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DESIGN, DEVELOPMENT AND TESTING OF A SOLAR-POWERED
MULTI-FAMILY RESIDENTIAL SIZE PROTOTYPE TURBOCOMPRESSOR
HEAT PUMP

Extended Test Program Interim Report

October 1982

Work Performed Under Contract No. AC03-77CS34510

United Technologies Research Center
East Hartford, Connecticut

and

United Technologies Corporation
Hamilton Standard Division
Windsor Locks, Connecticut

U.S. Department of Energy

Solar Energy

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Design, Development and Testing of a Solar-Powered
Multi-Family Residential-Size Prototype Turbocompressor Heat Pump

Extended Test Program Interim Report
Contract AC 03-77-CS34510

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October 1982

Prepared For

The U.S. Department of Energy
Division of Solar Applications
Office of the Assistant Secretary for
Conservation and Solar Applications

Design, Development and Testing of a Solar-Powered
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FOREWORD

The work described herein was performed by the Research Center of the United Technologies Corporation, in conjunction with the Hamilton Standard Division of United, for the United States Department of Energy, Office of Solar Applications, under Contract AC03-77-CS34510, formerly EG-77-C-02-4510, during the period October 1, 1979 to October 15, 1980. Included among those who contributed to this work are Messrs. T. J. Anderson, M. D. Krosney, C. E. Kepler, H. E. Khalifa, A. M. Landerman, M. D. Meader, T. N. Obee, G. T. Peters, B. W. Rhodes of UTRC and Messrs. H. Simpson, J. W. Sitler of Hamilton Standard. The Program Manager was Frank R. Biancardi (UTRC) and Gorken Melikian of UTRC and James W. Sitler of HSD were Co-Principal Investigators.

The DOE Program Manager was Mr. Fred Glaski of the DOE San Francisco Operations Office, ETEC PO. The work was conducted for Dr. Mike Davis, Headquarters Program Manager, Division of Solar Applications, Office of the Assistant Secretary for Conservation and Solar Applications, Washington, D.C. Mr. Robert Le Chevalier was the Headquarters Program Manager for the effort until February 1979. Mr. Glaski was supported in the technical administration of the program by Dr. Michael Wahlig, Messrs. Al Heitz and Harry Angerman of the DOE Lawrence Berkeley Laboratory, University of California. Their assistance, guidance and valuable comments throughout the program are gratefully acknowledged.

Design, Development and Testing of a Solar-Powered
Multi-Family Residential Size Prototype Turbocompressor Heat Pump

Extended Test Program Interim Report

SUMMARY

An experimental program was conducted to further define, improve and demonstrate the performance characteristics and operational features of an existing 18-ton solar-powered prototype heat pump designed, fabricated and partially tested during previous phases of this program (as reported in the interim technical report, UTRC R79-953050-1). The prototype heat pump is nominally sized for multi-family residential applications and provides both space heating and cooling. It incorporates a turbocompressor specially designed to operate at peak temperatures consistent with medium concentration collectors.

The major efforts in this program phase included modification and improvement of the instrumentation sensors, the laboratory simulation equipment and selected heat pump components. After implementing these modifications, performance testing was conducted for a total operating time of approximately 250 hours. Experimental test results compared favorably with performance data calculated using the UTRC computer prediction program for the same boundary conditions. A series of tests was conducted continuously over a 12-hr period to simulate operation (in the cooling mode) of the prototype heat pump under conditions typical of an actual installation. The test demonstrated that the heat pump could match the cooling load profile of a multi-family residential building.

During the system performance testing, sufficient data was taken to identify the performance of each of the major components (e.g. turbine, compressor, heat exchangers, R11 pump). Component performance is compared with that calculated using the UTRC computer predict program and with data supplied by their manufacturers.

The performance capabilities of the prototype heat pump system have been documented and recommendations are made for further design improvements which could be included in a MOD-2 configuration. The MOD-2 configuration would incorporate features that would improve system performance, reduce capital cost and most importantly improve system reliability.

CONCLUSIONS AND RECOMMENDATIONS

1. Measured heat pump system performance results from the extended tests continued to verify analytically predicted projections obtained from the UTRC computer modeling program. Performance levels achieved are considered high compared to other heating and cooling concepts.
2. A typical solar installation (i.e., in New York City area) using the UTC turbocompressor HP can match the building cooling load on a daily basis.
3. Field testing of a prototype turbocompressor HP system should be undertaken to demonstrate its performance and operational characteristics in an actual solar cooling and/or heating installation.⁽¹⁾
4. The addition of the thermal shield to the turbocompressor (T/C) markedly reduced the amount of coolant needed to cool the face seal and resulted in smoother operation.
5. Except for excessive fouling on the water side of the condensers, which could be controlled by proper water treatment, no serious signs of deterioration were detected in the heat pump components or in the R11 working fluid despite operation at temperature in excess of 300 F.
6. Additional research and development work directed toward incorporating advanced components and system features identified during testing of the prototype turbocompressor HP system should be undertaken to confirm the ultimate performance potential and operational, reliability, and durability features for such cooling and heating systems. Specific items which should be evaluated include:⁽²⁾
 - a) Alternate fluid level controls in the heat exchangers,
 - b) Alternate turbocompressor bearing feed system,
 - c) Higher efficiency power loop R11 feed pump,
 - d) Adding a regenerator to improve cycle efficiency,
 - e) Adding purge unit to automatically eliminate noncondensibles.
7. Additional improvements should be made to the UTC laboratory facility to make it more suitable for testing solar heat pump systems. These improvements should include:⁽²⁾
 - a) Use of a programmable microprocessor-based control system
 - b) Use of a more flexible automatic data acquisition system which will permit real time data analysis

(1) Subsequently, this H.P. system has been modified for cooling only duty using air-cooled condensers and is being field tested in Phoenix, Arizona as part of the SOLERAS program.

(2) These recommended improvements, except for the higher efficiency R11 pump, have been incorporated into the MOD-2 system.

INTRODUCTION

Extensive programs conducted under DOE sponsorship during the past several years have indicated that solar hot water, space heating, and space cooling systems have the potential to provide a significant reduction in the energy requirements of the U.S. during the next twenty-five years. Recent studies indicate that solar systems could be competitive on a life-cycle cost basis with conventional fossil-fueled and electrically-powered hot water, heating and air conditioning systems in major regions of the country during the 1980's if fossil fuel prices continue to increase and higher performance is achieved for various solar system elements. However, the widespread application of solar energy systems will require the detailed evaluation, development and demonstration of several potentially attractive heating and cooling systems.

UTC was an early advocate of solar-powered heating and cooling systems and was one of the first to demonstrate the performance, operational and reliability advantages of Rankine power cycle/Rankine cooling cycle systems designed for building applications. The 5-ton system built by UTC was successfully tested for almost 1000 hr (Ref. 1) over a wide range of operating conditions. However, off-the-shelf equipment was used in the laboratory system and thus the performance levels achieved were modest when compared to the ultimate potential of such systems.

Subsequently, the United Technologies Research Center, in cooperation with the Hamilton Standard Division of UTC, designed, fabricated and tested an 18-ton prototype heat pump under contract to the U.S. Department of Energy (Ref. 2). The heat pump incorporated a turbocompressor unit specifically designed to operate efficiently at temperatures achievable with medium concentration collectors. Tests conducted in a specially designed UTRC laboratory facility demonstrated operation in both cooling and heat pump modes over a range of simulated building, climatic and collector/storage subsystem conditions. Test results verified analytically predicted performance projections including achieving 18 tons of cooling at a COP approaching 0.60 at design-point conditions and a COP of about 0.80 at off-design conditions representative of summer in many U.S. locations. In spite of these encouraging achievements, testing was limited to approximately 25 hours of operation which did not permit proper assessment of the long-term reliability of the heat pump and its components. Furthermore, the laboratory simulation subsystems did not permit operation at some pertinent conditions.

Hence, the purpose of the extended testing program described herein was to perform the additional laboratory testing necessary to fully demonstrate heat pump system and component performance and reliability, as well as identify areas requiring further development. This effort would assist in the specification of a final system configuration and definition of equipment for future field test installations in various geographical locations.

This report describes the extended test program activities and presents detailed test results with comparisons of measured and analytically predicted values. An assessment of component and system performance, component durability and recommendations of system elements requiring additional refinement and development are also presented.

Detailed design information on this heat pump system and its components and the UTRC laboratory test facility were presented in Ref. 2.

TEST PROGRAM OVERVIEW

Scope of Tests

Operation and testing of the prototype heat pump system under the extended test program was carried out in the UTRC laboratory facility over a period of approximately nine months. At the end of the original program described in Ref. 2 the heat pump components were disassembled and inspected. Following reassembly of the turbocompressor and the heat pump system with selected modifications, tests were conducted over the full range of cooling mode and heating mode conditions during approximately 250 hours of operation. Utilizing the automatic data acquisition facility, data was recorded for over 2000 data slices taken at some 400 sets of water inlet conditions. Results provided by the UTRC Detail Data Reduction and Analysis Program (DDRAP) culminated in cooling mode and heating mode performance maps and evaluation of major heat pump component performance over the range of test operation.

The UTRC laboratory facility (designated Stand 28) was specially designed and constructed for the purpose of testing the prototype heat pump. To provide a means for testing the heat pump over the range of conditions and operating modes to be encountered in an actual installation, the facility contains subsystems which simulate; (1) the high-temperature water streams from the collector or furnace; (2) the ambient air temperature variation; and (3) the heating and cooling load. The high-temperature water is derived from a 12-hp furnace (steam boiler) purchased for this program. The cooling load is simulated using low-temperature water produced from house steam, and the air cooling (heat rejection from the condensers) is simulated using city water as the heat sink. The operating conditions for these three subsystems are summarized in Table I. Comprehensive instrumentation and sophisticated data acquisition systems have also been included to permit rapid and accurate analysis of heat pump performance and operating characteristics.

Various types of laboratory instrumentation sensors and equipment were provided to measure and record both water and R11 temperatures, pressures and flow rates entering and leaving each of the components comprising the heat pump module and the simulation subsystem. Visual displays of the sensor outputs provided the means for continuously monitoring the operation of the systems and when the desired run conditions were established, data (constituting one data slice) was digitized and recorded onto a flexible magnetic disk utilized in an Interdata minicomputer. The recorded data was later used in the UTRC data reduction program to establish the performance of the heat pump and each major component. In addition, mini-computer calculations made during a test run and using the measured parameters provided a quick look at the overall performance (based on water side measurements) to check the validity of the test.

A more complete description of the UTRC test facility and the procedures for taking test data are contained in Section V of the Interim Technical Report (Ref. 2).

TABLE I

SIMULATION SUBSYSTEM OPERATING CONDITIONS

<u>Service</u>	<u>High-Temp. Input to Vapor Generator</u>		<u>Low-Temp. Input to Evaporator</u>		<u>Heat Rejection from Condensers</u>	
	<u>Heating</u>	<u>Cooling</u>	<u>Heating</u>	<u>Cooling</u>	<u>Heating</u>	<u>Cooling</u>
Max. Water Flow Rate*, lb/min	503	503	334	334	334	668
Max. Pressure, psia	150	150	50	50	100	100
Supply Water Temp., F	175-295	275-325	55-125	50-60	75-95	80-115
Allowable Supply Water Temperature tolerance, F	3	3	2	2	2	2
Return Water Temp., F	160-280	260-310	45-115	40-50	90-120	95-130

*A throttle valve is located at the exit of the circulating pump in each loop to achieve any reduced flow rate.

Testing was conducted in four separate time periods to permit detailed data analysis for each test phase and evaluation of performance and advisability of procedural or configuration changes to be made before proceeding to the next tests. Discussion of the testing conducted in each time period and description of modifications incorporated are provided in the following sections.

Test Summary

A general summary of the tests completed during the extended test program is provided in Table II with the four test phases identified.

Phase I

In conjunction with reassembly of the heat pump equipment and before beginning the tests, many of the modifications identified as necessary or desirable during prior testing were incorporated. The major system changes made are summarized in Table III and described in the following paragraphs.

A major modification was the addition of a thermal barrier (heat shield) in the turbine housing of the turbocompressor as shown in Fig. 1. The barrier was designed to reduce the heat loss encountered at the turbine inlet and reduce the heat load on the seal and bearing of the turbocompressor. During this modification, the seal, bearings, compressor and turbine wheels were checked for wear and new lead seals were installed. The unit was then reassembled.

Modifications of the simulation equipment included upgrading the condenser water loop controller, repairing the vaporizer water mixing valve, the addition of drain valves in the water side of the power and cooling loop, and addition of a furnace water pressure regulator.

Other heat pump module modifications included relocation of the condenser liquid controllers, insertion of valves to permit isolation of the R11 pump, addition of a bearing temperature interlock, and the addition of new bearing feed and evaporator minimum pressure switches. The condenser liquid level controllers were installed with flexible lines so that their position might be optimized during the early tests in the program.

Instrumentation was installed to directly read the relatively small temperature differentials (ΔT) across the inlet and outlet water lines of the evaporator, vapor generator, and condensers rather than using the difference between two large independent thermocouple readings. This method of defining ΔT together with the flow rate provides a more accurate method of measuring the water side heat transfer rate.

Thermocouples were also added to the turbine bypass and compressor bypass lines. The readings from these thermocouples were used to indicate if and when the power loop pressure relief valve and/or the cooling loop surge valve inadvertently

TABLE II

TEST SUMMARY

<u>Phase</u>	<u>Date</u>	<u>Run #</u>	<u>Modifications</u>	<u>Tests Undertaken</u>
I	Sept. - Oct. (1979)	2001 - 2051	T/C w/thermal shield, carbon seal ΔT meters installed	Green runs R11 pump tests T/C removed after ~ 10 hr
II	Nov. - Feb. (1980)	2052 - 2191	T/C w/seal cooling system Flowmeters calibrated	Cooling performance tests, control testing completed ~ 75 hr tests
III	March - April (1980)	2192 - 2202	T/C w/pressure-balanced seal, level controllers adjusted	Leak checking on T/C, compressor stall
IV	May - June (1980)	2203 - 2353 <u>5000-5047</u>	Testing resumed Condenser fouling reported Chemical cleaning used	All system, component tests completed <u>(~ 135 hr completed)</u>
Total test points		~ 400		Total test time ~ 250 hr

TABLE III
SYSTEM MODIFICATIONS

Work statement items

- . Instrumentation & data acquisition
 - . Upgrade cooling loop flowmeter
 - . Install differential temperature sensors
- . Simulation equipment
 - . Improve condenser water loop controller
 - . Repair vaporizer water mixing valve
 - . Add water drain valves
 - . Add water pressure regulator
- . Pallet equipment
 - . Modify R11 level controllers
 - . Replace pressure switches
 - . Add bearing overtemp interlock
 - . Modify turbocompressor bearing lubrication system
 - . Add valves to isolate R11 pump

Additional items

- . Install new R11 pump seal
- . Relocate power loop R11 liquid flowmeter
- . Install pressure gauge on R11 pump

TURBOCOMPRESSOR WITH THERMAL BARRIER MODIFICATION

remained open; otherwise flow through these lines would go undetected. Under these conditions, performance data would not be valid because this is not a normal operating condition. The energy lost in the turbine bypass flow would appear erroneously as a smaller than actual (R11 side) vaporizer heat input. Similarly flow through the surge bypass line would incorrectly appear as a mechanical loss in the turbocompressor.

Other changes to the test instrumentation were completed to improve accuracy of data and to provide more insight into component performance.

The power loop flowmeter to measure pump liquid flow bypassed back to the power condenser was relocated upstream of the bypass valve to preclude distortion of the flow measurement by vapor inclusion.

Prior to testing, the power loop and cooling loop were flushed and leaks checked. The final leak check rates were 9.0×10^{-7} psi/min in the power loop, 282 times smaller than that established during the previous operating period and 4.9×10^{-5} psi/min in the refrigeration loop, more than 75 times smaller than previously established.

Green running of the turbocompressor was undertaken to set the turbocompressor seals and evaluate the impact of the thermal shield installed in the turbocompressor on the heat pump operating characteristics. Initial operation of the turbocompressor was found to be much smoother, the entire system was more quiescent and operation at the low-lift, low-flow conditions (i.e., 200 F vapor generator inlet water temperature levels, condenser inlet water temperatures of 85 to 100 F and evaporator inlet water temperatures of 50 to 80 F) was routinely achieved with no sign of surge or instabilities. Previous testing at these conditions had been marked by less than anticipated performance and unstable operation due primarily to oscillating heat exchanger liquid levels.

Tests were undertaken to determine the flow, head and efficiency characteristics of the R11 pump and motor. The impact of subcooling the condenser outlet R11 flow from 8 to 16 F at two different condenser temperatures was evaluated. Subcooling was found to substantially increase pump performance. Tests were suspended after about 10 hr of operation to investigate the cause of higher than anticipated vibration in the turbocompressor.

Phase II

Although prior testing had established that installation of the thermal shield could reduce heat loss from the turbine nozzle and inlet housing to other elements in the turbocompressor, the turbocompressor developed an abnormally high self-induced vibration level which examination of the disassembled parts indicated was caused by excessive shaft face seal temperature.

Analysis indicated that the cooling distribution system feeding the shaft face seal was inadequate. This was not evident during the previous system testing since bearing cartridge cooling flow volume was greater than the minimum necessary for seal cooling. The high coolant volume is believed to have promoted better flow distribution. As a result, HSD designed and installed a modified seal coolant distribution manifold intended to provide good flow distribution at normal coolant flow rates. The turbine section was refurbished and the turbocompressor was reassembled for resumption of testing.

Four additional thermocouples were added to the turbocompressor to monitor the seal cartridge temperature and six thermocouples were mounted externally to monitor the turbocompressor housing temperature. The temperatures of the seal cartridge were monitored throughout the tests via an additional visicorder to identify conditions wherein the seal coolant flow might be adequate. An external heater and means for increasing the coolant flow volume and velocity (by varying quality) were incorporated in the bearing cartridge feed system. To explore the effect of reducing the pressure differential across the seal as a means of reducing parasitic power consumption, a manual valve was installed in the cartridge coolant exit line. These changes were made to upgrade seal mechanical performance.

A number of tests were conducted to confirm the adequacy of the seal cooling design modification over the range of conditions anticipated during heat pump and cooling mode operation. Seal temperatures were found acceptable and normally below 165 F. The quality level of the coolant flow to the seal and bearing was varied from above 20 to less than 10 percent. Seal temperatures remained at satisfactory levels. Pressure to the bearing cartridge inlet was also reduced to equivalent condenser levels (i.e., 20 to 30 psia) and no deleterious effects were noted, although the running times were not extensive. These tests confirmed that the seal cooling distribution is adequate and the separate external heater (installed to achieve higher coolant vapor fractions and thus velocity) was eliminated.

Tests of the R11 pump and motor were completed and subcooling conditions for improved performance were identified.

Mapping of the system cooling performance was resumed. A wide range of run condition data points was obtained. Turbocompressor (T/C) speeds up to the normal maximum (42,000 rpm) were achieved and a broad range of operating conditions demonstrated. However, testing at the maximum lift conditions under which the highest pressures are experienced across the T/C seal was not attempted at that time due to temporary restrictions placed by Hamilton Standard personnel. Except for these selected high-lift conditions, mapping of the system cooling performance was completed.

Since the power consumption of the seal and bearing cartridge continued to be higher than anticipated, most dramatically at the lowest power conditions; additional means of improving the mechanical performance were explored. One approach is to

provide a pressure-balanced seal. In order to verify the benefits of this approach tests were planned to vary the pressure across the seal and measure mechanical efficiency changes. Such tests require very accurate measurement of the flow rates, temperatures and pressures in each loop in order to determine turbine and compressor work and hence mechanical losses. Prior to these tests, the output responses of the Eastech vapor flowmeters used to measure R11 cooling and power loop flows were carefully reviewed. The power loop flow meter readings appeared to be erratic at times and indicated higher than anticipated flow rates. Erratic output response of the cooling loop flow meter was also noticed and attributed to increased flow turbulence or the impact of liquid on the flow sensor element or both. It was speculated that vigorous boiling in the evaporator was reducing the demister effectiveness. The R11 charge in the cooling loop was reduced and the erratic Eastech flow meter response was eliminated.

Calibration of the flow meters in both the power and cooling loops was undertaken separately using a bypass around the turbocompressor. A steady-state flow was established using much of the normal system components, i.e., R11 feed pump. Analysis of the data indicated that the flow meters were measuring the correct flow rate to within a ± 2 percent error band.

In view of these results, it was decided to suspend testing to allow analysis and correlation of the test data obtained and to permit design and installation of a pressure-balanced turbocompressor shaft seal.

Phase III

The third phase of the test program was devoted to the design, production, installation (by HSD) and UTRC evaluation of the pressure-balanced turbocompressor shaft seal and testing several modifications of the bearing feed system. Only limited system performance data were obtained.

Minor modifications to the STHP test facility and heat pump module were undertaken in preparation for return by HSD of the turbocompressor unit to UTRC. Improvements to the bearing feed system were undertaken to accelerate the fill time of the pressurized tank currently used. Copper water line was wrapped around the tank and insulation was added to permit rapid condensation of the R11 entering the tank and also reduce tank pressure as a means of increasing fill rate. This modification reduced fill time by a factor of two and increased the daily running time.

The cooling loop liquid R11 line diameter was increased by replacing the existing line with one-inch copper tubing and using a larger flowmeter. Replacement and calibration of several pressure transducers on the heat pump module were completed and the water loop flowmeters were checked for proper operation.

The rig was leak checked without the turbocompressor. Since the turbocompressor has lower pressure limitations than the rig itself, leak checking can be completed quicker without the turbocompressor installed. The power loop and cooling loop were both checked at high pressure and then at vacuum. The minimum leak rates established were approximately 3×10^{-5} psi/min under vacuum conditions.

Testing of the original pressure-balanced seal carbon element configuration resulted in leakage at pressure differentials above 21 psi when tested with air. Leakage was due primarily to manufacturing deficiencies consisting of porosity of the carbon element sealing face due to insufficient lapping and the lack of precise control of the face diameters which altered the degree of pressure balance. These factors caused the seal to separate from the mating ring at moderate and higher pressure levels. In addition to correcting these deficiencies, it was decided to increase the mechanical spring-loading of the carbon element against the mating ring from 1.8 to 2.5 lb to make it more resistant to "pop-off" during initial run in and transient disturbances. Changes in the carbon element configuration resulted in an increase in the pressure balance ratio from 0.60 to 0.76. With this modification, the seal carbon element was tested at pressure differentials in excess of 50 psi in leak tests conducted at Hamilton Standard in a bench fixture and on the assembled turbocompressor unit. Pressure differentials of 50 psi are well beyond the normal operating range of 30 psi that are experienced in the turbocompressor. The maximum heat generated by the new pressure-balanced face seal was projected to be 34 percent less than the original unbalanced seal which operated for almost 75 hr with no difficulties in earlier tests of the heat pump module.

Efforts were continued to solve the turbocompressor bearing feed system difficulties encountered early in the test program. The gear pump originally selected and used to feed the turbocompressor bearings would not pump the R11 because of an inadequate net positive suction head. In addition, the Teflon gears had swollen after immersion in the R11 and the pump seized.

An oversized centrifugal pump with a stainless steel rotor was selected and the bearing feed system design was modified to utilize this pump. In this configuration, a portion of the flow provided by the centrifugal pump was bled off at the pump outlet and fed to the turbocompressor bearings. Bearing feed pressure was set by throttling the bypass loop flow. An ejector was installed upstream of the pump inlet to provide the suction head needed to avoid cavitation.

The new bearing feed system was tested in a separate R11 loop simulating the turbocompressor bearing feed system and worked well during steady operating inlet conditions. Although the tests conducted with the ejector upstream of the new centrifugal pump to provide the required suction head were successful in the separate R11 loop, when installed on the module, the pumping system would not operate due to widely varying inlet conditions. This feed system was abandoned and heat pump testing was continued utilizing the pressurized tanks as a source for bearing coolant flow.

Operation of the new pressure-balanced turbocompressor face seal was essentially trouble free. Mechanical efficiency levels appeared to have increased beyond the levels attributed to the previous seal. Monitoring of the seal and casing temperatures was continued although two of the thermocouples used became inoperative during the course of testing.

During test operation at some lower vapor generator supply temperatures, system output was noted to be significantly deficient and at least one compressor inlet temperature measurement indicated higher than anticipated temperatures. Reduced test data, while confirming the low output, indicated higher than expected compressor efficiency.

Further analysis of the compressor data was undertaken to determine the reasons for obtaining measured compressor efficiencies much higher than predicted levels during a small fraction of the tests. The analyses indicated the higher performance was observed at low vapor generator supply temperature (generally at less than 200 F). During these conditions, the turbine did not develop sufficient power so that the compressor speed was high enough to meet the head conditions imposed on the compressor by evaporator and cooling condenser water temperature levels. This produced partial compressor surge and flow recirculation leading to erroneous compressor inlet temperature readings. Compressor outlet temperature measurements were correct producing an erroneous compressor efficiency measurement.

Specific tests were conducted during which vaporizer inlet water temperature was progressively lowered, reducing power input to the compressor until this stall/surge condition was noted. Subsequent increase in vaporizer inlet water temperature resulted in a return to normal compressor operation. Repetition of this procedure substantiated prior observations and confirmed the cause of the data anomalies.

Phase IV

The final phase of testing included extended operation over a wide range of conditions to complete cooling mode and heating mode performance mapping and continuous operation over a full day's time at boundary conditions simulating those of a typical August day in the New York City area with a typical residential building demand.

Testing of the heat pump over a complete range of operating conditions from vapor generator inlet water temperatures of 300 F down to 200 F and below was conducted. Initial attempts to obtain the 300 F conditions were not successful due to reduced output of the furnace (boiler) used in the simulation laboratory. After installing a larger (4.0 gph) fuel nozzle, adjusting the fuel-air ratio and replacing the steam condensate trap with a manual valve, 300 F conditions in the vaporizer were routinely obtained.

Testing at conditions near the estimated surge line was found to be limited due to higher R11 temperature levels in the condensers than predicted. Tube fouling on the water-side of the condensers was identified as a major problem. The fouling was so severe that it restricted the water flow rate to less than 60 gpm. Subsequently a hydrochloric acid base-type scale remover was used to clean the tubes. The maximum water flow rate was increased from 57 to 88 gpm after cleaning indicating that most of the fouling material had been removed. However, some scale still remained when the cleaner supply had been exhausted. Condenser water flows of 80 gpm allowed completion of the cooling mode performance testing. Heat pump testing was conducted at 40 gpm. There is an obvious need for some sort of water treatment to avoid excessive fouling in future testing programs.

Analysis of the performance data after the condenser cleaning indicated that output increased by about 10 to 15 percent with results showing reasonable agreement with predicted performance levels that incorporate all recent loss factors. Performance data obtained from the tests were much more self-consistent, i.e., system heat balances as well as individual heat balances were much closer than in previous tests. In addition, the Eastech flow meters were operating much smoother and more consistently than in prior tests.

Operation of the turbocompressor unit was satisfactory except when the pressure differential across the new pressure-balanced seal was increased above 12 psi. Seal operating temperature levels were about as anticipated and the mechanical efficiency of the seal and bearing cartridge was higher than noted in previous testing with an unbalanced seal. Since fluid transfer occurred across the seal at high pressure differentials, testing was conducted by limiting the seal pressure differential to less than 12 psi. This introduced a performance loss but extensive operation in the heat pump mode was conducted over a range of conditions and all cooling and heat pump performance mapping was completed.

Heat pump mapping was conducted over vapor generator water inlet temperatures from 200 F to 275 F, evaporator temperatures from 55 F to 125 F, and condenser inlet temperatures from 90 F to 125 F.

Boundary conditions obtained from the analytical simulation of the cooling of a residential building (requiring an 18-ton chiller) for a typical August day were used as input to operate the heat pump module for 12 continuous hours. Vaporizer, condenser and evaporator water inlet temperatures were set every 15 minutes as defined by the UTRC-modified TRNSYS program simulation and the heat pump was cycled ON and OFF as dictated by the system controls. During the daily simulation, the inlet water temperature varied from 275 F all the way down to 220 F reflecting a continual draw-down on the storage tank. Ambient air temperature variations from 74 F to 89.1 F were simulated. A full set of test data was recorded for each 15-minute time interval.

Testing was continued to complete major work statement items and bring total test time to approximately 250 hr. The heat pump was run in many instances to accumulate operating time with no substantial data gathering.

SYSTEM PERFORMANCE RESULTS

Within the extended test program of approximately 250 hours duration, 64 channels of measured parameters which define heat pump system and component performance were recorded. Over 2000 data slices at over 400 combinations of water conditions were recorded on the Interdata minicomputer magnetic disks. Typically 5-data slices at 3 second intervals were recorded at each set of water inlet conditions. All data slices were reduced using the UTRC Detailed Data Reduction and Analysis Program (DDRAP) resulting in a volume of test information of prodigious proportions. Although this information is available, not all of it was needed or used to define system performance and some was considered invalid for various reasons.

An initial screening of the data was carried out based on the total system heat balance provided by the DDRAP. For each set of water inlet conditions, one of the 5 data slices was selected for further examination. The data indicate overall heat balances within 5 percent were attained in approximately 60 percent of the tests, with many test points within less than 1 percent. Data accuracy and consistency were steadily improved throughout the test program as obvious or suspected instrumentation reading errors were eliminated.

Most of the test data taken during the first three test phases of the extended test program provided detailed information to evaluate system operating characteristics and to indicate the effect of system modifications. In several instances, the detailed data indicated aspects of system operation which were not visually or audibly apparent. For example, a significant disparity between the two measured compressor inlet temperatures served to pinpoint the onset of compressor stall. Again the progressive decrease of measured condenser water loop flows and the degradation of heat exchanger UA values were a definite indication of progressive fouling of heat exchanger surfaces.

The test data presented in this section were recorded mainly during the final test phase and are typical of the measured performance of the final system configuration incorporating the turbocompressor with the thermal shield, improved seal cooling system, and pressure balanced seal. This configuration was operated in both cooling and heating modes over a wide range of conditions, was run continuously at boundary conditions representing a typical full day's cooling demand, and was subjected to systematic variation of selected parameters for component performance evaluation.

Cooling Mode

A summary of test points typical of operation in the cooling mode is provided in Table IV.* The range of heat pump module boundary temperature is tabulated below.

Vaporizer Inlet Water Temperature, 203.0 to 302.2 F
 Cooling Condenser Inlet Water Temperature, 85.1 to 107.9 F
 (Equivalent Ambient Air Temperature, 79 to 102 F)
 Evaporator Inlet Water Temperature, 46.0 to 60.3 F

During these tests the turbocompressor ran at speeds of 33,280 to 42,180 rpm and the system produced cooling of 5.5 to 19.9 tons operating with a Co-efficient of Performance of 0.36 to about 0.68. At boundary conditions near the design point conditions (i.e., 300 F vaporizer inlet water temperature, 103.1 F condenser inlet water temperature (95 F ambient) and 55 F evaporator inlet water temperature) the system provided (Reference Run 2342) 17.18 tons of cooling with a COP of almost 0.5. With lower ambient air temperatures (see Run 2299) the system provided almost 20 tons of cooling with a COP of 0.54.

During these tests the turbocompressor was operated over a wide range of conditions as is indicated by Fig. 2 which shows the combination of corrected head and corrected flow for the test points of Table IV. In some cases (for example runs 2289 and 2335) the compressor was operating in a marginal stall/surge mode and the performance was notably degraded from what might otherwise be anticipated for the boundary conditions.

Comparison of Test and Predicted Performance

The estimated performance of the UTRC turbocompressor heat pump was presented in the Ref. 2 Interim Technical Report (UTRC Report R79-953050-1) in Figs. 11-14 and 11-16 for operation in the cooling and heat pump modes, respectively. These performance maps were generated with the UTRC performance prediction program using computer models for system components representative of the typical installation of a production heat pump configuration.

As discussed in the Interim Technical Report (Ref. 2), characteristics of the test installation differ in many areas from those of the modeled configuration.

*Values shown in Tables IV thru IX were copied directly from computer data reduction printouts as a matter of convenience. Approximate parameter accuracies and RSS error percentages are as follows:

<u>Parameter</u>	<u>Accuracy</u>
Single Temperature	2F
Temperature Difference	0.2F
Pressure	0.5% F.S.
Water Flow Rate	0.5% F.S. ($\pm 2\%$)
Speed	0.03% F.S. (10 rpm)
Machine Efficiency	$\pm 2\%$
Heat Flow Rates	2.1% (RSS)
COP	2.9% (RSS)

TABLE IV

COOLING MODE TEST POINT SUMMARY

Run No.	Water Temperatures, * F Vaporizer/Condenser/Evaporator	Speed rpm	Cooling Output Tons	Cooling COP
2281	250.0/86.14/58.24	35180	13.397	0.6393
2282	251.7/86.47/58.20	35290	13.574	0.6811
2283	251.0/85.60/53.52	36050	12.948	0.5672
2284	250.9/86.35/48.31	37120	10.645	0.4650
2285	301.1/105.7/55.43	40990	15.436	0.4073
2286	251.6/85.14/59.71	35190	15.161	0.6328
2287	251.8/86.47/59.19	36510	14.505	0.5723
2288	249.1/97.01/58.80	36620	10.097	0.4469
2289	250.9/98.17/53.78	37660	8.780	0.3781
2291	251.5/86.18/53.44	37440	13.304	0.5877
2292	251.1/85.60/48.27	38420	11.994	0.5285
2295	251.6/86.81/59.15	36420	14.048	0.5915
2296	276.0/97.01/53.91	39100	14.002	0.4497
2297	275.5/85.73/54.39	39890	16.983	0.5447
2298	251.1/86.76/59.36	35030	13.683	0.5744
2299	300.4/84.93/58.89	41250	19.902	0.5400
2304	302.2/97.14/60.01	40400	19.059	0.5138
2305	251.2/86.02/59.58	35210	14.763	0.6258
2306	226.1/86.56/53.35	34030	8.461	0.4945
2307	203.0/85.52/60.35	30890	5.547	0.3740
2332	250.7/86.76/58.80	35910	14.462	0.6194
2333	251.0/85.97/58.28	35910	14.955	0.6620
2335	275.9/107.9/58.67	38420	11.140	0.3697
2336	252.1/85.89/58.71	36090	15.115	0.6706
2338	250.5/85.56/58.58	36110	14.670	0.6163
2341	250.8/86.56/58.24	35680	15.455	0.6265
2342	296.9/105.5/55.77	39940	17.179	0.4762
2343	297.2/105.1/51.01	41520	14.110	0.3605
2344	297.8/105.5/46.04	42180	13.545	0.3575
2346	223.2/86.35/52.48	33280	8.978	0.5185
2348	274.7/96.52/57.68	38420	16.545	0.5654
2350	251.1/86.60/58.63	35250	15.660	0.6798
2351	275.0/97.63/58.37	37550	16.535	0.5678
2352	250.3/97.51/58.33	35560	11.130	0.4922

* Inlet conditions

COOLING MODE TEST POINTS

82-6-134-1

Changes made to the UTRC performance prediction program to model the test installation were discussed and good agreement of revised performance predictions with test performance was demonstrated.

During the extended test program, the test installation was further modified and as a consequence additional changes were made to the performance prediction program computer models to represent the current configuration. These changes are listed and discussed below.

1. Since a thermal shield was incorporated into the turbocompressor, the computer heat loss function which calculates the heat transferred from the turbine inlet volute to bearing cooling flow was revised. Based on HSD estimates, the addition of the thermal shield reduces the heat transfer rate by 75 percent. The rate varies with the R11 volute and the evaporator temperature and volute flow rate. For example, the effect of adding the shield reduces the heat transfer from 122.4 Btu/min to 30.6 Btu/min at conditions corresponding to a 300 F vaporizer inlet water temperature. This difference represents an increase in energy available to the turbine.
2. Installation of a pressure-balanced turbocompressor shaft seal with increased spring preload required a change to the computer model for mechanical loss. The revised model is based on the new seal design characteristics and remains a function of machine speed and seal differential pressure. The provision to allow input of a multiplier to the basic mechanical loss function was retained to permit evaluation of the effect of variation in mechanical losses.

For the test installation, the multiplier value was specified as 0.66 based on accumulative test results. At an extreme high power input condition, with a machine speed 41,700 rpm and a seal pressure differential of approximately 31 psi, a calculated mechanical loss of 74 Btu/min is indicated. This value is significantly reduced at lower power input conditions more typical of average cooling operation.

3. Since test data continually indicated a difference between vaporizer water-side and refrigerant-side heat loads, the performance prediction program was revised to include a heat loss from the water side to ambient. The heat loss is a function of vaporizer inlet water temperature and varies from near zero at 150 F to 575 Btu/min at 300 F.

The computer model includes provision for an input multiplier to allow evaluation of the effect of variation in this heat loss. The major effect of the loss is a degradation in system COP, since the lost heat is included as input energy but is not available as turbine input energy and can produce no useful output.

With the above changes, the UTRC performance prediction program was run with input boundary conditions (water input temperatures and flow rates) identical to test values to allow direct comparison of predicted results with measured test parameters. Typical results are shown for 4 test runs in Table V.

TABLE V

COMPARISON OF COOLING PERFORMANCE TEST AND PREDICTED VALUES

Run. No	2350		2297	
	<u>Test</u>	<u>Predict</u>	<u>Test</u>	<u>Predict</u>
<u>Vaporizer</u> - Water Flow - lb/min	495.1	495.1	489.1	489.1
Inlet Temperature - F	251.1	251.1	275.7	275.7
<u>Condenser</u> - Water Flow - lb/min	667.5	667.5	661.2	661.2
Inlet Temperature - F	86.60	86.60	85.73	85.73
<u>Evaporator</u> - Water Flow - lb/min	324.5	324.5	324.3	324.3
Inlet Temperature - F	58.63	58.63	54.39	54.39
<u>Vaporizer</u> - Saturated Temperature - F	236.3	237.1	261.2	258.2
Heat Load - Btu/min	4607	4640	6235	5866
<u>Power Condenser</u> - Saturated Temperature - F	100.3	100.7	103.9	103.9
Heat Load - Btu/min	3830	3736	4845	4690
<u>Turbocompressor</u> - Speed rpm	35,250	35,850	39,890	39,610
Turbine Efficiency	0.7543	0.7619	0.7320	0.7631
Compressor Efficiency	0.8265	0.8293	0.7725	0.7786
<u>Refrigerator Condenser</u> - Saturated Temperature - F	98.67	94.15	103.5	94.57
Heat Load - Btu/min	3639	3558	4148	4112
<u>Evaporator</u> - Saturated Temperature - F	47.08	44.0	35.63	37.92
Heat Load - Btu/min	3132	3028	3396	3398
<u>Output</u> - Cooling Produced - Tons	15.66	15.14	16.98	16.99
Coefficient of Performance	0.6798	0.6448	0.5447	0.5705

TABLE V (Cont'd)

COMPARISON OF COOLING PERFORMANCE TEST AND PREDICTED VALUES

Run. No	2299		2342	
	<u>Test</u>	<u>Predict</u>	<u>Test</u>	<u>Predict</u>
<u>Vaporizer</u> - Water Flow - lb/min	482.5	482.5	483.5	483.5
Inlet Temperature - F	300.4	300.4	296.9	296.9
<u>Condenser</u> - Water Flow - lb/min	670.9	670.9	657.0	657.0
Inlet Temperature - F	84.93	84.93	105.5	105.5
<u>Evaporator</u> - Water Flow - lb/min	322.7	322.7	322.8	322.8
Inlet Temperature - F	58.89	58.89	55.77	55.77
<u>Vaporizer</u> - Saturated Temperature - F	276.2	279.0	279.5	276.7
Heat Load - Btu/min	7371	7266	7215	6819
<u>Power Condenser</u> - Saturated Temperature - F	100.9	106.0	126.8	124.3
Heat Load - Btu/min	5508	5820	6035	5488
<u>Turbocompressor</u> - Speed rpm	41,250	42,950	39,940	41,320
Turbine Efficiency	0.7457	0.7474	0.7646	0.7644
Compressor Efficiency	0.7571	0.7140	0.7550	0.7387
<u>Refrigerator Condenser</u> - Saturated	106.6	95.49	118.5	113.6
Temperature - F				
Heat Load - Btu/min	4907	4926	4213	3837
<u>Evaporator</u> - Saturated Temperature - F	38.86	39.77	45.20	40.98
Heat Load - Btu/min	3980	4013	3436	3029
<u>Output</u> - Cooling Produced - Tons	19.90	20.07	17.18	15.15
Coefficient of Performance	0.5400	0.5419	0.4762	0.4359

Comparison of measured test parameters with predicted values for these runs, as well as for the many more which have been analyzed, indicates good agreement in almost all areas and suggests causes where some disparity is found. For example, test results indicate that refrigeration condenser saturation temperature levels were consistently higher than predicted while power condenser saturation temperature levels show excellent agreement with predictions at lower heat loads and only small differences as heat load is increased. Such a result is entirely consistent with the effect of noncondensibles in the condenser. Since the refrigerant loop operates below atmospheric pressure, it is susceptible to leakage of air into the system, while the power loop is not.

Test Result Summary

The variation of performance of the test installation with evaporator water inlet temperature level for one vaporizer water inlet temperature is presented in Fig. 3 for two different ambient air temperature levels.* As anticipated, cooling capacity increases and COP is improved as evaporator temperature rise; less cooling is produced and COP degraded as ambient air temperature rise.

The combined test results at all vaporizer inlet water temperatures and for a nominal 55 F evaporator water inlet temperature were used to prepare the cooling mode performance map presented in Fig. 4.

Heat Pump Mode

A summary of test runs for operation of the test installation in the heat pump mode is provided in Table VI. The essential difference between the cooling and heat pump modes of operation is the condenser loop water flow rate. Approximately 40 gpm is used during the heat pump mode compared with 80 gpm during the cooling mode.

The range of heat pump module boundary conditions is tabulated below:

Vaporizer Water Inlet Temperature, 200.7 to 275.6 F
Condenser Water Inlet Temperature, 91.0 to 124.0 F
Evaporator Water Inlet Temperature, 59.15 to 126.6 F

The evaporator inlet water temperature represents storage water temperatures, and the condenser water inlet temperatures cover the range of return water temperature from fan/coil units operating with 70 F return air while delivering the heat load produced by the heat pump module.

*The condensing temperature shown is the measured temperature of the R11; whereas the corresponding ambient air temperature is a calculated value for a system configuration in which condenser cooling water rejects heat to the atmosphere via a dry cooling tower (forced-draft, air-to-water heat exchanger).

COOLING MODE PERFORMANCE DATAVAPOR GENERATOR INLET WATER TEMPERATURE $\cong 250\text{F}$

TYPICAL TEST POINTS WITH:

■ R11 CONDENSING TEMPERATURE $\cong 87\text{F}$ ● R11 CONDENSING TEMPERATURE $\cong 97\text{F}$

MEASURED HEAT PUMP PERFORMANCE — COOLING MODE

EVAPORATOR WATER INLET TEMPERATURE = 55F

TABLE VI

HEAT PUMP MODE TEST POINT SUMMARY

Run No.	Water Temperatures*, F <u>Vaporizer/Condenser/Evaporator</u>	Speed <u>rpm</u>	Heat Output <u>10³ Btu/Hr</u>	Heating <u>COP</u>
2308	252.2/98.7/60.18	36190	354.5	1.386
2309	250.2/103.4/79.48	31670	383.3	1.554
2310	251.2/108.5/79.01	32400	350.6	1.427
2311	250.6/108.7/100.4	27430	435.2	1.796
2312	251.2/113.6/99.32	28070	399.8	1.624
2313	273.4/113.9/100.9	29670	509.2	1.682
2314	275.4/119.1/99.73	30160	489.3	1.574
2315	272.8/113.8/80.03	34340	441.9	1.422
2316	273.9/108.2/79.52	33270	485.4	1.542
2317	275.6/102.5/59.15	37360	452.9	1.394
2318	273.7/117.5/1024.1	25960	543.7	1.802
2319	276.5/124.0/126.6	27810	546.6	1.794
2320	225.2/93.06/59.62	33480	275.1	1.396
2321	225.1/96.52/58.93	34430	253.0	1.287
2322	225.6/96.72/79.23	28100	325.2	1.685
2323	226.1/102.6/79.10	29950	280.9	1.471
2324	226.7/102.9/99.73	24620	371.6	1.959
2325	226.3/107.7/99.36	25390	336.3	1.787
2326	201.4/108.1/125.4	17820	319.5	2.365
2327	201.0/101.9/125.7	18280	348.5	2.552
2328	201.1/102.7/99.90	21810	259.0	1.856
2329	200.7/96.93/100.2	20850	297.8	2.103
2330	201.1/97.63/79.61	20470	209.9	1.426
2331	202.4/91.00/78.80	26460	246.7	1.635

*Inlet conditions

The turbocompressor operated at speeds of 17,820 to 37,360 rpm while the system provided heat outputs of 209.9 to 546×10^3 Btu/hr with Coefficients of Performance ranging from 1.29 to 2.55. Operating speeds in this mode are generally lower than for the cooling mode since the evaporator operates at higher temperature levels and lower lift is required of the compressor. This difference can be seen by comparing the operating points plotted on the compressor maps for the two modes of operation. The cooling mode data are presented in Fig. 2 and the heating mode compressor operating points are shown in Fig. 5. Compressor efficiencies of 85 percent were obtained at several test conditions. It can be noted that at least one operating point is definitely in the compressor surge region. This was confirmed by R11 compressor inlet temperature measurements made within the compressing housing. While one temperature was a normal 56.2 F, compared to evaporator water inlet temperature of 58.93 F, a second compressor inlet temperature was 112.3 F, indicating stalled flow.

Comparison of Test and Predicted Performance

With the UTRC performance prediction program modified as previously discussed, the computer program was run with heat pump module boundary conditions identical to test values to permit direct comparison of results. System parameters for two test points (Runs 2317 and 2324) are compared in Table VII. Again good agreement is noted between test performance and predicted values. The same minor disparity in refrigeration condenser saturation temperature discussed above is apparent in the heat pump operating mode, test values being slightly higher than anticipated by the prediction program.

Test Performance Summary

The results of these tests, conducted with a condenser water flow rate of 40 gpm, were used to prepare the heat pump mode performance map presented in Fig. 6. The heat output and COP are indicated for four storage (i.e., evaporator inlet) water temperatures from 60F to 125F and for four vaporizer water inlet temperatures from 200 F to 275 F.

Integrated System Simulation

An analytical and experimental effort was undertaken to demonstrate how well the UTRC turbocompressor heat pump system would satisfy the building cooling load in a representative solar installation. This installation was assumed to be a multi-family residential type building located in New York City. The building size was selected such that its cooling needs would be satisfied by a 18-ton air conditioning system at the standard 2 1/2-percent design conditions (i.e., 91 F dB outdoor, 80 F dB and 67 F wB indoor). In addition to the heat pump module which rejects heat via an outdoor air-to-water heat exchanger, the solar installation includes a 4000-ft²/6000-gal medium concentration collector/hot water storage subsystem and chilled water fan-coil units (no cold water storage used). The control system includes room thermostats set to turn on the fan coils and the heat pump when the room air temperature reaches 78 F and shuts the system off when it drops back to 76 F.

TYPICAL HEAT PUMP MODE OPERATING DATA

TABLE VII

COMPARISON OF HEATING PERFORMANCE TEST AND PREDICTED VALUES

	<u>Run. No</u>	<u>2317</u>		<u>2324</u>	
		<u>Test</u>	<u>Predict</u>	<u>Test</u>	<u>Predict</u>
<u>Vaporizer</u> - Water Flow - lb/min		489.1	489.1	500.6	500.6
Inlet Temperature - F		275.6	275.6	226.7	226.7
<u>Condenser</u> - Water Flow - lb/min		331.7	331.7	339.6	339.6
Inlet Temperature - F		102.5	102.5	102.9	102.9
<u>Evaporator</u> - Water Flow - lb/min		324.8	324.8	327.4	327.4
Inlet Temperature - F		59.15	59.15	99.73	99.73
<u>Vaporizer</u> - Saturated Temperature - F		255.0	258.9	210.9	216.1
Heat Load - Btu/min		5400	5591	3157	3415
<u>Power Condenser</u> - Saturated Temperature - F		127.1	127.9	121.5	122.9
Heat Load - Btu/min		4771	4531	2925	2854
<u>Turbocompressor</u> - Speed rpm		37,360	38,020	24,600	25,220
Turbine Efficiency		.7644	.7641	.7115	.7143
Compressor Efficiency		.6090	.6667	.8489	.8497
<u>Refrigerator Condenser</u> - Saturated Temperature - F		117.7	112.6	116.3	114.3
Heat Load - Btu/min		2778	2926	3259	3347
<u>Evaporator</u> - Saturated Temperature - F		45.64	47.74	84.50	86.20
Heat Load - Btu/min		2274	2321	2991	3050
<u>Output</u> - Heat Delivered - Btu/hr		452,900	447,400	371,000	372,100
Hot Water Temp. - F		123.2	124.8	119.7	120.9
Coefficient of Performance - Heating		1.394	1.313	1.956	1.800

MEASURED HEAT PUMP PERFORMANCE MAPRETURN AIR TEMPERATURE $\approx$ 70 F

The UTRC-modified TRNSYS computer program (with New York City weather data) was used to define the heat pump boundary conditions (i.e., hot storage tank temperature, room temperature and ambient temperature) for a typical August day. August 26 was selected as the typical day and the initial (1 A.M.) storage tank temperature was assumed to be 275 F. The analytically predicted prototype heat pump performance map was used in the TRNSYS analysis. The heat pump module boundary temperature profiles thus determined are shown in Fig. 7. Hot storage tank temperatures available to power the module vary from 275 F down to 220 F reflecting a continual extraction of energy. Ambient temperatures follow a typical pattern from night lows of approximately 75 F to a mid-day peak of 88F. UTRC-modified TRNSYS program predicted room temperatures in the range of 76 F to 78 F show the effect of system controls and the resulting cyclic operation. These temperature profiles were then used to set the water temperature in the laboratory simulation subsystems and tests for one full day of operation were conducted.

Test Results

Results of the one-day operation simulation test are shown in Fig. 8 where the heat pump module output profile is compared with the TRNSYS predicted building load. In the morning at lower ambient temperatures, the UTRC system output considerably exceeds the instantaneous demand and the system is turned off for a short period of time as room temperature is brought within the allowable range.

Detailed analysis of the data indicates that start-up and shut-down transients involve considerably shorter time intervals than modeled in the UTRC-TRNSYS simulation. Measured time to reach full output was approximately 15 minutes compared with 1 hour assumed in the simulation. Cooling produced ranged from 4 tons to 17 tons and a peak COP of 0.75 was attained.

A tabulation of the more significant test results is summarized in Table VIII. The total cooling produced for the day was 96 percent of that required by the building (i.e., 108.5 ton-hr versus 113 ton-hr) based on water-side measurements. The actual cooling produced by the heat pump as measured on the R11 side, is even closer to the building load and the heat pump more than adequately satisfies the building load. An average COP for the day would be 0.58 as indicated in Table VIII.

TYPICAL TEMPERATURE PROFILES

NEW YORK CITY,
MULTIFAMILY BUILDING
AUGUST DAY

TYPICAL FULL DAY SYSTEM OPERATION

NEW YORK CITY,
MULTIFAMILY BUILDING
AUGUST DAY

TABLE VIII

ONE-DAY SIMULATION RESULTS SUMMARY
Output/Input - Ton Hr

Building load	113.03
Measured STHP waterside cooling output/input	108.50/211.0
Measured STHP Rll-side cooling output	122.55
STHP TRNSYS predict output/input	125.73/186.4
Average daily COP*	0.58

* Based on Rll-side cooling output measurements

COMPONENT PERFORMANCE

Turbocompressor

The basic design features and physical characteristics of the turbocompressor used in the extended test program were presented in detail in Ref. 2. In anticipation of this program and during the course of the testing, modifications were incorporated as previously discussed. The performance presented here is that measured with all changes incorporated including 1) the turbine section thermal shield, 2) the revised shaft seal cooling flow distribution system, and 3) the pressure-balanced shaft seal.

The turbocompressor was run over the full range of speeds to an overspeed cutout of 46,000 rpm without incident, and performance data taken over the range of 17,820 to 42,180 rpm. Operation was notably quiet and smooth. Even operation in the surge region was found to be non-violent, the onset of surge in many instances being detectable only by instrumentation readings.

Compressor

The compressor was operated over a wide range of conditions as shown by the test points in Fig. 2 for the cooling mode and in Fig. 5 for the heating mode. Measured performance parameters showed general good agreement with predictions. Table IX gives a comparison of measured (TEST) compressor efficiency with predicted values (MAP) for typical operating conditions. Obvious anomalies are noted for some test points (for example, Runs 2070, 2136, and 2137) when the compressor was operating in a marginal surge condition with flow recirculation and consequent distortion of compressor inlet temperature measurements.

The comparison of test and predicted compressor efficiencies is more graphically shown by Fig. 9 whereon the variation of compressor efficiency with corrected flow is plotted for three values of constant corrected head. Test points indicated by open symbols (final test configuration after cleaning condensers) show excellent agreement with the predicted efficiency variation.

Turbine

While compressor operation covers a wide range of conditions, turbine operating parameters for the same system boundary conditions vary significantly less. This may be noted in Fig. 10 where typical test points are plotted. This figure shows turbine efficiency as a function of head and weight flow coefficients. A maximum predicted turbine efficiency of 0.780 indicates a map island efficiency of 0.98 times 0.78 or 76.44 percent.

TABLE IX

TYPICAL COMPRESSOR EFFICIENCY COMPARISON SUMMARY
 DATA AT VAPOR GENERATOR INLET TEMPERATURE OF 227-300F

<u>RUN NO.</u>	TURBOCOMPRESSOR	<u>COMPRESSOR EFFICIENCY</u>	
	<u>SPEED</u> <u>RPM</u>	<u>TEST</u>	<u>MAP</u>
2066	34,670	.7992	.8610
2067	35,910	.8248	.8179
2068	36,530	.8240	.8073
2070	33,450	1.3340	.7795
2071	35,650	.8467	.7380
2072	36,270	.8090	.7358
2073	35,490	.8249	.7773
2074	34,890	.8339	.8180
2109	35,410	.8320	.8248
2136	34,500	.9700	.7740
2137	34,400	.9540	.7539
2286	35,190	.8825	.8158
2297	39,890	.8201	.7725
2299	41,250	.7396	.7571
2338	36,110	.8544	.8141
2341	35,680	.8318	.8302
2350	30,100	.8649	.8265
5015	40,200	.7143	.7050
5038	32,850	.8431	.8446
5039	33,630	.8408	.8240
5041	33,340	.8252	.8048

COMPRESSOR EFFICIENCY COMPARISON

CORRECTED HEAD,		
Btu/lb	TEST	PREDICTED
~5.5	○	---
~6.0	△	---
~7.0	□	---

OPEN SYMBOLS AFTER CONDENSERS CLEANED

SOLID SYMBOLS PRIOR TO CLEANING

FIG. 9

The nominal variation of both weight flow and head coefficients is a consequence of operation over a limited inlet temperature range ($\sim 640-740$ R) with a fixed nozzle area configuration with choked flow. The head coefficient is influenced to some degree by power condenser temperature levels.

A comparison of measured turbine efficiency and predicted values is shown in Fig. 11 where the variation of efficiency with head coefficient is plotted for two values of constant weight flow coefficient. While the operating range is not great, good agreement of test measured efficiency with predicted values is indicated.

Mechanical Features

After completion of the extended testings, the turbocompressor was disassembled at Hamilton Standard and thoroughly inspected to assess the condition of all elements after more than 250 hours of operation. All parts were found to be in excellent condition with no indication of abnormal deterioration. Selected elements of the turbocompressor were photographed as disassembled and are shown in Figs. 12 and 13.

Elements of the turbocompressor, bearings and seals were returned to manufacturers for inspection and analysis. Examination of the turbocompressor bearings by the manufacturer revealed them to be in good condition. The degree of wear is consistent with the long service life anticipated for these bearings. Slight scratching in a few balls and axial marks on the bearing race were noted but these do not indicate any potential problems.

Detailed examination of the pressure-balanced shaft seal was carried out to determine the cause of anticipated fluid transfer across the seal at the pressure differential of approximately 12 psi although initial sealing tests showed no fluid transfer at pressure differentials in excess of 50 psi.

Basic to the design of the pressure-balanced seal is a precise balance and control of all forces acting on the carbon seal element against the tungsten carbide element. There are two major forces acting to seat the carbon seal element: 1) a mechanical spring force, and 2) a high pressure force acting on a small annular area produced by the differential operating pressures. Opposing these forces is a pressure force acting on the annular seal contact area - this pressure varying from high pressure at the annulus inside diameter to low pressure at the annulus outside diameter. This pressure force acts in a direction to unseat the carbon seal element.

As designed and initially installed with carefully lapped surfaces, the flow passage from annulus inside radius to outside radius exhibited a constant gap resulting in a parabolic pattern of pressure decay across the seal face from high to low pressure. Integration of this pressure force over the total face area results in an unseating force less than the combined seating force, thus providing a net element seating force adequate to prevent fluid transfer.

TURBINE EFFICIENCY COMPARISON

TEST FLOW COEFFICIENTS

 $\Delta < 1.0$ $\circ > 1.0$

82-6-134-8

TURBOCOMPRESSOR INSPECTION
TURBINE ELEMENTS

NOZZLE

NOZZLE 5X

HEAT SHIELD

BEARING CARTRIDGE END CAP LEAD SEAL

TURBOCOMPRESSOR INSPECTION
COMPRESSOR ELEMENTS

OUTER BEARING SLEEVE

ROTOR OUTBOARD SIDE

SEAL RING

INNER BEARING CARTRIDGE SLEEVE

While visual inspection of the seal disassembled after some 250 hours of operating showed all elements to be in apparently good condition; close examination of mating surfaces with a profilometer revealed that some wear had occurred on the carbide surface. The wear pattern was such as to result in an equal gap across the seal face with contact at or near the outside radius. With such a configuration the parabolic pattern of pressure decay is distorted, with higher pressures acting on more of the seal face area thereby increasing the unseating force. With progressive wear this process resulted in eventual unseating of the seal element allowing some fluid transfer across the seal.

After exploring several possibilities, it has been concluded that deterioration of the carbide surface must have resulted from a chemical reaction of the cobalt binder material. Carbide was transferred and found inbedded in the carbon face. Although revisions to spring forces and element dimensions and improved dimensional control could improve seal effectiveness and life, the long term solution to this problem lies in the use of a more corrosion resistant binder and the elimination of contaminants that are formed and cause the corrosion (wear).

Heat Exchangers

The actual performance of the four heat pump heat exchangers was assessed from the data collected during the additional overall system tests conducted from November 16, 1979 to June 26, 1980. This performance was analyzed and the valid data points were compared with performance values calculated using the UTRC prediction models and values provided by the manufacturer for the specified design point conditions. The heat exchanger specifications provided by the manufacturer are summarized in Table X. Heat transfer rates in each of the four water-to-R11 heat exchangers were determined from measured water flow rates and water temperature change (ΔT 's). The mean temperature difference was calculated from the measured water temperatures in and out of each heat exchanger and the R11 saturation temperature which was measured directly or established from pressure measurements. Finally, the overall heat transfer coefficient was calculated from the heat transfer rate, mean temperature difference and the total (installed) heat transfer area. The small amount of heat transfer and temperature difference profile represented by any superheating or subcooling which may have been observed was neglected when calculating the overall heat transfer coefficients from the test data. No attempt was made to establish the effective areas for boiling or condensing by measuring the liquid levels in the evaporator, vaporizer or condensers. The heat transfer rates in each heat exchanger were also calculated from R11 flow rate measurements and R11 enthalpies determined from temperatures measured entering and leaving. Only those data points showing a heat balance within 10 percent on the water and R11 sides were considered valid for the heat exchanger performance assessment.

The heat transfer coefficients determined using the UTRC prediction computer program and/or used by the heat exchanger manufacturing to size the four heat exchangers do not account for superheating R11 in the vaporizer or evaporator or

TABLE X
HEAT EXCHANGER SPECIFICATIONS

	<u>Vaporizer</u>	<u>Evaporator</u>	<u>Cooling condenser</u>	<u>Power condenser</u>
Capacity, 10^3 Btu	372	217	270	322
Heat transfer rate, Btu/hr-ft ² -F	72	81	95	120
Type	Shell/tube	Shell/tube	Shell/Tube	Shell/tube
Design pressure, psig	300/150	50/150	125/125	125/125
Design temperature	300/325	150/150	150/150	150/150
Fin spacing, fpi	28	19	19	19
Material	Steel/316SS	Steel/Cu	Steel/Cu	Steel/steel
Total heat transfer area, ft ²	266	285	355	310
Shell size, dia. (in.), length (ft)	12 - 8	14 - 8	12 - 8	12 - 7

for subcooling in the condensers. Therefore, any attempt to evaluate the performance of the heat exchangers by comparing the heat transfer coefficients where substantial superheating or subcooling (> 3 F) was noted with predicted values must account for this type of operation. Large amounts of superheating or subcooling can be real due to low R11 liquid levels in the vaporizer or evaporator or flooding the lower tubes in the condensers. It can also be apparent due to test error in measuring the saturation or exit temperature in each heat exchanger. A brief error analysis was conducted for selected test points to indicate the sensitivity of the overall heat transfer to possible measurement errors.

Further details as to the performance of each of the heat exchangers are discussed separately in the following paragraphs.

Vaporizer

The test data for the power loop vaporizer showed little or no superheating indicating that the liquid was maintained at a level sufficient to cover all of the heat transfer surfaces. Therefore, the test performance (i.e., the overall heat transfer coefficient) can be compared directly with predicted values. Heat transfer coefficients for several test points are depicted in Fig. 14 for water inlet temperatures from approximately 220 to 300 F and a relatively constant tubeside water velocity of 4.9 ft/sec. Predicted performance obtained from the UTRC computer modeling procedure is also shown for the same set of operating conditions. Instead of establishing an error band on the test performance, here it is applied to the calculated prediction values to show the sensitivity of a ± 0.5 F error in establishing R11 saturation temperature. As indicated, the calculated overall heat transfer coefficient changes approximately 10 percent even for this small difference (0.5 F) in saturation temperature. For the most part, the test performance is close to the UTRC predicted performance and well above the manufacturers' specification of 71.7 Btu/hr-ft²-F at a water inlet temperature of 310 F. Therefore, it can be concluded that this heat exchanger performed as expected.

The calculated (UTRC predicted) performance shown in Fig. 14 is based on a waterside fouling factor of 0.001 hr-ft²-F/Btu. Since the heat exchanger has been operated for more than one year without any cleaning of the surfaces and still provides the same level of expected performance, it appears that fouling is not a significant problem. The use of a moderate water velocity (~ 4.9 ft/sec), a closed water circuit and stainless steel tubes contributed to minimizing fouling of the waterside surfaces. At the end of the test program, inspection of the waterside surfaces (through the connections on the bonnet) showed all the surfaces (stainless steel) to be quite clean and free of deposits which would inhibit heat transfer (see Fig. 15). The tube outer surfaces were also inspected and found to be free of deposits or corrosion.

VAPORIZER TEST PERFORMANCEWATER VELOCITY = 4.9 ft/sec ($\pm 5\%$)BAND REPRESENTS $\pm 0.5^\circ\text{F}$ IN SATURATION TEMPERATURE ACCURACY

HEAT EXCHANGERS AFTER EXTENDING TESTING

VAPOFIZER WATER SIDE

EVAPORATOR WATER SIDE

Evaporator

The heat transfer coefficients achieved in the cooling loop evaporator at a constant water velocity appeared to correlate with the degrees of superheat as shown in Fig. 16. Superheating is a direct result of not maintaining a sufficient liquid level to effectively cover all of the heat transfer surfaces. Since the heat transfer coefficient for heating R11 vapor in this flooded heat exchanger is extremely low, the overall (average) value based on the total installed area would be expected to decrease with increasing superheat. The liquid level in the evaporator varied widely because fluid from the initial charge was withdrawn periodically to fill the bearing feed tank and subsequently was reintroduced into the evaporator as testing proceeded. The bearing feed tank held approximately 100 lb of R11 (with oil) which represented as much as 50 percent of the initial charge in the cooling loop and withdrawing this amount from the evaporator could lower the liquid level 2 to 3 inches, exposing the two top rows of tubes. As shown in Fig. 16, the overall heat transfer coefficients achieved when the liquid level was sufficient to limit superheating to less than 3 F were equal or better than the predicted values which are based on no superheat.

An error analysis of the evaporator test performance indicates that a 1 F error in measuring the saturation can result in a 5 to 15 percent change in the overall heat transfer coefficient.

Since the achieved performance with low superheat matches the predicted values, it can be concluded that progressive fouling on the waterside had little effect. Indeed, inspection of the waterside (through the water connections on the bonnet) after conclusion of the extended testing program revealed that the inside tube surfaces (copper) had only a very light deposit which would not significantly inhibit heat transfer. Some heavier deposits (from an undetermined source) were observed on the tube sheet (see Fig. 15) but they are of little importance to the successful operation of the evaporator. The evaporator was operated using untreated city water in a closed loop so that contaminants which could cause fouling were not continuously introduced. The tube outside surfaces were also inspected and found to be as clean or cleaner than when initially installed.

Condenser

The power loop and cooling loop condensers were cooled by untreated city water using a series flow, semi open arrangement so that the potential for waterside fouling was greater than in either the vaporizer or evaporator. Therefore, it was important that the evaluation of the condenser heat transfer achieved performance should consider the effects of waterside fouling. The other two parameters that significantly influence the heat transfer performance are the water velocity in the tubes and flooding of the tubes with R11 liquid which reduces the effective area for condensing duty but increases subcooling.

EVAPORATOR TEST PERFORMANCEWATER VELOCITY = 4.3 ft/sec ($\pm 5\%$)

82-6-134-10

The overall heat transfer coefficients achieved in the power condenser are depicted in Fig. 17 as a function of water velocity. The measured R11 saturation temperatures for test runs where unusually large amounts of subcooling were indicated were corrected to reflect a nominal subcooling of 3 F. It is believed that indications of larger amounts of subcooling are due to measurement errors in determining the R11 saturation temperature. Correcting the measured saturation temperature to reflect 3 F of subcooling increased the calculated overall heat transfer coefficient by 25 to 100 percent. The overall heat transfer coefficients achieved (after any necessary adjustments) during several test runs are compared to calculate (predicted) values for a range of waterside fouling factors and water velocities. Most of the test data was taken at water velocities close to the design point value of 5.9 ft/sec and the performance achieved approximates the predicted value based on a waterside fouling factor of $0.001 \text{ hr-ft}^2\text{-F/Btu}$ as well as the value provided by the manufacturer ($119 \text{ Btu/hr-ft}^2\text{-F}$). Initial tests conducted after being down several months for equipment modifications revealed that the normal water flow pumped through the condenser cooling loop ($\sim 80 \text{ gpm}$) could not be achieved due to added pressure losses. This water loop was subsequently flushed with a chemical cleaning agent to remove deposits which fouled the heat exchanger surfaces and restricted the flow. After cleaning, normal flow rates could once again be achieved. An evaluation of the power condenser heat transfer performance during the time of reduced flow ($\sim 58 \text{ gpm}$) indicated that fouling factor was approximately $0.002 \text{ hr-ft}^2\text{-F/Btu}$ (see Fig. 17). After cleaning the fouling factor apparently returned to its normal value of approximately $0.001 \text{ hr-ft}^2\text{-F/Btu}$.

The test performance for the cooling loop condenser is presented in Fig. 18 using the same format as for the power loop condenser. Again the R11 saturation temperatures for test runs where unusually large amounts of subcooling were adjusted to the nominal value of 3 F when assessing the condenser overall heat transfer coefficient. The effect of adjusting the saturation temperature to reflect 3 F of subcooling is approximately the same as for the power condenser (i.e., ~ 25 to 100 percent in the overall heat transfer coefficient). It is believed that this adjustment is justified because the saturation temperature was identified from a pressure measurement which could include the partial pressure of noncondensable gases which tend to collect in the condenser. The probability of noncondensibles (air) being present in the cooling loop condenser is much greater than in the power loop condenser because of the subatmospheric operating pressure in the evaporator. Not only do the noncondensable gases give a false indication of the true saturation temperature but they also degrade the condensing heat transfer coefficient to some extent. This effect could not be accurately assessed because little data of this nature are available in the published literature for refrigerants (more is available on steam condensation) and the instrumentation was insufficient to determine the quantity of noncondensibles

COOLING LOOP CONDENSER TEST PERFORMANCE

present. New turbocompressor systems being developed at UTRC will be equipped with a commercially available purge unit which will automatically remove non-condensibles from the system.

Nonetheless, the test performance (overall heat transfer coefficients) at water velocities close to the design point value (4.3 ft/sec) agree with the UTRC predicted values based on a waterside fouling factor of 0.001 to 0.002 hr-ft²-F/Btu as shown in Fig. 18. The test values achieved when the condensers were apparently fouled agree with predicted values based on a fouling factor of 0.002 to 0.003 hr-ft²-F/Btu at a apparent water velocity of 3.4 ft/sec.

Heat Exchanger Summary

Analyses of the test data indicate that all four heat exchangers can achieve the performance levels calculated using the UTRC prediction models, provided they are operated in the prescribed manner for which they were designed. Excessive superheating and subcooling can be avoided by proper level control in each heat exchanger. Significant fouling effects can likely be eliminated by the use of treated cooling water and normal periodic maintenance. The degrading effects of noncondensibles in the condensers can be reduced by the use of a purge unit. The heat exchanger test findings which have a bearing on the design of a new improved turbocompressor heat pump system are summarized in Table XI.

R11 Pump

The performance characteristics of the power loop R11 feed pump were derived from flow, pressure and electric power measurements taken during the overall heat pump system tests. Typical pump flow/heat data points are shown in Fig. 19 and compared with the performance curve provided by the manufacturer (solid line). During some tests, the power loop condenser was deliberately flooded to achieve specific amounts of subcooling and hence net positive suction head (NPSH) to assess the effect on pump performance. In the Ref. 2 test program, cavitation was identified as the possible reason for the test performance being lower than that specified by the manufacturer. The test data points presented in Fig. 19 show that both head/flow and overall pump and water performance are improved with increased subcooling which reduces cavitation. However, the 10 to 15 F of subcooling needed to approach the performance specified by the manufacturer is not achieved without penalty. Although a separate subcooler could be utilized to avoid flooding the condenser, which would significantly degrade its condensing capacity, the source of cooling water in an actual system installation would be derived from the outdoor water-to-air heat exchanger (dry cooling tower). Since this water would be available at a temperature not much more than 10 to 20 F below the condenser saturation temperature, it would be difficult to achieve more than 10 to 12 F of subcooling by this method. In future turbocompressors the follow-on programs, an evaluation will be made to assess the benefits and penalties of providing increased subcooling to the pump to improve its flow/head and efficiency characteristics.

TABLE XI

HEAT EXCHANGER TEST SUMMARY

<u>Component</u>	<u>Overall heat transfer coefficients, Btu/hr-ft²-F</u>		<u>Comment</u>
	<u>Test</u>	<u>Predicted</u>	
Vaporizer	100 - 145	118 - 144	No fouling or corrosion
Evaporator	95 - 110	92 - 102	Vapor space marginal, no tube fouling or corrosion
Power condenser	120 - 140	130	Fouls with untreated water
Cooling condenser	100 - 130 ⁽¹⁾	135	Fouls with untreated water

(1) Test saturation temperature corrected
for noncondensibles

R11 PUMP TEST RESULTS

REGENERATIVE TURBINE TYPE

PUMP MOTOR SPEED = 3500 rpm

— MANUFACTURER'S DATA (BASED ON 80% MOTOR EFFICIENCY)

--- TEST RESULTS WITH:

□ 15F SUBCOOLING

○ 10F SUBCOOLING

● < 10F SUBCOOLING

Fluid Stability

One of the tasks of the STHP extended test program was to assess the stability of the Rll working fluids in each loop of the heat pump module. Although the total operating time (≈ 250 hrs) for these tests is not considered sufficient to draw definite conclusions regarding long-term stability, an effort was made to detect any signs of decomposition. Chemical analyses were made of fluid (liquid) samples taken before, during and at the end of the test period. Metal surfaces in contact with hot Rll or Rll-oil mixtures were examined at the end of the tests for signs of corrosion, solid deposits or copper plating. As summarized in Table XII, no significant amounts of chloride ions were measured in fluid samples taken from both the power and cooling loops. There was also no sign of copper plating on turbocompressor and pump hot spots (i.e., the seals), corrosion on heat exchanger and other surfaces or solid deposits. These symptoms would have been present had there been significant amounts of Rll or oil decomposition. Some discoloration, however, was noted in fluid samples taken from the power loop at the end of testing. This may have been due to metallic impurities (mostly iron, silicon and calcium) which were apparently too small to be removed by the filter-drier unit.

Although these findings must be tempered by the knowledge that fresh Rll was periodically added to the power loop (and Rll-oil mix removed from the cooling loop) due to fluid transfer across the seal in the turbocompressor when the pressure-balanced seal was in use, there was no evidence of fluid decomposition which would discourage the use of Rll in future commercial STHP units similar in design to the prototype unit tested in the laboratory. Further data indicating possible fluid decomposition will be compiled as a result of operating the modified system under the SOLERAS program. Here the use of copper in the air-cooled condensers could increase the possibility of Rll decomposition in the power loop.

Control System

Several heat pump and facility simulator control modifications were made to improve performance and operation. The actions taken and the test results achieved are summarized in Table XIII.

Early attempts in the extended test program to operate at conditions requiring a 300 F vaporizer water temperature were unsuccessful because the output of the boiler in the Solar Lab could not meet the heat pump load requirement. Replacing the 3.5 gph fuel nozzle with a 4.0 gph nozzle, adjusting the fuel-air ratio and fuel oil pressure and insulating the vaporizer water lines provided some improvement. Problems with the steam condensate trap which also contributed to this situation could not be resolved so it was replaced with a manual valve. After some adjustment to this valve, the rig could be run continuously with a 300 F vaporizer temperature and operating points with this condition were tested.

TABLE XII

CHEMICAL ANALYSIS OF R11 WORKING FLUID CONTAMINATION AND BREAKDOWN

R82-953050-1

Contaminant Emission Spectrograph Analysis

Sample No.	Date	Description		Fe	Si	Al	Mn	Cr	Ni	Cu	Ca	Na	Total
83793	7/1/80	Power Loop Liquid - Sample #1	%	36.56	14.46	0.76	0.76	0.78	1.70	0.76	36.56	7.66	100.0
			*ppm	.0339	.0134	.007	.0007	.0007	.0016	.0007	.0339	.0071	.0927
83795	7/1/80	Power Loop Liquid - Sample #2	%	32.41	12.78	12.78	0.93	0.62	1.25	0.28	32.41	6.54	100.0
			*ppm	.0707	.0279	.0279	.0020	.0014	.0027	.0006	.0707	.0143	.2181
83794	7/1/80	Cooling Loop Liquid	%	22.44	48.08	4.80	4.81	1.60	1.12	1.12	11.22	4.81	100.0
			ppm	.0675	.1445	.0144	.0145	.0048	.0034	.0034	.0337	.0145	.3007

Calorimetric Analysis for Free Chlorine

Sample No.	Date	Description	Free Cl Ions - ppm	Comment
83793	7/1/80	Power Loop Liquid - Sample #1	< 10	None detectable
83795	7/1/80	Power Loop Liquid - Sample #2	< 10	None detectable
83794	7/1/80	Cooling Loop Liquid	< 10	None detectable

Evaporative Analysis for Oil Content

Sample No.	Date	Description	Oil % - by weight
83793	7/1/80	Power Loop Liquid - Sample #1	0.0789
83795	7/1/80	Power Loop Liquid - Sample #2	0.0681
83794	7/1/80	Cooling Loop Liquid	5.063

*by weight, based on total fluid sample

TABLE XIII
CONTROL SYSTEMS TEST RESULTS

<u>SIMULATOR CONTROLS</u>	<u>ACTION</u>	<u>PERFORMANCE</u>
Condenser Water Temp.	Modulator Installed	± 1 F Control
Vaporizer Inlet Water Temp.	Furnace Water Press. Regulator Installed	$T_{out} \leq 300$ F
<u>HEAT PUMP CONTROLS</u>		
Level Controllers		
- Power Loop	P.O.V. - Port Reduced 23/32 9/16 in. Plug Change 4 Notch 2 Notch Spring Rate Increased 5 10 lb/in P.F.V. - Cartridge Port Decreased	Some Improvement but controllers. Remains Unstable
- Cooling Loop	P.O.V. - Spring Rate Increased 1 5 lb/in. P.F.V. - Cartridge Port Decreased	Some Improvement Stability Marginal
<u>BEARING OVERTEMP. INTERLOCK</u>		
	T-C Embodied in Carbon Seal	Operated Up to 250 hr

Initial attempts at mapping heat pump performance were also hampered by an inability to maintain the condenser cooling water temperature at constant selected values. The use of a manual valve or a capillary controller could not hold the temperature within ± 3 F. These controls were subsequently replaced by a Honeywell Dialatrol controller (similar to those used in the evaporator and vaporizer water loops) which then provided temperature control within ± 1 F of the set point and allowed all set points in the heat pump rig to be adjusted from the control room.

A liquid level control is used in both the power and cooling loops to provide suitable liquid heads to insure proper operation of the R11 pump and the expansion valve. These controls have been plagued by unstable operation. Initial attempts to resolve this problem included making changes to orifices, plugs and springs in the level control and float chamber valves (see Table XIII). Generally, these initial changes to the level control components and the resulting performance showed the following:

Cooling Loop - The metering orifices appears to be oversized resulting in quick opening of the level control valve as the level rises in the float chamber. The level then drops before the level control system is able to respond to hold the level up in the float chamber. The problem may be solved by using a smaller metering plug.

Power Loop - The level control valve metering plug was reduced in size twice without changing the performance or reducing the freon pump bypass flow. This indicated that the valve is extremely oversized, requiring a metering plug smaller than any available at the time. Installing a smaller plug should cause the flow to stabilize between the vaporizer and the pump bypass so the level control can maintain a fixed liquid level.

Although problems with the two level controllers were not completely resolved during the extended tests, much useful information defining the operating performance of the overall system and its major components was obtained. Since these level controllers were to be used with the system modified for use in the SOLERAS program, further efforts were made to resolve this problem as described in the following paragraphs.

The initial effort to resolve the level control instability problem was to define exactly the nature of the instabilities. Therefore, a dynamic analysis of the two controllers was initiated. Both pilot operated valves are similar in design with the valve motion achieved by varying servo pressure which acts against one side of a piston. On the other side of the piston approximately half the area is exposed to upstream pressure while the remaining half is exposed to downstream pressure. Because of pressure drop in the line downstream of the valve, it was found that there was a net positive feedback effect due to an increase in downstream pressure as the valve is opened. This, in turn, causes the valve to open further, hence the positive feedback. Two things can be done to eliminate

this problem. First, a heavy spring can be installed that would cancel out the positive feedback effect. However, this tends to limit the pressures at which the valve can operate. The second modification is to reduce pressure drop between the valves and the receiver or evaporator. This was implemented by increasing line size and moving the valve to reduce the downstream line length in both the power and cooling loops. Some further changes of valve springs were also considered. However, at that point, those changes had not been tested to verify the predicted improvements in level controller operating stability.

HEAT PUMP SYSTEM DESIGN EVALUATION

Evaluation of the heat pump system and component performance and durability has been conducted throughout the extended test program. Design modifications for future heat pump configurations have been identified and those that might be implemented in a future unit selected for further analysis. A partial listing of factors impacting system performance, reliability, or cost is presented in Table XIV with possible approaches which may be considered to mitigate their negative aspects.

TABLE XIV

POTENTIAL IMPROVEMENTS FOR INCORPORATION IN FUTURE HEAT PUMPS

<u>Factors Impacting Performance, or Reliability or Cost Features</u>	<u>Description</u>	<u>Improvement Approach to be Evaluated</u>
1. Vapor Generator Heat Loss	Tests Indicate 5% heat loss possible.	Add insulation to heat exchanger and isolate unit from mounting
2. Turbine Inlet Heat Loss	Heat loss due to modulating valve and extended piping	Eliminate modulating valve and shorten piping
3. Turbine Volute Heat Loss	Machined aluminum housing increases heat conduction and loss to atmosphere	Use castings made of cast iron
4. Turbine nozzle bypass	Leakage path around turbine inlet possible	Use of brazed plate and elastomer seal to provide better seal
5. Turbine Nozzle Area	Part-load power slightly less than anticipated	Increased nozzle area to provide more power at part-load conditions
6. Turbocompressor Carbon Face Seal	Increased part-load performance required	Review performance and operating capabilities and improve upon seal designs tested to date
7. Evaporator Design	Exchanger tube area exposed and ineffective during some operation resulting in some liquid carryover	Provide more vapor space, eliminate demister and include more heat transfer area in submerged part of exchanger
8. Cooling and Power Condensers	Tests indicate subcooling required to improve feed pump performance	Investigate use of separate subcooler, modification to existing units, or modify water side flow distribution

TABLE XIV (Cont'd)

POTENTIAL IMPROVEMENTS FOR INCORPORATION IN FUTURE HEAT PUMPS

<u>Factors Impacting Performance, or Reliability or Cost Features</u>	<u>Description</u>	<u>Improvement Approach to be Evaluated</u>
9. Cooling and Power Loop Level Controllers	Stable operation sometimes limited for selected conditions	Evaluate alternative level control techniques
10. Bearing Feed System	Alternate systems being considered for SOLERAS	Use SOLERAS technique
11. Freon Feed Pump	Low efficiency, low suction head at selected conditions	Only better subcooling considered and possibly buying available hardware, no development possible
12. Bearing Supply Filter	Introduces P than desired with 7 micron filter	Real system should have low ΔP filter, remove or increase filter size
13. Surge Valve	Adds pressure drop and complexity to heat pump	Consider removal
14. Turbocompressor Manufacture Cost	Use of machined housings and rotors leads to high cost	Current design suitable for casting. Castings for housing should drastically reduce costs of subsequent housings
15. Instrumentation and Packaging	Freon side instrumentation introduces excessive piping and pressure losses, package more towards commercial item	Eliminate unnecessary instrumentation, use T/C castings, streamline design and provide more cooling/power loop separation
16. Regeneration Potential	Slight performance improvement possible with regeneration/intercooling	Evaluate potential and cost
17. Water Loop Treatment	Condensers were fouled and reduced performance	Consider adding inhibitor to closed-water loop

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2. Design, Development and Testing of a Solar-Powered Multi-Family Residential Prototype Turbocompressor Heat Pump. Interim Technical Report. UTRC Report R79-953050-1. Prepared under DOE Contract AC03-77-CS34510.