase phase delay rectifier (PDR) (Figure 5-1).

The PDR rectifies the 3-phase power from the utility and delivers the resulting dc current to the dc link. By adjusting the firing angle of the silicon controlled rectifiers (SCR's), it is possible to vary the current in the dc link from zero to maximum current.

TABLE 5-1

TRANSISTOR INVERTER VS LINE COMMUTATED SCR INVERTER

Item	Transistor Inverter	SCR Inverter	Comments
Junction temperature	200°C	125°C	Could affect size of heat sink.
Switching losses and turnoff time	Low--approximately 1 μsec	High--15 to 30 μsec	The SCR long turnoff time also contributes significantly to the motor control power factor.
Power handling capability	Low-30 kw range	High	Transistors may be paralleled for extra current capability.
Voltage	450 v	1300 v	Transistors can be cascaded to achieve more voltage capability.
Current gain	Low	High	Transistor inverter requires continuous base drive.
Power factor	Close to unity	Less than unity	Machine power factor depends on synch reactance + device switching time (T_Q) - effect of T_Q increases with speed.
Compensation for $\frac{di}{dt}$ and $\frac{dv}{dt}$	Yes	Yes	Snubbers and chokes required for both; this consumes power.

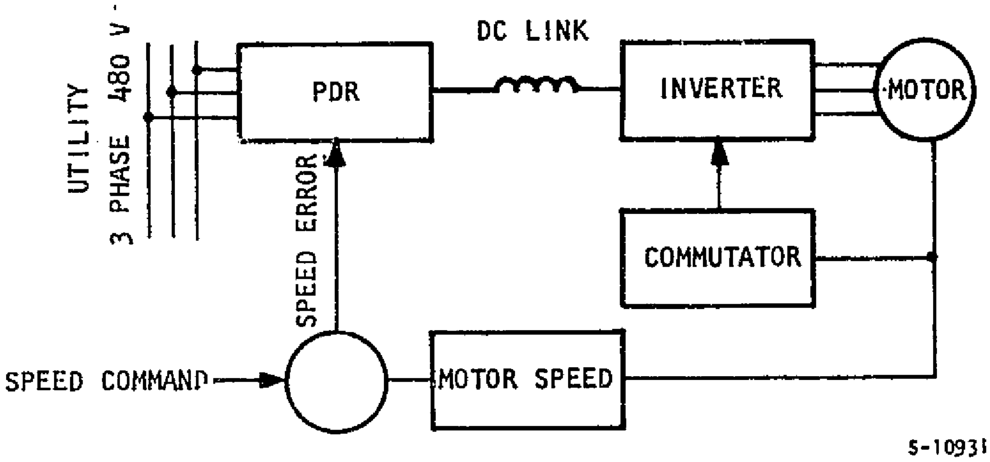


Figure 5-1. SCR Control Block Diagram

The dc link choke is designed to provide a current source for the line commutated inverter. The inverter provides the necessary current to the motor windings and is synchronized by the electronic commutator. A motor speed loop is closed around the control, and a change in required speed results in a change in the PDR firing angle; this in turn changes the dc link current and hence the motor speed.

5.3.1 Efficiency

The losses in the system are contributed by the following.

PDR--SCR forward voltage; SCR switching losses; SCR snubbers

Dc link--Choke I^2R losses

Inverter--SCR forward voltage drop; switching losses; SCR snubber losses

The total losses are expected to be on the order of 5 percent of 60 kw; i.e., 3 kw at 60 kw_e.

5.3.2 Thermal Design

The unit will be designed to reject 3 kw. A trade of liquid cooling vs forced air vs convection cooling has been performed, and liquid cooling was rejected due to its cost and complexity, especially since it also required deionized water for cooling. The forced-air cooling was rejected due to the increased cost of fans and from a reliability standpoint.

5.3.3 Size

The size of the unit is governed by the amount of heat sink required and by the dc link choke. Preliminary sizing shows that the package will be approximately 3.5 ft x 1.5 ft x 1.5 ft. About half of this volume will be required by the PDR, inverter SCR's, and heat sinks.

The choke has been designed to handle 10 percent ripple current at full load; the development unit weighs approximately 300 lb. A study is under way to determine the effects on the design if this choke is reduced in size.

5.3.4 SCR Control Circuit Description (See Figures 5-2 and 5-3)

The current supplied by the PDR is sensed by a Hall device and conditioned. The conditioned current signal is compared to a current waveform, and a current error signal is generated. The error signal is compared to a cosine ramp waveform, and the output of this comparator is used to turn on solid-state switches that drive the SCR gates.

5.3.4.1 Ramp Generators

The dc output voltage of a PDR is proportional to the cosine of the firing angle. Thus, if a cosine ramp is used in the generation of the firing angle, the output voltage becomes a linear function of the firing angle.

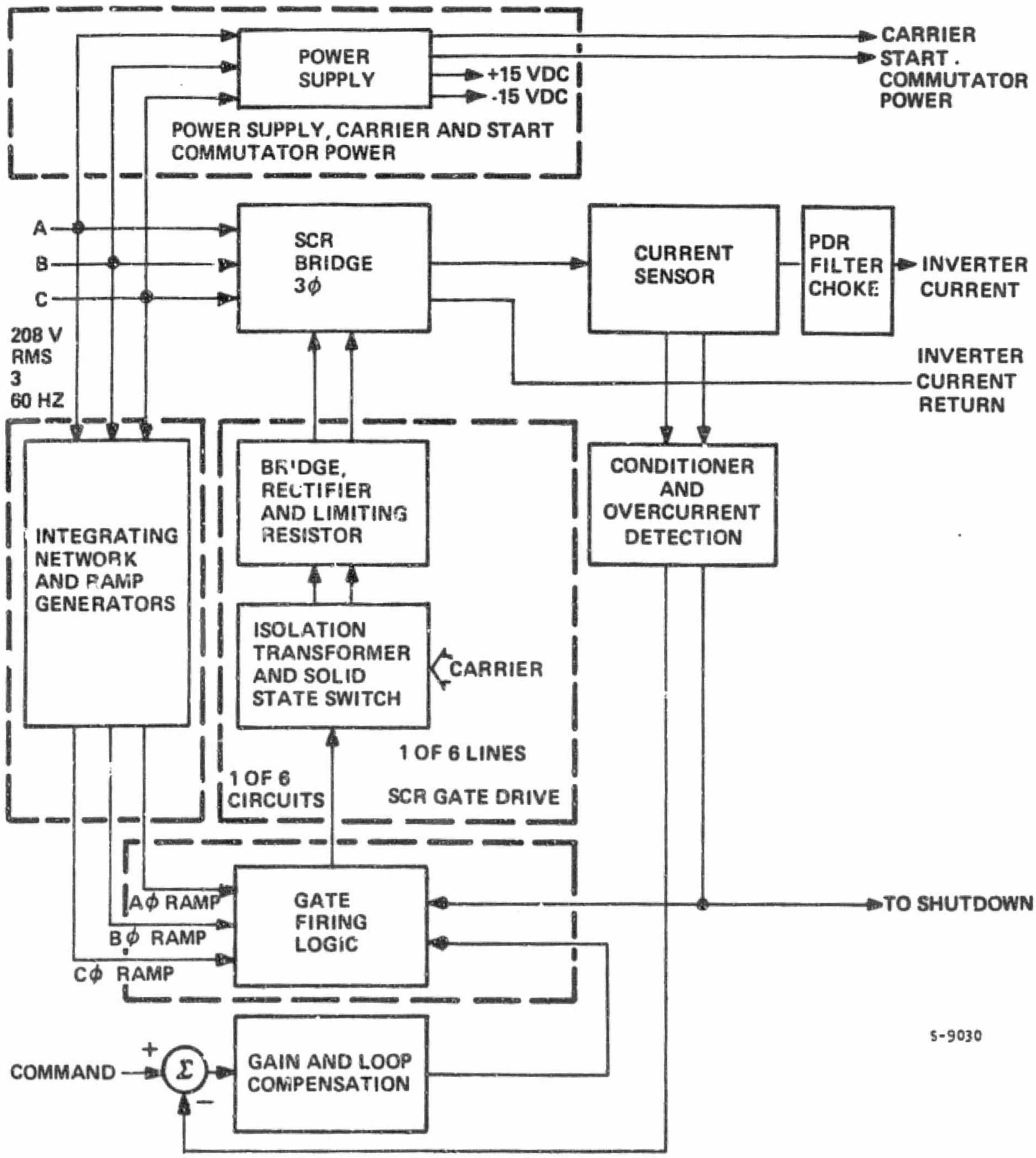


Figure 5-2. Phase Delay Rectifier

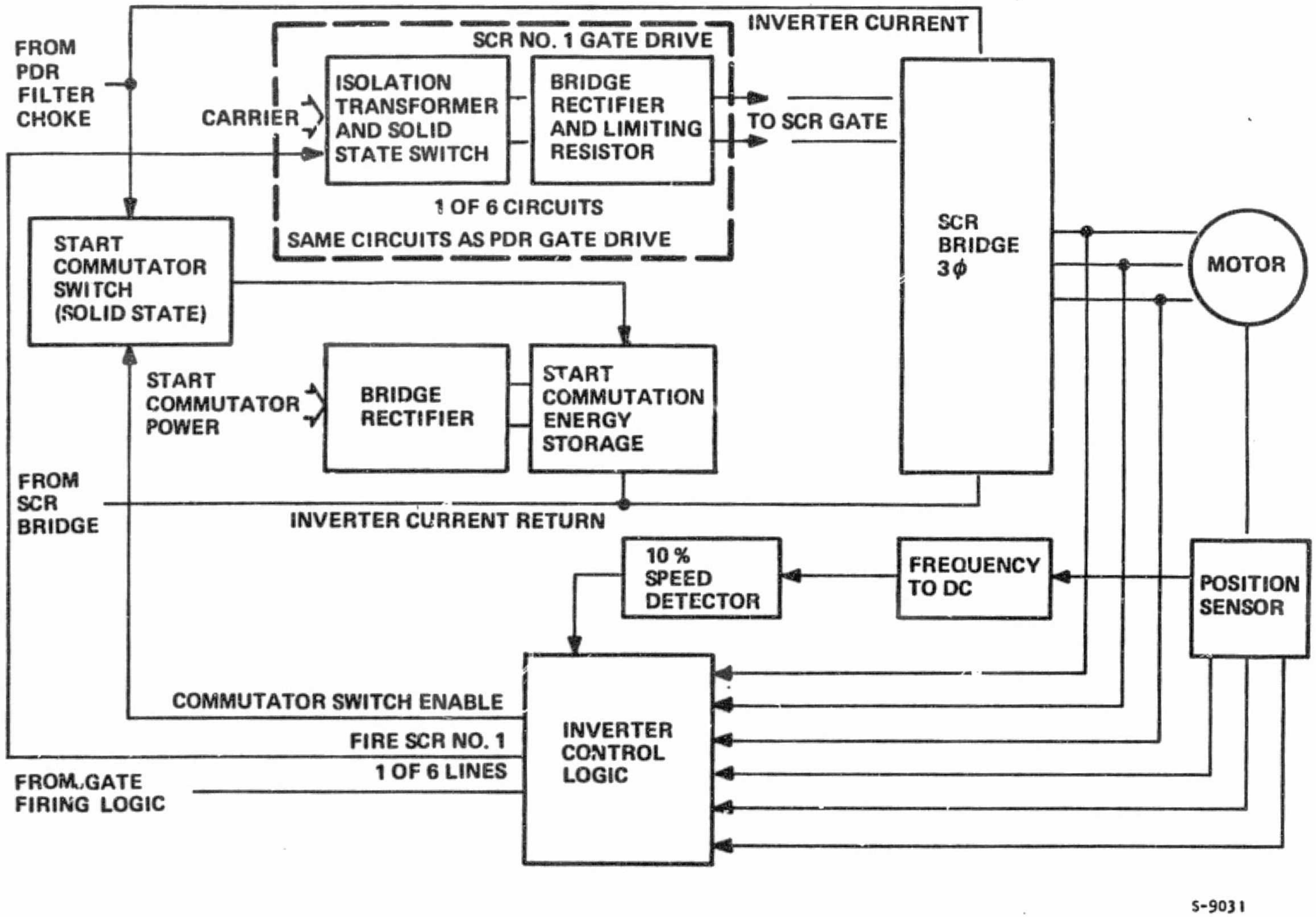


Figure 5-3. Inverter

For a 3-phase PDR this cosine ramp is a cosine waveform whose maximum occurs at the intersection of the positive half-cycles of two of the phases. The ramp can then be used to generate the firing pulse for the + SCR connected to the more positive of the two phases.

5.3.4.2 Start Commutator

Below 10 percent speed, the back EMF is not adequate for SCR commutation. An auxiliary commutation scheme is therefore needed to provide the required voltage for commutation.

This voltage is generated by charging a large capacitor, which is isolated by a solid-state switch from the inverter supply current. When commutation is required, the solid-state switch is turned on and the capacitor, which has previously been charged to a polarity opposite that required to drive the motor, is connected across the motor supply. The charge on this capacitor is adequate to enable it to absorb the PDR output current and reverse the polarity of the inverter supply, thereby causing SCR commutation.

5.3.4.3 Power Supply and Carrier

The power supply employs conventional series regulator circuitry. The input ac voltage is transformed to ±17 v min. The resulting dc voltage is series regulated down to ±14 vdc. This voltage is used to provide power for the control circuitry. The carrier generator is an ac square wave inverter circuit, which is used to translate the SCR gate drive from system ground to the SCR cathode reference. Transformers are used to isolate the SCR gates from each other. Thus, the same carrier is used for firing all SCR's (PDR and inverter).

5.3.4.4 Gate Firing Logic

Complementary metal oxide semiconductor (CMOS) integrated circuits have been employed in the gate firing logic circuitry because of the low cost, low power dissipation, and high noise immunity. The CMOS circuitry will be supplied with 7.5 v series regulated from the +14 v supply.

5.3.4.5 Circuit Design

Figures 5-4 through 5-13 give details of completed circuit designs for the SCR control (600 KBTUH heat pump compressor motor). The following circuits are shown:

(a) Power supply and carrier generator (Figure 5-4)

(b) Three-phase PDR ramp generator (Figure 5-5)

(c) Inverter integrating network (Figure 5-6)

(d) PDR/inverter gate logic (Figure 5-7)

(e) SCR gate drive (Figure 5-8)

(f) Back EMF conditioner (Figure 5-9)

(g) Inverter bridge (Figure 5-10)

(h) Auxiliary commutator (Figure 5-11)

(i) Three-phase PDR (Figure 5-12)

(j) Inverter ramp generator (Figure 5-13)

5.4 3-TON MOTOR CONTROL

5.4.1 General

The transistor system consists of a rectifier, which produces dc, followed by a transistor inverter. The inverter provides commutation for the motor and also controls the current by pulse width modulating one of the pair of inverter switches that is on at any one time (see block diagram of Figure 5-14).

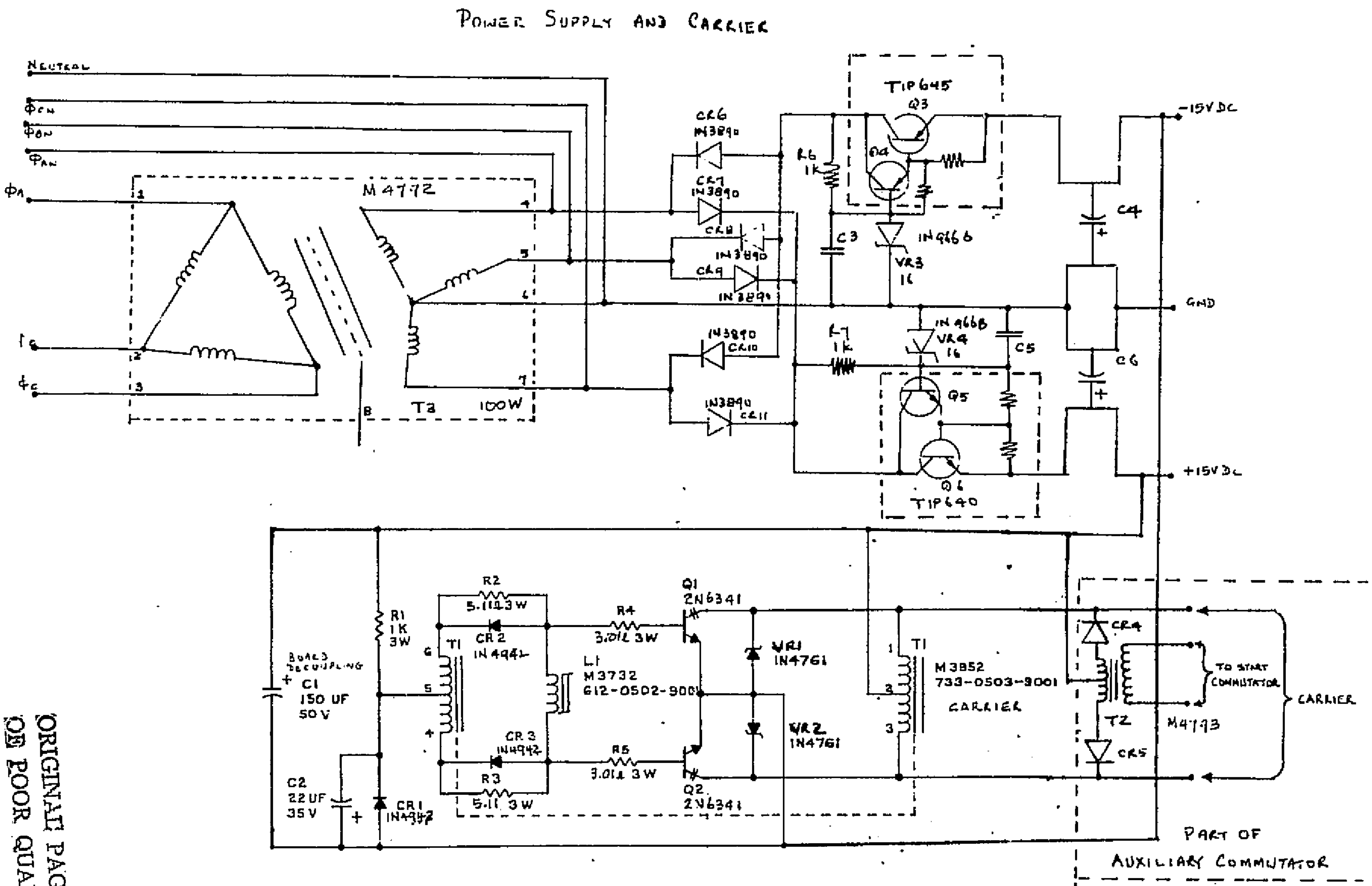


Figure 5-4. Power Supply and Carrier Circuit Design

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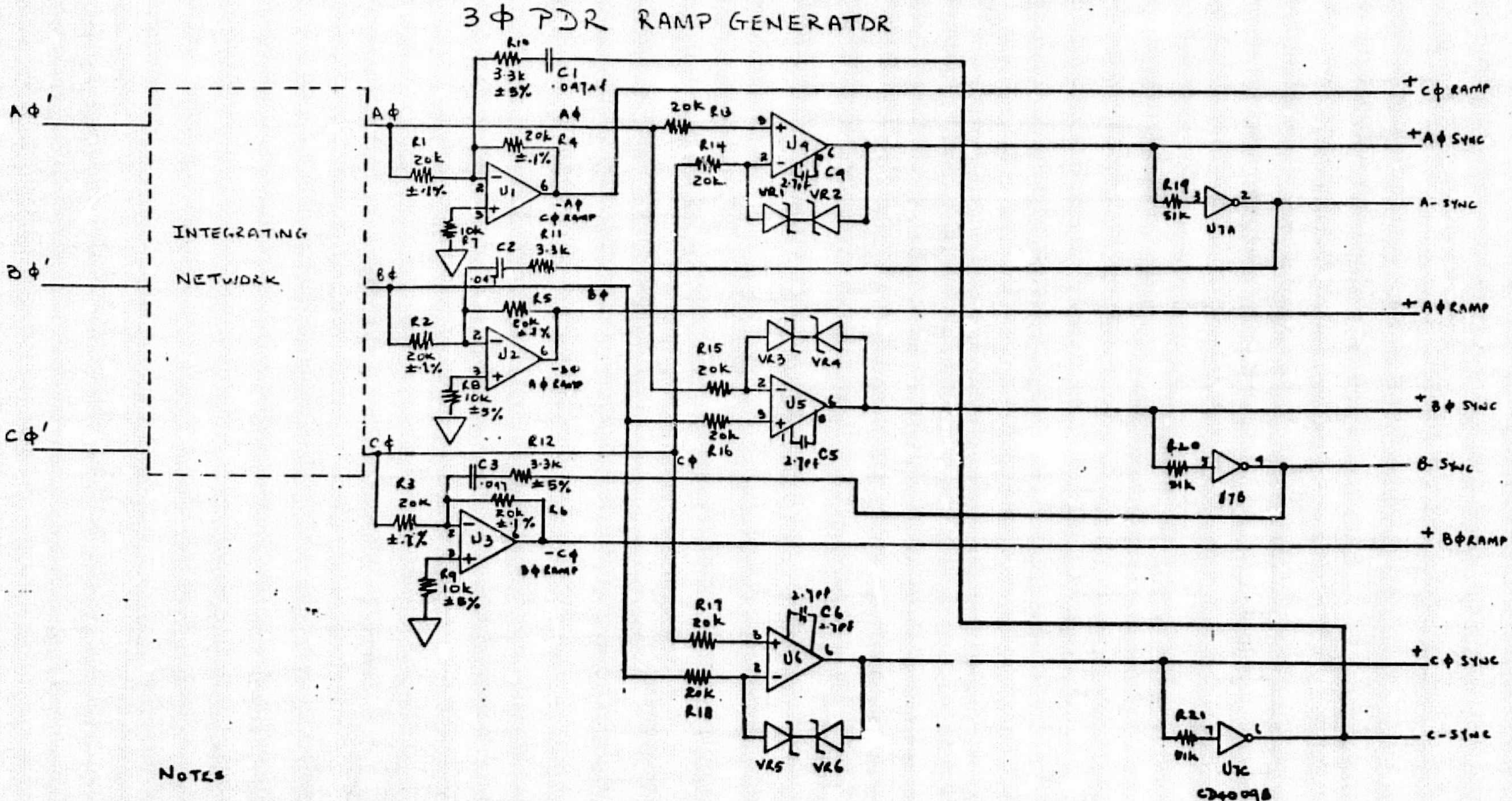


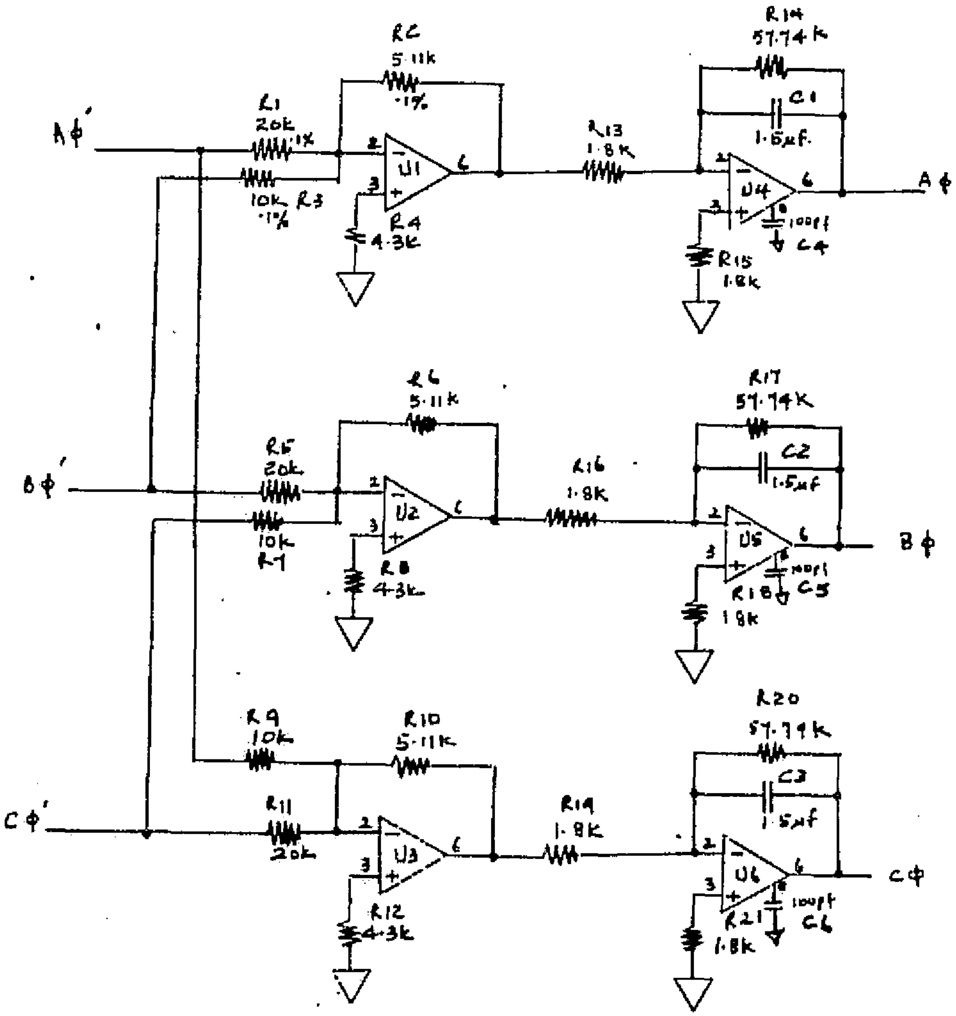
NOTES

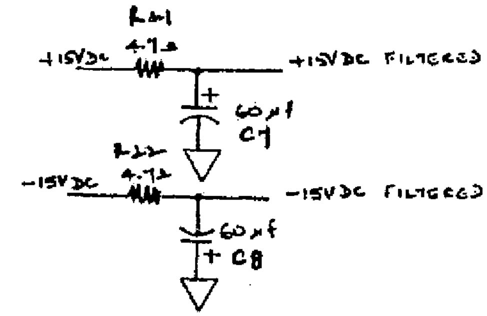
1 ALL ZENER DIODES ARE 1N756
2 U1-U3 ARE 431-005-9301 741
3 U4-U6 ARE 431-049-9301 748
4 U7 IS CD4009B
5 RESISTORS ARE IN OHMS ±5%, 1/4 W
6 +15V DC TO PIN 7 OF ALL AMPLIFIERS
7 -15V DC TO PIN 4 OF ALL AMPLIFIERS
8 3PF BETWEEN PINS 1 AND 8 OF U4-U6

Figure 5-5. Three-Phase PDR Ramp Generator Circuit Design

C-2

INVERTER INTEGRATING NETWORK





NOTES

1. RESISTORS ARE ±.1%, ·1W
2. U1-U3 431-005-9301 741
3. U4-U6 431-033-9302 LM108

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Figure 5-6. Inverter Integrating Network Circuit Design

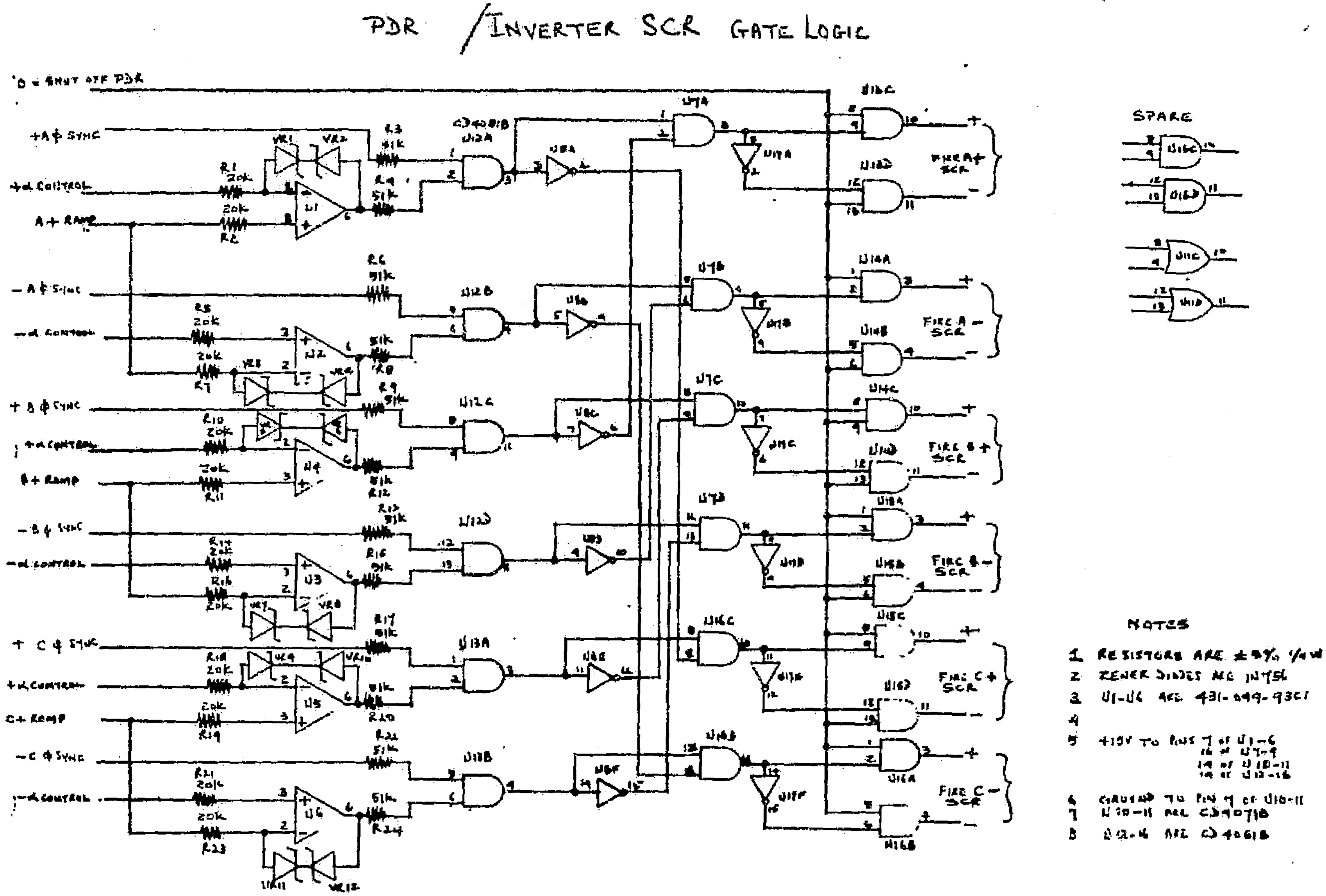

Figure 5-7. PDR/Inverter SCR Gate Logic Circuit Design

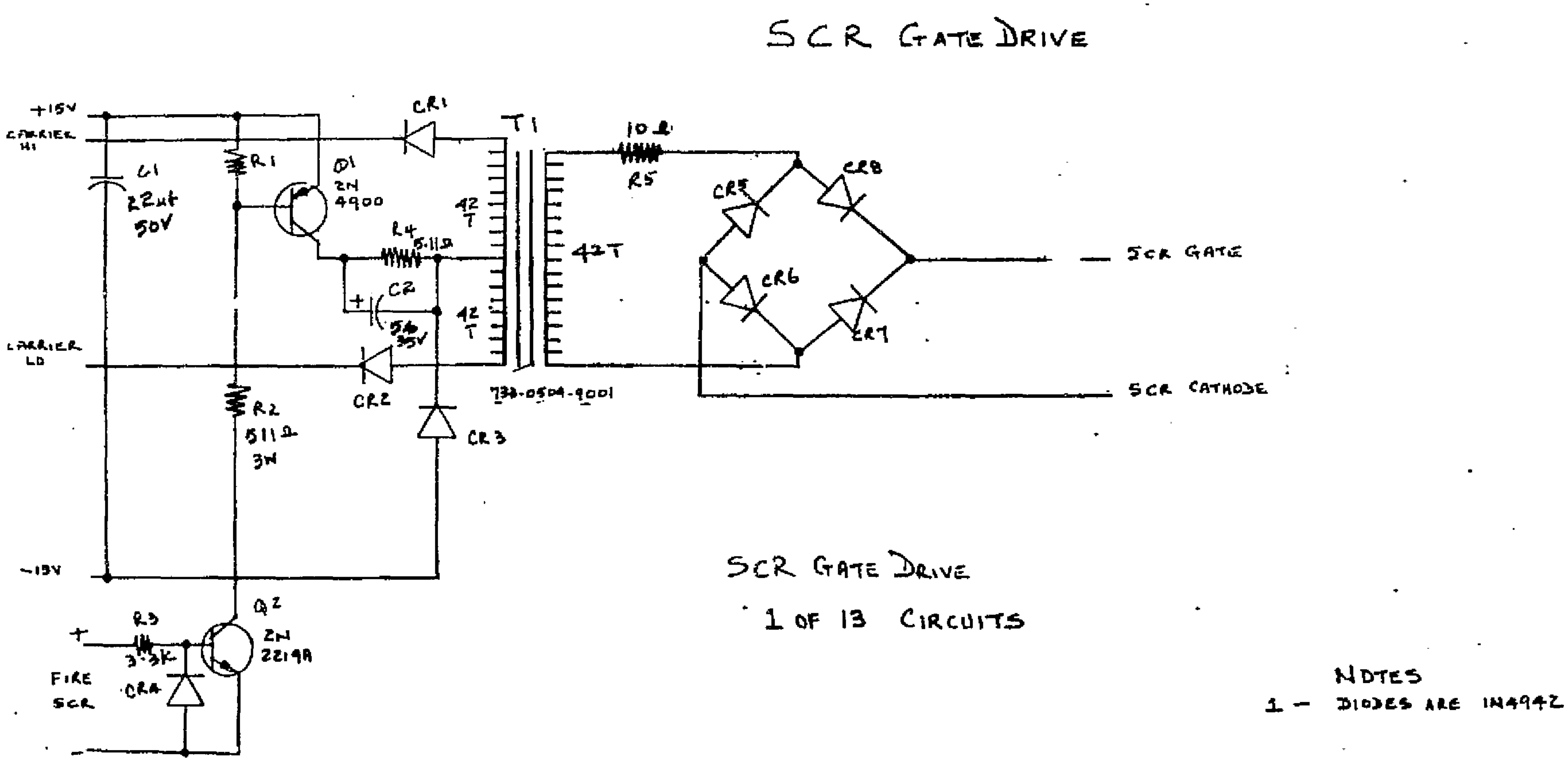

Figure 5-8. SCR Gate Drive Circuit Design

BACK EMF CONDITIONER

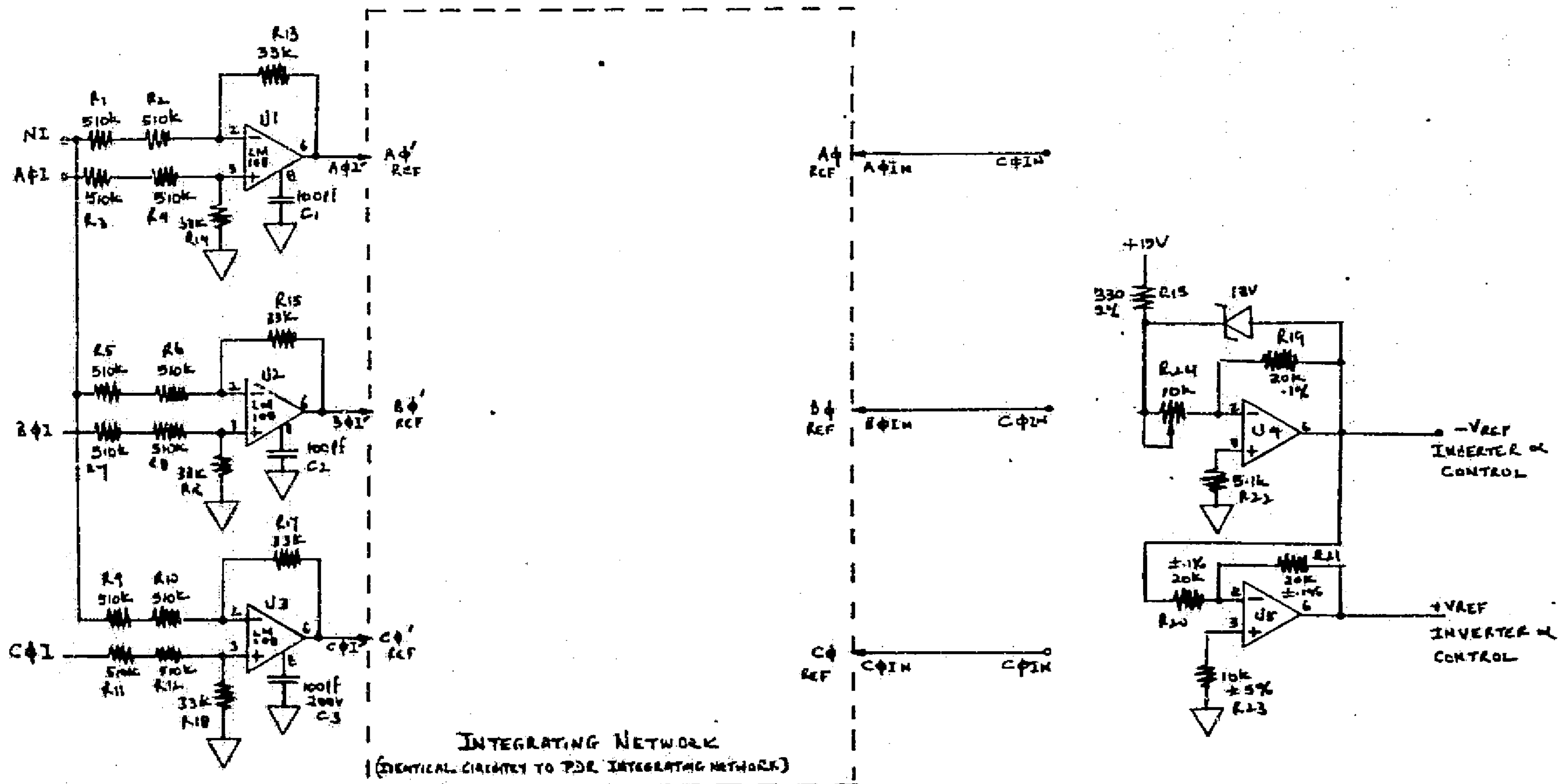

Figure 5-9. Back EMF Conditioner Circuit Design

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INVERTER BRIDGE

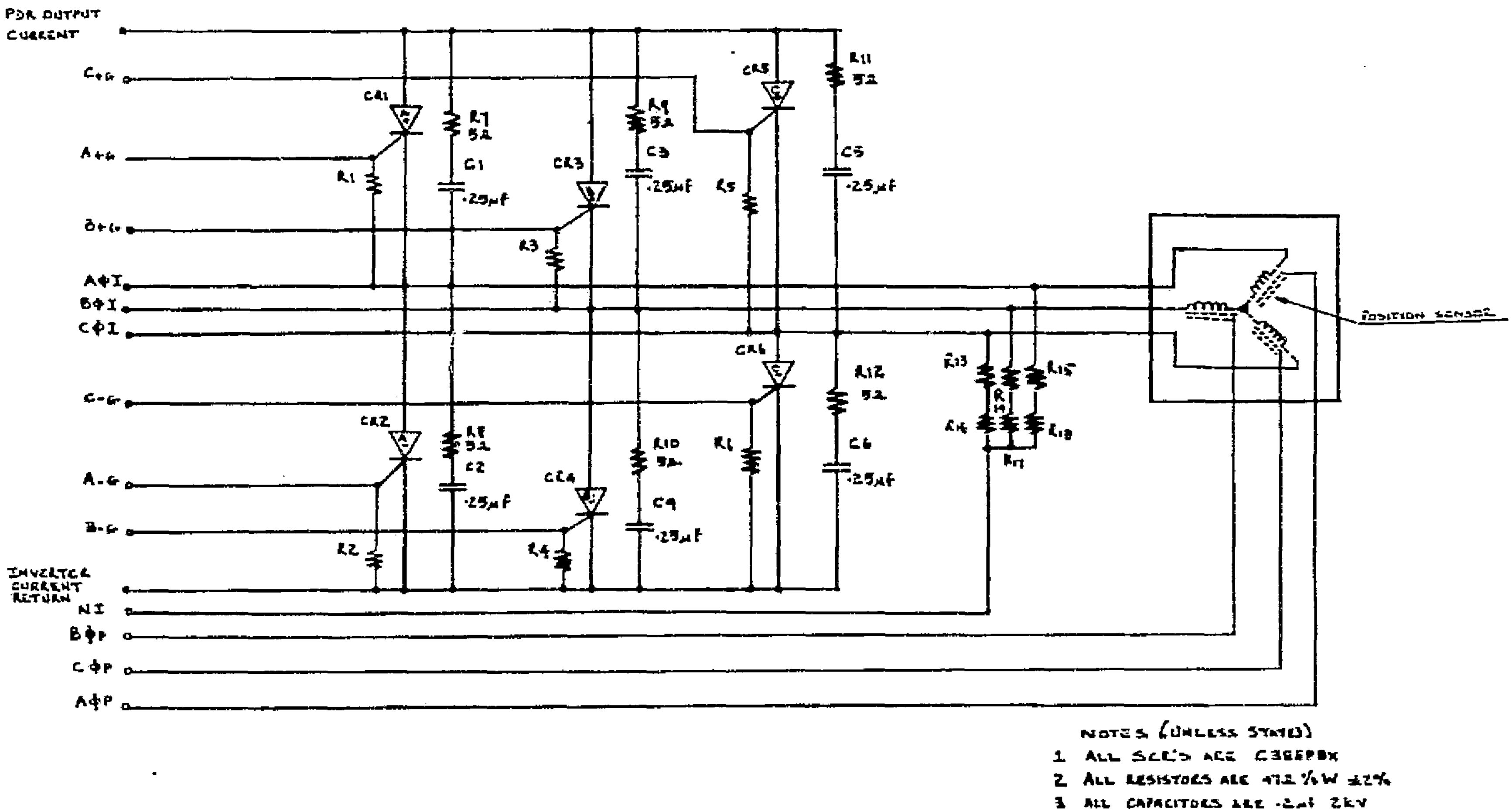


Figure 5-10. Inverter Bridge Circuit Design

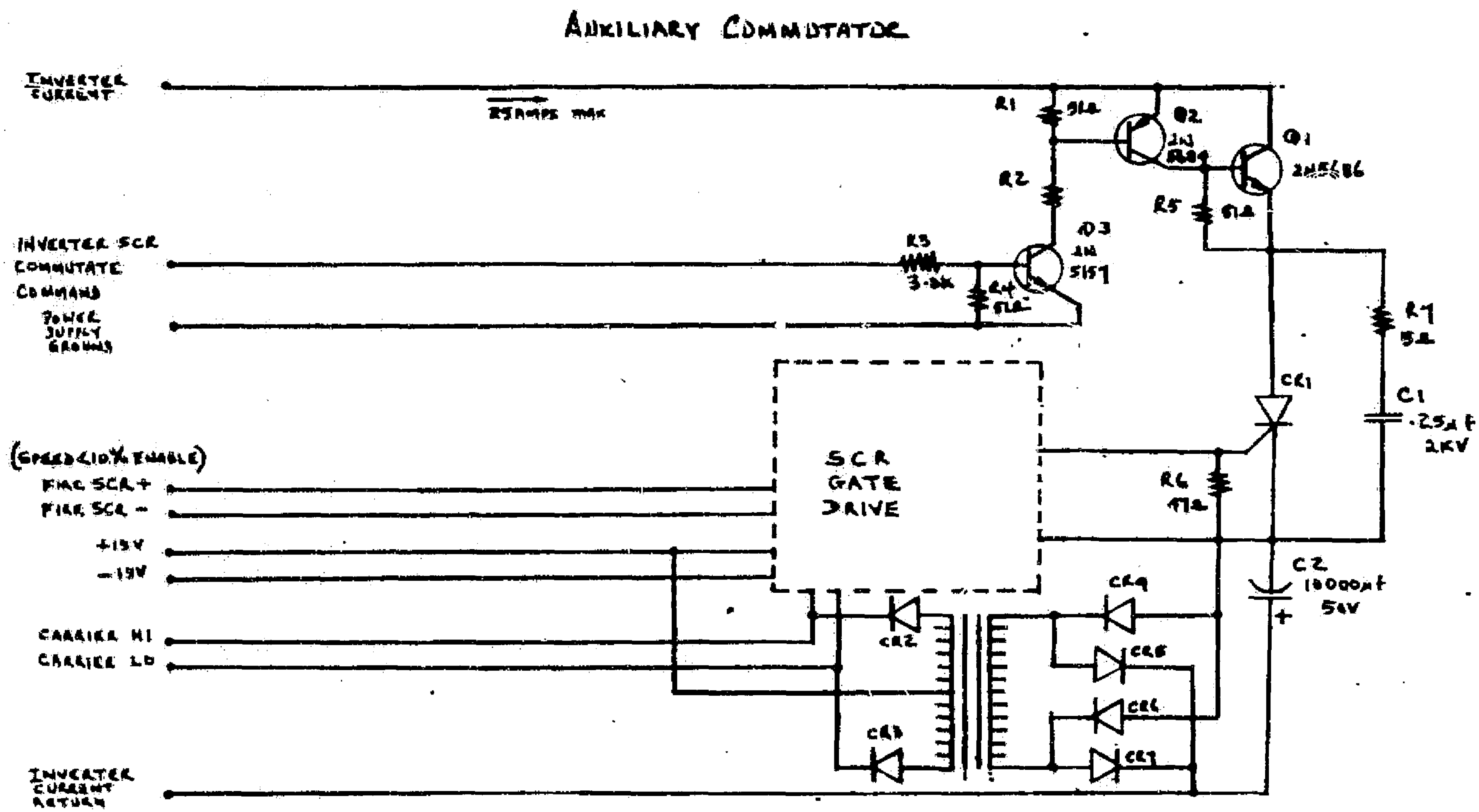


Figure 5-11. Auxiliary Commutator Circuit Design

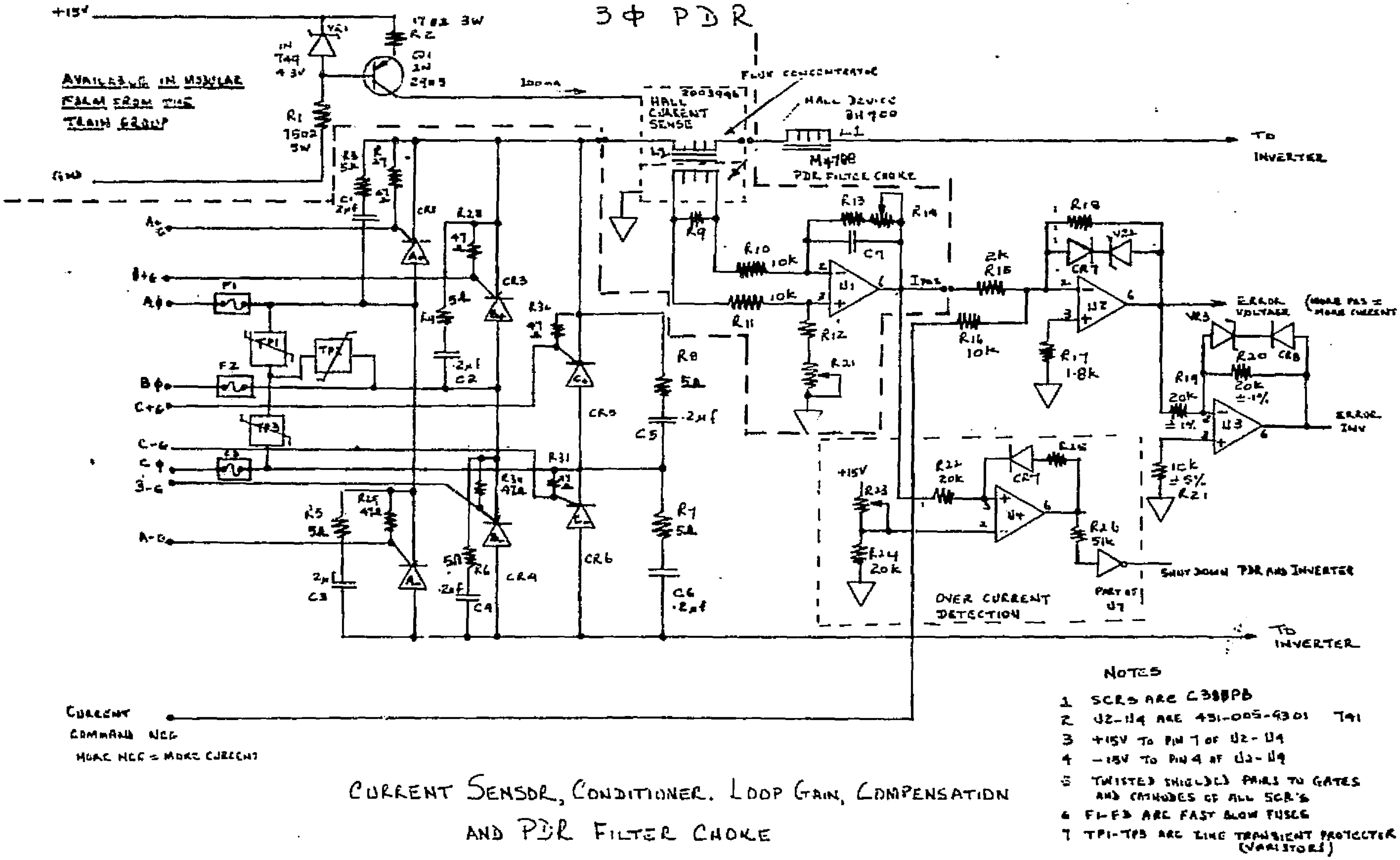
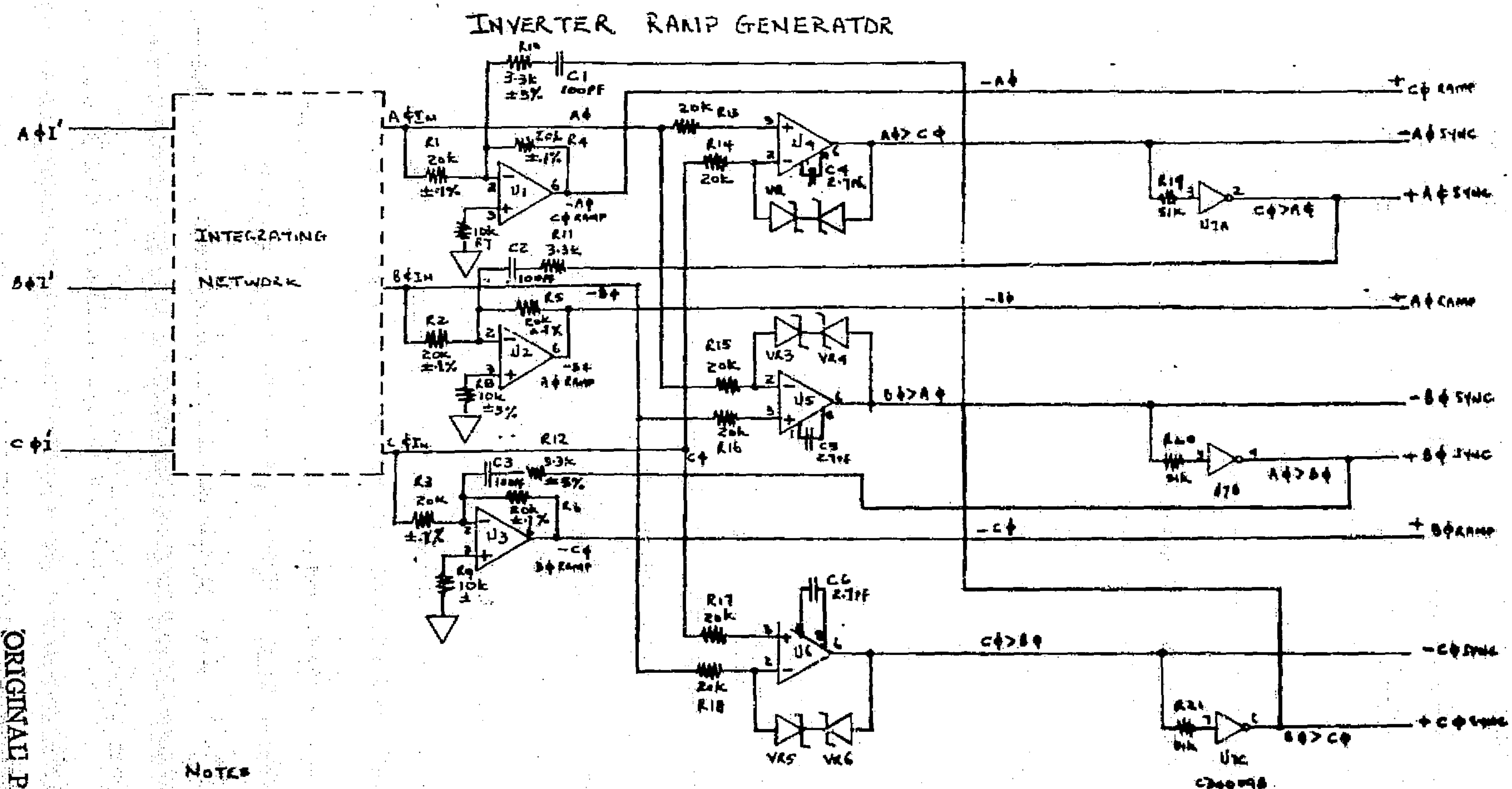


Figure 5-12. Three-Phase PDR Circuit Design



NOTES

1 ALL ZENER DIODES ARE IN766
2 U1-U3 ARE 431-005-9301 741
3 U4-U6 ARE 431-049-9301 748
4 U7 IS CD4049
5 RESISTORS ARE IN OHMS ±5%, 1/4 W
6 +15V DC TO PIN 7 OF ALL AMPLIFIERS
7 -15V DC TO PIN 4 OF ALL AMPLIFIERS
8 3PF BETWEEN PINS 1 AND 8 OF U4-U6

Figure 5-13. Inverter Ramp Generator Circuit Design

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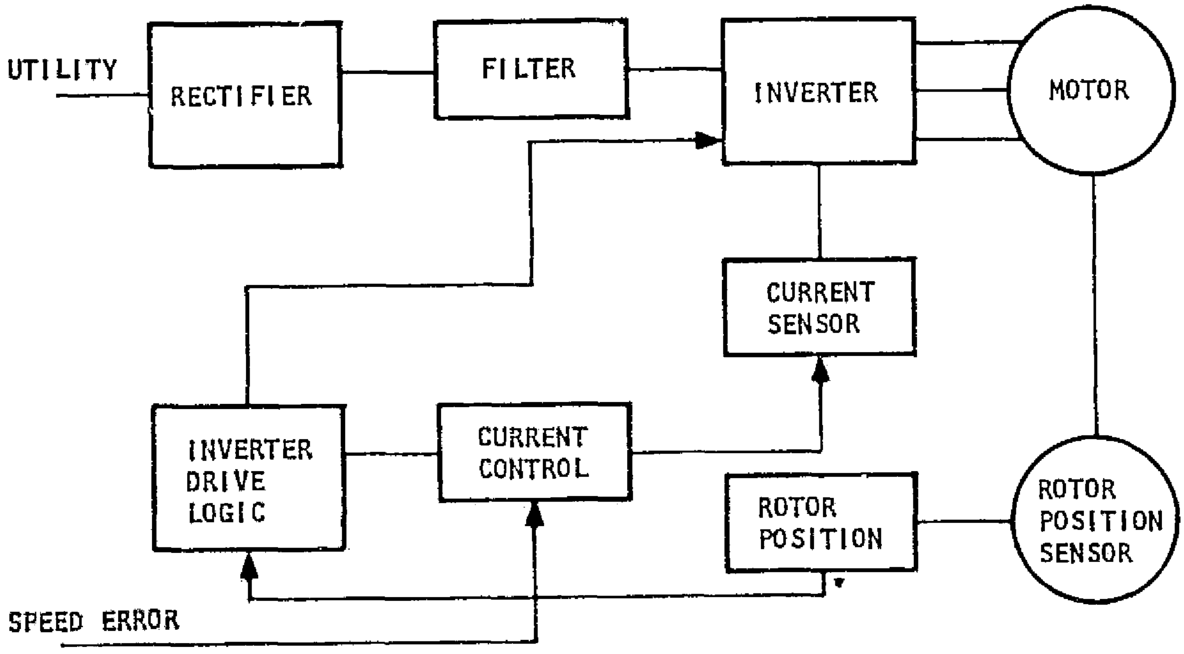


Figure 5-14. Transistor Control Block Diagram

Salient features of the transistor system are listed below.

- Motor current is conditioned twice--rectifier and inverter
- Motor current is controlled by pulse width modulation of the inverter
- Transistors are used for switches
- Rotor position sensor is required for starting
- Motor back EMF could be used for position sensor once motor is up to speed
- There are possible advantages to using back EMF for motor torque control
- Transistors require simpler drive circuitry

- High power transistors are not available in commercial quantities today at acceptable prices; they could be available in 1986 if there is a large enough market
- Rotor position sensor may have to work at high speeds

5.4.2 Circuit Description

The 3-ton motor controller employs transistors for the inverter switches and the dc power supply switches as mentioned previously. Figure 5-15 shows a simplified diagram of the motor controller. The motor controller can be divided into three sections for discussion--the dc power supply section, the inverter section, and the speed error generator.

The motor requires 23.5 amp driving current at 82,600 rpm and normal load. More current is required for starting. The motor controller is designed to limit the current to 50 amp. The load current is monitored continuously to ensure that the controller is operated within this limit.

5.4.2.1 Dc Power Supply

Single-phase 60-cycle ac power is full wave rectified and filtered before being fed into the switching power supply. This switching power supply is a current regulator using pulse width modulation and choke smoothing to supply the required current. The error signal from the motor speed is used to modulate the pulse width to supply the correct current to the motor. A current sensor will sense the load current to ensure that the supply current will not exceed 50 amp.

5.4.2.2 Inverter

The inverter consists of six sets of transistor switches, a commutating pulse generator, and a position sensor.

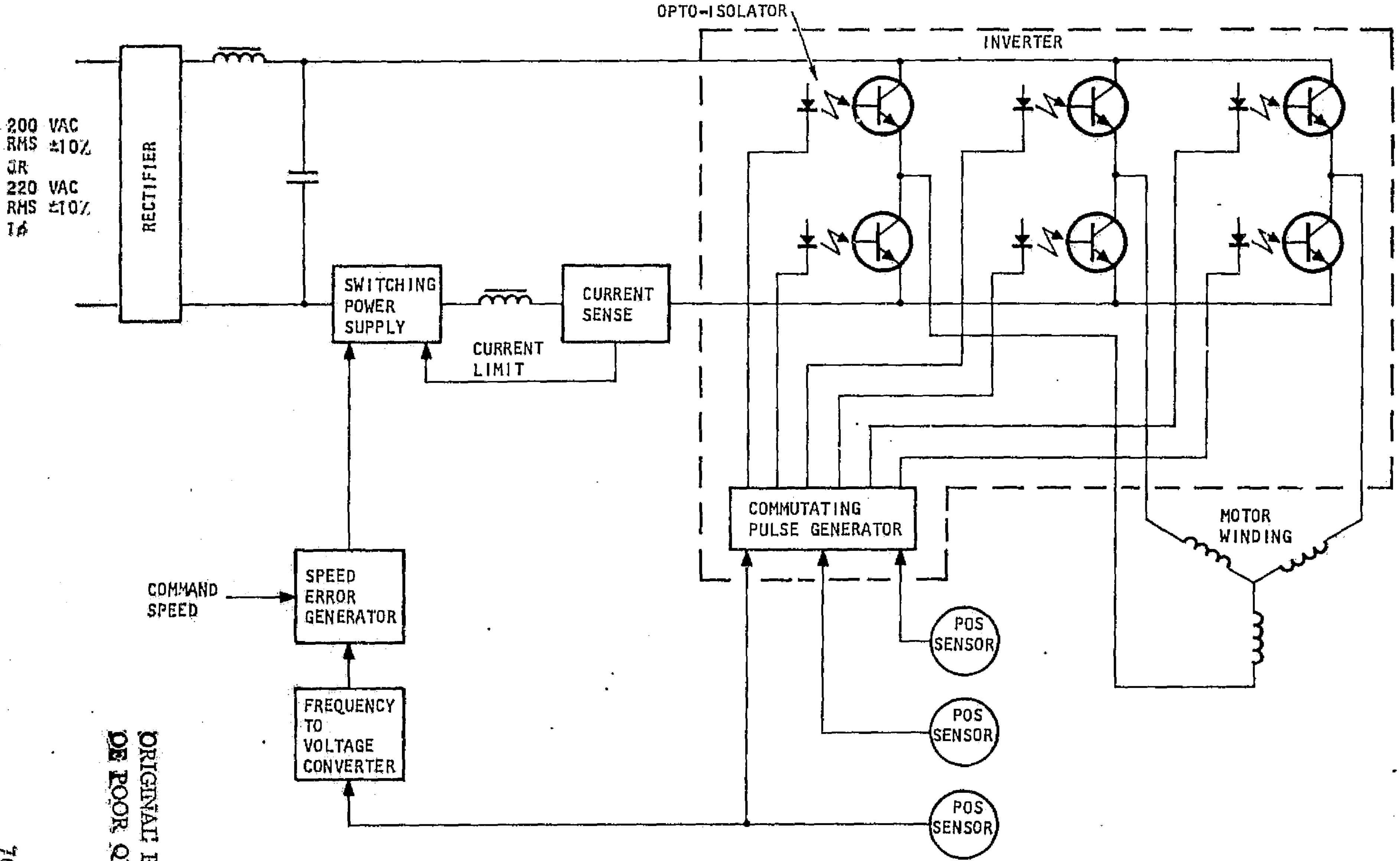


Figure 5-15. 3-Ton Motor Controller Diagram

The transistor switches are designed with four Darlington transistors connected in parallel sharing the base drive. The transistor is Baker clamped to reduce storage time (Figure 5-16), and hard reverse bias is applied during turnoff to further decrease storage time (Figure 5-17). This allows each of the four transistors to share the current equally during the turnoff period.

The switches are controlled by the commutating pulse generator through a photo-isolator. This reduces the possibility of false triggering induced by power supply and ground noise.

5.4.2.3 Speed Error Generator

A motor speed signal is generated by converting the pulse output from the position sensor into a voltage level (frequency to voltage conversion).

Motor speed signals are continuously compared with the motor speed command signal. The difference is processed, and a control signal is generated and sent to the current regulator. The magnitude of the control signal is a function of the motor speed, load condition, and magnitude of the error.

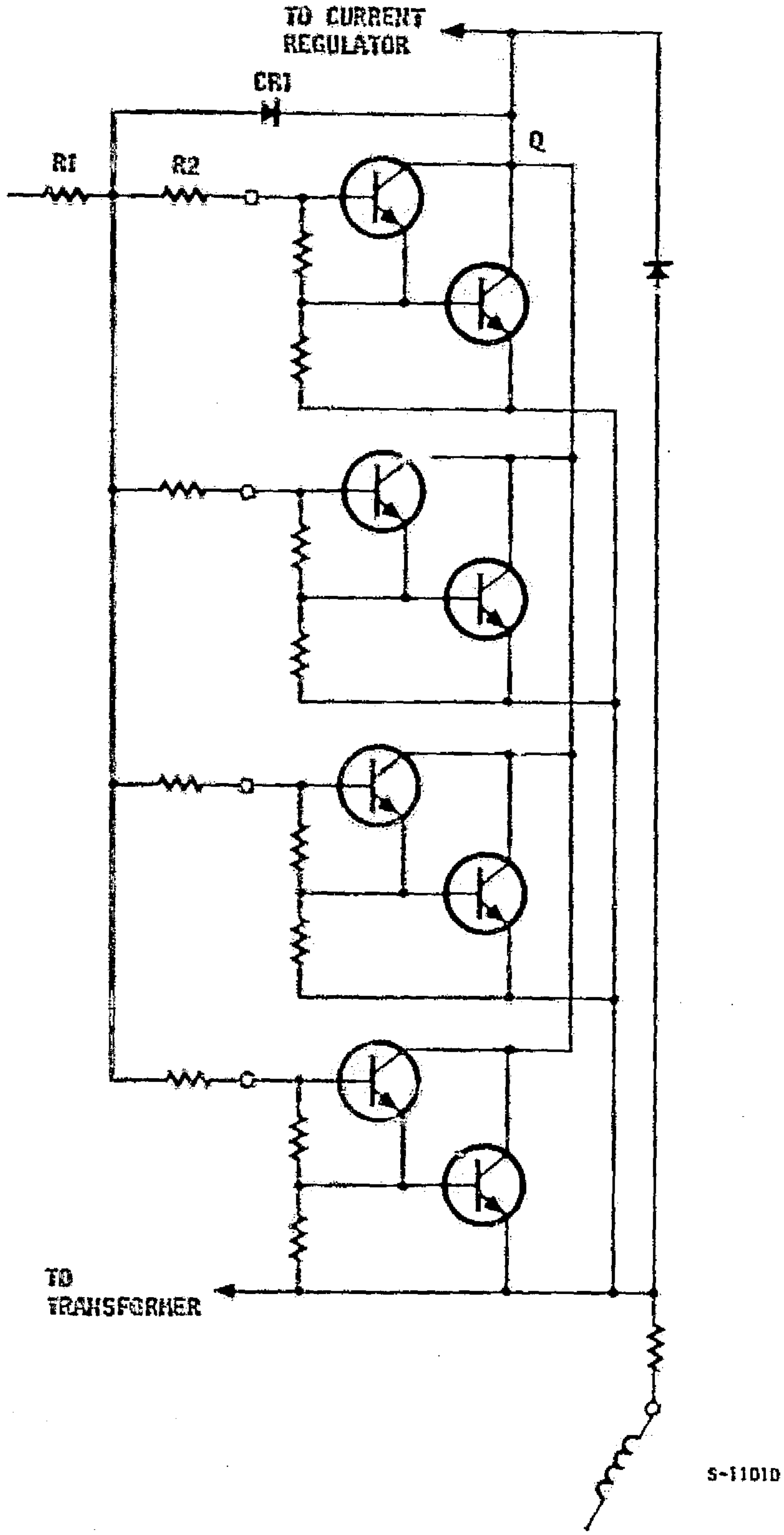


Figure 5-16. Inverter Switches

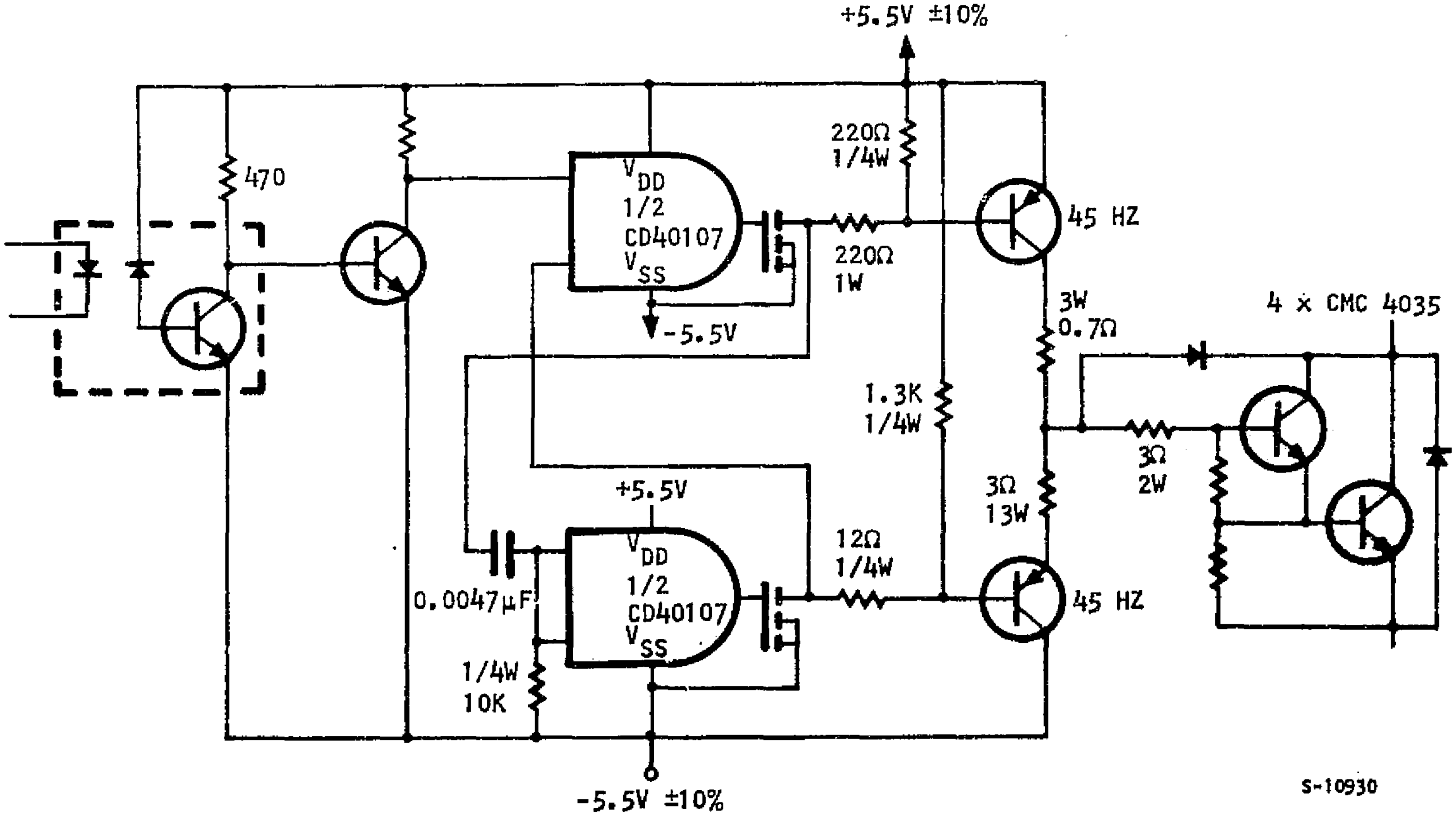


Figure 5-17. Inverter Driver

6. SYSTEM CONTROL

6.1 GENERAL

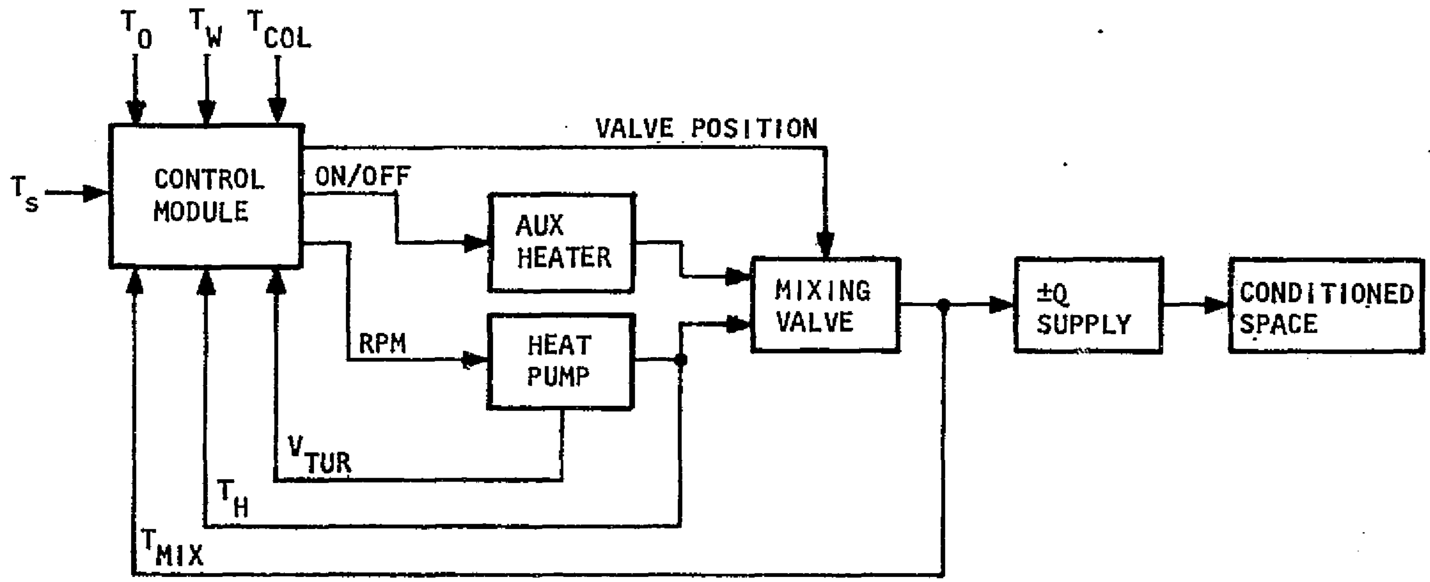
The control scheme for the heat pumps was described in Section 3. The control module function is to gather information from the system sensors, process this information, and issue appropriate commands to regulate the flow of heat into the conditioned space to achieve the desired heating and cooling effects with the minimum expense of utility power. Figure 6-1 is the control module functional diagram.

6.2 CONTROL MODULE MECHANIZATION

6.2.1 Heating Mode

The control module block diagram is shown in Figure 6-2. T_H is the controlled variable that sums up the heating effect coming from the heat pump. T_{ERROR} is the basic variable that computes the demand of heat to satisfy the desired temperature set point T_S. The temperature of water, T_W, in the storage tank acts as a safeguard in the heat pump operation to prevent compressor surge. T_W is also a measure of stored energy.

In the event the auxiliary heat is needed, T_{MIX} is used to regulate the mixing valve outlet temperature to satisfy the heat load. The mixing valve will add as much water as necessary in the 180° to 200°F range to bring T_{MIX} up to satisfy the heat load (required T_H). The basic control law is to allow the heat pump to run up to its maximum capacity before the auxiliary heater is brought on-line. Various control lockouts, time delays, etc. will be implemented to prevent premature use of the auxiliary heater. The heat pump will



T_S = TEMPERATURE SETTING

T_O = OUTSIDE TEMPERATURE

T_W = HEAT SOURCE WATER TEMPERATURE

T_H = HEAT PUMP OUTLET TEMPERATURE

T_{MIX} = MIXING VALVE OUTLET TEMPERATURE

V_{TUR} = FREON FLOW RATE

T_{COL} = COLLECTOR TEMPERATURE

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Figure 6-1. Multifamily System Control Module Functional Diagram

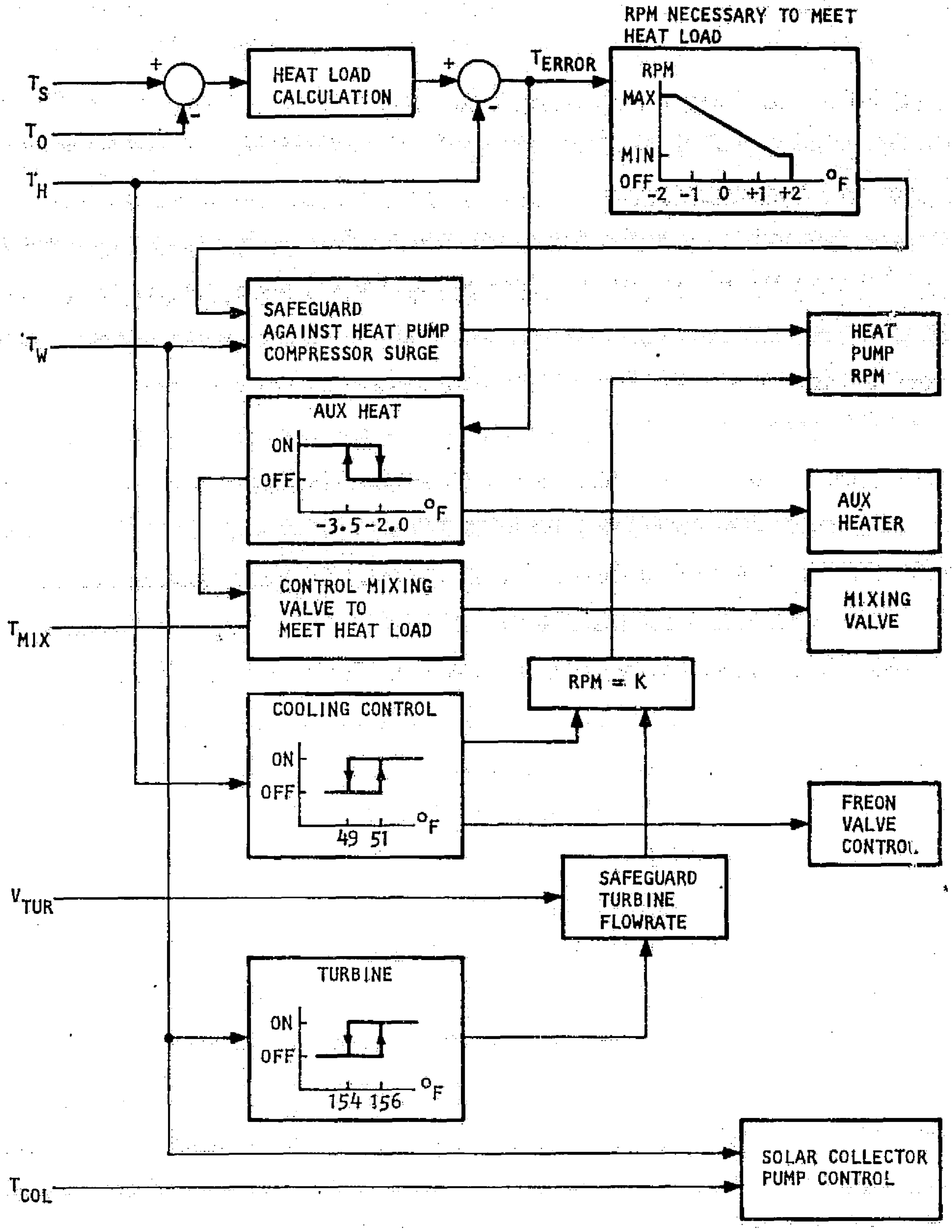


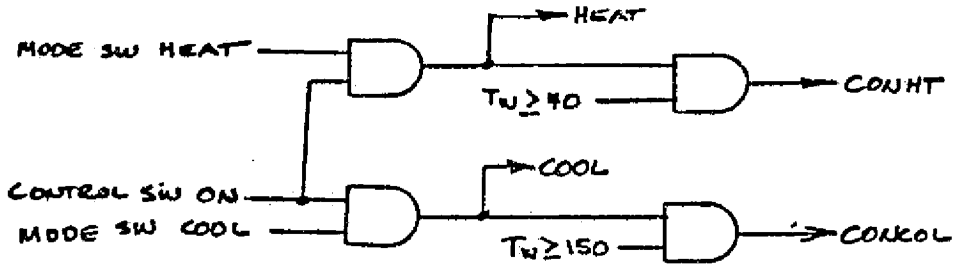
Figure 6-2. Multifamily System Control Module Block Diagram

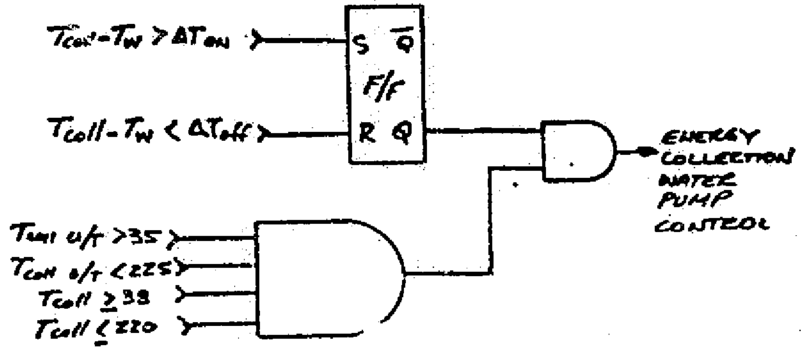
be given sufficient time to try to satisfy the heat load on its own. As a consequence, the heat pump will always be running at its maximum rpm whenever the auxiliary heater is used. The flow chart detailing this operation is shown in Figure 6-3.

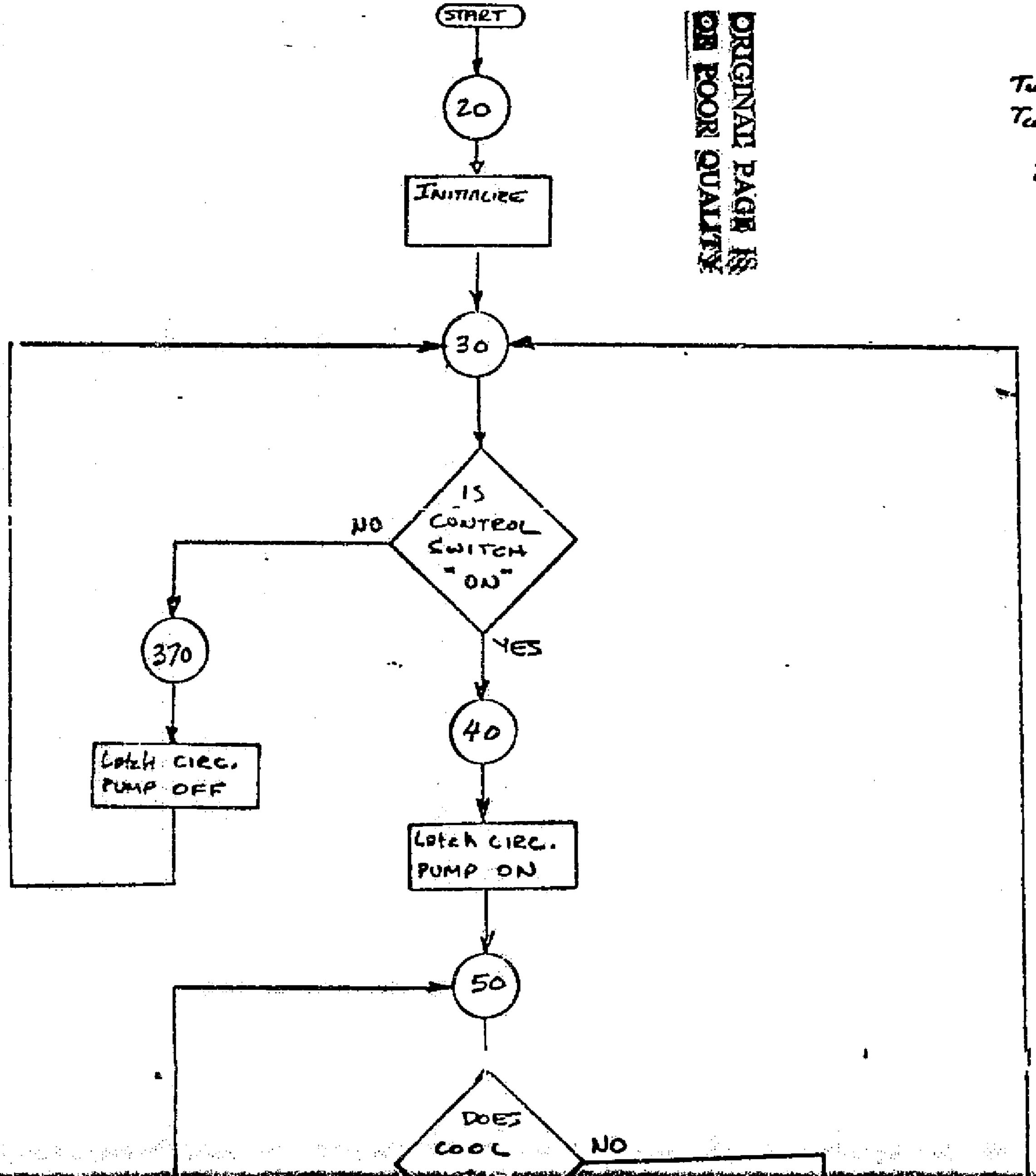
Necessary inputs for the system are T_S, T_O, T_H, T_W, T_{MIX}, T_{COL}, and V_{TUR}. Based on a set of control laws, these inputs are manipulated and in turn produce a set of outputs. There are four basic outputs from the control module: ON-OFF AUX HEATER; ON-OFF SOLAR COLLECTOR PUMP; LINEAR HEAT PUMP RPM; and LINEAR MIXING VALVE POSITIONING. Additional outputs are for turning on and off the various valves in shutdown or secondary controls.

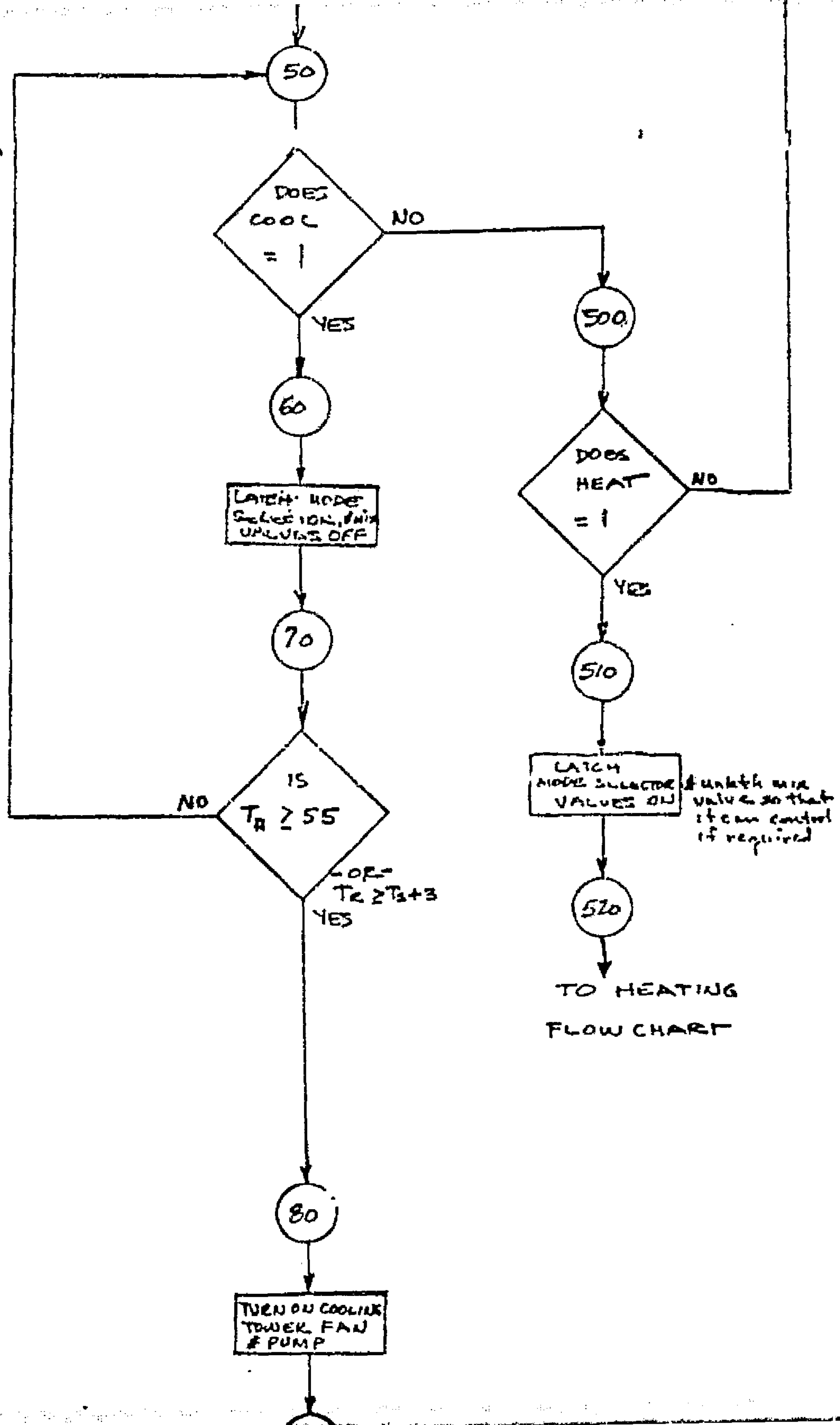
The control laws are a set of functions and rules governing the reaction to inputs. They are determined and executed by a digital microprocessor. The various rules and algorithms are stored in the memory. By programming the memory, the reaction of the control module to input changes and subsequent output controls is easily changed. The hardware or input/output interfacing will have a certain amount of flexibility already designed in, such as extra monitoring, input sensing channels, and output control channels. In this manner, system evolution can be accommodated without expensive hardware redesign. The block diagram of hardware organization of the control module is shown in Figure 6-4.

Hardware implementation of the digitial microprocessor is as shown conceptually in Figure 6-4. Various temperature measurements are sensed by inexpensive thermistors housed in brass bolt assemblies, which are commonly available. The various thermistors are scanned by the analog input multiplexing circuits, and each temperature is converted into an eight-bit digital word by the analog-to-digital (A/D) converter. Control system constants are

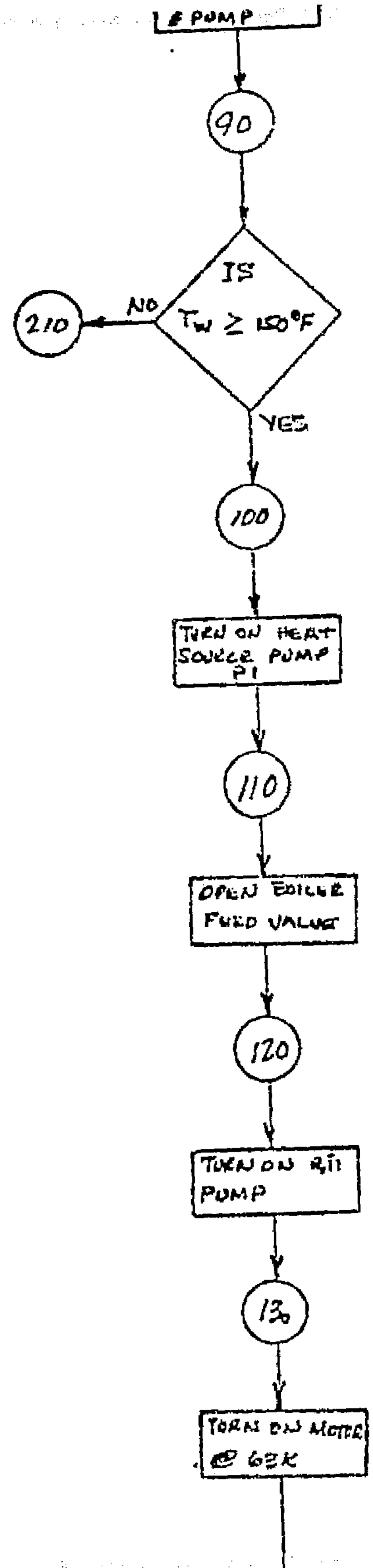

MODE SW HEAT
HEAT
Tw ≥ 40
CONHT
CONTROL SW ON
MODE SW COOL
COOL
Tw ≥ 150
CONCOL


Tcoll - Tw > ΔTon
Tcoll - Tw < ΔToff
S
R
Q
F/F
Tcoll ≥ 33
Tcoll ≤ 220
ENERGY COLLECTION WATER PUMP CONTROL

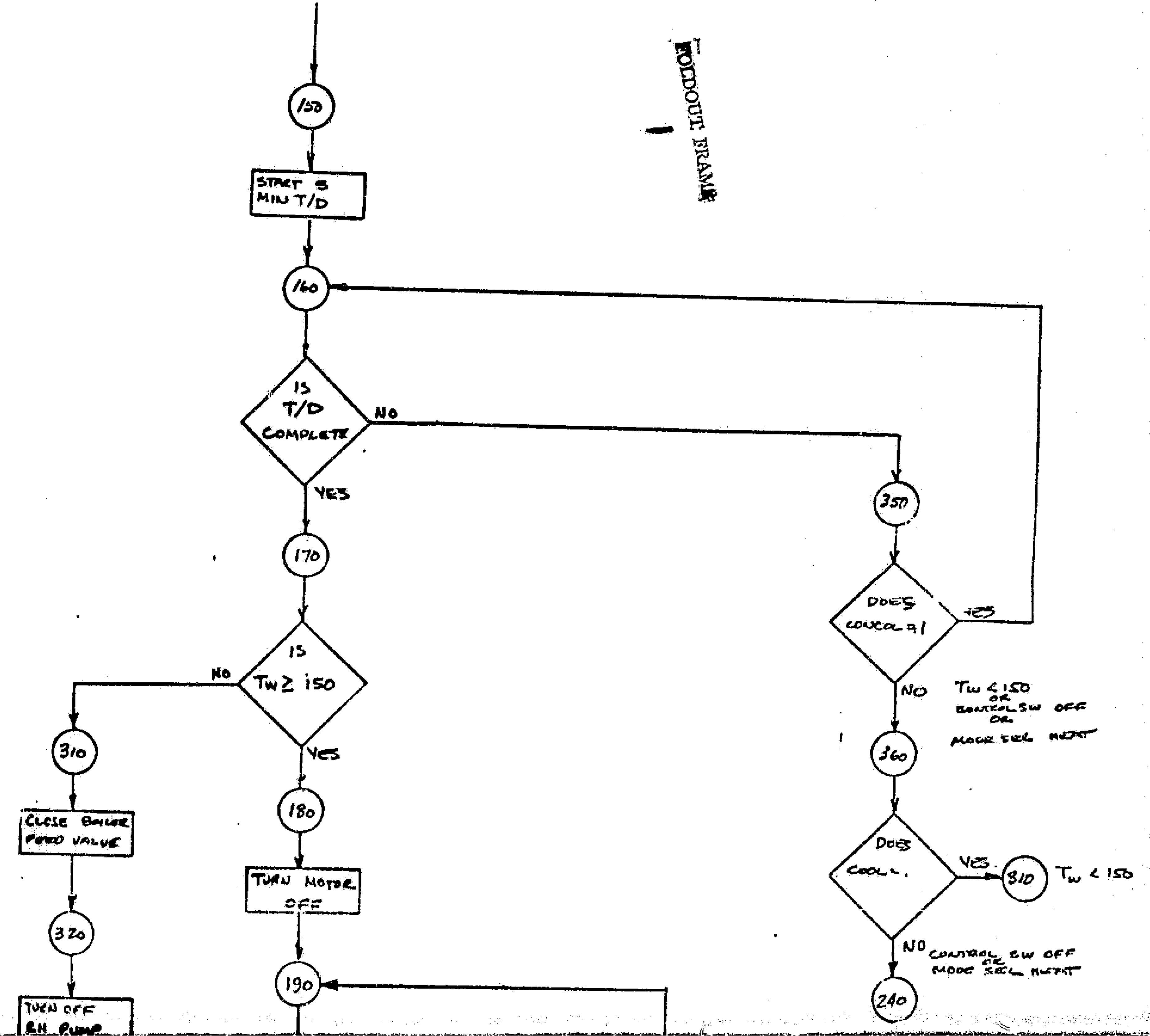

START
20
INITIALIZE
30
IS CONTROL SWITCH "ON"
NO
YES
370
Latch circ. pump off
40
Latch circ. pump on
50
DOES COOL
NO

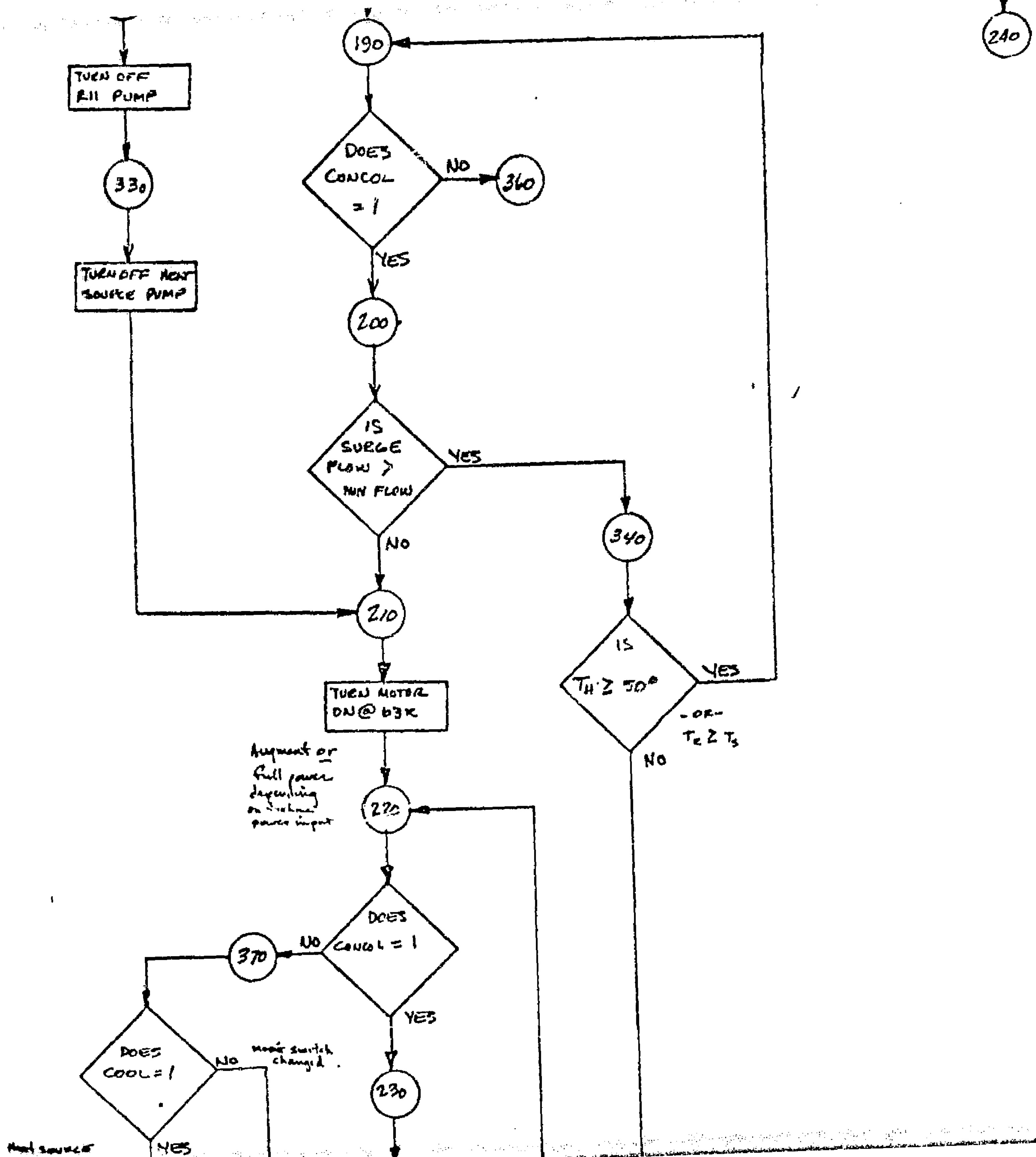


FOLDOUT FRAME 2


PUMP
90
IS
$T_W \geq 150°F$
NO
210
YES
100
TURN ON HEAT SOURCE PUMP P1
110
OPEN BOILER FEED VALVE
120
TURN ON R11 PUMP
130
TURN ON MOTOR @ 63K

FOLDOUT FRAME





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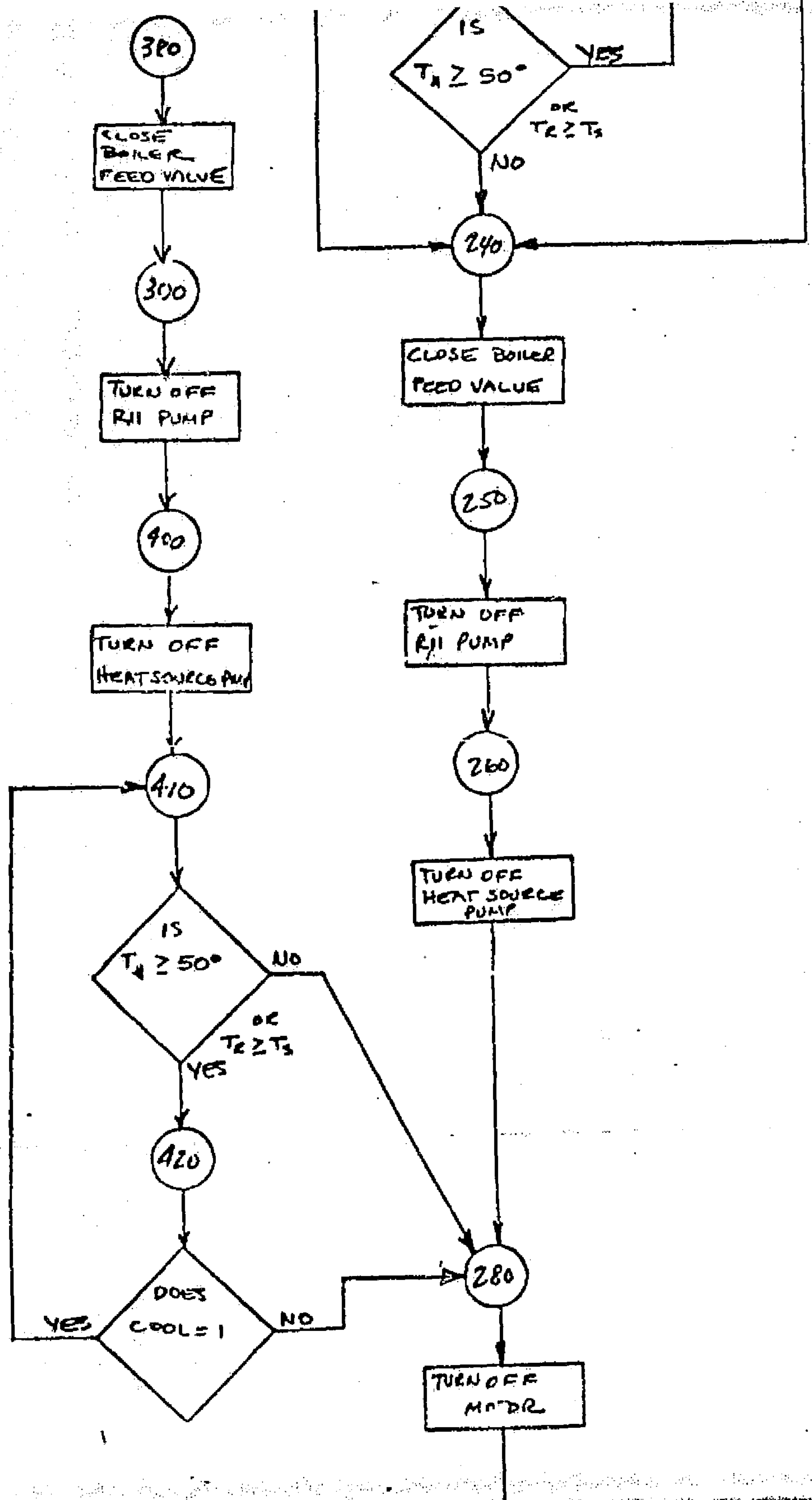
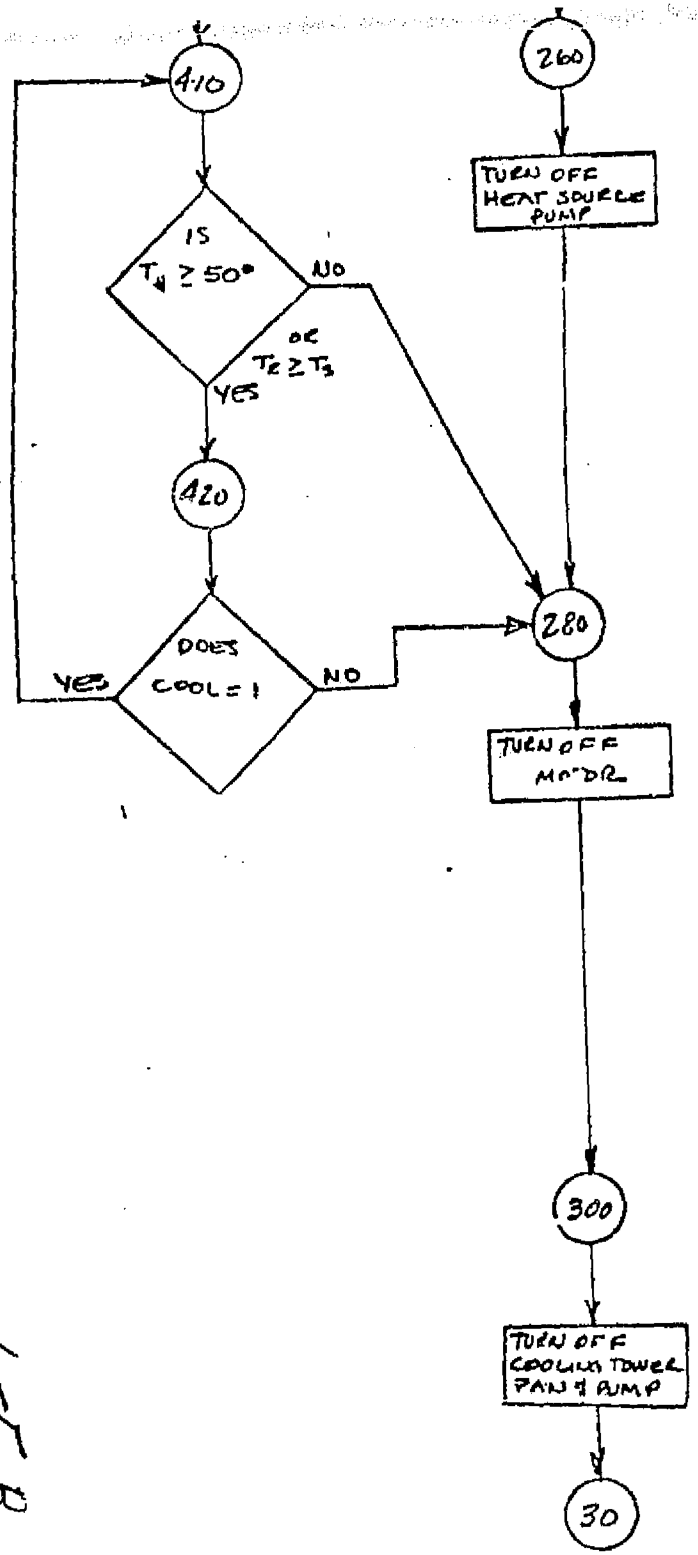
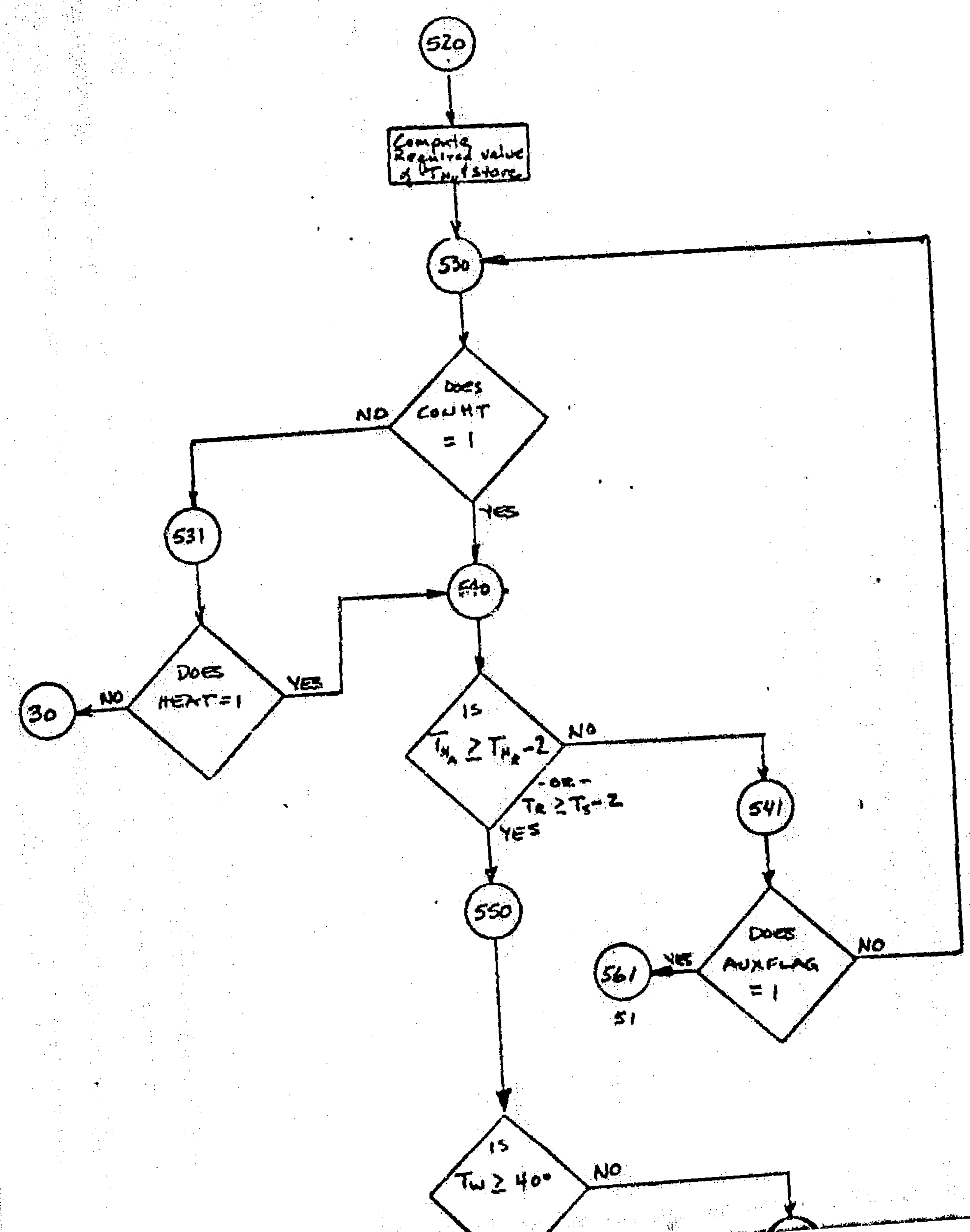

380
CLOSE BOILER FEED VALVE
300
TURN OFF R11 PUMP
400
TURN OFF HEATSOURCE PUMP
410
IS $T_4 \geq 50°$ OR $T_E \geq T_S$
NO
YES
420
DOES COOL = 1
YES
NO
IS $T_4 \geq 50°$ OR $T_E \geq T_S$
YES
NO
240
CLOSE BOILER FEED VALUE
250
TURN OFF R11 PUMP
260
TURN OFF HEAT SOURCE PUMP
280
TURN OFF MOTOR


410
IS
T4 ≥ 50°
OR
Tc ≥ T3
NO
YES
420
DOES
COOL = 1
YES
NO
260
TURN OFF
HEAT SOURCE
PUMP
280
TURN OFF
MOTOR
300
TURN OFF
COOLING TOWER
FAN & PUMP
30



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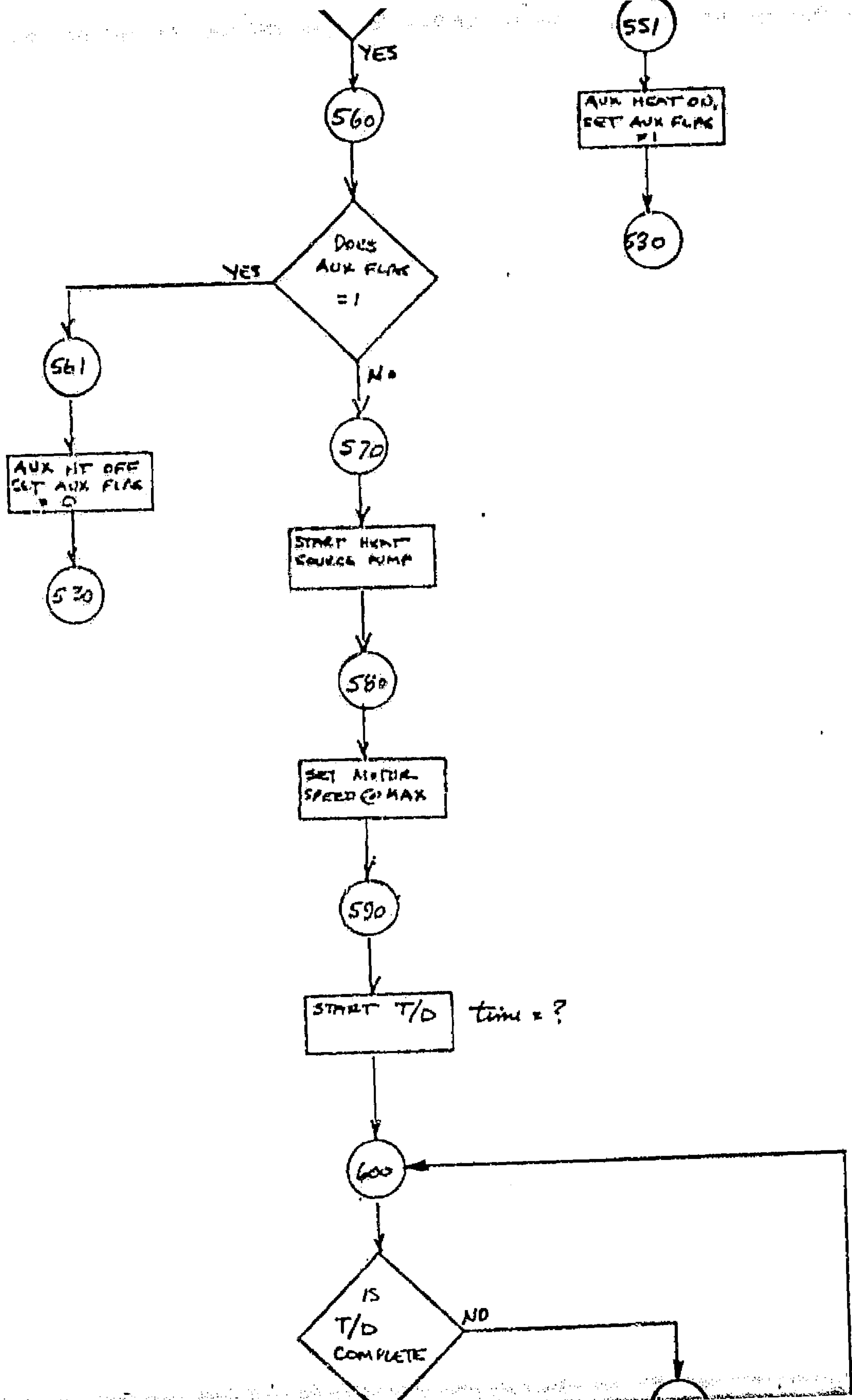
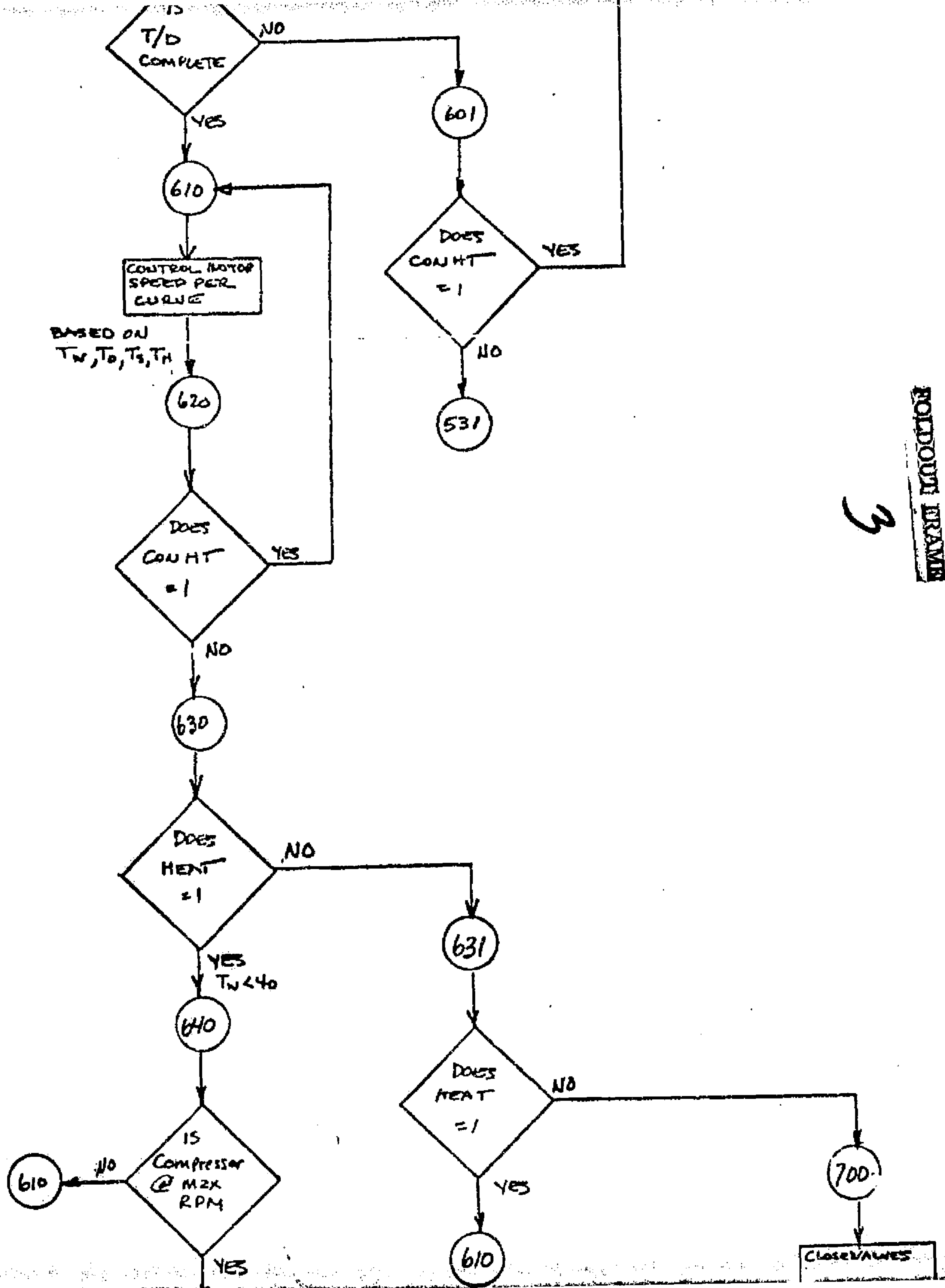


YES
560
551
AUX HEAT ON, SET AUX FLAG =1
530
DOES AUX FLAG =1
YES
NO
561
AUX HT OFF SET AUX FLAG = 0
530
570
START HEAT SOURCE PUMP
580
SET MOTOR SPEED @ MAX
590
START T/D
time = ?
600
IS T/D COMPLETE
NO



FOLDOUT FRAME 3

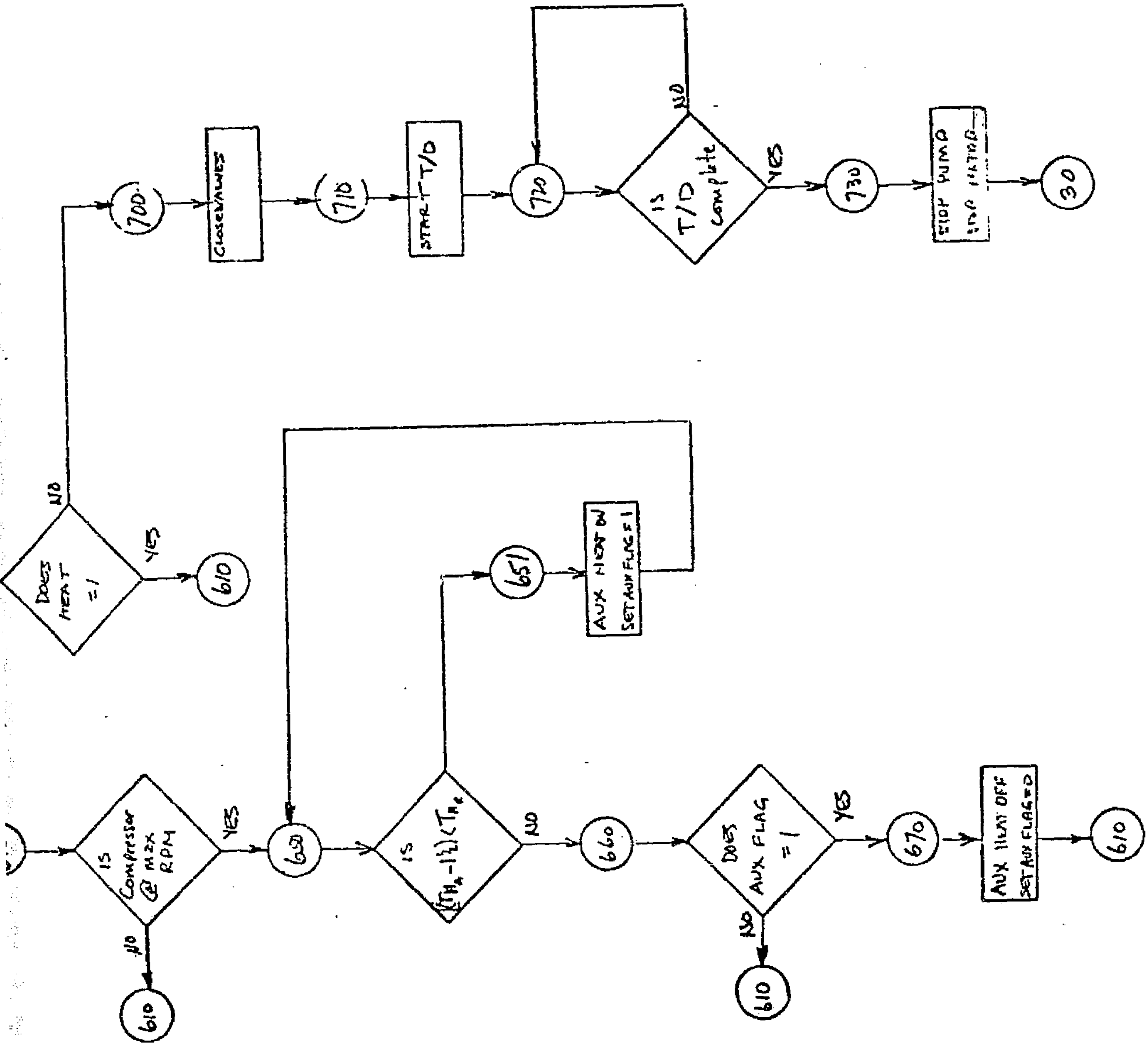


Figure 6-3. Multifamily Heating/Cooling Control System Flow Chart

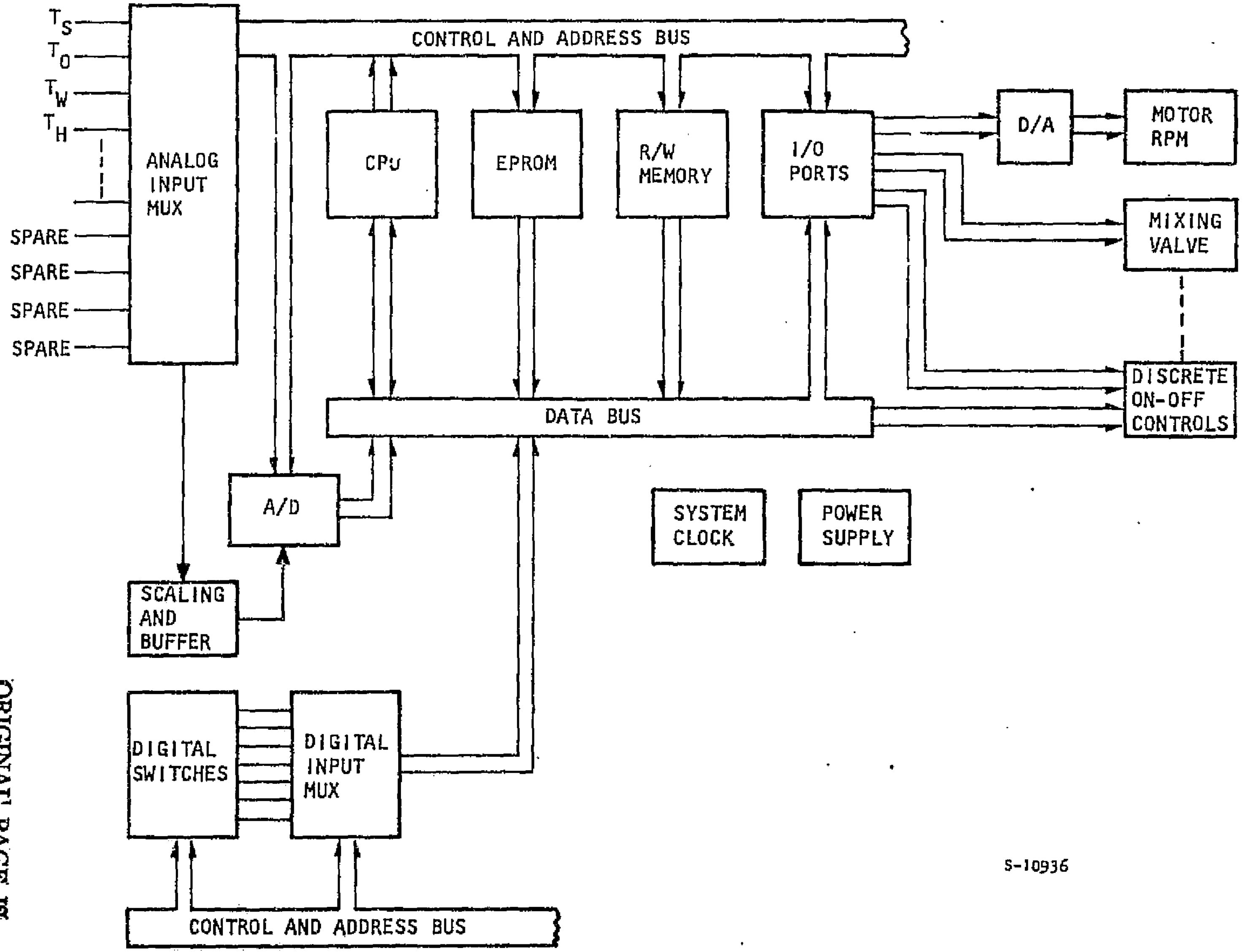


Figure 6-4. Control Module Hardware Block Diagram

made adjustable with digital switches. These switches are scanned by the microprocessor under program control. The CPU will execute instructions stored in the electrically programmable read-only memory (EPROM), and temporary scratch pad memory is provided by the read-write memory (RWM). After the necessary manipulation of data is completed, the CPU transmits the desired control signals to the outside world through the input/output (I/O) ports. In the case of motor rpm control, a digital-to-analog (D/A) converter is used. The mixing valve will have an incremental light intensity coupling with photocouplers. Other discrete on-off controls are compatible with direct digital bus interfacing.

The control module will be partitioned into two separate units. The analog input multiplexing, motor rpm control, mixing valve interfacing, discrete on-off controls, and power supply will be located with the heat pump package. The CPU, memory, I/O ports, and system clock will be located apart from the heat pump package in the residence or conditioned space, where ambient temperature range is moderate. This partitioning is compatible with sensor wiring and effects a cost saving on microprocessor components by eliminating the need for extended temperature parts.

6.2.2 Cooling Mode

An additional flowrate sensor is needed to safeguard the compressor operation in the cooling mode. T_H is now compared with a predetermined value to control the motor to either run at a constant rpm or turn off. T_W and the flowrate sensor will determine the need for the heat pump motor to cut in to supplement turbine power in order to meet the cooling requirement. The flow chart detailing this operation is shown in Figure 6-3.

6.3 COLLECTOR CONTROL

The circulation of water through the solar collectors is controlled by sensing T_{COL} and comparing it with T_W to preclude circulating T_W when a net heat gain is not possible. Lockout against freezing and boiling conditions is also implemented. The flow chart detailing this operation is shown in Figure 6-3.

7. HEAT PUMP HEAT EXCHANGERS

7.1 OVERALL DESIGN APPROACH

The heat exchangers (evaporators, condensers, boilers, and heating/cooling air coils) for the proposed heating/cooling systems are sized to meet the specified requirements at minimal cost. Standard Dunham-Bush air conditioning and heat pump products, which are built primarily for refrigerants R-12, R-22, and R-502, are used in this system with minor modifications to accommodate increased water flow rates and the lower vapor pressure R-11 refrigerant. Volume production tooling is available for the heat exchangers selected even though they are not standard catalog sizes. They will be fabricated as any other Dunham-Bush heat exchanger production run.

7.1.1 Design Procedure

The heat exchangers described in this section were designed according to the following procedure:

(a) Problem statements were generated as a result of computerized system optimization studies.

(b) Basic heat transfer data available for the Dunham-Bush standard heat transfer surfaces were reduced in a form suitable for use by the AiResearch heat exchanger design computer programs. In addition, single tube heat transfer tests were conducted to characterize the boiling/evaporating heat transfer and pressure drop behavior of R-11 under conditions similar to the actual heat exchanger design conditions.

(c) The computer programs were exercised to generate a number of heat exchanger configurations and sizes corresponding to various basic heat transfer surfaces. All these candidate heat exchanger designs met the performance requirements developed in (a) above.

(d) The candidate designs were reviewed by Dunham-Bush for selection on the basis of cost, packaging constraints, and suitability for the kind of service considered.

(e) As a result of the Dunham-Bush investigations, iterations affecting the system performance were made to arrive at an overall optimum solution.

7.1.2 Heat Exchanger Design Approach

The system heat exchangers use three types of basic Dunham-Bush heat transfer surfaces. These are described in the following paragraphs.

7.1.2.1 Finned-Tube Coil Heat Exchangers

These heat exchangers transfer heat between a liquid (or refrigerant) circulated within the tubes and air circulated outside the tubes over the extended surfaces. The extended surfaces (fins) are always exposed to the gas side to compensate for the heat transfer coefficient between the gas and metal, which is relatively low compared with that between the liquid and metal. This basic heat transfer surface is presently used in many different Dunham-Bush air conditioning and heat pump products.

A typical fin selection is depicted in Figure 7-1. The wavy aluminum fins are formed by high precision dies that draw fin collars in the fin stock. The aluminum fin stocks that run vertically are tightly bonded to the horizontal copper tubes by mechanical expansion of the tubes. The fin collars not only provide accurate control of the fin spacing, but completely cover the tube for greatest heat transfer efficiency and coil protection. The wavy fins produce a rippled airflow pattern throughout the coil, creating the air turbulence

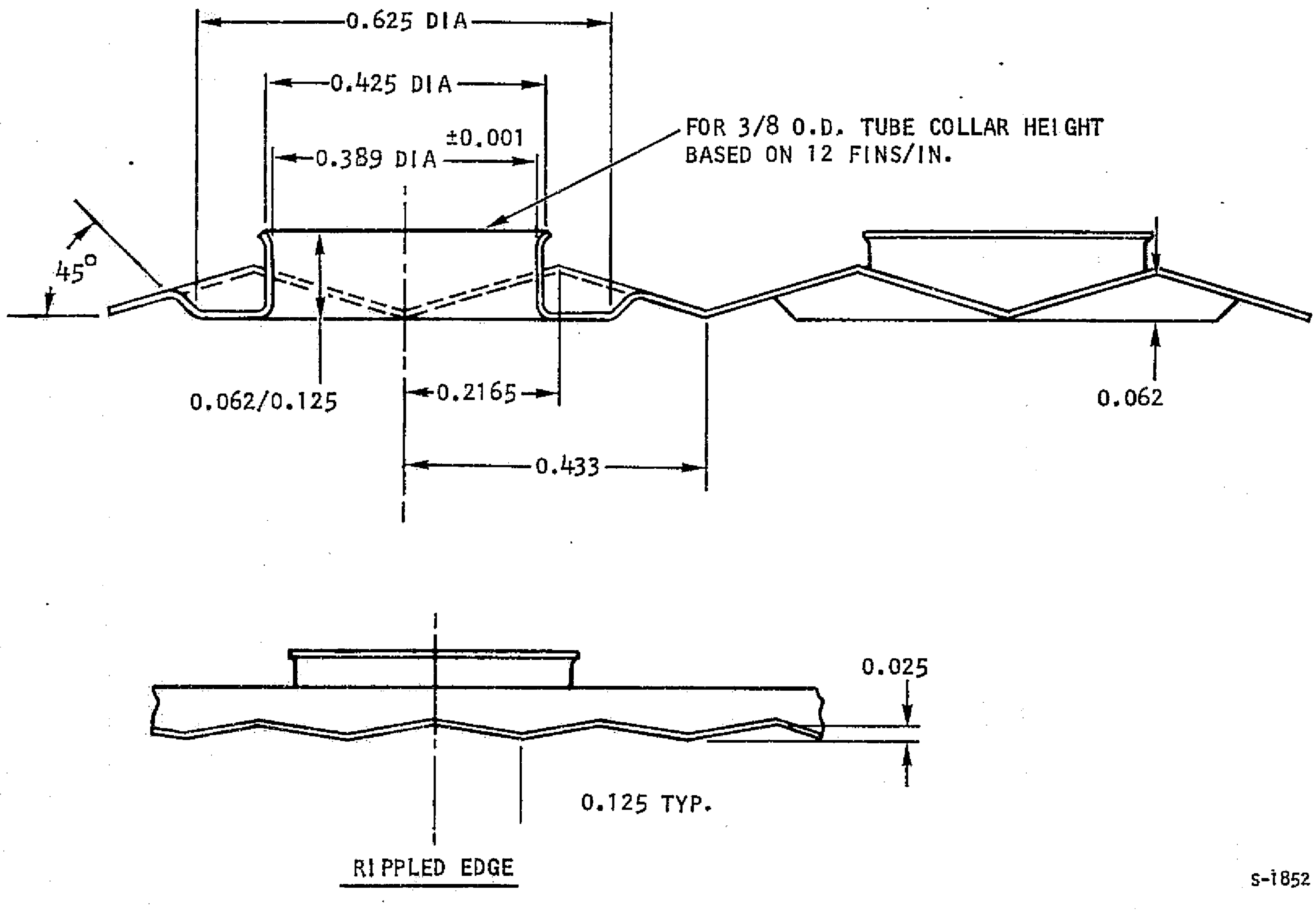


Figure 7-1. Typical Wavy Fin

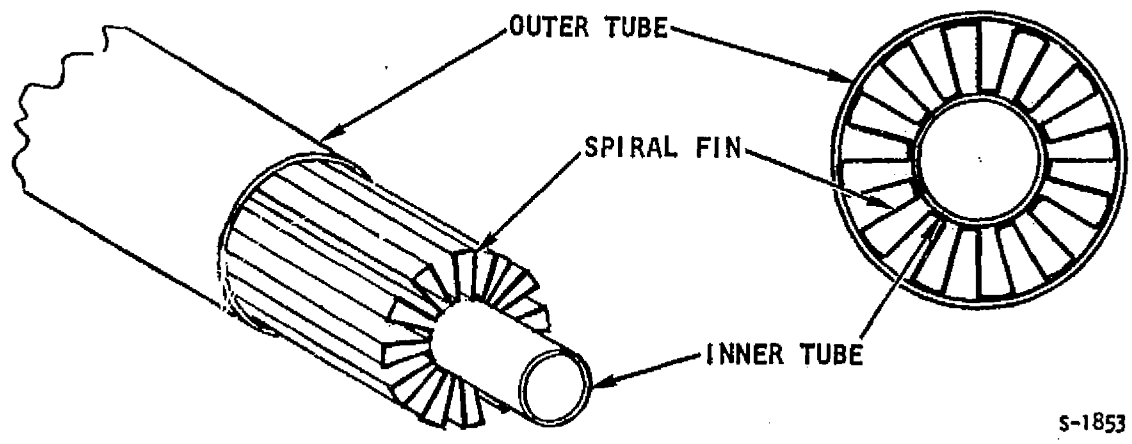


Figure 7-2. Inner-Fin Tube

necessary for efficient heat transfer. The tube size as well as the tube and fin spacing may vary to suit the constraints of design specification.

An inner-fin tube as shown in Figure 7-2 is used for the 3- and 10-ton R-11-to-air heat exchangers.

7.1.2.2 Inner Fin Water Chiller

The inner fin water chiller is a shell-and-tube heat exchanger. The copper tube bundle is contained within a steel shell; water moves over the outside of the tubes and refrigerant moves within. This is a standard heat transfer package that has been marketed by Dunham-Bush for many years.

A sketch of the inner-fin tube arrangement is shown in Figure 7-2. The inner fin is manufactured by inserting a small-diameter copper tube into a copper tube of a larger diameter. A folded spiral fin of copper sheet stock is wedged tightly between the OD of the inner tube and the ID of the outer tube.

The folds (points) vary in number around the periphery of the inner diameter; this variation is calculated to produce the most efficient heat transfer.

The inner tube is plugged so that the refrigerant flows only through the finned annulus between the inner and outer tubes while heat is being transferred to or from the water on the outside of the tube.

Water is directed back and forth across the tubes by using baffles. The baffle spacing is calculated to give the most efficient heat transfer on the outside of the tube.

7.1.2.3 Shell-and-Tube Condensers

The third type of heat exchanger is a horizontal low fin tube condenser. In this heat exchanger, water flows through the tubes and R-11 refrigerant condenses on the finned outer surface of the tubes. The condensate drips off of the tubes into the lower portion of the shell, where it can be subcooled by tubes submerged in the R-11 condensate.

Removable water headers are provided for easy cleaning of the inside of the tubes. These exchangers use copper water tubes, steel tube sheets, and cast iron headers constructed and certified to the ASME pressure vessel code. The copper water tubes are expanded into aluminum plate fins to form the outer R-11 condensing surface in this design. The fins are necessary to provide efficient condensation of the refrigerant. To provide subcooling, some of the tubes may be immersed in the R-11 condensate.

7.2 HEAT EXCHANGER CHARACTERISTICS

Tables 7-1, 7-2, and 7-3 summarize the sizes of the heating and heating/cooling system heat exchangers of the three types described above. In order to develop a heat pump package common to heating-only and heating/cooling systems, it has been necessary to size the cooling system boiler as well as the other heat exchangers. Further, the use of common heat exchangers for heating and cooling required sizing for both modes of operation.

TABLE 7-1

FINNED-TUBE COIL HEAT EXCHANGERS

Heat Exchanger	No. Required	Tube Dia., in.	Tube Wall Thickness, in.	Finned Tube Length, in.	Fin Width, in.	Rows Deep	Fins Per in.	Fin Thickness, in.
3-ton/60 KBTUH indoor coil evap/cond R-11 to air	1	0.5 outer tube* 0.25 inner tube	0.028	24	30	6	12	0.008
10-ton/200 KBTUH indoor coil evap/cond R-11 to air	1	0.5 outer tube* 0.25 inner tube	0.028	27	85.5	6	12	0.008
25-ton/800 KBTUH heater/cooler Water to air	12	0.375	0.016	24	24	8	12	0.006

* Dunham-Bush inner fin tube

TABLE 7-2

INNER-FIN HEAT EXCHANGER CHARACTERISTICS

Heat Exchanger	Shell Dia., in.	Outer Tube		Inner Tube		No. of Tubes	Effective Tube Length, in.	Baffle Spacing, in.
		Dia., in.	Thickness, in.	Dia., in.	Thickness, in.			
3-ton evaporator/ boiler in water/ R-11 circuit	8.0	0.5	0.028	0.25	0.025	122	30	6
10-ton evaporator/ boiler	12	0.5	0.028	0.25	0.025	317	24	6
25-ton evaporator/ boiler	20	0.75	0.032	0.375	0.025	330	36	6
25-ton evaporator/ condenser	20	0.75	0.032	0.375	0.025	330	40	4

TABLE 7-3

CONDENSER CHARACTERISTICS

Heat Exchanger	Shell Dia., in.	Tube Dia., in.	Tube Wall Thickness, in.	Fins per in.	Fin Thickness, in.	No. of Tubes	Effective Tube Length, in.
25-ton condenser	18	0.375	0.025	14	0.008	390	48
3-ton condenser	10	0.375	0.025	14	0.008	80	42
10-ton condenser	14	0.375	0.025	14	0.008	180	54

7.3 3-TON/60 KBTUH HEAT PUMP HEAT EXCHANGERS

In this heat pump, two heat exchangers must perform at high effectiveness under two modes of operation: evaporation and condensation for the R-11-to-air coil, and evaporation and boiling for the R-11-to-water heat exchanger. To accommodate both processes in the same heat exchanger requires a compromise solution with careful attention paid to pressure drop in both modes.

7.3.1 Water-to-R-11 Heat Exchanger

For the water-to-R-11 heat exchanger, the Dunham-Bush inner fin tubing provides the means of handling both evaporation and boiling in an efficient manner.

Table 7-4 lists the problem statements (performance requirements) for this unit in the heating and cooling modes of operation. Note that the pressure drop on the R-11 side results in a significant change in evaporating temperature.

The dimensions of this unit are given in Table 7-2.

7.3.2 Condenser

The R-11-to-cooling tower water condenser is a shell-and-tube unit with condensation occurring on the shell side finned tubes. Some of the tubes are submerged in the liquid refrigerant to provide the required subcooling. The condenser is designed with removable headers for cleanable tubes, as required in cooling tower water service.

The design conditions for this condenser are listed in Table 7-5. Table 7-3 shows the condenser characteristics.

7.3.3 Air Coil Heating/Cooling Unit

The finned-tube air heating and cooling coil design conditions for the single-family residence are summarized in Table 7-6. The size of the 3-ton

TABLE 7-4

3-TON/60,000 BTU/HR
WATER-TO-R-11 HEATING/COOLING UNIT DESIGN CONDITIONS

	Cooling Mode Boiler	Heating Mode Evaporator
R-11 Side		
Q, Btu/hr	65,862	47,661
Flow, lb/hr	744	715
Inlet pressure, psia	87.9	9.8
Inlet temperature, °F	90.9	55
Inlet enthalpy, Btu/lb	28.8	32.83
Evaporating temperature, °F	187 at inlet; 186.3 at outlet	55 at inlet; 51 at outlet
Latent heat, Btu/lb	66.57	65.96
Outlet pressure, psia	87	9.0
Subcooling/superheat, °F	5	3
Outlet temperature, °F	191.3	54
Outlet enthalpy, Btu/lb	115.3	99.18
Water Side		
Flow, lb/hr	12,000	12,000
Inlet temperature, °F	195	60
Outlet temperature, °F	189.5	56°F
Pressure drop, psia	3.5 psia	3.5 psi
Pressure, psia	TBD	TBD

TABLE 7-5

3-TON HEAT PUMP CONDENSER REQUIREMENTS

Parameter	Units
<u>R-11 Side</u>	
Q, Btu/hr	102,698
Flow, lb/hr	1256
Inlet pressure, psia	21.7
Inlet temperature, °F	124.1
Inlet enthalpy, Btu/lb	108.24
Condensing temperature, °F	94.5
Latent heat, Btu/lb	76.5
Outlet pressure, psia	21.4
Subcooling, °F	5
Outlet temperature, °F	89.5
Outlet enthalpy, Btu/lb	26.5
<u>Water Side</u>	
Flow, lb/hr	20,500
Inlet temperature, °F	84.4
Outlet temperature, °F	89.4
Pressure drop, psia	5.0
Pressure, psia	TBD

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TABLE 7-6

3-TON/60,000 BTU/HR
AIR COIL HEATING/COOLING UNIT DESIGN CONDITIONS

	Cooling Mode Evaporator	Heating Mode Condenser
R-11 Side		
Q, Btu/hr	36,709	60,000
Flow, lb/hr	504.6	715
Inlet temperature, °F	54.0	183.7
Inlet pressure, psia	9.6	35.5
Inlet density, lb/cu ft		
Inlet enthalpy, Btu/lb	26.5	116.7
Evaporating/condensing Temperature, °F	54.0 inlet; 51 outlet	123.9
Outlet pressure, psia	9.0	35.0
Heat of vaporization, Btu/lb	72.3	82
Outlet enthalpy, Btu/lb	99.25	32.7
Pressure drop, psia	0.45	0.2
Superheat/subcooling, °F	3.5	5
Air Side		
Flow, lb/hr	5400	5400
Cfm	1200	1200
Inlet drybulb temperature, °F	78	70.3
Inlet wetbulb temperature, °F	67	--
Outlet temperature, °F	57.0 (sat)	116.5
Inlet pressure, psia	14.7	14.7
Pressure drop, in H_2O	0.33	0.24

system air coil for the single-family residence is given in Table 7-1. No problems are anticipated in this case to obtain adequate superheating or subcooling.

7.4 10-TON/200 KBTUH HEAT PUMP HEAT EXCHANGER

This system is a variation of the smaller, single-family residence unit. Here again, dual mode of operation is necessary in the heat source/sink heat exchangers.

7.4.1 Water-to-R-11 Heat Exchanger

The water flow configuration in this heat exchanger will be the same as for the smaller unit described above. The problem statements corresponding to the heating and cooling modes of operation are listed in Table 7-7. Heat exchanger size and characteristics are listed in Table 7-2.

7.4.2 Condenser

The R-11-to-cooling tower water condenser is a shell-and-tube unit with condensation occurring on the shell side finned tubes. Some of the tubes are submerged in the liquid refrigerant to provide the required subcooling. The condenser is designed with removable headers for cleaning the tubes, as required in cooling tower water service.

The design conditions for this condenser are listed in Table 7-8. Table 7-3 shows the condenser characteristics.

7.4.3 Air Coil Heating/Cooling Unit

This coil uses the wavy fin surface geometry on the air side. It is similar to the corresponding unit in the 3-ton 60 KBTUH system but larger in dimension. The design conditions for this heat exchanger in the heating and cooling modes of operation are listed in Table 7-9. Heat exchanger sizes are given in Table 7-1.

TABLE 7-7

10-TON/200,000 BTU/HR
WATER-TO-R-11 HEATING/COOLING UNIT DESIGN CONDITIONS

	Cooling Mode Boiler	Heating Mode Evaporator
R-11 Side		
Q, Btu/hr	191,500	162,070
Flow, lb/hr	2135	2365
Inlet pressure, psia	86.4	9.6
Inlet temperature, °F	90.8	54
Inlet enthalpy, Btu/lb	25.7	30.8
Condensing/evaporating temperature, °F	185.3	54 at inlet; 50 at outlet
Latent heat, Btu/lb	66.7	67.8
Outlet pressure, psia	85.4	8.8
Subcooling/superheat, °F	4.6	5
Outlet temperature, °F	189.1	55
Outlet enthalpy, Btu/lb	115.4	99.33
Water Side		
Flow, lb/hr	36,000	36,000
Inlet temperature, °F	195	61
Outlet temperature, °F	189.7	56.5
Pressure drop, psia	4.0	4.2
Pressure, psia	TBD	TBD

TABLE 7-8

10-TON/200 KBTUH CONDENSER

Parameter	Units
<u>R-11 Side</u>	
Q, Btu/hr	311,600
Flow, lb/hr	3795
Inlet pressure, psia	21.6
Inlet temperature, °F	118.9
Inlet enthalpy, Btu/lb	107.6
Condensing temperature, °F	94.6
Outlet pressure, psia	21.4
Subcooling, °F	5
Outlet temperature, °F	89.6
Outlet enthalpy, Btu/lb	25.5
<u>Water Side</u>	
Flow, lb/hr	61,100
Inlet temperature, °F	[illegible]4.4
Outlet temperature, °F	89.5
Pressure drop, psia	5.0
Pressure, psia	TBD

TABLE 7-9

10-TON/200,000 BTU/HR
AIR COIL HEATING/COOLING UNIT DESIGN CONDITIONS

	Cooling Mode Evaporator	Heating Mode Condenser
R-11 Side		
Q, Btu/hr	119,600	205,300
Flow, lb/hr	1633	2440
Inlet temperature, °F	51	181
Inlet pressure, psia	9.0	31.2
Inlet enthalpy, Btu/lb	25.5	116.2
Evaporating/condensing temperature, °F	51 at inlet 47.9 at outlet	116
Outlet pressure, psia	8.4	31
Heat of vaporization, Btu/lb	72.89	74.7
Outlet enthalpy, Btu/lb	98.7	31
Pressure drop, psia	0.51	0.2
Superheat/subcooling, °F	3.1 (min)	5
Air Side		
Flow, lb/hr	18,000	22,500
Cfm	4,000	5,000
Inlet drybulb temperature, °F	78	70.6
Inlet wetbulb temperature, °F	67	--
Outlet temperature, °F	57.1 (sat)	108.6
Inlet pressure, psia	14.7	14.7
Pressure drop, in. H_2O	0.34	0.34

7.5 25-TON/600 KBTUH HEAT EXCHANGER

This system differs from the others in that it features a recirculating water loop to carry the heating/cooling effect from the heat pump to the terminal units.

7.5.1 Condenser

The condenser is a shell-and-tube unit with condensation occurring in the shell on the surface of low profile finned tubes. Water in the tube serves as the heat sink. Subcooling is achieved by providing additional water-cooled tubes submerged in the liquid refrigerant at the bottom of the shell.

The design conditions for this condenser are listed in Table 7-10. Table 7-3 shows the condenser characteristics.

7.5.2 Evaporator/Condenser

The flow configuration through this inner fin water-to-R-11 heat exchanger is similar to that of the smaller units described previously. Heating and cooling mode problem statements are given in Table 7-11. Evaporator characteristics are given in Table 7-2.

7.5.3 Boiler/Evaporator

The flow configuration through this inner fin water-to-R-11 heat exchanger is similar to that of the smaller units described previously. Design conditions are listed in Table 7-12, and characteristics are in Table 7-2.

7.5.4 Air-to-Water Terminal Unit Coil

Currently, it is assumed that this system will have 12 terminal units. The design requirements for each unit are listed in Table 7-13. The characteristics of this wavy fin coil are given in Table 7-1.

7.6 SINGLE TUBE HEAT TRANSFER TESTS

Generalized heat transfer and pressure drop data are available in the literature for the design of the various system R-11 phase change heat

TABLE 7-10

25-TON/600,000 BTU/HR
CONDENSER DESIGN CONDITIONS

	Cooling Mode
R-11 Side	
Q, Btu/hr	717,900
Flow, lb/hr	8800
Inlet pressure, psia	21.6
Inlet temperature, °F	115.4
Condensing temperature, °F	94.5
Inlet enthalpy, Btu/lb	107.1
Outlet pressure, psia	21.4
Outlet enthalpy, Btu/lb	25.5
Subcooling, °F	5
Water Side	
Flow, lb/hr	143,600
Inlet temperature, °F	84.4
Outlet temperature, °F	89.45
Pressure drop, psia	5.0 (max)
Pressure, psia	TBD

TABLE 7-11

25-TON/600,000 BTU/HR
EVAPORATOR/CONDENSER* DESIGN CONDITIONS

	Cooling Mode Evaporator	Heating Mode Condenser
R-11 Side		
Q, Btu/hr	299,420	603,800
Flow, lb/hr	4096	7085
Inlet temperature, °F	51	190.7
Inlet pressure, psia	9.1	34.4
Inlet enthalpy, Btu/lb	25.5	117.5
Saturation temperature, °F	51 at inlet; 47.9 at outlet	121.5
Outlet enthalpy, Btu/lb	98.7	32.3
Outlet pressure, psia	8.4	34.2
Pressure drop, psi	0.6	0.2
Superheat/subcooling, °F	3.1	5
Water Side		
Flowrate, lb/hr	37,500	37,500
Inlet temperature, °F	60	102.3
Outlet temperature, °F	52	118.4
Inlet pressure, psia	TBD	TBD
Pressure drop, psi	3.6	3.2

* Dual mode in Nashville only

TABLE 7-12

25-TON/600 KBTUH
BOILER/EVAPORATOR

	Cooling Mode Boiler	Heating Mode Evaporator
<u>R-11 Side</u>		
Q, Btu/hr	413,200	461,800
Flow, lb/hr	4656	6940
Inlet pressure, psia	87	9.27
Inlet temperature, °F	90.8	53.4
Evaporating/boiling temperature, °F	186.5 inlet; 185.0 outlet	53.4 inlet; 48.3 outlet
Inlet enthalpy, Btu/lb	25.8	32.3
Outlet pressure, psia	86	8.67
Outlet enthalpy, Btu/lb	115.5	98.84
Outlet temperature	189.6	51.2
Subcooling, °F	4.6	2.9
<u>Water Side</u>		
Flow, lb/hr	80,000	80,000
Inlet temperature, °F	195	62
Outlet temperature, °F	189.8	56.2
Pressure drop, psia	3.8	4.2

TABLE 7-13

25-TON/600,000 BTU/HR
AIR HEATER/COOLER DESIGN CONDITIONS
(12 Required)

	Heating Mode	Cooling Mode
Water Side		
Q, Btu/hr	50,320	24,952
Flowrate, lb/hr	3125	3125
Inlet temperature, °F	118.6	52
Outlet temperature, °F	102.3	60
Pressure, psia	TBD	TBD
Pressure drop, psi	1.6	1.7
Air Side		
Flowrate, lb/hr	5,420	3,750
Flowrate, cfm	1,200	830
Inlet drybulb temperature, °F	70.5	78
Inlet wetbulb temperature, °F	N/A	67
Outlet temperature, °F	109.2	57.5 (sat)
Pressure, psia	14.7	14.7
Pressure drop, in. H_2O	0.29	0.28

exchangers. Initially these data were used for this purpose; however, because of the relatively high effectivenesses of the heat exchangers used in the heat pump subsystems, and also because of the sensitivity of the designs, single-tube heat transfer and pressure drop tests have been conducted on inner fin tubes of the sizes used in the design of the heat pump heat exchangers.

The purpose of these tests is to verify performance prediction and to ascertain the designs prior to fabrication. This approach is common practice in the design of high-efficiency heat exchangers and obviates costly iterations later in the program, as well as schedule slips. Appendix A describes the test program and presents the results obtained.

8. COOLING TOWER

Specifications for the cooling system cooling towers are essentially dependent on the actual capacity of the systems and on the wetbulb design temperatures for the installations considered. Table 8-1 summarizes the 2-1/2 percent design air temperatures for the geographical locations selected for the cooling systems. Also given are the performance requirements for the nominal 3-, 10-, and 25-ton systems.

Cooling tower data were obtained from Marley; they are presented in Exhibit 8A at the end of this section. Final selection will only be made following site analysis. All cooling towers will be off-the-shelf units; however, the possibility of reducing the power requirements of these cooling towers is being investigated with Marley.

TABLE 8-1

COOLING TOWER PERFORMANCE REQUIREMENTS

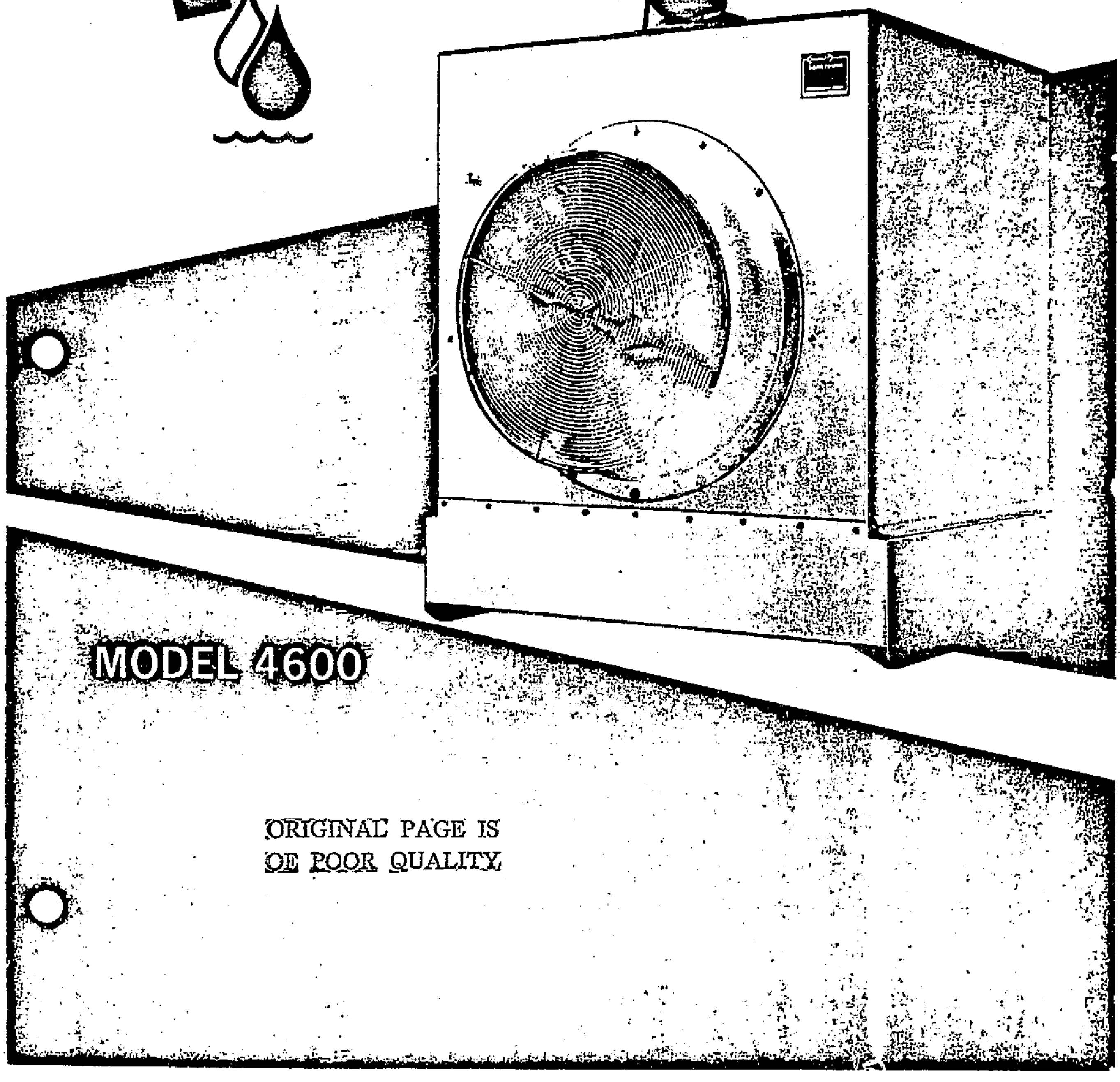
System capacity, tons	3		10		25	
Location	Des Moines, Iowa	Washington, D.C.	Los Angeles, Calif.	St. Louis, Missouri	Las Vegas, Nevada	Houston, Texas
Design outside temperature						
Drybulb, °F	92	92	90	94	106	92
Wetbulb, °F	77	77	70	78	71	80
Heat rejection rate, Btu/hr	102,700		311,600		717,900	
Tower inlet water temp., °F	89.4		89.5		89.4	
Tower outlet water temp, °F	84.4		84.4		84.4	
Water flow, gpm	41		122		287	

EXHIBIT 8A

MARLEY COOLING TOWER DATA

MARLEY

aquatower®



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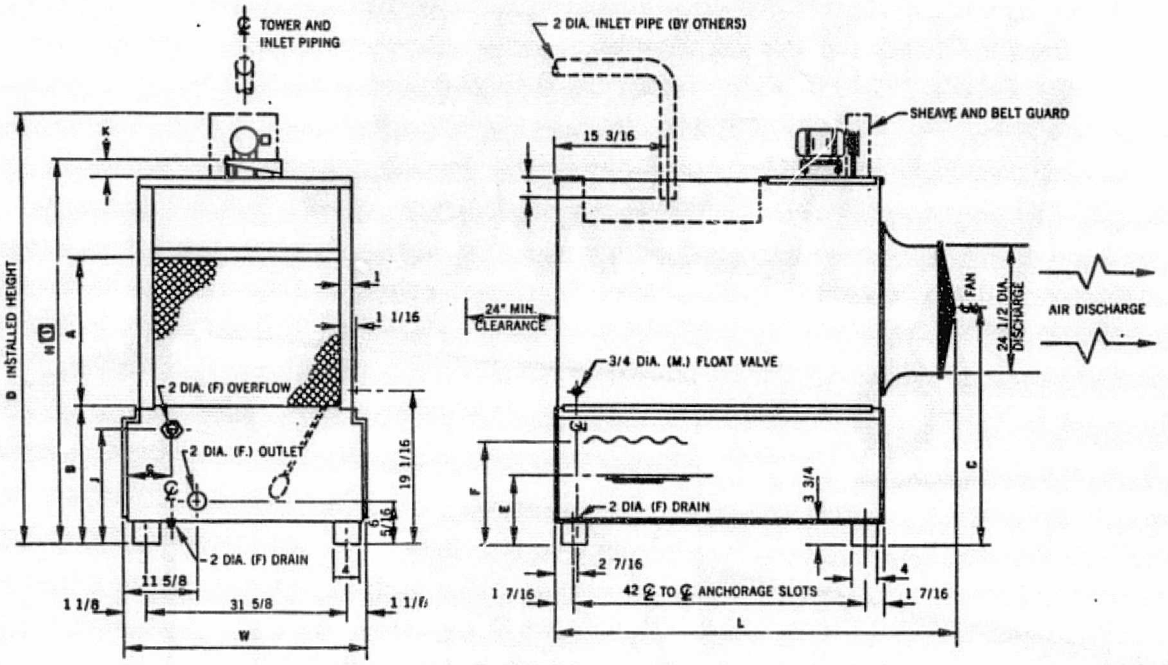
AIRESEARCH MANUFACTURING COMPANY
OF CALIFORNIA

MODELS 4613 THRU 4617



Tower Model No.	Shipping Dimensions			Dimensions in inches			
	W	L	H	A	B	C	D
4613	33 7/8	61 1/16	48 5/16	17 5/16	13 3/4	28 13/16	61 5/8
4615	33 7/8	61 1/16	57 3/8	22 13/16	15 7/16	30 1/2	68 13/16
4617	33 7/8	61 1/16	68 1/4	33 13/16	15 7/16	32 1/8	79 13/16

Tower Model No.	Fan RPM	Pump Head In Feet [3]	G.P.M.*		Shipping Weight	Operating Weight	Water Level		Overflow Location		K
			Min.	Max.			E Operating	F Overflow	G	J	
4613	765	4.5	16	25	365	725	7 3/4	12 1/2	6 13/16	10 7/8	3 1/2
4615	668	5.0	16	27	410	790	7 3/4	13 13/16	6 13/16	12 5/16	5
4617	668	6.5	23	40	445	840	7 3/4	13 13/16	6 13/16	12 5/16	5

GENERAL NOTES

[1] 'H' dimension is to top of motor base (which ships installed).

2. Motor and belt guard are shipped separate and require field installation.

[3] Pump head based on the following:
   A. Diameter of riser and inlet are equal.
   B. Total length of piping is static lift (H-4") + 5'-0".
   C. Two 90° elbows—same diameter as inlet.
   D. New steel pipe (by others).

*Based on standard nozzles. Water rates outside of these limitations require different nozzles.

GARRETT
AIRESEARCH MANUFACTURING COMPANY
OF CALIFORNIA

Performance Data

4600 AQUATOWER®

Tons of Refrigeration at 250 BTU/MIN/TON

Tower Model	Rated Tons	AT 3 GPM/TON						AT 4 GPM/TON								
		86	87	90	88	90	91	82.5	85	89	89.5	95	93.5	95	95	97
		76	77	80	78	80	81	75	77.5	81.5	82	87.5	86	87.5	87.5	89.5
		66	67	67	68	68	71	65	70	70	72	78	78	79	80	80
4613	5	5	5.2	7.3	5.4	6.8	6.0	5.0	4.1	6.9	6.4	7.3	5.9	6.6	5.8	7.9
4615	8	8.1	8.3	11.6	8.6	10.9	9.6	8.0	6.6	11.1	10.2	11.7	9.5	10.5	9.4	12.6
4617	10	10	10.3	14.0	10.6	13.1	11.7	9.6	8.1	13.2	12.0	13.8	11.3	12.5	11.2	14.7
4619	15	15	15.5	21.0	16.0	19.6	17.6	14.3	12.1	19.8	18.0	20.7	17.0	18.8	16.9	22.0
4621	20	20	20.6	28.0	21.3	26.2	23.5	19.1	16.2	26.4	24.1	27.5	22.7	25.0	22.5	29.4
4623	25	25	25.8	35.0	26.6	32.8	29.4	23.9	20.3	33.0	30.1	34.4	28.4	31.3	28.1	36.7
4625	30	30	31.0	42.0	32.0	39.3	35.3	28.7	24.3	39.7	36.1	41.3	34.1	37.6	33.8	44.1
4627	40	40	41.3	56.0	42.6	52.4	47.0	38.3	32.4	52.9	48.1	55.1	45.4	50.1	45.0	58.8
4629	50	50	51.7	70.1	53.3	65.6	58.8	47.9	40.5	66.1	60.2	68.9	56.8	62.6	56.3	73.5
4631	60	60	62.0	84.1	64.0	78.7	70.6	57.4	48.6	79.3	72.2	82.6	68.2	75.1	67.5	88.2
4633	75	75	77.5	105.1	80.0	98.4	88.2	71.8	60.8	99.2	90.3	103.3	85.2	93.9	84.4	110.2

GARRETT
AIRESEARCH MANUFACTURING COMPANY
OF CALIFORNIA

Performance Data

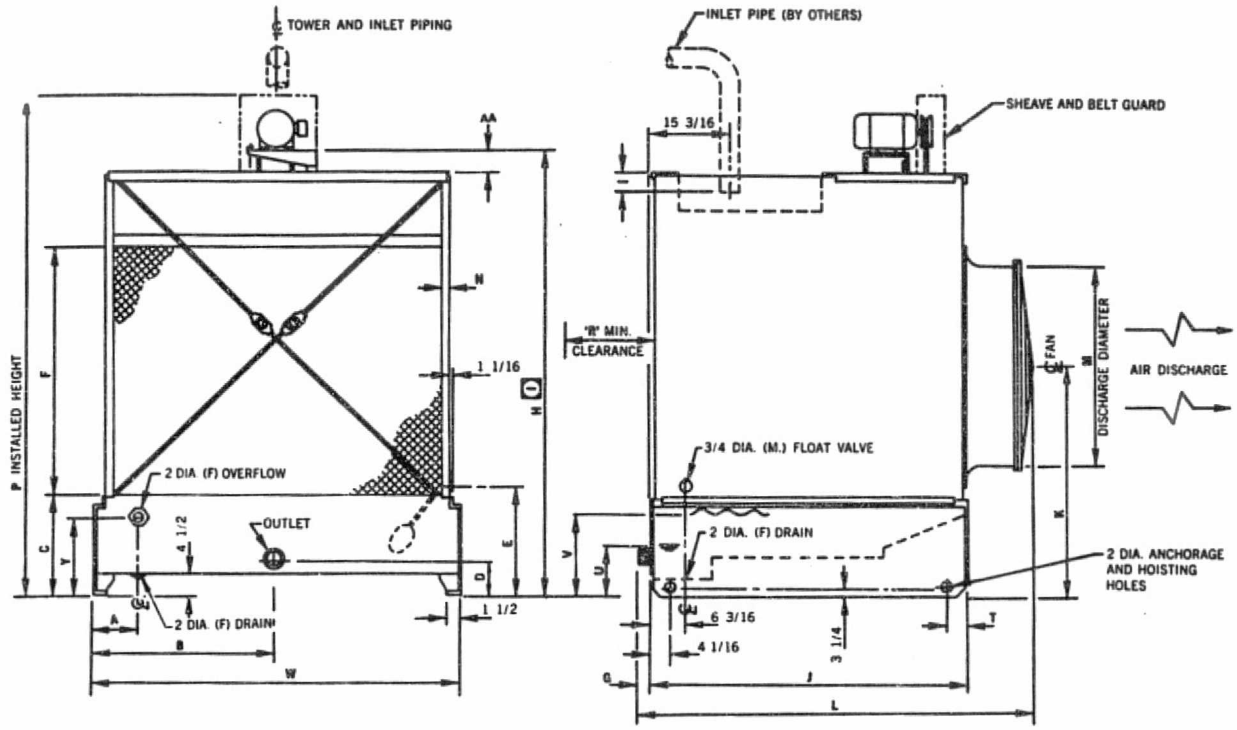
4600 AQUATOWER®

Tons of Refrigeration at 250 BTU/MIN/TON

AT 3 GPM/TON

Tower Model	Rated Tons	90 80 65	95 85 70	90 80 70	87 77 70	95 85 72	92 82 72	96 86 73	97 87 75	95 85 75	95 85 77	97 87 78	95 85 78	96 86 79	96 86 80	97 87 80
4613	5	8.2	9.8	5.8	3.8	8.7	6.2	9.0	8.7	6.9	5.7	6.8	5	5.2	4.5	5.4
4615	8	13.2	15.7	9.3	6.0	13.9	10.0	14.4	13.9	11.1	9.1	10.8	8	8.3	7.2	8.6
4617	10	15.5	18.3	11.4	7.7	16.5	12.1	17.0	16.5	13.4	11.2	13.1	10	10.3	9.1	10.7
4619	15	23.3	27.5	17.1	11.5	24.7	18.2	25.5	24.7	20.1	16.8	19.7	15	15.5	13.7	16.0
4621	20	31.1	36.6	22.8	15.4	33.0	24.3	34.0	3[illegible].0	26.8	22.4	26.2	20	20.6	18.2	21.3
4623	25	38.9	45.8	28.5	19.2	41.2	30.4	42.6	41.2	33.5	28.0	32.8	25	25.8	22.8	26.7
4625	30	46.6	55.0	34.2	23.1	49.4	36.5	51.1	49.5	40.2	33.6	39.4	30	31.0	27.3	32.0
4627	40	62.2	73.3	45.6	30.8	65.9	48.6	68.1	66.0	53.7	44.8	52.5	40	41.3	36.5	42.7
4629	50	77.7	91.6	57.0	38.5	82.4	60.8	85.1	82.5	67.1	56.0	65.6	50	51.6	45.6	53.4
4631	60	93.3	110.0	68.4	46.2	98.9	73.0	102.2	99.0	80.5	67.2	78.7	60	62.0	54.7	64.1
4633	75	116.6	137.4	85.5	57.8	123.6	91.2	127.7	123.7	100.6	84.0	98.4	75	77.5	68.4	80.1

MODELS 4619 THRU 4633



Tower Model No.	Shipping Dimensions			Dimensions in Inches														
	W	L	H	A	B	C	D	E	F	G	J	K	M	N	P	R	T	AA
4619	33⅞	64 3/16	79	5 5/16	9 11/16	17⅞	7 1/16	21⅛	42	X	48	38⅛	24½	1	90 5/16	24	4 1/16	5
4621	47⅞	68 3/16	77⅜	9 1/16	23 13/16	17⅞	7½	20 13/16	42	2 3/16	52	39¾	36¾	2	90 5/16	48	4 1/16	3½
4623	47⅞	68 3/16	77⅜	9 1/16	23 13/16	17⅞	7½	20 13/16	42	2 3/16	52	39¾	36¾	2	90 5/16	48	4 1/16	3½
4625	57⅞	72 3/16	77⅜	9 1/16	28 13/16	17⅞	7½	20 13/16	42	2 3/16	56	39¾	36¾	2	90 5/16	72	4 1/16	3½
4627	64	75 3/16	92 11/16	9⅛	32	20⅞	7½	23 13/16	52 13/16	2 3/16	57 13/16	47 13/16	42¾	2	104¼	72	4⅛	5
4629	76	81½	91 3/16	9⅛	3[illegible]	20⅞	7½	23 13/16	52 13/16	2 3/16	63 13/16	49 1/16	42¾	2	104¼	72	4⅛	3½
4631	88	85⅛	92 11/16	9⅛	44	20⅞	8½	23 13/16	52 13/16	2 3/16	66 13/16	49 5/16	48¾	2	104¼	72	4⅛	5
4633	100	85⅛	91 3/16	9⅛	50	20⅞	8½	23 13/16	52 13/16	2 3/16	66 13/16	49 5/16	48¾	2	104¼	72	4⅛	3½

Tower Model No.	Piping Layout Data - Inches		Fan RPM	Pump Head in Feet ③	GPM*		Shipping Weight	Operating Weight	Water Level		Y
	Inlet Piping By Others	Cold Water Outlet			Min.	Max.			U Operating	V Overflow	
4619	2	2F	850	9.0	34	59	505	860	9⅝	16¼	14⅝
4621	4	4M	478	6.5	48	83	625	1140	9⅝	16¼	14⅝
4623	4	4M	565	6.5	48	83	630	1155	9⅝	16¼	14⅝
4625	4	4M	634	6.5	62	106	795	1440	9⅝	16¼	14⅝
4627	4	4M	480	8.5	93	160	1070	1950	12⅝	19¼	17⅝
4629	4	4M	571	9.0	112	192	1320	2380	12⅝	19¼	17⅝
4631	6	6M	504	8.0	130	224	1420	2660	12⅝	19¼	17⅝
4633	6	6M	590	8.0	149	256	1590	3020	12⅝	19¼	17⅝

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GENERAL . . . Heavy gauge steel is used throughout to fabricate Aquatower basins and casings. Towers can easily be disassembled for special handling or installation in confined areas.

FILLING . . . High efficiency, film type cooling is provided by the use of non-combustible asbestos fill.

GRAVITY WATER DISTRIBUTION . . . Special target nozzles provide distribution over the fill sheets.

MECHANICAL EQUIPMENT . . . All models are V-belt driven. External lubrication lines are provided as standard equipment on all models. Motors on the 4613 through 4623 are totally enclosed. Models 4625 through 4633 are provided with protected motors designed and tested specifically for cooling tower use.

BASIN FIXTURES . . . All models are equipped with bronze, automatic make up water control valves. Overflow, drain, suction fitting and suction screens are furnished.

CORROSION PROTECTION . . . All steel utilized in the manufacture of Aquatowers is galvanized, providing a finish on both interior and exterior which is recognized for durability and long maintenance-free service life.

MOTOR DATA AND FRAME SIZES

				50 and 60 Hertz		
				1-Phase	3-Phase	
Tower Model	Tons	H.P.	RPM** @ 60 Hertz	1-speed	1-speed	Drive
4613	5	⅓	1730	56	—	V-Belt
4615	8	⅓	1730	56	—	V-Belt
4617	10	⅓	1730	56	—	V-Belt
4619	15	½	1720	56	56	V-Belt
4621	20	½	1720	56	56	V-Belt
4623	25	¾	1720	56	56	V-Belt

NOTE:
Single Phase Motors (1½ HP and larger) require 16 week shipping delay. Protected Single Phase Integral Motors are not available from Marley. Can be furnished in T.E. only.
**Approximate full load R.P.M.

MOTOR DATA AND FRAME SIZES

					60 Hertz 3-Phase			50 Hertz 3-Phase		60 Hertz 1-Phase	50 Hertz 1-Phase
Tower Model	Tons	H.P.	RPM** @ 60 Hertz	Drive	1 Speed	2-Speed 1-Winding	2-Speed 2-Winding	1 Speed	2-Speed 1-Winding	1 Speed	1 Speed
4625	30	1	1735	V-Belt	143T	143T	182T	143T	143T	143T	145T
4627	40	1	1735	V-Belt	143T	143T	182T	143T	143T	143T	145T
4629	50	1.5	1735	V-Belt	145T	145T	182T	145T	145T	145T	182T
4631	60	2	1730	V-Belt	145T	145T	182T	145T	182T	182T	184T
4633	75	3	1740	V-Belt	182T	182T	184T	182T	184T	184T	213T

We reserve the right to make changes in the dimensions and specifications without notice.

MARLEY

THE MARLEY COMPANY
5800 FOXRIDGE DRIVE
MISSION, KANSAS 66202

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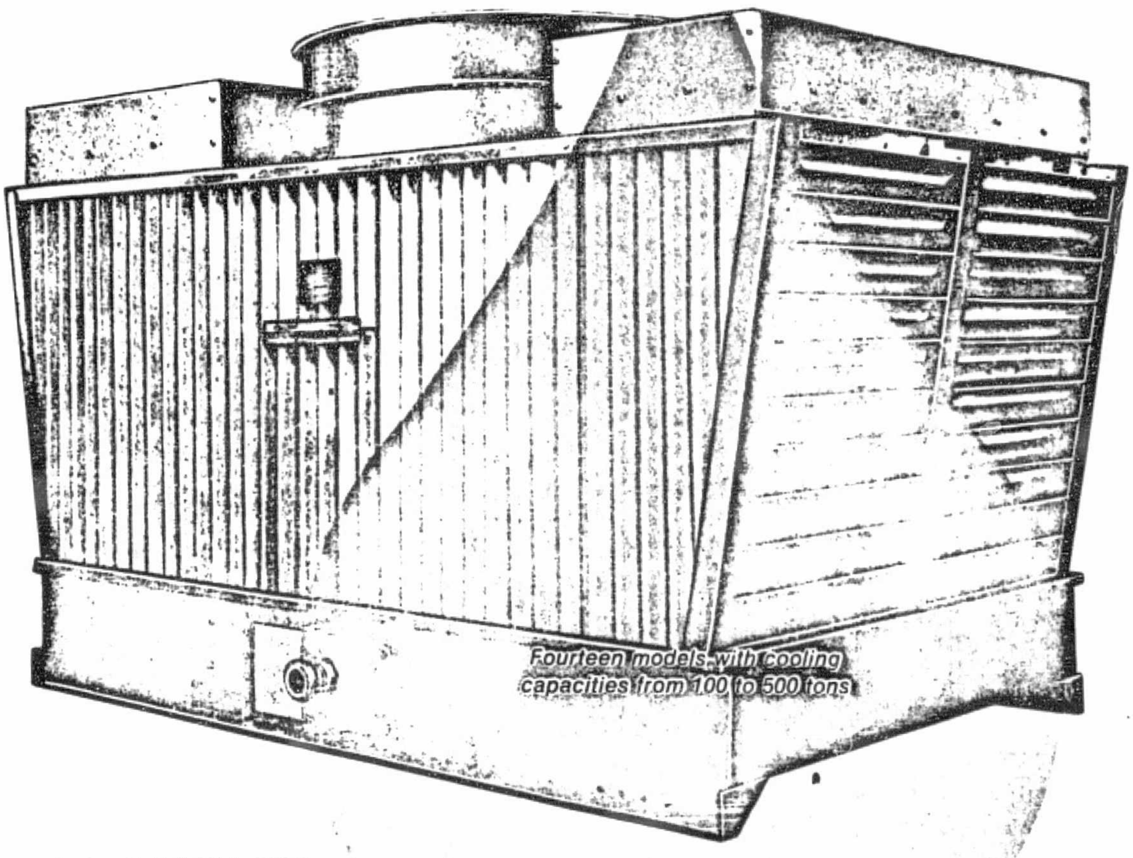
NC® MARLEY SERIES

DOUBLE FLOW® PACKAGED COOLING TOWERS



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High Performance
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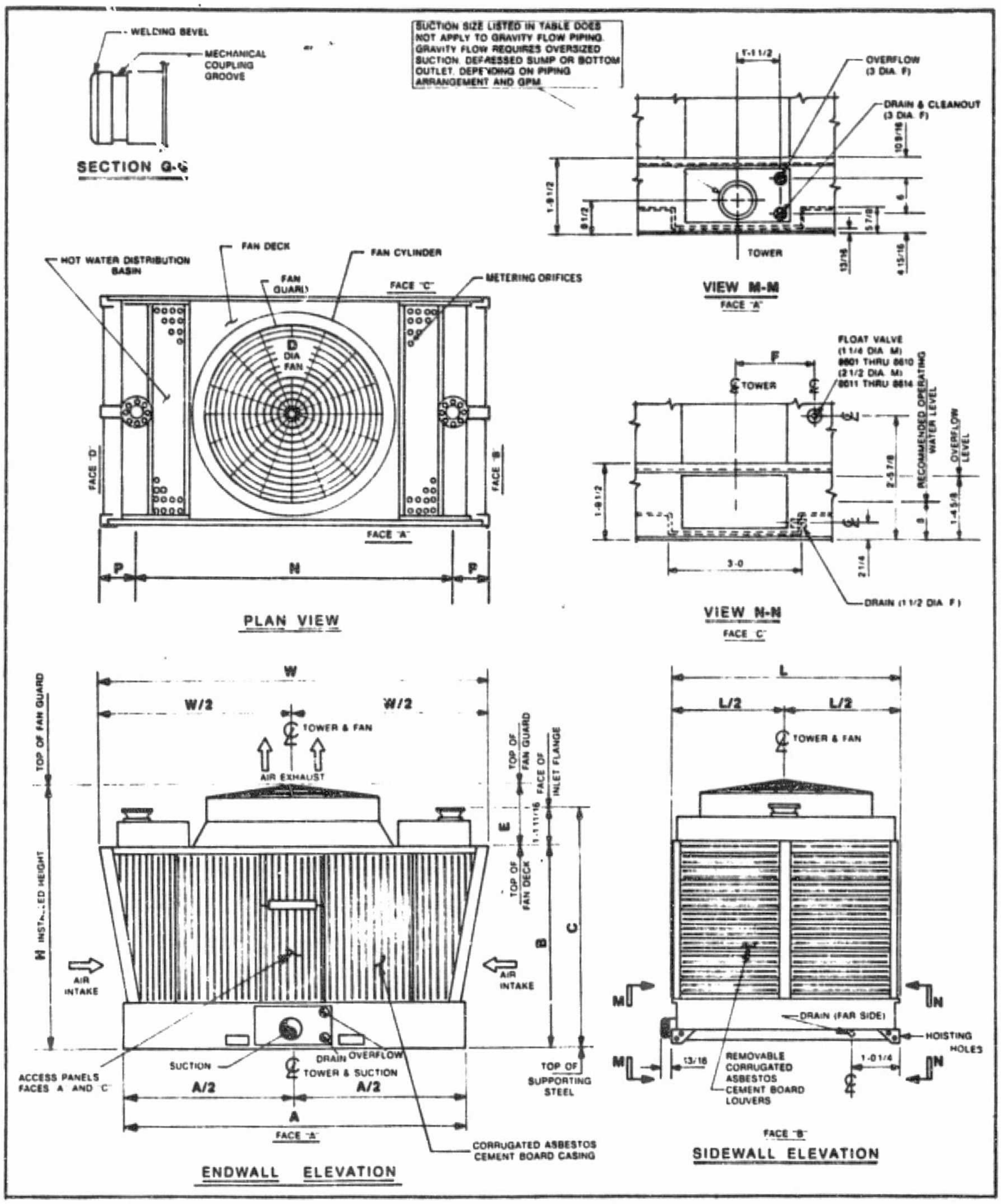
OF CALIFORNIA

DIMENSIONING DATA



2

TOWER MODEL	TOWER OVERALL (INSTALLED)			TOWER DIMENSIONS						PIPING DATA							
	L	W	H	A	B	C	D	E	F	[illegible]	[illegible]	Drain & Cleanout	Float Valve	[illegible]	[illegible]	P	[illegible]
8601	5'-4½	12'-0	6'-10⅝	11'-1⅝	5'-2⅜	6'-4⅛	54	1'-8½	1'-8⅛	6	3F	3F	1¼ M	4	9'-7⅜	1'-2⅜	5
8602	7'-3½	12'-0⅛	8'-2⅜	10'-10¼	6'-1⅛	7'-2⅞	72	2'-1¼	1'-7⅜	6	3F	3F	1¼ M	4	9'-9	1'-1⅜	5
8603	7'-3½	12'-0⅛	8'-2⅜	10'-10¼	6'-1⅛	7'-2⅞	72	2'-1¼	1'-7⅜	6	3F	3F	1¼ M	4	9'-9	1'-1⅜	7½
8604	7'-3½	13'-7⅜	8'-2⅜	12'-5¾	6'-1⅛	7'-2⅞	72	2'-1¼	1'-7⅜	8	3F	3F	1¼ M	6	11'-4½	1'-1⅜	7½
8605	7'-3½	13'-7⅜	8'-2⅜	12'-5¾	6'-1⅛	7'-2⅞	72	2'-1¼	1'-7⅜	8	3F	3F	1¼ M	6	11'-4½	1'-1⅜	10
8606	7'-10½	13'-7⅜	9'-1⅜	12'-1⅞	7'-0⅛	8'-1⅞	72	2'-1¼	1'-7⅜	8	3F	3F	1¼ M	6	11'-4½	1'-1⅜	10
8607	7'-10½	15'-3⅛	9'-1⅜	13'-9⅜	7'-0⅛	8'-1⅞	72	2'-1¼	1'-7⅜	10	3F	3F	1¼ M	6	13'-0	1'-1⅜	10
8608	7'-10½	16'-7⅜	9'-11⅜	15'-1⅞	7'-0⅛	8'-1⅞	84	2'-11¼	1'-11¾	10	3F	3F	1¼ M	6	14'-9	0'-11⅜	15
8609	9'-4½	16'-7⅜	9'-11⅜	15'-1⅞	7'-0⅛	8'-1⅞	84	2'-11¼	1'-11¾	10	3F	3F	1¼ M	8	14'-7	1'-0⅜	15
8610	9'-4½	16'-7⅜	9'-11⅜	15'-1⅞	7'-0⅛	8'-1⅞	84	2'-11¼	1'-11⅜	10	3F	3F	1½ M	8	14'-7	1'-0⅜	20
8611	9'-4½	17'-7⅜	10'-10⅜	15'-10⅜	7'-10⅞	9'-0⅜	96	2'-11¼	2'-5	10	3F	3F	2½ M	8	15'-7⅛	1'-0⅜	20
8612	9'-4½	19'-3⅛	10'-10⅜	17'-6⅜	7'-10⅞	9'-0⅜	96	2'-11¼	2'-5	10	3F	3F	2½ M	8	17'-3⅛	1'-0⅜	20
8613	9'-4½	19'-3⅛	10'-10⅜	17'-6⅜	7'-10⅞	9'-0⅜	96	2'-11⅜	2'-5	10	3F	3F	2½ M	8	17'-3⅛	1'-0⅜	25
8614	9'-4½	19'-3⅜	12'-8⅜	16'-10⅜	9'-8⅜	10'-10¾	96	2'-11¼	2'-4¾	12	3F	3F	2½ M	8	17'-3⅜	1'-0⅜	25

Installation Notes

1. All external piping by others, including inlet pipes. These inlet pipes are to be located to discharge into the distribution box in each distribution basin.

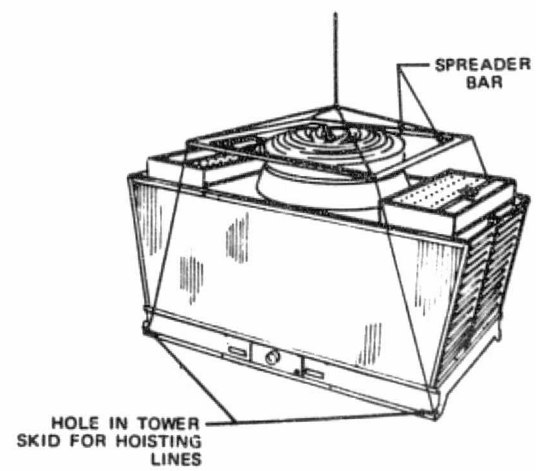
2. Piping on top of tower will be supported by tower structure. All other piping must not be supported by tower.

3. Fan, parallel shaft Geareducer and close coupled T.E.A.O. motor are mounted on steel support members, inside of tower.

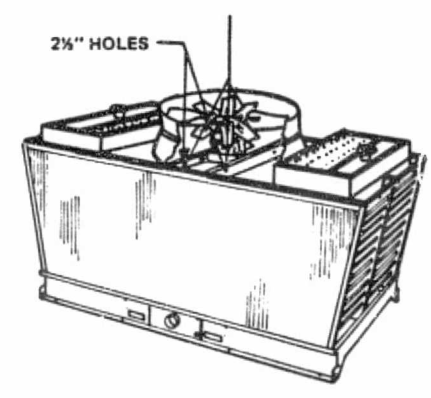
4. Wiring from motor conduit box to outside of tower by others.

5. If fan is cycled for capacity control, temperature limits should be set to prevent motor cycling more than 6 times per hour.

HOISTING INSTRUCTIONS

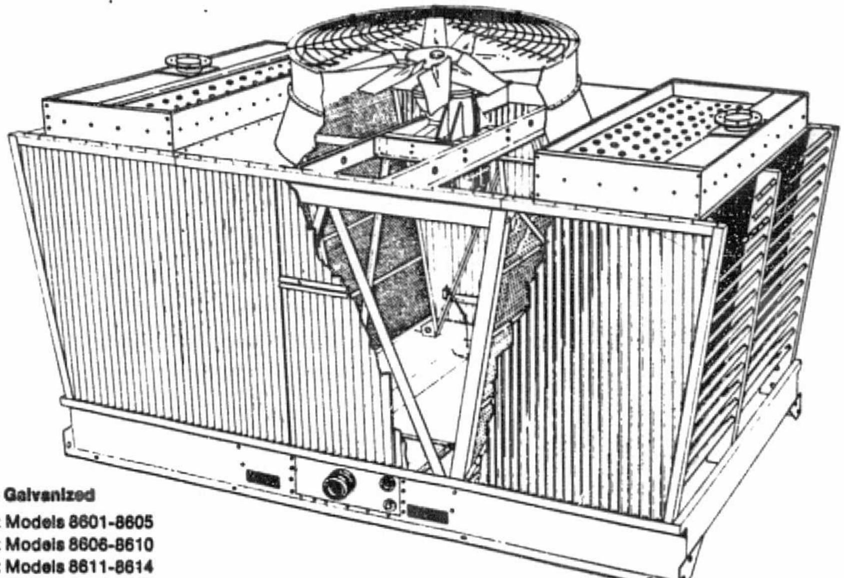


HOISTING FROM BOTTOM WITH SLINGS



2½" HOLES IN MECHANICAL EQUIPMENT SUPPORT MAY BE UTILIZED FOR HOISTING AND FINAL POSITIONING

ENGINEERING DATA



Cold water basin — Galvanized

Sides: 12 ga.: Models 8601-8605
10 ga.: Models 8606-8610
7 ga.: Models 8611-8614
Floor: 12 ga.: Models 8601-8605
10 ga.: Models 8606-8614

MODEL	8601	8602	8603	8604	8605	8606	8607	8608	8609	8610	8611	8612	8613	8614
† Nom. Tons	100	125	150	175	200	225	250	300	325	350	400	425	450	500
Length	5'-4½	7'-3½	7'-3½	7'-3½	7'-3½	7'-10½	7'-10½	7'-10½	9'-4½	9'-4½	9'-4½	9'-4½	9'-4½	9'-4½
Width	12'-0	12'-0⅛	12'-0⅛	13'-7⅝	13'-7⅝	13'-7⅝	15'-3⅛	16'-7⅝	16'-7⅝	16'-7⅝	17'-7⅝	19'-3⅝	19'-3⅝	19'-3⅝
Height	6'-10⅝	8'-2⅝	8'-2⅝	8'-2⅝	8'-2⅝	9'-1⅝	9'-1⅝	9'-11⅝	9'-11⅝	9'-11⅝	10'-10⅝	10'-10⅝	10'-10⅝	12'-8⅝
Shipping Wt.	3070	4050	4110	4480	4500	5370	5820	6850	7480	7530	8300	9060	9110	10050
Operating Wt.	6570	8720	8780	9820	9840	11040	12150	13770	15730	15780	16970	18560	186[illegible]	19220
*Motor H.P.	5	5	7½	7½	10	10	10	15	15	20	20	20	25	25
Fan Dia. & Type	54H3	72H3	72H3	72H3	72H3	72H3	72H3	84H3	84H3	84H3	96H3	96H3	96H3	96H3
RPM	666	547	547	547	547	547	547	426	426	426	426	426	426	426
CFM	29820	44450	50000	48630	54820	60180	58610	69050	76570	82630	91910	89590	94780	105270
Geareducer	11T	11T	11T	11T	11T	11T	11T	21.2T	21.2T	21.2T	21.2T	21.2T	21.2T	21.2T
Ratio/1 (60 Hertz)	2.65	3.20	3.20	3.20	3.20	3.20	3.20	4.105	4.105	4.105	4.105	4.105	4.105	4.105
Ratio/1 (50 Hertz)	2.23	2.65	2.65	2.65	2.65	2.65	2.65	3.22	3.22	3.22	3.22	3.22	3.22	3.22
GPM, Min.	100	110	120	125	175	190	190	200	200	210	240	250	250	280
GPM, Max.	500	625	750	875	960	1125	1250	1440	1625	1715	1880	2125	2160	2500
No. of Orifice/TWR	64	66	66	88	88	98	144	144	168	168	168	196	196	196
Suction (Beveled & Grooved)	6	6	6	8	8	8	10	10	10	10	10	10	10	12
Overflow (Female)	3	3	3	3	3	3	3	3	3	3	3	3	3	3
Drain (Female)	3	3	3	3	3	3	3	3	3	3	3	3	3	3
Float Valve (Male)	1¼	1¼	1¼	1¼	1¼	1¼	1¼	1¼	1¼	1¼	2½	2½	2½	2½
Inlet Size	4	4	4	6	6	6	6	6	8	8	8	8	8	8

Optional Equipment	
Air Inlet Screens	Field Installed # 2 Mesh x .063 Galv. Wire Screen # 16 Gage Metal "U" Edge Frame.
External Oil Sight Gage	Shop Installed — Glass Tube Brass Body Sight Gage is Installed Next to the Fill Pipe.
Vibration Isolation	Four Spring Isolators Under Tower. Marley Choice of Vendor. Vibration Isolation Under Mechanical Equipment Not Available. Two or more cells connected by equalizing flumes must be isolated beneath a common unitized support system. Consult your Marley Sales Engineer.
Basin Covers	⅜" Thick — Removable Corr. ACB Panels — Supported by Basin Sides. Top of Cover Flush With Top of Basin.
Flow Control Valves	Marley Flow Control Valves — Ship Separate — Bolt to Cast Iron Inlet Adapter.
Ladder & Handrail Field Installed	Ladder is 1'-6 wide with 1¼ Dia. pipe rails & ¾ Dia. steel rungs. Handrail is 3'-6 high above fan deck with 1¼ Dia. pipe rails & posts & 14 Ga. toe plates.
External Lube Line With Dip Stick	Stand pipe/dip stick assembly allows checking geareducer oil level & filling from outside tower.

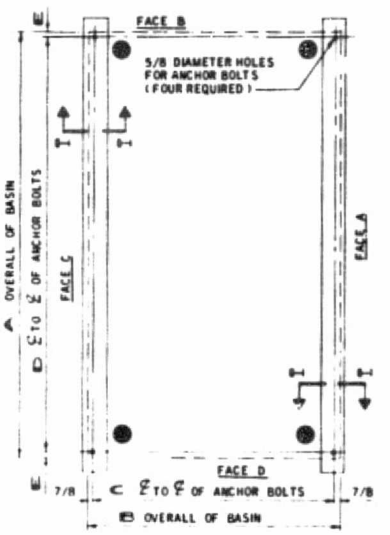
*Standard voltages are 200 volts or 230/460 volts
Two speed motors are one voltage only (200 volts, or 460 volts)
When two speed motors are used a time delay (20 seconds min.) must be provided by others when switching from high to low speed.

† 95°—85°—78° WB, 3 GPM/ton

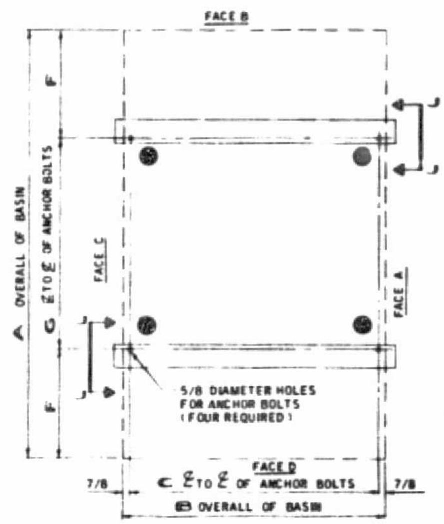
"We reserve the right to make changes in the dimensions and specifications without notice."

Some models are available for shipment in all Stainless Steel construction. Check with your Marley Sales Engineer.

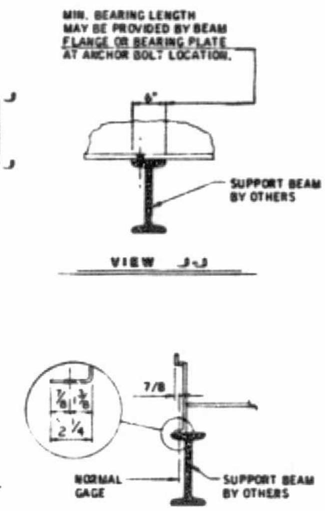
SUPPORTING STEEL DATA



PLAN OF SUPPORTING STEEL (RECOMMENDED)



PLAN OF SUPPORTING STEEL (ALTERNATE)



SECTION H-H

TOWER MODEL	TOWER DIMENSIONS							SHP. WEIGHT (LBS.)	MAX. OPERAT. WT. (LBS.)*	MAX. DEAD LOAD AT ● (LBS.)	MAX. LIVE LOAD AT ●
	A	B	C	D	E	F	G				
8601	11'-1⅛	5'-4½	5'-2¾	10'-7½	3¼	1'-11⅝	7'-2¾	3070	6570	1645	580
8602	10'-10¼	7'-3½	7'-1¾	10'-4½	2⅞	1'-10⅝	7'-2¾	4050	8720	2180	580
8603	10'-10¼	7'-3½	7'-1¾	10'-4½	2⅞	1'-10⅝	7'-2¾	4110	8780	2195	580
8604	12'-5⅝	7'-3½	7'-1¾	12'-0	2⅞	2'-7⅜	7'-2¾	4480	9820	2455	660
8605	12'-5⅝	7'-3½	7'-1¾	12'-0	2⅞	2'-7⅜	7'-2¾	4500	9840	2460	660
8606	12'-1⅛	7'-10½	7'-8¾	11'-8½	2¾	2'-7⅝	6'-10¼	5370	11040	2760	770
8607	13'-9⅝	7'-10½	7'-8¾	13'-4	2¾	3'-5⅝	6'-10¼	5820	12150	3040	860
8608	15'-1⅛	7'-10½	7'-8¾	14'-8½	2¾	3'-9⅜	7'-6½	6850	13770	3445	990
8609	15'-1⅛	9'-4½	9'-2¾	14'-8½	2¾	3'-9⅜	7'-6½	7480	15730	3935	850
8610	15'-1⅛	9'-4½	9'-2¾	14'-8½	2¾	3'-9⅜	7'-6½	7530	15780	3945	850
8611	15'-10⅝	9'-4½	9'-2¾	15'-5	2¾	3'-11⅝	7'-10⅝	8300	16970	4245	1110
8612	17'-6⅝	9'-4½	9'-2¾	17'-1	2¾	4'-4⅛	8'-8⅜	9060	18560	4640	1220
8613	17'-6⅝	9'-4½	9'-2¾	17'-1	2¾	4'-4⅛	8'-8⅜	9110	18610	4655	1220
8614	16'-10⅝	9'-4½	9'-2¾	16'-5¼	2¾	4'-2⅜	8'-4⅜	10050	19220	4805	1710

*SEE NOTE 2

GENERAL NOTES

1. SUPPORTING STEEL: Purchaser to design, construct and furnish supporting steel complete with 5/8 diameter holes for anchor bolts to suit the general dimensions of this drawing. All steel must be framed flush and level at top. Maximum beam deflection to be 1/360 of span, not to exceed 1/2 inch.

2. TOWER DEAD LOADS: Maximum wet operating weights and dead loads are based on water level in cold water basin at 1'-4 5/8 above top of supporting steel. This is the cold water basin overflow level.

3. TOWER LIVE LOADS: Wind loads are calculated on a basis of thirty pounds per square foot. Live loads are additive to dead loads.

4. ANCHOR BOLTS: All anchor bolts are 1/2 inch diameter and are to be furnished by others.

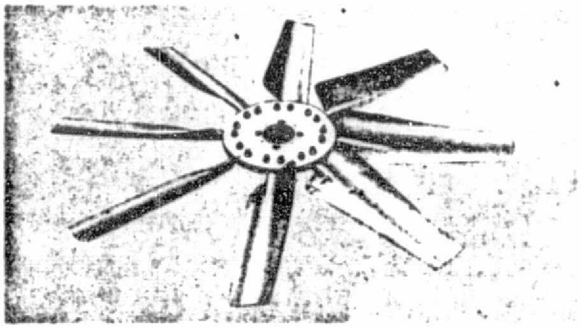
5. ALTERNATE BEAM LOCATION: If it is desired to place beams at a different location than shown, dimension F may be less, but not greater than tabulated.

6. PIER SUPPORTS: Tower may be supported from piers at the four anchor bolt locations if desired.

7. TOWER OBSTRUCTIONS: Louvered walls, faces B and D must have adequate air supply. If obstructions exist nearby, consult a Marley sales engineer.

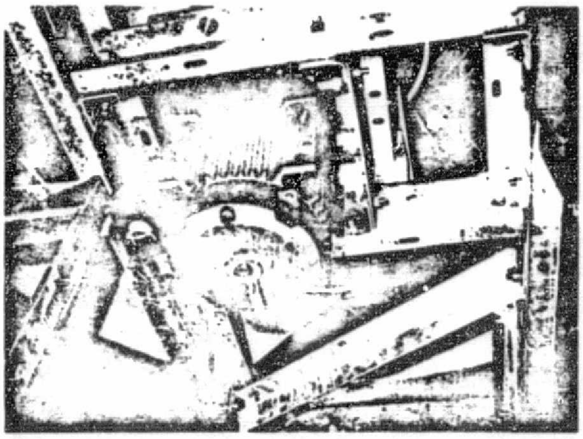
8. Consult your Marley Sales Engineer for proper application of towers subject to environmental factors which affect thermal capability.

NC FEATURES



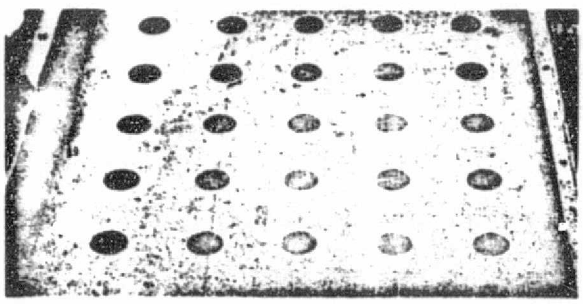
FAN

Fans used in the NC series are designed by Marley specifically for cooling tower duty. They are built for smooth, quiet operation and dependability against higher static pressures of high performance film-type fill materials. Adjustable pitch blades allow maximum utilization of rated horsepower. Blades are corrosion resistant solid cast aluminum alloy — hubs are aluminum.



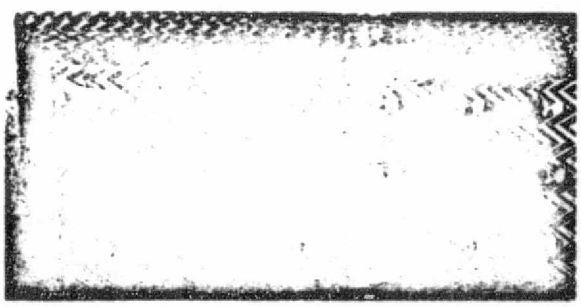
MECHANICAL EQUIPMENT

Heavy duty mechanical equipment and supports include a Marley-designed, parallel shaft, helical gear reduction unit. Large horsepower capacity and high speed reduction ratios are applied with a safety factor that assures long operating service in the hot, moist, corrosive atmosphere of cooling towers. Entire system has been proven in thousands of installations for efficiency and low sound levels.



HOT WATER DISTRIBUTION SYSTEM

Is designed for minimum pumping head and long service life. Open gravity type basin is heavy mill galvanized steel with specially designed "target nozzle" metering orifices. These polypropylene inserts deliver required water distribution and are highly resistant to temperature and weathering damage. They eliminate water channeling, wash-out, and the need for a separate diffusion deck. Clogging and maintenance are minimized.

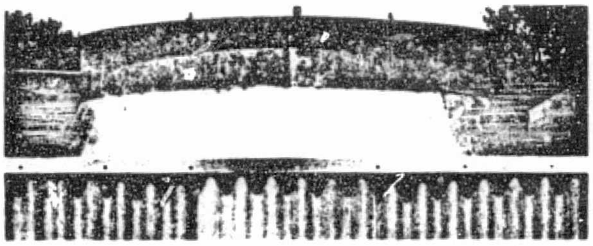


HIGH PERFORMANCE FILL

Polyvinyl chloride (PVC) fill has a flame spread rate of 25 (Steiner Tunnel Test per ASTM Standard E-84). Chevron configuration provides maximum air to water contact for highest thermal efficiency. Molded knob pattern assures constant, even spacing.

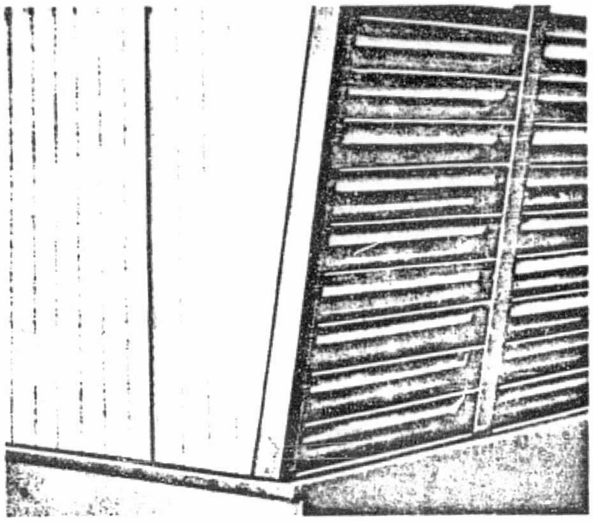
FAN CYLINDER

The hot dip galvanized steel fan cylinder is an "eased inlet" design for highest efficiency, allowing closer tolerances on fan tip clearance. The lower section of the two-piece cylinder and the fan deck are integral for higher structural integrity and better appearance.



CASING AND LOUVERS

Casing is vertically applied ⅜" corrugated asbestos cement board with lapped joints. Slipfit louvers are corrugated asbestos cement board, designed to properly direct the inlet air flow, minimize icing problems, and facilitate routine inspection and maintenance.

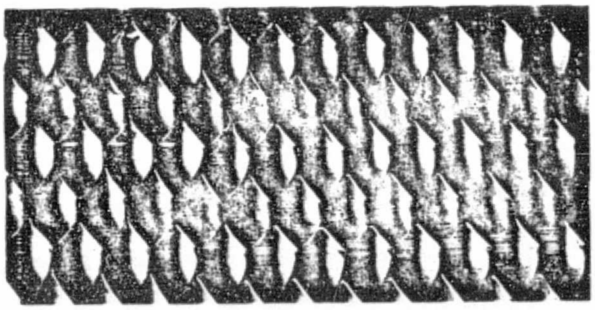


FLOW CONTROL VALVE

The Marley-built flow control valve has proven its reliability in over 20 years of application. Oversized valve body delivers best velocity dissipation for even water distribution. Heavy-duty cast iron body and stainless steel valve stem for long life and low maintenance.

HONEYCOMB DRIFT ELIMINATORS

Drift eliminators on the NC series towers are asbestos/neoprene in a special 2-pass honeycomb configuration designed to remove entrained moisture with minimum draft loss. Drift eliminator angle directs air into the plenum towards the fan, requiring less applied horsepower per CFM moved. Panel construction for easy fill inspection and lower maintenance costs.



SPECIFICATIONS

MARLEY
CORPORATE OFFICES
5800 FOXRIDGE DRIVE
MISSION, KANSAS 66202

GENERAL	Furnish and install Marley Model ____________ Induced Draft, Vertical Discharge, Double-Flow Cooling Tower(s). Tower dimensions shall be approximately ______ x ______ x ______ maximum height. Total horsepower requirement shall not exceed ____________ horsepower.
CAPACITY	The tower shall be guaranteed to cool ____________ GPM from ________°F to ________°F at a ________°F wet bulb.
CASING & LOUVERS	Casing shall be ⅜" corrugated asbestos cement board with lapped joints. Louvers shall be corrugated asbestos cement board of sufficient thickness to prevent sagging and shall be slipfit.
COLLECTION BASIN	Cold water basin shall be of one piece welded design with ______ ga. steel floor and ______ ga. steel sides. Basin shall be self-cleaning and complete with depressed center section, cleanout and drain fitting, side outlet sump with suction screen and anticavitation device and float operated make-up valve. The basin shall be designed to support the tower when resting on only two grillage beams.
DISTRIBUTION SYSTEM	Hot water distribution shall be of the open basin gravity type with plastic diffusing type metering orifices. Distribution basins shall be furnished with flanged connections suitable for direct piping connection or flow control valves as shown. Pump head shall not exceed feet, measured from base of tower.
FILLING & DRIFT ELIMINATORS	Fill and drift eliminator material shall be noncorrosive and nonferrous. Fill shall consist of vertical sheets of polyvinyl chloride plastic. Drift eliminators shall be two-pass asbestos neoprene honeycomb supported in steel frames. Drift loss shall be limited to 0.2 per cent.
HARDWARE & FINISH	All bolts, nuts and washers shall be galvanized steel. All steel shall be galvanized with minimum coating of 2¼ oz. per square foot.
MECHANICAL EQUIPMENT	There shall be ______ fan(s) of the propeller type with cast aluminum blades. Each blade shall be adjustable and individually attached to an aluminum hub. Fan drive(s) shall be through parallel shaft helical Geareducer. Motor(s) shall be flange mounted, Totally Enclosed, ______ HP, 1800 RPM, 3 phase 60 Hertz, ______ volts. A tapered hot dip galvanized steel fan cylinder shall be provided to minimize fan tip loss. V-belt drive will not be acceptable.
ACCESS & SAFETY	Access door shall be provided on both endwalls for access to the eliminator and plenum section. A 7 ga. hot dip galvanized wire grill type fan guard shall be provided over each fan cylinder.
SOUND	Reduced fan speed applications to meet restrictive sound level criteria are available.
WORK BY OTHERS	Supporting steel grillage, starting equipment and wiring shall be supplied by other contractors.

9. R-11 PUMP

9.1 GENERAL

A market survey has been made to determine the availability of hermetically sealed (canned motor) pumps suitable for handling R-11 at the flows and pressures shown in Table 9-1. The survey showed the following:

(a) Chempump has a hermetically sealed centrifugal pump that will meet the 3-ton, 10-ton, and 25-ton requirements; however, the overall efficiency would be less than 5 percent for the 25-ton unit and less than 1 percent for the 3-ton unit.

(b) Tuthill has an external gear pump with carbon bearings and sideplates that will meet the 3-ton unit requirement. This pump presently has a shaft seal that would have to be discarded, and the pump would have to be integrated with a hermetic motor. Efficiency of the Tuthill pump is 18 percent.

(c) There are several external gear pumps available, designed for lube oil service; they could be modified to include carbon-graphite bearings and sideplates and hermetic motors.

(d) Paco has a line of regenerative turbine pumps that could be modified to hermetic units. Pump efficiencies would be 13 percent, 20 percent, and 25 percent for the 3-, 10-, and 25-ton units respectively.

It was evident from the survey that it is essential to design, build, and develop three sizes of pumping units to meet the problem statements. Any intermediate solution, such as modifying existing pumps and motors for

TABLE 9-1

FREON R-11 PUMP REQUIREMENTS

Parameter	3-Ton Unit	10-Ton Unit	25-Ton Unit
Fluid	R-11	R-11	R-11
Inlet temperature, °F	89.5	89.6	89.5
Density, lb/cu ft	90.8	90.8	90.8
Specific gravity	1.45	1.45	1.45
Design flow, lb/hr	744	2135	4656
Design flow, gpm	1.02	2.93	6.39
Inlet pressure, psia	21.4	21.4	21.4
Outlet pressure, psia	120	120	120
Pressure rise, psi	98.6	98.6	98.6
Head, ft	156	156	156
Hydraulic hp	0.058	0.169	0.368
NPSP, psi	2.83	3.0	3.1
NPSH, ft	4.50	4.76	4.94
Power frequency, Hz	60	60	60
No. of phases	1	1	3
Voltage	115	115	208

operation on R-11 and hermetic sealing, would cost almost as much as a new pump design and would be a stop-gap solution at best.

9.2 SELECTION OF PUMP TYPE

Preliminary design tradeoffs have been made on four types of pumps--centrifugal, regenerative turbine, external gear, and sliding vane. In addition, the following pump types were briefly considered and then rejected for the reasons given:

Peristaltic (flexible tube)--Limited life

Internal gear--High galling probability with low lubricity fluids

Piston--Poor efficiency and high galling probability

Pilot--Poor efficiency and large rotating mass

Tables 9-2, 9-3, and 9-4 show the expected performance of the four types of pumps considered. The features of each type of pump are summarized in Table 9-5. The data show that a vane type pump is superior to the others, and the gear pump is a close second. The vane pump was selected for more detailed studies.

9.3 R-11 PUMP DESIGN

A program is proposed to design, build, and test three Freon R-11 pumps for the 3-, 10-, and 25-ton systems. All three pumps would be vane-type pumps of similar design, as shown in Drawing SK71756. The pumps would include a positive vane to cam ring tracking feature, which was the subject of a recent patent disclosure (Docket No. AL4153).

Because the 25-ton system has the most difficult requirements, it is proposed to build this pump first to show feasibility; it will be followed by the 3- and 10-ton designs.

TABLE 9-2

CENTRIFUGAL SINGLE-STAGE PUMP CHARACTERISTICS

Parameter	3-Ton Unit	25-Ton Unit
Speed, rpm	3400	3400
Specific speed, rpm	88	228
Suction specific speed, rpm	1050	2540
Impeller dia., in.	5.75	5.75
Pump efficiency, percent	6	28
Shaft hp	0.68	0.985
Rotor windage, hp	0.07	0.12
Air gap hp	0.75	1.105
Motor efficiency, percent	75	80
Input power, w	750	1030
Overall efficiency, percent	4	20

TABLE 9-3

REGENERATIVE TURBINE PUMP CHARACTERISTICS

Parameter	3-Ton Unit	25-Ton Unit
Speed*, rpm	1730	1730
Specific speed, rpm	44	114
Impeller dia., in.	4.2	4.2
Pump efficiency, percent	13	25
Shaft hp	0.315	1.104
Rotor windage, hp	0.05	0.10
Air gap hp	0.32	1.2
Motor efficiency, percent	75	80
Input power, w	318	1119
Overall efficiency, percent	10	18

* Insufficient NPSH to operate at 3400 rpm

TABLE 9-4

EXTERNAL GEAR OR SLIDING VANE PUMP CHARACTERISTICS

Parameter	3-Ton Unit	25-Ton Unit
Speed, rpm	1730	1730
Displacement, cu in/rev	0.121	0.818
Pump efficiency, percent	22	18
Shaft hp	0.186	0.575
Rotor windage, hp	0.05	0.07
Air gap hp	.236	0.645
Motor efficiency, percent	75	80
Input power, w	234	601
Overall efficiency, percent	13	34

TABLE 9-5

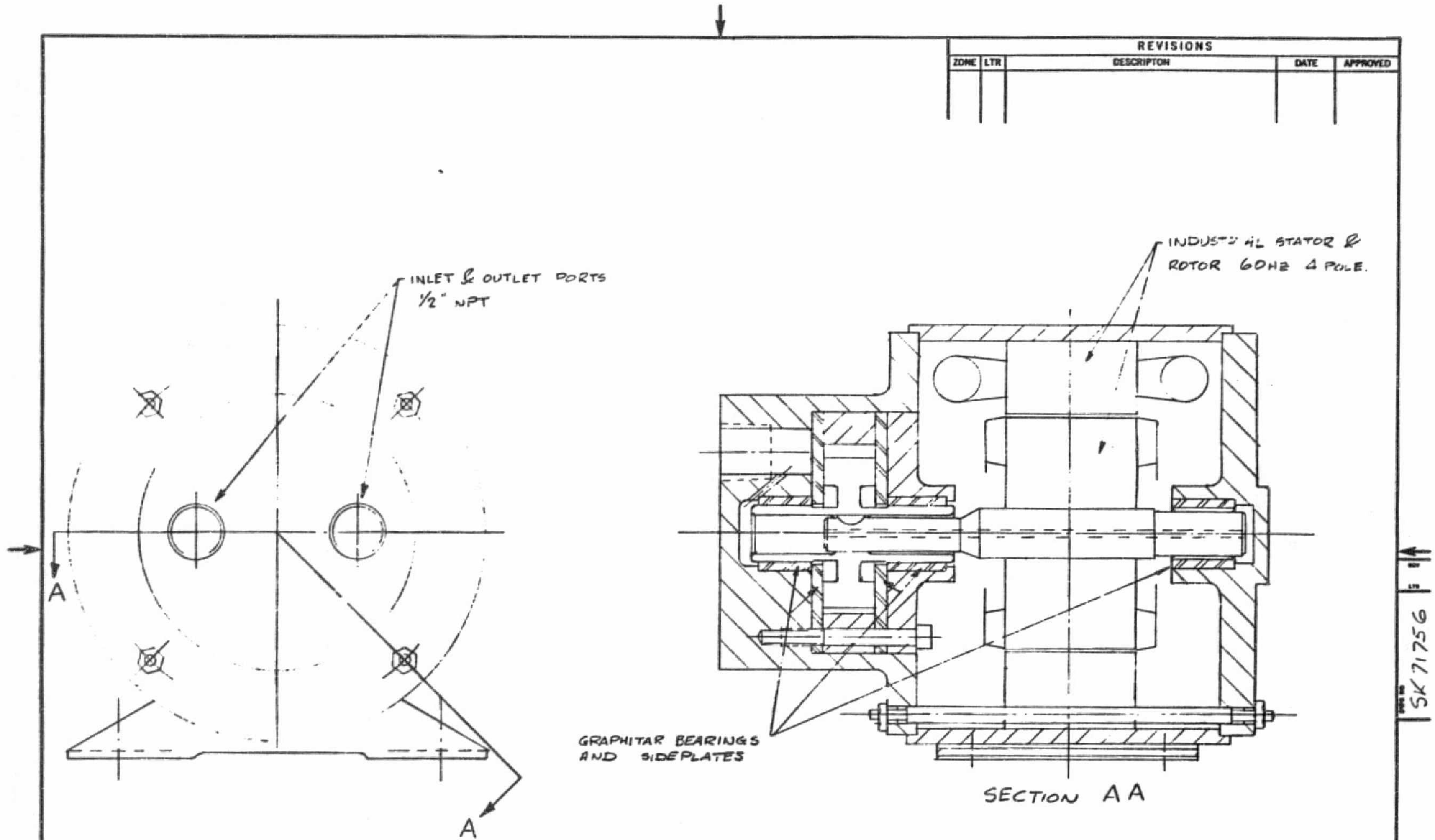
COMPARISON OF PUMP TYPES

Parameter	Desirability			
	Centrifugal	Regenerative Turbine	Gear on Gear	Vane
Efficiency	1	2	4	4
Size	2	3	4	4
Weight	2	3	4	4
Reliability/life	4	3	2	3
Development risk	4	3	2	2
Relative selling price, $ (100,000 quantity)	40	40	45	40

Legend:

4 = Most desirable characteristic

1 = Least desirable characteristic



PART NO

UNLESS OTHERWISE SPECIFIED:
BURR CONTROL PER SC653
STD INTERPRETATIONS PER P106
IDENTIFICATION MARKING PER MC16

MATERIAL

FINISH PROCESS

HEAT TREATMENT

CONTRACT NO

PREPARED A. Brown 11/17/72

CHK

DESIGN

VALUE ENGR

MATL

STRESS

APPD

DESIGN SUPERVISOR

PROJECT ENGINEER R Brown

GOVERNMENT APPD

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

PUMP, VANE

SIZE	CODE IDENT NO	DWG NO
C	70210	SK 71756

SCALE | SHEET | OF

REQD	NEXT ASSY	USED ON
APPLICATION		

Each program will be organized as follows.

(a) Purchase existing industrial motor, strip stator, and rewind, using insulation known to be compatible with Freon R-11.

(b) Cut fan off the rotor end rings and press out shaft.

(c) Design and fabricate a vane pump element using graphitar vanes and sideplates.

(d) Modify existing motor housing for hermetic sealing and incorporation of graphitar bearings.

APPENDIX

HEAT TRANSFER AND PRESSURE DROP MEASUREMENT FOR R-11 EVAPORATING IN DUNHAM-BUSH INNER FIN TUBES

APPENDIX

HEAT TRANSFER AND PRESSURE DROP MEASUREMENT FOR R-11 EVAPORATING IN DUNHAM-BUSH INNER FIN TUBES

INTRODUCTION

The single-tube heat transfer and pressure drop test described in the previous PDR data package (AiResearch report 76-12994) has been expanded in scope to accommodate changes in design approach as well as design conditions for the heat pump heat exchangers. In addition to horizontal tube evaporation, vertical tube evaporation was added in the test program as the latter emerges as a more viable choice for the phase-change heat exchangers. The number of test conditions was also increased substantially to obtain more reliable data in the neighborhood of design points. The test program was conducted in three parts, in the following sequence:

(1) Single-phase pressure drop test

(2) Two-phase pressure drop and heat transfer test--horizontal tube

(3) Two-phase pressure drop and heat transfer test--vertical tube

The single-phase pressure drop test was to establish experimentally the friction factor relationship for Dunham-Bush inner fin tubes. Such test was needed to confirm the friction factor curve recommended by Dunham-Bush and, at the same time, to analyze two-phase pressure drop on a basis consistent with that of the single-phase. Both the horizontal tube and the vertical tube two-phase flow tests were conducted using the test tubes selected from the single-phase test. The two-phase flow tests were conducted at several preselected design conditions for each heat exchanger so that the test results could be directly used either in design analysis or in confirmation of the predicted performance. Salient features of each test are described below.

TEST TUBE AND APPARATUS

Various parameters pertinent to the test tubes are summarized in Table A-1. Five tubes were tested for each model in the single-phase pressure drop test, and the one that was representative of the five was selected as the two-phase flow test tube.

An open system was used for both the single-phase and the two-phase flow tests to avoid the use of condensers, hence to simplify the control. Schematics of the test rig are shown in Figures A-1 and -2.

SINGLE-PHASE PRESSURE DROP TEST

Water was used as the working medium. Reynolds number considered ranges approximately from 200 to 10,000. Friction factor reduced from test data is shown in Figure A-3. The present results show generally higher values than those recommended by Dunham-Bush. A comparison of the present test results with an analytical calculation is shown in Figure A-4 for laminar flow region. The test results lie between the two analytical results*, supporting the validity of the test data. The best interpretations of the test results are shown in Figure A-5.

TWO-PHASE FLOW PRESSURE DROP AND HEAT TRANSFER TEST--HORIZONTAL TUBE

Low-Pressure Evaporation in 1/2-in.-OD Inner Fin Tube

The test conditions and the results derived are summarized in Table A-2. The instrument readings were quite stable in all the runs. Evaporation appears to be very smooth and no belching of liquid was observed at the tube exit when vapor quality reaches 95% or higher. Degree of superheat remained fairly steady when the exit vapor was superheated. Energy balance was satisfied within ±1%.

*The fully developed flow solution is that for a rectangular channel with aspect ratio of 3.8, the same aspect ratio for 1/2-in.-OD inner fin tube. The developing flow solution is based on the fully developed flow solution above and Langham's classic solution for circular tube (J. Appl. Mech., Vol. 9, June 1942).

TABLE A-1

SPECIFICATIONS OF TEST TUBES - DUNHAM-BUSH INNER FIN TUBE

	Model 12-14 (FT-45A)	Model 34-38 (CIC)
Outer tube	0.5" x 0.028", copper	0.75" x 0.035", copper
Inner tube	0.25" x 0.018", copper	0.375" x 0.028", copper
Test tube length, in.	40	60
Test tube L/D_n	1062	1060
Number of fins	36	44
Fin height, in.	0.09	0.147
Fin thickness, in.	0.008	0.008
Fin pitch, in.	1.125	1.125
Flow area, ft^2	0.000481	0.00135
Wetted area, ft^2/ft	0.613	1.14
Hydraulic diameter, ft	0.00314	0.00472
Free flow area/frontal area	0.655	0.769
Fin passage mean aspect ratio	3.8	6.2
Fin pitch/hydraulic diameter	30	20

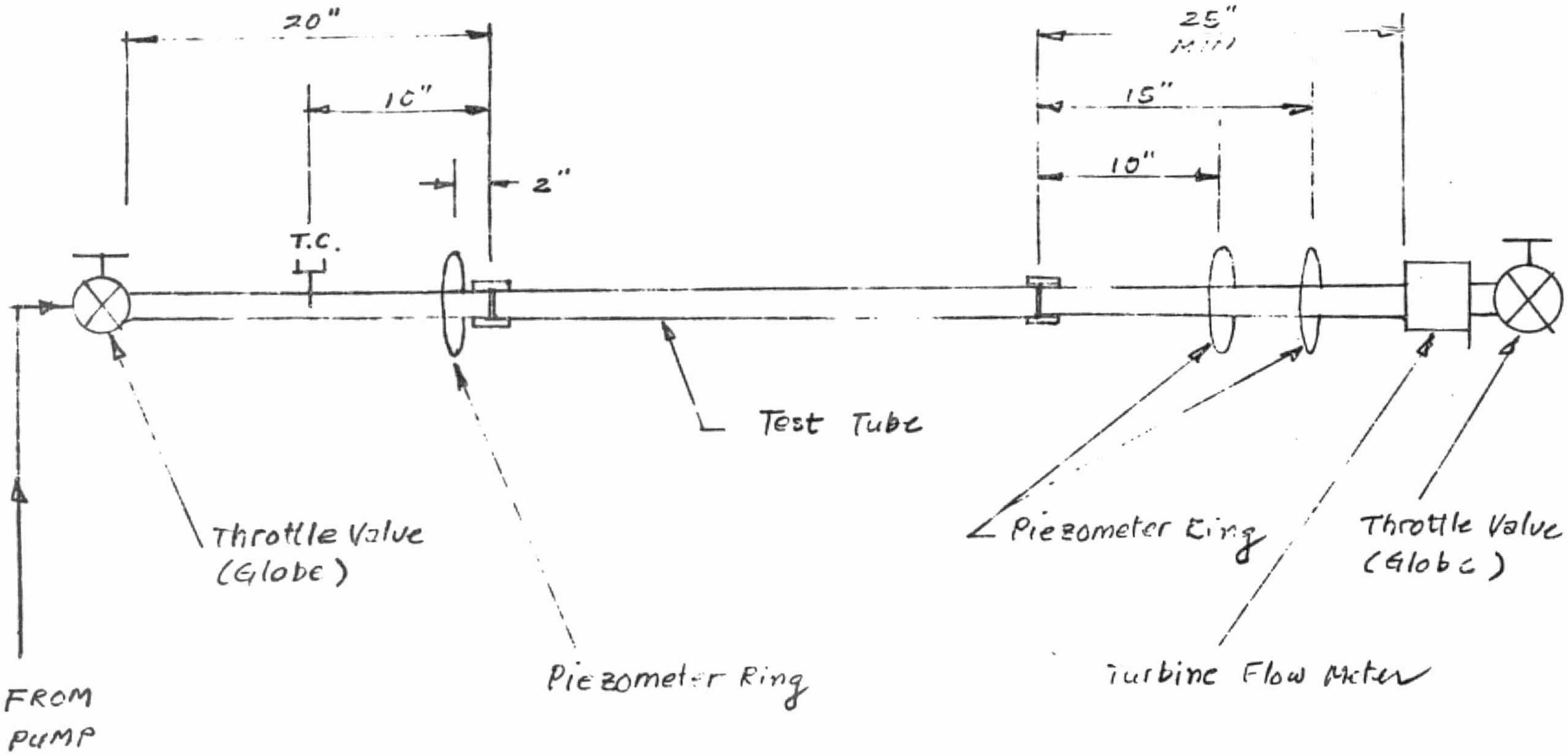


Figure A-1. Single-Phase Pressure Drop Test Setup

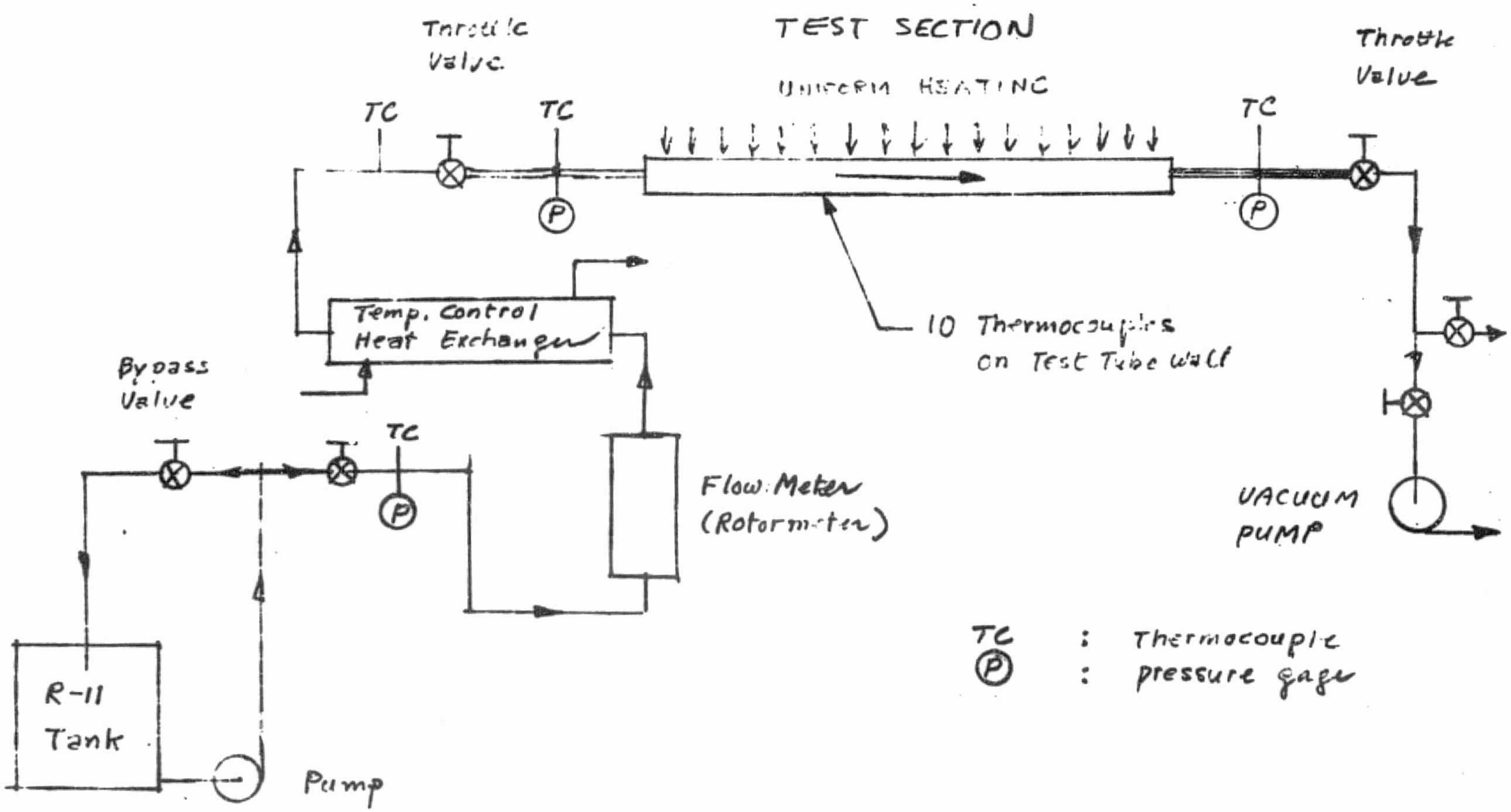


Figure A-2. Schematic, Two-Phase Flow Test Setup

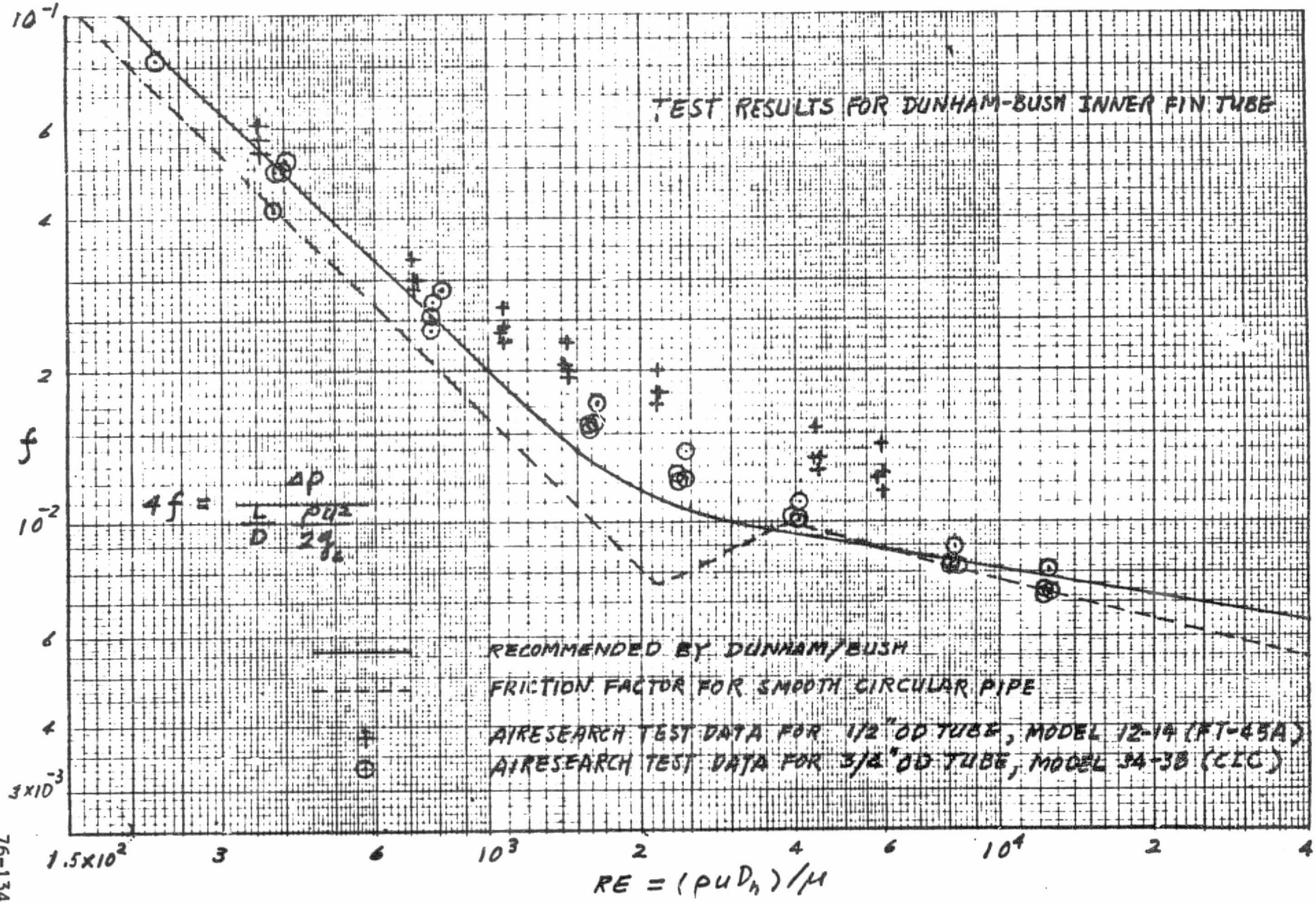


Figure A-3. Friction Factor for Inner Fin Tubes, AiResearch Test Results

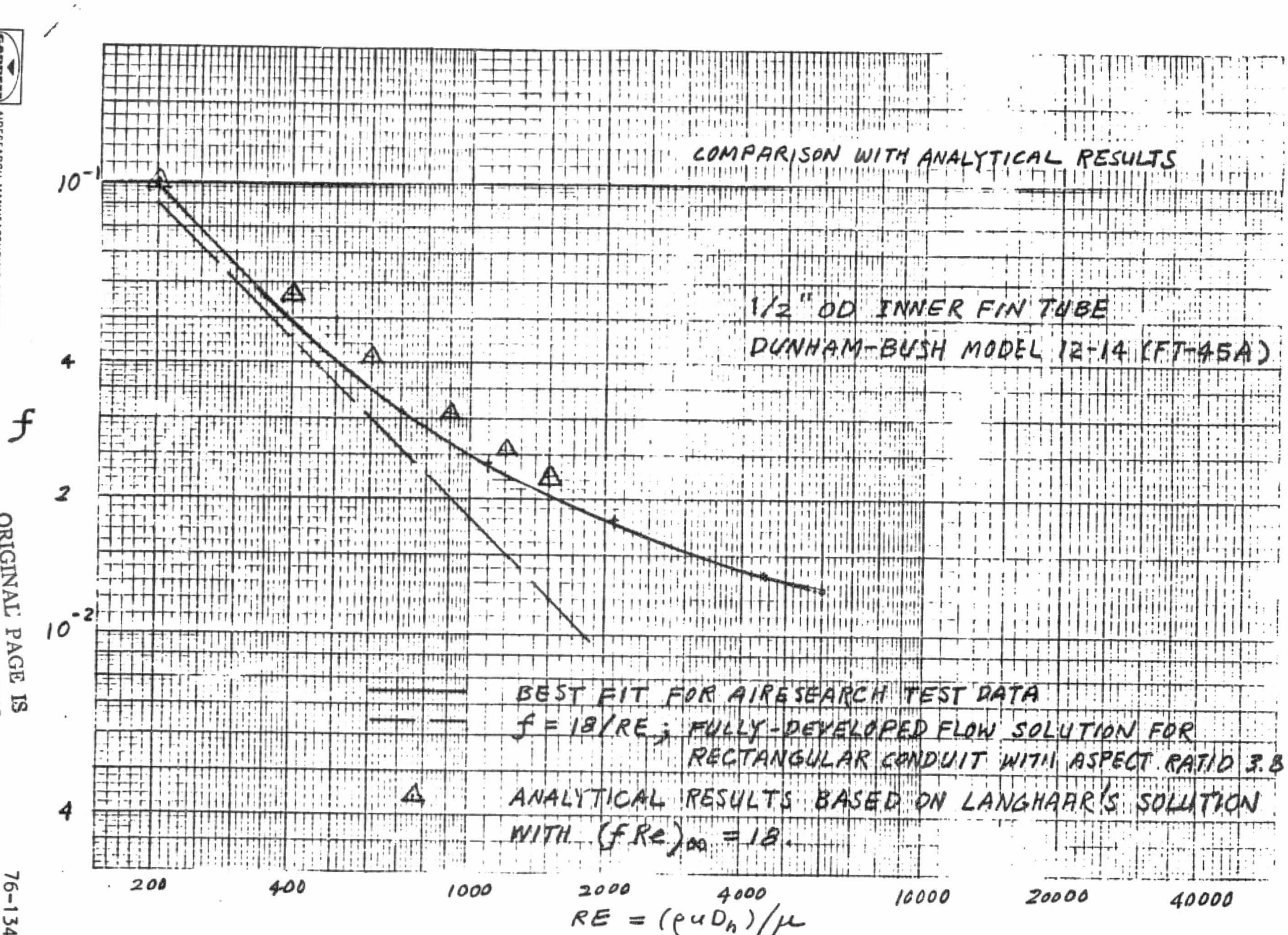


Figure A-4. Comparison of Friction Factor for Inner Fin Tube

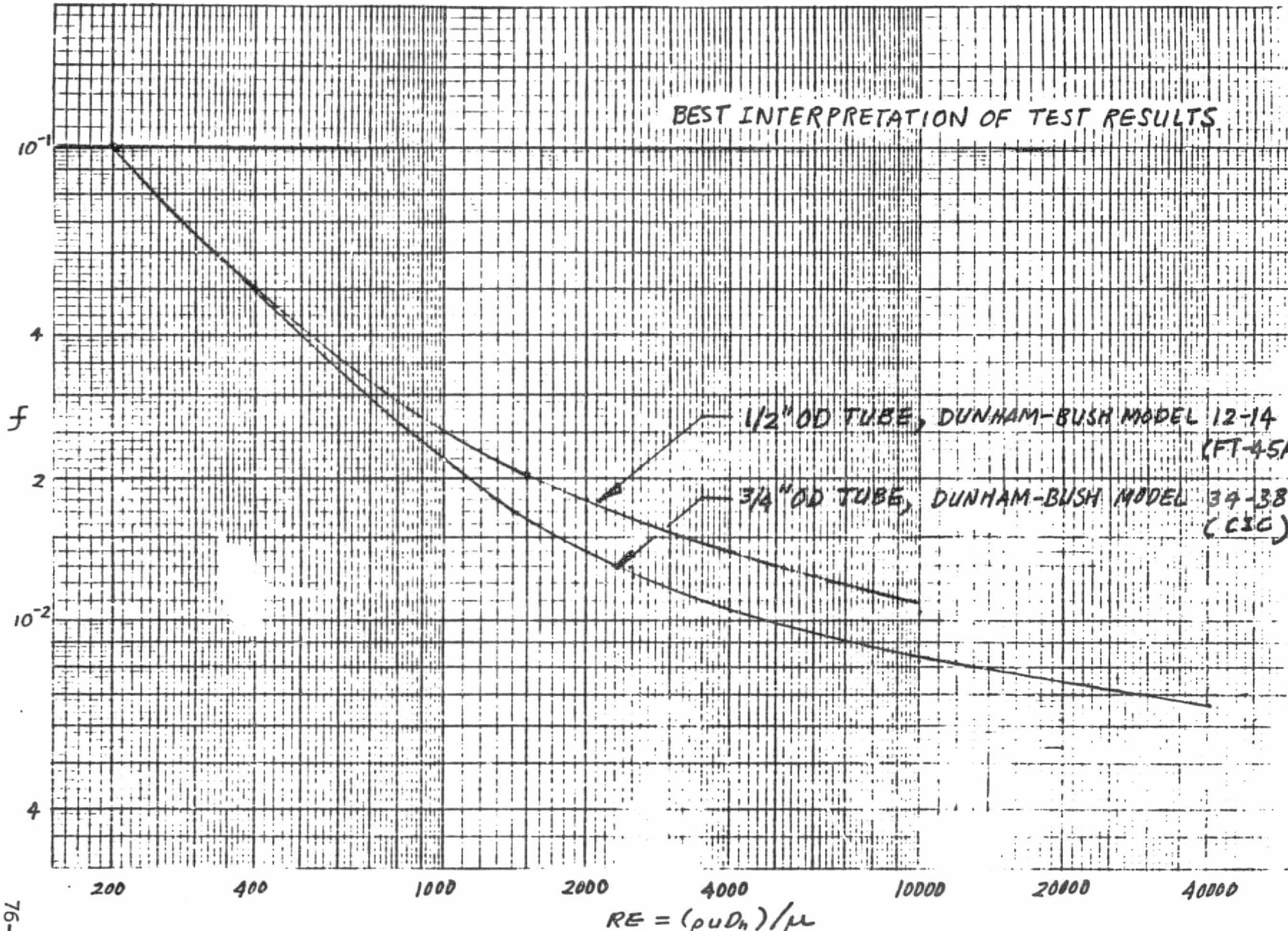


Figure A-5. Friction Factor for Inner Fin Tubes - Best Interpretation

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TABLE A-2

SUMMARY OF TEST RESULTS--EVAPORATION IN 1/2-IN.-OD HORIZONTAL TUBE

W, lb/hr	G, lb/ft^2-sec	P_{in}, psia	$T_{sat,ave}$, °F	Vapor Quality		$\Delta p/\ell$, psi/ft	ΔT_{w-b}, °F	$h_{boiling}$, Btu/hr-ft^2-°F
				Inlet	Exit			
5.9	3.41	9.81	54.3	0.0	1.0	0.09	2.2	124
6.2	3.58	9.55	53.1	0.0	1.0	0.105	2.0	108
8.0	4.62	10.28	55.9	0.0	1.0	0.270	2.6	126
8.3	4.79	10.20	55.0	0.0	0.98	0.200	2.5	130
12.0	6.93	10.24	54.7	0.0	0.96	0.354	3.1	156
12.0	6.93	10.29	54.9	0.0	1.0	0.360	2.6	177

$G = w/A$ mass flux

$T_{sat,ave} = 0.5(T_{sat,in} + T_{sat,out})$ mean saturation temperature

$\Delta T_{w-b} = (T_{w,ave} - T_{sat,ave})$ average wall-to-bulk temperature difference in the boiling region

The wall-to-bulk temperature difference in the boiling region was approximated by taking the difference between the average wall temperature measured and the mean of the inlet and the outlet saturation temperatures. This difference was of the same order of magnitude as the absolute error in the temperature difference, which was estimated to be ±1°F. As a conservative measure, therefore, the wall-to-bulk temperature difference calculated above was adjusted by adding 1.0°F. The adjusted results are shown in Table A-2. The heat transfer coefficients in Table A-2 represent lower limit, therefore, and could be conservative by as much as 50∼100 percent. The pressure drop given in Table A-2 contains a 20 percent margin, a magnitude the same as that of the relative error in the measured pressure difference.

The heat transfer test results compare favorably with Bo Pierre's* correlation as shown in Figure A-6. The correlation has been used in the design of horizontal tube evaporators and the present results confirm its applicability to the present applications. The pressure drop results are shown in Figure A-7 as a function of mass flux. The slope of the best-fit curve compares very well with that which can be determined using the single-phase pressure drop data in Figure A-5. The pressure drops were, in general, much lower than previous predictions.

LOW-PRESSURE EVAPORATION IN 3/4-IN.-OD INNER FIN TUBE

The test conditions and the results derived are summarized in Table A-3. The test data were analyzed in the same way that was used for 1/2-in.-OD tube described above. The test conditions remained fairly steady in most of the runs and energy balance was well satisfied; however, liquid-slugging was

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*Refer to Altman, M., Norris, R. H., Staub, F. W., "Local and Average Heat Transfer and Pressure Drop for Refrigerants Evaporating in Horizontal Tubes," Trans ASME, J. Heat Transfer, p 189, August 1960.

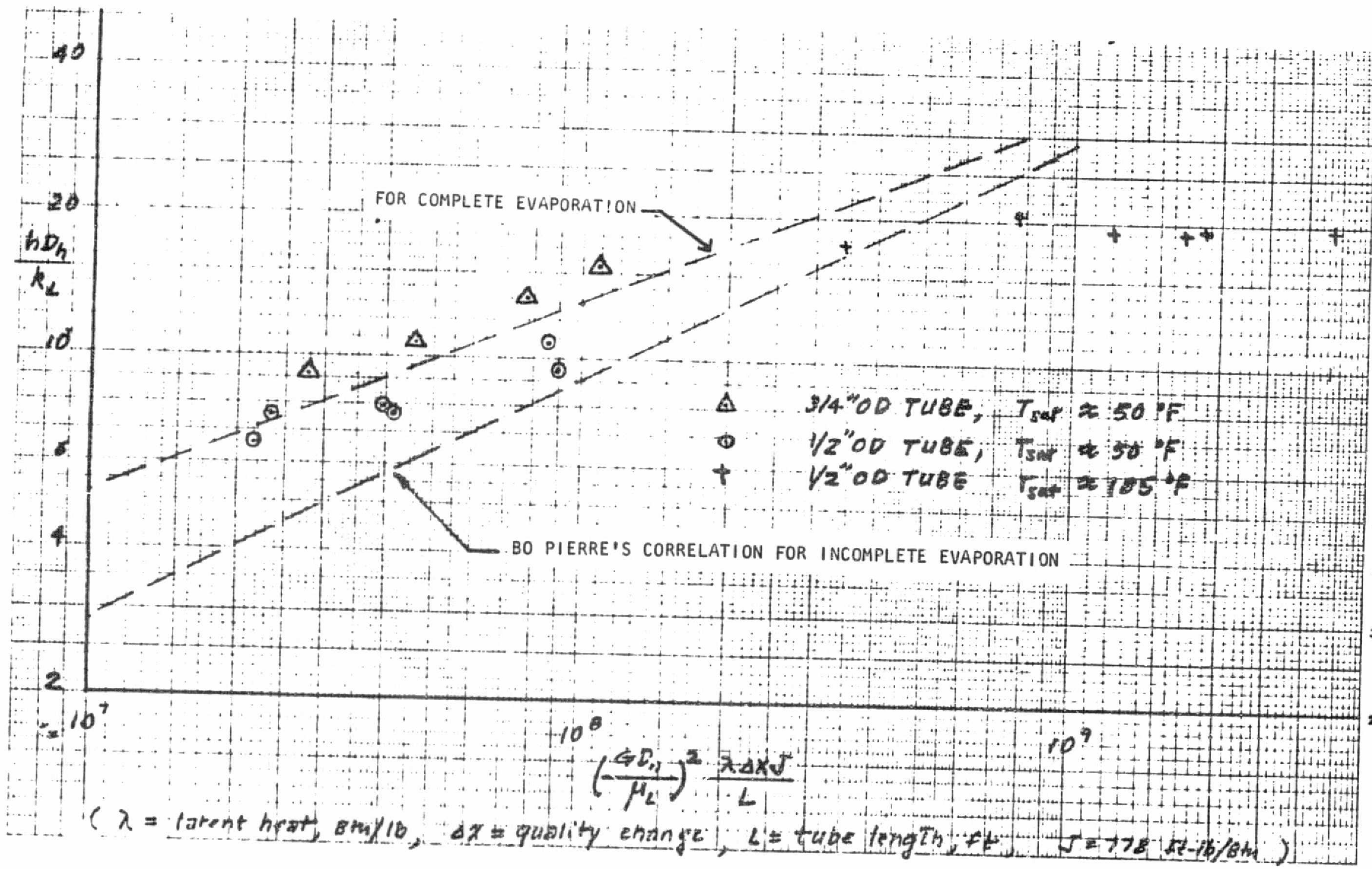

Figure A-6. Horizontal Tube Heat Transfer Results (Evaporation of R11 in Dunham-Bush Inner Fin Tube)

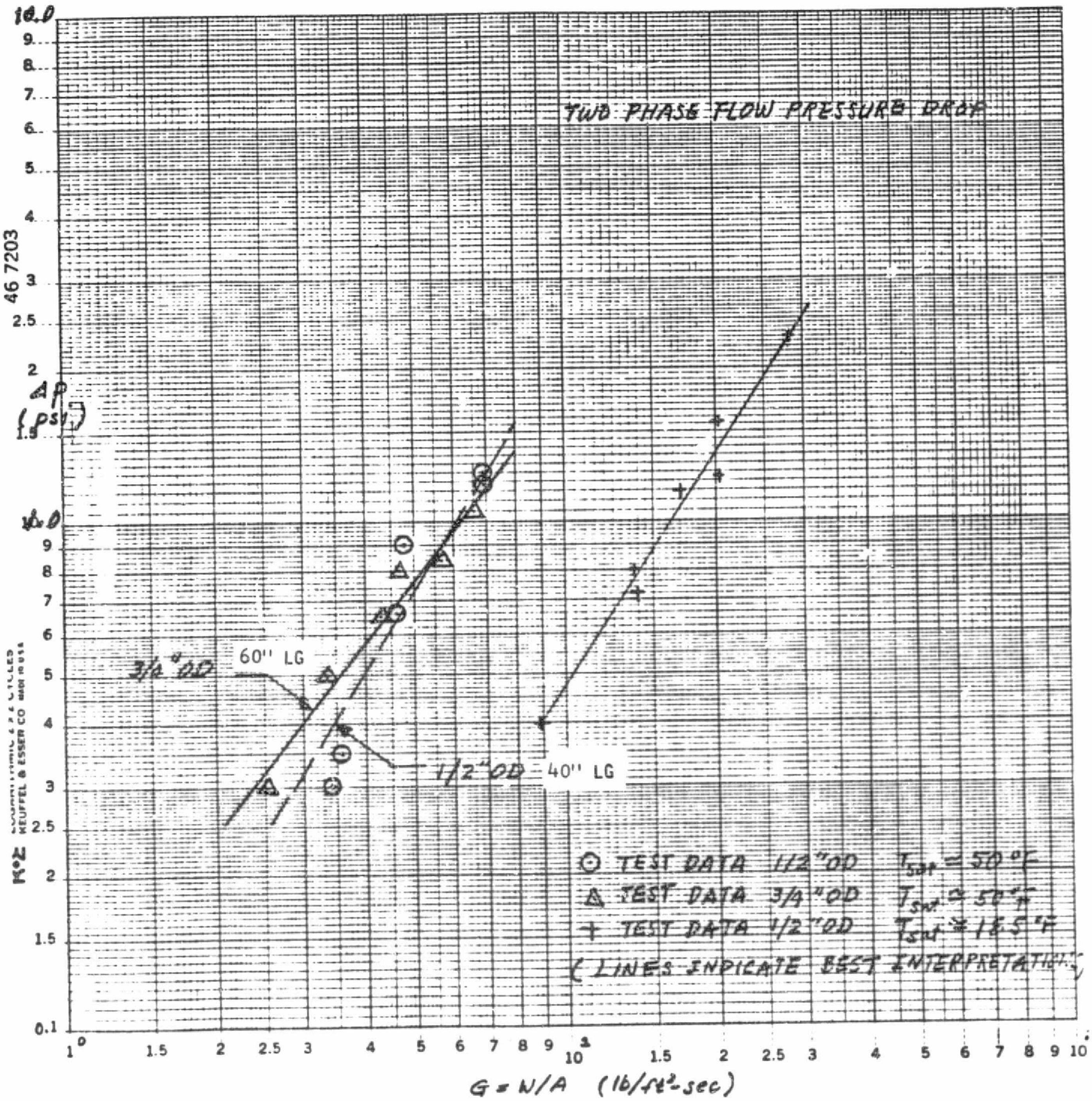

Figure A-7. Pressure Drop vs Mass Flux Rate, Horizontal Tube

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TABLE A-3

SUMMARY OF TEST RESULTS--EVAPORATION IN 3/4-IN.-OD HORIZONTAL TUBE

W, lb/hr	G, lb/sec-ft²	P_{in}, psia	T_{sat}, we °F	Vapor Quality		ΔP/ℓ, psi/ft	ΔT_{w-b}, °F	$h_{boiling}$ Btu/hr-ft²-°F
				Inlet	Exit			
12.3	2.53	8.55	48.0	0.0	1.0	0.06		
16.25	3.34	8.55	47.5	0.02	0.93	0.11	2.2	99.4
16.25	3.34	8.50	47.4	0.02	1.0	0.10		
20.84	4.29	9.61	52.6	0.02	0.93	0.15	2.4	117
20.84	4.29	9.45	51.9	0.03	1.0	0.13	--	''
27.54	5.67	9.55	52.2	0.02	0.93	0.16	2.5	143
27.54	5.67	9.60	52.0	0.02	1.0	0.19		
32.0	6.59	10.0	53.8	0.02	0.94	0.22	2.6	166

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often noticed at fairly high exit vapor qualities and degree of superheat was not as steady as in the 1/2-in.-OD tube test. Agreement with Bo Pierre's correlation is fairly good as shown in Figure A-6, and the pressure drop results show a trend similar to that of the single-phase flow. Again, the pressure drops were, in general, much lower than the predictions.

HIGH-PRESSURE BOILING IN 1/2-IN.-OD INNER FIN TUBE

Table A-4 summarizes the test conditions and the results obtained. In contrast to the low pressure evaporation described above, the high-pressure boiling runs were highly unsteady. Liquid-belching was observed even when the exit was supposedly at a saturated condition or even at a low superheat. A slight increase in heat input from this stage diminished the liquid-slugging but frequently accompanied a violent rise of the outlet vapor temperature. It was very difficult to maintain the exit at a steady superheat and, therefore, all of the boiling runs in Table A-4 were made with the outlet vapor quality varying between 0.90 and 0.99.

The wall-to-bulk temperature difference obtained was much larger than expected. At the same time the heat transfer coefficients calculated from the test results show a relatively flat characteristic with respect to flow rate. This is contrary to the general trend in flow boiling where overall heat transfer increases as flow rate increases. The cause of the above has not been determined yet, and the present results may have to be viewed with reservations until they are verified.

The pressure drops measured show a similar trend to that for the evaporation in 1/2-in.-OD tube, as shown in Figure A-7. Again, the measured pressure drops are much lower than those predicted. The exit vapor quality or degree of superheat was not found to affect the pressure drop to any significant extent. The same was true for both of the low-pressure evaporation tests.

TABLE A-4

SUMMARY OF TEST RESULTS--BOILING IN 1/2-IN.-OD HORIZONTAL TUBE

W, lb/hr	G, lb/ft²-sec	P_{in}, psia	T_{in}, °F	$T_{sat,ave}$, °F	Exit Vapor Quality	$\Delta P/\ell$, psi/ft	ΔT_{w-b}, °F	$h_{boiling}$, Btu/hr-ft²-°F
15.4	8.59	86.0	92.0	185.2	.96	0.12	3.2	209
23.7	13.69	85.8	74.0	185.4	.92	0.21	4.3	243
23.9	13.90	86.2	93.0	185.2	.79	0.18	5.0	226
29.5	17.04	86.5	91.0	185.4	.92	0.27	5.5	227
35.2	20.33	86.5	91.0	185.4	.91	0.27	6.6	225
35.2	20.33	86.9	91.5	185.6	.79	0.39	7.0	227
48.5	28.01	87.5	91.0	185.4	.95	0.57	9.1	231

TWO-PHASE FLOW PRESSURE DROP--HEAT TRANSFER TEST--VERTICAL TUBE

This test was initiated after the horizontal tube test above revealed that the vertical tube evaporation is viable from the pressure drop standpoint. At the same time it was realized that the vertical tube evaporation is preferable to the horizontal tube evaporation from the liquid slugging and flow distribution standpoint. The main emphasis of the test was put on the pressure drop because that is the primary factor affecting the design.

The test rig that was built for the horizontal tube test was used in the vertical tube test. This rig has a major drawback for a two-phase test in a vertical tube: the inlet temperature and pressure taps are located 11-in. below the test article and there is no TC measuring the fluid temperature in the boiling region. For the pressure drop across the test article to be determined, this means that the inlet pressure has to be corrected for an 11-in. head of liquid or liquid-vapor mixture whose void fraction is unknown. For the heat transfer in boiling region to be calculated, this means that the bulk fluid temperature should be determined using a local pressure estimated. In the present data analysis, the difficulty was resolved largely by estimating local pressures in such a way that it could give a realistic ΔP for the test article but fairly conservative ΔT_{b-w} for boiling region. The heat transfer results presented below for low-pressure evaporation thus could be conservative by as much as 50 to 100%. The boiling data are currently analyzed and are not reported here.

LOW-PRESSURE EVAPORATION IN 1/2-IN.-OD INNER FIN TUBE

The test conditions and the results derived are summarized in Table A-5. It is noted that the pressure drop is roughly two or three times larger than that for the horizontal tube of the same flow rate, reflecting the effect of

TABLE A-5

SUMMARY OF TEST RESULTS--EVAPORATION IN 1/2-IN. OD VERTICAL TUBE

Run	W, lb/hr	G, lb/ft^2-sec	$T_{sat,ave}$, °F	Vapor Quality		ΔP/ℓ, psi/ft	ΔT_{w-b}, °F	$h_{boiling}$, Btu/hr-ft^2-°F
				Inlet	Exit			
A1	7.69	4.44	49.8	0.15	1.0	0.33	1.1	240.
B1	6.45	3.72	51.0	0.15	1.0	0.32	1.2	180.
H1	6.32	3.65	50.6	0.15	0.93	0.30	1.1	160.
I1	5.96	3.45	50.3	0.15	0.96	0.31	1.4	116.
L1	8.51	4.92	49.0	0.18	1.0	0.39	1.5	175.
M1	6.26	3.62	49.3	0.18	0.93	0.36	1.4	124.
A2	5.77	3.34	50.8	0.18	1.0	0.33	0.9	193.
B2	5.93	3.43	49.2	0.0	1.0	0.36	1.3	171.
C2	5.67	3.27	50.1	0.0	1.0	0.35	1.5	147.
D2	6.51	3.76	49.3	0.0	0.95	0.36	1.4	159.
E2	6.51	3.76	49.4	0.0	0.90	0.38	1.4	157.
F2	7.37	4.26	49.7	0.0	1.0	0.34	1.3	227.
H2	7.37	4.26	49.7	0.0	0.96	0.33	1.2	232.
I2	8.15	4.71	49.7	0.0	1.0	0.33	1.4	237.
J2	8.25	4.77	48.8	0.0	0.95	0.37	1.7	183.

liquid head in the vertical tube. Heat transfer performance appears to be superior to the horizontal tube evaporations. Test conditions were very stable in most of the runs except for small continuous fluctuations of flow rate and station pressures. No belching of liquid was observed at a vapor quality higher than 0.95. Degree of superheat was relatively stable.

LOW-PRESSURE EVAPORATION IN 3/4-IN.-OD INNER FIN TUBE

The test conditions and the results are summarized in Table A-6. Again, pressure drop is higher than that of the horizontal tube evaporation by a factor of two to three in the range of flow rates considered. The heat transfer performance appears to be somewhat superior to the horizontal evaporation. Evaporation was observed to be very smooth. No belching of liquid was observed when the vapor quality approaches unity.

TABLE A-6

SUMMARY OF TEST RESULTS--EVAPORATION IN 3/4-IN. OD VERTICAL TUBE

Run	W, lb/hr	G, lb/ft^2-sec	$T_{sat,ave}$, °F	Vapor Quality		ΔP/ℓ, psi/ft	ΔT_{w-b}, °F	$h_{boiling}$, $Btu/hr-ft^2$-°F
				Inlet	Exit			
AA	12.67	2.61	43.8	0.11	0.95	0.28	1.2	143
BB	12.39	2.55	44.8	0.11	1.0	0.23		
CC	10.13	2.08	45.7	0.11	1.0	0.25		
DD	10.34	2.13	45.3	0.11	1.0	0.27		
EE	20.19	4.15	50.1	0.17	1.0	0.25		
FF	17.38	3.58	49.8	0.17	1.0	0.25		
GG	17.48	3.60	50.1	0.17	0.95	0.25	1.6	132
HH	17.48	3.60	48.5	0.0	1.0	0.30		
II	20.18	4.15	48.8	0.0	1.0	0.28		
JJ	20.18	4.15	49.5	0.0	0.99	0.28	1.2	254
KK	20.10	4.14	49.1	0.0	1.0	0.26		
LL	10.40	2.40	43.8	0.0	1.0	0.24		

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