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DEMONSTRATION OF A LOW-TEMPERATURE WASTE HEAT DRIVEN REFRIGERATION SYSTEM

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PHASE III

FINAL REPORT

SEPTEMBER 1983

Report prepared by

Foster-Miller, Inc.
350 Second Avenue
Waltham, MA 02254

under

Subcontract No. 41X-28906C

for

Oak Ridge National Laboratory
operated by
UNION CARBIDE CORPORATION
for the
U.S. DEPARTMENT OF ENERGY
under
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Report prepared by:

Hamed Borhanian
Scott Hynek

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ABSTRACT

As the cost of energy continues to rise faster than the cost of machinery, the minimum temperature for economically recoverable waste heat continues to decrease. The Department of Energy, anticipating the day when 140°F water will constitute an economically exploitable resource, has funded the development of several systems to convert 140°F water into process steam or industrial refrigeration. Foster-Miller, Inc. has designed, built, and tested one such system that is powered by the 140°F water and delivers 20 tons of refrigeration.

This system uses an organic Rankine power cycle to drive a standard vapor-compression refrigeration cycle. The power and refrigeration cycles are integrated in that they share a common working fluid (R-22), a common condenser, and a single hermetic housing for all moving parts.

The results of the Phase I Design and Phase II Component Tests were presented in a single report (Ref. 1) in March, 1982. This Phase III report duplicates very little of the information found in that earlier report, but refers to specific parts of it many times.

This report does not describe the system or its components, except where circumstances and experience dictated that the finished system differ from the design described in Ref. 1. Instead, it describes the measured performance of the system and its components, and it discusses the reasons for and results of those design refinements.

By and large, the system's performance met its goals. The performance of the developmental components even exceed predictions. This system could be made with 40 tons, or 80 tons capacity by simply using larger, and equally available components.

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1. EXECUTIVE SUMMARY

As the cost of energy continues to rise faster than the cost of machinery, the minimum temperature for economically recoverable waste heat continues to decrease. The U.S. Department of Energy, anticipating the day when water as cool as 140°F will constitute an economically exploitable resource, has funded the development of several systems to convert 140°F water into process steam or industrial refrigeration. Foster-Miller, Inc. has designed, built, and tested under subcontract to ORNL, one such system that is powered by 140°F water and delivers 20 tons of refrigeration.

This system could be made with 40 tons, or 80 tons, capacity by simply using larger, and equally available, compressors and multiple rotary valves. It can be used wherever a continuous source of 140°F (or hotter) waste water is collocated with a continuous requirement for refrigeration.

1.1 System Description

This waste heat driven chiller system uses an organic Rankine power cycle to drive a standard vapor-compression refrigeration cycle. The power and refrigeration cycles are integrated in that they share a common working fluid (R-22), a common condenser, and a single hermetic housing for all moving parts.

The power cycle is shown on the left hand side of Figure 1-1. 400 gpm of 140°F water heats the boiler to provide 300 psia freon vapor to the reciprocating expander. The expander exhausts wet freon vapor at 150-200 psia through an oil separator to the evaporative condenser. The condensate then passes through the boiler feed pump (a gerotor pump with a centrifugal inducer) to complete the power cycle. Boiler level is controlled by a level

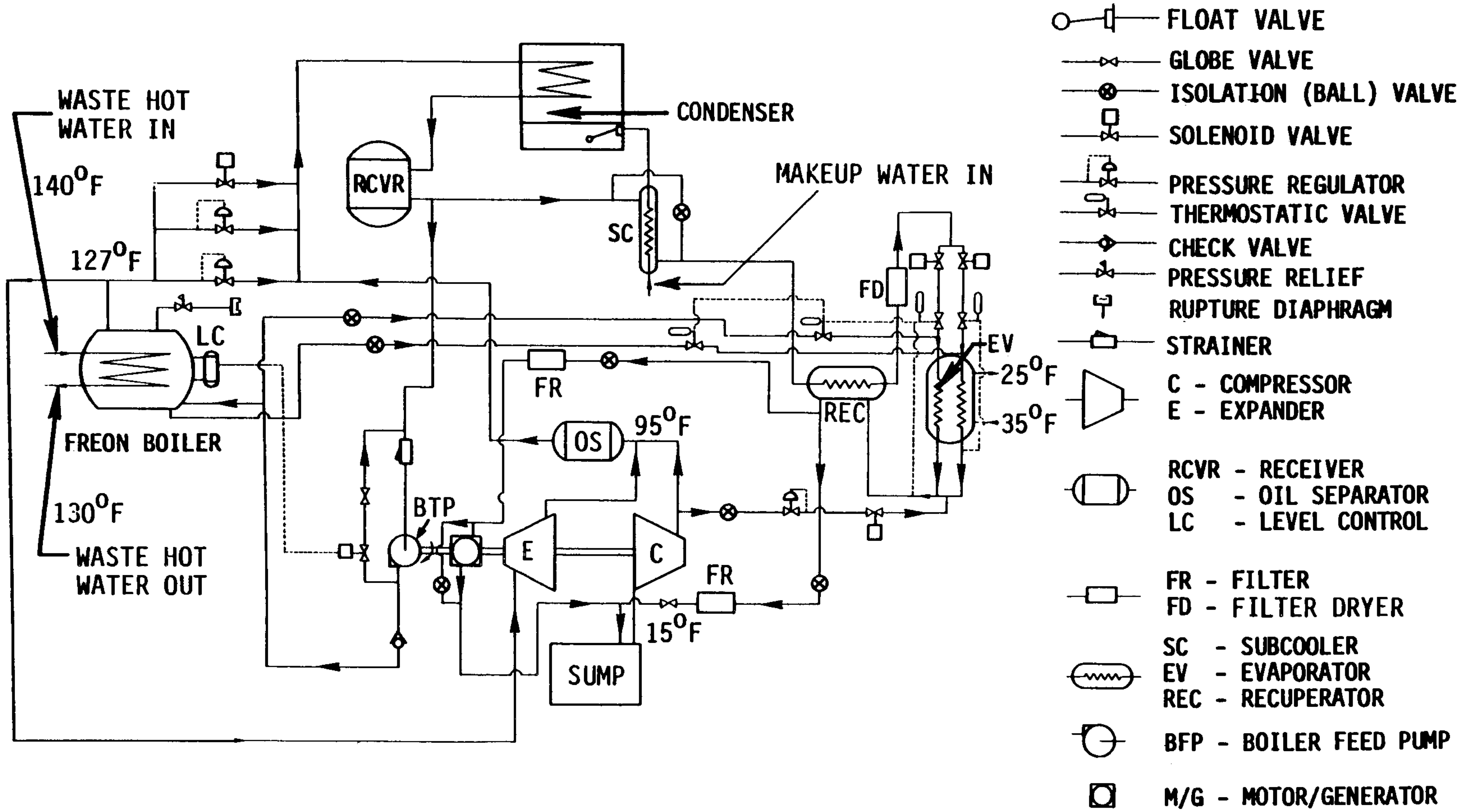


Figure 1-1. Waste Heat-Driven Chiller System Flow Schematic Diagram

control switch, which upon sensing high level opens a bypass line to reduce the flowrate from the boiler feed pump. Boiler pressure is controlled by pressure regulators that send any excess vapor directly to the condenser.

On the right hand side of Figure 1-1 is a conventional vapor-compression refrigeration cycle. 50 gpm of 40°F brine is chilled to 30°F by the dual circuit evaporator operating at 15°F. The freon vapor leaving the evaporator is superheated by 15°F in the recuperator to protect the compressor from ingesting liquid. The compressor discharges superheated vapor to the same condenser as is fed by the expander, through the same oil separator. That portion of the condensate that returns to the refrigeration cycle is subcooled twice (first by the condenser makeup water in the subcooler; second by the suction vapor in the recuperator) to make the evaporator more effective. Finally, it goes through the filter-drier to the thermostatic expansion valves, whence low pressure liquid and vapor return to the evaporator.

The expander provides the shaft power to run the compressor as supplemented by the integral induction motor/generator as follows. On hot days the condenser pressure will be high, limiting the shaft power generated by the expander, and requiring more of the compressor; the induction motor/generator then serves as a motor to make up the shaft power deficit. On cold days, the low condenser pressure permits the expander to generate more shaft power and requires less of the compressor; the resulting shaft power surplus is converted to 440 volt, 3 phase power by the induction motor/generator serving as a generator.

A four-cylinder York open-shaft refrigeration compressor was converted to a two-cylinder compressor plus two-cylinder expander, by replacing the cylinder head and the reed valves on one bank of two cylinders with a special head. This head contains a rotary valve, which valves the intake and exhaust flows to each expander cylinder through a single, large, port at the top of each cylinder. The rotary valve was selected because of its very large flow passages with resultant low pressure drops.

The crankshaft that serves both expander and compressor is coaxial with all other rotating components - the flywheel, the motor/generator, the gerotor pump, the centrifugal inducer, and the lubrication pump. All turn together at 1200 rpm within a hermetic housing, shown in Figure 1-2, so that there are no exposed shafts to require seals.

The evaporative condenser is supported by an 8 ft high steel framework, and the rest of the system is beneath the condenser. This arrangement (shown in Figure 1-3) minimizes the space required, and gives the condensate some static head with which to prevent flashing in the liquid lines. The evaporative condenser can be easily disassembled into two major components, making the system readily transportable by truck.

1.2 Significant Design Choices

Using the same working fluid for both the power cycle and the refrigeration cycle presents pressure restrictions that limit the number of refrigerants that can be used to R-12, R-22, and R-717. The saturation pressure should be above atmospheric pressure at the evaporator temperature (to prevent air inleakage) and below 300 psig (boiler test pressure) at 127°F, the maximum boiling temperature. Table 1-1 shows R-717 (ammonia) to have the best efficiency and highest capacity for a given compressor and expander. However, ammonia is incompatible with copper, and the windings of the hermetically enclosed motor are copper. R-22 is almost as good thermodynamically as ammonia, and is compatible with all materials in the system.

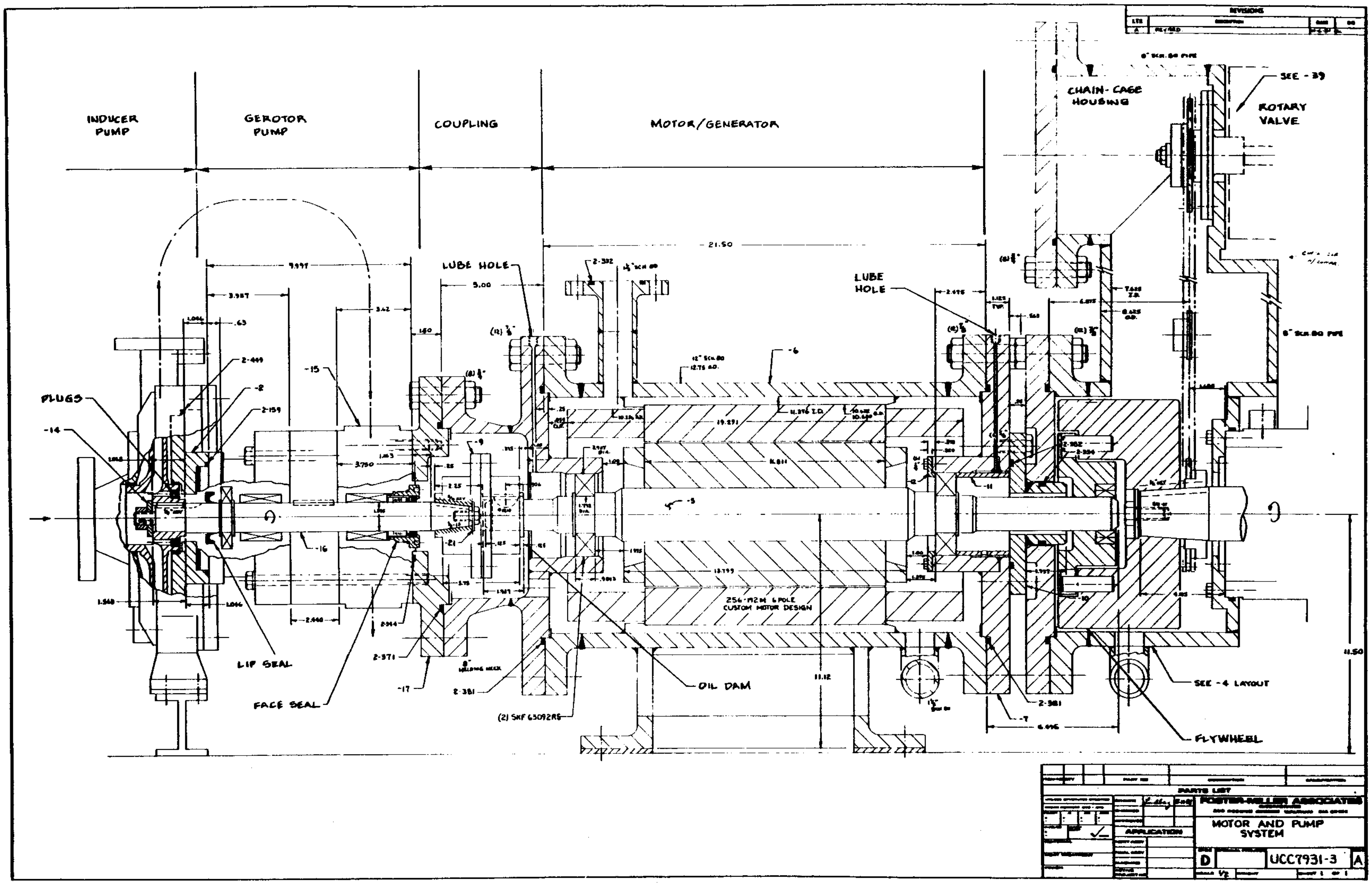
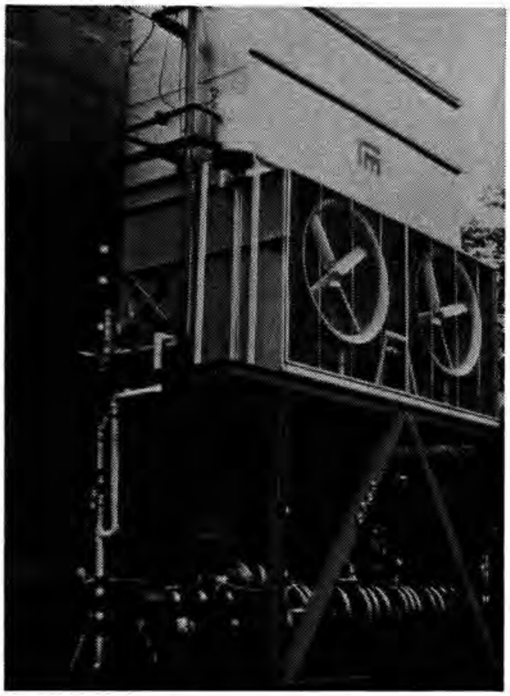
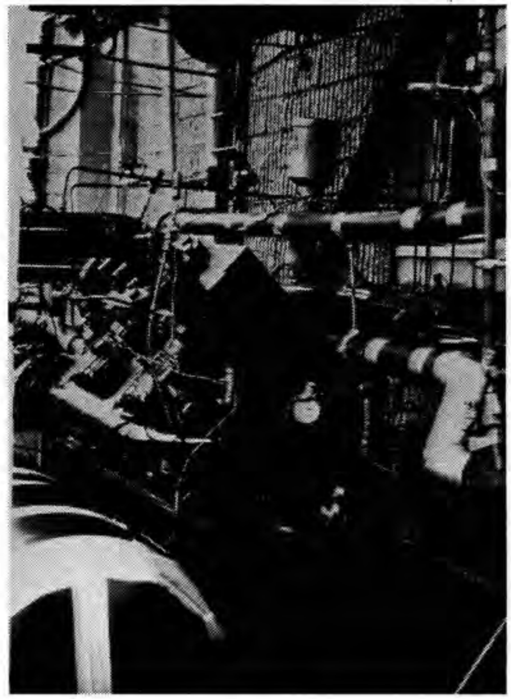


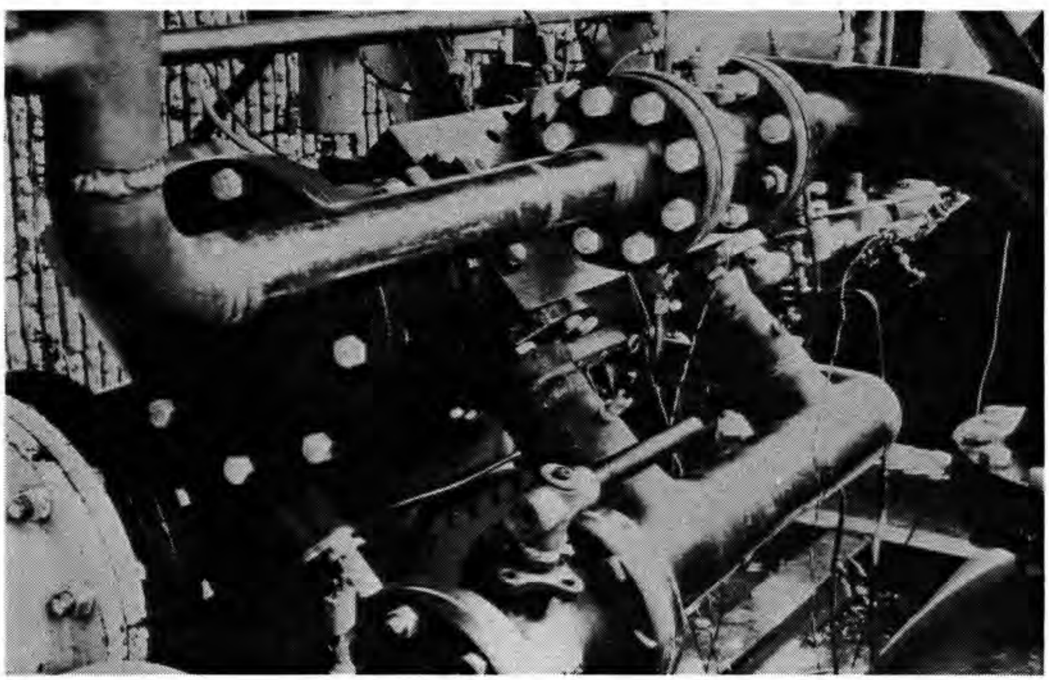
Figure 1-2. Hermetic Housing Enclosing All Rotating Parts - Inducer, Gerotor Pump, Motor/Generator, Flywheel, Compressor/Expander Shaft



a) ENTIRE CHILLER SYSTEM, WITH EVAPORATIVE CONDENSER ON TOP AND WASTE HEAT PIPING AT LOWER LEFT



b) ROTARY VALVE HOUSING, WITH SINGLE INTAKE AND DUAL EXHAUST PIPES MANIFOLDED TO COMPRESSOR DISCHARGE



c) CHAIN DRIVE HOUSING CONNECTING ROTARY VALVE TO FLYWHEEL, INDUCTION MOTOR/GENERATOR AT LOWER LEFT

Figure 1-3. Three Views of the Assembled Prototype Chiller System

Table 1-1. Comparison of Candidate Working Fluids

R	P_{sat} (15°F) (psig)	P_{sat} (127°F) (psig)	Cycle Efficiency*	Specific Size	
				Expander (cfm/hp)	Compressor (cfm/ton)
R-12	18	174	0.131	3.50	4.92
R-22	38	285	0.128	2.19	3.01
R-717	28	302	0.142	2.06	2.81

*Basis for Cycle Efficiency: $\frac{\text{Heat Duty of Evaporator}}{\text{Heat Duty of Boiler}}$, assuming no subcooler and no recuperator, 90% thermal and mechanical efficiencies of pump and expander.

Three of the five heat exchangers are large enough that their sizing affects total cost as well as system performance. Computerized calculations of the system's cost and performance as functions of the boiler, condenser, and evaporator sizes were used to determine the combination of heat exchanger sizes that optimized the ratio of capacity to cost; given the waste heat temperature, the required brine temperature, and the average ambient wet bulb temperature.

1.3 Component Design and Operation

Several major components of the system merit discussion; some are orthodox components used in unorthodox ways (with or without modification), while others are components that were designed specifically for this application.

1.3.1 Freon Boiler

A Bell & Gossett Model RCF 200-82 water-cooled freon condenser was used as a freon boiler with little modification. It was selected over other such condensers because of its large size (200 nominal tons in a single shell), its low water side pressure drop (due to only two tube passes), its absence of internal subcooling baffles (which would cause locally high vapor velocities), and its availability with a large (6 in. nominal) vapor exit nozzle (low vapor velocities, for less liquid entrainment). It worked well as long as the liquid level was kept near the middle of the shell; higher liquid levels could (and did) cause enough liquid entrainment to halt the expander.

1.3.2 Induction Motor/Generator

This 20 hp (15 kw) motor was designed by Custom Motor Design, Inc. to be cooled by the suction vapor. It drew 0.6 kw under no load, and the design efficiency was 96 percent under full load.

In addition to accommodating deficits and surpluses of shaft power, this motor/generator served as a starter motor and as a speed governor.

1.3.3 Gerotor Feed Pump and Centrifugal Inducer

Gerotor pumps are not generally used to pump freon, but only a positive displacement pump such as this would provide sufficient head at 1200 rpm.

The gerotor pump was demonstrated to require about 20 ft of net positive suction head to prevent cavitation. Because the condenser's subcooling and the static head could not always provide that, a Worthington Model D 1021-2x1x10 centrifugal water pump was adapted for use as an inducer. It raised the head available to the gerotor suction by 30 ft, and generally prevented cavitation.

1.3.4 Rotary Valved Reciprocating Expander

There are several things that one must do to convert part or all of a reciprocating compressor to a reciprocating expander:

1. Carefully consider the differential pressure limit across the piston. For this specific York compressor, the limit is 275 psi; more than this overloads the connecting rod bearings. This means that the expander intake pressure (on the top of the expander piston) cannot exceed the compressor suction pressure (below the expander piston, in the crankcase) by more than 275 psi.

2. Modify the wristpin bearings to accept a non-reversing load. The load on a compressor piston changes direction with each stroke, and adequate lubrication can be achieved by letting oil drip in through a hole in the top of the connecting rod; the wristpin moves back and forth enough in its bushing to surround itself with oil. The load on an expander piston, however, fluctuates but does not change direction; oil entering the top of the wristpin bushing cannot reach the highly loaded underside of the bushing. Our solution was to enlarge the wristpin bushings and to insert roller bearings, letting the wristpin itself serve as the inner race.

3. Re-shape the piston crown. The crown of a compressor piston conforms to the shape of the reed valve assembly. We filled in the dished crown with epoxy, reinforced with metal pins, to conform to the lower surface of the rotary valve housing. This served to decrease the clearance volume.

4. Damp the torsional vibrations. Two expander cylinders and two compressor cylinders yield torque fluctuations that are much more irregular than those produced by four compressor cylinders. Furthermore, the rotary valve drive train is sensitive to these fluctuations. The motor/generator could have served as a flywheel if the shaft coupling between it and the crankshaft were perfectly rigid, but it was not. We found it quite necessary to install a flywheel on the end of the crankshaft.

A single rotary valve assembly, shown in Figure 1-4, controlled the intake and exhaust of both expander cylinders. A single, large (61% of piston area) port in the head of each expander cylinder was alternately connected to the intake (at the center of the valve housing) or the exhaust (at the ends). It was this large port area that minimized the breathing losses; a typical poppet valve arrangement, with the exhaust valve area and the intake valve area each at perhaps 25% of the piston area, would have had roughly six times the breathing losses.

The radial clearances between the cylindrical steel valve and the cast iron valve block had to be very small to prevent leakage from intake to exhaust through this gap. At the same time, they had to be greater than the clearances in the ball bearings that supported the valve in the housing. There was an everpresent rotating force couple that tended to cock the valve within its housing, and contact between valve and housing could not be permitted. To combat this, we took two steps:

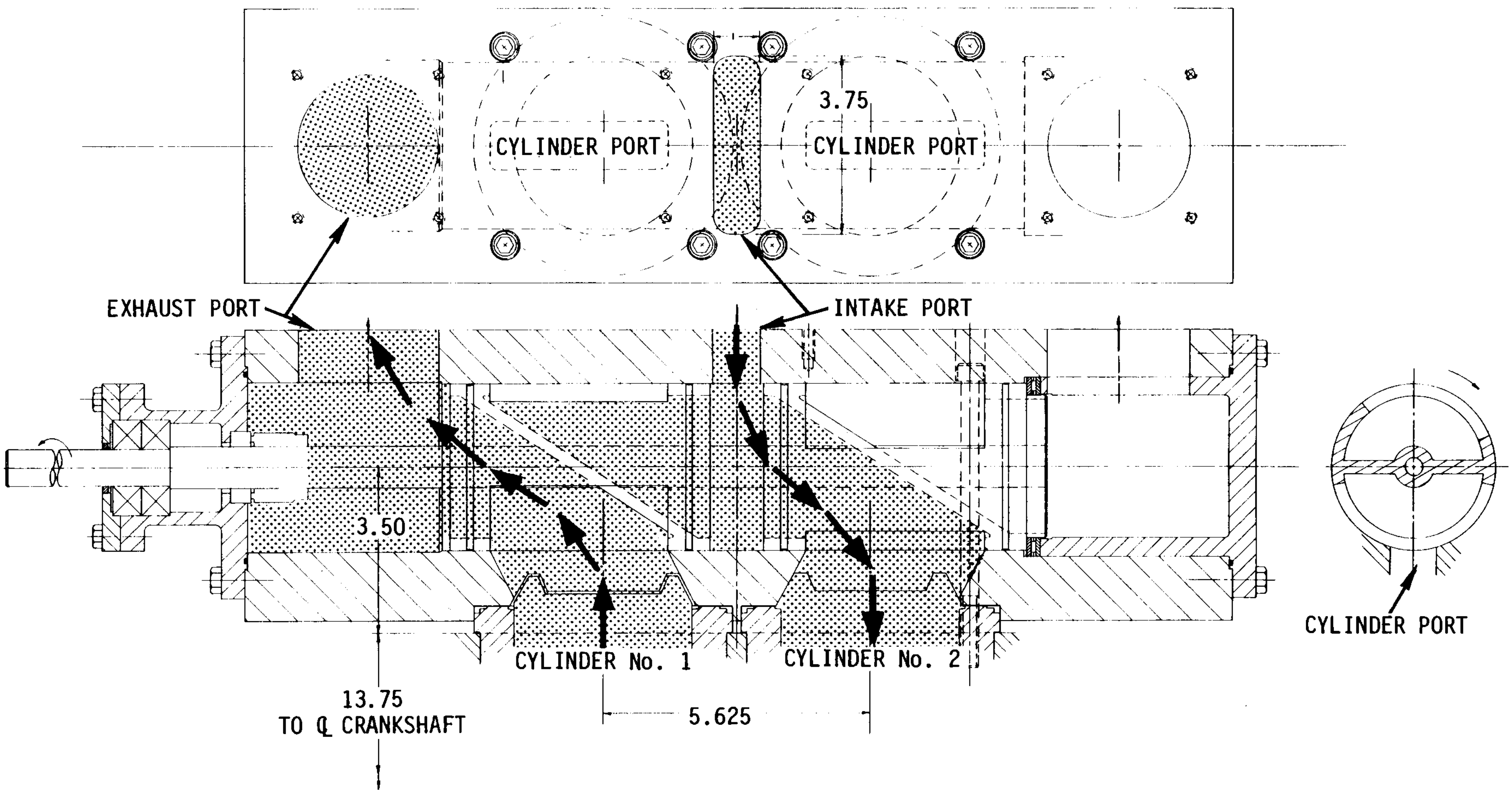


Figure 1-4. Operation of Rotary Valve Assembly, Showing Cylinder No. 1 Exhausting and Cylinder No. 2 on the Intake Stroke

1. The surface of the valve was ground off-axis, so that clearance was provided only where it was needed.

2. The surface of the valve was coated with Teflon, which abraded where the force couple brought the valve close to the housing, but filled the gap elsewhere.

1.4 Performance

The system was operated and tested for performance for 38 hours. Table 1-2 shows that the expander's performance exceeded expectations. Initial predictions of the expander's mechanical efficiency and its volumetric efficiency seem to have been conservative.

However, the overall system performance was slightly less than estimated. Predicted parasitic losses either were based on faulty measurements (bad wattmeter) or were incomplete, since the need for an inducer, with its power requirement, had yet to be established at the time of the predictions.

Figure 1-5 shows the sensitivity of the expander output (hence the system output) to the temperature available to the freon boiler. Figure 1-6 shows that while the electrical requirement of the system was quite sensitive to condensing pressure, the refrigeration capacity was not.

1.5 Economic Analysis

Using actual costs for the heat exchangers in the system that was tested, and costs for the developmental components as estimated for manufacture in production quantities (100 per year), we estimate a price of $91,951 for this system in 1983 dollars. By comparison, a conventional chiller system with comparable design capacity (we chose the Edwards model CC-40-Cl-AHP) costs $36,250. How quickly the energy savings will justify the initial

Table 1-2. Comparison of Actual Results with Those Predicted by Computer Simulations

Condensing temperature °C (°F)	Condensing pressure (psig)	Expander output (kW)	Compressor input (kW)	Feed pump input (kW)	Net power to generator (from motor) (kW)	Balance after pump and fan (kW) (2)	Net system output (kW) (3)
Predicted results							
35 (95)	182	13.0	18.5	1.7	(7.2)	(13.9)	-
29 (85)	156	17.7	17.2	2.1	(1.6)	(8.3)	-
24 (75)	132	22.2	15.7	2.4	4.1	(2.6)	-
18 (65)	111	25.9	13.8	2.7	9.4	2.7	-
13 (55)	93	29.3	11.6	3.0	14.7	8.0	-
Actual performance							
35 (95)	182	12.7	19.4	2.8	(9.5)	(16.9)	(18.0)
29 (85)	156	19.0	17.5	3.3	(1.8)	(9.2)	(10.3)
24 (75)	132	23.4	15.8	3.6	4.0	(3.4)	(4.5)
18 (65)	111	- (1)	-	-	-	-	-
13 (55)	93	-	-	-	-	-	-

(1) Missing values correspond to condensing temperatures never obtained during testing

(2) Condenser pump and fan power consumption: Predicted = 6.7 kW
Actual = 7.3 - 7.4 kW

(3) Parastic losses in motor/generator and controls = 1.1 kW.

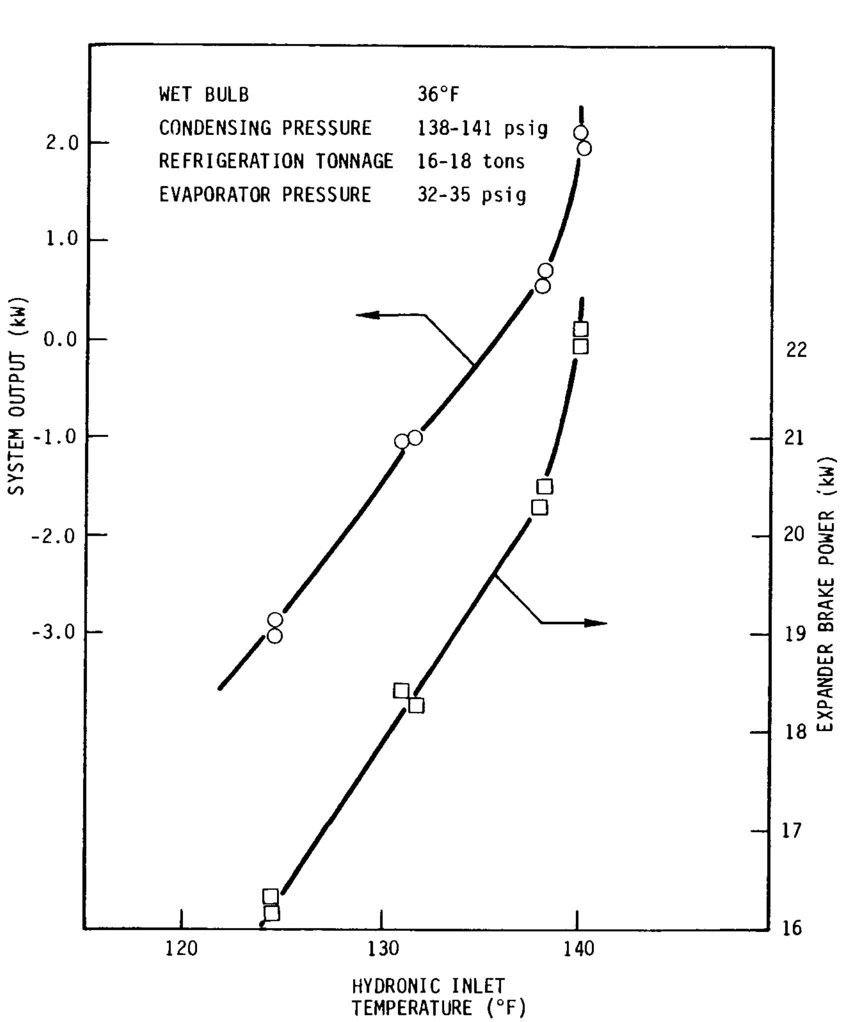


Figure 1-5. Effect of Waste Heat Temperature on the Expander Power Output and on the Net System Power Output

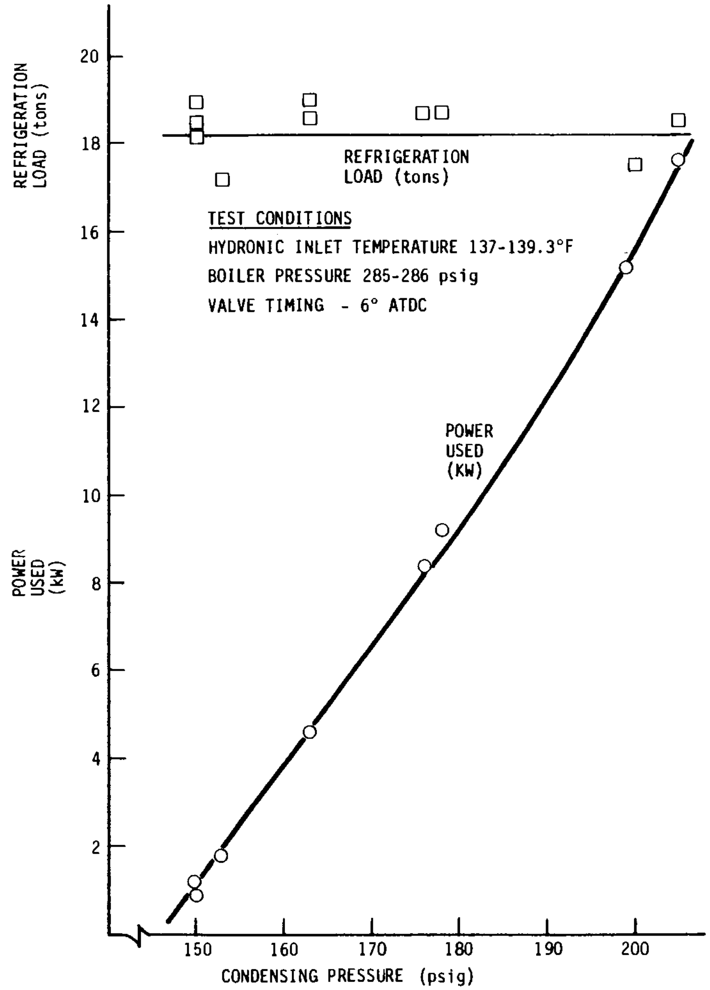


Figure 1-6. Effect of Condensing Pressure on System Performance

cost difference will depend on the electrical consumption of both systems, the cost of electricity, and the financial assumptions.

The electrical consumption of both systems is presented in Table 1-3. A typical correlation had to be assumed between the ambient dry bulb temperature (to which the air-cooled condenser of the Edwards chiller rejects its heat) and the ambient wet bulb temperature (to which the evaporative condenser of the waste heat driven chiller rejects its heat). The result is that the waste heat driven chiller uses 25 kw less electricity, independent of ambient temperature! This simplifies the analysis, and makes it applicable to any climate.

Table 1-3. Comparison of Electrical Power Consumed by the Waste Heat-Driven Chiller System with that of the Edwards Chiller System

Conditons		Waste Heat Driven Chiller System		Edwards Chiller System	
Ambient Wet Bulb Temperature (F)	Ambient Dry Bulb Temperature (F)	Power Consumed (less parasitics) (kW)	Power Consumed (inclusing parasitics)	Average Compressor Power (kW)	System Power Consumption (kW)
80	105	12.0	20.5	38.5	45.0
73	95	7.0	15.5	34.2	40.7
66	85	2.7	11.2	29.7	36.2
59	75	-0.8	7.7	25.9	32.4
52	65	-3.5	5.0	23.1	29.6
45	55	-5.7	2.8	20.7	27.2

For the cost of electricity, we used Boston Edison's general service rate G-2, applicable to commercial and light industrial customers, which averages about 5.7 cents/kwh for the conditions assumed in this analysis.

We took an investment tax credit of 10% for both systems, and an additional energy tax credit of 10% on the waste heat driven system (applicable to cogeneration systems). We assumed that the cost of electricity would increase at 5% per year. We used ACRS depreciation as 5-year property, a 46% marginal income tax rate, and a 10-year life. We assumed 8000 hours per year (virtually continuous) operation.

The result of this analysis is a 21% internal rate of return, or return on investment. This is a figure that could support either the proponents or the opponents of investing in a waste heat driven chiller system.

1.6 Applications

The applications for this waste heat driven chiller system are quite limited because a continuous source of waste heat must be collocated with a continuous requirement for refrigeration. Economic viability requires continuous operation; while refrigeration is often required continuously, year-round sources of 140°F waste heat are less common than one might think. Also, a heat source can be considered waste heat only if it would otherwise be thrown away, and if there is no other, more cost-effective way to reclaim this heat.

Still, analysis shows this system to be marginally economical in the Boston area. If placed where centrally generated electricity is extremely expensive, or where motor-generated electricity must be used, this system might look very attractive indeed.

This system might work very well with salt gradient solar ponds. The hot, salty lower layer often is much hotter than 140°F, and the cool upper layer could be used with a water cooled condenser, which would outperform and cost less than the evaporative condenser assumed in our economic analysis.

2. PERFORMANCE

2.1 Overall Basic System

The waste heat expander performance was appraised in two stages: first, the optimum valve timing was established and next, the effects of external variables, such as the waste heat temperature, ambient wet bult temperature, and brine temperature were determined. The results of these tests are discussed below.

2.1.1 Effect of Valve Timing

Valve timing can be optimized for either maximum expander efficiency or power output. We decided to maximize the power output because:

- Waste heat is considered effectively free
- Our Freon boiler was over-sized and could boil the additional Freon flow required (see following discussion and subsection 2.3.1.2 on Freon boiler)
- The increase in parasitic power consumption (hydronic pump power) was negligible compared to the gains in expander power.

Consider Figure 2-1 where a typical Pressure-Volume (P-V) diagram for our expander is shown plotted against crank angle degrees. The sequence of events is as follows:

- Point B (or B'): Rotary valve inlet port opens
- Point C (or C'): Rotary valve inlet port closes
- Point D (or D'): Rotary valve exhaust port opens
- Point A (or A'): Rotary valve exhaust port closes.

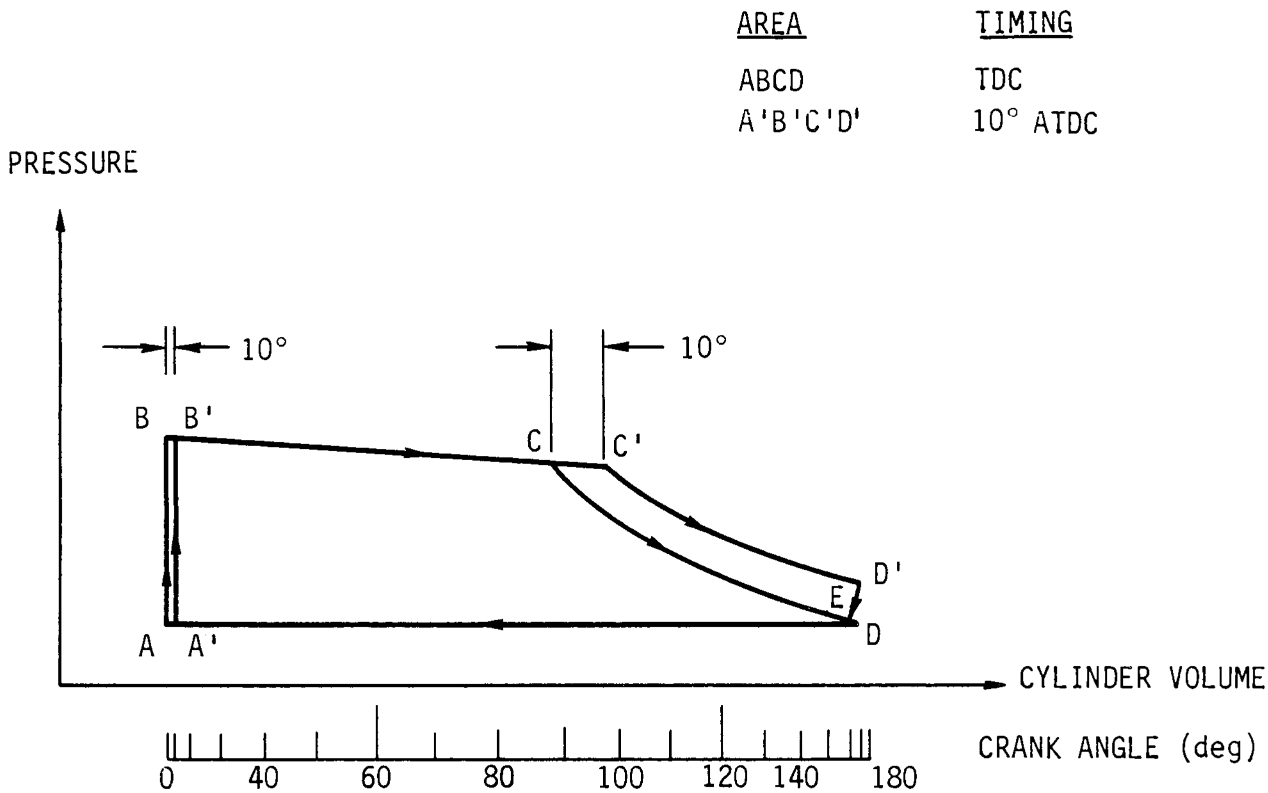


Figure 2-1. Rotary Valve P-V Diagram and Valve Timing Analysis

The area ABCD is the P-V trace for maximum diagram efficiency which in this example happens when the inlet opens at Top Dead Center (TDC). The area A'B'C'D' shows what takes place when the valve timing is retarded by 10°. In this transition, the work performed by the expander cylinder is raised by the difference between the areas CC'D'E and ABB'A'.

Since ABB'A' is a small area compared to CC'D'E, the expander power output is effectively increased by retarding the valve timing. However, since segment B'C' is longer than BC, the inlet flow throughput is also significantly boosted which requires more waste heat input, boiler, and condenser heat transfer area. Thus, the extent to which we could enhance the power output by retarding the valve timing was limited by the hydronic system input and the heat transfer area available in the Freon boiler.

Figure 2-2 shows actual P-V diagrams for the Freon expander operating at different valve timings. In these tests, the valve timing was retarded in steps of 3° starting from TDC. These results are derived from the data presented in Appendix 3, runs 3-8 to 3-17. As indicated in Figure 2-2d, the highest boiler pressure attained for the 9° after TDC (ATDC) timing was 277 psig. Our hydronic water heaters were not sized to adequately support full boiler pressure (285 psig) at 9° ATDC. Hence, 6° ATDC was chosen as the optimum valve timing and all subsequent tests were performed at this value.

Also of interest in Figure 2-2, is the effect of valve timing on the P-V area which shows a consistent increase in P-V work as the timing is retarded. This is borne out by the results in Figure 2-3, where the expander brake horsepower is plotted against the pressure difference across the expander for various valve timings. The lines drawn through the performance points are least-square fits. The 9° ATDC timing is represented only by one data point. For this valve-timing no further testing was deemed necessary since the boiler pressure would not reach design point.

The ultimate result of these tests, of course, was the selection of 6° ATDC as optimum valve timing. The results presented in the following subsections were all obtained at this valve setting.

2.1.2 Effect of Waste Heat Temperature

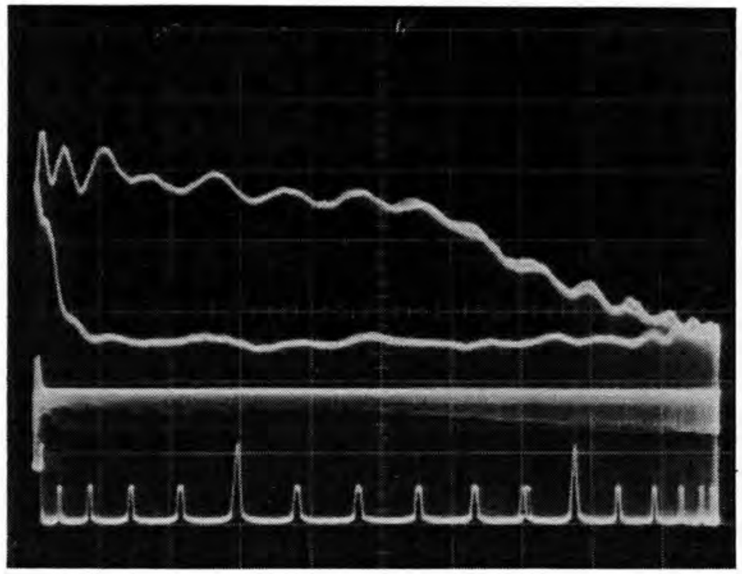
The waste heat temperature was set at the nominal 140°F during most of the testing period. However, one series of tests was made to investigate the effect of varying the hydronic input temperature on the performance of the system. During this test, the hydronic inlet temperature was adjusted by varying the

INLET PRESSURE = 295 psig

DISCHARGE PRESSURE = 160 psig

PRESSURE DIFFERENCE = 135 psi

INDICATED POWER = 19.8 kW



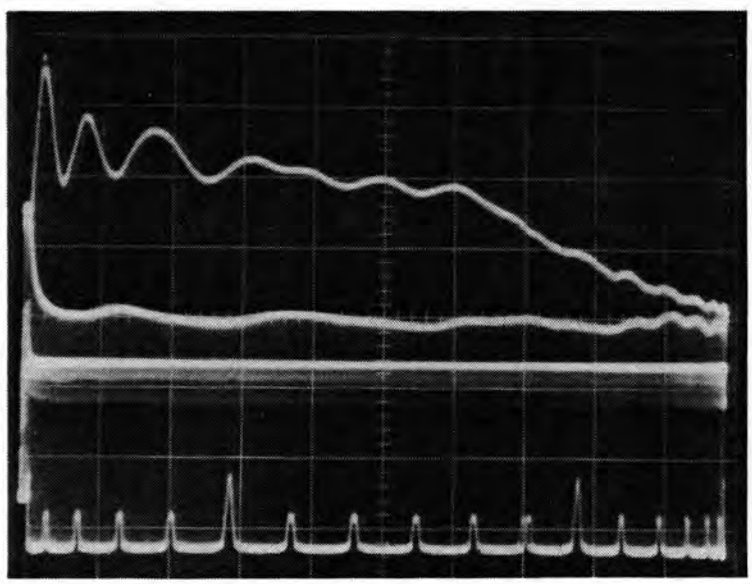
a) VALVE TIMING 0° ATDC

INLET PRESSURE = 288 psig

DISCHARGE PRESSURE = 152 psig

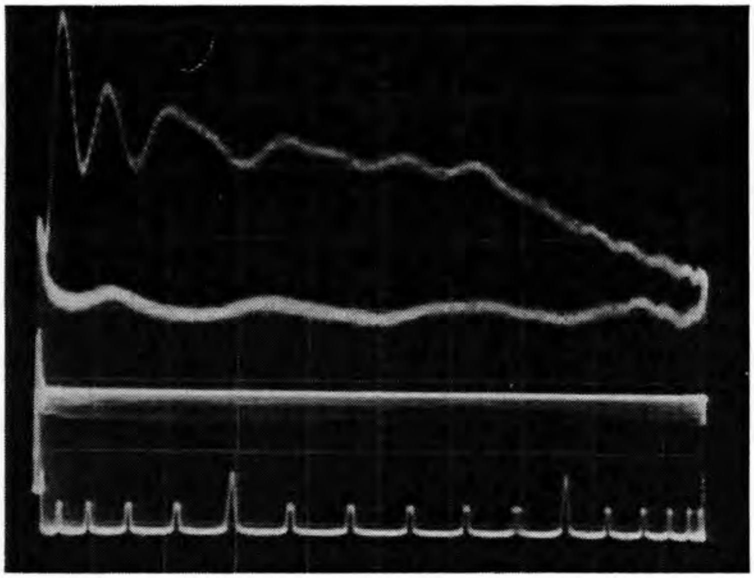
PRESSURE DIFFERENCE = 136 psi

INDICATED POWER = 21.8 kW



b) VALVE TIMING 3° ATDC

Figure 2-2. Effect of Valve Timing on the Weste Heat Expander P-V Diagrams



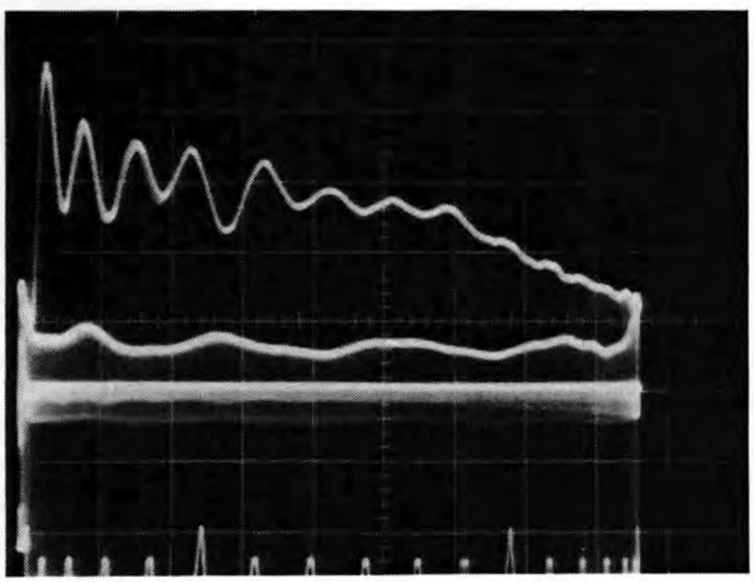
INLET PRESSURE = 290 psig

DISCHARGE PRESSURE = 147 psig

PRESSURE DIFFERENCE = 143 psig

INDICATED POWER = 23.8 kW

c) VALVE TIMING 6° ATDC



INLET PRESSURE = 277 psig

DISCHARGE PRESSURE = 136 psig

PRESSURE DIFFERENCE = 141 psi

INDICATED POWER = 24.5 kW

d) VALVE TIMING 9° ATDC

Figure 2-2. Effect of Valve Timing on the Waste Heat Expander P-V Diagrams (Continued)

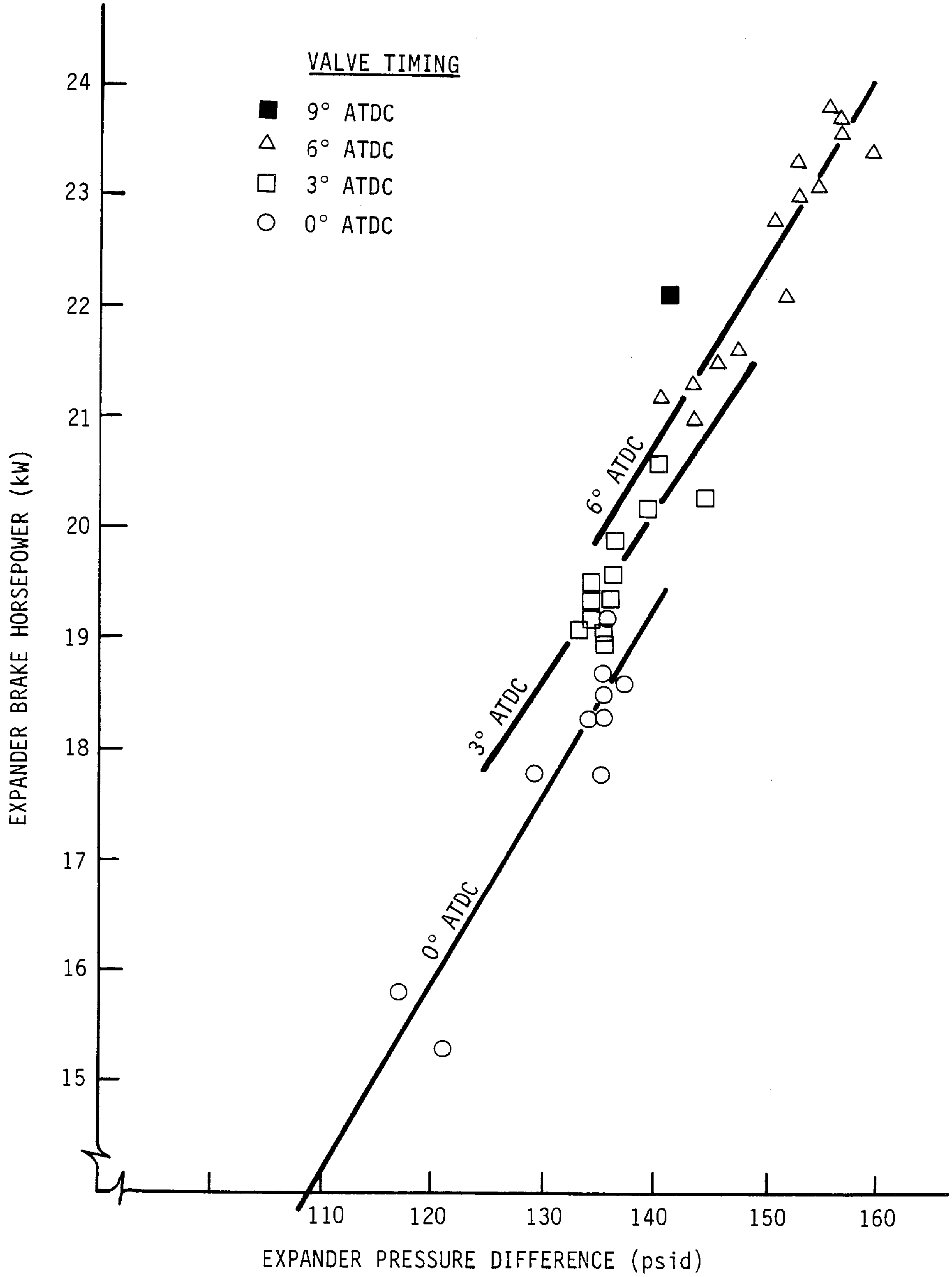


Figure 2-3. Effect of Valve Timing on Expander Performance as a Function of Expander Pressure Difference

firing rate on the water heater burners. The results of this test are shown in Appendix 3, run 3-19. The maximum temperature we could obtain was the nominal 140°F due to the 6° ATDC valve timing we had selected (as explained in subsection 2.1.1).

Figure 2-4 shows the effect of varying the hydronic inlet temperature from 140°F down to 124.5°F. Both system kW output and expander brake power are plotted. The results show strong dependence of expander power output on waste heat temperature.

This effect is more pronounced at higher temperatures indicating the benefits that could be derived by running this system on waste heat sources higher than 140°F.

As expected, the waste heat temperature was found to have no influence on the refrigeration load (between 16-18 tons for this test). The condensing pressure varied very slightly during the test (141 to 138 psig) and the wet bulb temperature was constant at 36°F.

2.1.3 Effect of Ambient Wet Bulb Temperature

An evaporative condenser rejects heat to the ambient wet bulb temperature. Thus, for a given load, the condensing pressure is determined by the wet bulb temperature which is the only factor we could not control during our tests. Our approach to investigating the effect of wet bulb temperature on system performance was twofold: in general, we tried to spread our runs over our testing season to cover as many different ambient conditions as possible; and specifically, we varied the condensing pressure during a dedicated series of tests by regulating the flow of water to the condenser spray nozzle. The results presented in this subsection reflect the outcome of these tests.

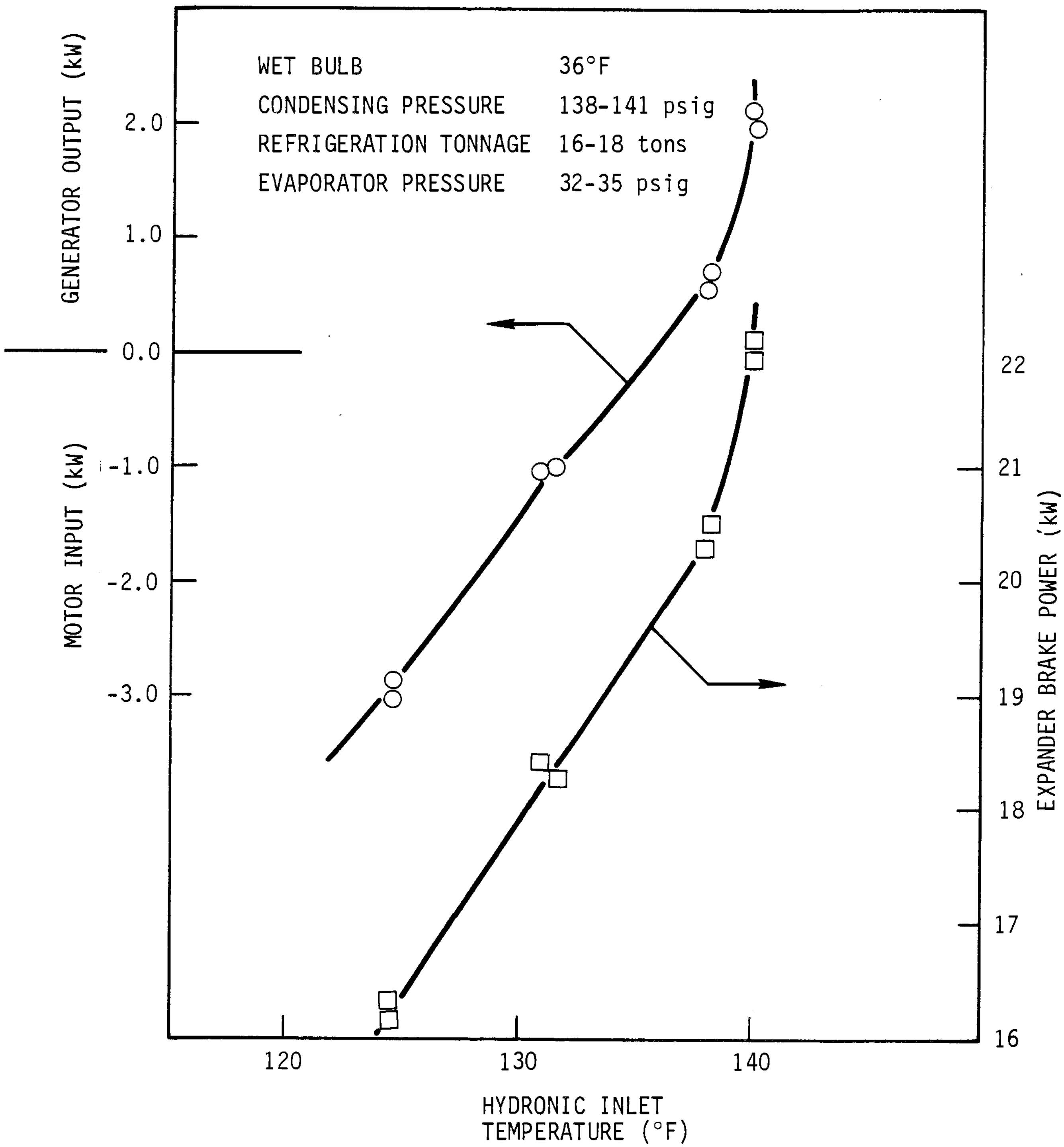


Figure 2-4. Effect of Hydronic Inlet Temperature on the System and Expander Power Output

Table 2-1 shows some of the data taken during diverse ambient conditions. These data represent typical operating points in tests run during the period from July 1982 to March 1983 (see Appendix C). Condensing pressure varied from 133 to 181 psig corresponding to wet bulb temperatures of 31 to 75°F. The refrigeration load in the early tests was limited by the inadequate Freon charge in the system (see subsection 2.3.1.3) and was restored to normal levels as we added more Freon.

Figure 2-5 is a plot of the condensing pressures in Table 2-1 against corresponding wet bulb temperatures. The dependence appears to be linear, but does not correspond to the data published by the manufacturer of the evaporative condenser (see subsection 2.3.1.1).

The effect of condensing pressure on system performance is presented in Figure 2-6. These results were derived from data presented in Appendix C, run 3-23. Net power consumption is shown to vary almost linearly with condensing pressure. Refrigeration load does not show a trend to change with condensing pressure. This is because changing the condensing pressure exerts two opposing effects on the refrigeration compressor:

High condensing pressures, on the one hand, tend to reduce the volumetric efficiency of the compressor curtailing its throughput; on the other hand, they tend to increase the evaporator pressure (due to high pressure differences across the thermal expansion valves - TEVs) thus increasing the Freon density at the compressor's suction and its throughput. In this test, raising the condensing pressure from 150 to 205 psig reduced the compressor volumetric efficiency from 70 to 67 percent and raised the evaporator pressure from 39 to 47 psig. The actual flow rate

Table 2-1. Effect of Ambient Wet Bulb Temperature on Condensing Pressure and System Performance Based on Chronological Data

Date	Wet bulb temperature (°F)	Condensing pressure (psig)	Refrigeration tonnage	System consumption (kW)*
7-15-82	75	181	6.96	-
7-16-82	72	177	10.0	-
10-6-82	63	173	12.0	6.3
11-30-82	45.5	150	18.2	0.3
12-2-82	50	156	18.4	1.8
12-2-82	49	154	18.6	2.0
12-30-82	31	133	19.9	-3.0
3-7-83	37	141	19.5	-1.9
3-7-83	36	138	17.9	-0.6
3-15-83	46	152	19.7	1.6
3-16-83	45	150	18.9	1.0
3-17-83	40	142	18.7	-0.7

*Wattmeter reading: positive values are net consumption, negative values are net generation, missing values are due to faulty wattmeter.

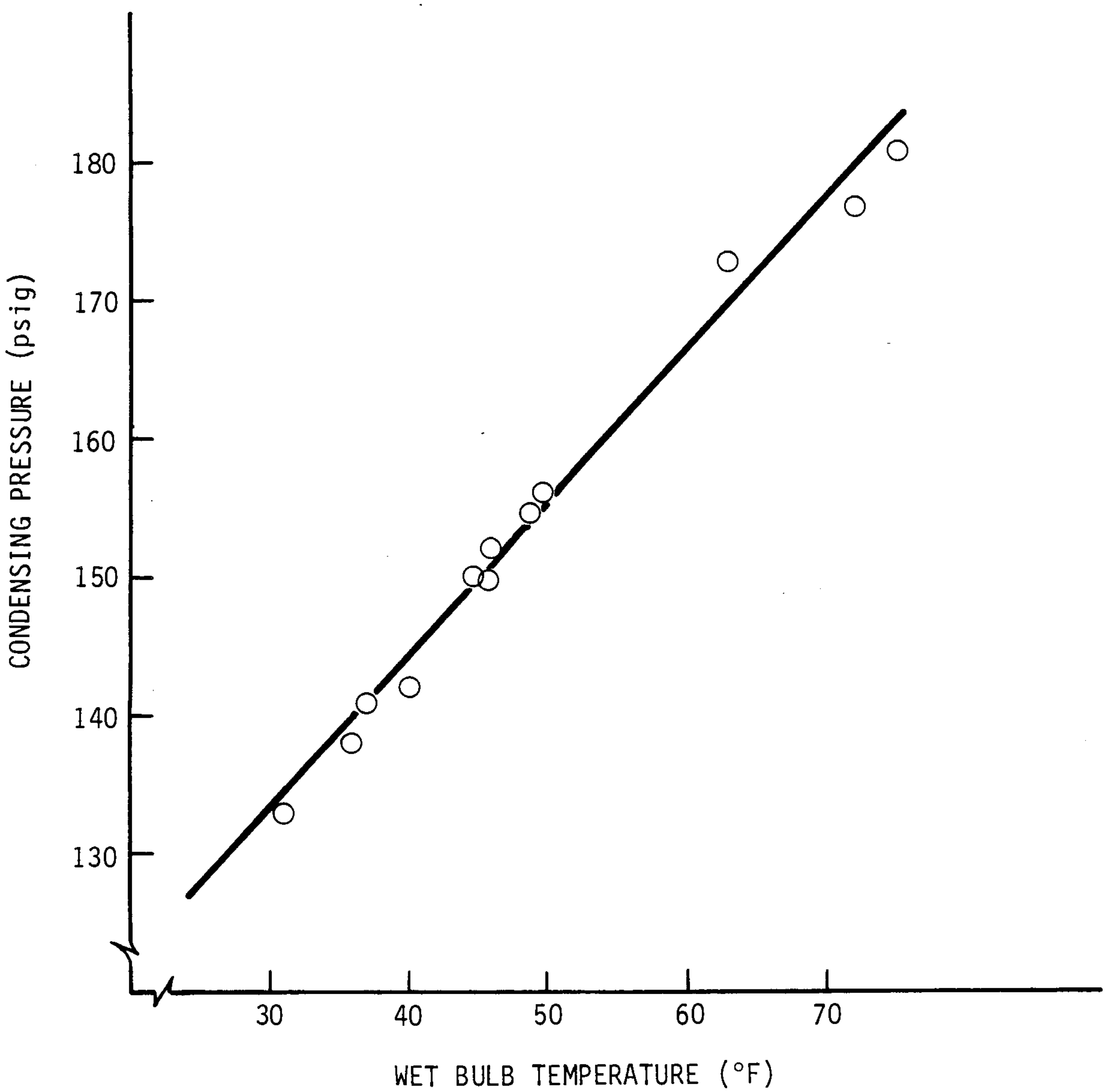


Figure 2-5. Wet Bulb and Condensing Pressure Correspondence for the Waste Heat Expander System

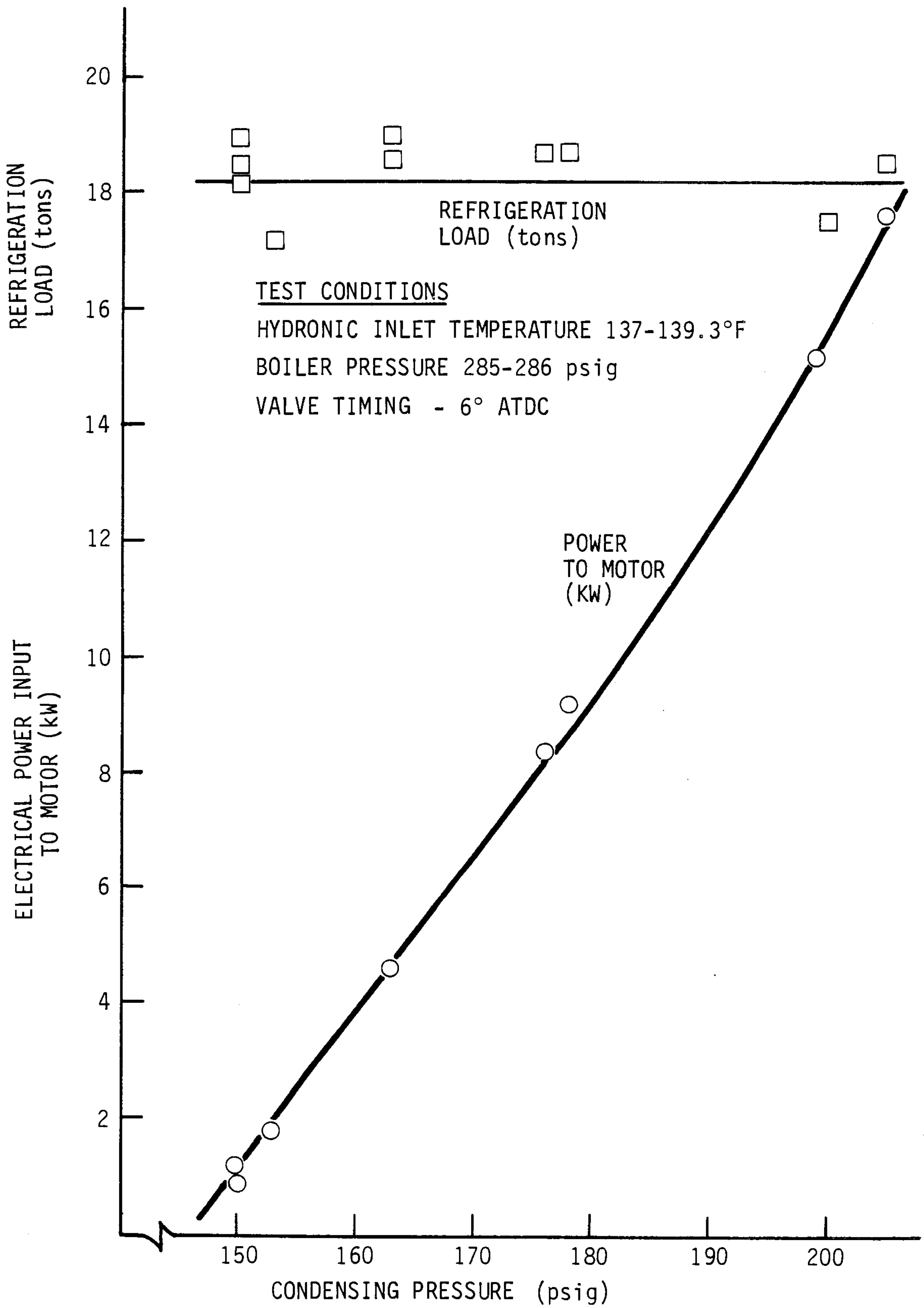


Figure 2-6. Effect of Condensing Pressure on System Performance

remained constant at approximately 2900 lb/hr as did the refrigeration effect at about 77 Btu/lb. Figures 2-5 and 2-6 can be used to predict the performance of the system under a broad range of ambient conditions.

2.1.4 Effect of Average Brine Temperature

Brine temperature controls the heat input to the brine loop which in turn determines the evaporator pressure and the refrigeration load. In our system we controlled the brine temperature by regulating the flow through the hydronic coil throttle valve (see Appendix B). This flow determines the heat input to the brine loop which, at equilibrium, must balance the heat rejected in the evaporator. When the valve is throttled down, the heat input to the brine loop is reduced and the evaporator temperature drops until the compressor capacity is lowered enough to match the reduced heat duty of the evaporator. This effect is accompanied by a drop in average brine temperature.

Figures 2-7 and 2-8 show the effect of closing the throttle valve on the refrigeration load and system power consumption. In Figure 2-7 the refrigeration load is plotted against the average brine temperature. The correlation is weak primarily because of large fluctuations in the evaporator temperature caused by the "hunting" (see subsection 2.3.3) of the expansion valves (TEVs). Brine thermal temperature follows the fluctuations in the evaporator temperature to keep the refrigeration load constant. During our tests, the average brine temperature floated over a range of approximately 8°F while the difference between the brine inlet and outlet temperatures, and hence the refrigeration load, remained constant. Figure 2-7, despite the weakness discussed above, does indicate a general trend in the relationship between refrigeration load and average brine temperature. Power consumption is also weakly correlated, yet the general trend is visible.

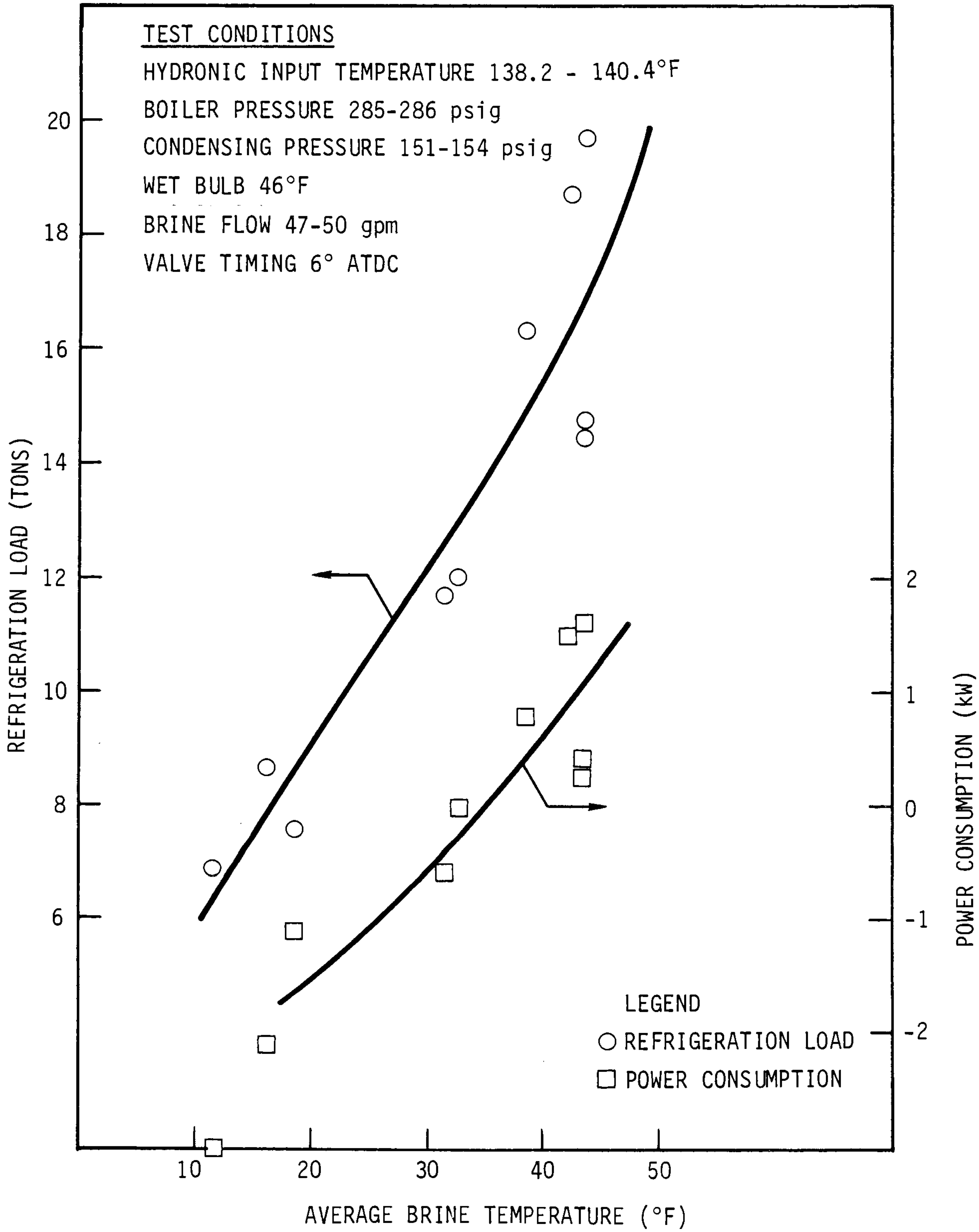


Figure 2-7. Effect of Average Brine Temperature on Refrigeration Load and System Power Consumption

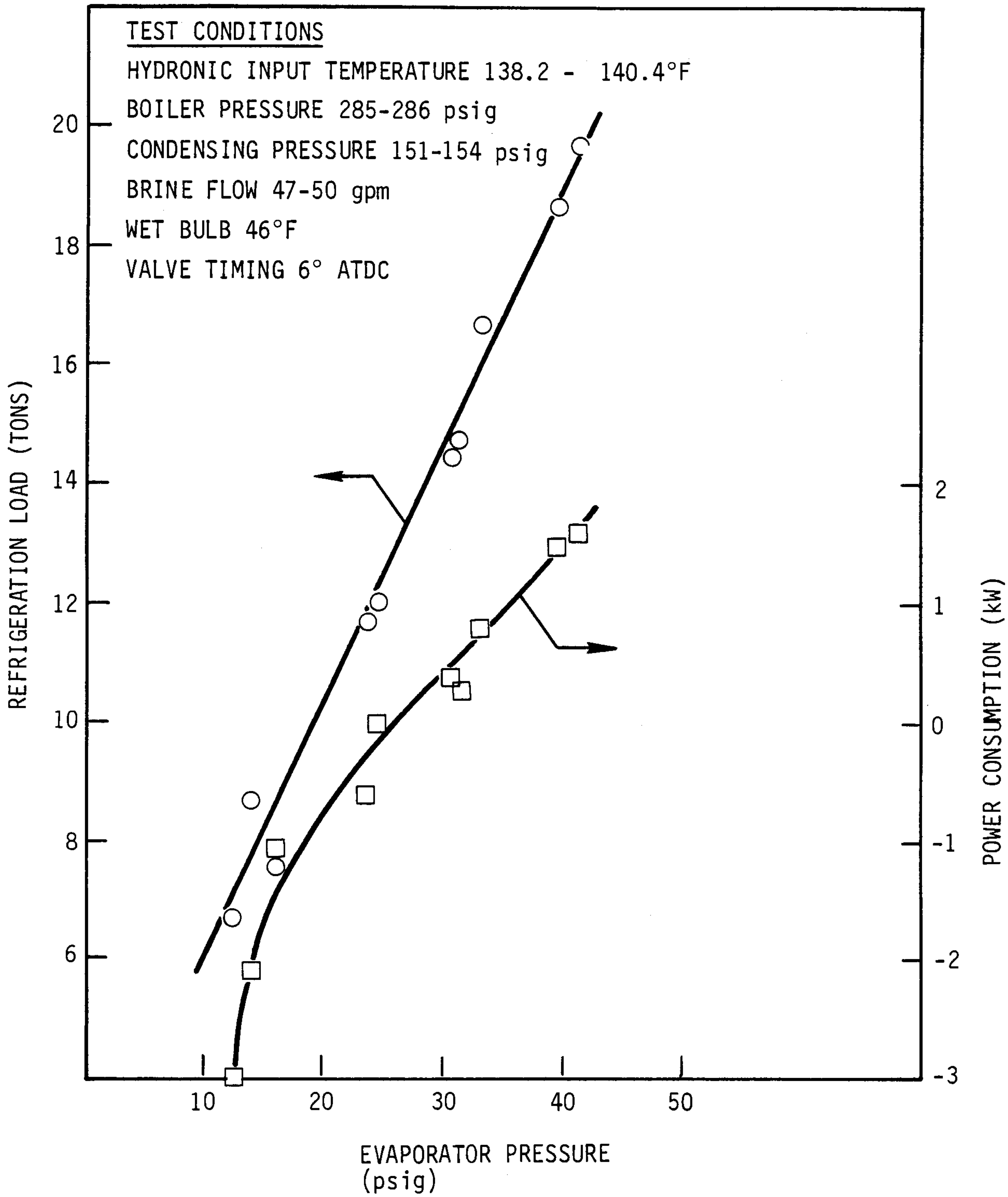


Figure 2-8. Dependence of Refrigeration Load and System Power Consumption on Evaporator Pressure

A better correlation is observed between refrigeration load and evaporator pressure in Figure 2-8. Evaporator pressure is not as strongly affected by the TEVs' hunting as is evaporator temperature and, hence, does not fluctuate as much. The results show a linear dependence between refrigeration load and evaporator pressure. The power consumption, however, does not exhibit a linear dependence. It shows a stronger dependence at lower evaporator pressures than at higher ones. This is due to the fact that the coefficient of performance (or COP: the ratio of evaporator heat input to shaft power input) of the refrigeration subsystem increases as the evaporator pressure is raised. In this test, the COP rose from a value of 2 to about 4 as the evaporator pressure was increased from 12.5 to 41 psig. This indicates the advantage of running at higher evaporator pressures. Figures 2-7 and 2-8 were obtained from test data presented in Appendix C, run 3-21.

2.1.5 Comparison of Test Results with Predicted Performance

In the previous sections, we discussed the effect of the external variables that control the performance of the Phase III waste heat driven brine chiller. In this subsection, we shall compare those results with the predictions made by computer simulations during Phase II of this project. Those predictions were presented in Section 5 of the Phase II Progress Report (1), and are reproduced in Table 2-2 for comparison. Test results presented in Table 2-2 were obtained from data in Appendix C, runs 3-16, 3-17, 3-18, 3-19, 3-21 and 3-23. These same data were used to produce Figure 2-9 which shows the effect of condensing pressure on expander brake power output. The actual expander power output values in Table 2-2 are taken from Figure 2-9. As shown in this table, condensing temperatures of 55 and 65°F were not obtained during actual runs. This is partially due to the fact that the condenser's heat rejection rate was lower than expected (see subsection 2.3.1.1). Also, the winter of 1982-83 had very few extremely cold days (in the

Table 2-2. Comparison of Actual Results with Those Predicted by Computer Simulations

Condensing temperature °C (°F)	Condensing pressure (psig)	Expander output (kW)	Compressor input (kW)	Feed pump input (kW)	Net power to generator (from motor) (kW)	Balance after pump and fan (kW) (2)	Net system output (kW) (3)
Predicted results							
35 (95)	182	13.0	18.5	1.7	(7.2)	(13.9)	-
29 (85)	156	17.7	17.2	2.1	(1.6)	(8.3)	-
24 (75)	132	22.2	15.7	2.4	4.1	(2.6)	-
18 (65)	111	25.9	13.8	2.7	9.4	2.7	-
13 (55)	93	29.3	11.6	3.0	14.7	8.0	-
Actual performance							
35 (95)	182	12.7	19.4	2.8	(9.5)	(16.9)	(18.0)
29 (85)	156	19.0	17.5	3.3	(1.8)	(9.2)	(10.3)
24 (75)	132	23.4	15.8	3.6	4.0	(3.4)	(4.5)
18 (65)	111	- (1)	-	-	-	-	-
13 (55)	93	-	-	-	-	-	-

(1) Missing values correspond to condensing temperatures never obtained during testing

(2) Condenser pump and fan power consumption: Predicted = 6.7 kW
Actual = 7.3 - 7.4 kW

(3) Parastic losses in motor/generator and controls = 1.1 kW.

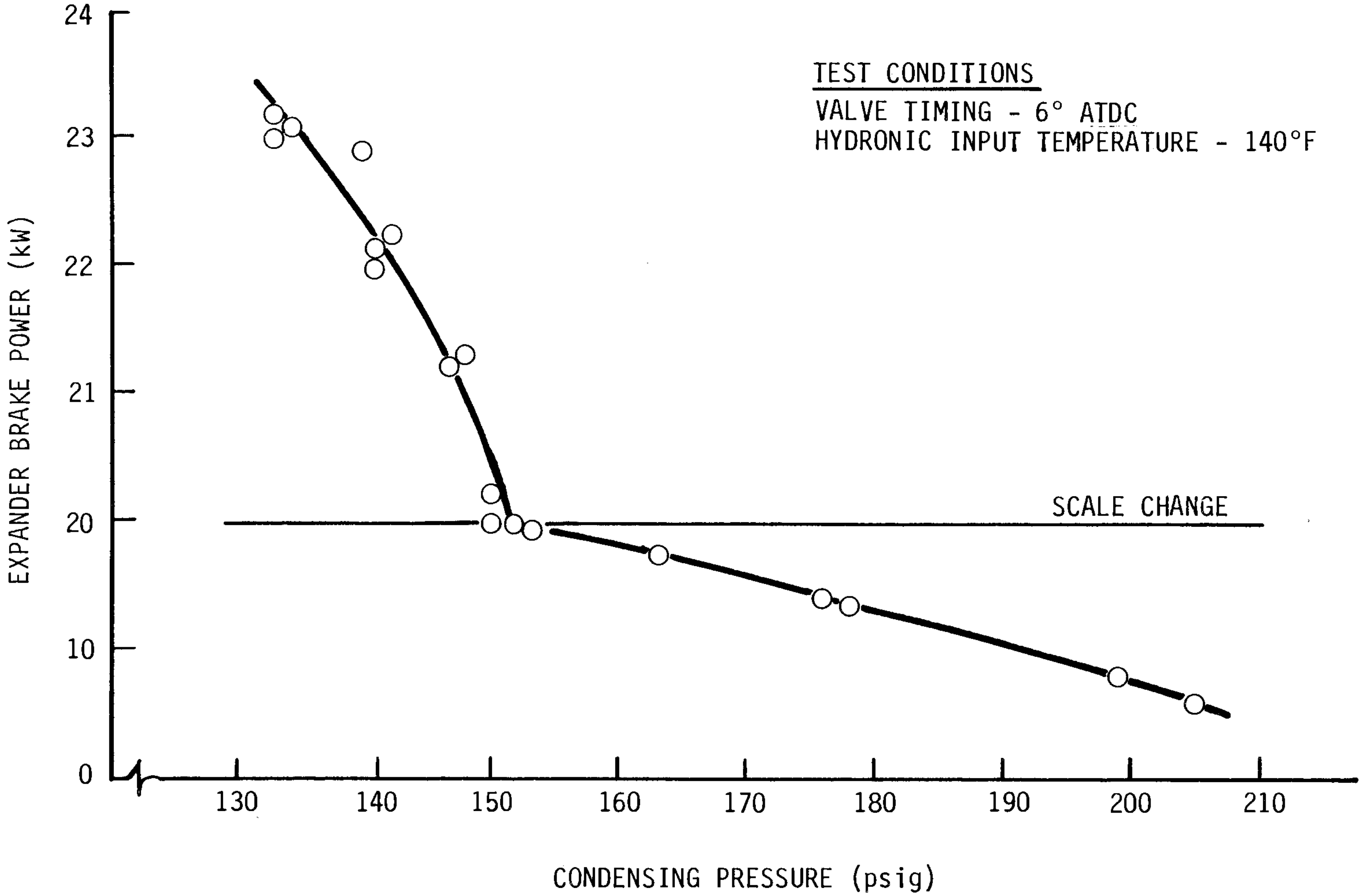


Figure 2-9. Effect of Condensing Pressure on Expander Power Output

Boston area) and we never got to run the system at a wet bulb temperature lower than 31°F.

Considering the fact that in our computer simulations, we underestimated the power requirement of all the components (compressor, feed pump, condenser fan and pump) and neglected other losses (controls, motor/generator no load power), the actual results compare favorably with the predicted values. The actual expander power output is higher than predicted, except for high condensing temperatures where they are almost equal. This is because the expander mechanical efficiency is generally better than 80 percent, which is the value assumed in the computer analysis. The mechanical efficiency is high at low condensing pressures (88 percent at 132 psig) and drops (to about 83 percent at 182 psig) as the pressure rises. This explains the trend in the expander output results. These results are even more favorable considering that the predicted values were calculated for a valve timing equivalent to 15° ATDC. The actual values were measured at 6° ATDC. As explained in subsection 2.1.1, as the valve timing is retarded, the expander power output increases. That the 6° ATDC expander output should exceed the results predicted for 15° ATDC, shows that the assumptions made in the computer simulations were conservative and that the expander performs much better than predicted.

The compressor input column was extracted from York literature on the power consumption of the RS44A compressor. The predicted results were based on a 15°F evaporator temperature, while the actual results are based on actual evaporator temperatures. Since the latter are higher than 15°F at high condensing temperatures, the power required is more than predicted under those conditions. This is because at high evaporator temperatures and pressures, Freon density at the suction and the compressor mass flow rate are greater raising the compressor's power consumption.

The computer simulations totally ignored the power requirement of the inducer end of the feed pump. (Probably because the necessity of adding an inducer was not demonstrated when these programs were written). For this reason, the feed pump power was substantially underrated.

The evaporative condenser's pump and fan power were also underestimated. This was due to the fact that the predicted values for pump and fan power were measured when we first received our wattmeter. As we later discovered, one of the transformers in the wattmeter was malfunctioning, producing erroneous readings on the wattmeter display. By coincidence, these erroneous readings were in the expected range and we did not discover the wattmeter problem until after the Phase II report was published. The actual power consumed by the pump and fan is about 7.4 kW compared to 6.7 kW used in the calculation of predicted performance.

Despite the increase in the actual values of parasitic power consumption, the balance of power, after the fan and pump are accounted for, corresponds closely to the predicted values. In fact, for the lower condensing pressures, they are within 1 kW, or 3 percent of the expander output. The results correspond to predictions even better, before the power consumed by the condenser pump and fan is considered. This can be observed in the "Net Power to Generator" column, where for 75 and 85°F condensing temperatures the actual power reading is within 0.2 kW of the predicted values. The net power generated by the system is further reduced by parasitic losses such as the power lost in the motor/generator (0.6 kW) and the power consumed in the control panel and other controls ($\approx$0.5 kW). These losses were completely ignored in the predicted values but were included in the actual results to illustrate the net power output of the system.

In general, the correspondence between predicted results and actual performance is striking. These results show that, at least from a technical standpoint, the rotary valve Freon expander concept was a sound and valid idea for extracting energy from low temperature waste heat. It is encouraging that the expander performance was better than predicted.

2.2 Auxiliary Systems

2.2.1 Hot Gas Bypass Capacity Control

The inclusion of the hot gas bypass regulator proved to be an effective means of controlling the refrigeration capacity. This pressure regulator bypasses some of the compressor exhaust gas to the compressor suction side (see Figure B-1), when suction pressure falls below the regulator setpoint. This is useful when it is required to keep the evaporator above a certain temperature and pressure as the heat input to the brine loop is reduced. The minimum temperature setpoint depends on the process for which the chilled brine is intended and varies for different applications. In our tests, the minimum setpoint used was 0°F evaporator temperature (24 psig evaporator pressure), since our 30 percent brine solution would freeze at about 0°F.

The first point in Table 2-3 shows what could happen when the hot gas bypass line is valved off. In this case, the brine loop heat load has fallen to the point where the evaporator pressure has dropped to 18 psig (-8°F). If this condition continues, the brine would freeze. Opening the ball valve in the hot gas bypass line, thus enabling the hot gas bypass line to function, raises the pressure to a value close to the setting of the regulator, as indicated in points 2 and 3. As shown, the process is accompanied by a drop in refrigeration tonnage and some reduction in system power output.

The decrease in tonnage takes place in spite of the increase in evaporator pressure and is due to the reduced effective flow rate through the evaporator. An increase in evaporator pressure, increases the refrigeration effect (enthalpy rise across the evaporator) and under any other circumstances would raise the refrigeration capacity of the system. The system power output suffers because the density at the compressor suction increases

Table 2-3. Hot Gas Bypass Capacity Control Results

Point	Time Relative to Point 1 (min)	System Power Output (kW)	Refrigeration Tonnage	Evaporator		Explanatory Notes
				Pressure (psig)	Saturated Temperature (°F)	
1	0	1.2	10.2	18	-8	Evaporator at partial load, hot gas bypass line closed
2	5	1.0	7.36	31	8	Evaporator at partial load, hot gas bypass line open, regulator set at about 30 psig
3	8	0.6	8.18	33	10	As in point 2, pressure floating
4	21	1.3	8.8	24	0	As in point 2, hot gas bypass regulator set at about 25 psig
5	31	-1.2	18.76	38	15	Evaporator full load, hot gas bypass line open but not operating

which results in higher compressor power consumption. The higher the pressure setting on the regulator, the larger the penalty paid in the refrigeration load. For this reason it is desired to set the regulator at the minimum evaporator pressure permissible. Point 4 illustrates the effect of lowering the regulator setting to about 25 psig. The tonnage increases as expected and the loss in power output is recovered. Point 5 shows the result of increasing the heat input to the brine loop to normal levels. As the pressure rises, the hot gas bypass regulator closes and no more penalty is paid in refrigeration tonnage. Thus, the load approaches normal values (18.7 tons) and the system operates at its full potential with a relatively small reduction in power output.

2.2.2 Compressor Unloader

One of the two compressor cylinders is equipped with a hydraulic unloading mechanism. This is a modification, as the compressor was made with two cylinders out of four unloading. The unloader lifts the suction reed valves and keeps them open, effectively eliminating one cylinder from the refrigeration circuit. The unloader is actuated when either of the following conditions are present:

- Low oil pressure - When the compressor oil pump discharge pressure is low (this happens when the compressor speed drops), the unloader is activated. Thus, during startup one cylinder is unloaded until oil pressure builds up.

- Low suction pressure - A pressure switch at the compressor suction activates the unloader when suction pressure drops below its setpoint.

Both these mechanisms have their merit. The first reduces the startup load and the current drawn by the motor by a factor of 1.83 (motoring load is not halved because of the remaining breathing losses in and out of the cylinder and piston friction). The second can be used as a method of capacity control. This method has one disadvantage and one advantage over the hot gas bypass technique; the disadvantage is that it is not a continuous means of capacity control and, for a given evaporator temperature, halves the refrigeration capacity. The advantage is that very little energy is wasted in unloading. Whereas in bypassing hot gas, that portion of compressor energy spent in compressing the bypassed gas is completely wasted. Thus, the compressor power consumption drops dramatically when the capacity is halved using the unloader. In run 3-14 in Appendix C, going from full to half load using the unloader, reduced the refrigeration capacity from 18.3 to 8.7 tons but increased the power output of the system by 8.3 kW. Reducing the refrigeration load by the same amount using the hot gas bypass method (Table 2-3 points 5 and 4) increased the power output by only 2.5 kW.

In the absence of large load fluctuations, the unloader can be used as a safety precaution. In this case, the unloader pressure switch should be set at the minimum suction pressure permissible (24 psig in our system).

The unloader performed troublefree during the entire testing period. It was found necessary during startup and very desirable for controlling refrigeration capacity in the presence of large load fluctuations or for safety reasons.

2.2.3 Thermal Expansion Valve Augmentation and Oil Let Down Lines

On cold days when condensing pressure is low, the pressure difference across the subcooler falls to levels insufficient to maintain the required flow of liquid freon to the evaporator TEVs.

This condition starves the evaporator and sharply reduces the refrigeration tonnage. Even if the subcooler was partially bypassed, at very low condensing pressures, the pressure drop across the TEVs would restrain the flow and starve the evaporator. In such cases, the TEV augmentation line supplies the evaporator with additional liquid. The thermal expansion valve in this line (see Figure B-1) is set to open when evaporator discharge is about 7 deg superheated. The above effects are shown in Table 2-4. Point 1 represents system performance under normal conditions during a relatively cold day at a wet bulb temperature of 40°F. When, as in point 2, the subcooler bypass line is completely shut, the evaporator is starved resulting in a drop of 6.6 tons in refrigeration capacity. This is an extreme case since the subcooler should be partially bypassed during cold weather operation (See subsection 2.3.1.4 and Appendix A - Operational Adjustments). Yet this condition helps to illustrate to full extent the capacity of the TEV augmentation line. Thus in point 3, the TEV augmentation line ball valve is opened and an increase of about 2.5 tons is registered in refrigeration tonnage. The substantial superheat of 64°F helps maintain the TEV in the TEV augmentation line completely open.

The oil let down line is identical in operation to the TEV augmentation line. However, its purpose is not solely to increase refrigeration capacity in case of evaporator starvation. The difference between the TEV augmentation and oil let down lines is their source of liquid freon. The TEV line is connected to the boiler feed pump discharge, the coolest point where high pressure liquid is to be found. The oil let down line, however, receives its liquid from the lower half of the freon boiler, just below the liquid-vapor interface. Its function is to route the stagnant oil on top of the liquid level to the evaporator so that the oil can find its way back to the crankcase, where it belongs. As shown in point 4, the oil let down line does not increase the refrigeration capacity as much as the TEV line does (2.1 compared to 2.5 tons). This has two causes: the supply of liquid to the

Table 2-4. Performance of the Thermal Expansion Valve Augmentation and Oil Let Down Lines

Point	Time Relative to Point 1 (min)	System Power Output (kW)	Refrigeration Tonnage	Evaporator			Notes OLD = Oil Let Down TEVA = TEV Augmentation
				Pressure (psig)	Outlet Temp. (°F)	Superheat (°F)	
1	0	0.2	19.5	42	20	1	Evaporator full load, subcooler bypassed, TEVA and OLD lines valved off.
2	25	2.8	12.9	23.5	71	71	As in point 1 except subcooler not bypassed.
3	41	2.0	15.4	31	64	56	As in point 2 except TEVA line operating.
4	49	1.8	15.0	30	64	57	As in point 2 except OLD line operating.
5	65	1.25	17.4	37	54	40	As in point 2 except both TEVA and OLD lines operating.
6	75	0.3	19.0	42.5	21	1.5	As in point 1 except TEVA and OLD lines open.

oil let down line is hotter, reducing the refrigeration effect; and the oil concentration is higher reducing the effective freon flow.

When both the TEV augmentation and oil let down lines are operating - point 5 - their effects are added with a resulting 4.5 ton increase in tonnage. This is because the 40°F superheat warrants the full operation of both TEVs in both lines. Point 6 is interesting in that it shows the TEVs in both the augmentation and oil let down lines closing automatically as the superheat is dropped to 1.5°F. This conclusion can be seen by comparing point 6 to point 1 where the ball valves in both lines are closed. The tonnage is slightly less in point 6, contrary to what would happen if the TEVs were operating.

The power output values in Table 2-4 are also of interest. Even though the power output increases as the refrigeration load drops, the gain in power output is completely overshadowed by the loss in refrigeration. This is as expected, but it is of interest to see how the numbers support this argument. Compare points 1 and 2: the first with the lowest power output and highest refrigeration, the second exactly the opposite. In going from point 1 to point 2, we gain 2.6 kW power output at the cost of 6.5 tons of refrigeration.

The oil let down line's main function of oil separation could not be easily evaluated. The oil separator seemed to work very effectively and the crankcase oil level remained steady during the testing period. It is left to more long-term operation or testing to determine the effectiveness (and indeed the necessity) of the oil let down line as an oil separator.

2.3 Components

2.3.1 Heat Exchangers

2.3.1.1 Evaporative Condenser

The evaporative condenser we selected was an Evapco Power-Mizer Model PMC-185, with vane axial fans and a rated capacity of 1,519,000 Btuh at 95°F condensing temperature and 80°F ambient wet bulb (see Phase II Progress Report, subsection 5.3.4 (1)).

Qualitatively, the evaporative condenser performed as expected. The pressure switch controls turned the fan and the spray pump on as the condenser pressure rose. Freezing the water in the spray pump and piping was avoided by draining the sump on cold nights. The freon charge in the condenser was determined by the requirements of the feed pump and the evaporator (see subsections 3.1.1 and 2.3.1.3).

During Phase III testing, the condenser was re-instrumented for the specific purpose of measuring its performance. A turbine flow meter was installed at the condenser outlet refrigerant piping to measure freon flow (see Appendix B). The spray flow was measured by installing a paddle wheel flow meter downstream of the spray pump. A differential pressure gauge, 1 in H_2O full scale, was installed to evaluate the pressure drop across the tube bundle. Test holes were drilled at different heights to measure the water temperature at the top (sprays), middle (just below the tubes) and bottom (the sump and just above the sump) of the condenser. To vary the condensing pressure, we installed a throttle valve at the spray pump discharge. This valve regulates the water flow through the sprays and enables the operator to smoothly adjust the condensing pressure. The range of adjustment is limited by the condensing pressures corresponding to the wet bulb and dry bulb temperatures.

Table 2-5 represents data taken during a condenser test. The actual heat rejection rate is 14 to 19 percent lower than the manufacturer's data. This loss in heat rejection can be attributed to contaminated freon and/or liquid level buildup in the condenser. In our case, the latter cause seems unlikely since we charged the system in measured steps, to levels just adequate for proper operation (see subsections 3.1.1 and 2.3.1.3). Freon contamination is the probable cause for this discrepancy. The system was opened to the atmosphere countless times and it was left open for long periods of time. Even though, during such periods, all holes were covered with plastic sheets, the system was not sealed to air, and moisture and other contaminants probably entered the system. Also, when cold weather burst some

Table 2-5. Evaporative Condenser Performance

Test Date: 11/23/82					
Point	1	2	3	4	5
Time relative to point 1 (min)	0	12	21	33	42
Condensing Temperature (°F)	76	80	81	81	81
Wet Bulb, when measured (°F)	52		52		
Rated heat rejection (MBH)	1.90	2.09	2.14	2.14	2.14
Actual heat rejection (MBH)	1.59	1.70	1.80	1.84	1.84
Percent difference between rated and actual heat rejection rates	-16.3	-18.7	-15.9	-14.0	-14.0

MBH = Million Btu/hr

tubes in the freon boiler (see subsection 2.3.1.2), some water found its way into the freon side of the system. This water was flushed and further pumped out, using the vacuum pump, for over a week. However, it is possible that the evacuation was not complete.

The presence of contaminants such as water, can reduce the heat rejection of the condenser dramatically. Water molecules are deposited on the cold surface of the condensing tubes and form a barrier against the flow of heat from the refrigerant to the tubes. The sales engineer from Evapco was of the opinion that, in time, we had contaminated our freon. This conclusion is also supported by some data we took prior to the official testing period of the system. During this period the actual heat rejection and the rated value were only about 6 percent apart (1.84 rated versus 1.74 actual MBH). This discrepancy grew in time, to the point that toward the end of the testing period the actual heat rejection was as much as 40 percent lower than the rated value. The increasing freon contamination during this period would explain the time-dependent nature of the results.

The spray pump's throughput was 160 to 165 gpm. The spray water temperature was found to stay almost constant as it dropped in the condenser. The make-up flow varied between 2.2 to 2.6 gpm. Tube bundle pressure drop was in the range of 0.7 to 1.0 in. of water. We measured an air flow of 26,300 cfm compared to the manufacturer's rating of 27,900 cfm.

Apart from the discrepancy in heat rejection values, the condenser performed well and had no mechanical problems.

2.3.1.2 Freon Boiler

The freon boiler was originally manufactured as a water cooled refrigerant condenser with a nominal 200 ton capacity. Hence, the heat transfer information available from the manufacturer was limited to condensing heat rejection data. We performed independent heat transfer calculations on the unit and picked the size conservatively. The unit selected is a two-pass Bell and Gossett, Model RCF-200-82 condenser. It has a total heat transfer area (based on the inside diameter) of 289.6 ft^2 of finned copper tubing with a fin-to-tube area ratio of 3.2. The rationale behind selecting this unit is described in the Phase II Progress Report (1), subsection 5.3.2.

The freon level in the boiler was controlled by a float level switch (see Appendix B). The optimum level was determined to be at the boiler mid-height. At this level, the total tube area submerged in liquid freon is 149.3 ft^2. The results shown in Table 2-6 are based on this heat transfer area.

As shown in Table 2-6, the boiler operates at a logarithmic mean temperature difference (LMTD) of 5.8 to 9.8°F depending on the waste heat inlet temperature. These results were obtained from test data in Appendix C, run 3-19. The effective overall heat transfer coefficient times area (UA) of the boiling section varied between 135,000 and 213,000 Btu/°F hr. Using conservative heat transfer coefficients, we had estimated an LMTD of 18°F under similar conditions to point 1 in Table 2-6. The results show that, as expected, the actual heat transfer coefficients are higher than estimated and that the boiler is adequately sized to perform its duty.

Table 2-6. Effect of Hydronic Inlet Temperature on the Freon Boiler Performance

Point*	Hydronic			Boiler			LMTD	UA
	Inlet Temperature (°F)	Outlet Temperature (°F)	Flow (gpm)	Pressure (psig)	Temperature (°F)	Heat Load (MBH)	(°F)	(1000 Btu/ ft^2 °F)
1	140.0	132.0	365	289	128	1.46	7.28	200
2	139.9	132.2	360	289	128	1.39	7.39	188
3	138.5	131.1	360	275	124.5	1.33	9.84	135
4	138.0	130.5	360	275	124.5	1.35	9.25	146
5	138.2	130.8	360	275	124.5	1.33	9.53	140
6	132.1	125.0	355	263	121	1.26	6.96	181
7	131.4	124.4	345	261	120.5	1.21	6.81	178
8	131.0	124.0	355	260	121	1.24	5.81	213
9	124.5	118.0	355	241	114.5	1.15	6.19	186
10	124.5	118.0	350	241	114.5	1.14	6.19	184

*Run 3-19, Date: 3/8/83 (Appendix C).

The control scheme of coupling the level control switch to the pump bypass solenoid valve proved an effective means of maintaining a level in the boiler (see Phase II Report (1), subsection 5.1.2). The pump bypass throttle valve (shown in Figure B-1), once set, kept a steady liquid level in the boiler and prevented it from fluctuating between the high and low set points of the level switch.

Boiler pressure was maintained at the nominal 285 psig by the boiler bypass regulators. These valves performed reliably with the minor problem of making noise at low throughputs. The nominally open expander bypass solenoid valve proved to be indispensable, as it dumped the boiler pressure to the condenser when the motor/generator stopped for any reason.

The only problem we experienced with the boiler was the bursting of tubes caused by freezing water. This happened during cold weather when the system was not in operation. Even though we drained the boiler after each shutdown, some of the tubes did not completely drain as the boiler was level. The burst tubes leaked freon ito the hydronic water when the system was under pressure, and allowed water in the system when it was open to atmosphere (no freon pressure present).

The leaking tubes were plugged at both ends using brass heat exchanger plugs. We kept the tubes from freezing during cold nights by running the hydronic pump and one of the water heaters. During long-term shutdowns, the boiler was first drained and the remaining water in the tubes was blown out using compressed air. We found that for this method to be effective, it was required to blow air for at least an hour so that no water would remain in the boiler.

2.3.1.3 Evaporator

The evaporator installed in our refrigeration system is a Standard LDX-21A chiller barrel (1). It is a dual circuit, two pass unit used in a direct expansion configuration (refrigerant inside tubes).

As previously shown in Table 2-1, during our early tests the refrigeration capacity was lower than the expected 18 to 20 tons. This was caused by inadequate freon charge in the system. Low freon charge causes the evaporator to starve and even lower freon charge is manifested by the feed pumps cavitation (see subsection 3.1.1). By adding freon to the system, we resolved the cavitation problem and restored the refrigeration capacity to normal levels. The freon charge was added in small, measured steps to avoid over filling the condenser (see subsection 2.3.1.1).

The refrigeration capacity of an evaporator, for a given process stream flow rate, is a function of two factors, range and approach:

- Range is the difference between the process stream inlet and outlet temperatures.
- Approach is the difference between the process stream outlet temperature and evaporator temperature.

For a given process stream flow rate, refrigeration capacity increases with an increase in range or approach.

The LDX-21A has a nominal 21 ton refrigeration capacity. For water as the process stream liquid, this translates into a water flow of 50 gpm at a range of 10°F and an approach of 9°F.

When we elected to use a 30 percent ethylene glycol brine solution (to protect the system from freezing), we had to run the system at a considerably higher approach. Antifreeze solutions have lower heat transfer coefficients compared to water and require a higher temperature drive for the same heat duty. Table 2-7 illustrates this point. The values for approach range from 12.2 to 31.7°F showing a trend of increasing in time. During early tests (3-08, 3-13 in Appendix C), the evaporator approach corresponded to values quoted by the manufacturer for 30 percent brine service. Progressively, in time the actual approach increased to values higher than predicted by the manufacturer's data. This is most likely due to the contamination of the freon and the fouling of the tubes with time.

As shown in Table 2-7, the approach is not steady during a run and varies considerably. This phenomenon parallels the discussion in subsection 2.1.4 where it was shown that brine temperature correlates weakly with refrigeration capacity. This might be caused by the TEVs' "hunting" (alternate overfeeding and underfeeding of the evaporator by the TEVs). Due to the physical distance between the TEV and its temperature sensing element (bulb), the effect of the TEV flow regulation is felt at the bulb after some time lapse. This phase lag between the control efforts of the TEV and its bulb, prevents the system from stabilizing at a single operating point and causes it to cycle about an equilibrium point. To illustrate this, consider Figure 2-10, where the time variations of the temperature at the inlet of the evaporator TEVs is plotted. This temperature fluctuates with the flow rate through the TEVs. As the flow drops (the TEVs cycle closed) the liquid flow is subcooled to about 55°F in the recuperator. As the flow picks up (the TEVs cycle open), the recuperator effectiveness drops and the inlet liquid is cooled only down to 60°F. The fluctuations in the inlet temperature,

Table 2-7. History of Evaporator Performance

Date	Run	Point	Brine			Evaporator			
			Flow	Temperatures		Temperature	Capacity	Range	Approach
			(gpm)	In (°F)	Out (°F)	(°F)	(ton)	(°F)	(°F)
10/6/82	3-08	1	69.5	32.8	27.2	14	15	5.6	13.2
11/30/82	3-13	1	53	38.1	29.2	17	18.2	8.9	12.2
11/30/82	3-13	2	48	40.6	30.8	17	18.2	9.8	13.8
12/2/82	3-14	5	51	40.8	31.3	16	18.7	9.5	15.3
12/2/82	3-14	6	50	44.2	34.6	17	18.5	9.6	17.6
12/2/82	3-14	7	50	47.9	39.0	14	17.2	8.9	25.0
12/8/82	3-15	1	36.5	48.0	34.1	19	19.6	13.9	15.1
12/29/82	3-16	1	55	40.3	31.8	17	18.0	8.5	14.8
12/29/82	3-16	5	55	44.3	35.9	15	18.0	8.5	20.9
12/30/82	3-17	5	54	53.0	44.7	15	17.3	8.3	31.7
12/30/82	3-17	7	55	40.0	31.0	16	19.1	9.0	15.0
3/8/83	3-19	3	50	49.0	40.2	11	17.0	8.5	29.8
3/8/83	3-19	10	51	40.6	32.2	9	16.5	8.4	23.2
3/15/83	3-21	1	33	54.0	38.7	20	19.5	15.3	18.7
3/15/83	3-21	3	49	48.7	38.3	18	19.7	10.4	20.3
3/16/83	3-23	2	33	58.6	43.8	16	18.9	14.8	27.8
3/16/83	3-23	7	33	59.8	45.3	23.5	18.5	14.5	21.8
3/17/83	3-24	1	33	55.2	40.5	18	18.7	14.7	22.5

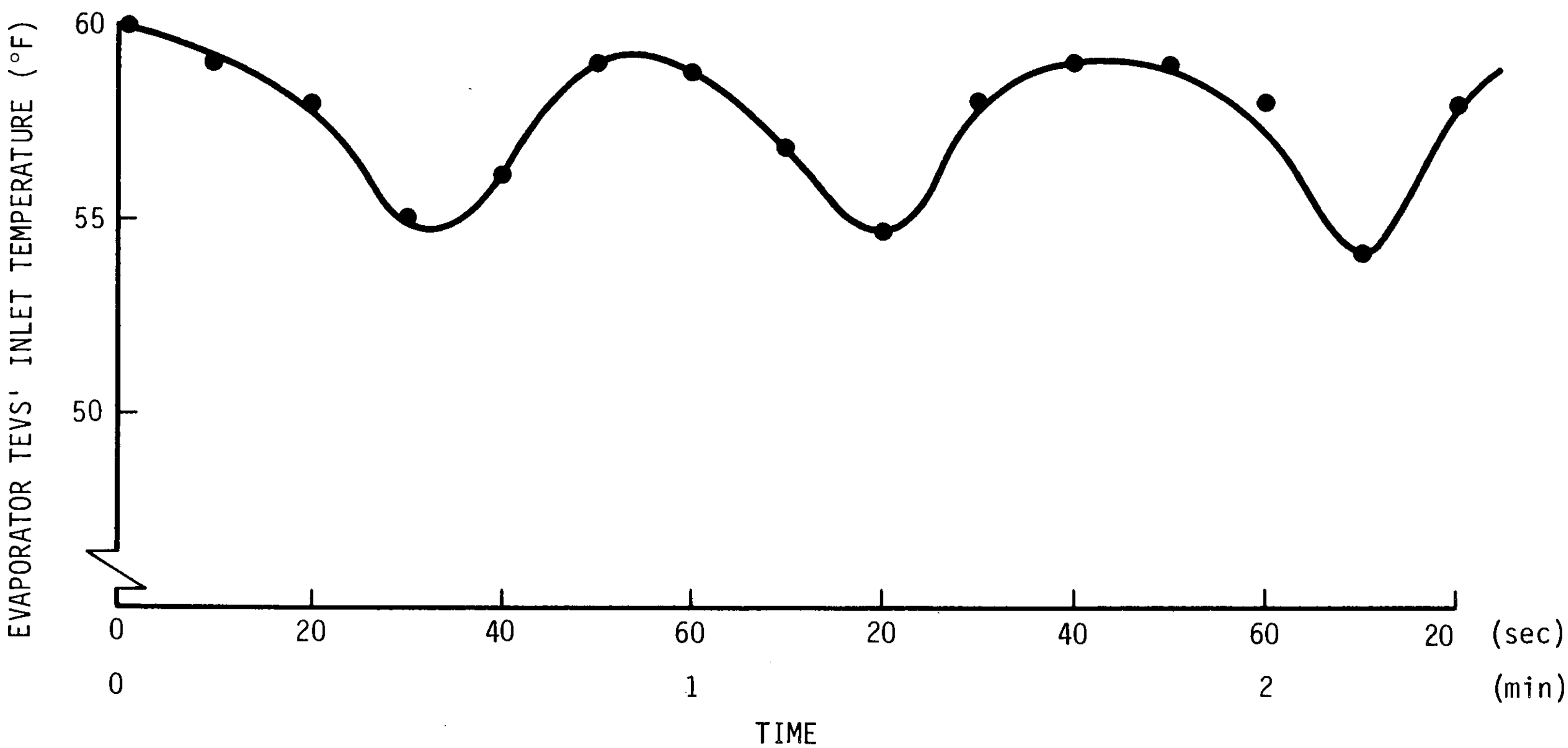


Figure 2-10. Time Fluctuations of Evaporator TEVs' Inlet Temperature due to "Hunting"

coupled to the variations in the flow itself, cause larger oscillations in the evaporator outlet temperature. The outlet temperature fluctuations, however, are not as well behaved as those shown in Figure 2-10. The only property which remains relatively immune to these fluctuations is the evaporator pressure. The evaporator temperature values in Table 2-7 are the saturation temperatures corresponding to the actual evaporator pressures and, hence, are stable in time.

The values for range in Table 2-7 are not affected by the TEVs' hunting. This is because the refrigeration capacity is relatively stable (due to the thermal inertia of the evaporator). The variations in range, in Table 2-7, are due to the changes in brine flow. For nominal tonnage (18 to 20 ton corresponding to 15 to 20°F evaporator temperature), the range varies between 9 and 15°F as the brine flow varies between 55 and 33 gpm.

Apart from the fouling problem discussed earlier, the evaporator performed satisfactorily. The TEVs' hunting was not large enough to affect the evaporator capacity and never caused evaporator flooding. This amount of hunting is natural for TEVs and overall, they performed trouble free and extremely well.

2.3.1.4 Subcooler

In refrigeration systems, subcooling the evaporator suction liquid increases the refrigeration effect and improves the system refrigeration capacity. We selected an Edwards Model B-2 coaxial heat exchanger (1) to subcool the liquid going to the evaporator. The subcooler was cooled by the evaporative condenser's make-up water, piped into its water side in a counter flow arrangement.

As explained in subsection 2.2.3, in order to avoid starving the evaporator, we installed a bypass valve across the subcooler's inlet and outlet ports. This valve bypasses all, or a portion, of the flow and reduces the pressure drop across the subcooler.

Table 2-8 illustrates the effect of the subcooler on system performance. In points 1 and 2 the subcooler is completely bypassed. As shown, inlet and outlet temperatures on both the freon and water sides are equal. A refrigeration tonnage of 19.4 to 19.9 ton is registered. As the bypass valve is partially closed (points 3 and 4), the liquid freon gets subcooled by about 6°F. As the valve is closed further (points 5, 6, and 7), the subcooling increases to about 11°F. Yet, despite the increase in refrigeration effect, the tonnage does not improve. This is simply because the pressure drop across the subcooler causes a reduction in the refrigerant flow rate (see appropriate column) which exactly counteracts the increase in refrigeration effect. Point 8 shows the result of closing the bypass valve completely. The freon outlet temperature drops to 26.5°F which is less than the water inlet temperature of 42°F. This indicates that the liquid is flashing in the subcooler which results in starving the evaporator. The evaporator pressure drops to 23 psig and the refrigeration capacity is reduced to 12.8 tons.

The natural conclusion of the foregoing discussion is that the subcooler is completely redundant. A larger subcooler will increase the tonnage but its inclusion must bear economic justification.

The subcooler's effectiveness could not be calculated from the data in Table 2.8 as the portion of flow that is not bypassed and the outlet temperature of that stream are not known except for point 8. Since the liquid is flashing in point 8, this point cannot be used for effectiveness calculations either. The heat

Table 2-8. The Effect of the Subcooler and Recuperator on System Performance

Subcooler Bypass Valve Position	Point	Time Relative to Point 1 (min)	System Output		Flows and Temperatures										Evaporator Pressure (psig)
					Freon Liquid Side					Freon Vapor Side		Water Side			
					Subcooler			Recuperator		Recuperator		Subcooler			
			Electric (kW)	Refrigeration (ton)	Flow (lb/hr)	In (°F)	Out (°F)	In (°F)	Out (°F)	In (°F)	Out (°F)	Flow (GPM)	In (°F)	Out (°F)	
Completely open	1	0	0.4	19.9	3105	72	72		68	20.5	35.5	2.2	42	42	43
Completely open	2	3	0	19.4	3031	72	72		68.5	21	37	2.2	42	42	42
Partially closed	3	14	0	19.5	2996	72.5	66.5		64	20	34.5	2.2	42	46	42
As in point 3	4	16	0	19.6	2906	73	66.5		54	20	34	2.2	42	47	42
Closed further	5	32	0.3	19.4	2844	72.5	61.5		51	19.5	30.5	2.2	42	50	41.5
As in point 5	6	35	0.3	19.9	3018	72	61.5		59.5	19	30.5	2.2	42	50.5	41
As in point 5	7	57	0.2	19.6	2973	72	61.5		59.5	19.5	33	2.2	42	50.5	40
Completely closed	8	79	2.8	12.8	1561	71	26.5		25.5	70.5	56	2.2	42	53	23

Operating Conditions: Wet Bulb Temperature = 40°F, Condensing Pressure = 142 psig - Date: 3/24/83 transferred in the subcooler during points 3 and 4 was 4400 and 5500 Btu/hr respectively. Heat transferred during points 5, 6 and 7 was around 9300 Btu/hr. The subcooler was left bypassed during the performance testing of the system.

2.3.1.5 Recuperator

A nominal 35 ton, Model LHXR-35 Packless recuperator, was selected and installed in our refrigeration system (1). The recuperator's duty is twofold: to further subcool the condensate at the TEVs' inlet; and to superheat the vapor at the compressor suction, thus protecting it from flooding.

According to Table 2-8, the recuperator performs 10 to 16°F of superheating, depending on the condition of the subcooler bypass valve. When the latter is open and the subcooler is isolated, the recuperator's liquid-side inlet temperature is higher and it does more superheating. When the subcooler is working, the recuperator superheats the suction vapor less. The amount of subcooling done by the recuperator is not well represented by the data in Table 2-7. According to this data, when the subcooler is bypassed, the recuperator subcools the liquid by 3.5 to 4°F and superheats the vapor by 15-16°F. Since the mass flow rate on both the liquid and vapor side are equal, a simple heat balance shows that the ratio of the degrees of subcooling to degrees of superheat should equal the ratio of the specific heats of the vapor to the liquid:

$$\left[\overset{\circ}{m}\ Cp\ \Delta T\right]_{vapor} = \left[\overset{\circ}{m}\ Cp\ \Delta T\right]_{liquid}$$

where

$\overset{\circ}{m}$ = mass flow rate,

Cp = specific heat,

ΔT = temperature rise or fall

Since

$$\overset{\circ}{m}_{vapor} = \overset{\circ}{m}_{liquid}$$

$$\therefore \quad \frac{\Delta T \text{ liquid}}{\Delta T \text{ vapor}} = \frac{Cp \text{ vapor}}{Cp \text{ liquid}}$$

Based on conditions in points 1 to 7 of Table 2-7, the specific heats are:

$$Cp_{vapor} = 0.17 \text{ Btu/lbm °F}$$

$$Cp \text{ liquid} = 0.295 \text{ Btu/lbm °F}$$

now

$$\frac{\Delta T \text{ liquid}}{\Delta T \text{ vapor}} \simeq \frac{4}{16} = 0.25 < 0.58 = \frac{0.17}{0.295} = \frac{Cp \text{ vapor}}{Cp \text{ liquid}}$$

This discrepancy is due to the fluctuations in the recuperator liquid-side outlet temperature and refrigerant flow, as explained in subsection 2.3.1.3. In this test, these fluctuations are evident in the degrees of subcooling performed by the recuperator which varies considerably between points that are otherwise similar: the subcooling due to the recuperator for points 3 and 4 is 2.5 and 12.5°F respectively. The actual value can be estimated from the degrees of superheat multiplied by the ratio of specific heats or $14 \times 0.58 \simeq 8$°F. For points 1 and 2 the effective subcooling is $15.5 \times 0.58 \simeq 9$°F and for points 5, 6 and 7 it is $12 \times 0.58 \simeq 7$°F.

Unlike the subcooler, the recuperator could not be isolated from the refrigeration circuit. Hence, its overall effect on system performance was not experimentally measured. When the subcooler is bypassed, the recuperator effectiveness (based on the vapor side) is approximately 30% with an overall UA (heat transfer coefficient times area) of 190 Btu/°F hr. The manufacturer's data indicate a UA of 300 Btu/°F hr and an effectiveness of 40 percent under similar conditions. Given the accuracy of the data, and the possibility of having fouled the recuperator or contaminated the freon, this comparison does not seem unrealistic.

2.3.1.6 Summary of Heat Exchanger Specifications and Performance

The heat exchangers discussed in subsection 2.3.1 and described in Section 5 of Reference 1 are summarized in Appendix E.

2.3.2 Machinery

2.3.2.1 Expander

As discussed in subsection 2.1.5, the expander performance exceeded our expectations. In subsection 2.1.1, we explained the rationale behind choosing the 6° ATDC valve timing. Figure 2-2 in the same subsection shows the effect of valve timing on the expander P-V diagrams, and Figure 2-3 portrays the variations in expander brake horsepower with the pressure difference across the rotary valve for different valve timings. Figure 2-4 in subsection 2.1.2 illustrates the sensitivity of the expander brake power to the inlet hydronic temperature, and Figure 2-9 in subsection 2.1.5 depicts the relationship between the expander brake power and condensing pressure.

In this section, we will examine the performance of the expander as a prime mover and discuss its mechanical, volumetric and isentropic efficiencies; these are defined as follows:

$$\text{Mechanical efficiency} = \frac{\text{brake power output}}{\text{indicated power}}$$

$$\text{Volumetric efficiency} = \frac{\text{actual volume of vapor displaced}}{\text{theoretical volume displaced}}$$

$$\text{(brake) isentropic efficiency} = \frac{\text{brake power output}}{\text{power based on isentropic enthalpy difference}}$$

where

brake power = power output measured at shaft

indicated power = power output based on P-V diagram

theoretical volume of vapor displaced = 2 x swept volume x RPM x cutoff

cutoff = the percentage of stroke taking place while the inlet valve is open.

The mechanical efficiency is dependent upon the power output of the system. As shown in Figure 2-11, mechanical efficiency improves as the brake power increases. This can be explained by introducing the friction power as the difference between indicated power and brake power; thus:

$$\text{mechanical efficiency} = \frac{\text{brake power}}{\text{brake power + friction power}}$$

The friction power is a weak function of operating pressures and is generally small compared to the brake power. By the nature of the above equation, as the brake power increases in proportion to friction power, the mechanical efficiency increases, explaining the trend in Figure 2-11.

The friction power can be calculated from the above equation, yielding the second graph in Figure 2-11.

The friction power is observed to increase at low brake power outputs. This is peculiar to the nature of this particular run (run 2-23, Appendix C) where the expander power output is reduced by raising the condensing pressure. Since high condensing pressures translate into high average cylinder pressures, the friction power is amplified. If the brake power were reduced by decreasing the expander inlet pressure, the opposite effect would have been observed. The friction power would have decreased with expander brake power.

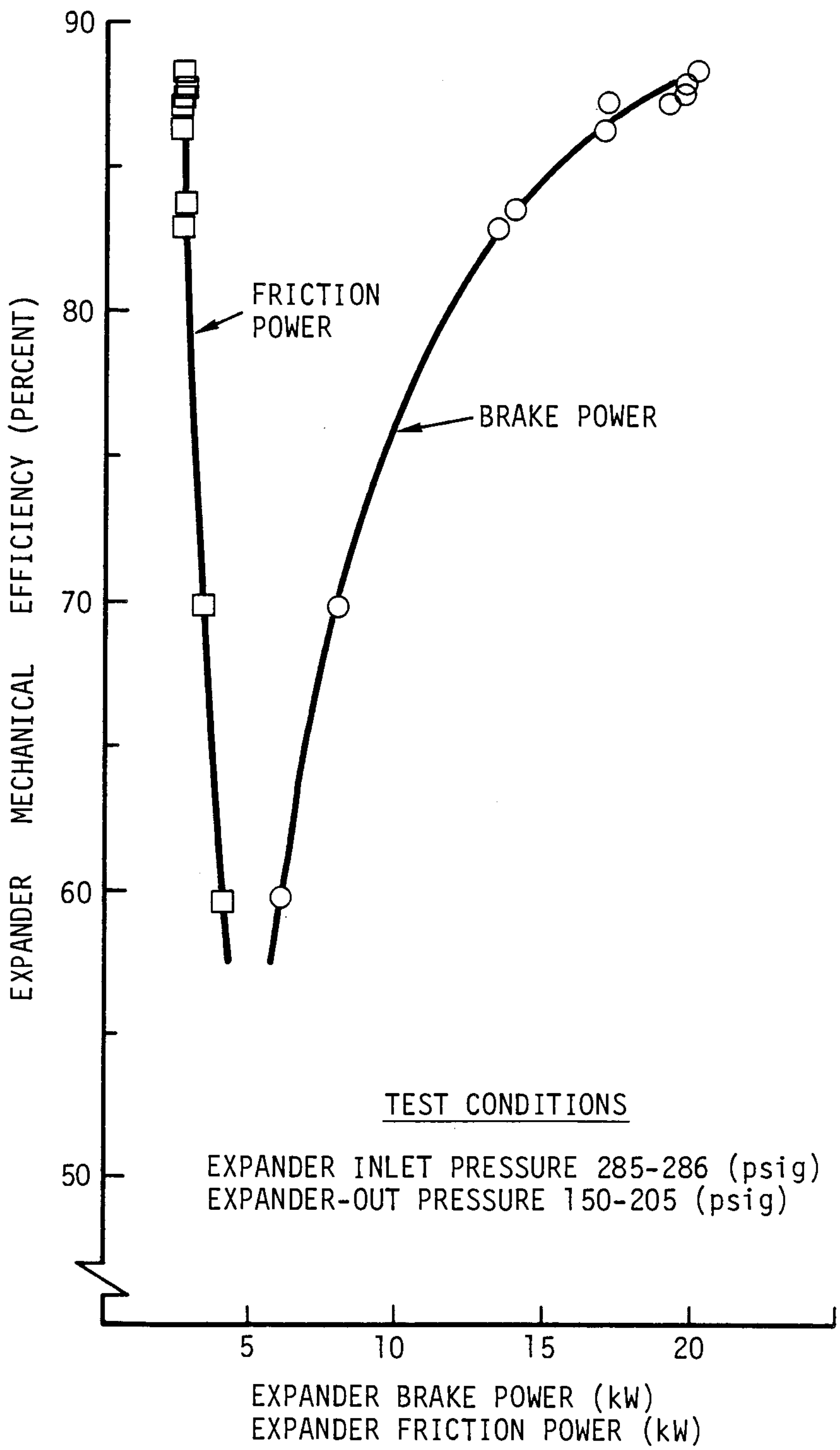


Figure 2-11. Variations in Mechanical Efficiency with Expander Brake Power and Friction Power

The expander volumetric efficiency could not be measured very accurately. This was due to the inaccuracy in the direct measurements of freon flow and the uncertainty in the flow calculated from the hydronic load. The hydronic heat load can be used to calculate the freon flow since the enthalpy difference across the boiler is known. The hydronic load divided by this enthalpy difference yields the amount of freon evaporated in the boiler. If the bypass regulators do not bypass any vapor, and if the level in the boiler does not change in time, the rate of evaporation equals the freon flow in the expander. The computer program used to reduce the data used both the flow obtained by this method and the flow measured by the turbine flowmeter to calculate volumetric effiency. These runs indicate an average volumetric efficiency of 100 percent, with results ranging between 80 and 120 percent. The results being so scattered, it is difficult to observe a trend in the change in volumetric efficiency with valve timing. In fact, since the volumetric efficiency is based on the cutoff portion of the stroke, valve timing does not necessarily affect the volumetric efficiency.

A volumetric efficiency of 100 percent is not suprising for a rotary valve. The pressure losses associated with our rotary valve are considerably lower than losses in poppet or reed valves. Other factors, as well, contribute to the high volumetric efficiency. Some of the vapor condenses on the cool walls of the cylinder, increasing the effective throughput and the apparent volumetric efficiency. Any leakage shows as an increase in volumetric efficiency when dealing with expanders.

Despite the scatter in the results, they strongly support the conclusion that the rotary valve has very low flow losses associated with its operation. This confirms the validity of selecting the rotary valve concept for this application from a technical standpoint. The volumetric efficiency data does not help to quantify the leakage across the valve. This leakage, however, is more a burden on the freon boiler than a threat to the expander efficiency.

The cutoff value at 6° ATDC valve timing is 68 percent. The overall volumetric efficiency based on total swept volume is, therefore, about 68 percent. This number would have been instructive if it could be accurately measured for various valve timings, but the scatter in results do not allow this. The overall volumetric efficiency should increase as the valve timing is retarded, indicating higher actual throughputs which support the rise in expander power.

Table 2-9 shows the combined effect of valve timing and condensing pressure on the brake isentropic efficiency. Both power output and isentropic efficiency increase as the condensing pressure drops. The scatter in the isentropic efficiency results, once again, is due to the spread in the values of the freon flow rate used in the calculations.

The effect of valve timing cannot be seen in Table 2-9 because it is completely overshadowed by the influence of the condensing pressure. In theory, the expander becomes less efficient as the timing of the inlet valve opening is delayed. This is because the increase in flow rate associated with retarding the valve (see subsection 2.1.1) does not do as much specific work (work per unit of heat input) as the rest of the flow.

The brake isentropic efficiency averages at about 68 percent. Dividing this number by the mechanical efficiency of 88 percent yields an indicated isentropic efficiency of 77 percent, which characterizes an efficient expander.

The expander, except for the rotary valve and its drive mechanism, ran trouble free during the entire testing period. We had no problems with the wrist pin bearings (see Section 5.3.3 in Phase II Report, (1)), the lubrication system, the shaft seal or any other subsystem of the expander. The rotary valve, however, required some modifications, which are described in subsection 3.3.

Table 2-9. Expander Brake Isentropic Efficiencies at Different Valve Timings and Condensing Pressures

Test*	Point	Valve Timing (^{o}ATDC)	Expander Brake Power (kW)	Expander Isentropic Efficiency (%)	Condensing Pressure (psig)
3-08	1	0	15.3	66	173
3-12	1	0	17.8	62	160
3-12	10	0	18.5	64	160
3-13	1	3	20.6	65	150
3-13	2	3	20.6	64	150
3-14	5	3	19.1	69	154
3-14	3	3	19.4	74	156
3-17	2	6	23.0	68	133
3-17	7	6	22.8	67.5	133
3-17	1	6	23.1	68	134
3-15	1	9	22.1	74	136

* See Appendix C

2.3.2.2 Compressor

Two of the four cylinders on the York RS44A compressor act as compressors. Both cylinders are on the same bank and both were originally equipped with an unloader mechanism. We modified one cylinder to stay permanently loaded. Thus, the unloader mechanism affects only one cylinder on the compressor bank. As described in subsection 2.2.2, this mechanism worked satisfactorily.

We experienced no mechanical problems with the compressor. During operation the reed valves made more noise than the rotary valve and got quite hot, but this is as expected. However, the compressor capacity did not match our expectations. Table 2-10 shows the compressor performance during the testing period of the system. The actual tonnage is 1.2 to 3.0 tons less than the rated values extracted from York data. What is not accounted for in the actual data are the losses in the piping and the amount of heat that the motor/generator adds to the refrigeration flow. The latter is about 0.6 kW or 0.17 tons. The former is unknown, but it includes losses due to: the copper line going to the motor/generator; other suction lines (even though they are mostly insulated); and heat flow from the enthalpy sampling lines (see Appendix B). Other possible losses include: crankcase heater (0.6 kW, was on most of the time); liquid carryover from the oil separator (some freon condenses in the oil separator and is carried to the crankcase with the oil); and pressure drop in the liquid lines (reduces the subcooling at the evaporator inlet). The York data are based on 5° subcooling at the evaporator inlet. The calculated values for subcooling are shown in Table 2-10. Since these values are based on the condensing pressure, the actual subcooling may be somewhat less due to the pressure loss between the condenser and evaporator inlet. The subcooler was bypassed during these tests. The recuperator was discounted from the calculations, since, what it adds to the refrigeration capacity on the subcooling side, it deducts from the superheating

Table 2-10. Compressor Performance

Date	Run	Point	Saturated Temperatures		Refrigeration Capacity		Subcooling (°F)
			Evaporator (°F)	Condenser (°F)	Actual (ton)	Rated (ton)	
10/6/82	3-08	1	14	92	15	18.15	6
11/30/82	3-13	1	17	81	18.2	20.75	6
11/30/82	3-13	2	17	81	18.2	20.75	6
12/2/82	3-14	5	16	84.5	18.7	19.86	8
12/2/82	3-14	6	17	84.5	18.5	20.51	8.5
12/2/83	3-14	7	14	84	17.2	19.02	8
12/8/82	3-15	1	19	76.5	19.6	22.15	7.5
12/29/82	3-15	1	17	81	18.0	20.75	10
12/29/82	3-15	5	15	82	18.0	19.68	10
12/30/82	3-17	5	15	75.5	17.3	20.4	11.5
12/30/82	3-17	7	16	75.5	19.1	20.86	11.5
3/8/83	3-19	3	11	77.5	17.0	18.5	10.5
3/8/83	3-19	10	9	77.5	16.5	17.65	11.5
3/15/83	3-21	1	20	83.5	19.5	21.78	6.5
3/15/83	3-21	3	18	83.5	19.7	20.87	7.5
3/16/83	3-23	2	16	82.5	18.9	20.08	7.5
3/16/83	3-23	7	23.5	103	18.5	21.16	6
3/17/83	3-24	1	18	79.5	18.7	21.34	7.5

side, having no net effect on the load. The York data are based on 15°F of superheat but the literature maintains that, for Freon-22, the amount of superheat does not affect the refrigeration capacity if kept below 25°F.

The discrepancy between the rated and actual data may be the result of the inaccuracies in the instrumentation. Ultimately, the York compressor may not be as efficient as advertised. But, as discussed above, we were not able to pinpoint a single cause.

2.3.3.3 Motor/Generator

The motor/generator was custom designed and manufactured for us by Custom Motor Design in Cincinnati, OH. We experienced a couple of manufacturer-related problems with the motor/generator and its housing: some welds leaked under pressure after the system had been assembled, and one internal lead, corresponding to one phase, was shorted against the shaft after rubbing against it for several hours. However, the problems were corrected and the unit then performed as expected.

The power drawn by the motor in the absence of any load intrigued us for a while, but the problem was due to a faulty wattmeter. After the wattmeter was fixed, we measured a no-load drain of 0.6 kW, which is what the manufacturer said it should be.

We had no problem with cooling the motor/generator and the precaution to route some of the compressor suction to its casing proved to be effective. This flow, however, washed the oil off the motor bearings, necessitating some external lubrication, as discussed in subsection 3.5.

The motor/generator performed as expected. It was as efficient as a motor as it was as a generator. It showed, once again, that induction motors are well suited to the task of alternately absorbing shaft power surpluses and making up shaft power deficits at essentially constant speeds.

2.3.3.4 Gerotor Pump

Our experience with the Phase III system supported our Phase II conclusion that pumping liquid freon was a difficult task. However, the gerotor pump proved adequate for this duty. One of the most important characteristics of the gerotor pump is its ruggedness and simplicity of design. This is indispensable in a pump that has to withstand cavitation-induced vibrations. The pump endured this abuse for several hours during the early testing of the system. Upon examination, we found a very slight increase in the play of the inner gerotor about the keyed shaft. This play was taken up by shimming the key and has not reoccurred during the subsequent cavitation-free operation, which totalled 40 hrs.

Most of the pump-related problems that we encountered during our testing are described in subsection 2.1.

Once the cavitation problem was resolved, the pump performed its function of filling the boiler satisfactorily. The pump-by-pass boiler-level control mechanism worked without trouble (as explained in subsection 2.3.1.2). The geroter pump is adequately sized for its flow duty and operates very efficiently (89 percent volumetric and 80 percent hydraulic efficiency).

2.3.3.5 Inducer Pump

The main function of the inducer pump is to provide enough net positive suction head (NPSH) for the gerotor pump. Once we filled the system with adequate freon, the inducer pump performed this function reliably.

The impeller on this pump was shaved at a machine shop to a diameter of 8.7 in. (from the original 10 in., see subsection 5.3.1 in Phase II Progress Report (1)). This was done in order to minimize the pressure head across this pump and, hence, its power requirement. The inducer pump develops 30-33 ft of head with the 8.7 in. impeller. This provides more NPSH than the 20 ft that the gerotor pump requires by a reasonable margin.

2.4 Economic Analysis

The economic attractiveness of installing this waste heat driven chiller system is best judged by means of differential costs, as compared with installing another chiller system of comparable capacity. We have selected the Edwards Engineering Corporation CC series of packaged water chillers for comparison, largely because data was readily available. Within that series, we selected model CC-40-C1-AHP, for it has a capacity of 220,000 Btu/hr at 105°F ambient dry bulb temperature, which is roughly equivalent to our 19 tons at 80°F ambient wet bulb temperature.

We have chosen to make the economic comparison on the basis of internal rate of return (IRR), (also called discounted cash flow, also called return on investment), for it avoids the problem of having to select the appropriate discount rate. The calculated value of IRR can be compared with any applicable discount rate to determine the economic viability of installing a waste heat driven chiller system.

Note that the base case which this differential analysis uses is buying the Edwards chiller, not replacing an existing chiller. Replacing an existing chiller would involve a much higher differential initial cost, thus a much lower IRR.

There are five input values which largely determine the result of this analysis, each of which merits discussion:

- Initial cost of the waste heat driven chiller
- Initial cost of the Edwards chiller
- Electrical consumption of the waste heat driven chiller

- Electrical consumption of the Edwards chiller
- Electric rates.

2.4.1 Initial Cost of the Waste Heat Driven Chiller

Subsection 5.5.1 of the Phase II Progress Report (1) calculates the first cost of the waste heat driven chiller to be:

System cost = $54,000 + 1.62 x cost of heat exchangers.

Table 2-11 presents the actual costs of the five heat exchangers used in the system that was built and tested, and calculates the 1982 price and (using 6% inflation) the 1983 price, at $91,951.

2.4.2 Initial Cost of the Edwards Chiller System

The price of the Edwards CC-40-C1-AHP packaged water chiller (air-cooled) was quoted as $36,250.

It was assumed that shipping costs, installation costs, and maintenance costs would be essentially the same for both chiller systems; the only difference would be that the evaporative condenser requires a makeup line. Hence, these costs do not enter into a differential cost analysis.

2.4.3 Electrical Consumption of the Waste Heat Driven Chiller System

Table 2-12 presents the calculation of the electrical consumption as a function of ambient wet bulb temperature by the system essentially as tested.

The relationship between condensing temperature and ambient wet bulb temperature is taken from the manufacturer's data, and not from our test data. As explained in subsection 2.3.1.1, our

Table 2-11. Initial Cost of Waste Heat Driven Chiller System

Chiller, less heat exchangers, 1982 (per ref. 1)		$54,003
Heat Exchangers		
Evaporative condenser	$11,500	
Freon boiler	5,929	
Evaporator	2,568	
Recuperator	123	
Subcooler	92	
	$20,212	
× 1.62 =		32,743
1982 price		86,746
plus 6% allowance for inflation		5,205
1983 price		$91,951

Table 2-12. Electrical Consumption*** of Waste Heat Driven Chiller System

Ambient Wet Bulb Temperature (°F)	Condensing Temperature (°F)	Condensing Pressure (psig)	Power Consumed* (less parasitics)** (kW)	Power Consumed (including parasitics)
80	97.5	189	12.0	20.5
73	92	174	7.0	15.5
66	86	158	2.7	11.2
59	80.5	144	-0.8	7.7
52	74.5	131	-3.5	5.0
45	69	119	-5.7	2.8

* Negative power consumptions correspond to power generation by the motor/generator.

** Parasitics include condenser spray pump, condenser fans, motor/generator residual, and controls.

*** With 18-20 tons of refrigeration at 15-20 °F evaporator temperature, 30 °F chiller water leaving temperature.

condenser did not perform up to specifications, probably because of contaminated freon. This relationship is presented in Figure 2-12.

Condensing pressure is related to condensing temperature as per the saturation properties of R-22.

The power consumed by this chiller system (less parasitics) as a function of condensing pressure is presented in Figure 2-13. In this figure is summarized the essential results of the entire development program. To be added to these values are the power consumptions of the condenser's pump and fans (7.4 Kw), and the losses of the motor/generator and controls (1.1 Kw), all of which are independent of ambient temperature.

2.4.4 Electrical Consumption of Edwards Chiller System

The Edwards chiller uses an air-cooled condenser, and rejects its heat to the ambient dry bulb temperature. It is necessary to assume an average, typical relationship between ambient dry bulb and ambient wet bulb; this relationship (shown by the first two columns of Table 2-13) is necessarily somewhat subjective, but has a good basis in fact. Condensers are typically rated at 105°F ambient dry bulb, 78°F ambient wet bulb; letting 105°F dry bulb correspond to 80°F wet bulb, as we did, slightly penalizes the waste heat driven chiller. The spread between dry bulb and wet bulb should diminish as both go lower; our letting it drop from 105/80 to 55/45 perhaps overstates this effects, but conservatively so.

Figure 2-14 shows the capacity of the chiller system as a function of ambient dry bulb, assuming a 30°F chilled water leaving temperature. No so obvious is the relationship of compressor

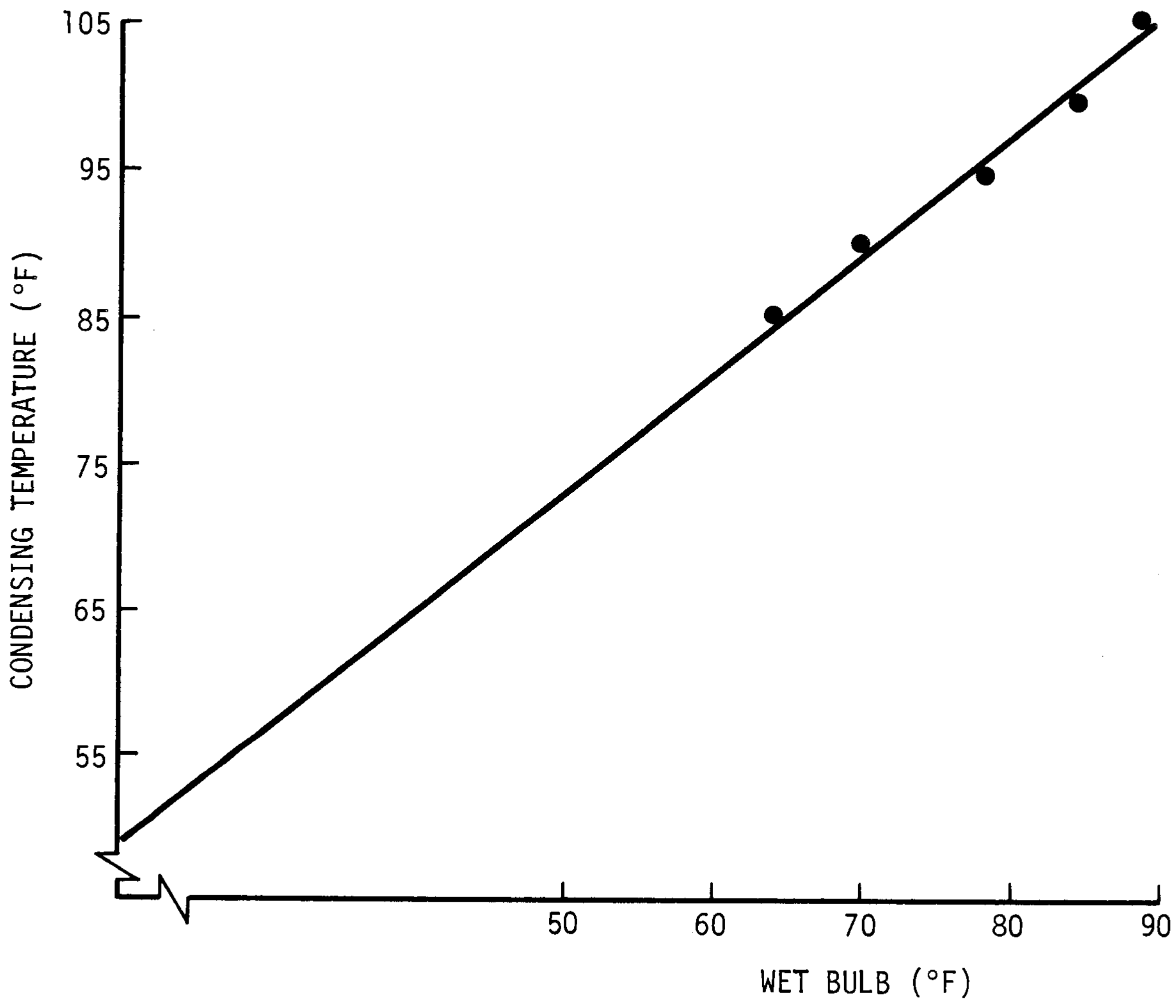


Figure 2-12. Condensing Temperature Versus Ambient Wet Bulb Temperature

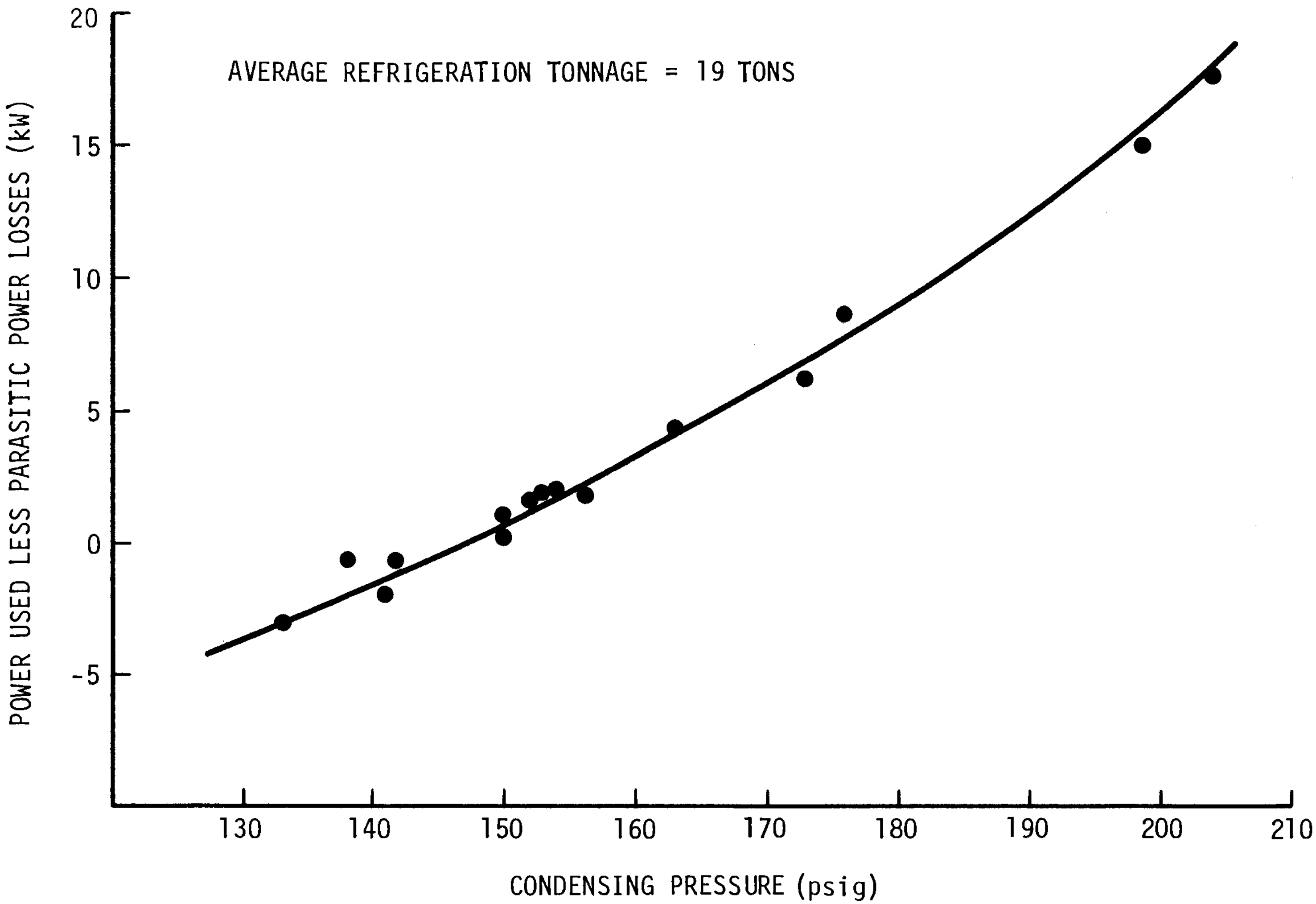


Figure 2-13. Actual Power Consumption (less Parasitic) of the Brine Chiller System Versus Condensing Pressure

Table 2-13. Performance of the Edwards Chiller

Ambient Wet Bulb Temperature (F)	Ambient Dry Bulb Temperature (F)	Capacity (10^3 Btuh)	Compressor Power (kW)	Average** Compressor Power (kW)	System* Power Consumption (kW)
80	105	220	38.5	38.5	45.0
73	95	240	37.3	34.2	40.7
66	85	260	35.1	29.7	36.2
59	75	280	32.9	25.9	32.4
52	65	300	31.5	23.1	29.6
45	55	320	30.1	20.7	27.2

* System includes 3 fans, each at 460V, 3 phase, 4.8 amps, 90% power factor.

** Assumes system is cycled on and off to maintain an average capacity of 220,000 Btuh.

AIR-COOLED MODEL CC-40-C1-A
CAPACITY IN 1000'S OF B.T.U./HR.

Temp. of Water/Glycol Leaving Chiller	Ambient Temperatures °F				(Note 1)
	75°	85°	95°	105°	115°
60°F	505	470	440	420	395
55	477	445	415	390	360
50	450	420	385	355	335
45	405	375	345	320	290
40	360	335	305	280	260
35	320	295	270	250	230
30	280	260	240	220	200
25	245	230	210	195	175
20	215	205	185	170	160

WATER-COOLED MODEL CC-40-C1-W
CAPACITY IN 1000'S OF B.T.U./HR.

Temp. of Water/Glycol Leaving Chiller	Entering Cond. Water Temp. °F				
	75°	85°	95°	105°	115°
60°F	560	520	490	255	430
55	515	475	445	415	390
50	465	430	400	370	345
45	420	390	360	330	305
40	375	350	320	295	270
35	330	310	285	260	240
30	290	270	250	230	210
25	255	235	220	200	185
20	225	210	195	180	165

The above are net ratings with allowance made for the heat input of the recirculation (by-pass) pump, but not for the system pump.
Note 1: For ambient temperatures above 105°F, consult factory for special design and pricing.

ELECTRICAL DATA:

Volts	Phase	Cycles	Compressor(s)		Fan(s)		By-Pass Pump		System Pump		Total Unit FLA	Suggested Field Wiring Disconnect Size *Circuit Breaker	Fuse Size (Time Lag)
CC-40-C1-A Air-Cooled			H.P.	FLA Each	H.P.	FLA Each	H.P.	FLA Each	H.P.	FLA Each			
200	3	60	35	163.0	(3) 3	11.0	1	4.1	7½	25.3	225.4	400 AMP	300 AMP
230	3	60	35	163.0	(3) 3	9.6	1	3.6	7½	22.0	217.4	400 AMP	300 AMP
460	3	60	35	81.5	(3) 3	4.8	1	1.8	7½	11.0	108.7	200 AMP	150 AMP
CC-40-C1-W Water-Cooled													
200	3	60	35	163.0	–	–	1	4.1	7½	25.3	192.4	400 AMP	250 AMP
230	3	60	35	163.0	–	–	1	3.6	7½	22.0	188.6	400 AMP	250 AMP
460	3	60	35	81.5	–	–	1	1.8	7½	11.0	94.3	200 AMP	125 AMP

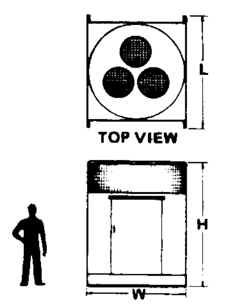
H.P. RATINGS:

Models	Compressor(s)		Condenser Fan		By-Pass Pump	System Pump		
	No.	H.P.	No.	H.P.	H.P.	H.P.	Ft.Hd.†	GPM
CC-40-C1-A	1	35	3	3	1	7.5	100	160
CC-40-C1-W	1	35	–	–	1	7.5	100	160

NOTE: † For p.s.i., multiply ft. by 0.434.

DATA AND DESCRIPTION

Model CC-40-C1	Type of Enclosure	Walk-in Enclosures Air	Water
Physical Dimension	Width (in.) W	84½	84½
	Length (in.) L	96	96
	Height (in.) H	108	78
Condensing Air Required (CFM)		32,000	–
Condensing Water Pipe Size (in.)		–	2½
Piping Size	Supply Pipe Size (in.)	3	3
	Return Pipe Size	3	3
Chilled Water Storage Tank Capacity (gal.)		167	167
Operating Weight (lbs.)		6850	6850
Crating	Crate: Width (in.)	No Crating	
	Crate: Length (in.)		
	Crate: Height (in.)		
	Shipping Weight, lbs.	5400	5400



END VIEW
WALK-IN ENCLOSURE

Figure 2-14. Specification Sheet for Edwards Chiller power to ambient dry bulb, which is taken from Figure 2-15; one has to recognize that the compressor capacity of Figure 2-14 for:

- Ambient dry bulb temperature + 25 = condensing temperature

- Saturated suction temperature + 12.5 = chilled water leaving temperature.

and read off the power inputs accordingly.

Finally, we add the (constant) power consumption of the three condenser fans.

Comparison of the rightmost column of Table 2-13 with the rightmost column of Table 2-12 yields the somewhat astonishing result that the waste heat driven chiller system uses 25 kW less power than does the Edwards chiller system, <u>independent of ambient temperature</u>. This fortuitous result simplifies the analysis markedly, for the distribution of ambient temperatures applicable to any given site does <u>not</u> enter into the economic analysis; all that matters is the local electric rate.

2.4.5 <u>Electric Rates</u>

Even for a given class of service (residential vs. commerical vs. industrial), there are as many electric rate schedules as there are electric utilities. Also, they change frequently, and are complicated.

Figure 2-16 shows Boston Edison's General Service Rate G-2, applicable to commercial (such as supermarkets) and light industrial customers (such as Foster-Miller). The annual bill for adding a constant 25 kW for 8000 hrs is calculated using the marginal rates, and by summing the demand charge, the energy charge, and the fuel adjustment charge as follows:

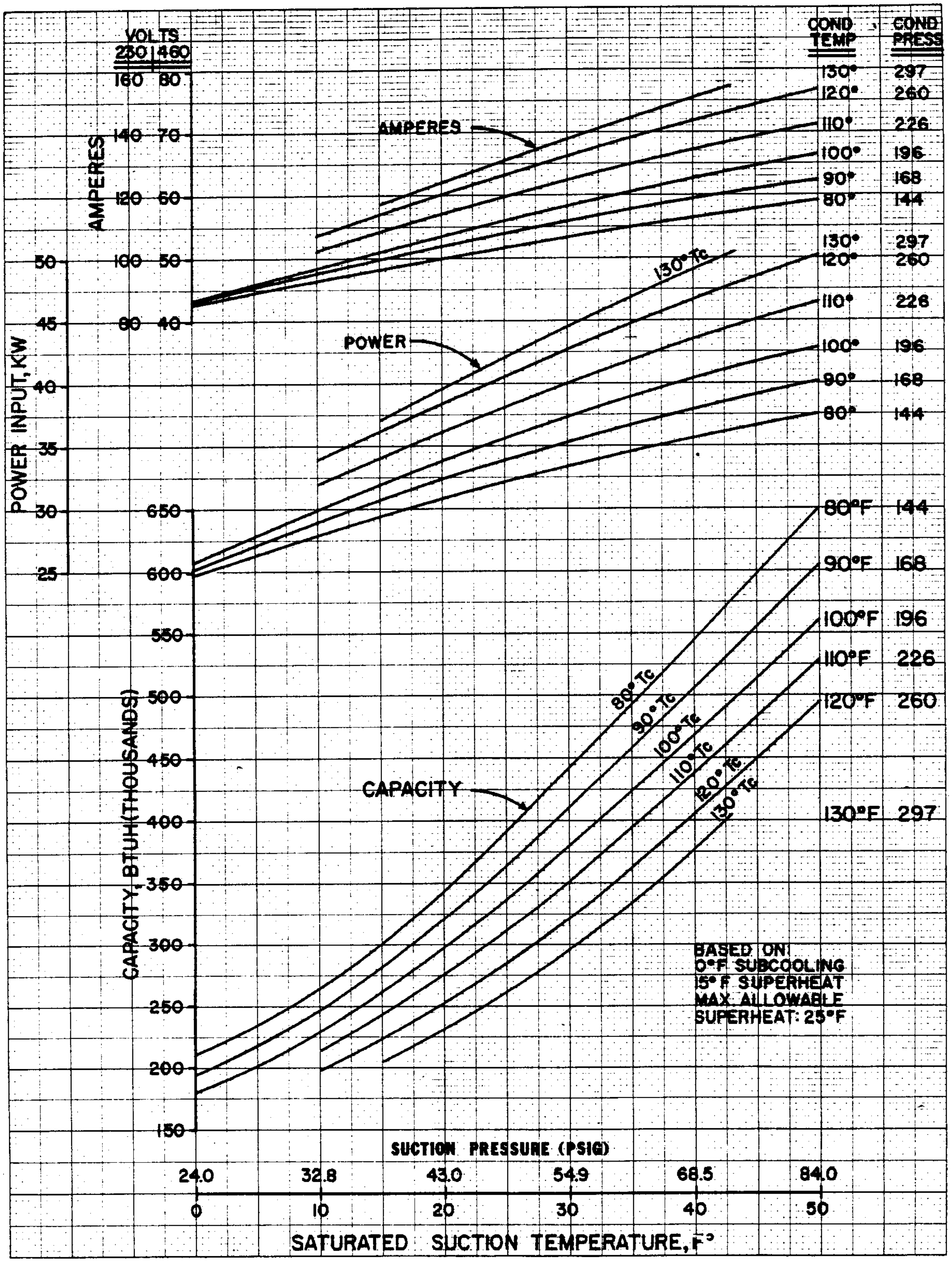


Figure 2-15. Specification Sheet For Compressor Used in Edwards Chiller

Boston Edison Company
800 Boylston Street, Boston

Sheet No. 1 Cancels

M.D.P.U. No. 579A
M.D.P.U. No. 531

GENERAL SERVICE RATE G-2

Available for commercial and industrial uses at a single location where the service voltage is less than 5000 volts and the monthly demand is equal to or greater than 20 kilowatts. Not available for resale or domestic service in residential premises.

Rate:

Demand Charge

During the billing months of:

July-October	November-June	
$171.00	$171.00	per month for the first 20 kilowatts of demand or any portion thereof.
6.74	5.64	per kilowatt per month for the next 130 kilowatts of demand.
5.39	4.29	per kilowatt per month for the excess.

Energy Charge

2.84 cents per kilowatthour for the first 290 hours' use of the demand per month.

1.74 cents per kilowatthour for the next 150 hours' use of the demand per month.

1.19 cents per kilowatthour for the excess.

During the billing months of July through October, all use will be billed an additional charge of

.68 cents per kilowatthour.

Additional Energy Charge as provided in "Direct Current Rate", applicable to all DC kilowatthours on this rate.

Fuel and Purchased Power Adjustment as provided in "Fuel and Purchased Power Adjustment," applicable to all kilowatthours on this rate.

Conservation Service Charge as provided in the "Conservation Service Charge," applicable to all bills rendered under this rate.

Determination of Demand: The maximum fifteen-minute demand (either kilowatts or 80 percent of the kilovolt-amperes) will be determined by meter during the monthly billing period except, any demand recorded between 12 p.m.* and 9 a.m.* and all day Saturday and Sunday will be reduced by

*All times are Eastern Daylight Savings Time and pertain to the summer off-peak period. Corresponding Eastern Standard Times are 11 p.m. and 8 a.m.

(Continued)

Date Filed,
May 14, 1982
Pursuant to Amended Order in
D.P.U. No. 906 dated May 13, 1982

Issued by
Thomas J. Galligan, Jr., Chairman and Chief Executive Officer

Date Effective,
May 14, 1982

Figure 2-16. Boston Edison Commercial Rate Schedule

Demand Charge

$25 \times (4 \times \$5.39 + 8 \times \$4.29) =$ $1,397

Energy Charge

$25 \times [(\frac{4}{12} \times \$0.0187) + (\frac{8}{12} \times \$0.0119)] \times 8000 =$ 2,833

Fuel Charge

$25 \times \$0.035906 \times 8000 =$ 7,181

Total Annual Charge $11,411

Therefore, the annual average cost per kWh is 5.7¢.

2.4.6 Calculation of Internal Rate of Return

In calculating the IRR, we have assumed

- That the cost of electricity will increase by 5% annually
- That the system has a ten-year life
- That the owner is a corporation with a 46% marginal federal income tax rate and no state or local income taxes.

We further take a 10% investment tax credit on the differential cost, a 10% energy tax credit (applicable to cogeneration systems) on the entire cost of the waste heat driven chiller system, and Accelerated Cost Recovery System (ACRS) depreciation as 5-year property.

The net present value (NPV) of this differential investment depends, of course, on the rate at which the future cash flows are discounted. The dependence is shown in Figure 2-17. The

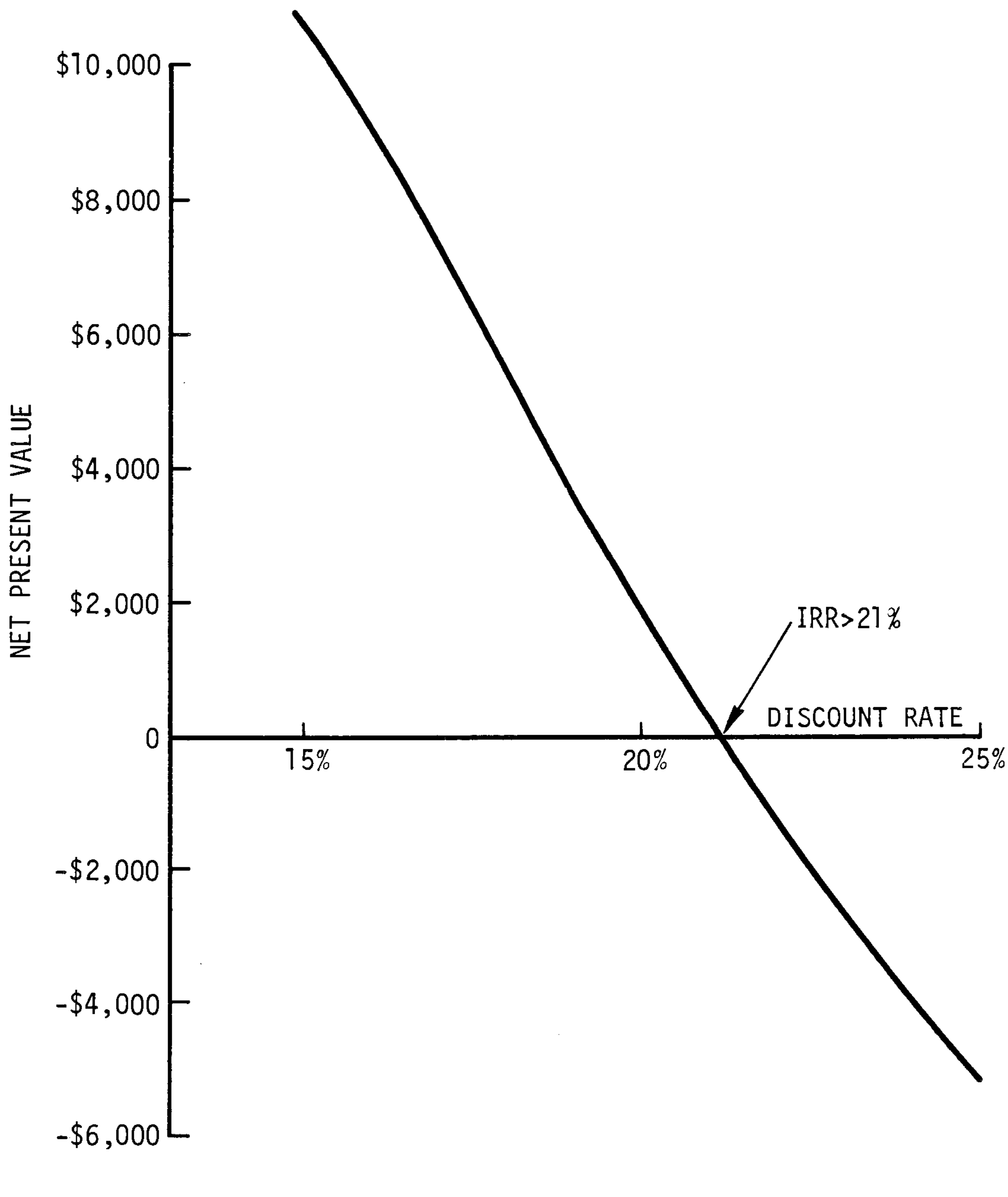


Figure 2-17. Net Present Value versus Discount Rate

value at which the NPV equals zero is the IRR, just over 21%. The NPV calculation for a 21% discount rate is shown in Figure 2-18, from which the details of the calculation are readily apparent.

2.4.7 Economic Summary

This economic analysis, like any other, is no better than the assumptions and estimates on which it is based. There is room for discussion as to the cost to manufacture and maintain the waste heat driven chiller. There is also room for conjecture as to the future cost of electricity. Still, it is accurate enough to justify consideration of this system as a candidate in areas of high electrical energy costs. For example, Table 2-14 shows many areas where commercial and industrial electricity costs in 1982 already exceeded those used in this analysis.

YEAR	0	1	2	3	4	5	6	7	8	9	10
PRICE OF WASTE HEAT CHILLER	91951										
PRICE OF EDWARDS CHILLER	36250										
DIFFERENTIAL INVESTMENT	-55701										
INVESTMENT TAX CREDIT		5570									
ENERGY TAX CREDIT		9195									
ENERGY CHARGE SAVINGS		1397	1467	1540	1617	1698	1783	1872	1966	2064	2167
FUEL ADJUSTMENT SAVINGS		2833	2975	3123	3280	3444	3616	3796	3986	4186	4395
DEMAND CHARGE SAVINGS		7181	7540	7917	8313	8729	9165	9623	10104	10610	11140
TOTAL ANNUAL SAVINGS		11411	11982	12581	13210	13870	14564	15292	16056	16859	17702
DEPRECIATION		8355	12254	11697	11697	11697	0	0	0	0	0
NET OPERATING INCOME		3056	-273	883	1512	2173	14564	15292	16056	16859	17702
INCOME TAXES		1406	-125	406	696	1000	6699	7034	7386	7755	8143
NET INCOME		1650	-147	477	817	1173	7864	8258	8670	9104	9559
DEPRECIATION		8355	12254	11697	11697	11697	0	0	0	0	0
NET OPER. PROFIT AFTER TAX		10005	12107	12174	12514	12871	7864	8258	8670	9104	9559
TOTAL CASH FLOW		24771	12107	12174	12514	12871	7864	8258	8670	9104	9559
PRESENT VALUE	-55701	20471	8269	6872	5838	4962	2506	2174	1887	1637	1421
NET PRESENT VALUE	337										

DISCOUNT RATE	0.21					
MARGINAL TAX RATE	0.46					
ENERGY ESCALATION RATE	0.05					
FUEL ESCALATION RATE	0.05					
DEMAND ESCALATION RATE	0.05					
DEPRECIATION SCHEDULE		0.15	0.22	0.21	0.21	0.21
INVESTMENT TAX CREDIT	0.10					
ENERGY TAX CREDIT	0.10					

Figure 2-18. Net Present Value Calculation for 21% Discount Rate

Table 2-14. 1982 Electricity Costs for Commercial and Industrial Users in Major United States Metropolitan Areas*

City	Commercial	Industrial
	cents/kWh	
Buffalo	8.5	6.4
Kansas City	8.2	5.3
Louisville	5.3	4.0
Milwaukee	7.4	4.6
Nashville	5.4	4.6
Newark	10.0	7.6
Richmond	6.8	5.2
San Antonio	7.0	5.5
Atlanta	8.0	4.9
Chicago	9.8	7.1
Detroit	7.7	6.0
Fort Worth	6.9	4.8
Los Angeles	5.9	6.0
Miami	6.2	5.6
New York	15.1	12.0
Seattle	1.4	0.9

*As reported in Energy User News Newspaper, April 4, 1983 and January 17, 1983.

3. DESIGN REFINEMENTS

The system as designed during Phase II required several design refinements for it to work as expected. These refinements were successfully implemented.

3.1 Feed Pump

The feed pump performed well and exceeded our expectations based on Phase II results. This, however, was the result of some design refinements and our ability to develop an understanding of its legitimate requirements.

3.1.1 Pump Cavitation

During early runs, cavitation occasionally set in, disrupting pump performance. We identified as a potential cause the leakage between the gerotor and the inducer housings (see Figure 3-1). The pressure in the gerotor pump housing lies between the suction and discharge pressures, probably closer to its discharge pressure. This pressure is much higher than the inducer housing pressure (as much as 100 psid). As a result, freon leaked and expanded through the labyrinth seal separating the two pumps. The inducer's impeller was equipped with equalizer holes to balance the pressure across the faces of the impeller to reduce axial thrust. These passages, shown in Figure 3-1, allowed the leaking freon to mix directly with the inducer pump suction flow decreasing its available net positive suction head (NPSH). The result was that the inducer pump would cavitate. The gerotor pump, which relies on the inducer to provide it with enough NPSH, would in turn start cavitating and the effective pump throughput would drop drastically. These phenomena were accompanied by loud cavitation noise and high amplitude vibrations, which increased the chances of a component failure.

We plugged the passages in the impeller - axial thrust was taken by the gerotor pump thrust bearing - and replaced the labyrinth seal with the original lip seal that came with the gerotor pump. Further, to eliminate the pressure drop across the lip seal and increase its life expectancy, we added a bleed line from the gerotor pump housing to the condenser.

These changes reduced the occurrence of cavitation but did not eliminate it. We eventually came to realize that the foremost cause of cavitation is a drop in liquid level in the freon receiver. This caused vapor and liquid to be sucked simultaneously into the pump's suction port, effecting a low NPSH condition. Thus, we added more freon to the system until cavitation noises ceased.

3.1.2 Pump Seal Noise and Motor/Generator Frosting

The feed pump, as designed (see Figure 3.1), was equipped with two brass labyrinth seals; one on the inducer side, the replacement of which was described in subsection 3.1.1, and one on the motor/generator side, discussed here. This seal limited shaft leakage from the gerotor pump housing to the motor/generator. Early in the tests we determined that a portion of the pump noise emanated from the vicinity of this seal. Upon examination, we found that the shaft rubbed against the seal's teeth. Pump bearing clearance was large enough to allow contact under pressure. We enlarged the inside diameter of the seal by 0.004 in. to accommodate the bearing clearance. This eliminated the noise but caused enough leakage to frost up the motor/generator (which is at evaporator pressure) and to slightly raise the evaporator pressure. The labyrinth seal was temporarily replaced by a lip seal (the same kind used in the opposite end).

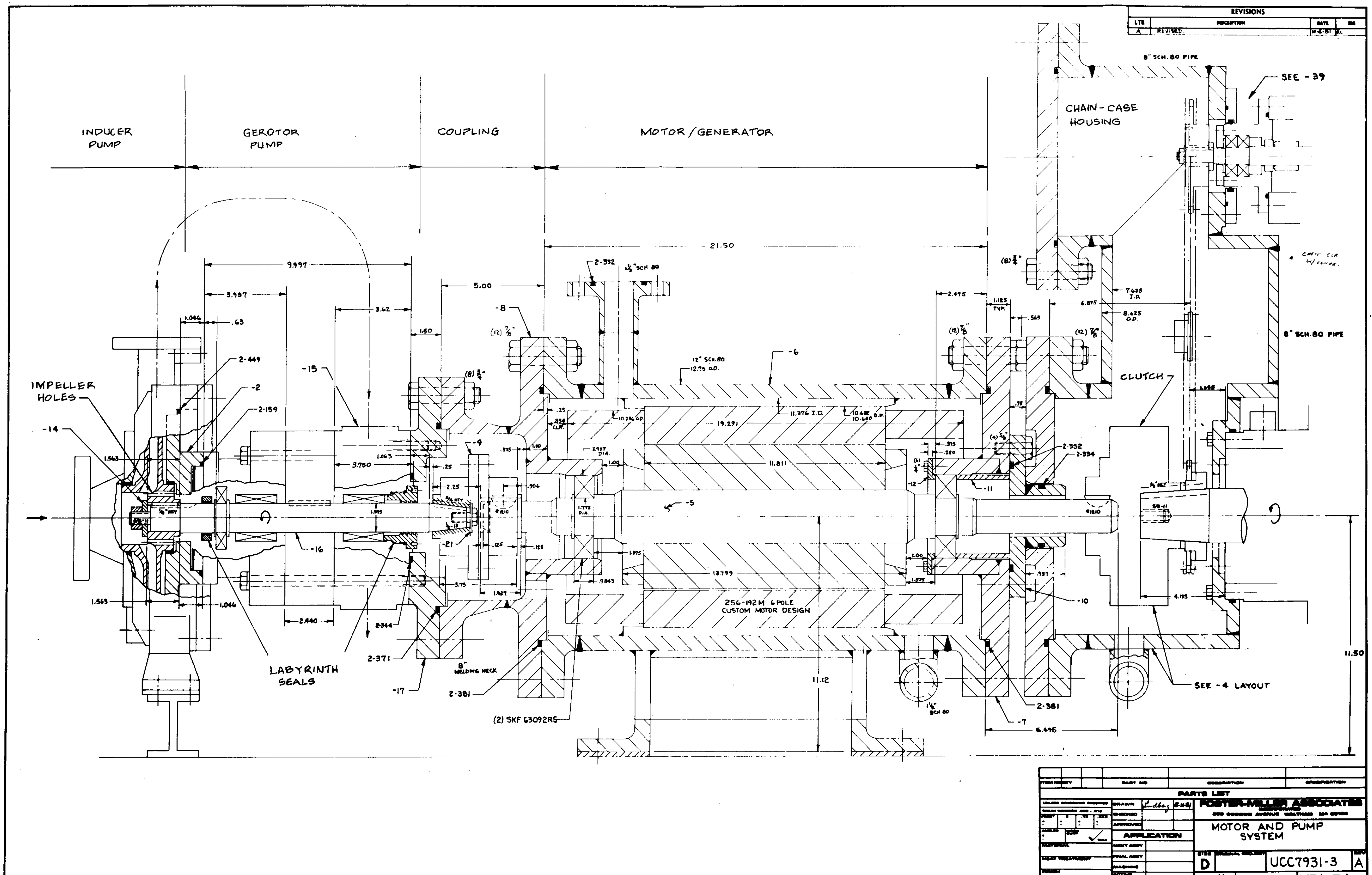


Figure 3-1. Feed Pump, Motor/ Generator and Chain Case Housing Before Modification

The frosting stopped and the measure proved effective; however, the lip seal could not long withstand the pressure difference across it (over 200 psid). We redesigned the housing to accommodate a mechanical face seal. When we replaced the lip seal with the face seal, the former had started deteriorating and portions of the lip were broken loose. The face seal's life is limited by the wear rate of its graphite gland against its ceramic counterpart. The microfinish on the ceramic gland should extend this life to several thousand hours. Figure 3-2 shows the pump layout after the above modifications.

3.2 Clutch

3.2.1 Modifications

The clutch was originally included in the design for the following reasons:

- Ease of start-up
- Testing of the power cycle (feed pump - boiler - condenser) independent of the expander
- Safety feature in case of a catastrophic blockage.

After a few runs, it became apparent that the clutch was too delicate a device for the rugged service it was called to render.

The first manifestations of clutch problems were in terms of slippage. At start-up, when the motor/generator was turned on, the clutch would slip and fail to crank the expander. Even though the clutch was oversized, this did not adequately compensate for the effect of oil in its environment. To increase the torque rating, we doubled the voltage across its magnet's windings. This reduced the slippage but did not eliminate it. During one such test, the clutch burned out after a shear pin failure brought the expander to a halt (see subsection 3.3). We decided to bolt the clutch plates to each other. The fix was promptly accomplished transforming the clutch into a coupling which, apart from the spring leaves on the backplate, was completely rigid. This arrangement worked for a short time.

3.2.2 Replacement by Chain Coupling

The bolted clutch coupling eventually failed during an isolated occurrence of liquid ingestion by the expander. Excessive liquid entered a cylinder when the boiler level control was raised above the boiler midpoint in an attempt to establish the optimum freon level. Even though relief valves are built in the valve housing to relieve any liquid in the cylinder, the impact of the moving piston on the incompressible liquid transmitted a strong enough shock to the crankshaft to break the spring leaves in the clutch and to break another shear pin. Cylinder overpressure also damaged the pressure transducer in the affected cylinder. The boiler level control was lowered and the clutch was replaced by a chain coupling. This coupling performed its function (as a coupling) satisfactorily but generated new problems which resulted in its replacement by a flywheel as described in subsection 3.4.

3.3 Rotary Valve

3.3.1 Clearances, Asymmetry

The clearance between the rotary valve and its housing was chosen to minimize the leakage of high pressure gas into the low pressure chambers. When we received the housing block and the rotary valve from contracted machine shops, the diametral

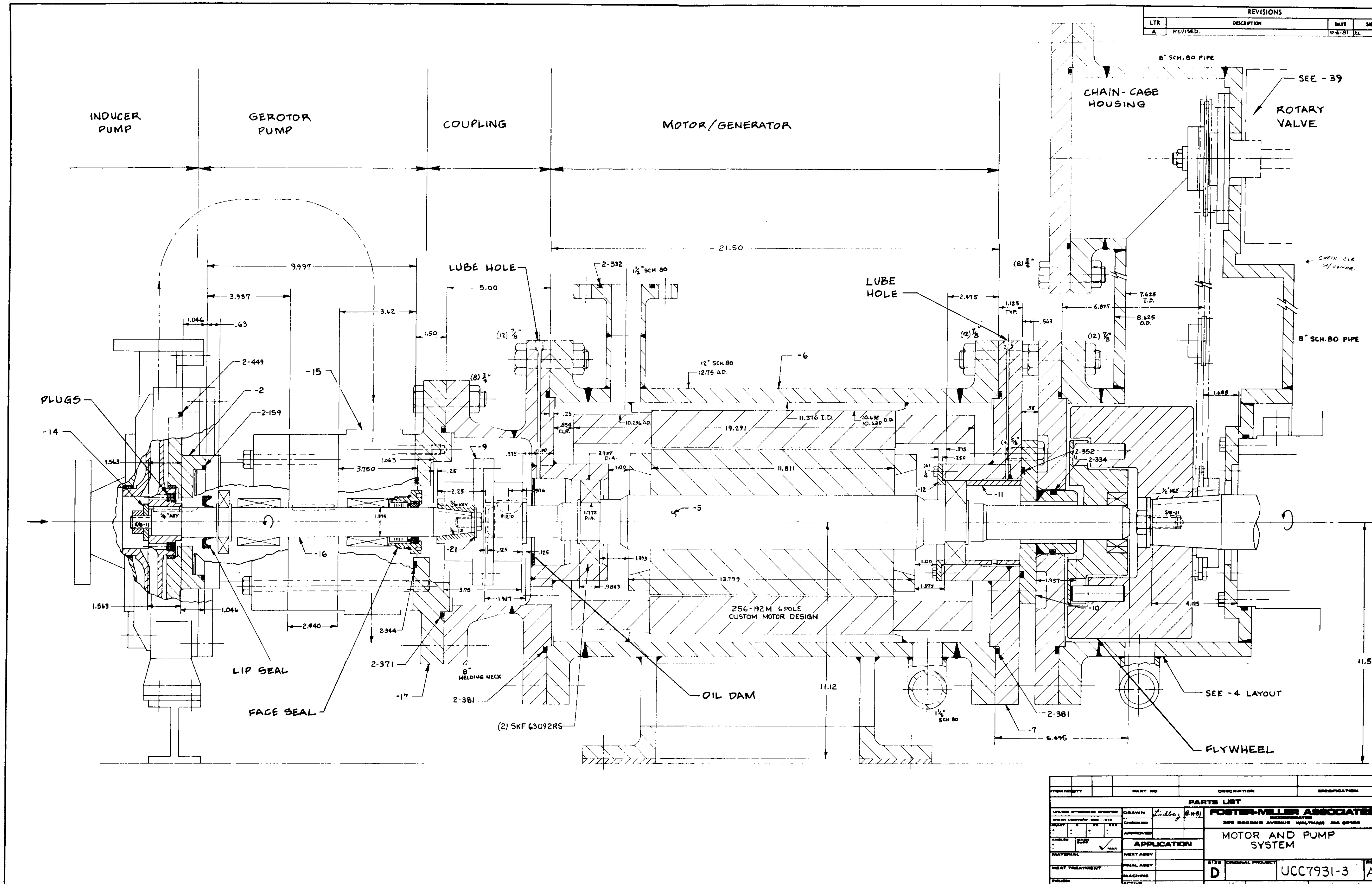


Figure 3-2. Pump, Motor/Generator and Chain Case Housing After Final Modifications

clearance was between 2.2 to 2.6 mil (0.0022 to 0.0026 in.). This clearance did not, however, prevent the valve from rubbing against its housing when the bearings were loaded up by differential pressure across the valve. Here, we describe the measures we took to eliminate this problem.

During operation, the pressure differences across valve surfaces induce a moment that tends to cock the valve (see Figure 3-3). Contrary to expectations, the clearances between the bearing surface and the housing, superimposed to the clearance in the bearings themselves, were found large enough to allow contact between the valve and the housing under the action of this moment.

To eliminate contact, the valve was sent back to a machine shop. To minimize the increase in leakage, it was desired to shave the contact areas marked in Figure 3-3 exclusively. Hence, the valve was clamped in the lathe chucks at a slight angle. The metal removed varied linearly from 0 mil at the valve midpoint to 1 mil at the edge. Thus, the increase in clearance was asymmetric in the sense that only the areas in the second and fourth quadrants of Figure 3-3 were machined. Quadrants 1 and 3 remained unchanged. The resulting geometry was equivalent to two truncated asymmetric cones with bases joined at the midpoint.

When we received the valve, we assembled the expander (indoors), rented an air compressor, and ran the unit on compressed air. This test was a complete success. The expander was moved outside and mounted in its final configuration. Due to higher pressure drops across the valve, runs using freon revealed additional contact areas which did not surface in the air tests. This friction repeatedly caused the shear pin to fail and arrested the operation. Each time, the contact areas were filed to remove metal and the tests repeated until we obtained a completely contact-free valve.

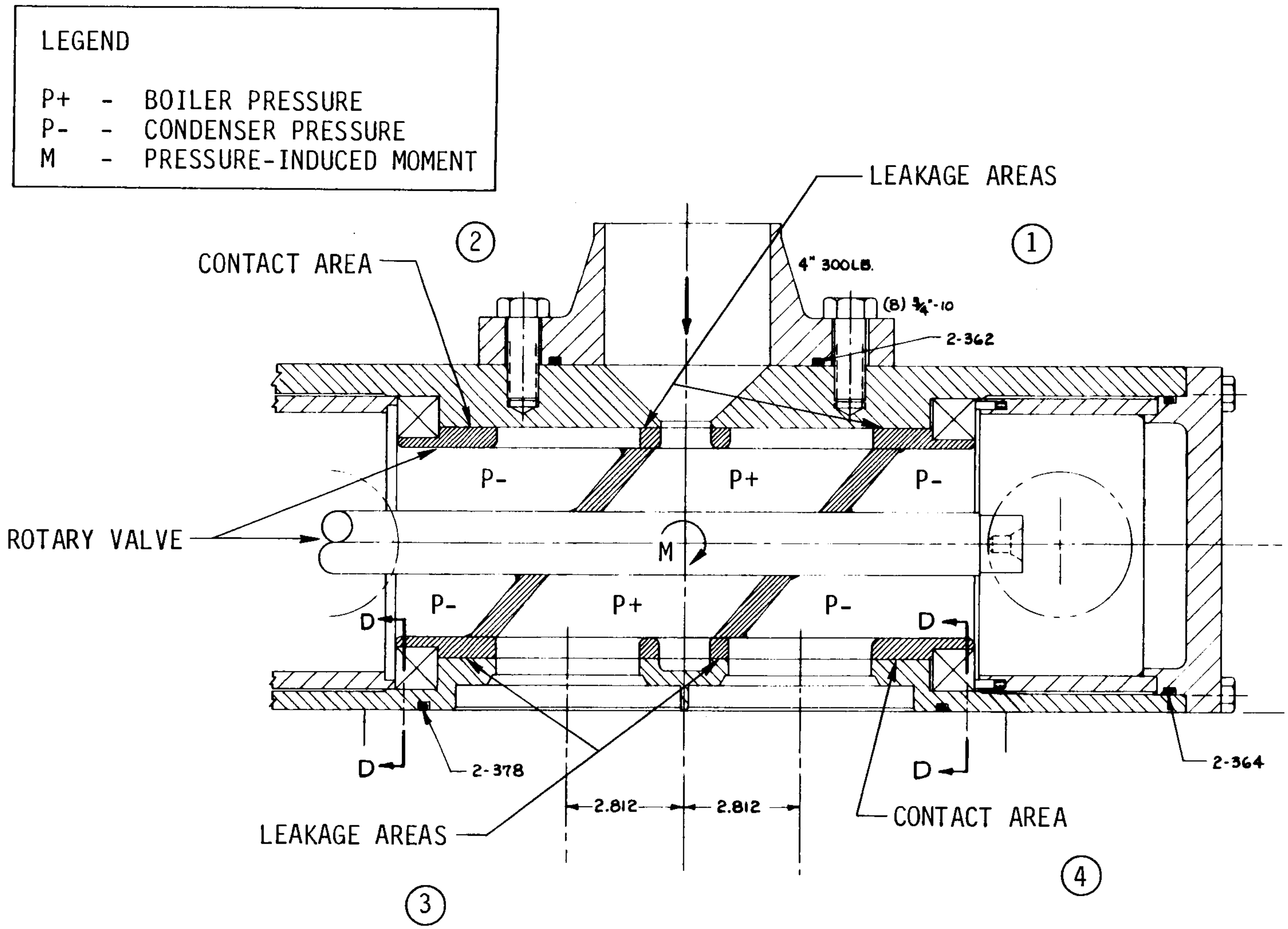


Figure 3-3. Rotary Valve Friction and Leakage

3.3.2 Leakage - Labyrinth Grooves, Coating

The problems caused by the valve's cocking were not limited to friction torque. Those areas where the clearance increases as the valve tilts (see Figure 3-3) cause a surge in gas leakage, reducing both expander volumetric efficiency and mean effective pressure (MEP). We checked this loss by coating the valve with Teflon and machining labyrinth grooves in those parts of the valve we felt would most benefit by them.

The rate of seepage across the leakage areas in Figure 3-1 is inversely proportional to the width of the sealing section (the path the gas has to cross). Thus, the narrow sealing areas close to the midpoint of the valve are of greater concern than the wider ones toward the ends.

We considered increasing the effective resistance to leakage by three methods:

- Labyrinth grooves
- Graphite rings (placed in grooves)
- Flame spraying or coating.

The graphite rings idea was discarded once we recognized the difficulties involved in inserting a ring-equipped valve into the housing. When assembling the valve, the compressed graphite rings would expand into the portholes of the housing and prevent the valve from going in any further. To resolve this problem would involve removing the housing and machining the interior edges of the port areas to allow smooth entry of graphite rings. Even though the rotary valve itself is easy to remove, the removal of the housing would involve the disassembly of a major portion of the entire system. Time limitations did not permit such an undertaking. Furthermore, there was no guarantee that this operation would indeed permit easy assembly of the graphite rings.

Labyrinth grooves were only deemed necessary for the narrow sealing sections and two such grooves were machined on each side of the intake ports.

After considering a variety of coatings, we selected a relatively soft Teflon coating with polyamide binder. The coating contractor believed he could apply a coating of less than 1.5 mil radial thickness. However, after the valve was returned to us, it failed to fit the valve housing. The average thickness of the actual coating was 2.5 mil radial, approaching 3 mil on the high points. The valve was sent back, the old coating was baked off and this time a pure Teflon coating, 1 mil thick radially, was applied under more controlled conditions. The valve fit the housing with no friction and after a few tests, was worn off only at the contact areas of Figure 3-3. Thus, we were able to reduce the diametral clearance at leakage points without causing a permanent interference at the contact points. Unfortunately, the test data obtained before these modifications were not reliable enough for us to be able to make a quantitative evaluation of these improvements.

3.3.3 Shaft - Run-Out, Spline

Shear pin failure occurred at least four times, at different stages during the testing period, to challenge our understanding of the principles which governed our machine. These cases were the following:

- Liquid ingestion (discussed in subsection 3.1.2)
- Friction torque (discussed in subsection 3.3.1)
- Shaft whip (discussed in subsection 3.3.3)
- Torsional vibrations (discussed in subsection 3.4).

When the resolution of the friction torque problem failed to stop shear pin failures, we focused our attention on other possible causes. We had noticed that after the recent valve seizures, the valve shaft was slightly bent. This was attributed to the length of the cantilever arm (see Figure 3-4) and the high instantaneous bending loads on the shaft during failures. When the valve seizes, chain tension climbs momentarily, pulling (bending) the valve shaft toward the crankshaft. Shaft run-out could in turn induce a fluctuating bending load on the shear pin, causing its failure.

To prevent the valve shaft from transmitting any bending load to the valve, a spline coupling was employed. A male spline (external spline) was welded to the shaft while its female counterpart (internal spline) was rigidly pinned to the rotary valve. To support the bending load, a ball bearing was mounted in a housing bolted on the open face of the seal hub.

This bearing supports the shaft less than 1 in. from the line of force and hence reduces the cantilever effect drastically. Figure 3-5 shows the valve with the foregoing modifications.

3.4 Torsional Vibrations

When the system was originally designed, the motor/generator rotor was expected to serve as a flywheel to dampen expander torque fluctuation. This function it faithfully performed until we were obliged to substitute the clutch with a chain coupling (see subsection 3.2.2). A chain coupling is not a perfectly rigid connection; therefore, most of the torque fluctuations at the crankshaft were not transmitted and absorbed by the motor/generator. Instead, these fluctuations were absorbed by the rotary valve.

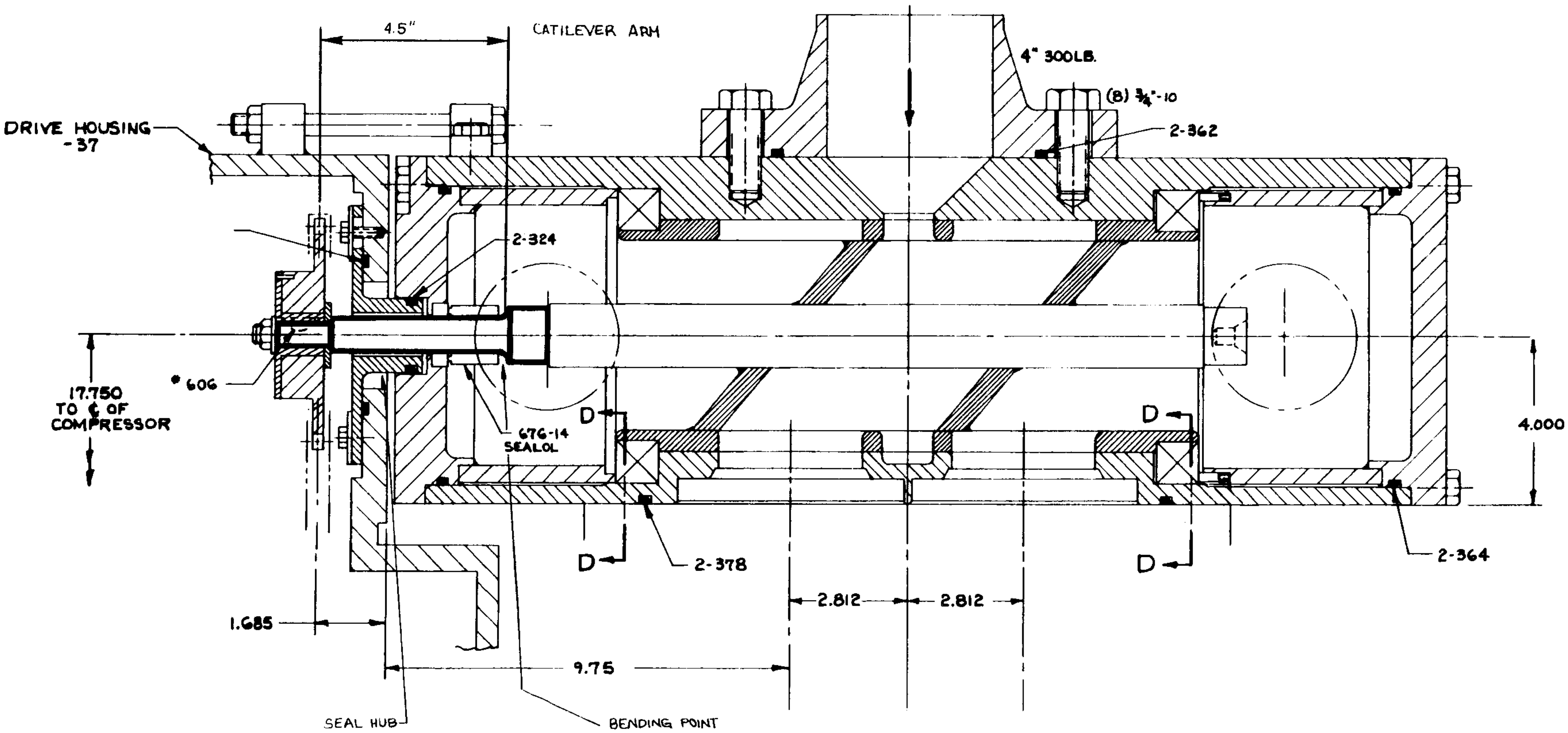


Figure 3-4. Rotary Valve Drive Shaft Before Modifications

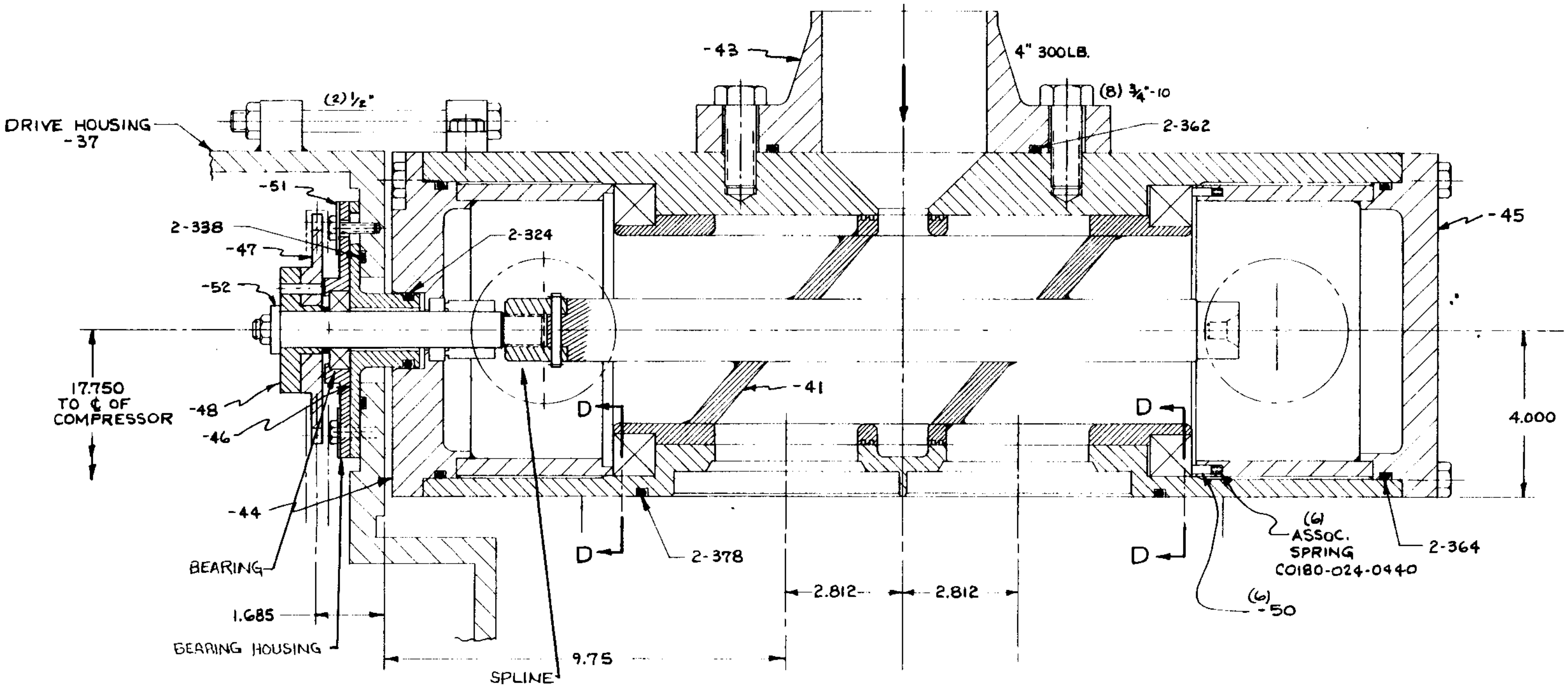


Figure 3-5. Final Rotary Valve Configuration

Early in the testing, the shear pin failed due to friction torque. When this was eliminated, the valve became a free mass of substantial inertia. To impose high torque fluctuations on this mass was bound to cause very high instantaneous torsional loads on the valve drive train. The shear pin, being the weakest link in this train, was the first to fail.

The first attempt to reduce the fluctuating load on the valve was by adding rubber pads in the timing gear. These had the effect of a torsional spring, coupling the drive sprocket to the valve shaft. This modification was inadequate, for the rubber pads were so vigorously pounded by torque fluctuations that they were extruded out of their housing. More shear pin failures were recorded. At last it was decided to redesign the drive train and add a flywheel with a substantial moment of inertia.

3.4.1 Flywheel

When one bank of the York compressor cylinders were converted to act as expanders, the torque output balance of the machine was substantially disrupted. Due to the V configuration, when one expander cylinder is at its maximum torque output position (approximately 90 deg after top dead center) the corresponding compressor cylinders are at top and bottom dead center or their minimum compound torque requirement stage. Conversely when a compressor cylinder is at its maximum torque requirement position, both expander cylinders deliver virtually no torque output. Thus, during each cycle, crankshaft torque fluctuates between a large positive value (maximum expander torque output) and a large negative value (maximum compressor torque input). This dramatic reversal in the acceleration of the crankshaft causes its angular speed to fluctuate about the nominal 1200 rpm. Our calculations showed that it was sufficient to contain angular fluctuations within 1 deg to avoid excessive local stresses in the drive train. To accomplish this required a flywheel with a moment of inertia of about 10 $lb\text{-}ft^2$.

Figure 3-6 shows the flywheel design. The chain coupling was to be removed, hence the flywheel was to couple the expander and the motor/generator shafts as well. The massive end was mounted directly on the crankshaft to absorb torque variations right at their origin. A specially designed hub was clamped on the motor/generator shaft using a Ringfeder® locking assembly. The coupling is achieved by four dowel pins closely fitted into holes drilled in the flywheel. The other ends of these pins are fitted into square blocks which fit into special compartments in the hub. These compartments are wide enough to allow hard rubber pads to be permanently squeezed between the square blocks and the compartment walls. The result is a semi-rigid coupling between the motor/generator and the crankshaft. The rubber pads act as torsional springs of high stiffness to dampen abrupt fluctuations.

Figure 3-2 shows the chain case housing after the inclusion of the flywheel. Since the installation of the flywheel, no shear pin failure has occurred.

3.5 Motor/Generator Bearings

After 15 hr of accumulated operation, one of the motor/generator bearings became noisy and eventually seized, bringing the motor to a stop. The failure was caused by the freon vapor washing off the oil from the bearing. The accelerated wear had cracked the retainer which was jammed between the balls and the races. We decided to lubricate both motor bearings using one of the ports of the Bijur lubricator. Two lube holes were drilled in the motor/generator flanges and oil dams were

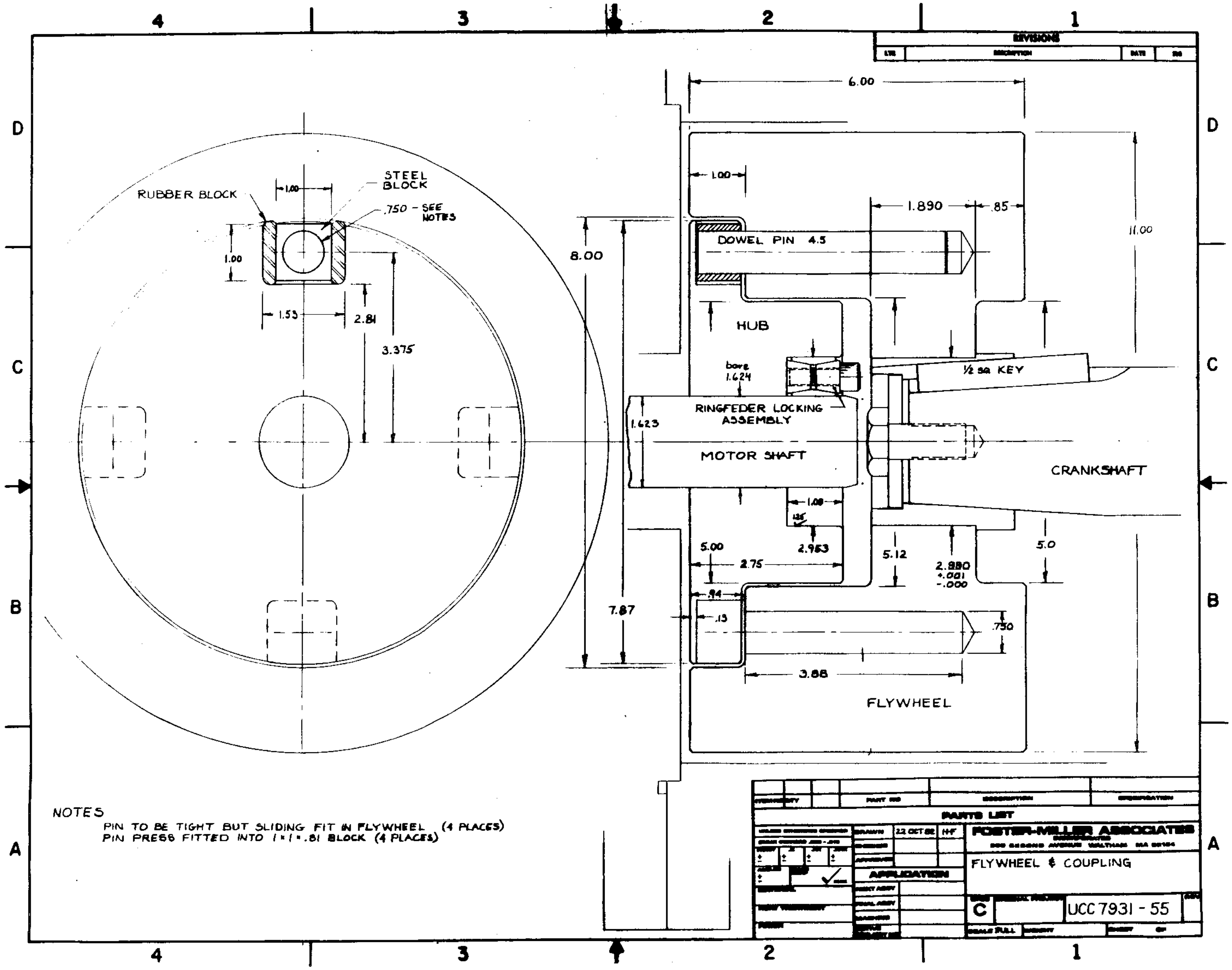


Figure 3-6. The Flywheel

pressed in the bearing housing to trap oil (see Figure 3-2). The Bijur lubricator was piped to the lube holes with a check valve to prevent back flow. The bearings are lubricated with one squirt of oil per minute during operation. The motor/generator has run trouble-free ever since.

3.6 Summary

As is to be expected with any mechanical development of this magnitude, unforeseen problems arose. Some were complex, especially those in which several coexistent problems yielded the same symptoms. Some of the solutions involved considerable effort, such as installing a massive flywheel in the least accessible part of the system. However, all of those problems were indeed solved by means of the design refinements discussed in this section.

4. CONCLUSIONS AND RECOMMENDATIONS

It could be concluded that we have developed an expensive but high-performance mechanical equivalent of an absorption chiller. Its performance is high in the sense that it can accept its driving heat at relatively low (and presumably more available) temperatures, and that it can provide its refrigeration at relatively low (and presumably more useful) temperatures. It is expensive because it uses large moving components with close tolerances. However, it would be a mistake to conclude that this is the only result of this effort.

4.1 Contributions to the Technology

We have demonstrated some things to be possible which were not previously demonstrated in operating machinery.

4.1.1 Rotary Valve

Reciprocating expanders have generally been found to be impractical to operate at low pressure ratios because the breathing losses imposed by the valves were a significant fraction of the power output. A typical poppet valve arrangement might provide an intake valve area and an exhaust valve area each equal to 25% of the piston area.

The rotary valve developed in this project uses the same large piston head port for intake and exhaust, and provides intake and exhaust valve area each equal to 61% of the piston area. With breathing losses proportional to the inverse of the area squared, this cuts breathing losses by a factor of 6. Because this rotary valve cuts the breathing losses so drastically, we were able to achieve isentropic efficiencies as high as 63%, and volumetric efficiencies as high as 70%, with pressure ratios on the order of 3:2.

This does not mean that all reciprocating engines should use rotary valves. At high pressure ratios, breathing losses can be relatively insignificant, so their prevention assumes less importance. Also, the bearings supporting a rotary valve are exposed to the exhausting working fluid; this presents constraints on the nature and exhaust temperature of the working fluid. Still, the rotary valve developed in this project should be applicable to a greater class of expanders.

4.1.2 Conversion of Compressors to Expanders

Given the above solution to breathing problems at low pressure ratios, the reciprocating expander competes seriously with the turbine expander. It is useful to know that part or all of a reciprocating compressor can be converted to a reciprocating expander. It is even more useful to know what steps and what caveats are involved in this conversion.

The conversion of compressor to expander is described in subsections 5.2.3 and 5.3.3 of Reference 1. The steps include replacing the existing head(s) and reed valves with a rotary valve and housing, replacing the wristpin sleeve bearings with needle bearings, and modifying the piston crown contour.

The caveats refer mostly to pressure limits and depend on whether part or all of the compressor is being transformed into an expander. The crankcase design pressure limit must be respected; in an expander-only configuration, the crankcase will be at the exhaust pressure (if vented to exhaust), but in a compressor-expander combination the crankcase will be at compressor suction pressure, which is much lower.

There is also a maximum pressure difference to be allowed across the expander piston, as determined by the maximum journal bearing loading; in an expander-only configuration, this is the same pressure difference as is applied to the expander, but in a compressor-expander combination it is the difference between expander intake and compressor suction pressures. Finally, there must be at least two expander pistons, lest the pressure difference across the expander presents a formidable starting load.

4.2 Successful Design Concepts

Section 3 of this report can leave one with the inaccurate impression that none of the design concepts of Phase I could be successfully reduced to practice in Phase III without major modifications. As with any development project, there were problems to be identified and solved after the initial design stage. Yet several of the original design concepts survived intact and deserve mention for their success.

Foremost among these was the very notion that the power and refrigeration cycles could be so thoroughly integrated. They were integrated with respect to a common working fluid, a common condenser, and a common hermetic housing for the moving parts.

Both cycles used the same working fluid very nicely. Neither cycle bled the other dry while flooding itself, for example; there was an inherent stability in this respect. An inadequate charge would manifest itself first in the refrigeration cycle in the form of reduced capacity due to a starved evaporator. Still less charge would affect the power cycle in the form of a cavitating boiler feed pump. Given an adequate charge, however, both cycles shared the working fluid well.

Putting all moving parts in a single hermetic housing was intended to eliminate the troubles associated with open shaft seals. This it did. There was still a requirement for one fairly elaborate internal shaft seal between the gerotor pump and the motor/generator. A labyrinth seal at this point did not prevent a significant flow of liquid from entering the motor/generator housing and flashing there, which decreased the compressor capacity available to the evaporator. The gerotor pump shaft bearing clearance prevented tightening the clearances on this labyrinth seal, which made a mechanical face seal necessary.

Another successful design concept was the use of a gerotor pump as a freon boiler feed pump. This did require some machining to achieve the optimum clearance (per subsection 3.1 of reference 1) and the incorporation of a centrifugal inducer to prevent cavitation (ibid), but the pump achieved volumetric efficiencies of typically 85% and hydraulic efficiencies in the range of 70 to 80%.

4.3 Barriers to Commercialization

While it is true that this system has met its performance goals, it is also true that its commerical prospects are limited. The applications for this system are not as numerous as one might think. This becomes apparent upon carefully considering the constraints on what comprises an appropriate application.

First, there must be a continuous source of waste heat at or above 140°F. While the system will operate with lower waste heat temperatures, the motor/generator will require additional electric power, reducing the return on investment. If the system is operated anything less than full time, the payback time will be extended proportionately. Many sources of waste heat are available only seasonally, and not all continuous sources are always hotter than 140°F; see Figure 2-4 to see the effect of lower temperatures on performance.

Furthermore, for the purposes of this system, a heat source can be considered waste heat only if it would otherwise be thrown away <u>and</u> if there is no other, more cost-effective way to reclaim this heat. For example, suppose there is a source of heat that is currently being thrown away. Suppose also that nearby there is a need for hot water. If by simply installing an inexpensive heat exchanger this heat source <u>could</u> be reclaimed with a higher return on investment than our system could provide, then that currently unused heat source cannot be considered waste heat.

Second, there must be a <u>continuous</u> requirement for refrigeration at an evaporator temperature between <u>0°F</u> and <u>35°F</u>. Both the waste heat source and the refrigeration requirement must be continuous for the system to operate continuously and to pay back quickly. The lower limit on evaporator temperature (0°F) is due to pressure limits on the expander (see subsection 4.1.2). The upper limit (35°F) is the lower limit at which a lithium bromide/water absorption chiller can operate; above this limit the less expensive absorption chiller is more cost effective.

Third, the waste heat source and the refrigeration requirement must be close to each other. Both the additional first cost and the heat losses associated with long piping runs detract from the return on investment.

Fourth, and perhaps the least obvious of all the constraints, the waste heat source and the refrigeration requirement must be associated with the same industrial process. If this system couples two processes such that the refrigeration requirement of Process A depends on the waste heat generated by Process B, then any shutdown in Process B could force a shutdown of Process A if the power demanded by the refrigeration cycle happened to exceed the capacity of the motor/generator.

Each of these constraints is economic and each would be lessened by any increase in the return on investment of the system (such as is afforded by operating at higher temperatures). However, they do seem to account for the observed lack of interest shown by industry (see Appendix C of Reference).

4.4 Future Pursuit of the Concept

We should keep in mind the fact that this system does what no other system can do. It transforms 140°F waste heat into 0°F refrigeration.

What would it take to enable this system to improve its return on investment?

1. Higher electric rates or lower discount rates, either of which is quite possible.

2. Higher waste heat temperatures, which would permit a smaller freon boiler. Higher pressures, though also possible with higher waste heat temperatures, would exceed component pressure limits.

3. Larger systems (economies of scale), or more systems (production-oriented design).

4. The opportunity to do without the evaporative condenser, which is an expensive component in terms of both initial and operating costs; a source of cooling water would permit the use of a water-cooled condenser.

Perhaps this system could be used in conjunction with a solar pond, in remote areas where grid electricity is expensive. The lower (hot) layer could be pumped directly to the boiler and the upper (cool) layer could be pumped directly to a water-cooled condenser that would replace the evaporative condenser. Such areas are hard to find in this country, but are found overseas; perhaps this system would be attractive to the military.

Another possibility is that certain components might find application where the entire system might not. By converting all four cylinders to expander cylinders and eliminating the regrigeration cycle entirely, the system could be used to generate electricity in remote regions.

Our recommendation is that the singular capabilities of this system and of certain of its components be remembered by those who need refrigeration or electricity in unusual circumstances. The identification of such an application might then justify further testing for durability.

APPENDIX A

OPERATING PROCEDURE

The Phase III waste heat expander was designed to operate with minimum supervision. In fact, once the system is started, it does not require continuous operator intervention. During the initial hours of the operation, several adjustments should be made which, once set, require only periodic attention. These adjustments define the system parameters and are set to optimize the performance of the system. A periodic check on these settings is necessary to ensure continuous operation and optimum performance.

This section discusses the sequence of events that lead to starting, adjusting and shutting off the system. Those instructions marked by an asterisk are pertinent to the testing period of the machine and are not required during continuous operation. They were included to illustrate our testing procedure.

Refer to Appendix B for valve identification.

a. Start-Up Procedure

1. Start with a pumped down system (see shut down procedure steps 9 and 10).

2. Check all valve positions against the following list:

Subcooler, in	open
Subcooler bypass	open
Evaporator filter dryer, in	closed
Evaporator filter dryer, out	open
Evaporator, out	both open
Bristol inlets	both closed
Bristol, out	open

Pump bleed	closed
Hot gas bypass	closed
M/G cooling	open
Compressor, in	completely open
Enthalpy samplings	both closed
Crankcase equalizing	closed
Receiver, inlet	open
Pump bypass	open 1-1/2 turns
Pump, out	open
Receiver equalizing	closed
Condenser level indicator	closed
Oil separator valves	all open
Charge lines	all closed
*Tecumseh bypass	open
Expander out pressure gauge	open
Pump, in	closed
Expander, out	closed
Expander, bypass	closed

3. Check oil level in crankcase and Bijur lubricator (the Bijur will require a larger oil reservoir for continuous operation).

* 4. Connect pressure transducers' and the rotary function generator's cables. Turn the oscilloscope and the pressure transducers' power supply on.

* 5. Record Motor/Generator (M/G) and pump bypass hour meters' readings. Record initial dry bulb and wet bulb temperatures.

6. At the electrical console: turn unloader switch on, hot gas bypass off, oil separator off, Tecumseh off. Set evaporator load to off and expander bypass to auto.

7. Open the evaporator filter dryer inlet ball valve.

8. Turn electrical console on (both 110V and 480V).

9. Start hydronic and brine pumps.

10. Start condenser pump and fan.

11. Open the expander bypass valve slowly.

12. When boiler pressure builds up to condenser pressure, open expander-out, and pump-inlet ball valves.

13. Turn Bijur and wattmeter on.

14. Simultaneously, start the motor/generator and turn pump-bypass switch to auto. Consequently, turn the evaporator load switch to half.

*15. Turn Peerless boiler on immediately and turn other hydronic heaters on as the pressure in the freon boiler rises. Make sure the brine coil throttle valve is open sufficiently for the brine not to freeze.

16. Once the wattmeter reads below 10 kW, turn evaporator load switch to full.

17. Open pump bleed valve.

b. Operational Adjustments

1. Boiler level

- Float level position: open the pump bypass throttle valve completely. The boiler level will cycle (so will the boiler pressure) as the feed pump bypass solenoid valve alternates between its open and closed states. If the maximum boiler pressure obtained during this cycling does not reach 295 psig, the float level sensor should be raised in position until the maximum boiler pressure equals 295 psig. (Note: do this in conjunction with step 2 and only when all water heaters are operating and the hydronic-in temperature equals or exceeds 140°F, otherwise boiler pressure may never reach design point, regardless of the float level position, due to inadequate hydronic input).

- Pump bypass throttle position: close the pump bypass throttle valve in small steps until a steady level is established in the freon boiler. accomplished when the pump bypass solenoid remains open and does not cycle (make sure you don't overfill the boiler).

2. Pressure regulators: adjust the large boiler back pressure regulator for a boiler pressure of 295 - 300 psig. (This step is most safely done in conjunction with the boiler level adjustment described in step 1). Adjust the small regulator setting for a boiler pressure of 285 - 295 psig.

3. TEV superheat: adjust the two evaporator main TEV superheat settings to a completely open (least superheat) position. Adjust the oil let down line TEV to

one-third way closed and the augmentation line TEV to two-thirds way closed position. Open the oil let down line and TEV augmentation line ball valves.

4. Subcooler bypass: close subcooler bypass valve (Freon side) to a position just short of flashing at the subcooler exit, which starves the evaporator; this condition is apparent when the subcooler water inlet temperature exceeds the subcooler freon outlet temperature. Close subcooler water side bypass valve.

*5. Evaporator pressure: adjust the hydronic heat coil throttle valve to obtain desired evaporator pressure (design point = 43 psig).

*6. Brine-in temperature: see evaporator pressure

*7. Hydronic-in temperature: adjust aquastats on water heaters to obtain desired hydronic-in temperature.

8. Unloader pressure switch: set unloader pressure switch to the minimum crankcase pressure permissible (25 psig).

9. Hot-gas-bypass regulator: starve evaporator by turning the unloader switch off and evaporator switch to half load. Open hot-gas bypass ball valve, turn the hot-gas by-pass regulator pressure adjustment to the lowest setting (all the way counterclockwise) and turn hot-gas-bypass switch on. Crank up the pressure setting on the regulator until a pressure slightly less than the desired evaporator pressure is established (40 psig design point). Turn evaporator load to full and unloader switch to on.

10. Oil separator line: turn the oil separator switch on and watch the crankcase oil level for an hour until a steady level is established. If this level is too high or too low, drain or fill the crankcase with refrigeration oil (400 SSU).

*11. Enthalpy sampling lines: open both needle valves a quarter of a turn. The Tecumseh is not needed for their operation and its use is discouraged.

12. Hot-gas-bypass desuperheating TEV: adjust the superheat setting to a high superheat value (3/4 closed) and open the ball valve just upstream of the TEV.

13. Condenser pressure switches: Set the condenser fan pressure switch to the minimum condenser pressure permissible. Since the TEV's require a minimum pressure drop of 75 psid, for a 20°F evaporator temperature this pressure switch should be set at 43 + 75 = 118 psig. The spray pump pressure switch can be adjusted to any pressure above 118 psig. This is particularly useful when it is desired to maintain condenser pressure around a certain value.

*14. Rotary Function Generator (RFG) Position: Slacken RFG clamp screw and turn RFG until its TDC mark output coincides with the TDC mark output of the magnetic pick up (on the oscilloscope). Tighten RFG clamp.

Operational adjustments periodic checks: All the above settings should be checked from time to time. Those that need the most frequent inspection are: the crankcase oil level, the pump bypass throttle, and the subcooler bypass which should be readjusted every month so that variations in condensing pressure will not starve the evaporator. Condensing pressure also exerts a secondary influence on the performance of the TEV's and pressure regulators which are affected by the pressure difference across them. The settings on these should be inspected at least seasonally. Least likely to change are the settings on the pressure switches which however, for safety reasons, should be checked semi-annually. The float level position should not require readjustment, yet a seasonal check on boiler pressure will ensure whether it is properly positioned. The Bijur lubricator oil reservoir should be refilled at a frequency dependent upon its size. Use refrigeration oil with 400 SSU viscosity.

c. Shutdown Procedure

1. Close the valves in the following lines:

   TEV augmentation line
   Oil let down line
   Desuperheating TEV line
   *Enthalpy sampling lines.

2. Switch system to full load operation (if running at half load).

3. Turn oil separator switch to off.

4. Close evaporator filter-dryer inlet ball valve, immediately afterwards stop the motor generator and turn pump bypass switch to open.

5. Shut the pump inlet ball valve and the pump bleed valve.

6. Turn Bijur lubricator off.

*7. Turn all water heaters off.

8. When all the liquid in the Freon boiler is boiled off, shut the expander bypass and expander-out ball valves.

9. Pumpdown: turn Bristol compressor on, open both Bristol suctions and the crankcase equalizing ball valve (make sure you don't slug the Bristol with liquid).

10. Pump system down to 10 psig, turn Bristol off, close both Bristol suctions.

11. Stop condenser fan and pump.

12. Stop hydronic and brine pumps.

13. Turn the electrical console off (both 110V and 480V).

*14. Turn pressure transducer power supply and oscilloscope off.

15. Turn wattmeter off.

*16. Record final wet bulb and dry bulb temperatures, record final motor/generator and pump bypass hours.

d. <u>After Shutdown</u>

1. Keep crankcase and oil separator heaters on all the time.

2. To adjust timing: pump system down to 2 psig, open system to atmosphere. Remove chaincase housing cover and flywheel sighthole cap. Crank engine to TDC position and check valve timing on timing wheel. Adjust timing mechanism until desired timing is established. Put cover and cap back in place and pull a vacuum on the system.

APPENDIX B

FLOW DIAGRAM AND VALVE IDENTIFICATION

A complete list of all valves and pressure switches is given below. The valves are listed under their category in alphabetical order. The corresponding number shows their location on the following flow diagram.

a. Isolation and Globe Valves

Designation	Valve Number
Brine fill valves	1,2
Bristol inlets	3,4
Bristol, out	5
Bypass lines	6,7,8,9,10,11,12
Charge Valves	13,14,15,16,17,18,19,20
Compressor, in	21
Condenser drain	22
Condenser level indicator	23,24
Condenser spray throttle	25
Crankcase equalizing	26
Enthalpy samplings	27,28
Evaporator filter dryer, in	29
Evaporator filter dryer, out	30
Evaporator, out	31,32
Expander bypass	33
Expander, out	34
Hot gas bypass	35
Hot gas bypass desuperheating	36
Hydronic fill valve	37
Hydronic heat coil bypass	38
Hydronic heat coil throttle	39
Hydrotherms' butterfly valve	40

Designation	Valve Number
Motor/Generator cooling	41
Oil let down line	42
Oil separator	43
Peerless butterfly valve	44
Pump bleed	45
Pump bypass	46
Pump, in	47
Pump, out	48
Receiver equalizing	49
Receiver, inlet	50
Subcooler bypass	51
Subcooler, in	52
Subcooler water side, in	53
Subcooler waterside bypass	54
Tecumseh bypass	55
TEV augmentation line	56

b. Solenoid Valves

Designation	Valve Number
Bristol bypass	60
Evaporator, in	61,62
Expander bypass	63
Oil separator	64
Pump bypass	65

c. Pressure Regulators

Designation	Valve Number
Boiler	70,71
Hot gas bypass (also a solenoid valve)	72

d. Thermal Expansion Valves (TEVs)

Designation	Valve Number
Evaporator	80,81
Hot gas bypass desuperheating	82
Oil let down line	83
TEV augmentation line	84

e. Pressure Switches

Designation	Switch Number
Bristol suction	90
Bristol discharge	91
Condenser fan	92
Condenser spray pump	93
Feed pump discharge	94
Unloader	95

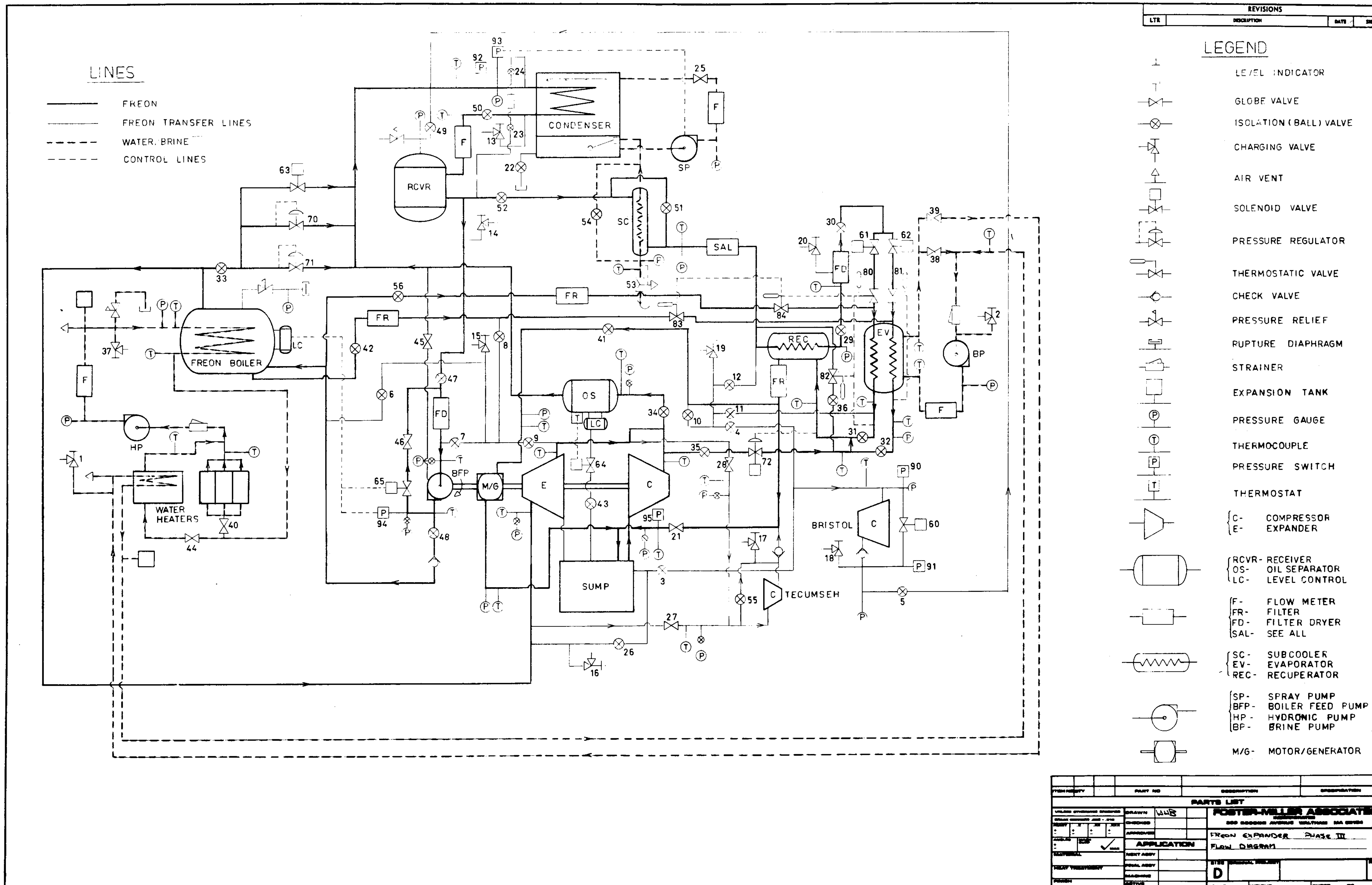


Figure B-1. System Flow Diagram

APPENDIX C

TEST RESULTS

This appendix contains test results covering the period of November 6, 1982 to March 17, 1983. The missing test numbers represent either aborted tests or tests run for a purpose other than measuring system performance.

All pressures are given in terms of lbs/sq in. gage.

PHASE III BRINE CHILLER SYSTEM

RUN NUMBER:3-08   DATE:10/06/82   VALVE TIMING= 0.0 DEGREES ATDC   WET BULB= 63 DEG. F

**** TEST RESULTS ****

POINT	HYD. FLOW GPM	HYD. T-IN DEGF	HYD. T-OUT DEGF	BRINE FLOW GPM	BRINE T-IN DEGF	BRINE T-OUT DEGF	EVAP. T-IN DEGF	EXP. T-IN DEGF	EXP. T-OUT DEGF	COMP. T-IN DEGF	COMP. T-OUT DEGF	COND. T-IN DEGF	COND. T-OUT DEGF	PUMP T-IN DEGF	PUMP T-OUT DEGF	E.S. T-IN DEGF	E.S. T-OUT DEGF	EVAP. T-OUT DEGF
1	400.0	141.6	134.4	69.5	32.8	27.2	79.0	133.0	90.0	40.0	193.0	99.0	68.0	86.0	89.0	64.0	48.0	26.0
2	390.0	141.0	132.9	62.0	48.0	43.0	32.0	128.0	90.0	111.0	220.0	91.0	67.0	85.0	90.0	24.0	12.0	46.0

ABBREVIATIONS: HYD.=HYDRONIC, T=TEMPERATURE, EVAP.=EVAPORATOR, EXP.=EXPANDER
COMP.=COMPRESSOR, COND=CONDENSER, E.S.=ENTHALPY SAMPLING.

NOTE: COMP. T-IN VALUES IN THIS TEST ARE INCORRECT DUE TO THERMOCOUPLE MALFUNCTION.

PHASE III BRINE CHILLER SYSTEM

RUN NUMBER:3-08 DATE:10/06/82 VALVE TIMING= 0.0 DEGREES ATDC WET BULB= 63 DEG. F

**** TEST RESULTS - CONTINUED ****

POINT	POWER USED KW	M/G HOURS	EXP. P-IN PSI	EXP. P-OUT PSI	EVAP. P. PSI	COMP. P-IN PSI	COND.* P-IN PSI	COND.* P-OUT PSI	GPUMP P-IN PSI	E.S. P. PSI	FREON** FLOW GPM	E.S. H-IN B/LB	E.S. H-OUT B/LB	EVAP. H-OUT B/LB	COMP. RO-IN LB/CFT	NOTES
1	6.3	32.60	294.0	173.0	37.0	37.0	173.0	173.0	189.0	15.0	37.0	115.2	112.7	107.8	0.902	F,OF,B
2	5.0	32.94	293.0	176.0	30.0	30.0	179.0	179.0	186.0	15.0	37.0	108.8	106.9	111.4	0.660	F,OF,N

EXPLANATION OF NOTES: EVAPORATOR LOAD- F=FULL, H=HALF
UNLOADER- O=ON, OF=OFF
SUBCOOLER- N=NOT BYPASSED, B=BYPASSED
TAL= TEV AUGMENTATION LINE OPEN
OLL= OIL LET-DOWN LINE OPEN

ABBREVIATIONS: M/G=MOTOR/GENERATOR, EXP.=EXPANDER, P=PRESSURE, EVAP.=EVAPORATOR, COMP.=COMPRESSOR, COND.=CONDENSER
GPUMP=GEROTOR PUMP, E.S.=ENTHALPY SAMPLING, H=ENTHALPY, B=BTU, RO=DENSITY.

* CONDENSER PRESSURE IS TOO HIGH DUE TO CONDENSATION IN PRESSURE GUAGE LINES (CONDENSER IS HIGH ABOVE GROUND).
CLOSEST PRESSURE TO CONDENSING PRESSURE IS EXP. OUT PRESSURE WHICH IS LESS THAN 1 PSI HIGHER.

** FREON FLOW MEASURED BY TURBINE FLOW METER. ACTUAL FLOW IS LESS THAN 90% OF THIS VALUE (SINCE THE FLOW METER READ TOO HIGH).

PHASE III BRINE CHILLER SYSTEM

RUN NUMBER:3-12 DATE:11/22/82 VALVE TIMING= 0.0 DEGREES ATDC WET BULB= 53 DEG. F

**** TEST RESULTS ****

POINT	HYD. FLOW GPM	HYD. T-IN DEGF	HYD. T-OUT DEGF	BRINE FLOW GPM	BRINE T-IN DEGF	BRINE T-OUT DEGF	EVAP. T-IN DEGF	EXP. T-IN DEGF	EXP. T-OUT DEGF	COMP. T-IN DEGF	COMP. T-OUT DEGF	COND. T-IN DEGF	COND. T-OUT DEGF	PUMP T-IN DEGF	PUMP T-OUT DEGF	E.S. T-IN DEGF	E.S. T-OUT DEGF	EVAP. T-OUT DEGF
1	380.0	144.7	136.6	59.0	48.6	44.8	77.0	137.0	85.0	75.0	183.0	83.0	81.0	81.5	84.0	60.0	40.0	41.0
2	380.0	145.0	135.9	56.0	42.9	35.8	80.0	134.0	89.0	83.0	187.0	86.0	84.0	83.0	86.0	56.0	34.0	16.0
3	380.0	144.3	136.4	57.0	33.9	29.5	76.0	138.0	85.0	95.0	189.0	83.0	81.0	82.0	83.0	61.5	39.5	30.0
4	380.0	142.9	135.2	57.0	58.9	55.0	29.0	137.0	84.0	103.0	225.0	82.0	80.0	81.0	82.0	59.0	37.5	56.0
5	370.0	145.1	136.1	56.0	72.5	67.6	31.0	137.0	86.0	110.0	220.0	84.0	82.0	82.0	85.0	57.0	36.0	67.0
6	370.0	144.0	136.0	55.0	66.7	60.8	34.0	138.0	85.0	113.0	204.5	83.0	81.0	81.0	83.0	60.5	39.0	61.0
7	380.0	142.0	134.1	55.0	68.4	65.1	26.5	133.0	84.0	116.0	235.0	82.0	80.0	80.5	83.0	53.5	34.0	65.0
8	380.0	142.2	134.2	55.0	75.4	71.1	28.0	134.0	84.0	116.0	230.0	82.0	80.5	81.0	83.0	54.0	34.0	70.0
9	380.0	142.2	134.2	55.0	75.0	70.7	28.0	134.0	84.0	118.0	230.0	82.5	80.5	81.0	83.0	55.0	34.0	70.0
10	380.0	142.0	134.4	52.0	37.8	30.7	79.5	137.5	86.0	118.0	187.0	84.0	83.0	81.5	86.0	57.0	35.0	32.0
11	380.0	142.0	134.4	53.0	39.1	32.2	75.0	136.5	85.5	119.5	187.5	83.4	81.0	82.0	83.0	58.0	36.5	29.0

ABBREVIATIONS: HYD.=HYDRONIC, T=TEMPERATURE, EVAP.=EVAPORATOR, EXP.=EXPANDER
COMP.=COMPRESSOR, COND=CONDENSER, E.S.=ENTHALPY SAMPLING.

NOTE: COMP. T-IN VALUES IN THIS TEST ARE INCORRECT DUE TO THERMOCOUPLE MALFUNCTION.

PHASE III BRINE CHILLER SYSTEM

RUN NUMBER:3-12 DATE:11/22/82 VALUE TIMING= 0.0 DEGREES ATDC WET BULB= 53 DEG. F

**** TEST RESULTS - CONTINUED ****

POINT	POWER USED KW	M/G HOURS	EXP. P-IN PSI	EXP. P-OUT PSI	EVAP. P. PSI	COMP. P-IN PSI	COND.* P-IN PSI	COND.* P-OUT PSI	GPUMP P-IN PSI	E.S. P. PSI	FREON** FLOW GPM	E.S. H-IN B/LB	E.S. H-OUT B/LB	EVAP. H-OUT B/LB	COMP. RO-IN LB/CFT	NOTES
1	-3.8	35.84	295.0	160.0	51.0	51.0	162.5	162.5	173.0	13.5	37.0	114.5	111.4	109.4	1.070	H,ON,B
2	4.4	35.92	295.0	166.0	43.0	43.0	171.0	170.0	167.0	13.0	50.0	113.9	110.4	106.4	0.915	F,OF,B
3	3.0	36.06	295.0	160.0	48.0	48.0	165.0	164.0	174.0	13.5	38.0	114.8	111.3	107.7	0.972	F,OF,B
4	0.1	36.18	295.0	158.0	22.0	22.0	163.0	161.0	172.0	11.5	35.5	114.6	111.2	113.5	0.547	F,OF,N
5	1.5	36.30	295.0	160.0	28.0	28.0	166.0	165.0	173.0	12.5	50.0	114.5	111.1	114.9	0.631	F,OF,N,TAL
6	2.8	36.41	295.0	161.0	36.0	36.0	165.0	164.0	174.0	13.5	39.0	113.6	110.6	113.6	0.749	F,OF,N,TAL,OLL
7	0.3	36.54	295.0	160.0	19.0	19.0	165.0	163.0	171.0	11.0	41.0	114.1	110.8	115.1	0.489	F,OF,N
8	1.2	36.60	295.0	160.0	24.0	24.0	164.0	162.0	170.0	12.0	42.0	113.6	110.8	115.6	0.564	F,OF,N
9	1.0	36.68	295.0	160.0	25.0	25.0	165.0	163.0	171.0	12.0	42.0	113.8	110.8	115.6	0.576	F,OF,N
10	2.8	36.89	295.0	160.0	38.0	38.0	165.0	163.5	172.0	14.0	39.5	114.0	110.5	108.6	0.772	F,OF,B
11	3.0	36.95	295.0	160.0	38.0	38.0	165.0	163.5	172.0	14.0	39.5	114.1	110.6	108.2	0.769	F,OF,B

EXPLANATION OF NOTES: EVAPORATOR LOAD- F=FULL, H=HALF
UNLOADER- O=ON, OF=OFF
SUBCOOLER- N=NOT BYPASSED, B=BYPASSED
TAL= TEV AUGMENTATION LINE OPEN
OLL= OIL LET-DOWN LINE OPEN

ABBREVIATIONS: M/G=MOTOR/GENERATOR, EXP.=EXPANDER, P=PRESSURE, EVAP.=EVAPORATOR, COMP.=COMPRESSOR, COND.=CONDENSER
GPUMP=GEROTOR PUMP, E.S.=ENTHALPY SAMPLING, H=ENTHALPY, B=BTU, RO=DENSITY.

* CONDENSER PRESSURE IS TOO HIGH DUE TO CONDENSATION IN PRESSURE GUAGE LINES (CONDENSER IS HIGH ABOVE GROUND).
CLOSEST PRESSURE TO CONDENSING PRESSURE IS EXP. OUT PRESSURE WHICH IS LESS THAN 1 PSI HIGHER.

** FREON FLOW MEASURED BY TURBINE FLOW METER. ACTUAL FLOW IS LESS THAN 90% OF THIS VALUE (SINCE THE FLOW METER READ TOO HIGH).

PHASE III BRINE CHILLER SYSTEM

RUN NUMBER:3-13 DATE:11/30/82 VALVE TIMING= 3.0 DEGREES ATDC WET BULB= 45.5 DEG. F

**** TEST RESULTS ****

POINT	HYD. FLOW GPM	HYD. T-IN DEGF	HYD. T-OUT DEGF	BRINE FLOW GPM	BRINE T-IN DEGF	BRINE T-OUT DEGF	EVAP. T-IN DEGF	EXP. T-IN DEGF	EXP. T-OUT DEGF	COMP. T-IN DEGF	COMP. T-OUT DEGF	COND. T-IN DEGF	COND. T-OUT DEGF	PUMP T-IN DEGF	PUMP T-OUT DEGF	E.S. T-IN DEGF	E.S. T-OUT DEGF	EVAP. T-OUT DEGF
1	400.0	141.2	134.2	53.0	38.1	29.2	71.0	136.0	81.0	87.0	179.0	79.0	75.0	75.0	76.5	58.0	33.0	25.0
2	400.0	141.7	134.6	48.0	40.6	30.8	68.0	137.0	81.0	98.0	178.0	79.0	75.0	75.0	76.0	60.0	36.0	34.0

ABBREVIATIONS: HYD.=HYDRONIC, T=TEMPERATURE, EVAP.=EVAPORATOR, EXP.=EXPANDER
COMP.=COMPRESSOR, COND=CONDENSER, E.S.=ENTHALPY SAMPLING.

NOTE: COMP. T-IN VALUES IN THIS TEST ARE INCORRECT DUE TO THERMOCOUPLE MALFUNCTION.

PHASE III BRINE CHILLER SYSTEM

RUN NUMBER:3-13 DATE:11/30/82 VALVE TIMING= 3.0 DEGREES ATDC WET BULB= 45.5 DEG. F

**** TEST RESULTS - CONTINUED ****

POINT	POWER USED KW	M/G HOURS	EXP. P-IN PSI	EXP. P-OUT PSI	EVAP. P. PSI	COMP. P-IN PSI	COND.* P-IN PSI	COND.* P-OUT PSI	GPUMP P-IN PSI	E.S. P. PSI	FREON** FLOW GPM	E.S. H-IN B/LB	E.S. H-OUT B/LB	EVAP. H-OUT B/LB	COMP. RO-IN LB/CFT	NOTES
1	0.4	39.16	290.0	150.0	40.0	40.0	152.0	153.0	160.0	13.5	39.0	114.2	110.3	107.4	0.857	F,OF,B
2	0.4	39.31	290.0	150.0	40.0	40.0	152.0	153.0	163.0	15.0	37.5	114.4	110.7	108.8	0.836	F,OF,B,TAL,OLL

EXPLANATION OF NOTES: EVAPORATOR LOAD- F=FULL, H=HALF
UNLOADER- O=ON, OF=OFF
SUBCOOLER- N=NOT BYPASSED, B=BYPASSED
TAL= TEV AUGMENTATION LINE OPEN
OLL= OIL LET-DOWN LINE OPEN

ABBREVIATIONS: M/G=MOTOR/GENERATOR, EXP.=EXPANDER, P=PRESSURE, EVAP.=EVAPORATOR, COMP.=COMPRESSOR, COND.=CONDENSER
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* CONDENSER PRESSURE IS TOO HIGH DUE TO CONDENSATION IN PRESSURE GUAGE LINES (CONDENSER IS HIGH ABOVE GROUND).
CLOSEST PRESSURE TO CONDENSING PRESSURE IS EXP. OUT PRESSURE WHICH IS LESS THAN 1 PSI HIGHER.

** FREON FLOW MEASURED BY TURBINE FLOW METER. ACTUAL FLOW IS LESS THAN 90% OF THIS VALUE (SINCE THE FLOW METER READ TOO HIGH).

PHASE III BRINE CHILLER SYSTEM

RUN NUMBER:3-14 DATE:12/2/82 VALVE TIMING= 3.0 DEGREES ATDC WET BULB= 50-48 DEG. F

**** TEST RESULTS ****

POINT	HYD. FLOW GPM	HYD. T-IN DEGF	HYD. T-OUT DEGF	BRINE FLOW GPM	BRINE T-IN DEGF	BRINE T-OUT DEGF	EVAP. T-IN DEGF	EXP. T-IN DEGF	EXP. T-OUT DEGF	COMP. T-IN DEGF	COMP. T-OUT DEGF	COND. T-IN DEGF	COND. T-OUT DEGF	PUMP T-IN DEGF	PUMP T-OUT DEGF	E.S. T-IN DEGF	E.S. T-OUT DEGF	EVAP. T-OUT DEGF
1	405.0	139.9	133.3	60.0	37.0	29.3	75.0	128.0	83.0	73.0	183.0	83.0	78.0	78.0	81.5	26.0	8.0	32.0
2	410.0	139.1	132.7	61.0	38.9	31.1	74.0	129.0	83.0	82.0	182.0	84.0	77.0	77.5	81.5	42.0	24.0	28.0
3	410.0	139.9	132.3	61.0	37.4	29.7	73.0	128.5	82.5	95.5	177.0	85.0	77.0	77.0	79.0	44.0	25.0	27.5
4	410.0	138.6	132.1	61.0	37.1	29.2	71.0	129.0	82.0	95.0	177.0	85.5	76.0	76.5	78.5	47.5	28.0	15.0
5	410.0	138.4	131.9	51.0	40.8	31.3	69.5	131.0	82.0	94.0	174.5	88.5	76.0	76.5	78.0	51.0	31.0	29.0
6	410.0	137.8	131.4	50.0	44.2	34.6	33.5	127.5	82.0	108.0	176.0	84.5	76.0	76.0	78.0	43.0	24.0	39.0
7	410.0	137.3	130.8	50.0	47.9	39.0	29.5	127.0	82.0	111.0	184.0	81.5	76.0	76.0	79.0	35.0	17.0	45.0
8	410.0	137.4	131.0	49.0	54.6	46.9	27.0	127.0	81.0	113.0	199.0	80.5	76.0	75.5	78.5	28.0	7.0	52.0
9	405.0	137.7	131.3	50.0	63.7	57.2	24.0	127.0	81.0	114.0	217.0	80.0	75.0	75.0	78.0	24.5	0.0	61.5
10	410.0	140.0	133.3	50.0	39.1	29.6	70.5	134.0	82.0	110.0	172.0	88.0	75.0	76.0	78.0	54.0	32.0	20.0
11	400.0	142.9	135.8	58.0	35.6	31.7	66.0	138.0	80.0	116.0	180.0	83.0	73.5	74.0	76.0	58.0	33.0	16.0

ABBREVIATIONS: HYD.=HYDRONIC, T=TEMPERATURE, EVAP.=EVAPORATOR, EXP.=EXPANDER
COMP.=COMPRESSOR, COND=CONDENSER, E.S.=ENTHALPY SAMPLING.

NOTE: COMP. T-IN VALUES IN THIS TEST ARE INCORRECT DUE TO THERMOCOUPLE MALFUNCTION.

PHASE III BRINE CHILLER SYSTEM

RUN NUMBER: 3-14 DATE: 12/2/82 VALVE TIMING= 3.0 DEGREES ATDC WET BULB= 50-48 DEG. F

****** TEST RESULTS - CONTINUED ******

POINT	POWER USED KW	M/G HOURS	EXP. P-IN PSI	EXP. P-OUT PSI	EVAP. P. PSI	COMP. P-IN PSI	COND.* P-IN PSI	COND.* P-OUT PSI	GPUMP P-IN PSI	E.S. P. PSI	FREON** FLOW GPM	E.S. H-IN B/LB	E.S. H-OUT B/LB	EVAP. H-OUT B/LB	COMP. RO-IN LB/CFT	NOTES
1	1.8	40.94	294.0	158.0	39.0	39.0	163.0	161.0	160.0	10.0	45.0	109.4	106.7	108.6	0.876	F,OF,B
2	1.8	41.02	291.0	156.0	39.0	39.0	162.0	160.0	167.0	10.0	40.0	111.9	109.1	108.0	0.850	F,OF,B
3	1.8	41.31	290.0	156.0	39.0	39.0	161.0	159.0	167.0	10.0	40.5	112.2	109.2	108.0	0.824	F,OF,B
4	1.9	41.47	289.0	155.0	39.0	39.0	161.0	159.0	168.0	10.3	38.5	112.6	109.7	105.8	0.826	F,OF,B,TAL
5	2.0	41.54	287.0	154.0	39.0	39.0	160.0	158.0	158.0	10.3	37.0	113.2	110.0	108.1	0.827	F,OF,B,TAL,OLL
6	2.1	41.72	288.0	154.0	40.0	40.0	160.0	158.0	167.0	11.0	39.5	112.0	109.0	109.7	0.819	F,OF,N,TAL,OLL
7	1.7	41.81	287.0	153.0	37.0	37.0	159.0	156.0	164.0	10.8	42.5	110.8	108.0	111.2	0.767	F,OF,N,TAL,OLL
8	0.4	41.92	288.0	152.0	32.0	32.0	158.0	155.0	160.0	10.0	43.0	109.7	106.5	112.3	0.688	F,OF,N,TAL
9	-1.1	41.99	290.0	151.0	22.0	22.0	157.0	154.0	155.0	9.0	43.0	109.1	105.5	114.3	0.536	F,OF,N
10	1.4	42.20	287.0	153.0	38.0	38.0	159.0	156.0	167.0	10.0	35.0	113.7	110.3	106.7	0.785	F,OF,B
11	-6.9	42.33	294.0	150.0	43.0	43.0	155.0	154.0	165.0	10.0	31.0	114.3	110.5	106.4	0.852	H,ON,B

EXPLANATION OF NOTES: EVAPORATOR LOAD- F=FULL, H=HALF
UNLOADER- O=ON, OF=OFF
SUBCOOLER- N=NOT BYPASSED, B=BYPASSED
TAL= TEV AUGMENTATION LINE OPEN
OLL= OIL LET-DOWN LINE OPEN

ABBREVIATIONS: M/G=MOTOR/GENERATOR, EXP.=EXPANDER, P=PRESSURE, EVAP.=EVAPORATOR, COMP.=COMPRESSOR, COND.=CONDENSER
GPUMP=GEROTOR PUMP, E.S.=ENTHALPY SAMPLING, H=ENTHALPY, B=BTU, RO=DENSITY.

* CONDENSER PRESSURE IS TOO HIGH DUE TO CONDENSATION IN PRESSURE GUAGE LINES (CONDENSER IS HIGH ABOVE GROUND).
CLOSEST PRESSURE TO CONDENSING PRESSURE IS EXP. OUT PRESSURE WHICH IS LESS THAN 1 PSI HIGHER.

** FREON FLOW MEASURED BY TURBINE FLOW METER. ACTUAL FLOW IS LESS THAN 90% OF THIS VALUE (SINCE THE FLOW METER READ TOO HIGH).

PHASE III BRINE CHILLER SYSTEM

RUN NUMBER:3-15 DATE:12/08/82 VALVE TIMING= 9.0 DEGREES ATDC WET BULB= DEG. F

**** TEST RESULTS ****

POINT	HYD. FLOW GPM	HYD. T-IN DEGF	HYD. T-OUT DEGF	BRINE FLOW GPM	BRINE T-IN DEGF	BRINE T-OUT DEGF	EVAP. T-IN DEGF	EXP. T-IN DEGF	EXP. T-OUT DEGF	COMP. T-IN DEGF	COMP. T-OUT DEGF	COND. T-IN DEGF	COND. T-OUT DEGF	PUMP T-IN DEGF	PUMP T-OUT DEGF	E.S. T-IN DEGF	E.S. T-OUT DEGF	EVAP. T-OUT DEGF
1	405.0	136.1	128.7	36.5	48.0	34.1	61.5	126.0	76.0	35.0	156.5	74.0	68.0	69.0	69.5	69.0	54.0	20.0

ABBREVIATIONS: HYD.=HYDRONIC, T=TEMPERATURE, EVAP.=EVAPORATOR, EXP.=EXPANDER
CONP.=COMPRESSOR, COND=CONDENSER, E.S.=ENTHALPY SAMPLING.

PHASE III BRINE CHILLER SYSTEM

RUN NUMBER:3-15   DATE:12/08/82   VALVE TIMING= 9.0 DEGREES ATDC   WET BULB=   DEG. F

**** TEST RESULTS - CONTINUED ****

POINT	POWER USED KW	M/G HOURS	EXP. P-IN PSI	EXP. P-OUT PSI	EVAP. P. PSI	COMP. P-IN PSI	COND.* P-IN PSI	COND.* P-OUT PSI	GPUMP P-IN PSI	E.S. P. PSI	FREON** FLOW GPM	E.S. H-IN B/LB	E.S. H-OUT B/LB	EVAP. H-OUT B/LB	COMP. RO-IN LB/CFT	NOTES
1	-1.7	43.75	277.0	136.0	42.0	42.0	141.0	138.0	140.0	0.0	37.5	112.9	110.8	106.4	1.010	F,OF,B

EXPLANATION OF NOTES: EVAPORATOR LOAD- F=FULL, H=HALF
UNLOADER- O=ON, OF=OFF
SUBCOOLER- N=NOT BYPASSED, B=BYPASSED
TAL= TEV AUGMENTATION LINE OPEN
OLL= OIL LET-DOWN LINE OPEN

ABBREVIATIONS: M/G=MOTOR/GENERATOR, EXP.=EXPANDER, P=PRESSURE, EVAP.=EVAPORATOR, COMP.=COMPRESSOR, COND.=CONDENSER
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* CONDENSER PRESSURE IS TOO HIGH DUE TO CONDENSATION IN PRESSURE GUAGE LINES (CONDENSER IS HIGH ABOVE GROUND).
CLOSEST PRESSURE TO CONDENSING PRESSURE IS EXP. OUT PRESSURE WHICH IS LESS THAN 1 PSI HIGHER.

** FREON FLOW MEASURED BY TURBINE FLOW METER. ACTUAL FLOW IS LESS THAN 90% OF THIS VALUE (SINCE THE FLOW METER READ TOO HIGH).

PHASE III BRINE CHILLER SYSTEM

RUN NUMBER:3-16 DATE:12/29/82 VALVE TIMING= 6.0 DEGREES ATDC WET BULB= 41 DEG. F

**** TEST RESULTS ****

POINT	HYD. FLOW GPM	HYD. T-IN DEGF	HYD. T-OUT DEGF	BRINE FLOW GPM	BRINE T-IN DEGF	BRINE T-OUT DEGF	EVAP. T-IN DEGF	EXP. T-IN DEGF	EXP. T-OUT DEGF	COMP. T-IN DEGF	COMP. T-OUT DEGF	COND. T-IN DEGF	COND. T-OUT DEGF	PUMP T-IN DEGF	PUMP T-OUT DEGF	E.S. T-IN DEGF	E.S. T-OUT DEGF	EVAP. T-OUT DEGF
1	390.0	139.6	132.4	55.0	40.3	31.8	66.5	134.0	78.5	92.0	165.0	77.0	70.0	71.0	72.0	57.5	32.0	23.0
2	390.0	140.3	133.1	55.0	40.7	32.5	67.0	134.5	79.5	92.0	168.0	77.0	70.5	72.0	74.0	57.0	30.0	21.0
3	380.0	140.6	133.4	55.0	41.1	32.8	64.0	134.5	80.0	98.0	165.0	78.0	72.0	72.0	74.0	55.0	28.0	23.0
4	385.0	141.8	134.1	56.0	41.4	32.9	67.0	134.0	80.0	103.0	167.0	77.0	73.0	72.0	74.0	52.0	24.5	22.0
5	385.0	140.6	133.3	55.0	44.4	35.9	31.0	134.0	80.0	108.0	171.0	78.0	71.0	72.0	74.0	58.5	32.5	35.0
6	385.0	141.2	133.2	55.0	60.2	54.1	24.0	133.0	79.0	112.0	206.0	76.0	72.0	71.0	73.0	28.0	4.0	56.0
7	385.0	141.6	133.7	54.0	58.4	51.8	67.5	136.0	78.0	111.0	155.0	76.0	72.0	71.0	73.0	14.0	0.0	49.0

ABBREVIATIONS: HYD.=HYDRONIC, T=TEMPERATURE, EVAP.=EVAPORATOR, EXP.=EXPANDER
COMP.=COMPRESSOR, COND=CONDENSER, E.S.=ENTHALPY SAMPLING.

NOTE: COMP. T-IN VALUES IN THIS TEST ARE INCORRECT DUE TO THERMOCOUPLE MALFUNCTION.

PHASE III BRINE CHILLER SYSTEM

RUN NUMBER:3-16 DATE:12/29/82 VALVE TIMING= 6.0 DEGREES ATDC WET BULB= 41 DEG. F

**** TEST RESULTS - CONTINUED ****

POINT	POWER USED KW	M/G HOURS	EXP. P-IN PSI	EXP. P-OUT PSI	EVAP. P. PSI	COMP. P-IN PSI	COND.* P-IN PSI	COND.* P-OUT PSI	GPUMP P-IN PSI	F.S. P. PSI	FREON** FLOW GPM	E.S. H-IN B/LB	E.S. H-OUT B/LB	EVAP. H-OUT B/LB	COMP. RO-IN LB/CFT	NOTES
1	-0.5	44.29	286.0	146.0	40.0	40.0	151.0	148.0	160.0	14.0	50.0	114.0	110.0	107.1	0.847	F,OF,B
2	-0.5	44.37	290.0	147.0	40.0	40.0	151.0	149.0	162.0	14.0	50.0	114.0	109.8	106.9	0.838	F,OF,B
3	-0.2	44.45	290.0	147.0	40.0	40.0	151.0	150.0	161.0	15.0	37.5	113.6	109.4	107.1	0.836	F,OF,B
4	-0.8	44.64	293.0	146.0	40.0	40.0	150.0	149.0	161.0	15.0	50.0	113.1	109.0	106.9	0.821	F,OF,B
5	-0.7	44.81	293.0	148.0	38.0	38.0	151.0	150.0	161.0	14.5	41.2	114.2	110.1	109.2	0.783	F,OF,N,TAL,OLL
6	-3.3	45.05	294.0	144.0	27.0	27.0	149.0	147.0	158.0	13.0	50.0	109.5	105.8	113.2	0.613	F,OF,N
7	-8.2	45.17	295.0	144.0	60.0	60.0	150.0	148.0	157.0	19.0	50.0	106.9	104.8	110.2	1.131	H,O,B

EXPLANATION OF NOTES: EVAPORATOR LOAD- F=FULL, H=HALF
UNLOADER- O=ON, OF=OFF
SUBCOOLER- N=NOT BYPASSED, B=BYPASSED
TAL= TEV AUGMENTATION LINE OPEN
OLL= OIL LET-DOWN LINE OPEN

ABBREVIATIONS: M/G=MOTOR/GENERATOR, EXP.=EXPANDER, P=PRESSURE, EVAP.=EVAPORATOR, COMP.=COMPRESSOR, COND.=CONDENSER
GPUMP=GEROTOR PUMP, E.S.=ENTHALPY SAMPLING, H=ENTHALPY, B=BTU, RO=DENSITY.

* CONDENSER PRESSURE IS TOO HIGH DUE TO CONDENSATION IN PRESSURE GUAGE LINES (CONDENSER IS HIGH ABOVE GROUND).
CLOSEST PRESSURE TO CONDENSING PRESSURE IS EXP. OUT PRESSURE WHICH IS LESS THAN 1 PSI HIGHER.

** FREON FLOW MEASURED BY TURBINE FLOW METER. ACTUAL FLOW IS LESS THAN 90% OF THIS VALUE (SINCE THE FLOW METER READ TOO HIGH).

PHASE III BRINE CHILLER SYSTEM

RUN NUMBER:3-17 DATE:12/30/82 VALVE TIMING= 6.0 DEGREES ATDC WET BULB= 31 DEG. F

**** TEST RESULTS ****

POINT	HYD. FLOW GPM	HYD. T-IN DEGF	HYD. T-OUT DEGF	BRINE FLOW GPM	BRINE T-IN DEGF	BRINE T-OUT DEGF	EVAP. T-IN DEGF	EXP. T-IN DEGF	EXP. T-OUT DEGF	COMP. T-IN DEGF	COMP. T-OUT DEGF	COND. T-IN DEGF	COND. T-OUT DEGF	PUMP T-IN DEGF	PUMP T-OUT DEGF	E.S. T-IN DEGF	E.S. T-OUT DEGF	EVAP. T-OUT DEGF
1	365.0	139.5	131.7	56.0	39.4	30.2	58.0	128.0	75.0	33.0	160.0	73.5	67.0	67.0	68.5	38.5	30.5	17.0
2	365.0	139.6	131.6	56.0	39.3	30.1	58.0	129.0	74.0	43.0	158.0	73.5	66.5	66.5	67.5	49.5	23.0	17.0
3	360.0	138.8	131.0	55.0	58.4	52.0	21.0	127.0	72.5	33.0	198.5	70.5	65.0	65.0	66.0	24.5	0.0	55.0
4	335.0	140.0	132.0	54.0	62.7	55.3	25.0	129.5	73.5	43.5	191.0	73.0	65.5	66.0	67.0	48.0	17.5	57.5
5	352.0	139.6	131.6	54.0	53.0	44.7	28.0	130.0	73.5	44.5	176.0	75.5	65.5	66.0	67.0	54.0	25.0	48.0
6	354.0	139.1	131.0	55.0	53.6	47.1	21.0	128.5	73.0	42.0	201.0	72.0	65.0	65.5	66.0	38.0	11.0	51.0
7	355.0	138.6	130.6	55.0	40.0	31.0	61.0	128.0	74.0	27.0	140.0	72.5	66.0	66.0	67.0	49.5	19.0	20.0
8	352.0	140.8	132.6	55.0	47.2	40.9	60.0	133.0	72.0	45.5	156.0	71.0	64.0	64.0	66.0	63.0	30.5	38.5

ABBREVIATIONS: HYD.=HYDRONIC, T=TEMPERATURE, EVAP.=EVAPORATOR, EXP.=EXPANDER
COMP.=COMPRESSOR, COND=CONDENSER, E.S.=ENTHALPY SAMPLING.

PHASE III BRINE CHILLER SYSTEM

RUN NUMBER:3-17 DATE:12/30/82 VALVE TIMING= 6.0 DEGREES ATDC WET BULB= 31 DEG. F

**** TEST RESULTS - CONTINUED ****

POINT	POWER USED KW	M/G HOURS	EXP. P-IN PSI	EXP. P-OUT PSI	EVAP. P. PSI	COMP. P-IN PSI	COND.* P-IN PSI	COND.* P-OUT PSI	GPUMP P-IN PSI	E.S. P. PSI	FREON** FLOW GPM	E.S. H-IN B/LB	E.S. H-OUT B/LB	EVAP. H-OUT B/LB	COMP. RO-IN LB/CFT	NOTES
1	-3.0	45.86	288.0	134.0	38.0	38.0	140.0	136.0	146.0	35.5	37.5	109.9	108.6	106.0	0.938	F,OF,B
2	-3.0	46.00	285.0	133.0	38.0	38.0	139.0	136.0	146.0	36.5	40.5	111.6	107.3	106.0	0.914	F,OF,B
3	-5.0	46.13	286.0	130.0	24.0	24.0	136.0	134.0	144.0	21.5	38.5	108.5	104.7	113.2	0.675	F,OF,N
4	-4.4	46.42	288.0	132.0	30.0	30.0	138.0	135.0	146.0	29.0	38.0	111.8	106.9	113.3	0.766	F,OF,N,TEV
5	-3.5	46.56	285.0	133.0	36.0	36.0	138.0	136.0	146.0	35.5	38.0	112.4	107.7	111.4	0.873	F,OF,N,TEV,OLL
6	-5.2	46.66	285.0	130.0	24.0	24.0	136.0	134.0	145.0	22.0	37.5	110.6	106.3	112.6	0.657	F,OF,N
7	-2.8	46.84	283.0	133.0	39.0	39.0	138.0	136.0	146.0	39.0	37.5	111.4	108.1	106.6	0.973	F,OF,B
8	-10.7	46.99	287.0	128.0	42.0	42.0	134.0	131.0	144.0	51.0	35.0	113.1	107.6	109.5	0.982	H,O,B

EXPLANATION OF NOTES: EVAPORATOR LOAD- F=FULL, H=HALF
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TAL= TEV AUGMENTATION LINE OPEN
OLL= OIL LET-DOWN LINE OPEN

ABBREVIATIONS: M/G=MOTOR/GENERATOR, EXP.=EXPANDER, P=PRESSURE, EVAP.=EVAPORATOR, COMP.=COMPRESSOR, COND.=CONDENSER
GPUMP=GEROTOR PUMP, E.S.=ENTHALPY SAMPLING, H=ENTHALPY, B=BTU, RO=DENSITY.

* CONDENSER PRESSURE IS TOO HIGH DUE TO CONDENSATION IN PRESSURE GUAGE LINES (CONDENSER IS HIGH ABOVE GROUND).
CLOSEST PRESSURE TO CONDENSING PRESSURE IS EXP. OUT PRESSURE WHICH IS LESS THAN 1 PSI HIGHER.

** FREON FLOW MEASURED BY TURBINE FLOW METER. ACTUAL FLOW IS LESS THAN 90% OF THIS VALUE (SINCE THE FLOW METER READ TOO HIGH).

PHASE III BRINE CHILLER SYSTEM

RUN NUMBER:3-18 DATE:03/07/83 VALVE TIMING= 6.0 DEGREES ATDC WET BULB= 37 DEG. F

**** TEST RESULTS ****

POINT	HYD. FLOW GPM	HYD. T-IN DEGF	HYD. T-OUT DEGF	BRINE FLOW GPM	BRINE T-IN DEGF	BRINE T-OUT DEGF	EVAP. T-IN DEGF	EXP. T-IN DEGF	EXP. T-OUT DEGF	COMP. T-IN DEGF	COMP. T-OUT DEGF	COND. T-IN DEGF	COND. T-OUT DEGF	PUMP T-IN DEGF	PUMP T-OUT DEGF	E.S. T-IN DEGF	E.S. T-OUT DEGF	EVAP. T-OUT DEGF
1	385.0	141.4	133.9	51.0	47.1	37.2	64.0	136.0	78.0	35.5	155.0	84.0	70.5	70.5	71.0	68.0	39.0	18.0
2	375.0	145.0	137.2	49.5	28.6	22.1	44.0	139.0	77.0	32.5	184.0	81.0	70.0	70.0	71.5	65.5	36.0	2.0
3	365.0	144.4	136.3	54.0	4.7	1.2	23.0	138.0	77.0	28.0	206.0	80.3	70.0	70.0	71.0	62.0	31.5	-19.0
4	360.0	144.9	135.7	50.5	11.9	7.0	55.0	136.0	77.0	30.5	215.0	78.5	71.0	71.0	71.5	60.5	29.5	-11.5
5	365.0	144.0	135.9	52.0	16.9	11.2	61.0	138.0	77.0	33.0	215.0	79.5	70.0	70.0	71.0	62.0	30.5	6.5
6	360.0	140.4	132.6	52.0	53.7	45.5	63.5	131.0	77.0	32.0	172.0	78.0	71.0	71.0	72.0	59.0	29.0	13.5
7	355.0	140.2	132.4	52.0	56.3	48.0	61.5	130.0	77.5	25.5	168.0	77.5	71.5	71.5	72.0	58.0	28.0	14.0

ABBREVIATIONS: HYD.=HYDRONIC, T=TEMPERATURE, EVAP.=EVAPORATOR, EXP.=EXPANDER
COMP.=COMPRESSOR,COND=CONDENSER, E.S.=ENTHALPY SAMPLING.

PHASE III BRINE CHILLER SYSTEM

RUN NUMBER:3-18 DATE:03/07/83 VALVE TIMING= 6.0 DEGREES ATDC WET BULB= 37 DEG. F

**** TEST RESULTS - CONTINUED ****

POINT	POWER USED KW	M/G HOURS	EXP. P-IN PSI	EXP. P-OUT PSI	EVAP. P. PSI	COMP. P-IN PSI	COND.* P-IN PSI	COND.* P-OUT PSI	GPUMP P-IN PSI	E.S. P. PSI	FREON** FLOW GPM	E.S. H-IN B/LB	E.S. H-OUT B/LB	EVAP. H-OUT B/LB	COMP. RO-IN LB/CFT	NOTES
1	-1.9	49.85	287.0	141.0	38.0	38.0	148.0	145.0	155.0	38.0	35.0	114.6	109.8	106.0	0.932	F,OF,B
2	-3.8	50.00	293.0	140.0	25.0	25.0	146.0	143.0	153.0	26.0	47.5	114.8	110.0	104.6	0.649	F,OF,B
3	-7.0	50.19	297.0	140.0	14.0	14.0	147.0	145.0	157.0	12.0	32.5	114.9	110.0	103.1	0.500	F,OF,B
4	-8.3	50.27	297.0	141.0	9.0	9.0	148.0	145.0	155.0	16.0	33.0	114.5	109.6	102.2	0.408	F,OF,B
5	-7.3	50.32	296.0	139.0	18.0	18.0	145.0	142.0	153.0	18.0	34.0	114.6	109.6	103.7	0.566	F,OF,B
6	-2.4	50.53	289.0	140.0	35.0	35.0	146.0	144.0	155.0	30.0	36.0	113.5	108.6	105.6	0.833	F,OF,B
7	-2.2	50.57	289.0	141.0	34.0	34.0	147.0	144.0	155.0	30.0	36.0	113.2	108.3	105.5	0.879	F,OF,B

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* CONDENSER PRESSURE IS TOO HIGH DUE TO CONDENSATION IN PRESSURE GUAGE LINES (CONDENSER IS HIGH ABOVE GROUND).
CLOSEST PRESSURE TO CONDENSING PRESSURE IS EXP. OUT PRESSURE WHICH IS LESS THAN 1 PSI HIGHER.

** FREON FLOW MEASURED BY TURBINE FLOW METER. ACTUAL FLOW IS LESS THAN 90% OF THIS VALUE (SINCE THE FLOW METER READ TOO HIGH).

PHASE III BRINE CHILLER SYSTEM

RUN NUMBER:3-19 DATE:03/08/83 VALVE TIMING= 6.0 DEGREES ATDC WET BULB= 36-35 DEG. F

**** TEST RESULTS ****

POINT	HYD. FLOW GPM	HYD. T-IN DEGF	HYD. T-OUT DEGF	BRINE FLOW GPM	BRINE T-IN DEGF	BRINE T-OUT DEGF	EVAP. T-IN DEGF	EXP. T-IN DEGF	EXP. T-OUT DEGF	COMP. T-IN DEGF	COMP. T-OUT DEGF	COND. T-IN DEGF	COND. T-OUT DEGF	PUMP T-IN DEGF	PUMP T-OUT DEGF	E.S. T-IN DEGF	E.S. T-OUT DEGF	EVAP. T-OUT DEGF
1	365.0	140.0	132.0	51.0	56.4	48.3	62.5	129.0	77.5	34.0	166.0	77.0	71.0	71.0	72.0	54.5	25.0	14.0
2	360.0	139.9	132.2	52.0	56.0	48.0	65.0	129.0	77.5	36.0	165.0	77.0	71.5	71.5	72.0	55.0	24.5	15.0
3	360.0	138.5	131.1	50.0	49.0	40.2	49.0	135.0	77.5	43.5	164.5	85.5	67.0	67.0	68.0	68.0	40.0	11.5
4	360.0	138.0	130.5	52.0	49.7	40.8	63.0	132.0	76.0	42.5	163.0	82.0	69.0	69.0	70.0	65.0	35.5	17.0
5	360.0	138.2	130.8	51.5	49.3	40.5	58.5	133.0	76.0	43.5	164.5	83.5	68.0	68.0	69.0	67.5	38.5	16.5
6	355.0	132.1	125.0	52.0	45.1	36.8	60.0	122.0	74.0	41.0	167.0	76.0	68.0	68.0	68.5	51.0	22.0	13.0
7	345.0	131.4	124.4	52.0	45.0	36.5	58.0	121.0	74.0	39.0	165.0	74.0	69.0	69.0	69.0	44.5	15.0	13.0
8	355.0	131.0	124.0	51.0	44.5	36.0	62.5	121.0	74.5	37.0	164.0	74.0	69.0	69.0	69.0	39.0	12.5	13.0
9	355.0	124.5	118.0	52.0	41.1	32.6	54.0	115.5	73.5	12.0	156.0	71.0	67.0	67.0	67.0	12.0	11.5	12.5
10	350.0	124.5	118.0	51.0	40.6	32.2	56.0	115.0	72.0	10.5	157.0	70.0	66.0	66.0	66.5	12.0	11.0	13.0

ABBREVIATIONS: HYD.=HYDRONIC, T=TEMPERATURE, EVAP.=EVAPORATOR, EXP.=EXPANDER
COMP.=COMPRESSOR, COND=CONDENSER, E.S.=ENTHALPY SAMPLING.

PHASE III BRINE CHILLER SYSTEM

RUN NUMBER:3-19 DATE:03/08/83 VALVE TIMING= 6.0 DEGREES ATDC WET BULB= 36-35 DEG. F

**** TEST RESULTS - CONTINUED ****

POINT	POWER USED KW	M/G HOURS	EXP. P-IN PSI	EXP. P-OUT PSI	EVAP. P. PSI	COMP. P-IN PSI	COND.* P-IN PSI	COND.* P-OUT PSI	GPUMP P-IN PSI	E.S. P. PSI	FREON** FLOW GPM	E.S. H-IN B/LB	E.S. H-OUT B/LB	EVAP. H-OUT B/LB	COMP. RO-IN LB/CFT	NOTES
1	-2.0	50.64	289.0	141.0	34.0	34.0	147.0	144.0	155.0	33.0	39.0	112.6	107.8	105.5	0.840	F,OF,B
2	-2.1	50.67	289.0	141.0	34.0	34.0	146.0	144.0	155.0	34.0	39.0	112.7	107.7	105.5	0.836	F,OF,B
3	-1.0	51.10	275.0	138.0	34.0	34.0	142.0	139.0	147.0	34.0	50.0	114.8	110.2	105.5	0.820	F,OF,B
4	-0.6	51.25	275.0	138.0	34.0	34.0	143.0	140.0	154.0	37.0	32.0	114.2	109.3	105.5	0.823	F,OF,B
5	-0.7	51.29	275.0	138.0	35.0	35.0	144.0	140.0	154.0	33.0	32.0	114.7	110.0	105.6	0.857	F,OF,B
6	0.8	51.57	263.0	138.0	33.0	33.0	139.0	136.0	148.0	32.0	34.0	112.1	107.4	105.4	0.826	F,OF,B
7	1.0	51.61	261.0	138.0	32.0	32.0	140.0	137.0	148.0	33.0	35.0	110.9	106.2	105.3	0.811	F,OF,B
8	1.0	51.65	260.0	138.0	34.0	34.0	140.0	137.0	148.0	33.0	35.0	110.1	105.8	105.5	0.834	F,OF,B
9	3.0	51.94	241.0	138.0	33.0	33.0	135.0	132.0	144.0	33.0	35.0	105.8	105.7	105.4	0.859	F,OF,B
10	2.9	51.98	241.0	138.0	32.0	32.0	135.0	132.0	142.0	33.0	35.0	105.8	105.6	105.3	0.877	F,OF,B

EXPLANATION OF NOTES: EVAPORATOR LOAD- F=FULL, H=HALF
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CLOSEST PRESSURE TO CONDENSING PRESSURE IS EXP. OUT PRESSURE WHICH IS LESS THAN 1 PSI HIGHER.

** FREON FLOW MEASURED BY TURBINE FLOW METER. ACTUAL FLOW IS LESS THAN 90% OF THIS VALUE (SINCE THE FLOW METER READ TOO HIGH).

PHASE III BRINE CHILLER SYSTEM

RUN NUMBER:3-21 DATE:03/15/83 VALVE TIMING= 6.0 DEGREES ATDC WET BULB= 46 DEG. F

**** TEST RESULTS ****

POINT	HYD. FLOW GPM	HYD. T-IN DEGF	HYD. T-OUT DEGF	BRINE FLOW GPM	BRINE T-IN DEGF	BRINE T-OUT DEGF	EVAP. T-IN DEGF	EXP. T-IN DEGF	EXP. T-OUT DEGF	COMP. T-IN DEGF	COMP. T-OUT DEGF	COND. T-IN DEGF	COND. T-OUT DEGF	PUMP T-IN DEGF	PUMP T-OUT DEGF	E.S. T-IN DEGF	E.S. T-OUT DEGF	EVAP. T-OUT DEGF
1	425.0	137.9	131.7	33.0	54.0	38.7	68.0	131.0	83.0	36.0	161.0	86.0	77.0	77.0	77.0	66.0	40.0	22.0
2	430.0	138.0	131.9	33.0	54.5	39.5	69.0	132.0	82.0	37.0	163.0	87.0	76.0	76.0	77.5	67.0	41.0	22.0
3	425.0	138.3	132.1	49.0	48.7	38.3	67.0	133.0	83.0	37.0	164.0	88.0	76.0	76.0	77.5	68.0	44.0	19.0
4	425.0	138.6	132.3	49.0	47.6	37.7	70.0	134.0	82.0	36.0	166.0	89.0	76.0	76.0	77.0	68.0	44.0	20.0
5	420.0	139.8	133.3	49.0	42.6	34.0	63.0	135.0	82.0	31.0	173.0	90.0	76.0	76.0	76.0	69.0	44.0	12.0
6	440.0	138.6	132.1	47.0	47.7	39.7	57.0	133.0	82.0	23.0	178.0	87.0	76.0	76.0	77.0	65.0	39.0	10.0
7	420.0	138.8	132.2	50.0	47.3	39.7	53.0	133.0	82.0	22.0	178.0	87.0	76.0	76.0	77.0	66.0	40.0	9.0
8	430.0	140.4	133.6	48.0	34.7	28.4	52.0	136.0	82.0	4.0	182.0	89.0	77.0	77.0	77.0	65.0	40.0	3.0
9	430.0	138.2	131.7	48.0	35.8	29.3	58.0	132.0	82.0	2.0	184.0	86.0	77.0	77.0	77.0	62.0	37.0	2.0
10	425.0	139.8	132.3	48.0	20.4	16.3	43.0	132.0	84.0	-9.0	204.0	87.0	78.0	78.0	78.0	61.0	37.0	-9.0
11	425.0	138.7	131.9	48.0	18.3	13.6	44.0	133.0	82.0	-8.0	205.0	86.0	76.0	76.0	77.0	61.0	37.0	-12.0
12	425.0	140.2	133.3	47.0	13.4	9.7	14.0	135.0	81.0	-13.0	210.0	87.0	75.0	75.0	76.0	64.0	40.0	-15.0

ABBREVIATIONS: HYD.=HYDRONIC, T=TEMPERATURE, EVAP.=EVAPORATOR, EXP.=EXPANDER
COMP.=COMPRESSOR, COND=CONDENSER, E.S.=ENTHALPY SAMPLING.

PHASE III BRINE CHILLER SYSTEM

RUN NUMBER:3-21   DATE:03/15/83   VALVE TIMING= 6.0 DEGREES ATDC   WET BULB=   46 DEG. F

**** TEST RESULTS - CONTINUED ****

POINT	POWER USED KW	M/G HOURS	EXP. P-IN PSI	EXP. P-OUT PSI	EVAP. P. PSI	COMP. P-IN PSI	COND.* P-IN PSI	COND.* P-OUT PSI	GPUMP P-IN PSI	E.S. P. PSI	FREON** FLOW GPM	E.S. H-IN B/LB	E.S. H-OUT B/LB	EVAP. H-OUT B/LB	COMP. RO-IN LB/CFT	NOTES
1	1.5	55.52	287.0	152.0	43.5	43.5	156.0	154.0	165.0	50.0	39.5	113.7	109.3	106.7	1.027	F,OF,B,TAL,OLL
2	1.5	55.62	287.0	152.0	44.0	44.0	156.0	154.0	165.0	52.0	39.0	113.7	109.3	106.5	1.043	F,OF,B,TAL,OLL
3	1.6	55.74	286.0	152.0	41.0	41.0	158.0	155.0	167.0	49.0	38.5	114.0	110.0	106.3	0.985	F,OF,B,TAL,OLL
4	1.5	55.78	286.0	152.0	39.5	39.5	157.0	155.0	166.0	48.0	38.0	114.1	110.1	106.6	0.950	F,OF,B,TAL,OLL
5	0.8	55.92	286.0	151.0	33.0	33.0	156.0	154.0	166.0	42.0	35.0	114.6	110.4	105.7	0.847	F,OF,B,TAL,OLL
6	0.4	56.14	286.0	151.0	30.5	30.5	157.0	154.0	165.0	40.0	36.0	114.0	109.7	105.6	0.807	F,OF,B,TAL,OLL
7	0.3	56.18	286.0	152.0	31.0	31.0	156.0	154.0	165.0	40.0	36.0	114.2	109.9	105.3	0.828	F,OF,B,TAL,OLL
8	-0.6	56.33	286.0	151.0	23.5	23.5	156.0	154.0	164.0	35.5	44.5	114.3	110.2	105.0	0.708	F,OF,B,TAL,OLL
9	0.0	56.41	285.0	152.0	24.5	24.5	158.0	155.0	166.0	35.0	38.5	113.8	109.7	104.8	0.731	F,OF,B,TAL,OLL
10	-1.1	56.59	285.0	154.0	16.0	16.0	160.0	156.0	165.0	30.0	47.0	113.9	110.0	103.6	0.589	F,OF,B,TAL,OLL
11	-2.1	56.63	285.0	152.0	14.0	14.0	156.0	154.0	166.0	28.0	36.0	114.0	110.1	103.2	0.548	F,OF,B,TAL,OLL
12	-3.0	56.69	285.0	150.0	12.5	12.5	157.0	154.0	166.0	27.0	35.0	114.5	110.6	102.9	0.514	F,OF,B,TAL,OLL

EXPLANATION OF NOTES: EVAPORATOR LOAD- F=FULL, H=HALF
UNLOADER- O=ON, OF=OFF
SUBCOOLER- N=NOT BYPASSED, B=BYPASSED
TAL= TEV AUGMENTATION LINE OPEN
OLL= OIL LET-DOWN LINE OPEN

ABBREVIATIONS: M/G=MOTOR/GENERATOR, EXP.=EXPANDER, P=PRESSURE, EVAP.=EVAPORATOR, COMP.=COMPRESSOR, COND.=CONDENSER
GPUMP=GEROTOR PUMP, E.S.=ENTHALPY SAMPLING, H=ENTHALPY, B=BTU, RO=DENSITY.

* CONDENSER PRESSURE IS TOO HIGH DUE TO CONDENSATION IN PRESSURE GUAGE LINES (CONDENSER IS HIGH ABOVE GROUND).
CLOSEST PRESSURE TO CONDENSING PRESSURE IS EXP. OUT PRESSURE WHICH IS LESS THAN 1 PSI HIGHER.

** FREON FLOW MEASURED BY TURBINE FLOW METER. ACTUAL FLOW IS LESS THAN 90% OF THIS VALUE (SINCE THE FLOW METER READ TOO HIGH).

PHASE III BRINE CHILLER SYSTEM

RUN NUMBER:3-23 DATE:03/16/83 VALVE TIMING= 6.0 DEGREES ATDC WET BULB= 45-43 DEG. F

**** TEST RESULTS ****

POINT	HYD. FLOW GPM	HYD. T-IN DEGF	HYD. T-OUT DEGF	BRINE FLOW GPM	BRINE T-IN DEGF	BRINE T-OUT DEGF	EVAP. T-IN DEGF	EXP. T-IN DEGF	EXP. T-OUT DEGF	COMP. T-IN DEGF	COMP. T-OUT DEGF	COND. T-IN DEGF	COND. T-OUT DEGF	PUMP T-IN DEGF	PUMP T-OUT DEGF	E.S. T-IN DEGF	E.S. T-OUT DEGF	EVAP. T-OUT DEGF
1	435.0	137.6	131.5	33.0	57.5	43.2	64.0	128.0	82.0	37.0	167.0	81.0	76.0	76.0	77.0	72.0	51.0	30.0
2	435.0	137.6	131.6	33.0	58.6	43.8	71.0	128.0	82.0	37.0	166.0	80.0	75.0	75.0	77.0	47.0	26.0	22.0
3	435.0	137.9	132.0	33.0	60.7	47.2	61.0	131.0	83.0	34.0	164.0	85.0	76.0	76.0	77.0	62.0	38.0	21.0
4	435.0	139.3	132.9	33.0	59.4	44.5	62.0	133.0	87.0	36.0	171.0	94.0	78.0	78.0	79.0	69.0	46.0	22.0
5	435.0	138.6	132.3	33.0	58.8	44.2	74.0	131.0	88.0	38.0	173.0	93.5	80.5	80.5	82.0	67.0	45.0	22.0
6	435.0	137.8	131.5	33.0	58.8	45.1	84.0	128.0	101.0	42.5	187.0	105.5	95.0	95.0	94.0	60.0	42.0	23.0
7	435.0	138.3	131.9	33.0	59.8	45.3	89.0	130.5	104.0	46.0	191.0	113.0	97.0	97.0	97.0	67.5	54.0	26.0
8	435.0	138.4	132.1	33.0	58.1	43.4	79.0	132.0	93.0	45.0	182.0	103.0	85.0	85.0	87.0	69.0	52.0	24.0
9	435.0	138.1	131.8	33.0	57.3	42.6	75.0	131.0	92.0	45.0	182.0	100.0	86.0	86.0	87.0	67.0	48.0	23.0
10	435.0	137.0	131.0	33.0	55.9	41.5	71.0	128.0	81.0	41.0	165.0	83.0	75.0	75.0	77.0	60.5	35.5	22.0

ABBREVIATIONS: HYD.=HYDRONIC, T=TEMPERATURE, EVAP.=EVAPORATOR, EXP.=EXPANDER
COMP.=COMPRESSOR, COND=CONDENSER, E.S.=ENTHALPY SAMPLING.

PHASE III BRINE CHILLER SYSTEM

RUN NUMBER: 3-23 DATE: 03/16/83 VALVE TIMING= 6.0 DEGREES ATDC WET BULB= 45-43 DEG. F

**** TEST RESULTS - CONTINUED ****

POINT	POWER USED KW	M/G HOURS	EXP. P-IN PSI	EXP. P-OUT PSI	EVAP. P. PSI	COMP. P-IN PSI	COND.* P-IN PSI	COND.* P-OUT PSI	GPUMP P-IN PSI	E.S. P. PSI	FREON** FLOW GPM	E.S. H-IN B/LB	E.S. H-OUT B/LB	EVAP. H-OUT B/LB	COMP. RO-IN LB/CFT	NOTES
1	1.1	58.36	285.0	150.0	40.0	40.0	155.0	152.0	162.0	47.0	42.0	114.9	111.4	108.2	0.966	F,OF,B
2	1.0	58.41	285.0	150.0	39.0	39.0	155.0	152.0	161.0	47.0	43.0	110.7	107.2	107.0	0.947	F,OF,B
3	1.8	58.68	286.0	153.0	42.0	42.0	157.0	154.0	165.0	52.0	38.5	112.9	108.8	106.6	1.013	F,OF,B
4	4.5	58.83	286.0	163.0	44.0	44.0	169.0	166.0	176.0	52.0	38.5	114.1	110.2	106.5	1.046	F,OF,B
5	4.6	58.88	286.0	163.0	43.3	43.3	169.0	166.0	176.0	52.0	39.0	113.7	110.0	106.7	1.021	F,OF,B
6	15.2	59.05	285.0	199.0	45.0	45.0	205.0	202.0	211.0	56.0	43.0	112.3	109.3	106.7	1.047	F,OF,B
7	17.6	59.20	285.0	205.0	47.0	47.0	211.0	208.0	217.0	58.0	41.5	113.5	111.2	107.1	1.075	F,OF,B
8	9.2	59.32	285.0	178.0	46.5	46.5	184.0	181.0	191.0	56.0	40.0	113.9	111.0	106.8	1.059	F,OF,B
9	8.4	59.40	285.5	176.0	45.0	45.0	182.0	179.0	189.0	54.0	40.0	113.6	110.4	106.7	1.040	F,OF,B
10	1.0	59.56	286.0	150.0	43.0	43.0	155.0	152.0	163.0	50.0	38.5	112.7	108.5	106.7	1.031	F,OF,B

EXPLANATION OF NOTES: EVAPORATOR LOAD- F=FULL, H=HALF
UNLOADER- O=ON, OF=OFF
SUBCOOLER- N=NOT BYPASSED, B=BYPASSED
TAL= TEV AUGMENTATION LINE OPEN
OLL= OIL LET-DOWN LINE OPEN

ABBREVIATIONS: M/G=MOTOR/GENERATOR, EXP.=EXPANDER, P=PRESSURE, EVAP.=EVAPORATOR, COMP.=COMPRESSOR, COND.=CONDENSER
GPUMP=GEROTOR PUMP, E.S.=ENTHALPY SAMPLING, H=ENTHALPY, B=BTU, RO=DENSITY.

* CONDENSER PRESSURE IS TOO HIGH DUE TO CONDENSATION IN PRESSURE GUAGE LINES (CONDENSER IS HIGH ABOVE GROUND).
CLOSEST PRESSURE TO CONDENSING PRESSURE IS EXP. OUT PRESSURE WHICH IS LESS THAN 1 PSI HIGHER.

** FREON FLOW MEASURED BY TURBINE FLOW METER. ACTUAL FLOW IS LESS THAN 90% OF THIS VALUE (SINCE THE FLOW METER READ TOO HIGH).

PHASE III BRINE CHILLER SYSTEM

RUN NUMBER:3-24 DATE:03/17/83 VALVE TIMING= 6.0 DEGREES ATDC WET BULB= 40 DEG. F

**** TEST RESULTS ****

POINT	HYD. FLOW GPM	HYD. T-IN DEGF	HYD. T-OUT DEGF	BRINE FLOW GPM	BRINE T-IN DEGF	BRINE T-OUT DEGF	EVAP. T-IN DEGF	EXP. T-IN DEGF	EXP. T-OUT DEGF	COMP. T-IN DEGF	COMP. T-OUT DEGF	COND. T-IN DEGF	COND. T-OUT DEGF	PUMP T-IN DEGF	PUMP T-OUT DEGF	E.S. T-IN DEGF	E.S. T-OUT DEGF	EVAP. T-OUT DEGF
1	435.0	137.2	131.2	33.0	55.2	40.5	67.0	128.0	79.0	33.0	157.0	77.0	72.0	72.0	73.5	66.0	47.0	24.0

ABBREVIATIONS: HYD.=HYDRONIC, T=TEMPERATURE, EVAP.=EVAPORATOR, EXP.=EXPANDER
COMP.=COMPRESSOR, COND=CONDENSER, E.S.=ENTHALPY SAMPLING.

PHASE III BRINE CHILLER SYSTEM

RUN NUMBER:3-24 DATE:03/17/83 VALVE TIMING= 6.0 DEGREES ATDC WET BULB= 40 DEG. F

**** TEST RESULTS - CONTINUED ****

POINT	POWER USED KW	M/G HOURS	EXP. P-IN PSI	EXP. P-OUT PSI	EVAP. P. PSI	COMP. P-IN PSI	COND.* P-IN PSI	COND.* P-OUT PSI	GPUMP P-IN PSI	E.S. P. PSI	FREON** FLOW GPM	E.S. H-IN B/LB	E.S. H-OUT B/LB	EVAP. H-OUT B/LB	COMP. RO-IN LB/CFT	NOTES
1	-0.7	62.53	286.0	142.0	41.0	41.0	148.0	145.0	155.0	51.5	42.0	113.6	110.4	107.2	0.966	F,OF,B

EXPLANATION OF NOTES: EVAPORATOR LOAD- F=FULL, H=HALF
UNLOADER- O=ON, OF=OFF
SUBCOOLER- N=NOT BYPASSED, B=BYPASSED
TAL= TEV AUGMENTATION LINE OPEN
OLL= OIL LET-DOWN LINE OPEN

ABBREVIATIONS: M/G=MOTOR/GENERATOR, EXP.=EXPANDER, P=PRESSURE, EVAP.=EVAPORATOR, COMP.=COMPRESSOR, COND.=CONDENSER
GPUMP=GEROTOR PUMP, E.S.=ENTHALPY SAMPLING, H=ENTHALPY, B=BTU, RO=DENSITY.

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** FREON FLOW MEASURED BY TURBINE FLOW METER. ACTUAL FLOW IS LESS THAN 90% OF THIS VALUE (SINCE THE FLOW METER READ TOO HIGH).

APPENDIX D

LIST OF DRAWINGS

Drawing Number	Title	Revision	Size*	Sub-Assembly†
1	Compressor coupling	A,obsolete	C	CC
2	Impeller housing adaptor		D	FP
3	Motor and pump system	A	D	
4	Drive housing layout	A	D, 2 sheets	
5	Motor shaft & rotor		D	MG
6	Motor housing		D, 2 sheets	MG
7	Motor bearing flange, comp. end		D	MG
8	Motor bearing flange, pump end		D	MG
9	Pump coupling		C	FP
10	Seal flange		C	MG
11	Bearing sleeve		B	MG
12	Bearing retainer		B	MG
13	Valve housing mod., Phase II		E	-
14	Impeller washer		B	FP
15	Pump rework		C	FP
16	Pump shaft		C	FP
17	Pump flange		D	FP
18	Pump seal	obsolete	B	FP
19	Pump end cap	A	C	FP
20	Clutch coupling	part obsolete	C	CC
21	Pump coupling washer		B	FP
22	Clutch washer	obsolete	B	CC
23	Sprocket mount		C	CC
24	Drive sprocket		C	CC
25	Drive sprocket spacer		C	CC
26	Clutch spacer		C	CC
27	Idler cap		C	CC
28	Idler spacer		B	CC
29	Idler screw		C	CC
30	Idler screw cover		C	CC

Drawing Number	Title	Revision	Size*	Sub-Assembly†
31	Idler pivot plate		C	CC
32	Idler yoke		C	CC
33	Idler sprocket		C	CC
34	Sprocket shaft & pin		C	CC
35	Upper drive cover		C	CC
36	Drive housing flange		D	CC
37	Drive housing	B	D, 2 sheets	CC
38	Compressor neck rework		C	CC
39	Two cylinder valve layout	A	D	RV
40	Valve housing	A	D, 2 sheets	RV
41	Valve	A	D	RV
42	Exhaust flange		C	RV
43	Intake flange		C	RV
44	Cap, drive end	A	C	RV
45	Cap, plugged end		C	RV
46	Seal cylinder		C	RV
47	Valve sprocket	obsolete	C	RV
48	Drive wheel	obsolete	C	RV
49	Rod bracket		B	RV
50	Spring pin	A	B	RV
51	Washer (seal cylinder)		B	RV
52	Sprocket spacer	obsolete	B	RV
53	Compressor cover rework		B	CC
54	Timing adjuster	A	C	CC
55	Flywheel & coupling		C	CC

* Description of sizes: B = 11 x 17 in., C = 17 x 22 in., D = 22 x 34 in., E = 34 x 44 in.

† Abbreviations: CC = Chain-case housing, FP = Feed pump, MG = Motor/generator, RV = Rotary valve.

Drawing originals are stored at Foster-Miller, Inc., 350 Second Avenue, Waltham, MA 02254.

APPENDIX E

SUMMARY OF HEAT EXCHANGER SPECIFICATIONS AND PERFORMANCE *

Description	Model	Specifications	Nominal Rating	Actual Performance
Evaporative condenser	Evapco PMC-185	Vane axial fans 6.3 kW fan 1 kW pump	Heat rejection: 1.52 MBH at 95°F Condensing pressure and 80°F wet bulb	60 to 95% of rated capacity
Freon boiler	Bell and Gossett RCF-200-82 Shell and tube condenser	Total heat transfer area = 289.6 ft^2 Boiler heat transfer area = 149.3 ft^2	Not rated as a boiler	Boiling load: 1.1-1.5 MBH LMTD = 5.8-9.8°F
Evaporator	Standard LDX-21A-2 Chiller-Barrel	Dual circuit, two pass	21 tons 10°F range 9°F approach For pure water at 50 gpm	18-20 tons 9°-15°F range 12°-31°F approach For 30% ethylene glycol solution, 55-33 gpm
Subcooler	Edwards B-2 Coaxial, counter flow	Tube length = 20 ft Cooled by condenser Makeup water = 3 gpm	2 tons as condenser u** = 72 Btu/hr-°F	4400-9300 Btu/hr Subcooling u** not measured
Recuperator	Packless LHXR-35	For 35 tons system capacity	15°F superheat 7.5°F sub-cooling For 16°F evaporator and 55°F condensing temperatures u** = 300 Btu/hr-°F	10°-16°F superheat 6°-8°F subcooling For 17°-20°F evaporator and 72°-93°C condensing temperatures u** = 190 Btu/hr-°F

*Extracted from Section 5 of Phase II Progress Report (1) and subsection 2.3.1 of this Report.
**u ≡ overall heat transfer coefficient.

APPENDIX F - REFERENCE

1. S. Hynek, H. Borhanian, I. Krepchin, D. Walker, C. Mariano, H. Fuller, and K. Lee, "Design and Component Testing of a Low Temperature Waste Heat Driven Refrigeration System, Phases I & II Progress Report," ORNL/Sub/80-28906/1, March, 1982.

APPENDIX G - DISTRIBUTION

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39. H. M. Leibowitz, Mechanical Technology, Inc., 968 Albany-Shaker Road, Latham, New York 12110

40. J. N. Osborne, Department of Energy, CE-124, FORSTL, 1000 Independence Avenue, S.W., Washington, D.C. 20585

41. Office of the Assistant Manager, Department of Energy, Oak Ridge Operations, Oak Ridge, Tenn. 37830

42-68. Technical Information Center, Department of Energy, P.O. Box 62, Oak Ridge, Tenn. 37830