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LUNAR BASE HEAT PUMP

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PHASE II FINAL REPORT

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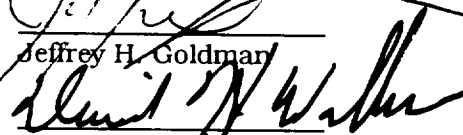
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1. SUMMARY

The criteria established at the beginning of Phase I were followed during Phase II in order to bring the concept to a design ready for fabrication. These criteria were:

1. Lowest possible overall cost including DDT&E, launch, operation, and maintenance.
2. Lowest possible overall system mass and volume, not necessarily the lowest heat pump mass and volume. This translates to the highest heat pump COP at the optimal heat rejection temperature.
3. Safest, environmentally compatible and most readily available working fluid.
4. Highest reliability through simplicity and use of proven off-the-shelf components.
5. Greatest flexibility in development.

At the start of the Phase II program, HCFC-123 was chosen over the Phase I recommended CFC-11 because of environmental concerns related to ground development. This resulted in a similar, but different, system. The LSSIF design resulting from the new system still uses similar technology to the base and lander. It is still packaged on an open pallet to allow frequent component changes, and still uses commercial off-the-shelf components to allow less expensive development and evolution toward the flight system.

A general system schematic of the LSSIF high lift heat pump system is shown in Figure 1 and Foster-Miller drawing NAS9614-1003. The heat pump is a two-stage system utilizing HCFC-123 as the working fluid/refrigerant. Stage one uses three Trane scroll compressors and stage two uses four Mars rotary compressors. The heat exchangers are IIT standard industrial grade plate-fin type. All hardware is commercial or industrial grade to control cost and to take advantage of terrestrial experience.

The overall layout of the high lift (Lunar Base) heat pump is shown in Figure 2 and Foster-Miller drawing NAS9614-1001. The mechanical package sits on a steel and aluminum pallet with an aluminum, forced ventilation, enclosure. Overall dimensions are approximately 120 cm wide x 183 cm long and 120 cm high. The package layout is intentionally open to allow easy access to all components in order to facilitate development. The first-stage and second-stage compressors are located at the opposite ends of the skid to permit the easiest access during package development. Heat exchangers are inboard, as it is expected they will require less access during package development. Critical control valves are also kept near the skid extremities to facilitate maintenance and/or replacement.

The package enclosure prevents damage to the heat pump components and protects personnel from accidentally touching hot components (i.e., compressor discharge manifolds) during operation. The ventilation fan maintains temperature within the enclosure.

Mechanical systems are described and specified in the Mechanical Systems Description and Specification, Document NAS9614-1.

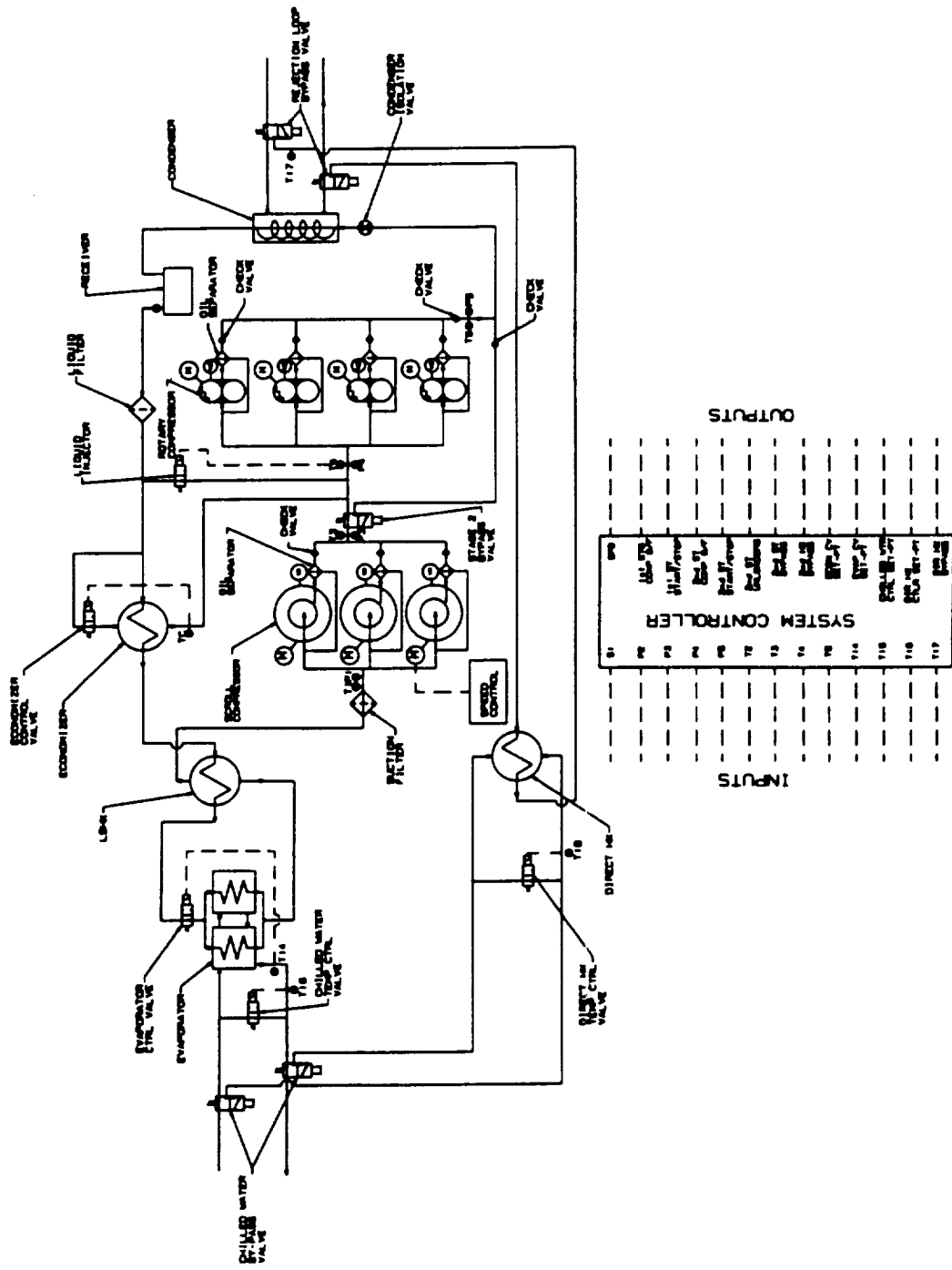


Figure 1. System schematic of the LSSIF high temperature lift heat pump

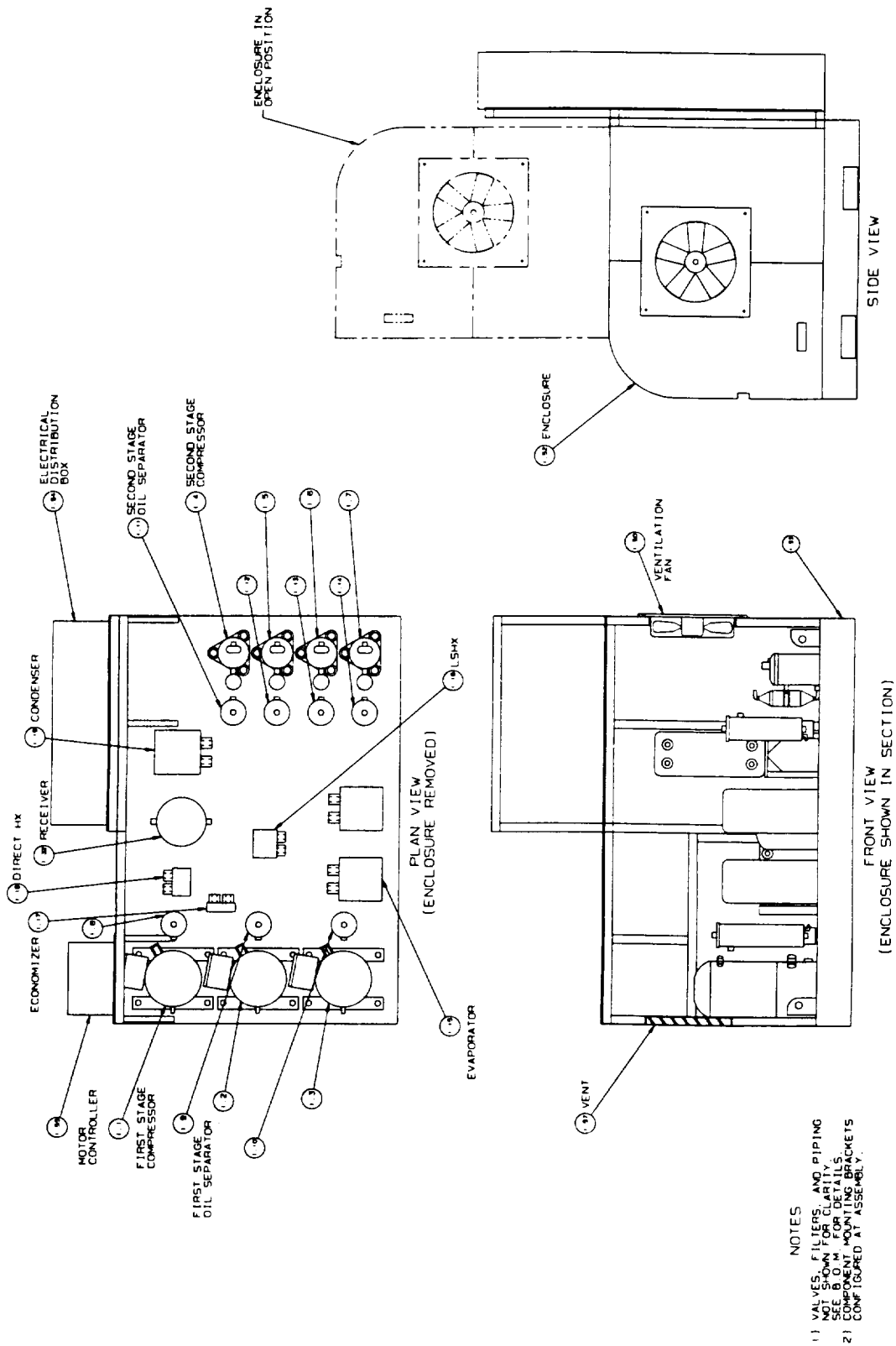


Figure 2. Package layout of the LSSIF high temperature lift heat pump

Data acquisition and control system measurements are described and specified in the Control and Data Acquisition Systems Description and Specification, Document NAS9614-2.

Electrical systems are described and specified in the Electrical Systems Description and Specification, Document NAS9614-5.

2. INTRODUCTION

A heat pump is a device which elevates the temperature of a heat flow by means of an energy input. By doing this, the heat pump can cause heat to transfer faster from a warm region to a cool region, or it can cause heat to flow from a cool region to a warmer region. The second case is the one which finds vast commercial applications such as air conditioning, heating and refrigeration. Aerospace applications of heat pumps include both cases.

In the first case, when heat can be rejected directly to space by radiation, a heat pump can reduce the amount of radiator area required by elevating the rejection temperature. If the heat pump can be operated efficiently enough that the radiator weight savings is more than the extra weight of the heat pump and its power system, then there is a net benefit. Since radiator weight savings go up and heat pump performance goes down with increasing temperature lift, there is a tradeoff in deciding how high to raise the heat rejection temperature.

Some aerospace applications fall into the second case where heat is rejected (often by radiation) to a surrounding thermal environment which is warmer than the area to be cooled. One example of this is a lunar base habitat near the moon's equator. Although NASA has sent humans to the moon before, they were never sent for extended periods of time. The Apollo missions were relatively short in duration and were conducted during the early part of the lunar day/night cycle. Hence the Apollo lunar module Thermal Control Systems (TCS) could reject waste heat via evaporative cooling through a water boiler. During future missions to the moon, or to other planets, the crew and support equipment will be exposed to more severe thermal environments for longer periods of time. Therefore, using a consumable fluid such as water for thermal control will no longer be feasible. A heat pump of some type must be used to enable rejection of moderate temperature waste heat to these more severe environments. For example, a lunar base TCS will collect waste heat from the crew habitat at a temperature of about 275K. That waste heat must then be elevated to a temperature above that of the effective thermal sink temperature (i.e., local environment) in order to enable its rejection. For a lunar base near the equator, this will require a heat pump with a relatively high temperature lift.

The NASA Johnson Space Center is currently developing a Life Support Systems Integration Facility (LSSIF, previously the SIRF) to provide system-level integration, operational test experience, and performance data that will enable NASA to develop flight-certified hardware for future planetary missions. A high lift heat pump is a significant part of the TCS hardware development associated with the LSSIF.

The high lift heat pump program discussed here is being performed in three phases. In Phase I, the objective is to develop heat pump concepts for a lunar base, a lunar lander and for a ground development unit for the SIRF. In Phase II, the design of the SIRF ground test unit is being performed, including identification and evaluation of safety and reliability issues. In Phase III, the SIRF unit is manufactured, tested and delivered to the NASA.

The following actions were taken to meet the Phase I objectives and to provide adequate data to select the optimum heat pump concept.

1. Conduct a systems optimization study to determine the rejection temperature that minimizes the overall power generation and heat rejection system mass, not necessarily the heat pump's mass.
2. Conduct a tradeoff study of refrigerants and thermodynamic cycles to determine which combination yields the highest $COP_{Cooling}$ at the optimal heat rejection temperature.
3. Research and select the major components, compressor and heat exchangers, that can be used to implement the thermodynamic cycle selected. Special attention was paid to using the same technologies for the SIRF and flight heat pumps.
4. Generate a package concept for the components selected, including mass and volume estimates.
5. Identify the technology similarities and differences between the SIRF and flight units in order to identify the steps necessary to evolve to flight configuration.

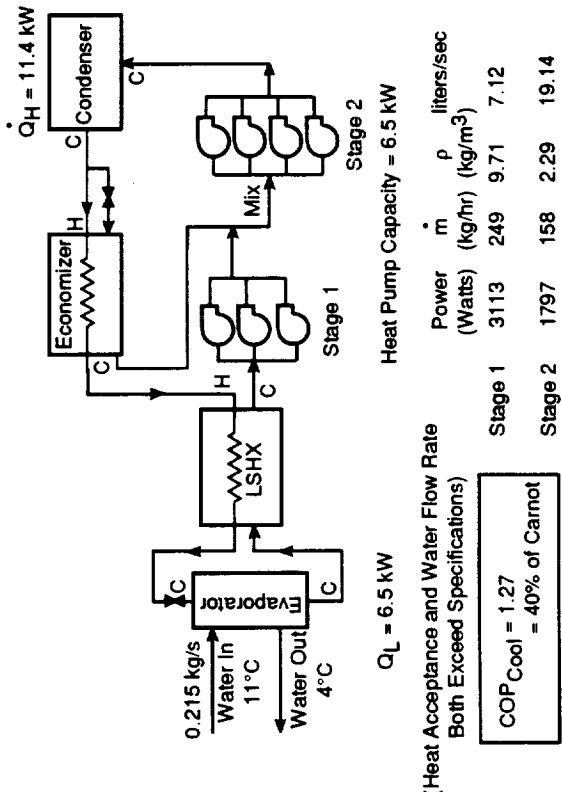
Performing the above steps led us to recommending a CFC-11 refrigerant based system as noted in the Phase I Final Report.

During Phase II the system was redefined based on the HCFC-123 refrigerant because of environmental concerns and regulations of the ground development (LSSIF) unit. A detailed design ready for fabrication was produced during Phase II based on this new system. The results of this design effort are outlined in this report.
HCFC-123 System Definition

3. HCFC-123 SYSTEM DEFINITION

The system was redesigned based on HCFC-123. The process used to define the new system was identical to the process used during Phase I and outlined in the Phase I Final Report. Figure 3 shows the HCFC-123 System State Points and Figure 4 shows the Pressure-Enthalpy Diagram for this system. Changing refrigerant to HCFC-123 resulted in Coefficients of Performance of 1.58 (50 percent of Carnot) for the flight unit, and 1.27 (40 percent of Carnot) for the LSSIF unit. This represents a performance penalty of only 5 to 6 percent for the flight unit, using optimally designed components for this application. However, there are fewer available commercial components for HCFC-123, and the most suitable ones will still result in a 15 percent performance penalty, from CFC-11, for the ground based prototype.

In addition to performance penalties, HCFC-123 created other concerns. These included safety issues, because HCFC-123 has lower acceptable exposure limits, greater development risk, because there is less industrial experience, and greater development costs, because there are fewer commercial components available.

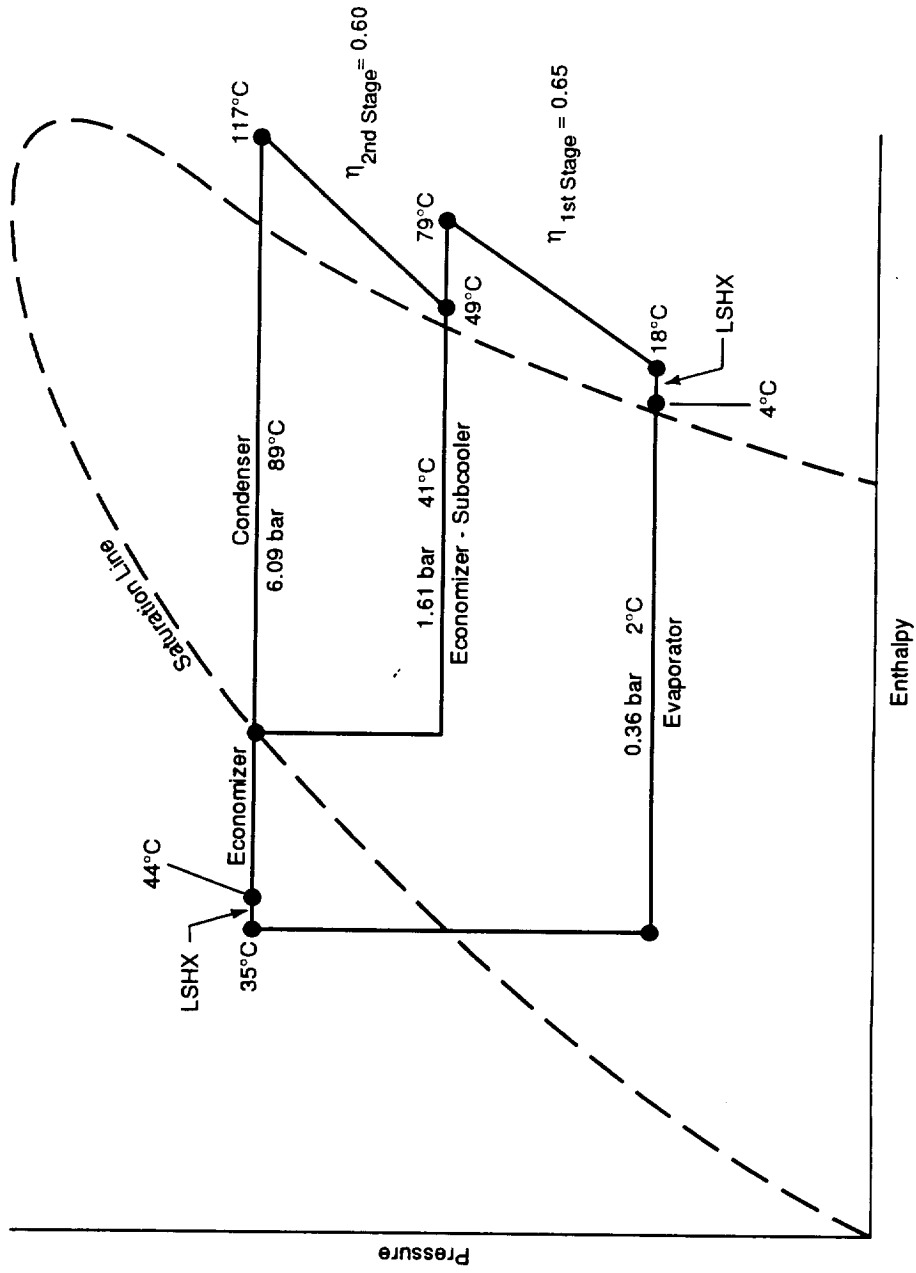


	Inlet Hot			Outlet Hot			Inlet Cold			Outlet Cold		
	P (bar)	T (°C)	\dot{m} (kg/hr)	P (bar)	T (°C)	\dot{m} (kg/hr)	P (bar)	T (°C)	\dot{m} (kg/hr)	P (bar)	T (°C)	\dot{m} (kg/hr)
EVAP							0.37	2	158	0.36	4	158
LSHX	5.85	44	158	5.61	35	158	0.36	4	158	0.34	18	158
Economizer	6.09	89	158	5.85	44	158	1.68	41	91*	1.61	44	91*
Mix				1.61	49	249						
Condenser	6.34	117	249	6.09	89	249						

* The Economizer Cold Side Massflow Rate Consists of 63 kg/hr for Subcooling Liquid on its Way to the Evaporator and 28 kg/hr Overfeed (or Liquid Injection) to Desuperheat The Second Stage Intake.

A931-2

Figure 3. HCFC-123 LSSIF system state points



A931-1

Figure 4. HCFC-123 LSSIF system pressure-enthalpy diagram

4. MECHANICAL SYSTEMS

The system schematic of the LSSIF high lift heat pump system is shown in Figure 5 and Foster-Miller drawing NAS9614-1003. The heat pump is a two-stage system utilizing HCFC-123 as the working fluid/refrigerant. Stage one uses three Trane scroll compressors and stage two uses four Mars rotary compressors. The heat exchangers are ITT standard industrial grade plate-fin type. All hardware is commercial or industrial grade to control cost and to take advantage of terrestrial experience.

The overall layout of the high lift (Lunar Base) heat pump is shown in Figure 6, Foster-Miller drawing NAS9614-1001 and Foster-Miller parts list NAS9614-100. The mechanical package sits on a steel and aluminum pallet with an aluminum, forced ventilation, enclosure. Overall dimensions are approximately 120 cm wide x 183 cm long and 120 cm high. The package layout is intentionally open to allow easy access to all components in order to facilitate development. The first-stage and second-stage compressors are located at the opposite ends of the skid to permit to easiest access during package development. Heat exchangers are inboard, as it is expected they will require less access during package development. Critical control valves are also kept near the skid extremities to facilitate maintenance and/or replacement.

The package enclosure prevents damage to the heat pump components and protects personnel from accidentally touching hot components (i.e., compressor discharge manifolds) during operation. The ventilation fan maintains temperature within the enclosure.

4.1 Compressors

The first stage consists of three Trane CSH3-093 scroll-type compressors. The second stage consists of four Mars MDB240211A rotary piston compressors. The minimum number of compressors required in each stage is determined by the design mass flowrate based on comparisons of capacity charts and expected operating conditions. These are best estimates as actual capacity charts for R-123 do not yet exist for most commercially available compressors. Both scroll and rotary piston machines are used because they are less susceptible to damage from liquid slugging or wet compression compared with other compressor types. At least three compressors per stage are required to provide reasonable redundancy and control for a ground test system, and to conform to standard commercial practice. Redundancy requirements for a space or planetary based system will not be met by this system, as it is expected all seven compressors will be required for operation at the maximum design load. However, off-design conditions should permit selective compressor isolation if maintenance or repair is required.

Three first-stage compressors are actually used to meet the minimum flow requirements, the redundancy requirements for commercial level reliability, and the flow variability to allow reasonable load following over the entire operating range. One of the first-stage compressors is capable of variable speed control. The other two require on/off operation only. None of the first-stage compressors requires unloading capability, neither partial nor full. The compressor control scheme is described in a separate section of this report.

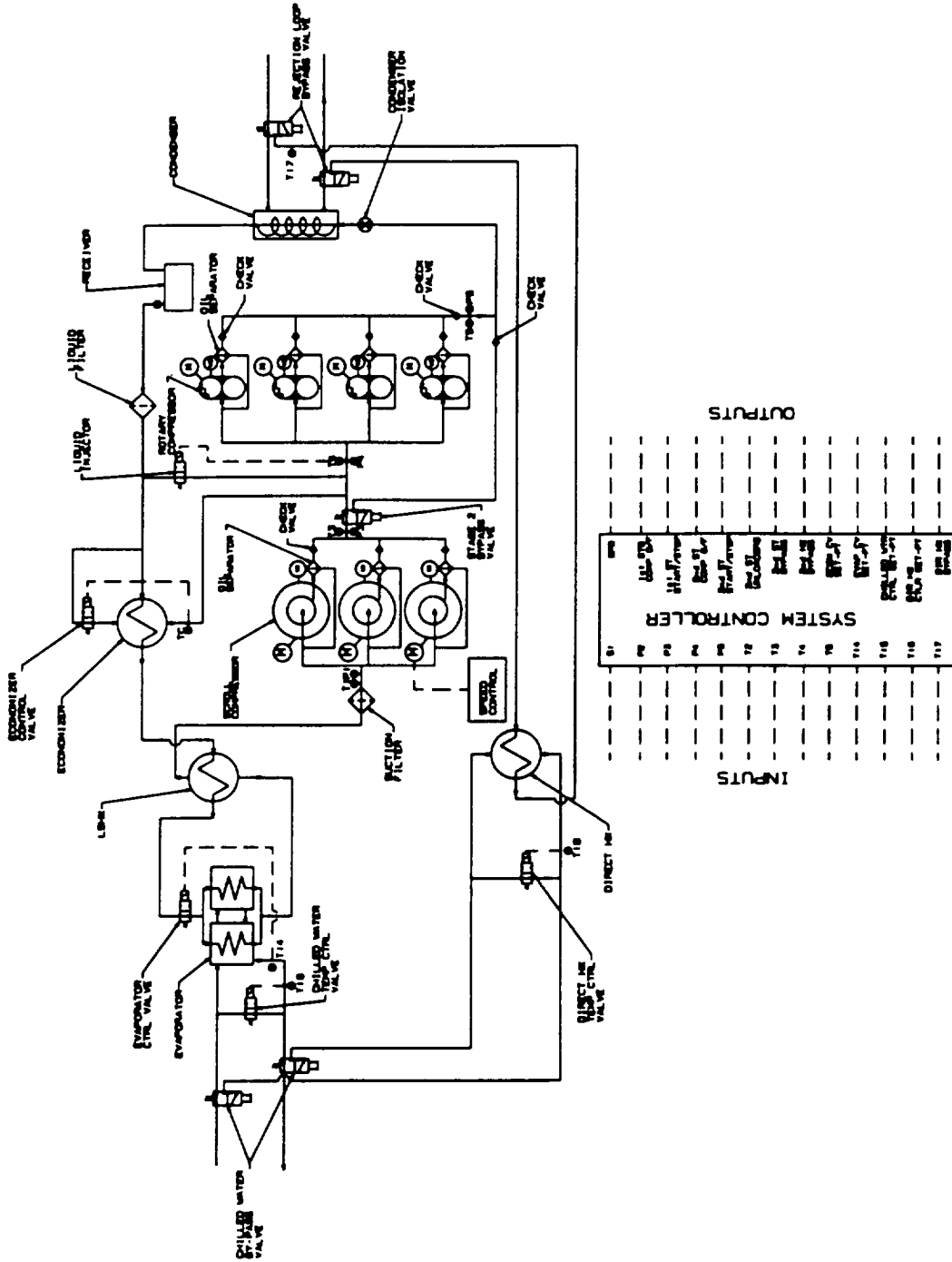


Figure 5. System schematic of the LSSIF high temperature lift heat pump

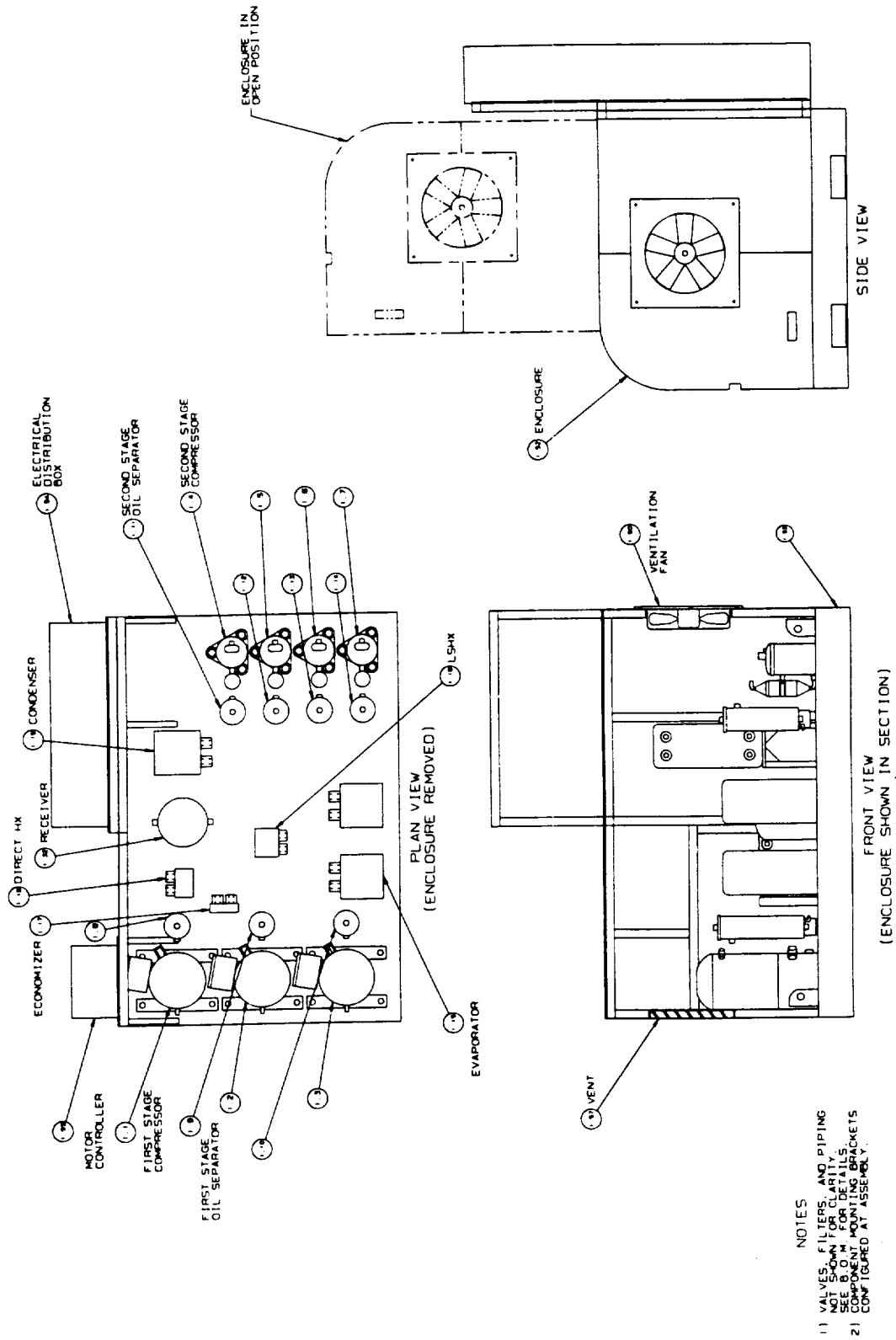


Figure 6. Package layout of the LSSIF high temperature lift heat pump

Four second-stage compressors are used to meet the minimum flow requirements, the redundancy requirements for commercial level reliability, and the flow variability to allow reasonable load following over the entire operating range. Each compressor is either on or off. No unloading or speed variation is required with the four compressors. Four compressors operating in on/off mode will yield 25 percent load increments of the second-stage.

Each first stage compressor, operates on 230 VAC, 3 phase power, and each second stage compressor operates on 120 VAC single phase power.

Each compressor, first and second stage, has "HOT" warning signs and protective guards at their discharges to avoid personnel burn hazards.

4.2 Oil Separation and Separators

Each compressor discharge has its own oil separator. These are commercial grade, cyclone type similar to the Simons 5000 Series of separator. Each separator has its own return lines to its corresponding compressor sump. The first and second stage separators are not interchangeable due to the different flow and pressure ratings.

4.3 Compressor Sump Heaters

Each compressor has a sump heater in order to prevent refrigerant condensation when it is shut down. Condensation can cause liquid slugging and excess power surges upon compressor start-up. Each heater is commercial grade, wrap around type of at least 50 Watts rating, similar to the Mars model 3240 sump heater.

4.4 Heat Exchangers

Five heat exchangers are used in the LSSIF high-lift heat pump system as shown in Table 1. They are, or are equivalent to, commercial Holt (ITT Standard) heat exchangers, all of similar construction, using brazed parallel plates to maximize heat transfer while minimizing size, weight and cost.

Table 1. Heat exchanger descriptions

Heat Exchanger	Heat Transfer Capacity (BTU/hr)	Type	Fluids Hot Side	Fluids Cold Side
1. Evaporator (1.15)	22,185 BTU/hr 6.5 kW	Parallel Plate	H ₂ O (15% Glycol)	R-123
2. LSHX (1.16)	1,380 BTU/hr 0.40 kW	Parallel Plate	R-123	R-123
3. Economizer (1.17)	6,995 BTU/hr 2.05 kW	Parallel Plate	R-123	R-123
4. Direct (1.18)	22,185 BTU/hr 6.5 kW	Parallel Plate	H ₂ O (15% Glycol)	H ₂ O (50% Glycol)
5. Condenser (1.19)	38,942 BTU/hr 11.41 kW	Parallel Plate	R-123	H ₂ O (50% Glycol)

Capacity numbers shown in Table 1 represent the maximum design heat transfer rates expected. To ensure the design point of 5 kW of cooling could be met, each heat exchanger was oversized by approximately 30 percent to account for system development uncertainty. The direct heat exchanger was sized to provide 6.5 kW of cooling when the water inlet temperature is at or below 35°F.

The evaporator is a Holt, ITT Standard PS600-60, plate-fin heat exchanger, or equivalent. The evaporator is single phase water on one side and two-phase HCFC-123 on the other side. It absorbs heat from the chilled water loop, that simulates the habitat cooling loop, and transfers it to the refrigeration cycle, boiling the refrigerant in the process.

The liquid suction heat exchanger (LSHX) is a Holt, ITT Standard PS400-32, plate-fin heat exchanger, or equivalent. It is a liquid to vapor heat exchanger. It transfers heat from the condensate entering the evaporator to first-stage compressor suction vapor. This subcools the condensate to increase evaporator efficiency and superheats the suction vapor to prevent liquid from entering the compressors.

The economizer heat exchanger is a Holt, ITT Standard PS400-10, plate-fin heat exchanger, or equivalent. It is a single phase liquid to two-phase refrigerant “flash” evaporator. Refrigerant condensate is routed from its loop, undergoes a pressure drop through the economizer control valve, flashing to vapor, and is injected into the second-stage suction stream. This cools the second-stage suction vapor to prevent over heating of the second-stage compressor motor windings.

The condenser is a Holt, ITT Standard PS600-64, plate-fin heat exchanger, or equivalent. The evaporator is single phase water (50 percent Glycol) on one side and two-phase HCFC-123 on the other side. It transfers heat from the HCFC-123 vapor, condensing it in the process, to the rejection loop.

The direct heat exchanger is a Holt, ITT Standard PS400-14, plate-fin heat exchanger, or equivalent. It is a liquid to liquid heat exchanger that transfers heat directly from the chilled water loop to the rejection loop when vapor compression heat pumping is not required.

4.5 Check Valves

Each compressor has a check valve at its discharge to prevent high pressure back flow when shutdown. The second-stage branch of compressors also has a separate check valve to prevent back flow into the stage when it, only, is shutdown. This check valve is redundant with the four compressor discharge check valves. The second-stage bypass line has a check valve to prevent back flow to the first-stage. All check valves are commercial grade.

4.6 Filters

There is a vapor filter at the suction to the first-stage compressors and a liquid filter at the receiver discharge.

These filters are of commercial grade and their allowable particle size will be determined after the compressors, valves and heat exchangers have been purchased.

4.7 Receiver

A liquid refrigerant receiver is located downstream of the condenser. Its purpose is to store high pressure condensed refrigerant, and is used specifically to accommodate different operating charges in the evaporator and condenser during varying load conditions. The

receiver comes with its own outlet isolation valve. This valve used in conjunction with the condenser isolation ball valve is used to isolate the refrigerant in the high pressure side of the heat pump, and away from most major components, to allow maintenance on the major components without removing the refrigerant charge.

4.8 Bypass Valves

Three way, two position solenoid operated valves are placed in the chilled water supply and return lines to divert chilled water from the evaporator to the direct heat exchanger. These valves are commercial grade and allow full system flow to either the evaporator or direct heat exchanger. They also isolate chilled water flow to the bypassed heat exchanger. These valves and their function work in unison with the condenser rejection loop working fluid bypass valves so that chilled water and rejection loop working fluid flows will be either fully to the direct heat exchanger or fully to the evaporator or condenser.

Three way, two position solenoid operated valves are placed in the rejection loop supply and return lines to divert the rejection loop working fluid from the condenser to the direct heat exchanger. These valves are commercial grade and allow full system flow to either the condenser or direct heat exchanger. They also isolate rejection working fluid flow to the bypassed heat exchanger. These valves and their function shall work in unison with the evaporator chilled water bypass valves so that chilled water and rejection loop working fluid flows will be either fully to the direct heat exchanger or fully to the evaporator or condenser.

A three way, two position solenoid operated valve is used to divert first stage discharge gas around the second stage compressors when they are not required.

4.9 Control Valves

All control valves are pulse width modulated type, which are two position (open/closed) valves, with flow control achieved by varying the length of time the valve is open. Each valve will perform its control function independently based on a monitored outlet temperature and local controller. However, the temperature set-point will be provided by the central system controller.

The controllers modulating refrigerant flow into the evaporator and condenser will adjust flow rate depending on outlet temperatures; approximately 5°F of superheat will be maintained. The actual temperature will depend on heat exchanger pressure.

The economizer control valve regulates the “flashing” of refrigerant condensate to the second-stage compressor suction in order to control second-stage compressor temperatures to within safe operating limits, protecting motor windings and avoiding lubricate breakdown. This process is supplemented by the liquid injection system.

The liquid injection control valve controls a relatively small liquid flow from the relatively cool liquid return line to be injected into the first-stage discharge/second-stage suction line. This liquid flow mixes with the first-stage vapor, evaporating, to cool the second-stage suction. Cooling the second-stage suction helps control second-stage compressor temperatures to within safe operating limits, protecting motor windings and avoiding lubrication breakdown. The liquid injection has an adverse effect on heat pump COP and is only used intermittently when economizer flow is inadequate to control second-stage suction temperature.

The chilled water loop control valve bypasses water around the evaporator to maintain a post-mixed stream temperature between 38 to 40°F. If the control valve cannot maintain the desired temperature range, it will:

1. Divert all water through the evaporator if the outlet temperature is greater than 40°F.
2. Completely bypass the evaporator if the outlet temperature is 37°F or less.

The direct heat exchanger control valve operates primarily to prevent freeze-up of the chilled water loop. It will also attempt to control the temperature of the chilled water to within 38 to 40°F, but will bypass rejection water around the direct heat exchanger in order to do so. This is because rejection water can be as low as 17°F, and, if chilled water is bypassed, the lower chilled water flowrates, coupled with very low temperatures in the rejection water loop, could present a freezing condition. In the event that the valve cannot maintain the desired temperature, it will:

1. Divert all rejection water to the direct exchanger if the chilled water outlet temperature is greater than 40°F.
2. Completely bypass the direct heat exchanger if the outlet temperature is 37°F or less.

The evaporator control valve maintains the refrigerant flow and provides the refrigerant pressure drop (expansion) that creates the cooling refrigerant phase change.

4.10 Isolation Valves

Isolation valves at the exit of the receiver and inlet of the condenser are supplied in order to isolate the refrigerant charge from the rest of the system to allow maintenance.

4.11 Enclosure

The enclosure, shown in Figure 7, is two pieces. One piece is the back wall that supports the electrical enclosures. It is removable with bolts or equivalent fastener. It does not need to be removable with the electrical panels in place.

The second piece covers the remaining three sides and is hinged at the top of the back wall for one man lifting. Lifting is assisted via gas springs to limit the lifting load to 30 pounds.

Enclosure construction will be aluminum frame (3003 or 6061-T6) and .060 aluminum sheet (3003) covering. Fastening will be by welding, riveting, and/or bolting, in any combination. It does not overlap the skid base except for its lifting handle and its lift assisting devices (air springs).

The electrical enclosures are commercial grade steel with commercial type cable pass through points and component mounting.

4.12 Enclosure Ventilation

The enclosed volume of the heat pump will be ventilated with ambient air. The ventilation fan flow rate is at least 2000 CFM at 0" standard pressure. The fan is thermostatically controlled to come on at 80°F (300K) increasing and shut off at 75°F (297K) decreasing. Temperature control is variable as this may need adjusting when a correlation between ambient temperature, heat pump load and enclosure temperature is established. Control temperature is taken centrally and not on an enclosure wall.

The fan is mounted in the end of the liftable portion of the enclosure. Its diameter is not limited, however, its overall dimensions will not exceed the end dimensions of the enclosure. Fan weight is not restricted as long as the enclosure does not distort to an unacceptable degree. An acceptable distortion is any distortion that still allows opening the enclosure. Its weight must also be compensated for in the enclosure lifting limit.

4.13 Skid

The skid will be capable of supporting 1500 pounds by fork lift lifting points. Its construction is mild steel and/or aluminum and fastening is by welding or bolting. Its foot print is 120 cm by 182 cm. Figure 8 shows the skid details.

The mechanical components, including manufacturer and part number, are listed in Table 2.

Table 2. Mechanical components

Item No.	Nomenclature	Part No.	Manu- facture
1-3	First stage scroll compressor	CSH3-093	Trane
4-7	Second stage rotary compressor	MDB240211A	Mars
8-10	First stage oil separator	S5185	AC&R
11-14	Second stage oil separator	S5182	AC&R
15	Evaporator		Standard
16	Liquid suction heat exchanger		
17	Economizer		
18	Direct heat exchanger		
19	Condenser		
20	Suction filter	C-4313-S-T-HH	Sporlan
21-23	Stage 1 check valve	803B-10S	Superior
24-27	Stage 2 check valve	802B-8S	Superior
28	Stage 2 bypass valve	12Z-B8X-JBP	Parker
29	Stage 2 check valve	805C-14S	Sporlan
30	Condenser isolation ball valve	590-14ST	Superior
31	Second stage bypass check valve	805C-14S	Sporlan
32	Receiver (with valves)	UR-20	Standard
33	Liquid filter	C-084-S	Sporlan
34	Desuper heater valve	Tbd	Valcor
35	Economizer temperature control valve and actuator	Tbd	ETI
36	Evaporate temperature control valve and controller	Tbd	ETI
37-38	Chiller water bypass valves	12F34C2140APF	Parker
39-40	Rejection loop bypass valve	12F34C2140ADF	Parker
41-47	Compressor sump heaters	3240	Mars
48	Chilled water temperature control valve	08F20C2128ADR	Parker
49	Direct heat exchanger temperature control valve	08F2DC2128ADR	Parker

5. MATERIALS

The LSSIF heat pump uses commercial grade components used and therefore its materials are determined by the component manufacturer's experience. Many component materials are proprietary to manufacturer who will not supply data. There is also little need to know the material for most components. Where material is critical, such as in the compressors, a sample unit will need to be cut apart and analyzed following limited endurance testing. Therefore, a selected compressor (hottest) should be cut apart after limited endurance testing to verify lubrication and wear and to analyze materials.

Heat pump frames and brackets will be machined Aluminum 3003, 6061-T6. The plumbing will be brazed stainless steel. Braze and weld processes will be to commercial standards. The skid will be of mild steel (machined, welded), and Aluminum 6061-T6 (machined, bolted). The enclosure will be aluminum 3003 (formed, riveted).

6. ELECTRICAL SYSTEMS

The electrical system schematic is shown in Figure 9.

6.1 Electrical Distribution Box or Enclosure

The electrical distribution box or enclosure is commercial grade and made of steel, plastic (PVC) or aluminum. It will be 152 cm high, 91 cm wide, 25 cm deep. The layout of the components inside the box will be determined during assembly.

6.2 Motor Starters and Drives

All motor starters and drives are commercial grade. The two on/off first-stage compressor starters are size 2, 15 hp, 230 VAC, 3-phase. The one variable speed first-stage compressor is driven by a variable frequency drive, 15 hp, 230 VAC, three-phase. The four second-stage compressor (on/off) starters are size 00, 15 hp at 120 VAC, single-phase.

6.3 Circuit Breakers

There is one circuit breaker of commercial grade and it is three-pole, 300 VAC, 100 Amp.

6.4 Power Distribution Terminal Block and Terminals

There is one power distribution terminal block with commercial grade terminals rated at 300 VAC, 100 Amp or greater.

6.5 Push Buttons

There are 20 commercial grade push buttons rated at 300 VAC, 100 Amp unless used to actuate a relay. When used to actuate a relay, their rating is equal to or greater than the actuation voltage and current used to actuate the relay.

6.6 Indicator Lights

There are ten commercial grade indicator lights. Their wattage and voltage ratings will be determined at assembly.

6.7 Independent Temperature Controllers and RTDs

There are five commercial grade independent temperature controllers, with RTDs. Each is a Phasetronics Series EP1, single-phase, SCR power control, pulse-width modulated, phase angle controller, or equivalent. They are compatible with the General Electric Fanuc Model 90-30 controller and will receive temperature set-point either from that controller or by local manual input directly to the controller.

Each RTD is compatible with the working fluid in which it will reside. The fluids are:

SEE SHEET 1

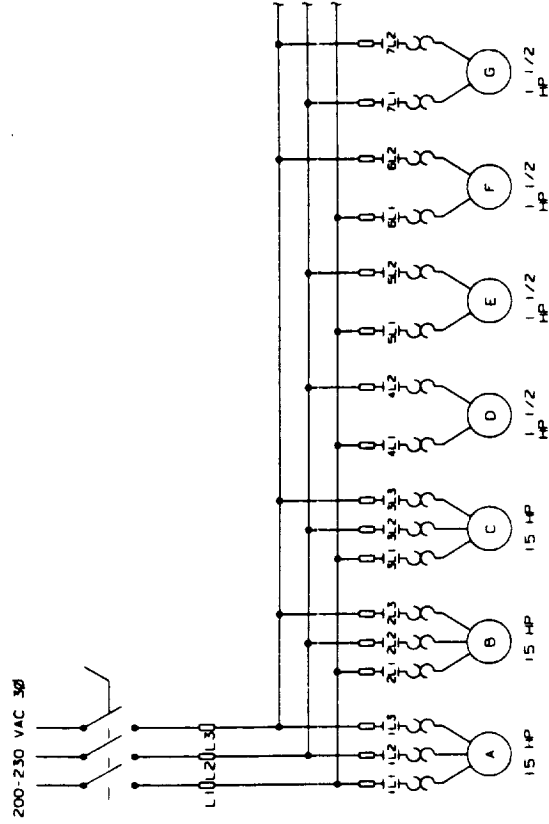
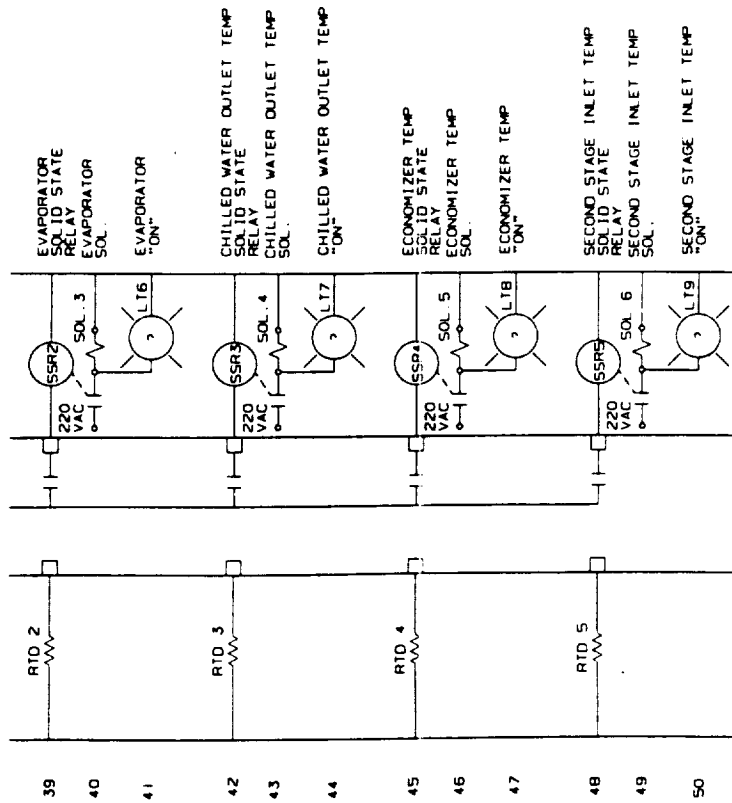


Figure 9. LSSIF heat pump electrical schematic

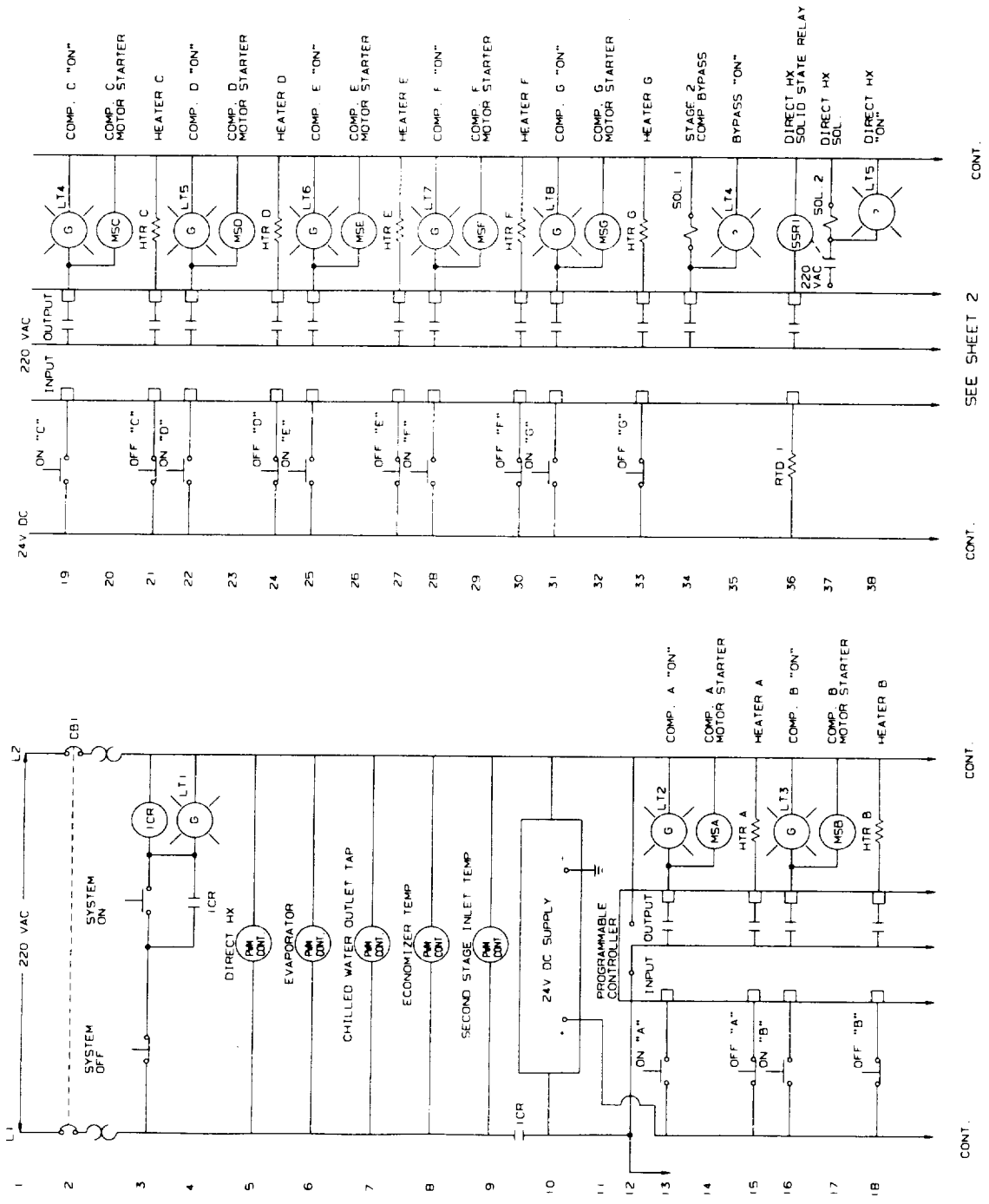


Figure 9. LSSIF heat pump electrical schematic (continued)

- HCFC-123 for the refrigerant loop.
- Water (50 percent Glycol) for the rejection loop.
- Water (15 percent Glycol) for the chilled water loop.
- Water for the chilled water loop.

Each RTD's temperature range is compatible with its individual function. The functions are:

- Economizer control valve regulation.
- Liquid injection control valve regulation.
- Chilled water loop control valve regulation.
- Direct heat exchanger control valve regulation.
- Evaporator control valve regulation.

6.8 Solid-State Relays

There are five commercial grade solid-state relays. Their ratings will be determined when the major electrical load components (i.e., compressors) are received. This will ensure that they are compatible with the eventual component, in case substitutions are required.

6.9 Wiring, Harnesses, Connectors

Wiring, harnesses and connectors are commercial grade and their ratings are consistent with the components to be interconnected.

6.10 System Controller

The system controller is a General Electric Fanuc Model 90-30 controller. The controller functions are discussed in more detail in another section of this report.

7. CONTROL SYSTEM

The overall control scheme is shown in Figure 10. Figure 11 shows the location of the instrumentation. Appendix A includes the detailed control diagram.

7.1 Major Control Parameters

The control system for the two stage high-lift heat pump will control the parameters shown in Table 3.

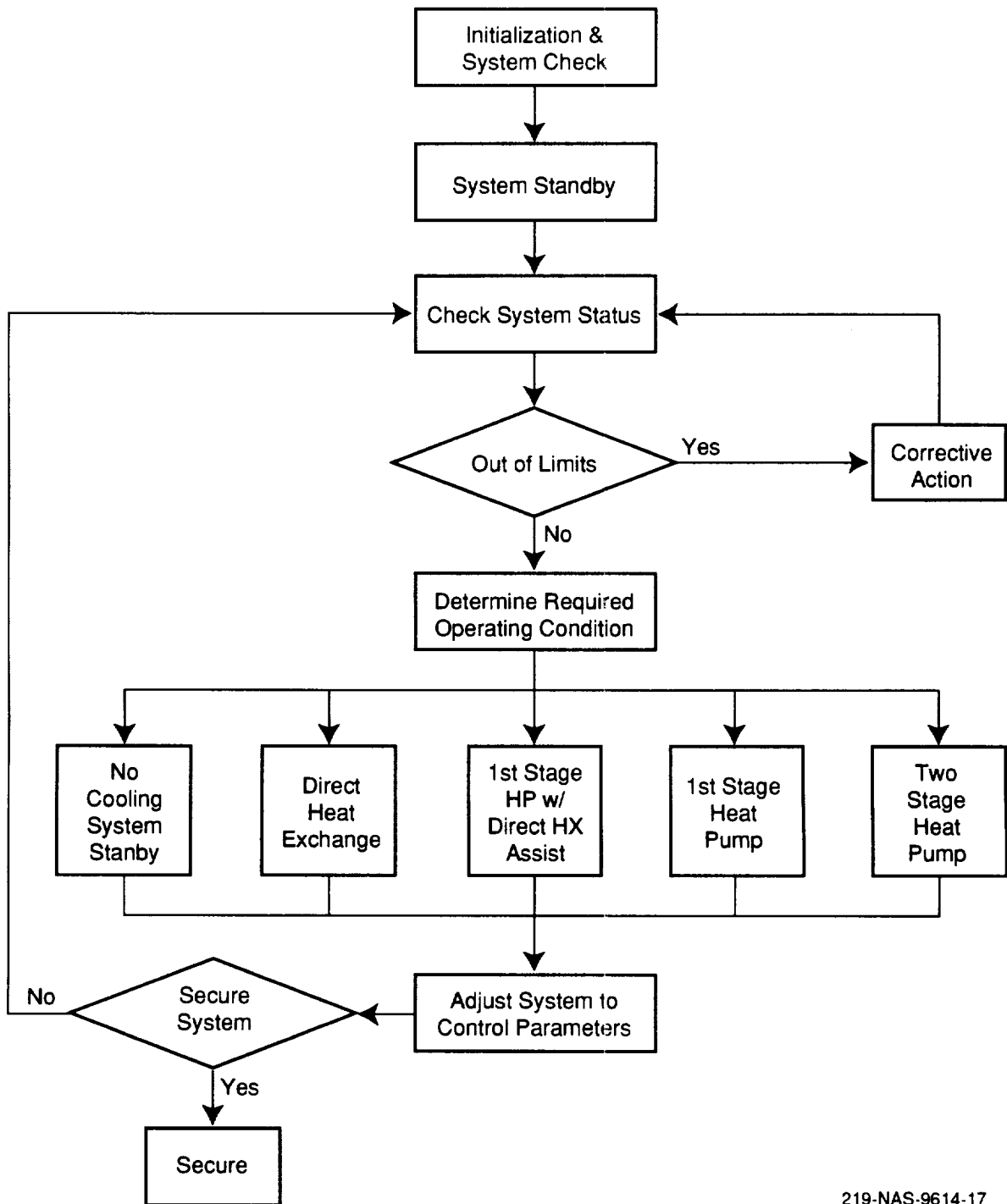
7.2 Primary Control Logic

The high lift heat pump will be designed to operate in one of the following five states:

1. CHILLED WATER INLET TEMPERATURE $< 40^{\circ}\text{F}$ - No cooling is required. Therefore, both direct cooling and the heat pump will be secured. The heat pump system is in "STANDBY" status. The thermal transport loop is probably bypassed or in standby as well.
2. CHILLED WATER INLET TEMPERATURE $> 40^{\circ}\text{F}$, REJECTION LOOP TEMPERATURE $< 35^{\circ}\text{F}$ - The direct cooling mode can provide the required cooling load, and will be used alone. Heat pump in "STANDBY". The thermal transport loop must be active.
3. CHILLED WATER INLET TEMPERATURE $> 40^{\circ}\text{F}$, REJECTION LOOP TEMPERATURE $35^{\circ}\text{F} - 45^{\circ}\text{F}$ - Direct cooling mode can be used to assist heat pump (to minimize power consumption), but the first stage is used to provide the additional cooling required.
4. CHILLED WATER INLET TEMPERATURE $> 40^{\circ}\text{F}$, REJECTION LOOP TEMPERATURE $45^{\circ}\text{F} - 100^{\circ}\text{F}$ - No assist from direct cooling, but first stage is expected to be adequate. Second stage compressors and will be off and bypassed.
5. CHILLED WATER INLET TEMPERATURE $> 40^{\circ}\text{F}$, REJECTION LOOP TEMPERATURE $> 100^{\circ}\text{F}$ - Two stages will be required. (Note: A 10 degree overlap will be used to determine the cut-out and cut-in temperatures for the second stage to prevent cycling. Additionally, these temperatures may be altered due to testing, or if it is found the system operates more effectively with both stages operating at lower rejection temperatures.)

It is expected that the system will start in state 1, then transition through each state to state 5 as the rejection water temperature increases. The overall control logic diagram is shown in Figure 10. Although it is expected that the states will occur sequentially (from 1 to 5, then back to 1), the main control loop will handle state transitions in any order. In addition, it will be capable of startup in any state. A more detailed control logic diagram is found in Appendix A.

The control system may need to be designed to calculate refrigerant state-points so that conditions potentially hazardous to the equipment can be flagged. These include suction gas



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Figure 10. LSSIF heat pump overall control scheme

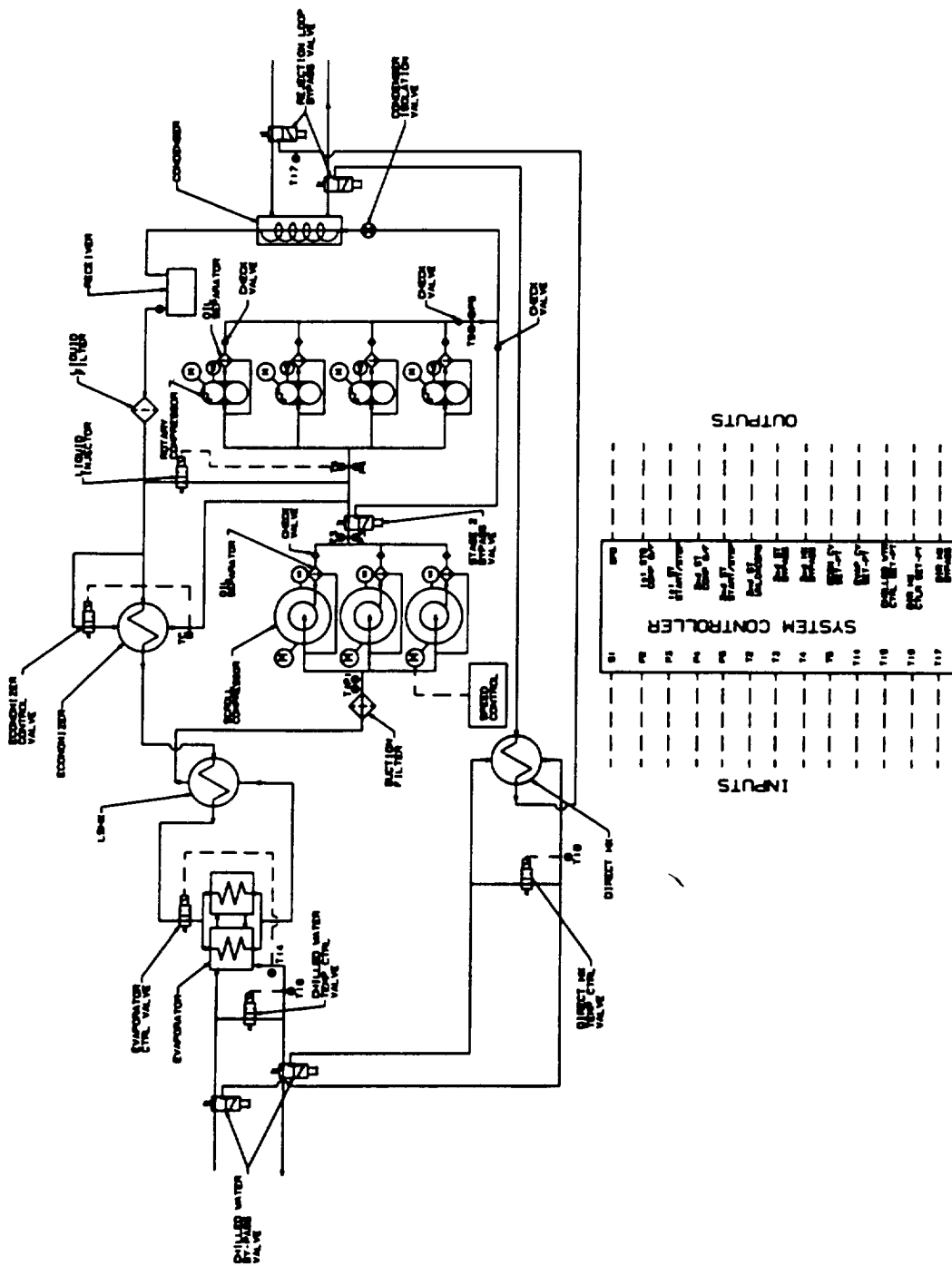


Figure 11. LSSIF heat pump instrumentation

Table 3. Parameter controls

Parameter	Method of Control
1. Chilled Water Return Temp (38°F to 40°F) Setpoint adjustable by controller	By independent bypass around the evaporator: 1. If the outlet temperature is greater than 40°F - A Pulse-Width Modulated valve will remain closed, sending all chilled water through the evaporator 2. As the outlet temperature drops below 40°F - Flow will begin bypassing the evaporator. The flow will be split, then remixed, with the bypass amount proportional to the amount required to maintain 38°F to 40°F 3. When the outlet temperature drops below 38°F - It will completely bypass the evaporator
2. Evaporator Temperature, Refrigerant Side (34°F to 36°F)	By holding first stage compressor suction pressure fixed. Temperature will correspond to the saturated suction pressure. There are three first stage compressors, two on/off controls, and one variable speed control: 1. As suction pressure drops - first stage capacity will be reduced by slowing the variable speed compressor, or shutting off one of the other two compressors, as shown in the control logic diagram (attachment) 2. As the pressure increases - the variable speed compressor speeds up, or additional compressors are brought on line as required.
3. Second Stage Suction Pressure	Suction pressure will be controlled in twelve increments as shown in the logic sequence diagram. There are four compressors, each compressor is capable of being turned on or off Compressor sequencing can be varied to spread compressor on-time and wear evenly. A setpoint has not yet been determined for the second stage. Two scenarios are possible: 1. Second stage suction pressure is held constant, giving a constant pressure ratio on the first stage 2. Second stage suction pressure floats with condenser pressure, giving equal pressure ratios between the stages. Both can be incorporated in the control logic; the determination of which scenario is used will be made during the testing program, and will depend on best mode of operation at off-peak conditions (power consumption, avoidance of wet compression).
4. Discharge Pressure	This is not a controlled parameter, but will float with rejection loop temperature. However, it may be used to determine stage pressure ratios as discussed above.
5. Refrigerant Flowrate to Evaporator	This will be controlled by a Pulse-Width Modulated control valve, set to maintain about 5°F degrees of superheat out of the evaporator. (This setpoint may be adjusted by the controller.)
6. Direct Cooling Mode (38°F to 40°F)	The direct cooling heat exchanger will be in use whenever the rejection loop temperature is at or below 45°F, and the chilled water temperature is greater than the rejection loop temperature: 1. A Pulse Width Modulated (PWM) control valve on the rejection loop will start to bypass a portion of the flow around the direct cooling heat exchanger when the chilled water flow out the exchanger drops below 40°F 2. The PWM control valve will adjust to control chilled water temperature between 38°F -to 40°F 3. The PWM control valve will fully bypass the heat exchanger if chilled water temperature drops below 37°F, and cannot be controlled in the specified range. (This is to prevent freeze-up, as the heat rejection loop can drop to 17°F.)

which contains liquid, as well as compression in the two-phase region inside each compressor. If these features need to be designed into the system, the program will initially go to an alarm condition so that the situation can be closely monitored, and corrective action taken where appropriate and possible. Exact parameters for action and/or system shutdown will then be determined during the test phase.

Another situation which will be monitored and flagged for action will be a chill water loop temperature below 35°F. This situation can occur if either the direct heat exchanger control is not operating properly, or the suction pressure control for the evaporator continues to run low. Freeze-up and damage could occur in these two heat exchangers, so action will be necessary to prevent it.

7.3 Control System Hardware

The system controller is a GE FANUC 9030 which has 512 channel input, 512 channel output, digital, analog, T/C capable. Table 4 contains a list of control system input and output.

Table 4. Control system components list

List Number	Figure 11 Reference Number	Measurement	Type
CONTROLLER INPUTS			
1.	(T14)	Chill Water Inlet Temperature	RTD
2.	(T15)	Chill Water Evaporator Inlet Temperature	RTD
3.	(T16)	Chill Water Outlet Temperature	RTD
4.	(T17)	Rejection Water Inlet Temperature	RTD
5.	(T2)	1st Stage Suction Temperature	RTD
6.	(T4)	2nd Stage Suction Temperature	RTD
7.	(T3)	1st Stage Discharge Temperature	RTD
8.	(T5)	2nd Stage Discharge Temperature	RTD
9.	(P2)	1st Stage Suction Pressure	Analog
10.	(P3)	1st Stage Discharge Pressure	Analog
11.	(P4)	2nd Stage Suction Pressure	Analog
12.	(P5)	2nd Stage Discharge Pressure	Analog
13.	(S1)	Compressor "A" Speed Signal	Analog
		2nd Stage Compressor On/Off (OPTIONAL)	Digital
		1st Stage Compressor On/Off (OPTIONAL)	Digital
CONTROLLER OUTPUTS			
1.		First Stage Variable Speed Compressor A	Analog
2.		First Stage Compressor Power On/Off (3)	Digital
3.		Second Stage Compressor Power On/Off (3)	Digital
4.		First Stage Compressor Start/Stop (3)	Digital
5.		Second Stage Compressor Start/Stop (3)	Digital
6.		Second Stage Valve Indicator	Digital
7.		Second Stage Bypass On/Off	Digital
8.		Direct Heat Exchanger Bypass	Digital
9.		Evaporator Control Valve Setpoint	Analog
10.		Chilled Water Controller Setpoint	Analog
11.		Direct Cooling Controller Setpoint	Analog
12.		Economizer Control Valve Setpoint	Analog
13.		Second-stage desuperheater control valve	Analog

7.4 Data Acquisition Hardware

The GE FANUC 9030 can also provide the data acquisition interface function for the system. The capability of each type for number of data channels exceeds our requirements by about an order of magnitude. Input channel types are compatible with the expected instrumentation. However, the controller does not have enough onboard memory to store appreciable data, pending reduction, so it can only function as an interface.

A list of the data acquisition instrumentation is contained in Table 5, with positions of the measurements shown in Figure 11. This list reflects measurements required to fully assess individual component operation. Instrumentation may be reduced in the future as the operating parameters are understood and characterized.

Table 5. Data acquisition system components list

Figure 11 Reference Number	Measurement
	PRESSURE TRANSDUCERS
(P1)	Refrigerant, Evaporator Outlet Pressure
(P2)	Refrigerant, 1st Stage Suction Pressure
(P3)	Refrigerant, 1st Stage Discharge Pressure
(P4)	Refrigerant, 2nd Stage Suction Pressure
(P5)	Refrigerant, 2nd Stage Discharge Pressure
(P6)	Refrigerant, Liquid Discharge Pressure
(P7)	Refrigerant, Evaporator Inlet Pressure
(P8)	Refrigerant, Economizer Outlet Pressure
	TEMPERATURES (RTD WITH TRANSMITTERS)
(T1)	Refrigerant, Evaporator Outlet Temperature
(T2)	Refrigerant, 1st Stage Suction Temperature
(T3)	Refrigerant, 1st Stage Discharge Temperature
(T4)	Refrigerant, 2nd Stage Suction Temperature
(T5)	Refrigerant, 2nd Stage Discharge Temperature
(T6)	Refrigerant, Condenser Liquid Outlet Temperature
(T9)	Refrigerant, Economizer Outlet Temperature (Hot)
(T10)	Refrigerant, LSHX1 Outlet Temperature
(T12)	Refrigerant, Economizer Outlet Temperature (Cold)
(T14)	Chilled Water, Inlet Temperature
(T15)	Chilled Water, Evaporator Inlet Temperature
(T16)	Chilled Water, Outlet Temperature
(T17)	Rejection Water, Inlet Temperature
(T18)	Rejection Water, Condenser Inlet Temperature
(T19)	Rejection Water, Condenser Outlet Temperature
(T20)	Rejection Water, Direct HX Inlet Temperature
(T21)	Rejection Water, Direct HX Outlet Temperature
	FLOWMETERS
(F1)	Refrigerant, Evaporator Inlet Flow
(F2)	Refrigerant, Economizer Inlet Flow
(F3)	Chilled Water, Inlet Flow
(F4)	Rejection Water, Inlet Flow
	POWER (WATT TRANSDUCERS)
(W1)	Power, 1st Stage Compressors
(W2)	Power, 2nd Stage Compressors
	SPEED
(S1)	Speed, Compressor A

For initial programming, debugging, and laboratory analysis, it will be necessary to link the GE FANUC controller with a IBM compatible PC. During operation, this will provide real-time display of critical data, supervisory control of the system controller, as well as onboard storage of data files for later analysis. The GE FANUC 9030 has limited storage capability, so it will have to be linked to a storage device to retain data. We will use the IBM PC to emulate the functions of the LSSIF system controller. Data acquisition and storage requirements will be coordinated with the NASA.

8. QUALITY ASSURANCE

Prototype fabrication conforms to industry prototyping convention. This is the most cost-effective and satisfies the LSSIF application needs. This convention consists of:

- Minimal definition during the design.
- Build and test the unit to confirm performance.
- Reverse engineer working unit.
- Update the definition and drawings to meet working unit.

Configuration control is per the Foster-Miller Drawing Requirements Manual which conforms to DoD-STD-100 and ANSII14.5-1982.

Inspection at Foster-Miller is conducted by engineering personnel. Discrepancies, failures and corrective action are reported memorandum to JSC technical monitor. Flight program discrepancy reports are not used in order to minimize cost during this prototype effort. Material disposition while at Foster-Miller is performed by the NASA technical and contracts personnel, Foster-Miller engineers and property administrators, and our local DCAS representative.

Since lubrication failure can occur without immediate hardware failure, we will perform limited destructive inspection of the hottest and longest operating compressor or compressors after performance testing and prior to shipping unit. This will verify normal operation and wear with no pending disasters.

9. SAFETY

The LSSIF heat pump contains a relatively small quantity of refrigerant. The actual quantity will be calculated at the end of fabrication. However, Foster-Miller anticipates it will be in the order of magnitude of 5 kg. This should not present a personnel safety hazard, even in the event of a complete loss of charge. Therefore, no leak detector is required.

This unit is designed within manufacturer's component and piping pressure ratings. Brazing is performed per industry standards. Therefore no burst pressure testing is required. Proof pressure will be performed to 140 psig (1.5 x 92 psig maximum operating pressure).

Several other safety features are included in the design. A forced ventilation, metal enclosure protects surrounding personnel and internal equipment. The LSSIF is protected from power surges by a 100 ampere main circuit breaker. Warning signs and touch guards are placed on the compressor discharges to protect personnel when the enclosure is open. Hermetic compressors, no open drives, protect personnel from entanglement with an drive mechanisms.

General workmanship also contributes to safety as no sharp edges on the enclosure or internal parts and components will be allowed.

10. TEST PROGRAM

The test program is preliminary and will evolve as both the LSSIF heat pump design and fabrication evolve and as the LSSIF itself evolves. Therefore many items of this program are to be determined at a later date.

The general philosophy for testing at the vendor is to run the heat pump autonomously through a complete lunar day cycle to verify that it can meet the maximum design load and any decreasing or increasing incremental load between zero and the maximum design load. Several peripheral tests will be performed prior, during and after the lunar day cycle test to verify operation of safety features and to gather informational data for other purposes such as off-design performance predictions.

There are no requirements for operating environment so no environmental tests (i.e., thermal vacuum, vibration, hot/cold ambient conditions) will be performed. However, some parameters such as enclosure internal temperature and compressor head temperature will be recorded and correlated to operating load, conditions and external ambient conditions.

There will be no individual component or subassembly tests performed prior to assembly into the heat pump. Such tests are not warranted because:

- All components are mature commercial components previously tested at the manufacturer with a history of operational reliability.
- The risk of component failure does not warrant the cost of pre-assembly component testing.
- The cost of component replacement at the heat pump level of assembly does not warrant the cost of pre-assembly component testing.
- The LSSIF heat pump is the only contractual end item in this program. So no other end item testing is required.
- A destructive examination of the "hot" compressor in each stage will be performed.

Table 6 represents a test matrix for anticipated tests where: "P/F" is a pass or fail test, "Verify" is a test to verify required or predicted performance, and "Data" is a test performed to collect data with no requirements in place.

10.1 Vendor Testing

Vendor testing will be performed in the thermal systems laboratory at Foster-Miller, Inc., Waltham, Massachusetts. The test setup is shown in Figure 12.

Table 6. Test matrix

	Type	Foster-Miller	SIRF
Proof pressure tests			
• Refrigerant loop	P/F	X	
• Rejection loop	P/F	X	X
• Internal Thermal Control System	P/F	X	X
Safety shut-off verifications	P/F	X	X
Oil sump heater check-out	P/F	X	X
Lunar cycle performance test	Verify	X	X
Transient response testing	Data	X	X
Limited endurance testing	Data	X	
Degraded refrigerant charge performance	Data	X	
Hot test compressor destructive inspection	Data	X	
Integrated SIRF testing	Verify		X

The following tests will be performed at the vendor (Foster-Miller):

1. Refrigerant loop proof pressure test.
2. Rejection loop proof pressure test.
3. Chilled water loop proof pressure test.
4. Safety features and shut off tests.
5. Oil sump heater operation check-out.
6. Lunar day cycle performance testing.
7. Transient response testing.
8. Limited endurance testing.
9. Degraded refrigerant charge performance.
10. Hot compressor destructive inspection.

10.2 Test Equipment Requirements

Test equipment requirements will be established by the cognizant test engineer. The test engineer will ensure that the allowable equipment limits are compatible with the design limits of the LSSIF heat pump. Limits will be placed on the parameters shown in Table 7 plus on any parameters denoted by the manufacturer. Safety features and warning alarms will be installed in the test equipment to ensure these limits are not reached in the test equipment or in the test article.

10.3 Reliability

No reliability requirements exist on the LSSIF heat pump unit, therefore, reliability is dependent on manufacturer's reliability predictions and use of each component within manufacturer design limits.

The only critical design components are the compressors. These will be monitored for compression head, suction and discharge temperature through out testing. The compressor with the hottest operating history will be destructively inspected to verify adequate lubrication and no lubricant degradation.

10.4 Loads

The cognizant test engineer will determine the test rig operating temperatures, flows, and pressures to ensure testing to the appropriate refrigeration loads.

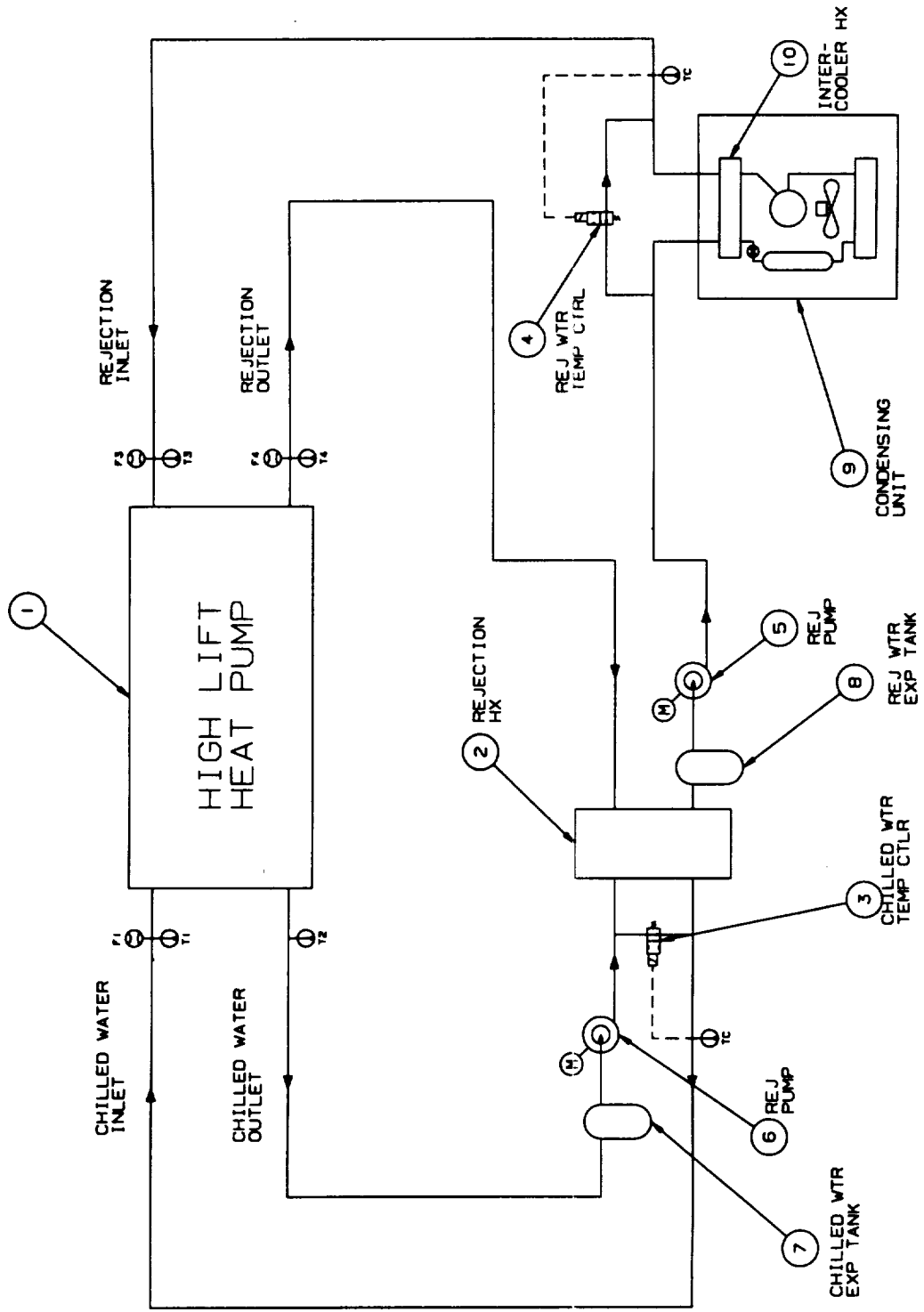


Figure 12. LSSIF heat pump test loop at Foster-Miller

Table 7. Parameters to be limited

Parameter	Alarm	Safety Feature
Internal pressure		Relief valves
Temperature	X	Automatic equipment shut-off
Electrical current		Circuit breaker

10.5 Predicted Environments

There are no requirements for operating environment so no environmental tests (i.e., thermal vacuum, vibration, hot/cold ambient conditions) will be performed. However, some parameters such as enclosure internal temperature and compressor head temperature will be recorded and correlated to operating load, conditions and external ambient conditions.

10.6 Test Program Evolution

Due to the nature of this prototype and the commercial prototype development approach adapted, several test items will be determined as the unit is fabricated. These include allowable tolerances and applicable standards.

10.7 LSSIF Testing

LSSIF testing will be performed in the LSSIF, in building 7, at the NASA Johnson Space Center. The following tests will be performed in the LSSIF:

1. Rejection loop proof pressure test.
2. Chilled water loop proof pressure test.
3. Safety features and shut-off tests.
4. Oil sump heater operation check-out.
5. Lunar day cycle performance testing.
6. Transient response testing.

Test equipment requirements, reliability, loads, predicted environments, allowable tolerances and applicable standards will be determined as the LSSIF heat pump and NASA facility evolve.

11. RECOMMENDATION

Based on the manufacturer research and discussions and based on Foster-Miller's design effort on this item, Foster-Miller recommends procuring the major components and fabricating the LSSIF heat pump per the following Foster-Miller drawings, specifications, and plans:

Specifications:

NAS9614-1	Mechanical Systems Spec. and Description
NAS9614-2	Control and Data Acquisition Systems Spec. and Description
NAS9614-5	Electrical Systems Spec. and Description

Plans:

NAS9614-6	Master Test Plan
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Drawings:

NAS9614-1001	Package Layout
NAS9614-1002	Electrical Schematic
NAS9614-1003	System Schematic
NAS9614-1004	Test Loop
NAS9614-1005	Enclosure
NAS9614-1006	Skid

12. RELATED DOCUMENTS

Specifications:

NAS9614-1	Mechanical Systems Spec. and Description
NAS9614-2	Control and Data Acquisition Systems Spec. and Description
NAS9614-5	Electrical Systems Spec. and Description

Plans:

NAS9614-6	Master Test Plan
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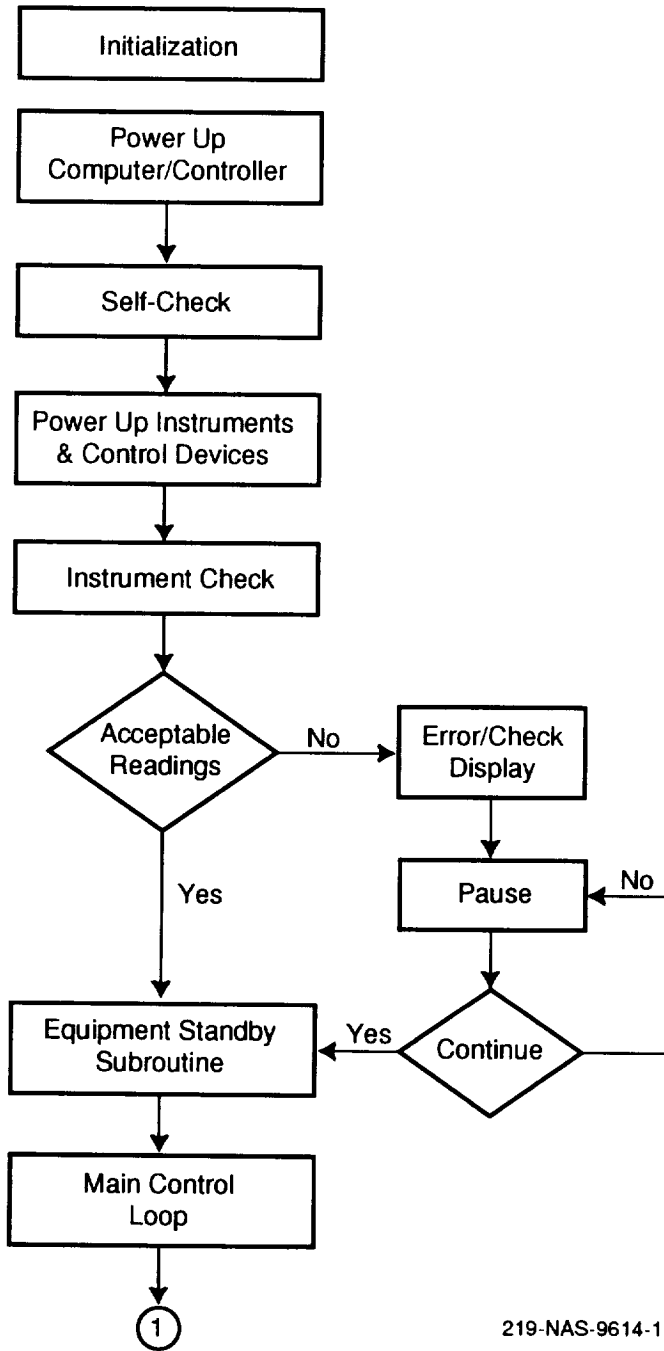
Drawings

NAS9614-1001	Package Layout
NAS9614-1002	Electrical Schematic
NAS9614-1003	System Schematic
NAS9614-1004	Test Loop
NAS9614-1005	Enclosure
NAS9614-1006	Skid

APPENDIX A

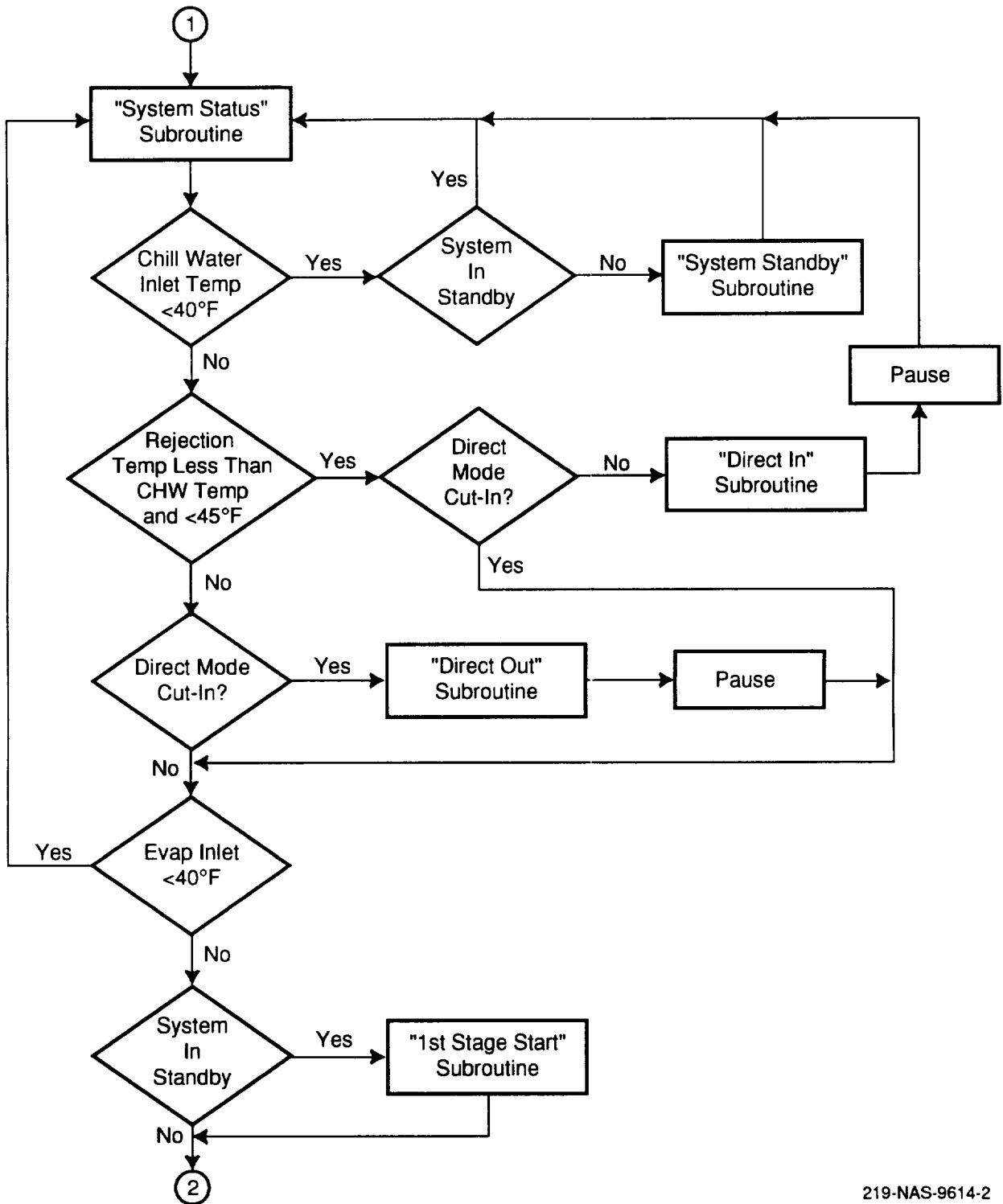
DETAILED CONTROL LOGIC DIAGRAM

Control Logic Diagram

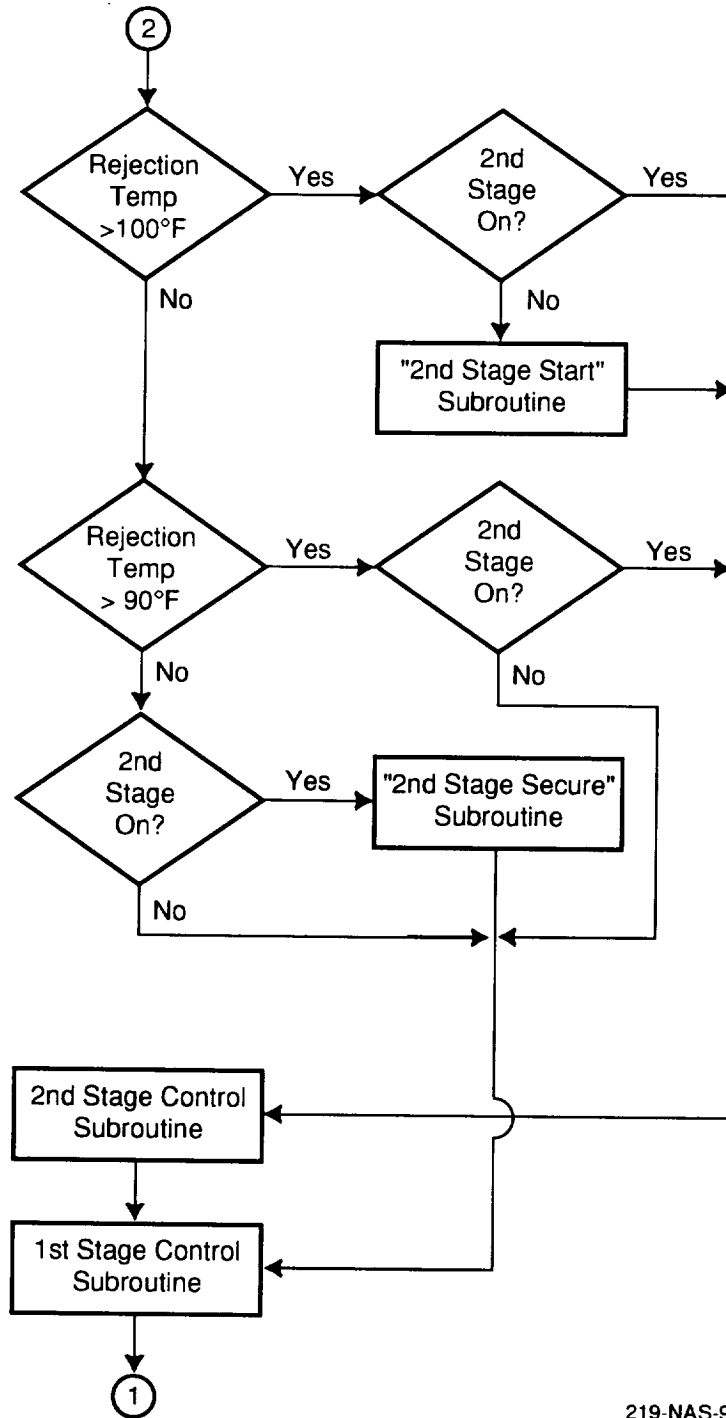


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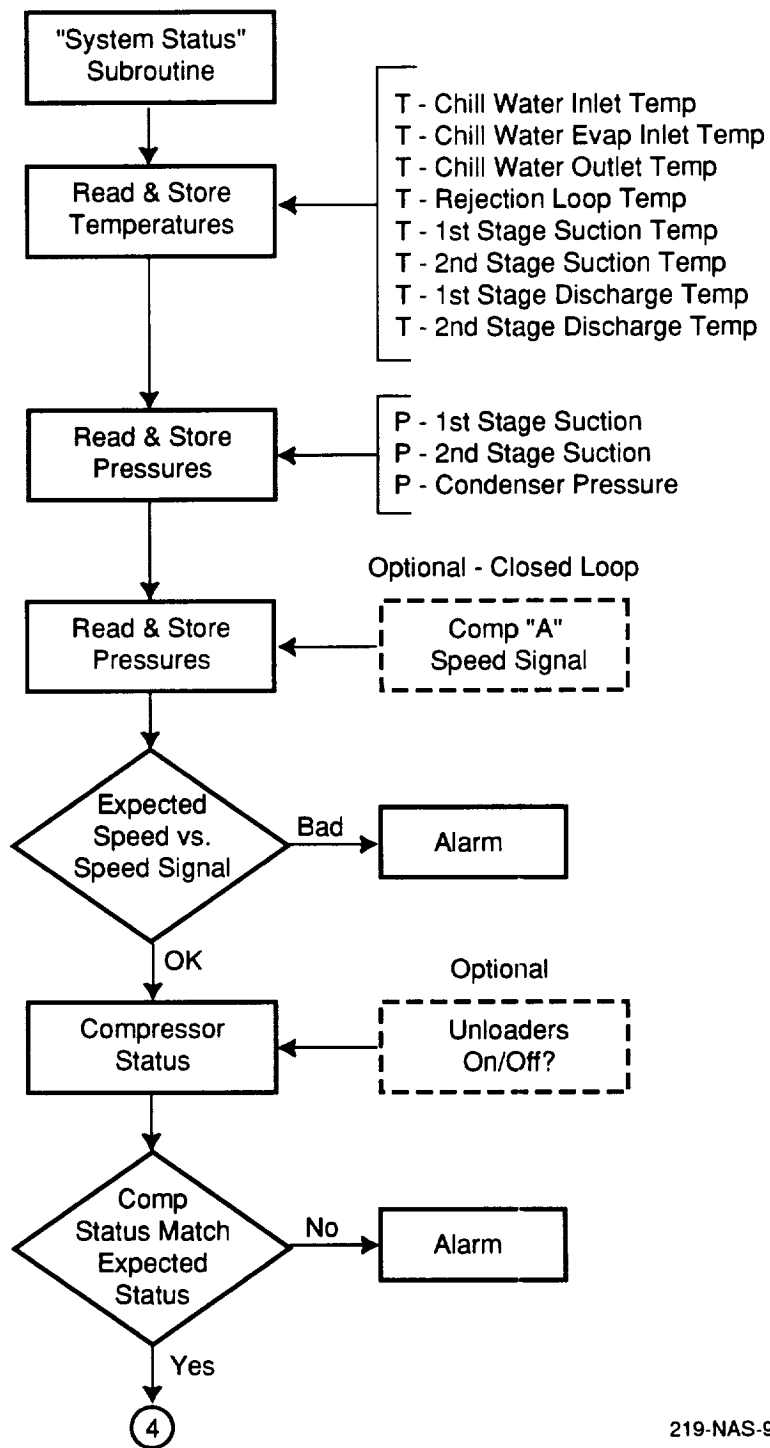
Main Control Loop



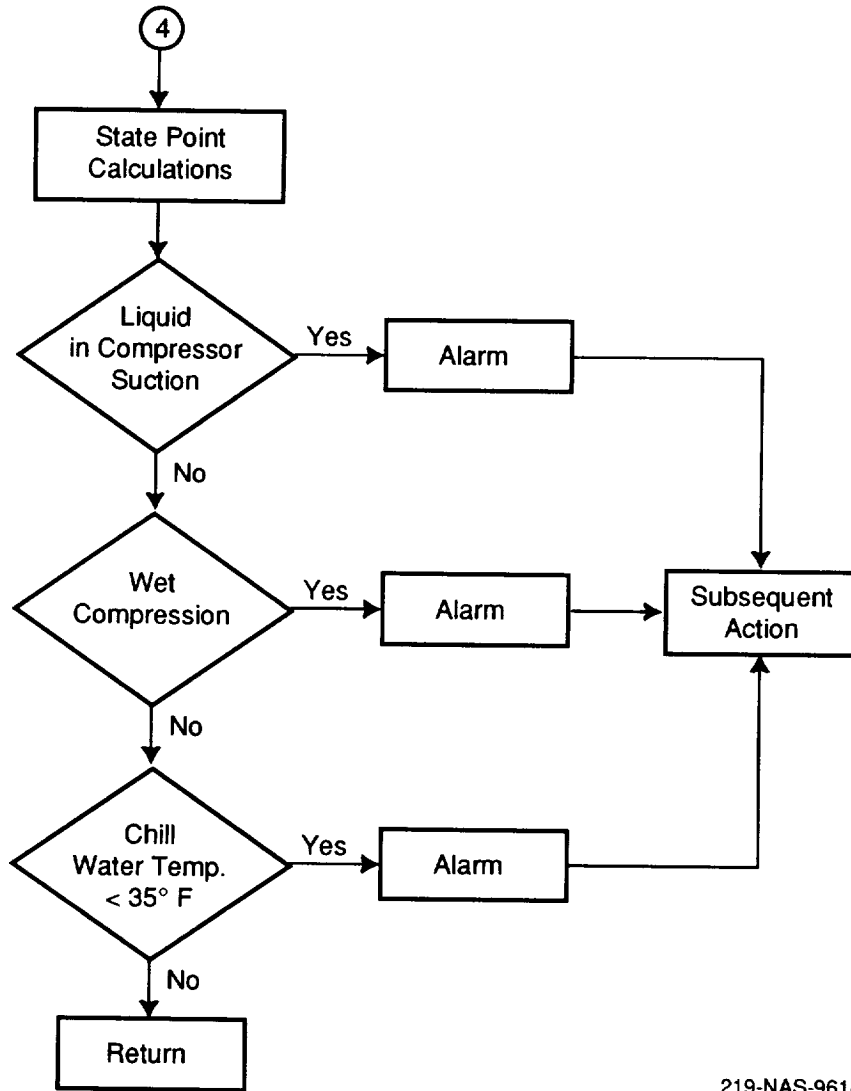
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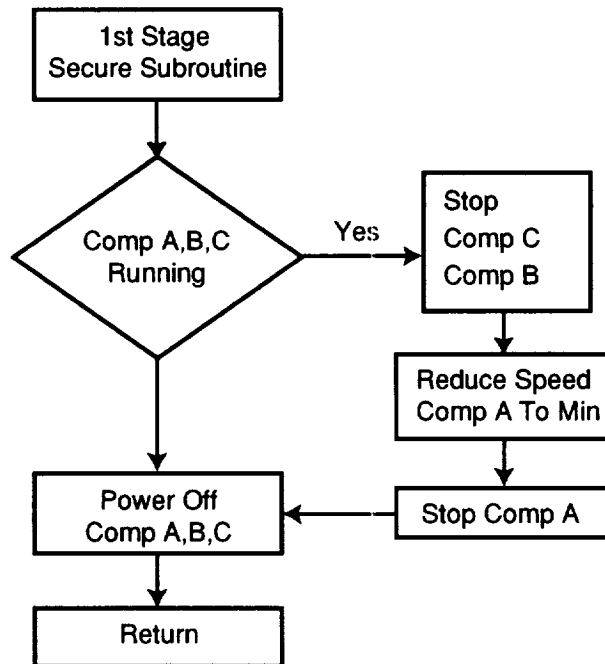
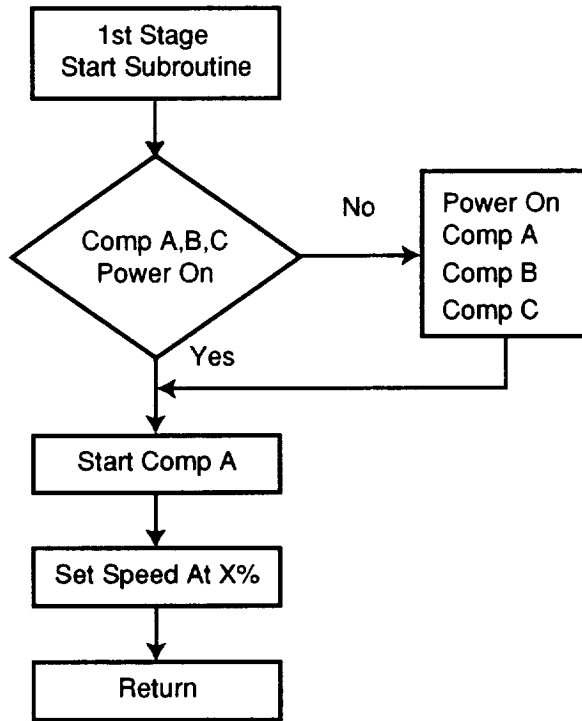
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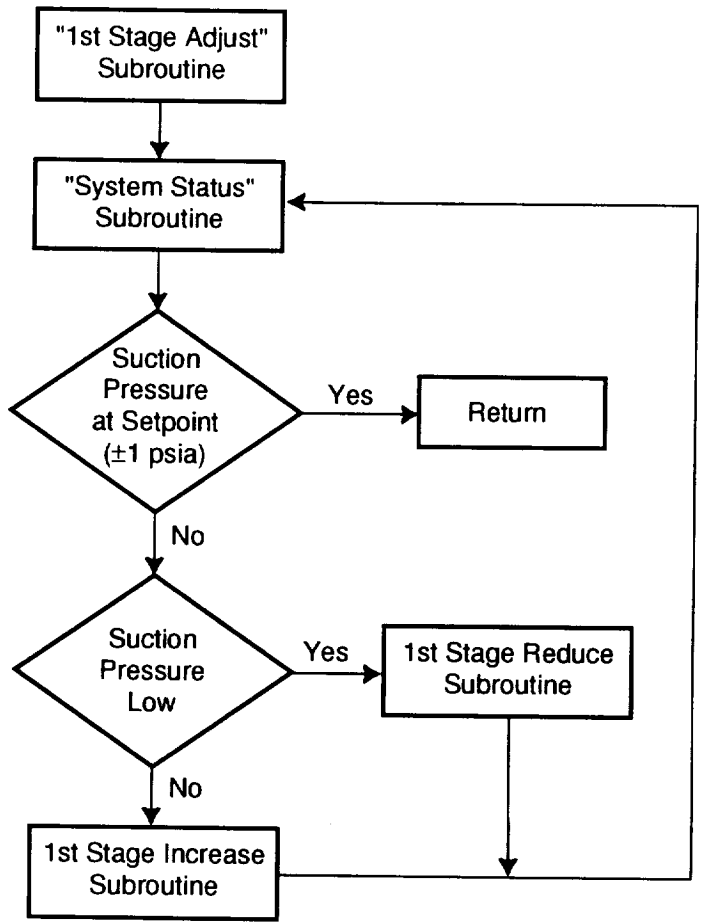
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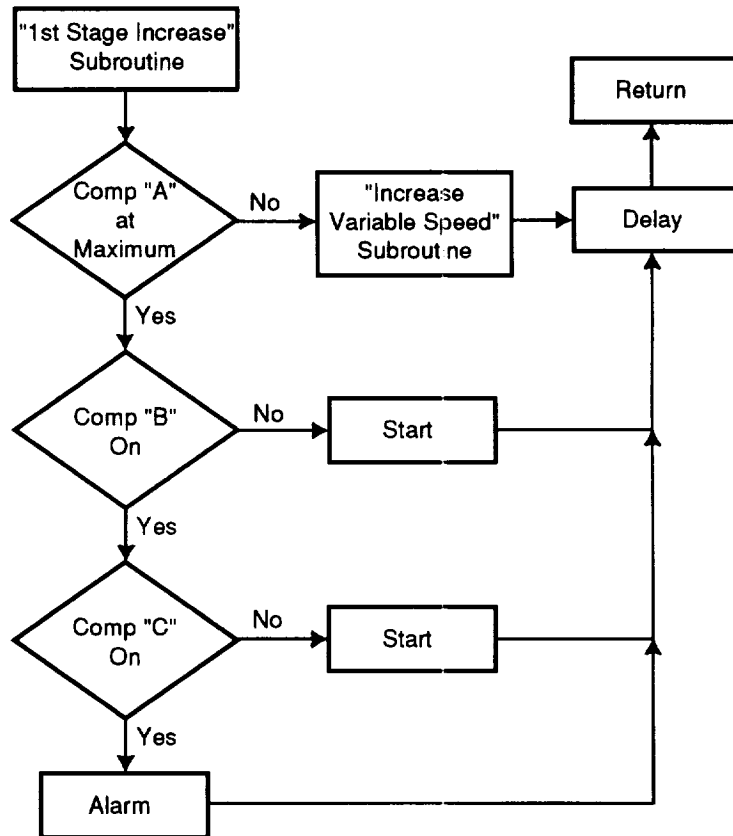
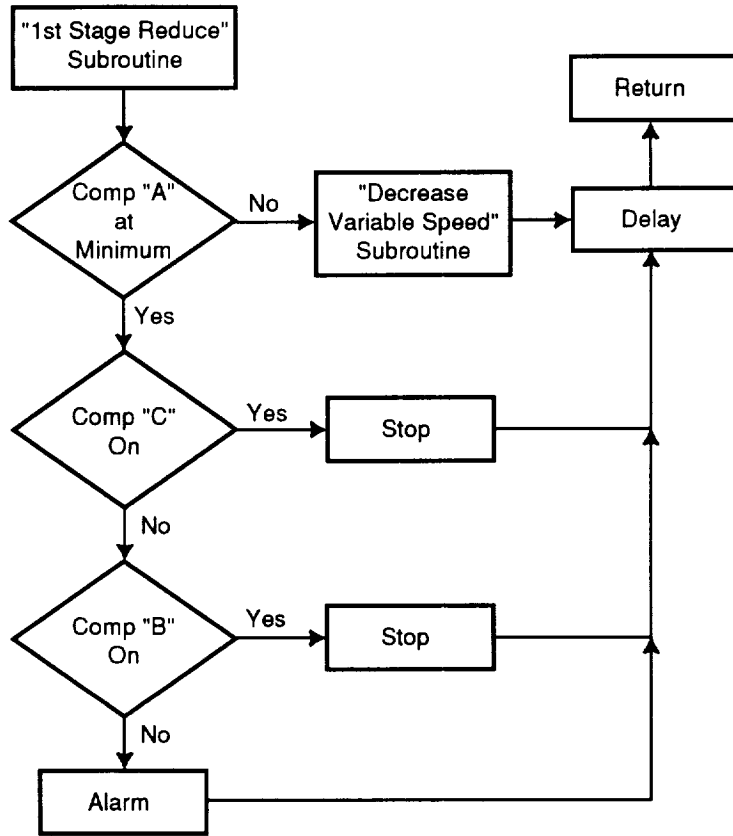
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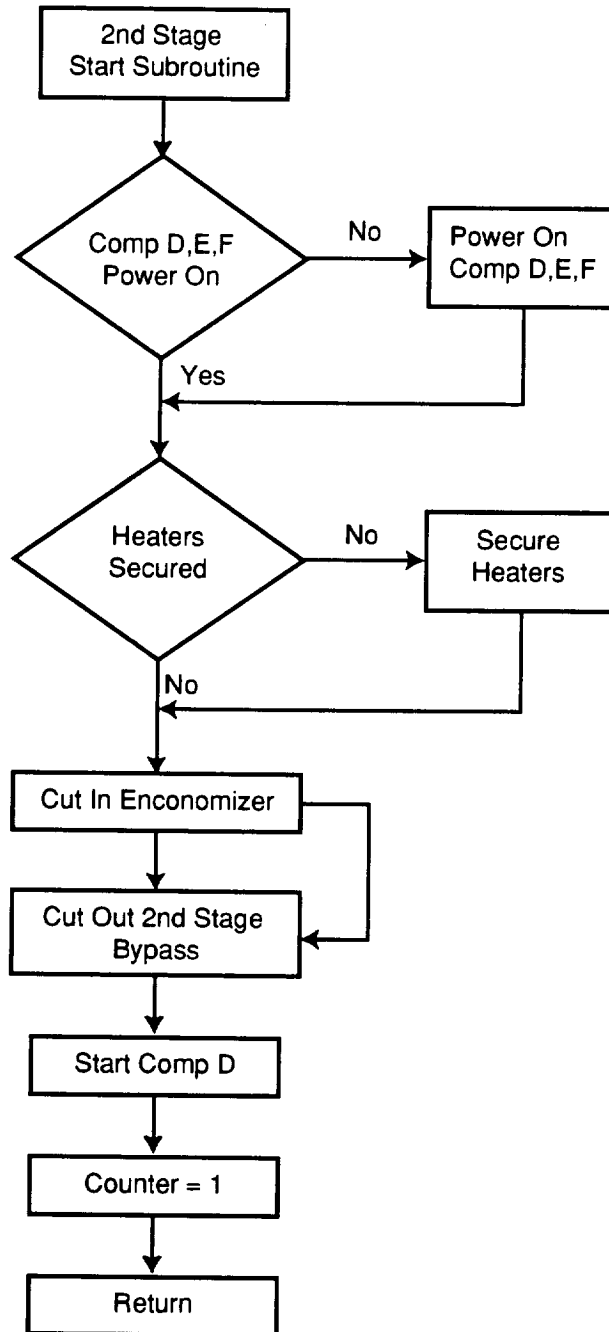


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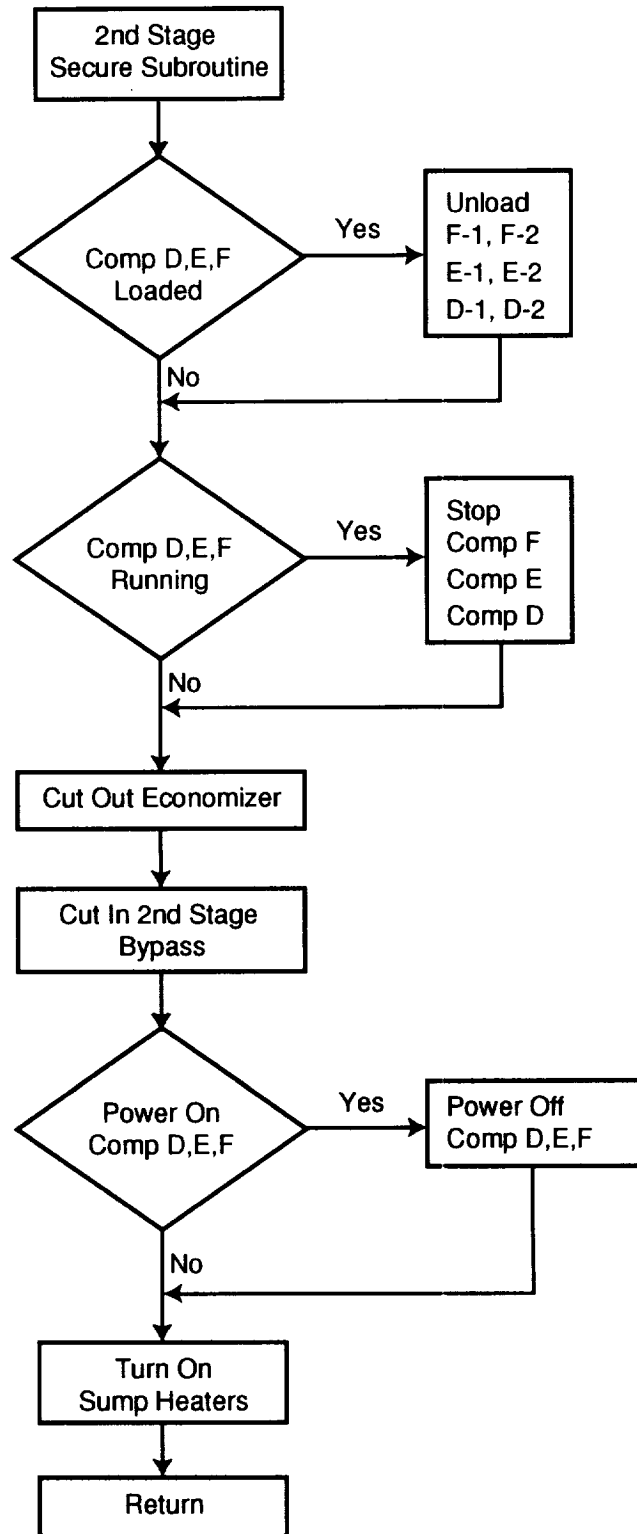


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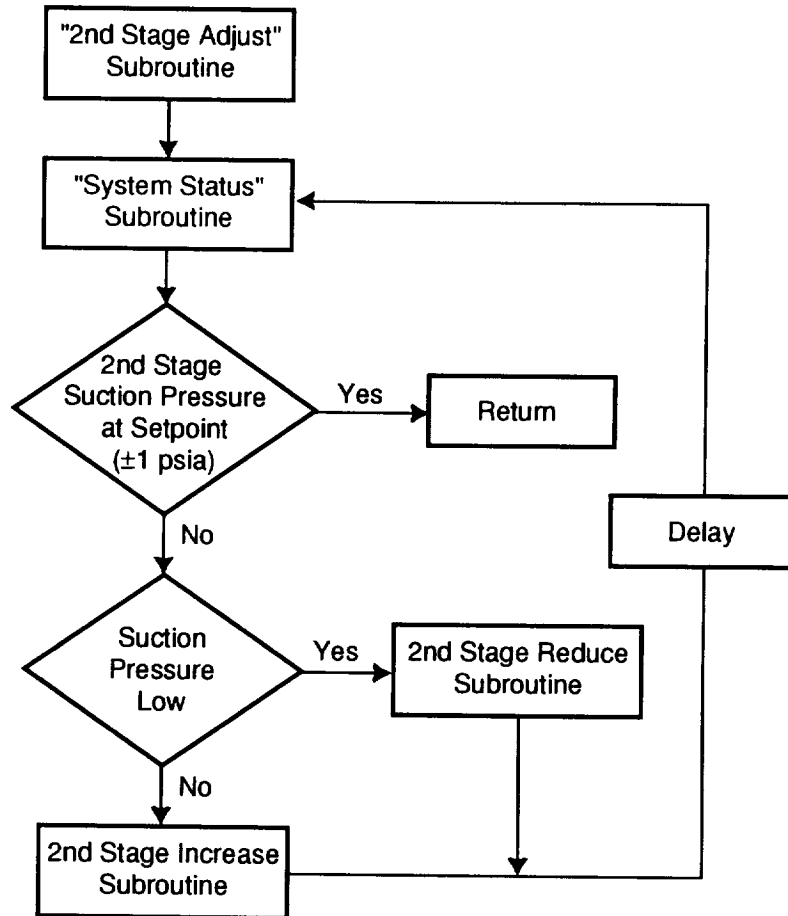




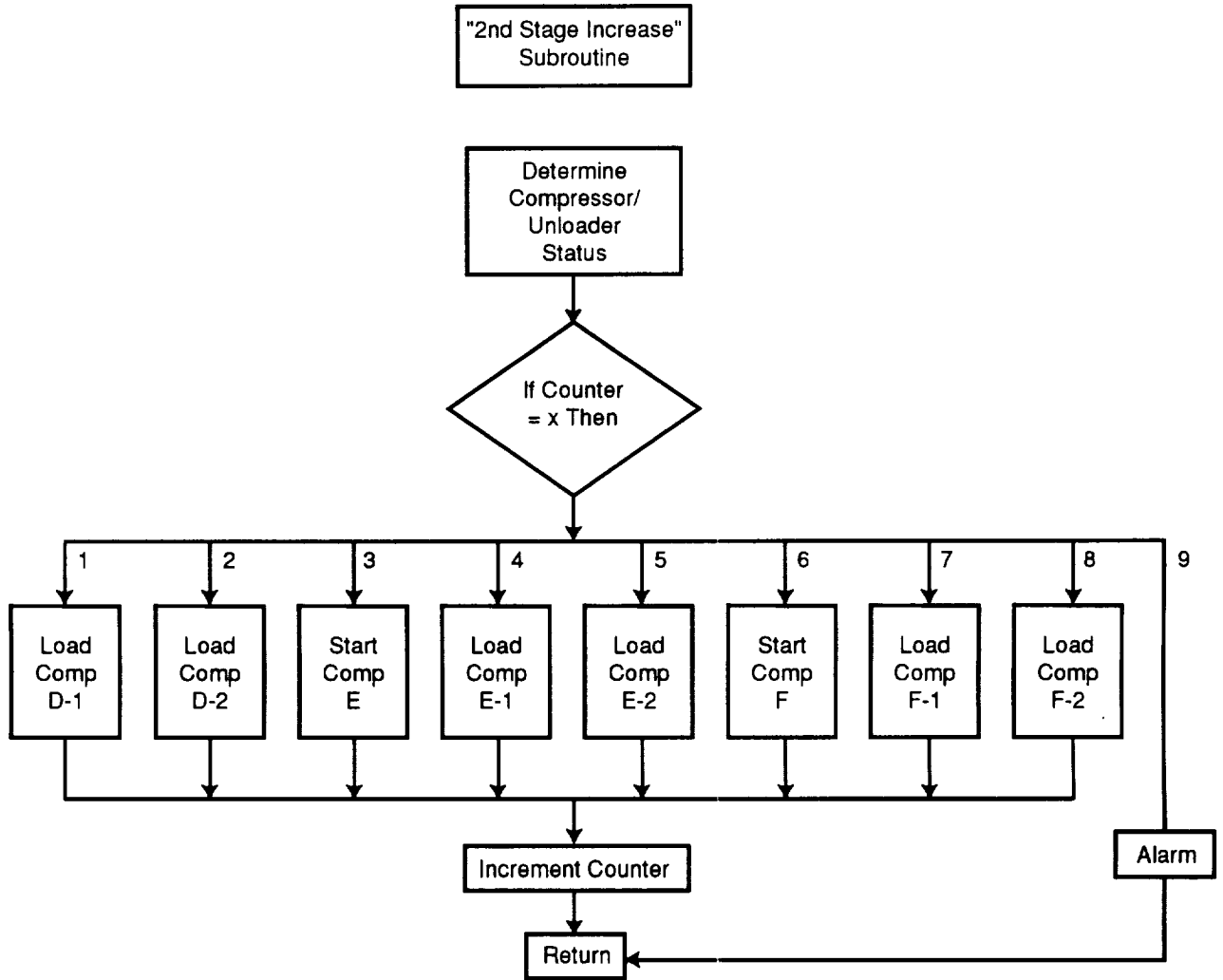
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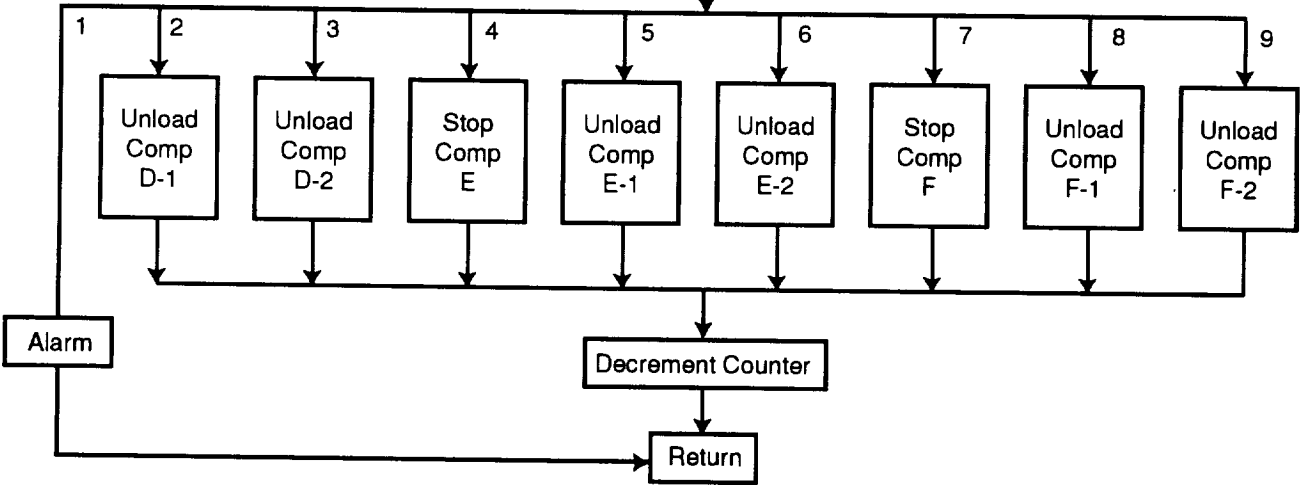


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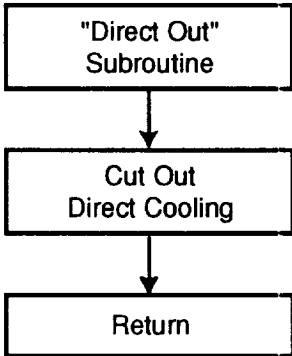
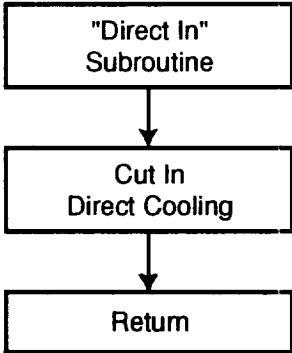
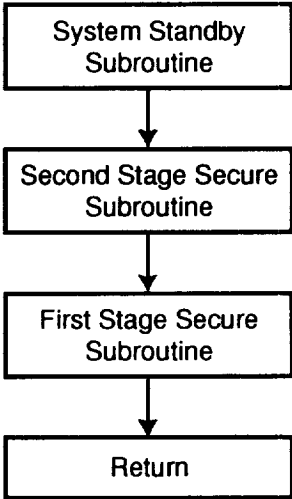
"2nd Stage Reduce"
Subroutine

Determine
Compressor/
Unloader
Status

If Counter
= x Then



219-NAS-9614-9



219-NAS-9614-14

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13. ABSTRACT (Maximum 200 words)

This report describes the Phase II detailed design of the Lunar Base Heat Pump which is a vapor compression heat pump requiring a high temperature lift. Use of a vapor compression heat pump versus other types was based on prior work performed for the Electric Power Research Institute. A high lift heat pump is needed to enable a thermal control system to remove heat down to 275K from a habitable volume when the external thermal environment is severe. For example, a long term lunar base habitat will reject heat from a space radiator to a 325K environment.

Referring to the Phase I Final Report, a selection process was performed that included an optimization trade study, quantifying the effect of radiator operating temperature and heat pump efficiency on total system mass. This process also included selecting a radiator operating temperature corresponding to the lowest system mass. Total system mass included radiators, all heat pump components and the power supply system. The study showed that lunar night operation, with no temperature lift, dictated the radiator size. To operate otherwise would require a high mass penalty to store power. With the defined radiation surface, and heat pump performances assumed to be from 40 percent to 60 percent of the Carnot ideal, the optimum heat rejection temperature ranged from 387K to 377K, as a function of heat pump performance.

Refrigerant and thermodynamic cycles were then selected to best meet the previously determined design conditions and to meet ground based environmental regulations. This dictated the use of HCFC-123 instead of the best performing candidate, CFC-11.

During Phase II, the detailed design was performed for the ground based test unit (LSSIF, previously the SIRF) based on the HCFC-123 refrigerant. The results of this design effort are contained in this report.

During Phase III, the design will be fabricated, tested at Foster-Miller, and shipped to the NASA Johnson Space Center for integration into the Advanced Life Support Systems Integration Facility (LSSIF, previously known as the SIRF).

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