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SYSTEMS EVALUATION OF THERMAL BUS CONCEPTS

9 FEBRUARY 1982

PREPARED FOR

THE NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
LYNDON B. JOHNSON SPACE CENTER
HOUSTON, TEXAS

BY

VOUGHT CORPORATION
DALLAS, TEXAS

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1.0 SUMMARY

This report summarizes the results of a study of thermal bus concepts. The purpose of the thermal bus will be to provide a centralized thermal utility for large, multi-hundred kilowatt space platforms projected for the 1990's. Objectives of the study were identification of thermal bus concepts and selection of the most promising concept(s) for development based on the results of system level trade studies.

The study was conducted in three major tasks. A schedule of the study is presented in Figure 1. In Task I, concepts were generated, defined, and screened for inclusion in system-level thermal bus trades. In Task II, parametric trade studies were conducted with the concepts surviving Task I screening in order to define the operational envelope, performance, and physical characteristics of each. As a result of this task, two concepts were selected as offering the most promise for thermal bus development. Task III consisted of the study reporting including monthly progress reports, the final program briefing, and this final report.

Four concepts were generated as a result of Task I. All of the concepts involved two-phase flow in order to meet the required isothermal nature of the thermal bus. Two of the concepts employ a mechanical means to circulate the working fluid, a liquid pump in one case and a vapor compressor in another. Another concept utilizes direct osmosis as the driving force of the thermal bus. The fourth concept was a high capacity monogroove heat pipe. After preliminary sizing and screening, three of these concepts were selected to carry into the trade studies. The monogroove heat pipe concept was deemed unsuitable for further consideration because of its heat transport limitations. One additional concept utilizing capillary forces to drive the working fluid was added at the Concepts Briefing at NASA's request.

Parametric system-level trade studies were performed on the four concepts which were carried into Task II. Sizing and weight calculations were performed for thermal bus sizes ranging from 5 to 350 kW and operating temperatures in the range of 4 to 120°C (39 to 248°F). System level considerations such as heat rejection and electrical power penalties and interface temperature losses were included in the weight calculations. The following conclusions were reached as a result of the thermal bus trade studies:

1. System weight is not a significant factor in selecting a thermal bus concept. The weight variation between the concepts was between 2 and 4% for all concepts except the osmotic concept which was approximately 15% heavier than the other concepts.

2. Ammonia is the best working fluid from a weight and performance standpoint. It's toxicity and flammability, however, make it unsuitable for use in a manned cabin. Water is unattractive for a low temperature bus ($<40°C$) because of its low vapor pressure.

3. The mechanical pump driven concept offers the most promise for near term development. It has good performance characteristics, requires minimum development, and requires little power (only 12 W for a 350 kW bus).

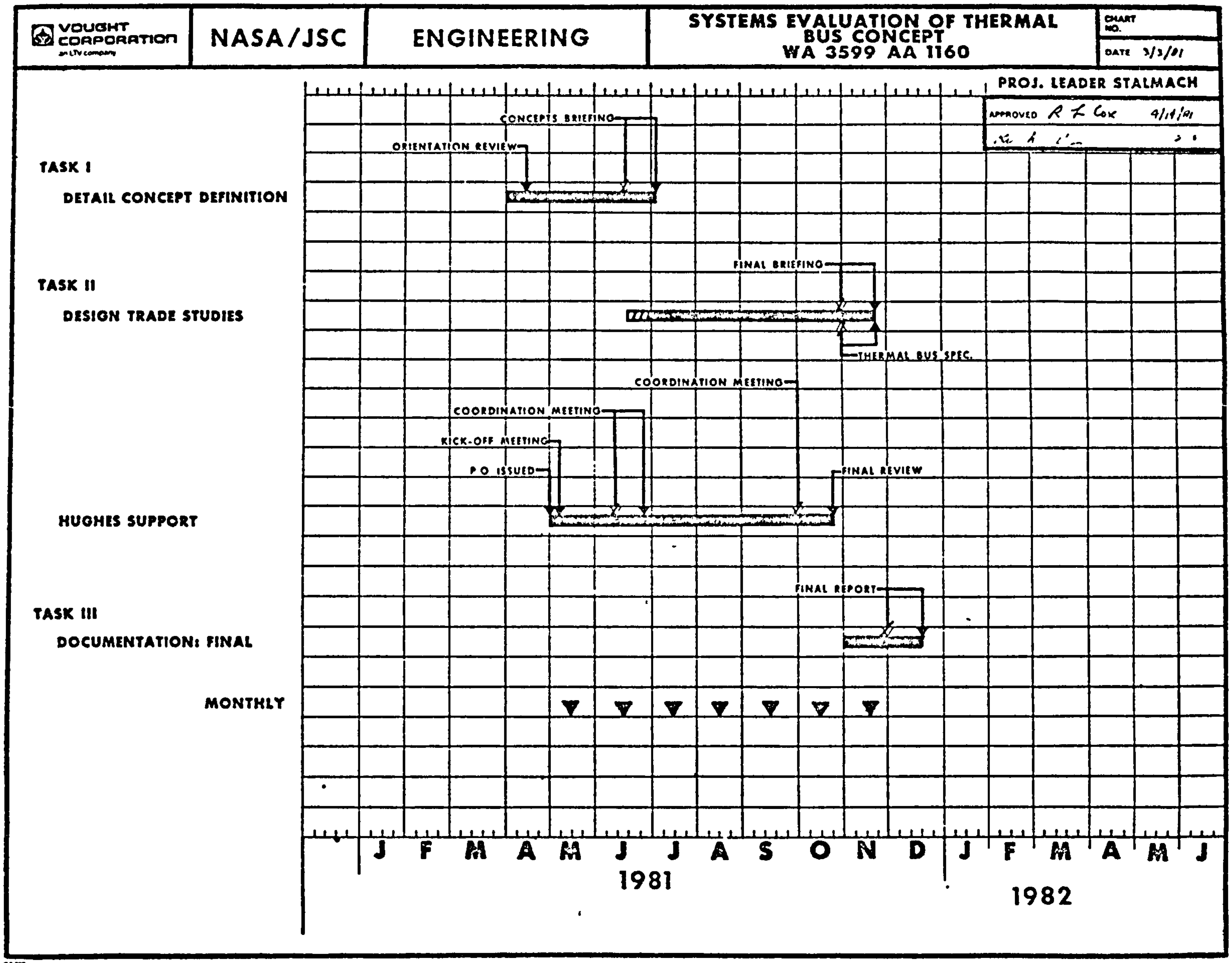


FIGURE 1 STUDY SCHEDULE

4. The capillary pumped concept offers promise as a passive heat transport system. Such a system would be self regulating and would require no power input. However, its development risk is higher and up-front research and development of the capillary pump would be required.

5. The osmotic pumped thermal bus concept offers good performance and a passive system. It has problems however with flow control and requires much research and development work.

6. The electrical power weight penalty has a small effect on the total system weight for the compressor driven concept. However, the optimum system configuration is significantly impacted.

7. A redundant thermal bus is required for all of the concepts in order to reach the reliability goal of 0.99 for 10 years. In addition, redundant components and/or scheduled maintenance is required on a component level.

As a result of this study the following recommendations are made for development of a thermal bus.

1. Develop the thermal bus using the pump driven approach initially: build and test a thermal bus prototype, develop heat rejection and temperature control techniques, develop the pump and heat exchangers.

2. Develop the capillary pump separately: build and test a capillary pump and establish its characteristics and limitations.

3. Continue laboratory materials and concepts studies for the osmotic pump module.

2.0 INTRODUCTION

In previous space systems, thermal management has been achieved either passively or through the use of pumped liquids and electrical heaters. The Shuttle Orbiter and Spacelab are examples of this technology. Evolving future space platforms, however, will require a much more significant role of thermal management because of the multi-year mission durations, large quantities of waste heat to be dissipated, long physical distances involved, and variety of payloads and missions which must be accommodated by the platform.

The idea of a thermal "utility" has evolved to effectively serve these growing thermal management needs. The central element of the utility concept is a Thermal Bus, which would provide the function of heat transport at a given temperature level(s). Desirable performance requirements include:

- Provide a uniform thermal control source (cooling and heating insensitive to the addition or removal of loads) for space platform electrical, life support, mechanical, scientific, and experimental equipment.

- Provide heat load/payload interfaces. Accommodate payload change with maximum ease of interfacing (connection and disconnection) and with minimum impact to other payloads.

- Transport the heat from payloads to the heat rejection system for rejection.

- Provide interface with the heat rejection system which permits flexibility in maintenance, growth, and reconfiguration. Provide the heat load control.

The primary purpose of this study was to perform a parametric system level trade study of promising thermal bus concepts in order to select one or more concepts for development.

3.0 REQUIREMENTS, GUIDELINES AND CONSTRAINTS

In the overall scheme of central thermal management, the thermal bus would be most useful if heat could be added or subtracted at several locations, and if the bus provided a constant temperature source or sink insensitive to the quantity or distribution of heat added or subtracted. Since heat sinks are needed at various temperature levels (i.e., about 4°C (40°F) for condensing heat exchangers in manned modules, 20°C - 40°C (68°F - 104°F) for many equipment items, and on the order of 120°C (248 F) for space processing furnaces), and interfacing modules may have shorter thermal bus length requirements, it is possible that a single bus concept will not be optimum for all cases. The objectives of our Requirements and Guidelines were to permit the determination of the most promising concepts and their regions of applicability.

Figure 2 lists the Requirements, Guidelines and Constraints used in this study. They were meant to provide the needed guidance to conduct a meaningful trade study while not being restrictive to the point of excluding potential concepts which may offer advantage in only a part of the entire spectrum of interest.

FIGURE 2

REQUIREMENTS, GUIDELINES AND CONSTRAINTS

PERFORMANCE

- Desired capability for localized heat removal or delivery
- Heat Removal Load : 5 kW to 350 kW, Max/Min Load Ratio 10 to 1
- Temperatures : 4° to 120° (40°F to 250°F)
- Localized Heat Delivery : 0 to 50 kW
- Isothermal Character : Goal of 5°C band

PHYSICAL CHARACTERISTICS

- Centralized System Length : 15 m to 50 m (50 ft to 164 ft)
- Interfacing Module Length : 3 m to 15 m (10 ft to 50 ft)
- Minimum Weight and Volume

DESIRED OPERATIONAL CHARACTERISTICS AND TIMING

- Modular growth capability
- On-orbit reconfiguration capability
- Capability for simple make and break of interface with equipment
- Minimize monitoring and control required
- Minimize on-orbit maintenance
- Early 1990's technology readiness

ENVIRONMENTS

- Pressurized compartment or unpressurized area
- Assume thermal control and micrometeoroid protection is provided by surrounding structure in either case
- Launch vibroacoustic and acceleration per Spacelab User's Handbook (pressurized module and pallet)

INTERFACES

- Not to be addressed in detail in this parametric study - but concepts should not preclude feasible interfaces
- Consider open-fluid or heat exchanger interfaces between modules

FIGURE 2 (CONT'D)

REQUIREMENTS, GUIDELINES AND CONSTRAINTS

TRADE PENALTIES

- Power : 45 to 159 kg/kW (100 to 350 lb/kW)
- Launch Costs : \$1540/kg (\$700/lb)
- Heat Rejection : From constructable radiator concepts in NAS3-22270 at -40°C environmental sink

SAFETY

- No toxic or flammable fluids in pressurized compartments
- Fluid toxicities compatible with practical ground handling for bus in unpressurized areas
- No contact temperatures above 45°C (113°F)
- General guidelines from Rockwell Phase B Modular Space Station

RELIABILITY

- 10 year life design goal at 0.99 probability (where practical)
- Redundancy and minimal maintenance to achieve life goal
- Indefinite life with further maintenance
- Minimize moving parts

OTHER

- Minimum life cycle cost
- Minimum vibration
- Minimum EMI generation
- Minimum contamination threat to payloads

4.0 CONCEPT GENERATION

The purpose of a thermal bus, in analogy to an electrical bus, is to provide a uniform thermal control source (cooling or heating insensitive to the addition or removal of loads) for space platform electrical, life support, mechanical, scientific, and experimental equipment.

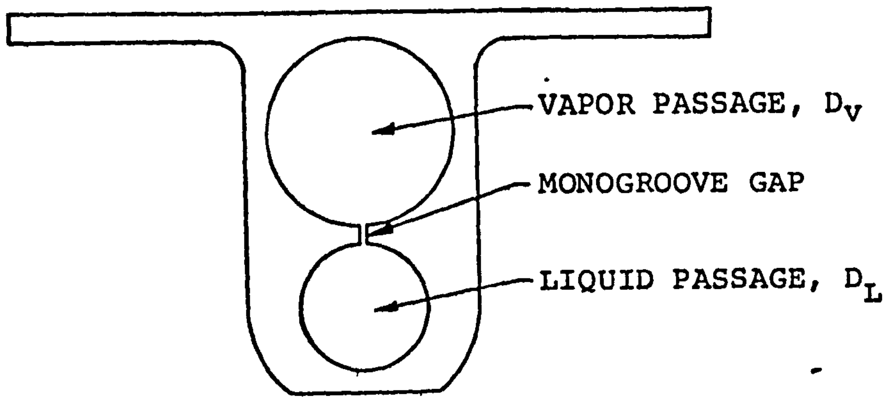
Because of the isothermal requirement of a 5°C (9°F) temperature band for the thermal bus, only concepts employing two-phase flow were considered. During Task I, four concepts were identified as candidates for a thermal bus. Those concepts were a mechanical pump driven system, a vapor compressor driven system, an osmotic heat pipe system, and a high-capacity, monogroove heat pipe. Preliminary sizing of these concepts were performed during Task I in order to screen these concepts for inclusion in the system level trade studies.

As a result of this preliminary screening, it was decided that the monogroove heat pipe concept of Reference 1 was unsuitable for use over much of the heat load range of interest because of its heat transport limitations. Figure 3 illustrates a cross-section of the monogroove heat pipe design. This design ends up being wall wick limited for most cases, i.e., there is a limit on how large the diameter may be without drying out the circumferential grooves on the wall of the evaporator vapor passage. As a result, the capacity of the heat pipe cannot be increased by increasing the heat pipe diameter past some limit. Figure 4 presents the maximum heat transport capability in kilowatt-meters of the monogroove heat pipe for various temperatures and fluids. Figure 5 shows how this heat transport limit translates into maximum heat load as a function of heat pipe transport length. Because of the relatively low heat loads that can be transferred over long distances, this concept was not carried into the Task II trade studies.

At the suggestion of NASA/JSC another concept was added to replace the monogroove heat pipe. This concept is a capillary pumped concept which was first proposed by the Lewis Research Center in 1966 (Reference 2) and is described in Section 4.2.

4.1 CONCEPT 1 : MECHANICAL PUMP DRIVEN

The mechanical pump driven concept is illustrated schematically in Figure 6. A pump located in the liquid portion of the loop provides the driving force for circulation of the working fluid. The evaporators, located in parallel, could be flow-through heat exchangers such as a tube-in-shell heat exchangers or coldplates. Flow to each evaporator could be controlled by metering the liquid flow. This may be done by controlling the pressure drop through each evaporator with orifices to get the desired flowrate. The flow distribution would be set beforehand to allow adequate flow for the maximum heat load expected at each evaporator. At less than full heat input, all of the liquid will not be evaporated and there will be a vapor/liquid mixture exiting the evaporator. The vapor flows to the condenser where it is condensed to a slightly sub-cooled state. A pre-charged accumulator (pressurized with gaseous nitrogen) located just upstream of the pump sets the saturation pressure of the loop (and therefore the evaporating and condensing temperatures) at the desired level. The loop pressure may be set at a fixed value or, if different temperature set points are desired, a pressure regulating device may be used to vary the accumulator pressure.



HIGH CAPACITY MONOGROOVE HEAT PIPE DESIGN

FIGURE 3

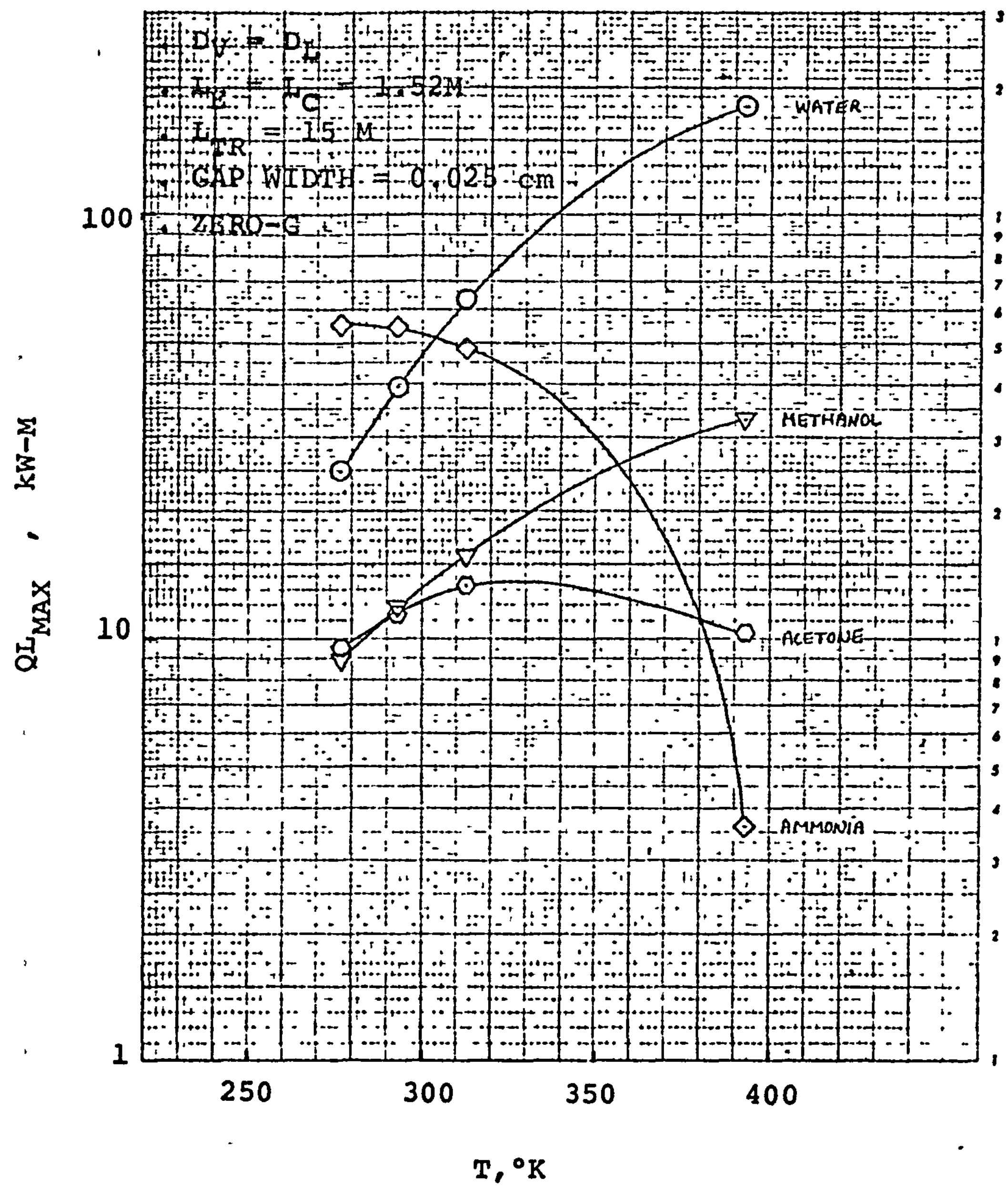


FIGURE 4

HEAT TRANSPORT CAPABILITY OF MONOGROOVE HEAT PIPE

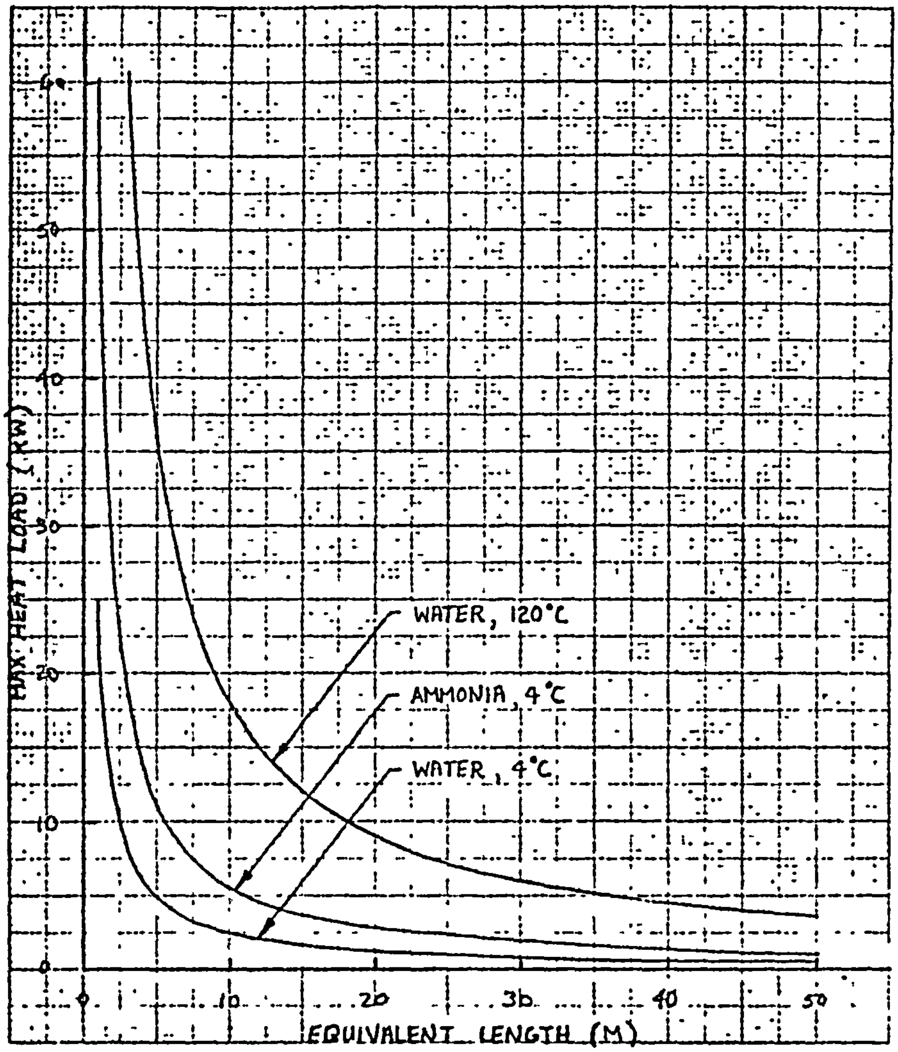
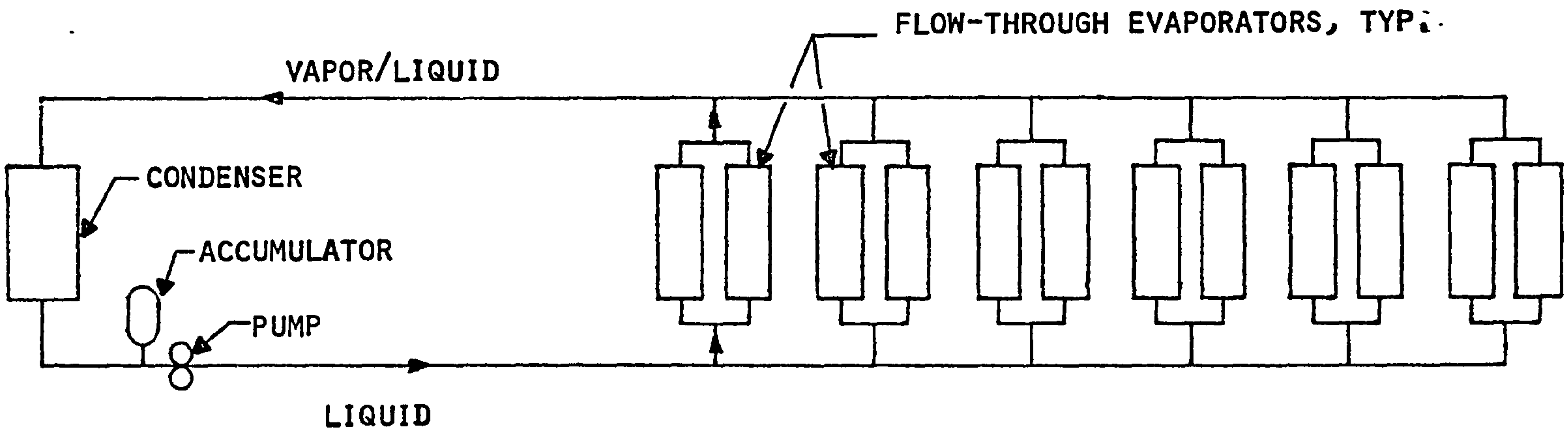


FIGURE 5
MAXIMUM HEAT LOAD AS A FUNCTION OF HEAT
TRANSPORT DISTANCE FOR MONOGROOVE HEAT PIPE

FIGURE 6

CONCEPT 1 - MECHANICAL PUMP DRIVEN



The pump for this concept could be a centrifugal pump. The pump flowrates required are presented in Figure 7 for various heat loads, working fluids, and temperatures. Figure 8 gives the performance characteristics of the Orbiter coolant pumps. It can be seen that two Orbiter Freon 21 pumps in parallel could pump 28.2 l/min (7.45 gal/min) which is close to the requirement of 29 l/min (7.66 gal/min) for a 20°C, 350 kW thermal bus with ammonia as the working fluid. A problem with using a centrifugal pump is its limited lifetime. It seems unlikely that such a pump could operate continuously for the desired 10 year lifetime of the thermal bus. Redundancy or replacement of the pump package would be necessary to meet this goal.

Another possibility for pumping of the liquid is a bimorph pump concept such as the one described in Reference 3 and illustrated in Figure 9. The advantage of this pump concept is that it has few moving parts and therefore should have a long lifetime. This concept makes use of the unique characteristics of bimorph elements. A bimorph is made up of two layers of ferroelectric crystal or ceramic cemented together with electrodes attached as illustrated in Figure 9. Application of an electric voltage to the bimorph causes it to bend due to the piezoelectric effect of the ferroelectric crystals. Reversing the voltage causes the bimorph to bend in the opposite direction. The author of Reference 3 constructed a cylindrical pump made up of bimorph elements separated by an elastic material. By supplying an alternating current to the electrodes, the volume of the pump is varied resulting in liquid flow through the check valves as shown. The pump Narasaki built and tested is illustrated in Figure 9 with dimensions given in mm. Figure 10 presents experimental flowrates achieved with this pump. The fluid used in the experiments was not identified but reportedly had a viscosity of 2 cSt. The pressure rise available with this pump was not reported but the efficiency was given as between 4 and 10%. A literature search was conducted to find any other references of bimorph vibrator pumps but none were found.

4.2 CONCEPT 2 : CAPILLARY PUMP DRIVEN

The capillary pumped concept is illustrated schematically in Figure 11. This concept makes use of capillary forces to drive the working fluid as in a heat pipe. The evaporator and the pump are one and the same. In this concept the liquid and vapor phases are separated by a wick in the evaporator and there is vapor flow only in the vapor line. As in the mechanical pump driven concept, the temperature of the thermal bus is set by a pressurized accumulator.

The evaporator in this concept is a capillary pump illustrated in Figure 12. This evaporator concept is similar to one reported in References 2 and 4. The pump is made up of an axially grooved pipe with a wick on its interior surface as shown. This wick may be a quartz felt as it was in Reference 2 or a metal felt. The liquid, which should be slightly subcooled, enters the pump as illustrated and saturates the wick. Heat is applied to the exterior surface of the evaporator and is conducted through the fins to the wick interface. At this point the liquid is evaporated out of the saturated wick. The evaporation of liquid results in highly curved liquid-vapor interfaces (menisci) in the wick pores which provides the capillary pressure rise to drive the fluid. The magnitude of this capillary pressure is a function of the wick pore size and the surface tension of the working fluid. By choosing a wick with a very small pore size, a relatively large pressure rise may be achieved. Figure 13 shows the capillary pressure rise available

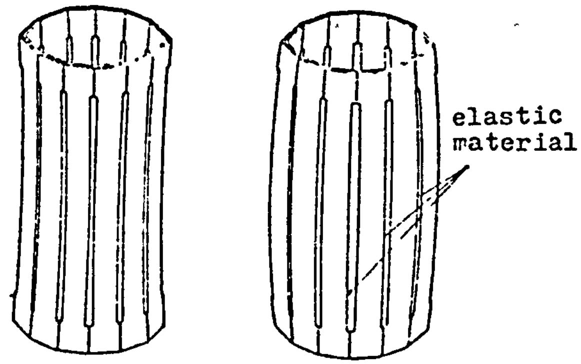
FIGURE 7
PUMP REQUIREMENTS

HEAT LOAD (kW)	TEMP (°C)	FLOW RATE (ℓ/min)		
		NH_3	R-11	H_2O
5	4	0.37	1.1	-
	20	0.42	1.1	-
	120	2.0	1.8	0.14
25	4	1.9	5.2	-
	20	2.1	5.5	-
	120	9.9	9.1	0.70
350	4	26	73	-
	20	29	77	-
	120	138	128	9.8

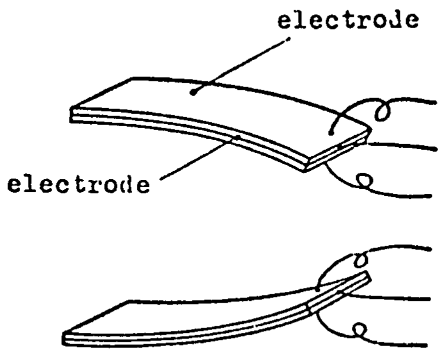
FIGURE 8
PERFORMANCE CHARACTERISTICS OF ORBITER PUMPS

PUMP	FLUID	FLOW (ℓ/MIN)	ΔP (kPa)
SPACELAB	WATER	3.90	134
ORBITER	WATER	7.23	372
SPACELAB	F-21	16.7	352
ORBITER	F-21	14.1	483

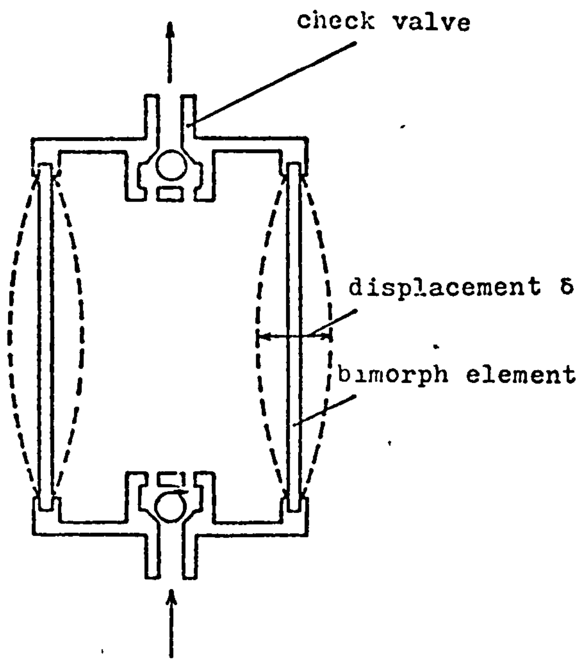
FIGURE 9 BIMORPH PUMP CONCEPT



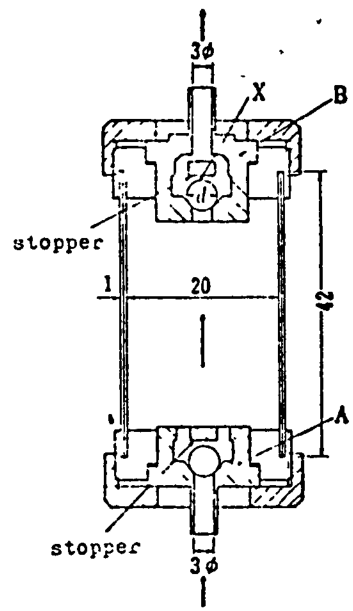
Cylindrical bimorph vibrator



Bimorph



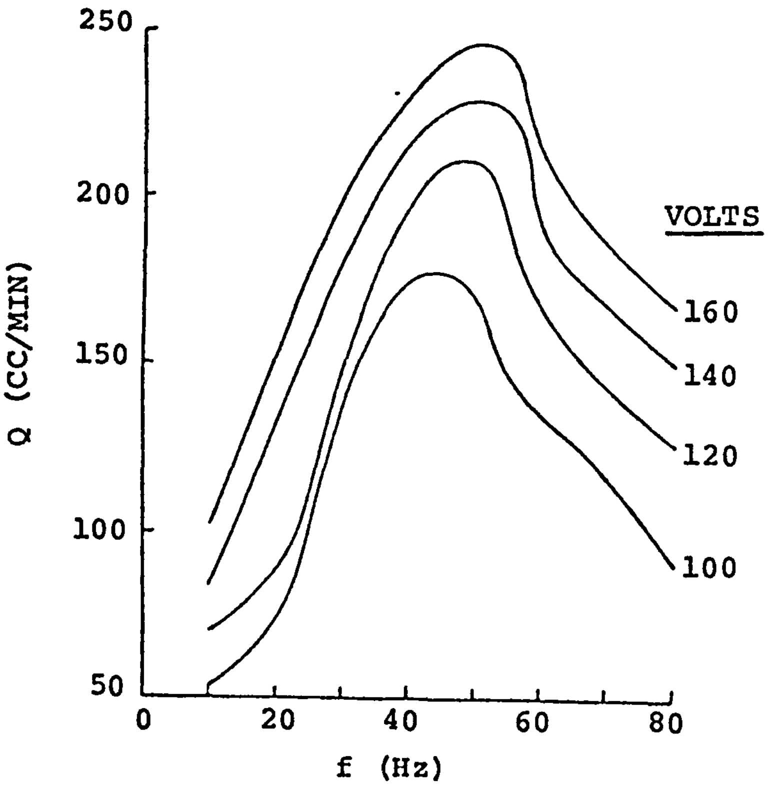
Bimorph vibrator pump



Cylindrical bimorph pump

FIGURE 10

EXPERIMENTAL CAPACITIES OF BIMORPH PUMP*



D = 2.0 cm, L = 4.2 cm

*REFERENCE 3

FIGURE 11

CONCEPT 2 - CAPILLARY PUMP DRIVEN

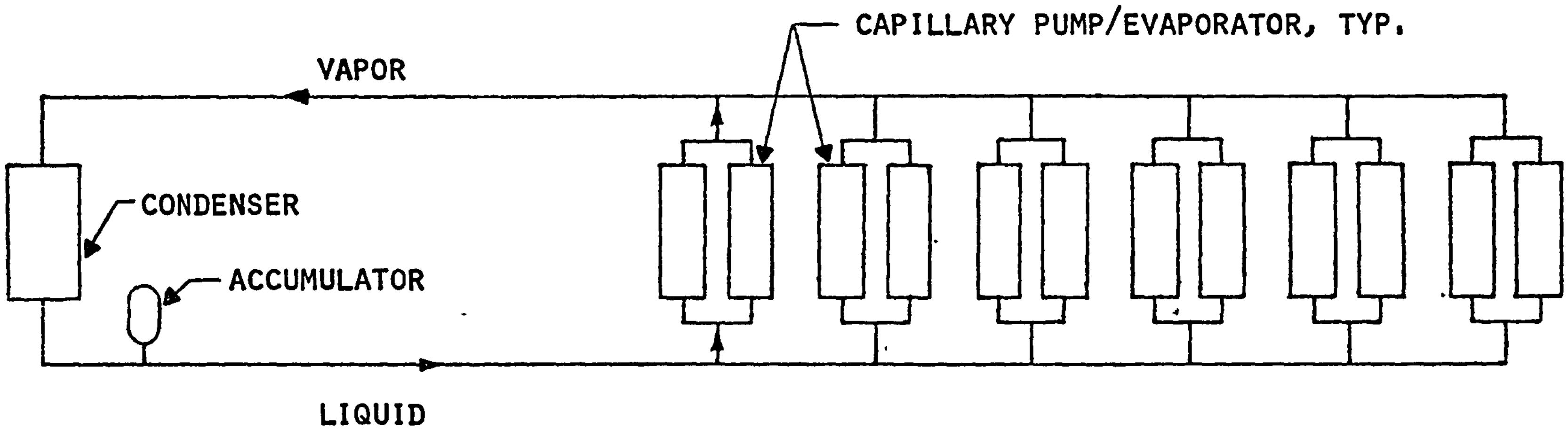
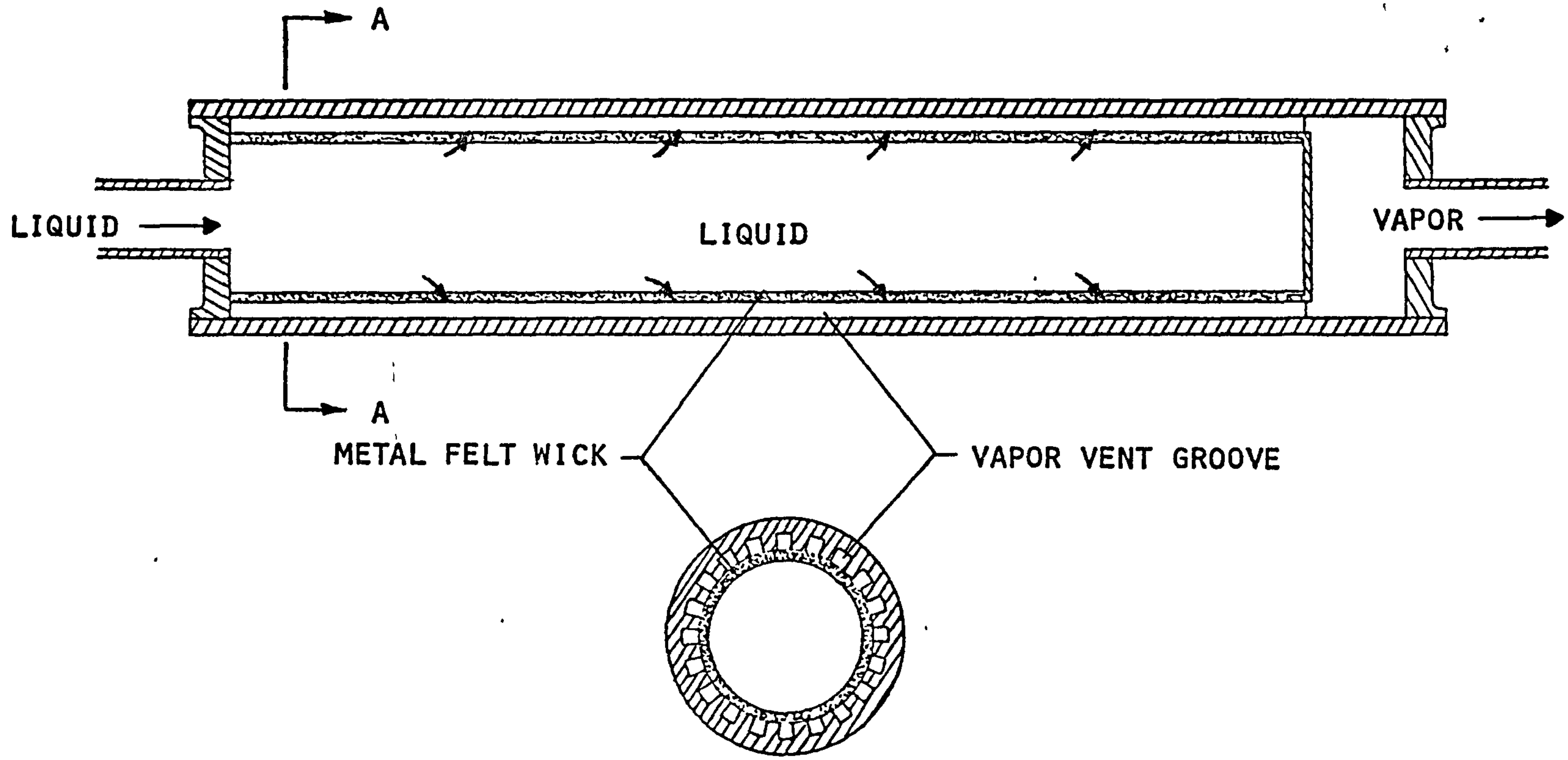


FIGURE 12
CAPILLARY PUMP/EVAPORATOR CONCEPT



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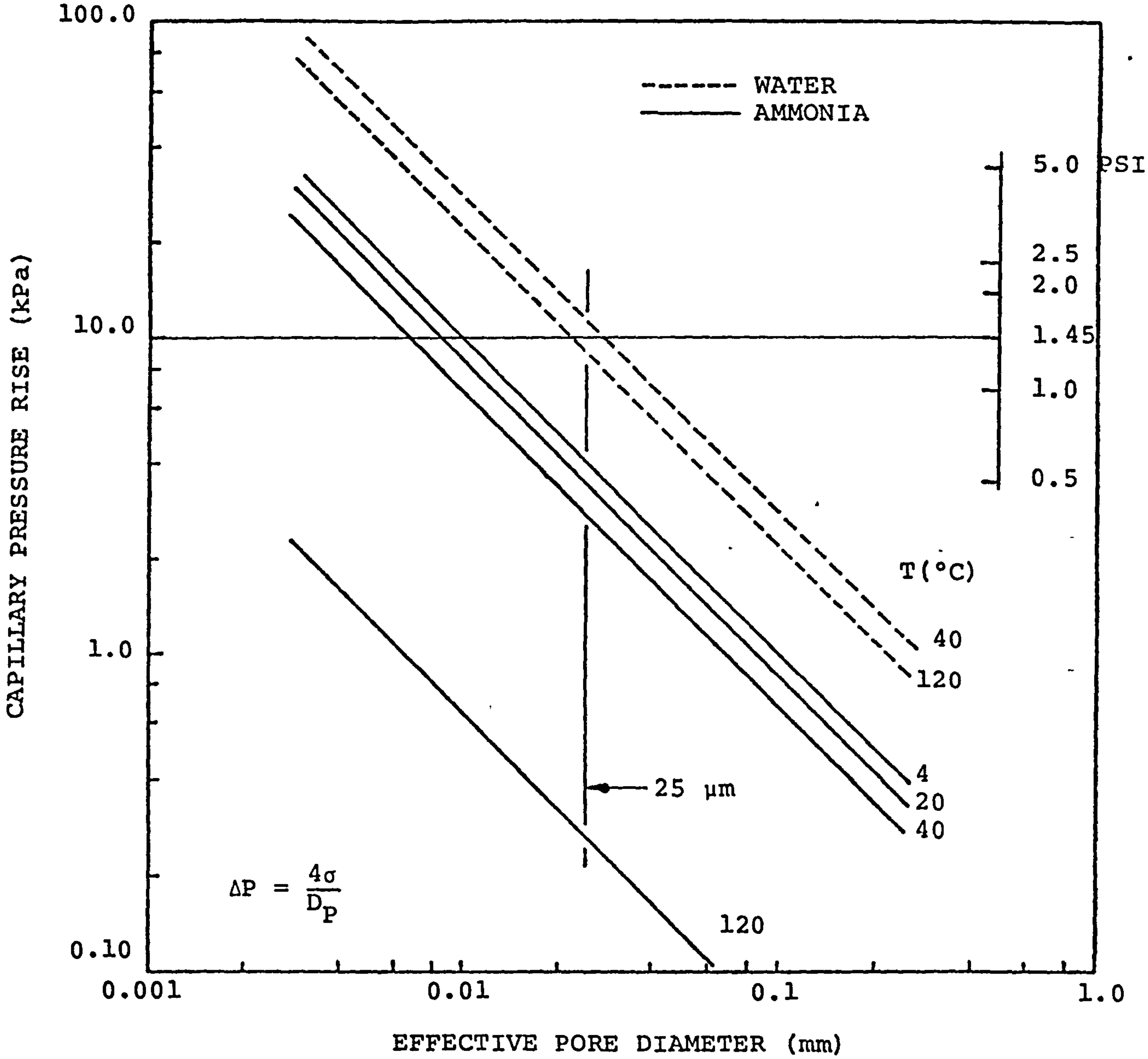


FIGURE 13
EFFECT OF WICK PORE DIAMETER ON CAPILLARY PRESSURE

as a function of pore diameter for ammonia and water at various temperatures. For this study an effective pore diameter of 25 μm (9.8×10^{-4} in) was assumed. This results in a pressure rise of approximately 3.5 kPa (0.5 lb/in^2) with ammonia and 10 kPa (1.5 lb/in^2) with water. A challenge with such a system is to design it such that the total pressure drop is lower than these values. This requires larger flow cross sectional areas than in the mechanically pumped concept.

4.3 CONCEPT 3 : OSMOTIC PUMP DRIVEN

The osmotic pump driven concept is illustrated in Figure 14. This concept employs direct osmosis as the driving force to circulate the working fluid. An osmotic pump module is located just upstream of each evaporator. This osmotic pump module consists of an osmotic membrane separating pure solvent on one side of the membrane and a solvent-solute mixture (solution) on the other side. The pure solvent flows through the semi-permeable membrane and into the concentrated solution due to direct osmotic forces. The membrane is impermeable to the solute. Large pressure differentials, orders of magnitude greater than that of capillary wicks, may be achieved with osmotic pumping. This provides the capability to transport large amounts of heat over long distances with a passive device.

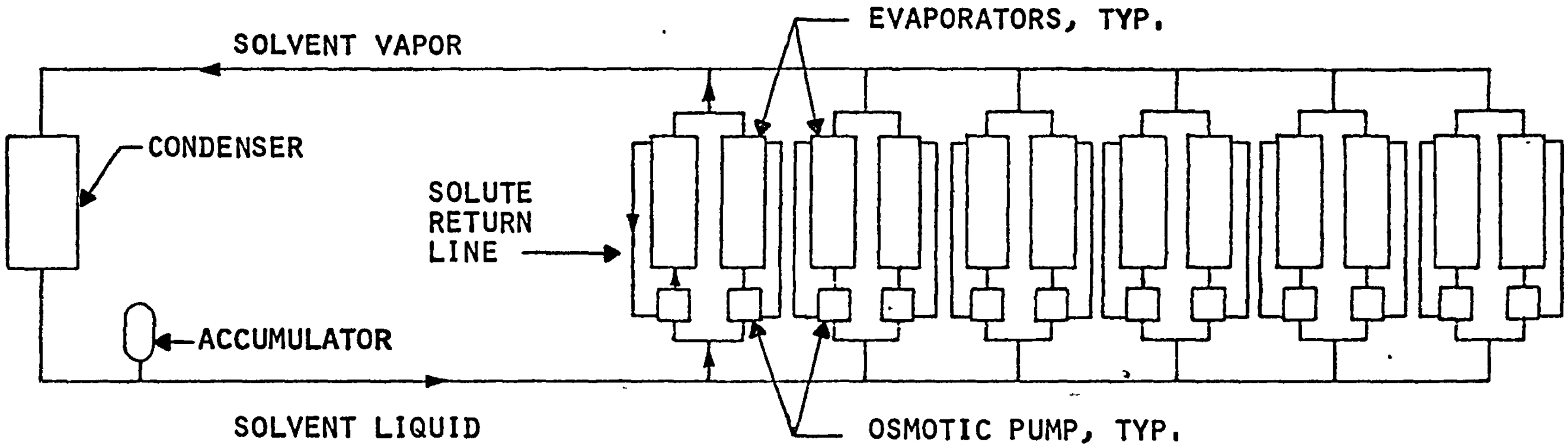
In the evaporator, the solvent is evaporated leaving behind the solute. A wick structure must be employed in the evaporator to prevent solute from being carried over with the solvent vapor and "poisoning" the system. The pumping force of the osmotic membrane is dependent on the concentration gradient across it. If solute is carried to the solvent side of the membrane this pumping action will be degraded as the concentration gradient is lessened. Eventually the concentration gradient could become so low that pumping would cease.

Extensive testing of osmotic heat pipes has been performed by Hughes Aircraft Co. under Air Force funding (References 5 and 6). During this study Hughes was given a subcontract to provide support in generating concepts for an osmotic thermal bus and in predicting performance and weights of such a system.

To date, all of the testing of osmotic heat pipes has been with cellulose acetate membranes, water as the solvent, and sucrose as the solute. This combination, however, is not compatible over the entire range of operating temperatures to be addressed in this study. The vapor pressure of water is too low for operation at temperatures much below 40°C (104°F). Sucrose breaks down and oxidizes at temperatures near 100°C (212°F). Also, the cellulose acetate membrane cannot withstand sustained operation at temperatures higher than 75°C (167°F).

For these reasons, an advanced technology approach is proposed which would extend the operating capability throughout the desired temperature range of 4°C to 120°C (39°F to 248°F). The recommended approach consists of a Polybenzimidazole (PBI) membrane, ammonia or methanol as the solvent, and aluminum sulfate as the solute. This approach will require extensive development before it can be implemented. It is felt that methanol and ammonia will be compatible with PBI. However, compatibility and membrane pumping experiments are recommended. A detailed investigation of solutes was not conducted. Aluminum sulfate was selected primarily because of it's good

FIGURE 14
CONCEPT 3 - OSMOTIC PUMP DRIVEN



solubility and separation characteristics with water. It's solubility and separation characteristics with methanol and ammonia must be investigated, however.

In order to apply osmotic heat pipe technology to space applications, a means for operating in a zero-g environment must be devised. With the non-existence of gravitational forces to contain the liquid, the capillary attraction forces of wicking material is required to eliminate any free-floating or unattached liquid. In addition, a means is necessary to actively direct the concentrated solution at the evaporator towards the membrane where it can mix with dilute solution and increase the concentration in the vicinity of the membrane to promote increased osmotic pressure. In one-g operation this function is carried out by free-convection currents induced by gravitational forces. Several techniques have been considered to induce return of solute to the osmotic membrane during zero-g operation. The two most promising of these are an electrostatic technique and a displacement technique.

The electrostatic technique is illustrated in Figure 15. By using an electrolytic solute such as aluminum sulfate, a voltage could be imposed across the membrane to maintain the solute concentration in the vicinity of the membrane surface. The required voltages are very small. Figure 16 shows the voltage required for several electrolytes. This concept is not a circulation technique but merely a means of holding the solute ions in place. Consequently it requires little or no power consumption. Any slight current leakage would be due to secondary currents in the electrolyte and could be determined experimentally.

If the solvent flow can be preferentially channelled (e.g. path of least resistance) then the solvent pumping velocity can be used to drive the solution circulation. As the solvent flows through the membrane into the solution compartment, it carries a portion of the solution with it toward the evaporator. If a return line is provided, new solution will be drawn into the module to maintain continuity. Likewise, when the dilute solution reaches the evaporator the solvent will evaporate and the resulting concentrated solution will be displaced from the evaporator. Figure 17 illustrates an evaporator concept that will provide for return of the concentrated solution back to the pump module. It is similar in construction to the capillary pump described earlier.

Another consideration for 0-g operation is the control technique. A flow control technique is necessary since the osmotic pumping rate is not coupled to the heat input rate as in a capillary pumped heat pipe. Without a means of control, the solvent will continue to flow at a constant rate into the solution compartment even after the heat input is reduced or shut off. If this is allowed to happen, the solution will eventually be forced through the wick and into the solvent compartment.

Three possible techniques for 0-g control are solvent depletion, control valve, and potential difference. The most simple and direct approach is to size the solvent condensate volume such that it can be stored in the evaporator or an accumulator on the solution side of the membrane under a no-load condition. Pumping will stop when all of the solvent has been pumped into the solution side. A control valve may be used to regulate the flowrate if it is located on the solvent side of the loop. Otherwise the solvent would

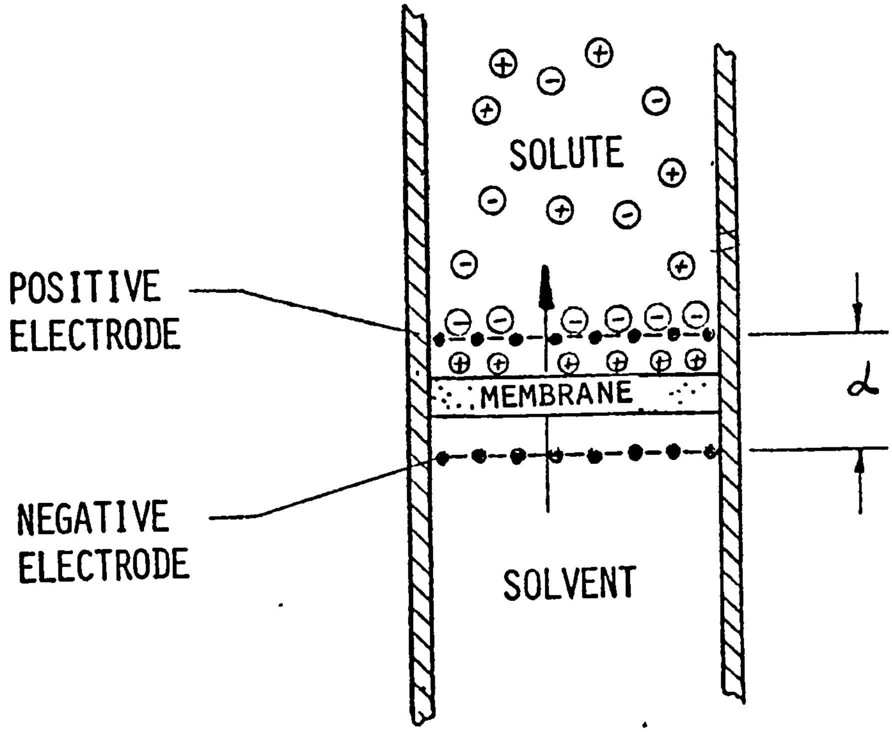


FIGURE 15

ELECTROSTATIC TECHNIQUE FOR SOLUTE RETENTION

ION	VOLTAGE*
K^+	0.092
Mg^{++}	0.127
Na^+	0.135
Li^+	0.175
SO_4^{--}	0.084
NO_3^-	0.095

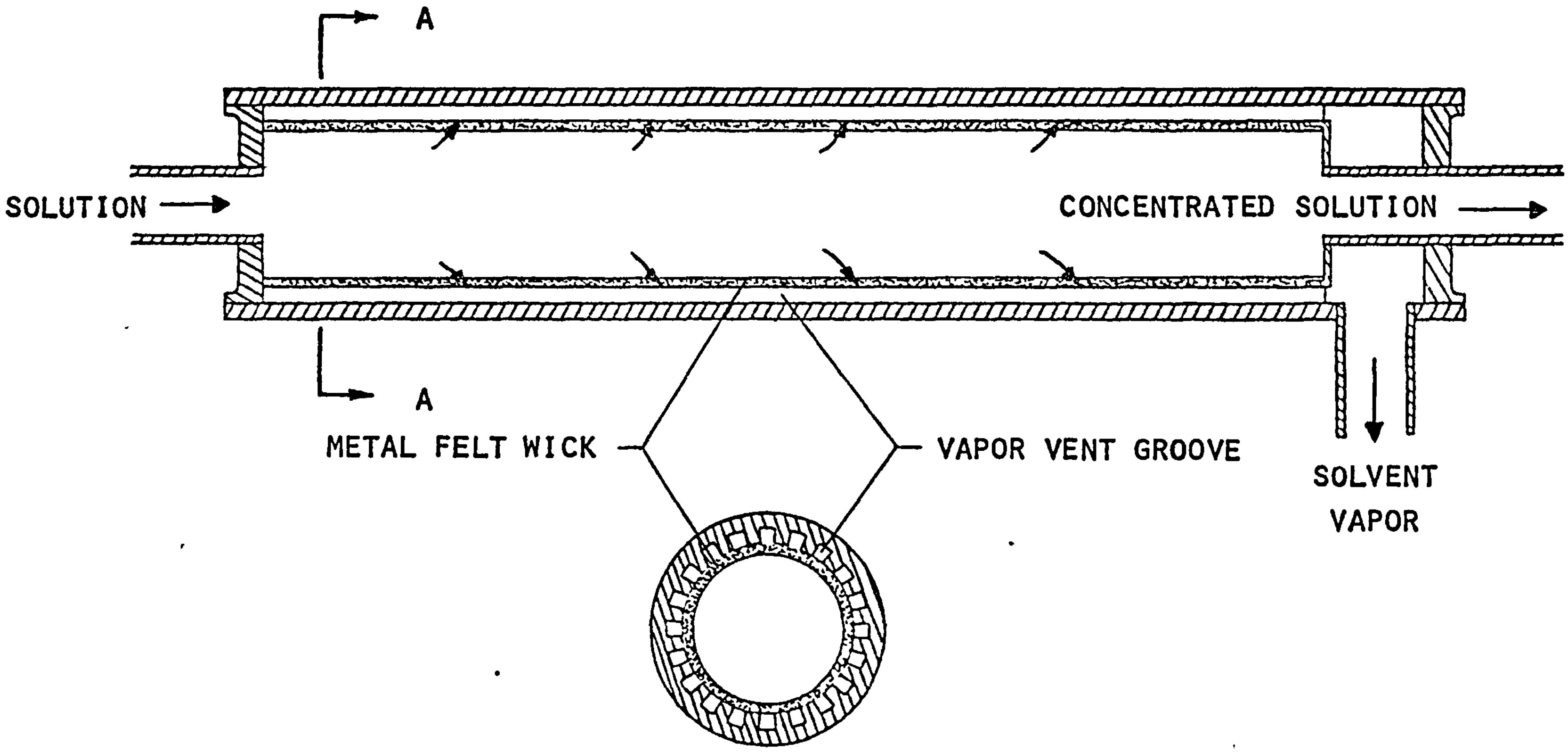
*ESTIMATED VOLTAGE REQUIRED TO COUNTERACT SOLVENT VELOCITY, $J_V = 7 \times 10^{-5}$ CM/SEC.

FIGURE 16

VOLTAGE ESTIMATED FOR ELECTROSTATIC TECHNIQUE

FIGURE 17

OSMOTIC PUMPED THERMAL BUS EVAPORATOR CONCEPT



SECTION A-A

continue to flow across the membrane until the full osmotic pressure of several hundred psi is reached. This could damage the membrane or membrane seals. Finally, if the electrostatic method of solute retention is used, the applied voltage could be varied to control the concentration of ions near the membrane surface and therefore control the flowrate of solvent across the membrane. Although each of these control concepts appear feasible for on/off control, their ability to accurately control flowrates at reduced loads is uncertain and would have to be verified by laboratory experiments.

In order to reduce weight, it is desirable to reduce the volume of the osmotic membrane modules. This may be accomplished by increasing the ratio of membrane surface area per unit volume. Two types of compact osmotic membrane modules, spiral wound and hollow fiber, have been investigated. Figure 18 illustrates the increased performance per unit volume of these types of module over a shell-and-tube type. Figures 19 and 20 illustrate the spiral wound and hollow fiber modules respectively.

4.4 CONCEPT 4 : COMPRESSOR DRIVEN

The compressor driven concept is illustrated schematically in Figure 21. This concept employs a vapor compressor, located just upstream of the condenser, to circulate the working fluid. The advantage of this concept over the mechanical pump driven is that the condenser is at the highest pressure (and therefore the highest temperature) point in the loop rather than the lowest pressure point. This results in reduced radiator area and weight. A disadvantage however, is the increased power required by the compressor compared to a pump. Temperature control in this concept, as in the previous ones, is maintained by a pressurized accumulator.

Because of the nature of vapor compressors it is necessary to avoid a liquid/vapor mixture entering the compressor. This requires a wicked evaporator to separate the liquid and vapor phases. The proposed evaporator design for this concept is the same as for the capillary pump shown previously in Figure 12. In order to control the pressure differential across the wick so that liquid is not forced into the vapor passage, a pressure regulator is required as illustrated in Figure 21.

4.5 RADIATOR CONTROL CONCEPTS

In order to assure that the liquid entering the evaporators is saturated or slightly subcooled under varying loads and radiator sink temperatures, a means of radiator or heat rejection control must be devised. Otherwise, at low loads or low sink temperatures, the liquid returning from the radiator/condenser interface could become subcooled to the point that evaporation would not take place in the evaporators and the thermal bus would no longer operate as a isothermal, two-phase device. Two concepts proposed for radiator control with any of the thermal bus concepts are a passive regenerative concept and an active radiator bypass control concept.

The passive regenerator concept is illustrated in Figure 22. Vapor flowing to the condenser first flows through a regenerative heat exchanger where it reheats the subcooled return liquid to a slightly subcooled or saturated state. At the design maximum heat rejection level, the temperature of the liquid leaving the condenser will be only slightly lower than the vapor temperature and no heat transfer will occur in the regenerator. However, at

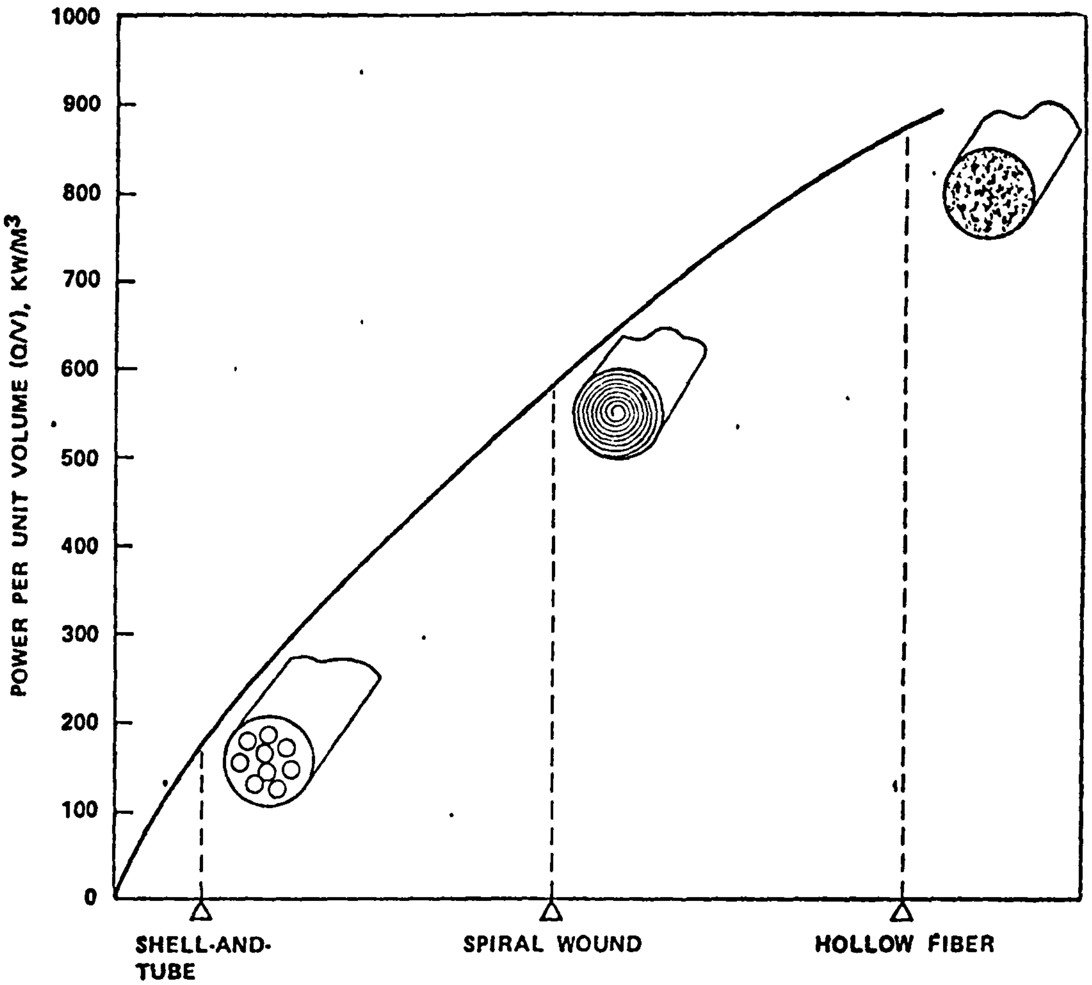
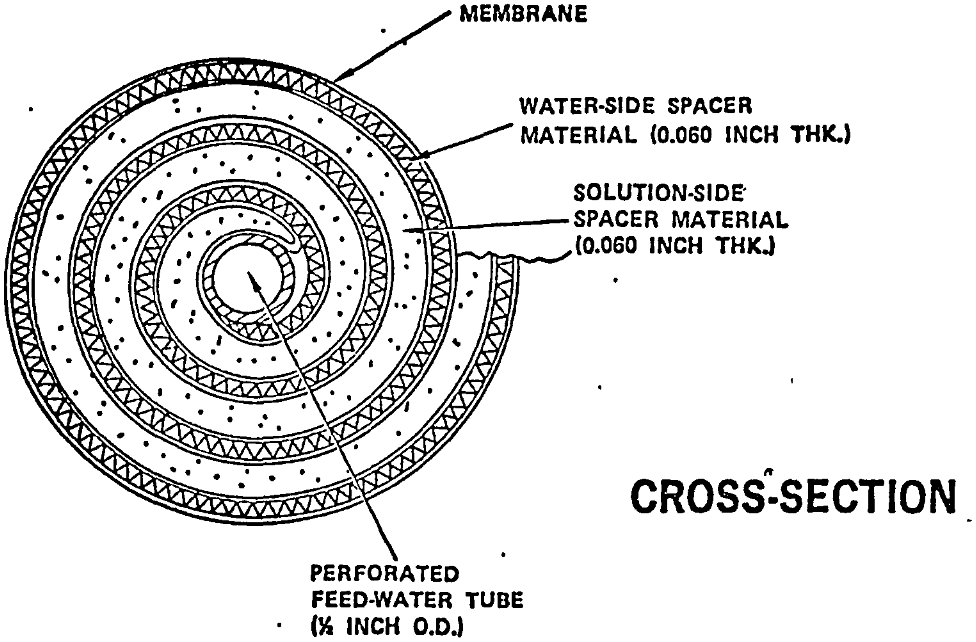


FIGURE 18

STATE-OF-THE-ART MEMBRANE MODULES



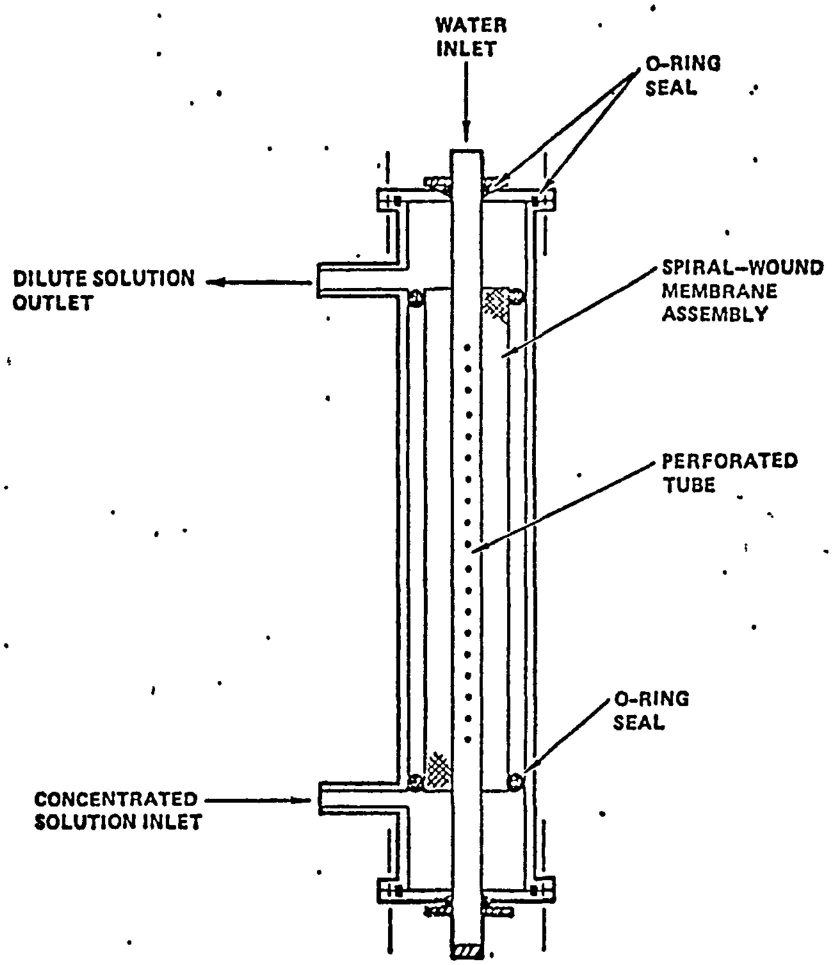
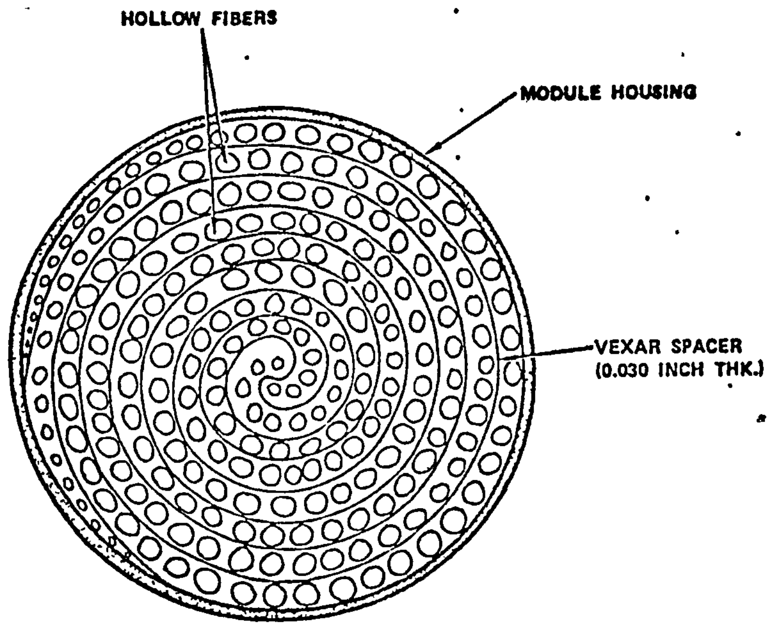


FIGURE 19

SPIRAL-WOUND MODULE CONSTRUCTION



CROSS-SECTION

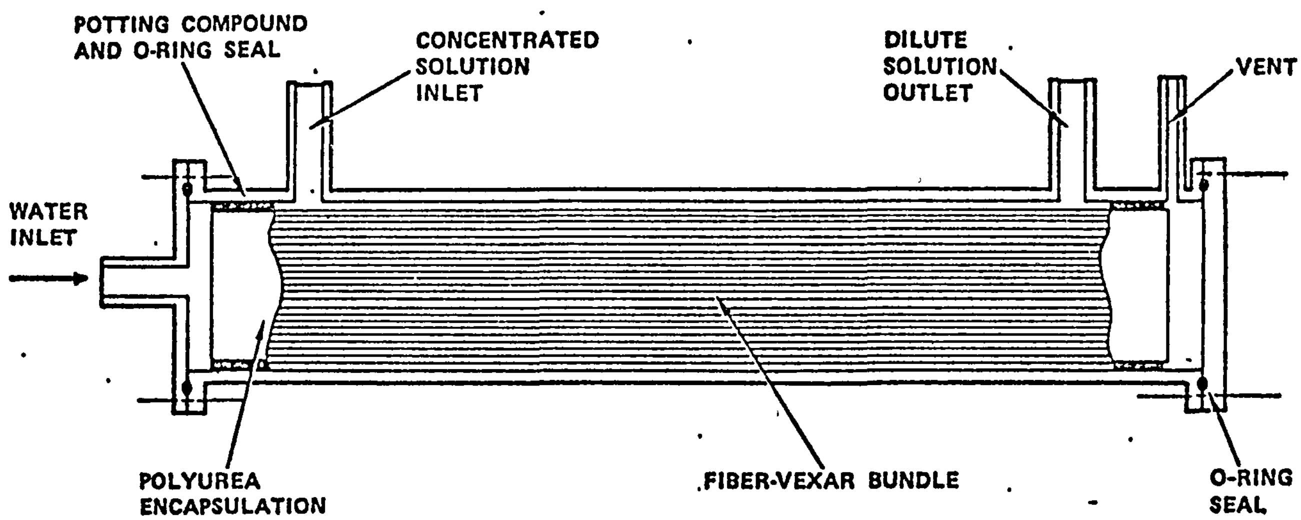
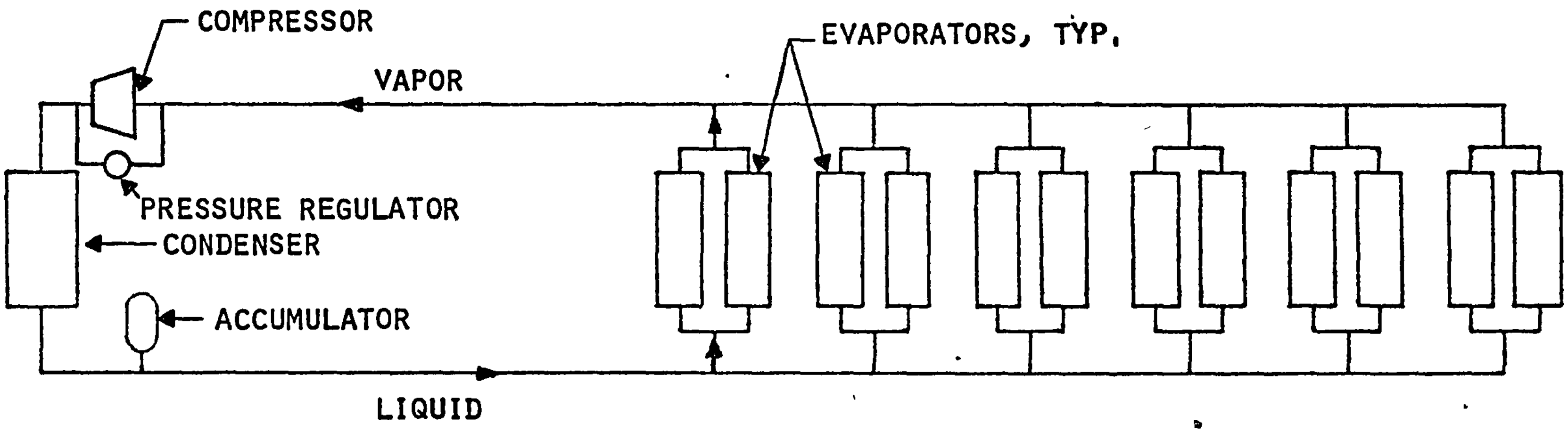


FIGURE 20

HOLLOW-FIBER MODULE CONSTRUCTION

FIGURE 21
CONCEPT 4 - COMPRESSOR DRIVEN



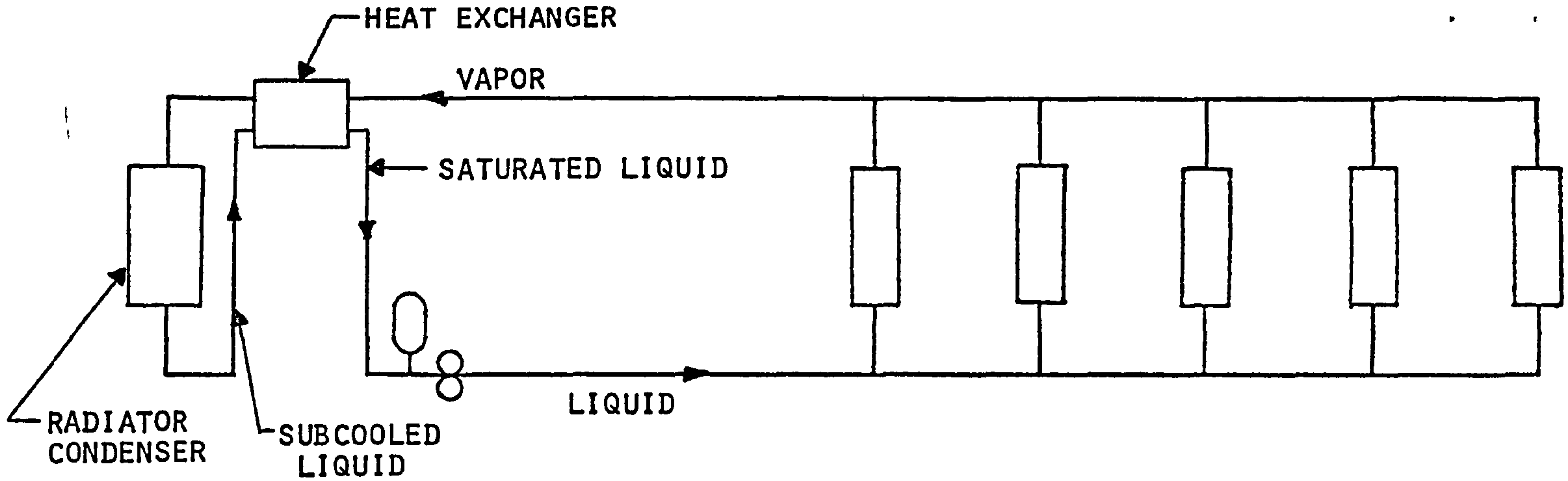


FIGURE 22 PASSIVE REGENERATOR HEAT REJECTION CONTROL

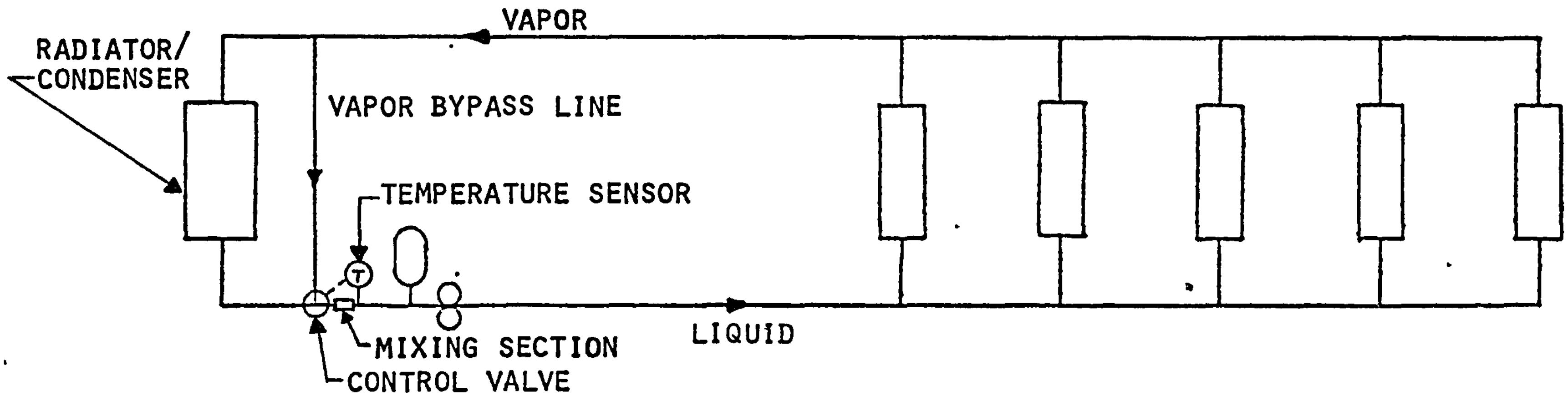


FIGURE 23 ACTIVE RADIATOR BYPASS CONTROL

a reduced heat load the condenser return will be subcooled and heat will be transferred to the liquid return. As a result, some of the vapor will be condensed in the regenerator and the vapor entering the condenser will be at a lower quality. This in turn results in an even lower liquid outlet temperature and, therefore, a lower average radiator temperature. The radiator will eventually reach a steady state temperature that will allow the radiator heat rejection to match the heat input at the evaporators. This concept offers promise of a passive, highly reliable means of heat rejection control, however, the method is untested and would require some development. Because of its passive nature this concept was baselined for all thermal bus concepts.

An alternate, active radiator bypass control concept illustrated in Figure 23 uses an active control valve to bypass some of the vapor by the radiator to mix with the liquid condenser return. A temperature sensor located downstream of a mixing chamber measures the mixed liquid temperature and an electronic controller adjusts the amount of bypass to control this temperature to a slightly subcooled value. This concept is less reliable than the regenerative method and would require development of a control valve to handle two phase flow.

5.0 TRADE STUDY RESULTS

The four thermal bus concepts surviving the Task I screening were included in parametric, system-level trade studies during Task II. System level considerations such as heat rejection penalties and electrical power penalties were included in order to obtain a fair comparison between the concepts since radiator temperature and pump power are both functions of pressure drop. A heat rejection penalty of 5.9 kg/m^2 (1.2 lb/ft^2) was imposed based on the required radiator plan area. The radiator area was based on the following assumptions: a two-sided radiator, equivalent sink temperature = -40°C (-40°F), fin effectiveness = 0.90, emissivity = 0.76 (silver Teflon coating), and a 5°C (9°F) temperature drop between the condenser and the radiator. A power penalty of 159 kg/kW$_E$ (350 lb/kW$_E$) was assumed based on current power system designs. In addition, the compressor driven concept was also evaluated at a power penalty of 45 kg/kW$_E$ (100 lb/kW$_E$) in order to evaluate the effect of low-weight, advanced technology power systems on this concept. System weights were calculated parametrically as a function of thermal bus size (heat transport capacity and transport length), temperature, transport line diameters, and working fluid. Pressure drops, power requirements, radiator area, weights, and 10-year reliabilities were calculated. A trade matrix was developed in order to compare the thermal bus concepts based on these quantitative as well as qualitative elements.

Four sizes of thermal bus were evaluated for each concept. The heat loads for these sizes ranged from 5 to 350 kW and the thermal bus length varied from 19m to 50m (62 to 164 ft.). The number, arrangement, and maximum heat input of the payload interfaces are shown in Figures 24 through 27 for each of the four cases. Thermal bus evaporator temperatures assumed were 4, 20, 40, and 120°C. Working fluids evaluated included ammonia, Freon 11, and water. Figure 28 presents a matrix that shows the design points at which each concept was evaluated.

5.1 WEIGHT AND SIZING ANALYSIS

At each design point, thermal bus weight was calculated as a function of vapor line diameter in order to find the weight optimum diameter. For simplicity in calculations, the liquid line diameters were sized such that the liquid flow pressure drop equaled the vapor flow pressure drop. The weight calculated included the line weight, fluid weight, radiator weight, and power penalty weight. Heat exchanger and pump weights were not included at the time since they were not functions of line diameter. After the weight optimum line size was determined, the heat exchanger, pump, and accumulator weights were calculated to include in the total thermal bus system weight. For the capillary pump driven concept, the weight optimum line diameter usually resulted in a pressure drop greater than the pressure rise available with capillary pumping. In this case the minimum diameter that resulted in a suitably low pressure drop was determined, as discussed in Section 5.1.2.

The results of the optimum weight calculations are presented in Figure 29. The conclusions that can be reached from this figure are:

1. Ammonia is the best working fluid (lowest thermal bus weight) over the entire temperature range.

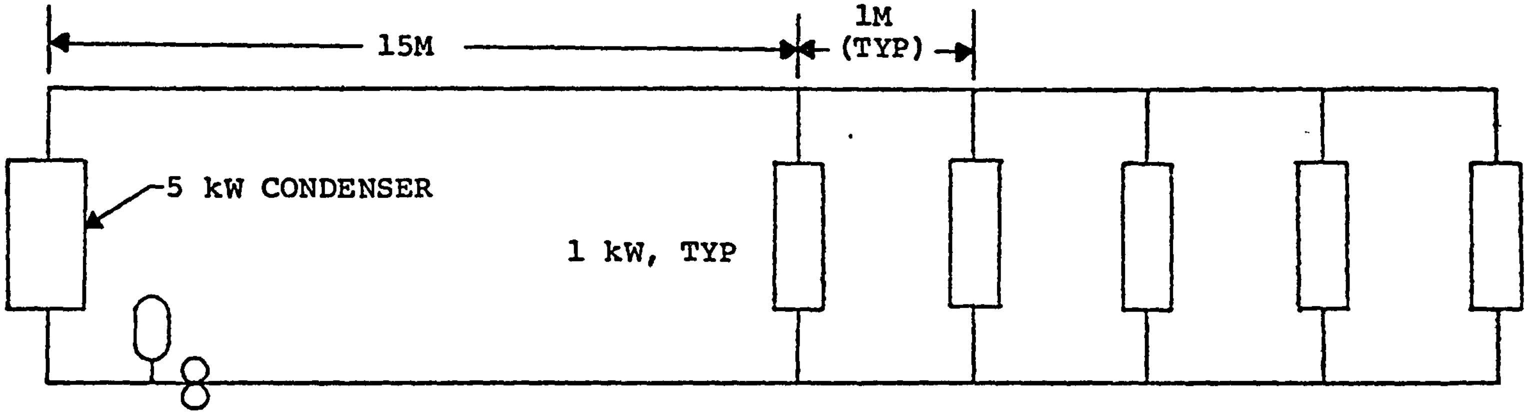


FIGURE 24 CASE 1 - 5 kW BUS

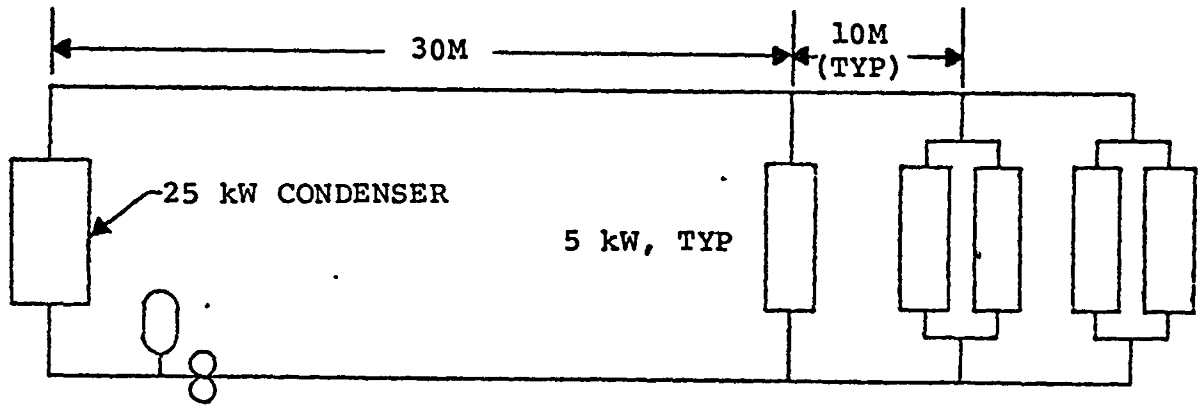


FIGURE 25 CASE 2 - 25 kW BUS

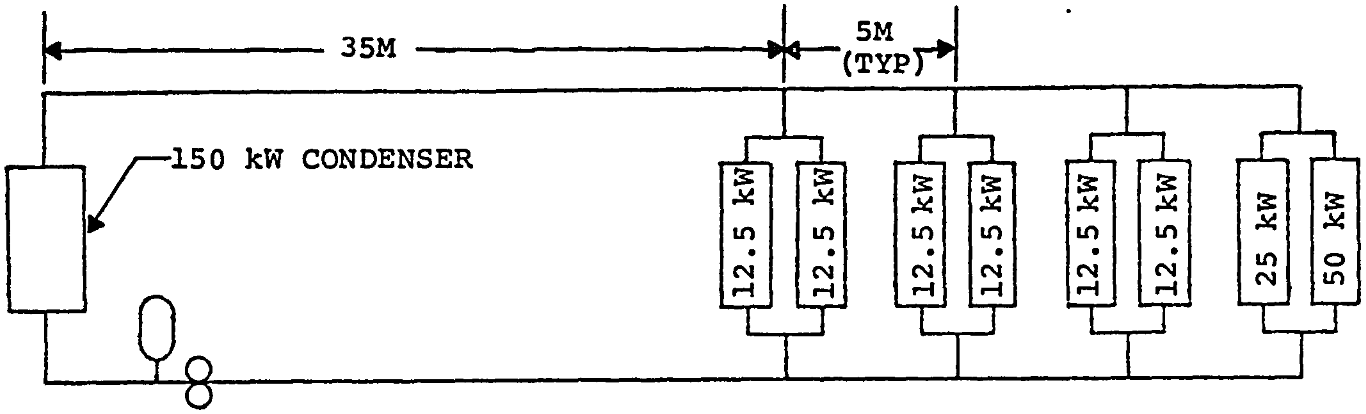


FIGURE 26 CASE 3 - 150 kW BUS

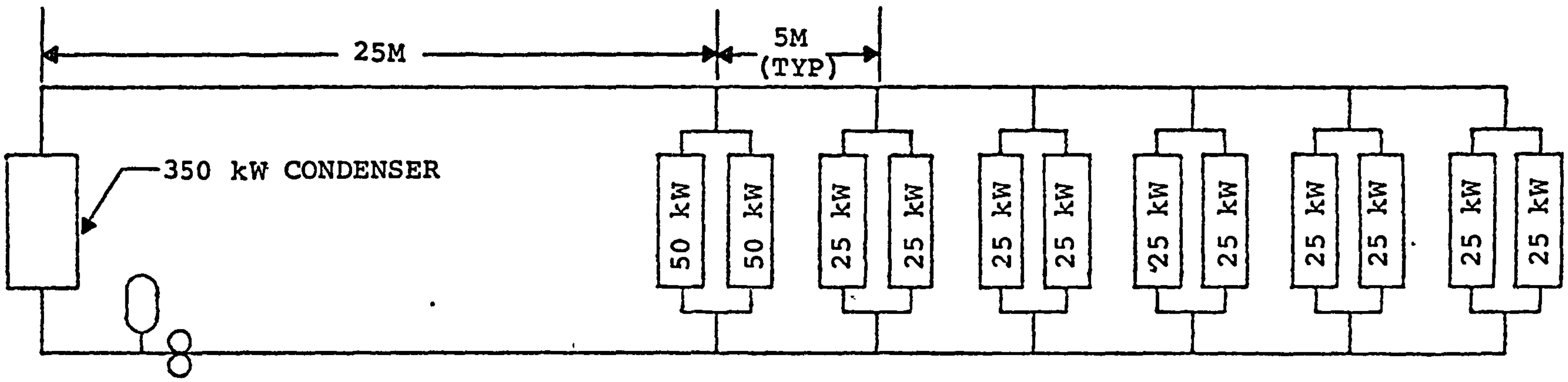


FIGURE 27 CASE 4 - 350 kW BUS

FIGURE 28 - DESIGN POINTS EVALUATED

CASE	CONCEPT	AMMONIA				FREON 11				WATER	
		4°C	20°C	40°C	120°C	4°C	20°C	40°C	120°C	40°C	120°C
1 - 5 kW	1	X	X	X	X	X	X	X	X	X	X
	2	X	X	X	X	X	X	X	X	X	X
	3	X	X	X						X	X
	4A	X	X	X	X	X	X	X	X	X	X
	4B	X	X	X	X	X	X	X	X	X	X
2 - 25 kW	1	X	X	X							
	2	X	X	X							
	3	X	X	X							
	4A										
	4B										
3 - 150 kW	1	X	X	X							
	2	X	X	X							
	3	X	X	X							
	4A										
	4B										
4 - 350 kW	1	X	X	X	X	X	X	X	X	X	X
	2	X	X	X	X	X	X	X	X	X	X
	3	X	X	X						X	X
	4A	X	X	X	X	X	X	X	X	X	X
	4B	X	X	X	X	X	X	X	X	X	X

CONCEPT 1 = MECHANICAL PUMP
CONCEPT 2 = CAPILLARY PUMP
CONCEPT 3 = OSMOTIC PUMP
CONCEPT 4A = COMPRESSOR, 159 kg/kWe
CONCEPT 4B = COMPRESSOR, 45 kg/kWe

FIGURE 29 - THERMAL BUS SYSTEM WEIGHTS

CASE	CONCEPT	THERMAL BUS SYSTEM WEIGHT, KG AMMONIA 4°C	20°C	40°C	120°C	FREON 11 4°C	20°C	40°C	120°C	WATER 40°C	120°C
1 - 5 kW	1	168	111	76	32	196	134	92	44	108	40
	2	166	110	75	37	187	129	94	52	97	39
	3	182	125	91						107	49
	4A	169	112	76	31	187	129	92	44	121	46
	4B	169	114	77	31	180	128	91	43	107	44
2 - 25 kW	1	848	556	372							
	2	849	566	389							
	3	911	624	447							
	4A										
	4B										
3 - 150 kW	1	4999	3268	2181							
	2	5001	3311	2253							
	3	5440	3770	2690							
	4A										
	4B										
4 - 350 kW	1	11582	7578	5047	1890	13705	9018	6128	2690	6929	2225
	2	11557	7620	5154	2585	12952	8861	6415	3492	5724	2166
	3	12643	8700	6266						6337	2807
	4A	11543	7606	5113	1922	12605	8607	6068	2786	6619	2317
	4B	11020	7485	5036	1871	11664	8263	5858	2691	5861	2239

CONCEPT 1 = MECHANICAL PUMP
CONCEPT 2 = CAPILLARY PUMP
CONCEPT 3 = OSMOTIC PUMP
CONCEPT 4A = COMPRESSOR, 159 kg/kWe
CONCEPT 4B = COMPRESSOR, 45 kg/kWe

2. With ammonia, the concept weights are within 5% of each other except for the osmotic concept which is 10-24% heavier.

3. The capillary pumped concept is heavy at 120°C with both ammonia and Freon 11 but is the lowest weight concept with water.

4. If the low power penalty (45 kg/kW) is assumed, the vapor compression concept is the lowest weight in all cases except with water.

A detailed breakdown of weights, line sizes, power, and radiator area for each concept and design point are presented in the Appendix.

5.1.1 Mechanical Pump Driven

The mechanical pump driven thermal bus was sized as discussed in the previous section. In calculating weights, it was assumed that the evaporator was a flow-through heat exchanger such as a shell-and-tube type. Only the weight of the tubes was calculated; the payload side of the heat exchanger was not included. The assumption of a flow-through heat exchanger required a larger accumulator than would have been necessary with a wicked evaporator. Since under a low or no-load condition the vapor passage can be almost completely filled with liquid, the accumulator was sized large enough to contain the entire vapor volume of the thermal bus. In calculating pump weight it was assumed that there were two centrifugal pumps with one being redundant.

5.1.2 Capillary Pump Driven

As mentioned previously, the weight optimum line diameters for the capillary pump driven concept usually resulted in a pressure drop larger than the capillary pump could overcome. By plotting the thermal bus pressure drop as a function of vapor line diameter, the minimum diameter that would still allow capillary pumping could be found. In calculating the maximum pressure rise available with capillary pumping, a 25 μm (9.8×10^{-4} in) pore diameter was assumed. Metal felt wicks of this pore size are readily obtainable.

Figures 30 through 33 illustrate the effect of line diameter on the capillary pump thermal bus weight for different bus sizes assuming ammonia as the working fluid. The weights shown include lines, fluid, power penalty, and radiator penalty. Also shown in the figures is the point at which pressure drop equals capillary pressure rise. At diameters larger than these values, capillary pumping is achievable. It is evident from these figures that, although line diameters much larger than the optimum values are required, the impact on weight is small due to the flatness of the curves. Part of the reason for this flatness is that the radiator weight is dominant and the line weight is a relatively small percentage of the total. Also, the increase in line weight with increasing diameter is partly offset by the decreasing power penalty and radiator weights which occur because of reduced pressure drop.

5.1.3 Osmotic Pump Driven

Experimental performance data was used by Hughes to scale-up and predict osmotic pump performance and sizes. Baseline design parameters were established from experimental data of a prototype Hughes osmotic heat pipe

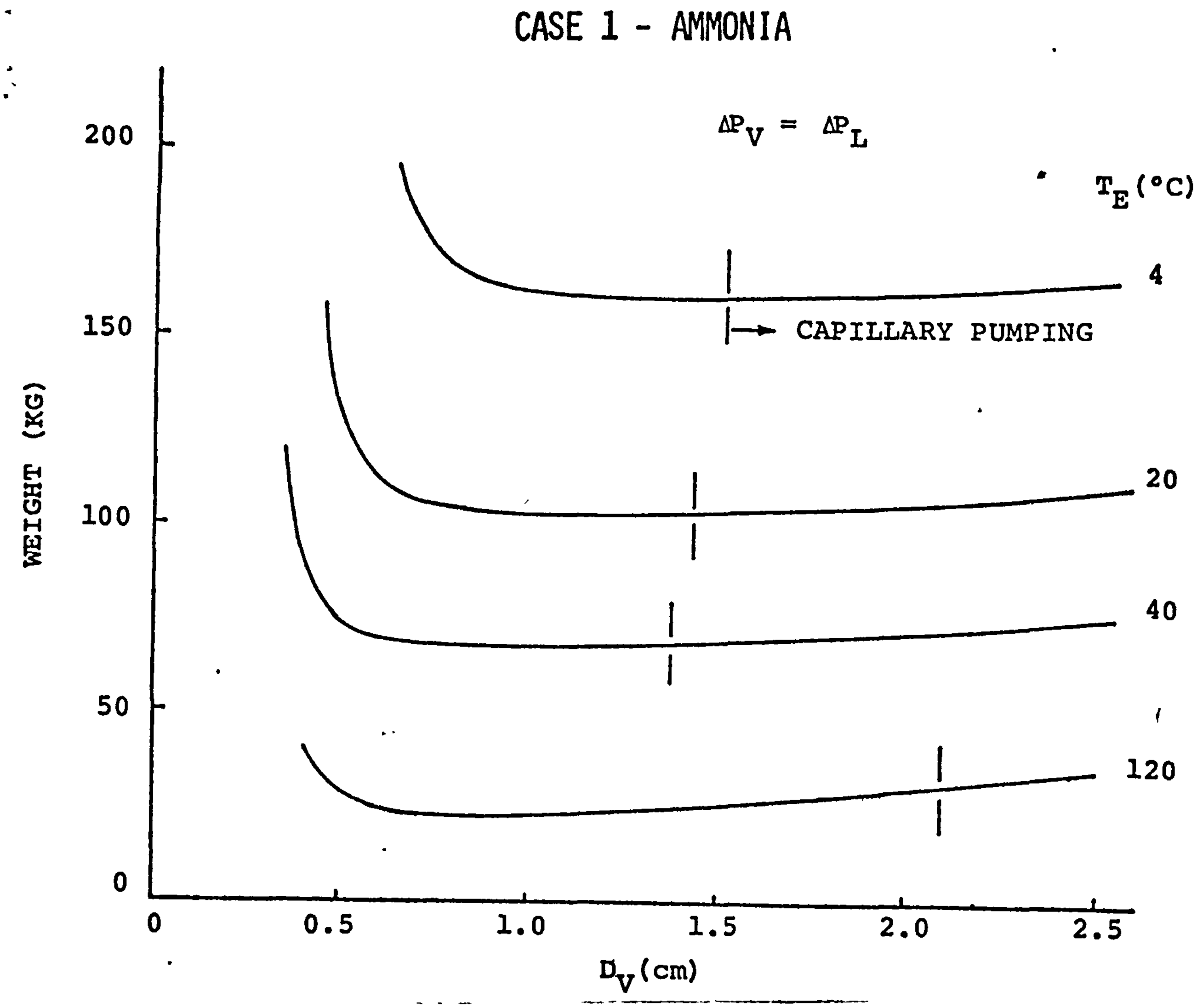


FIGURE 30
EFFECT OF LINE DIAMETER ON CAPILLARY PUMPED THERMAL BUS WEIGHT

CASE 2 - AMMONIA

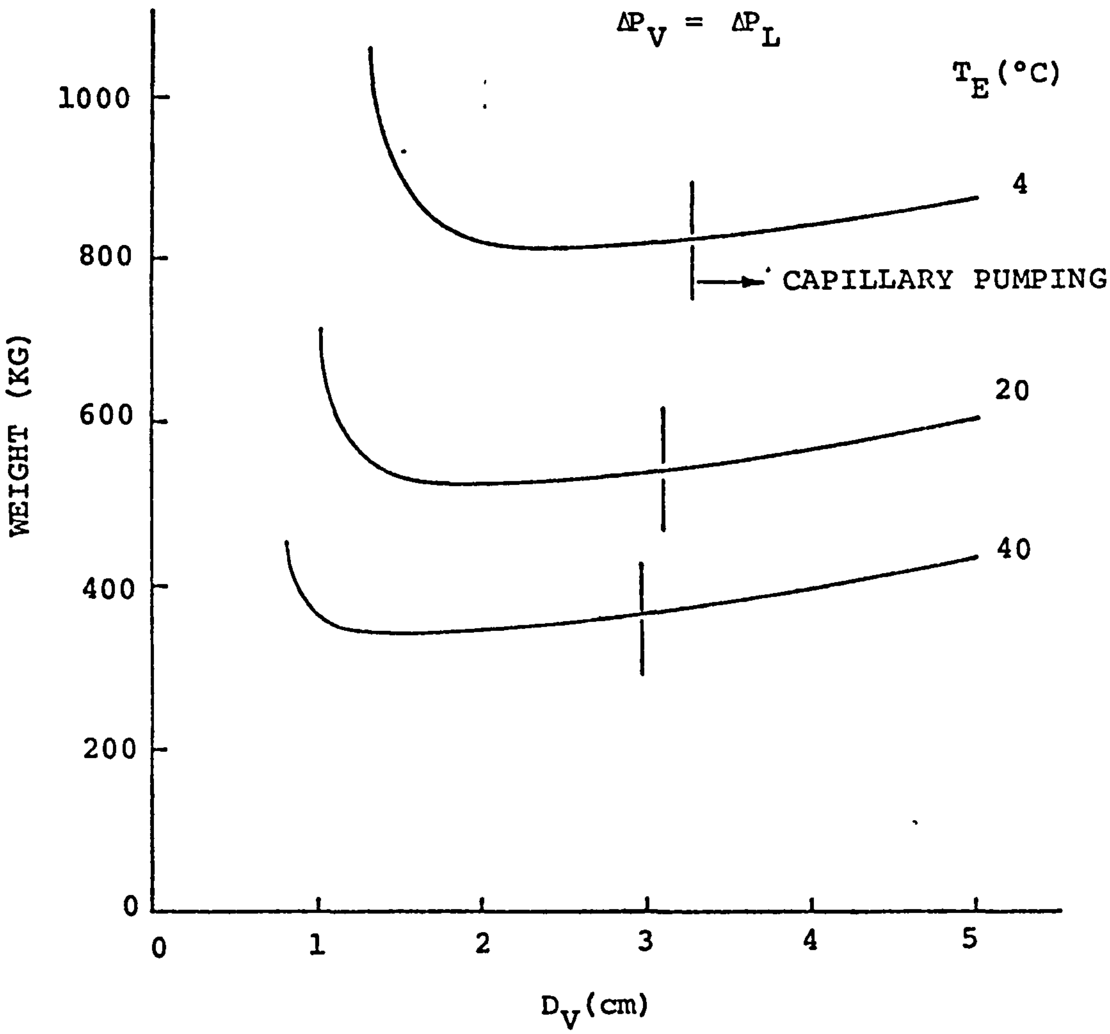


FIGURE 31
EFFECT OF LINE DIAMETER ON CAPILLARY PUMPED THERMAL BUS WEIGHT

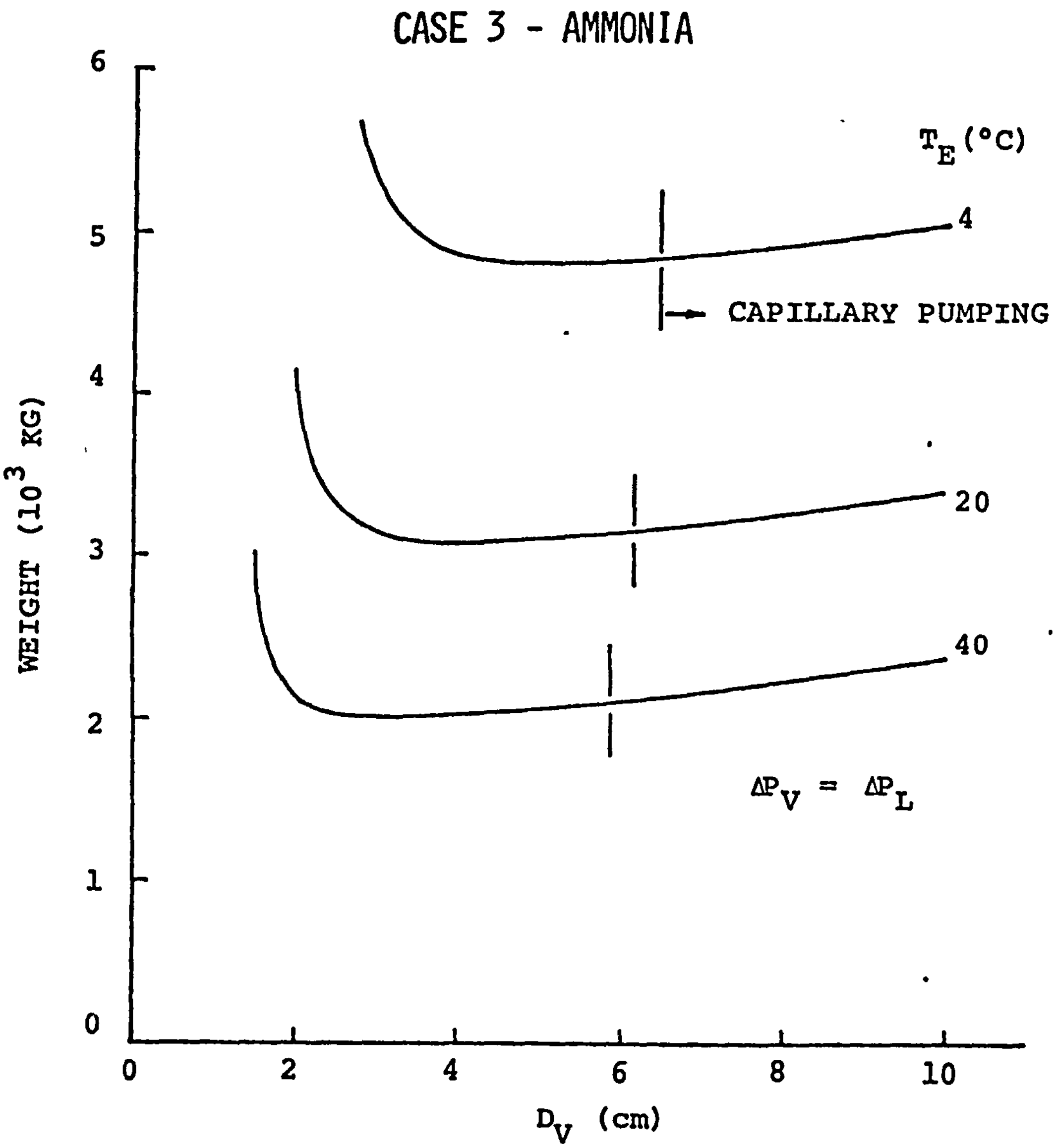


FIGURE 32
EFFECT OF LINE DIAMETER ON CAPILLARY PUMPED THERMAL BUS WEIGHT

CASE 4 - AMMONIA

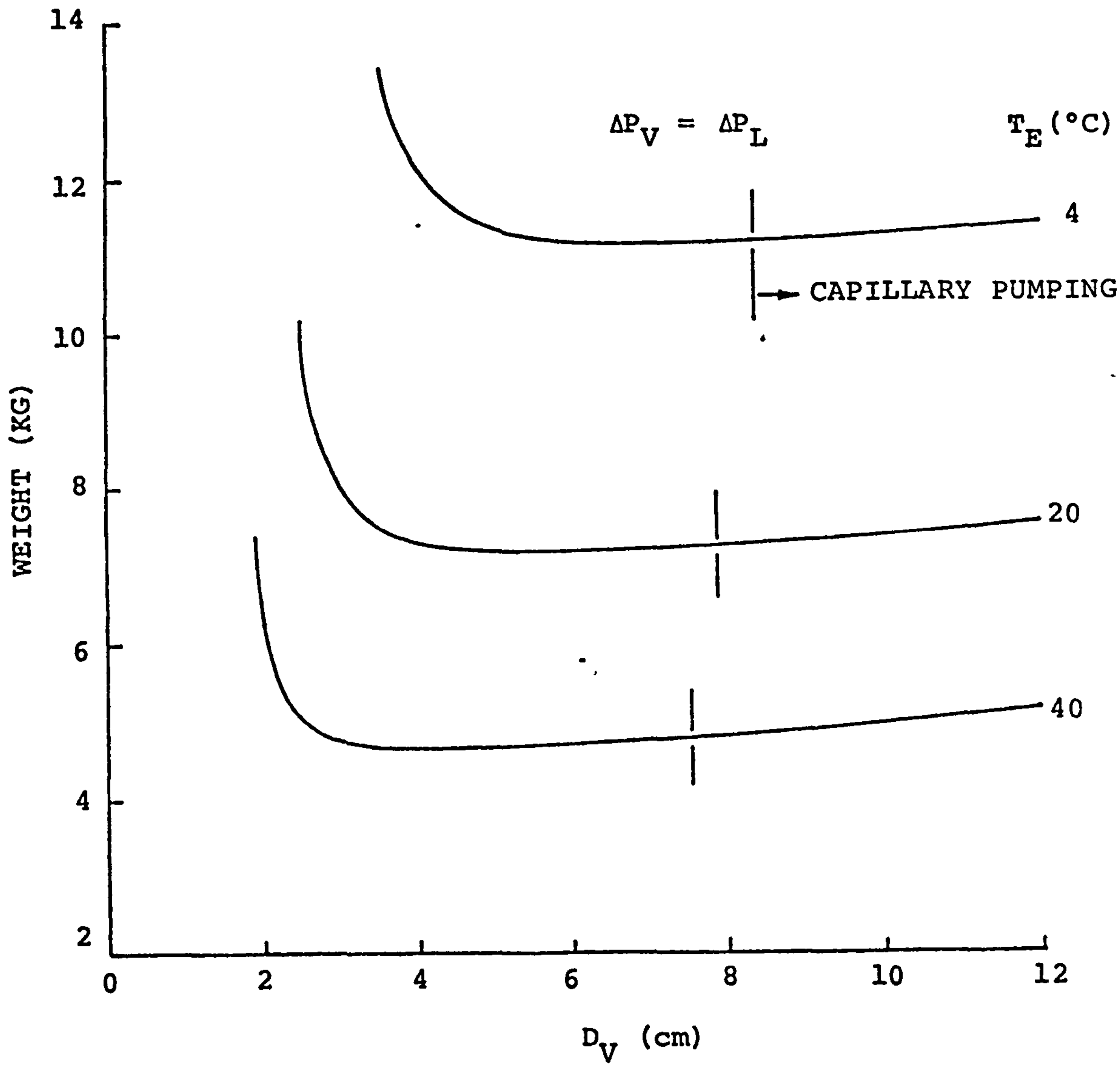


FIGURE 33
EFFECT OF LINE DIAMETER ON CAPILLARY PUMPED THERMAL BUS WEIGHT

containing a cellulose acetate, spiral-wound membrane and a 1 molality (molality is defined as the number of moles of solute per 1000g of solvent) water/sucrose working solution (Reference 6). Figure 34 shows that the osmotic pressure rise with such a combination is approximately 2.8×10^6 N/m^2 (400 lb/in^2). The membrane was contained in a cylindrical module 7.32 cm (2.88 in) in diameter and 28.58 cm (11.25 in) in length. The unit was tested to 500 watts in 1-g at temperatures of 40 to 60°C (104 to 140°F). Dimensional and detail information of a more compact hollow-fiber membrane module was used in conjunction with performance data from the spiral-wound module to obtain a second set of baseline design parameters. Inherent in this approach is the assumption of a viable solution circulation technique to permit equal pumping rates across a unit membrane area. Although this is an optimistic assumption, it does portray the improvement expected from future development. Performance predictions at temperatures below 20°C (68°F) introduces the evaluation of ammonia as a working fluid and corresponding new membrane materials. Sufficient osmotic pump performance data with these fluids is not available. The expectation of successful future developments justifies assuming new membrane/solution combinations performing equivalent to the baseline. The baseline units were constructed of copper and were not weight optimized. For this study it was assumed that the module container was made of stainless steel and wall thicknesses were based on containment of the working fluid vapor pressure.

Osmotic pump performance and size predictions were performed for both the spiral-wound and hollow fiber designs. Scaling up for heat load increases was accomplished by keeping the module length constant and increasing module diameter as necessary. Figures 35 and 36 present the results of module sizing for the spiral-wound and hollow fiber designs, respectively. The compactness advantage of the hollow fiber construction is evident in the smaller volume and weight. Figures 37 and 38 present the module component weights for the spiral-wound and hollow fiber designs, respectively, with water as the working fluid. For the purposes of this study the hollow-fiber module design was assumed in calculating the osmotic thermal bus weights. The osmotic pump weights used for the various evaporator heat loads are presented in Figure 39.

5.1.4 Compressor Driven

In sizing the compressor driven thermal bus concept, two electrical power penalties were used. The standard 159 kg/kW_E penalty (used for the mechanical pumped concept also) was meant to be typical of current power system designs. The lower 45 kg/kW_E penalty was used to evaluate the effect of potential advanced technology power systems on this concept. Figures 40 through 42 compare the weights of the compressor driven concept with these two power penalties. Also shown for comparison is the mechanical pumped concept weights. These figures show that with the 159 kg/kW_E power penalty, the compressor driven concept offers no significant weight advantage over the mechanical pumped concept and, in fact, is heavier in most cases. However, with the 45 kg/kW_E power penalty, the compressor driven concept does offer a weight advantage over the mechanical pump concept in some cases. The most significant weight advantage is at the low design temperature (4°C), where the optimum diameter also varies significantly. At low temperatures, close to the assumed sink temperature, a change in radiator temperature has a larger effect than at higher radiator temperatures. Reducing the flow passage diameters increases the pressure drop which both decreases the radiator weight (by increasing radiator temperature) and increases the power penalty weight.

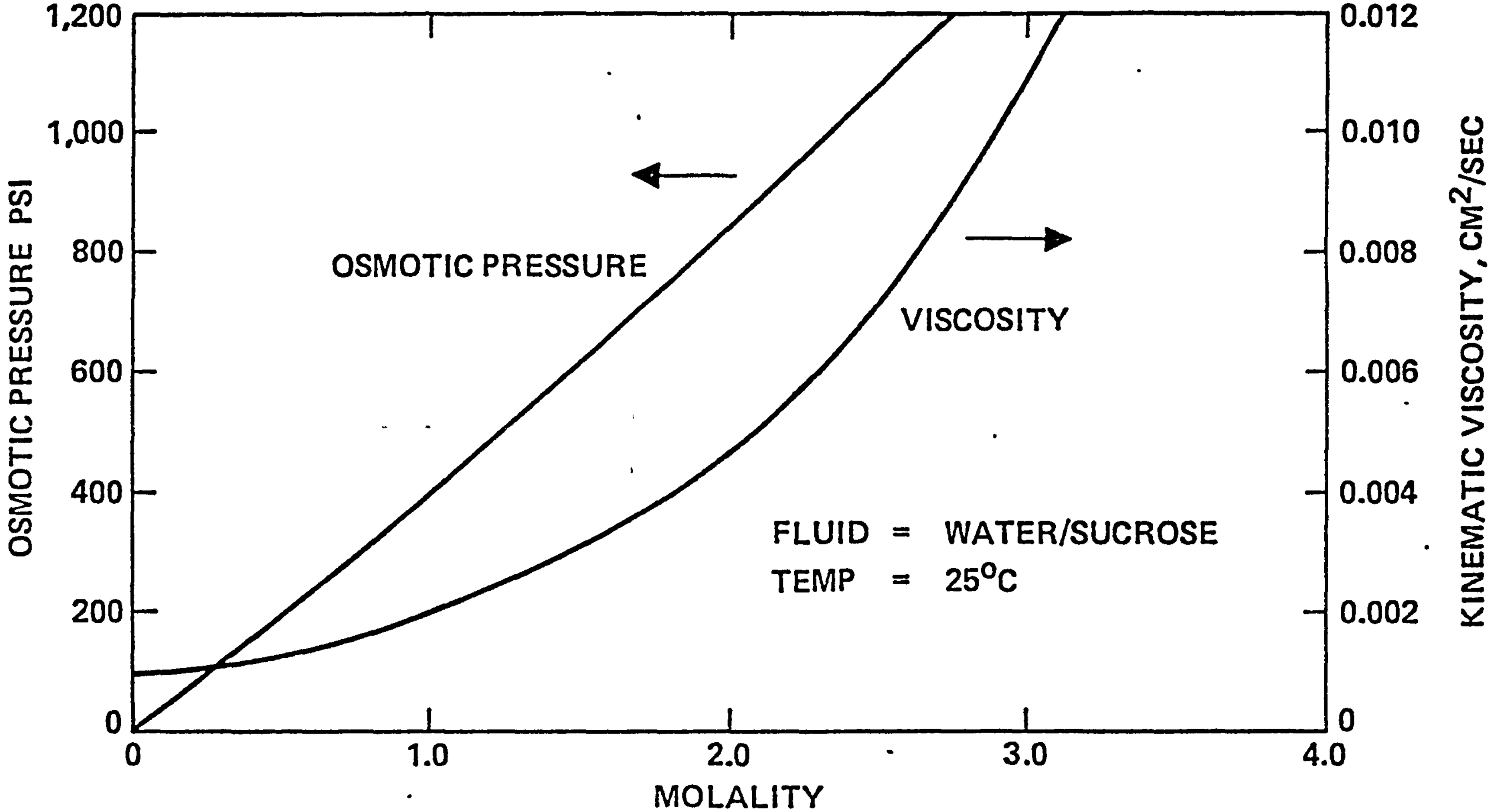


FIGURE 34 OSMOTIC PRESSURE HEAD AND VISCOSITY

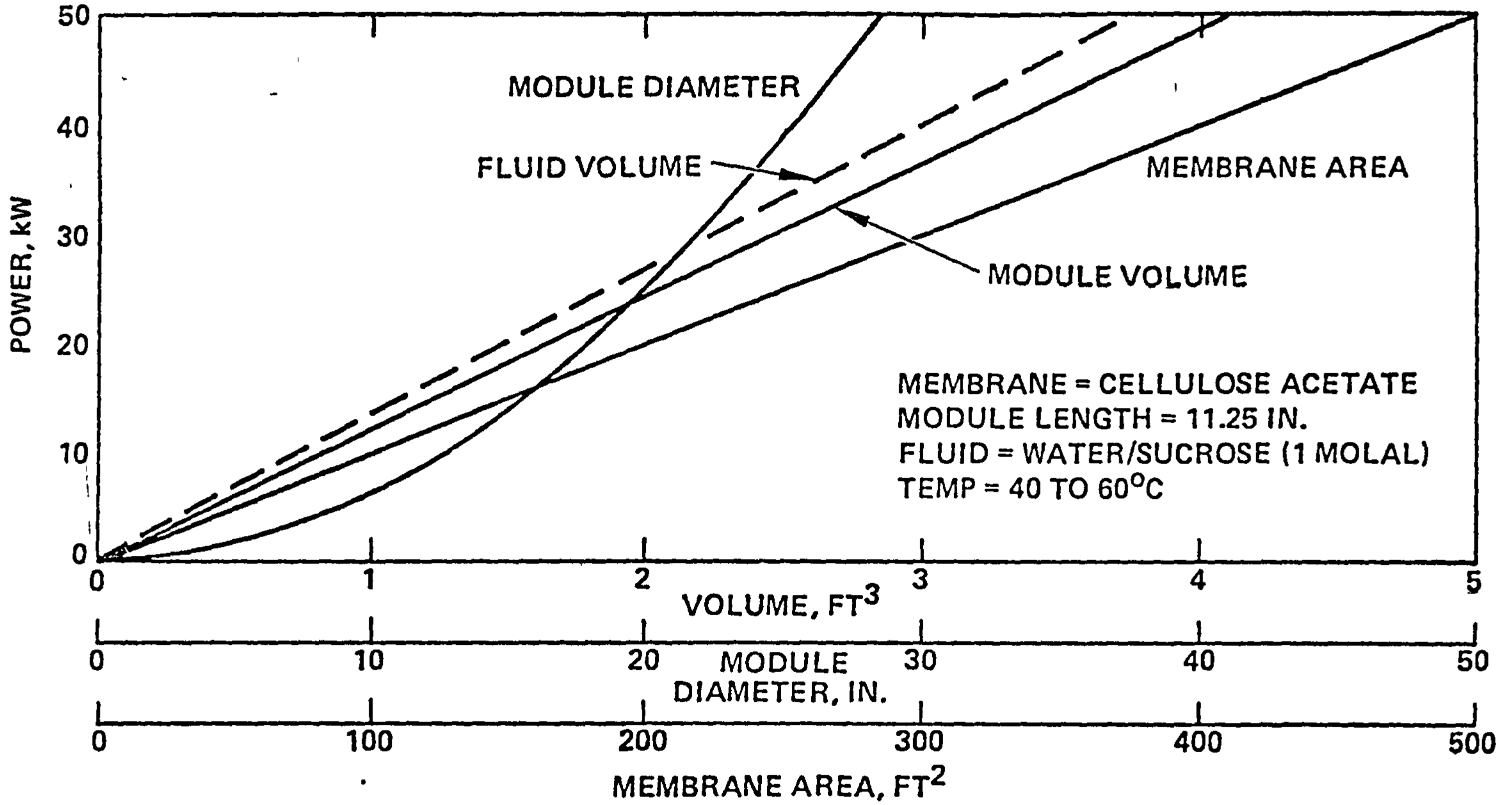


FIGURE 35 SPIRAL-WOUND MODULE SIZE

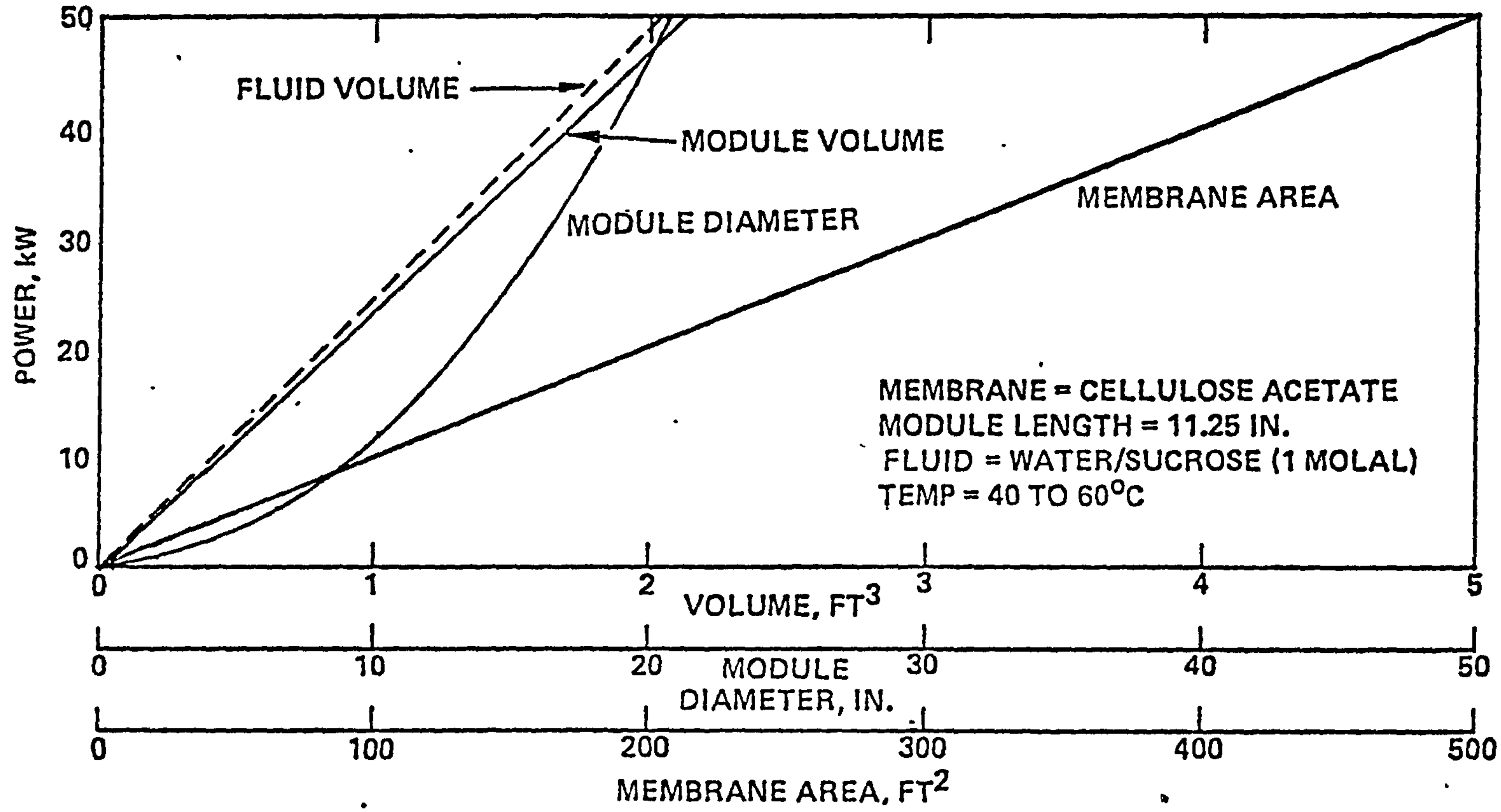


FIGURE 36 HOLLOW-FIBER MODULE SIZE

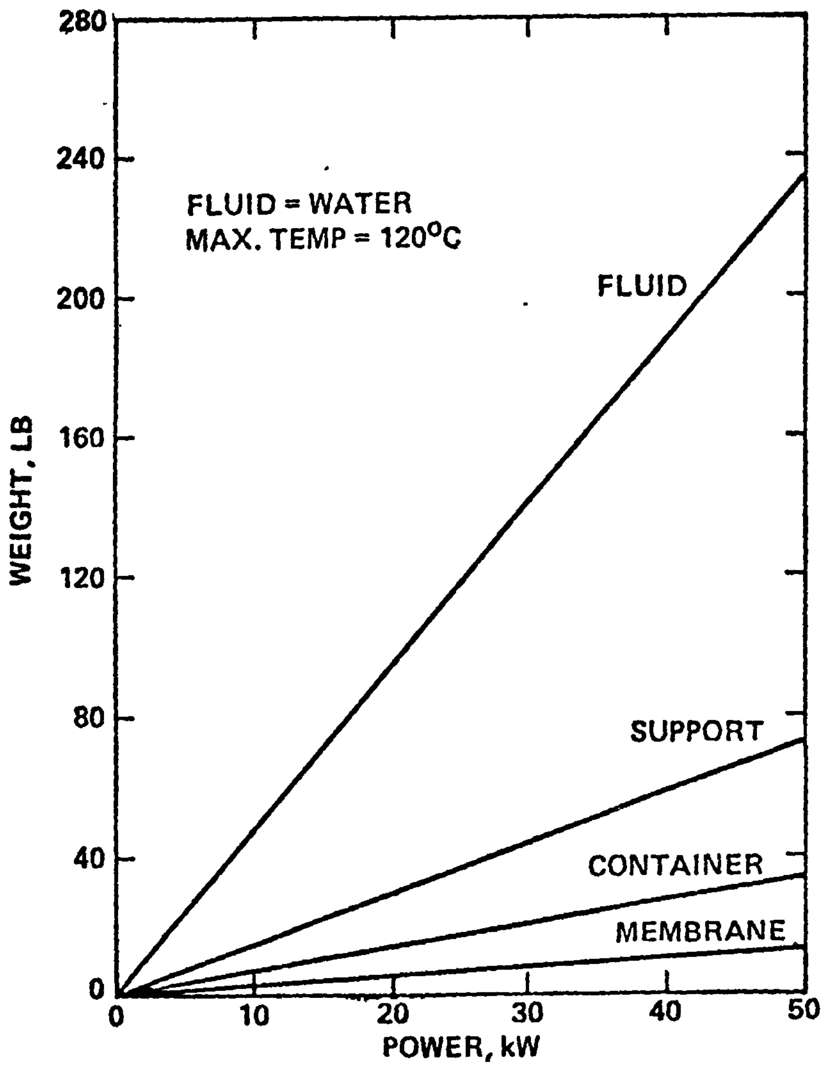


FIGURE 37

SPIRAL-WOUND MODULE COMPONENT WEIGHTS

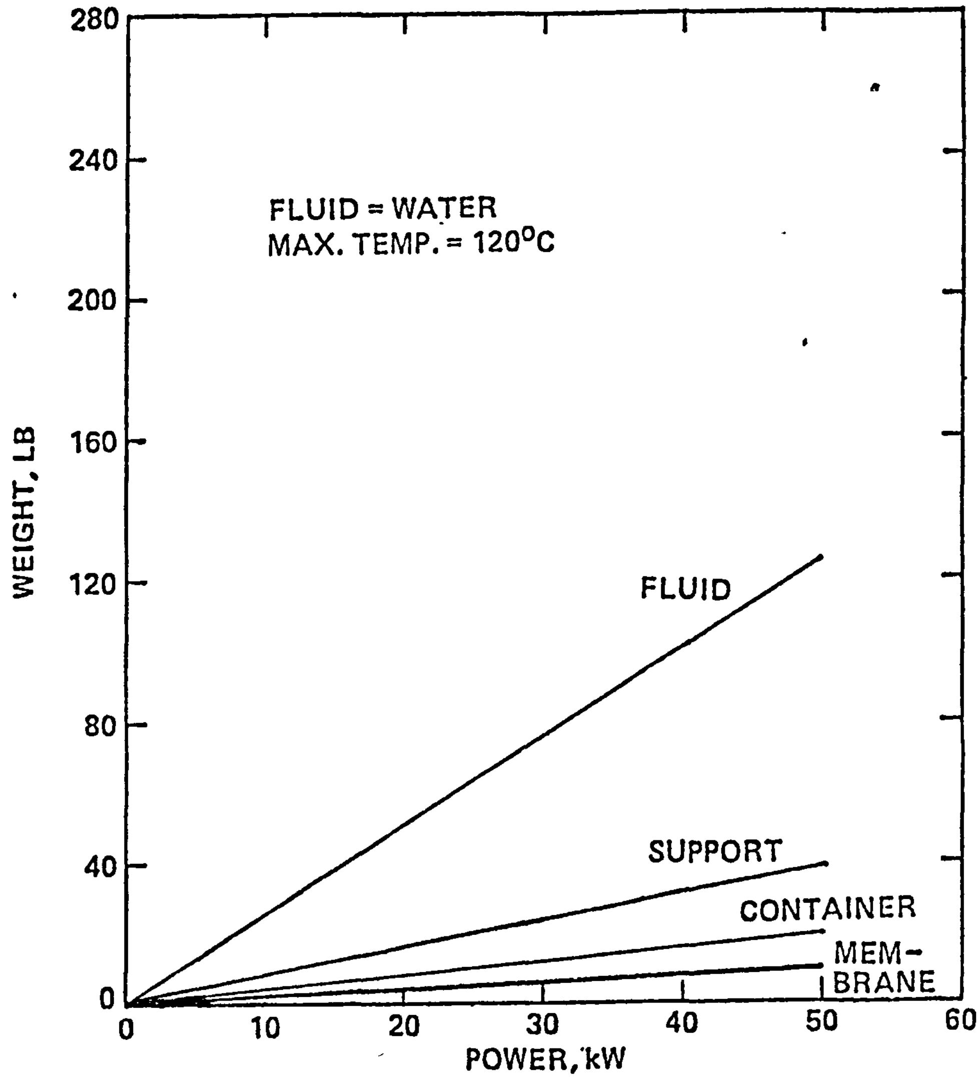


FIGURE 38

HOLLOW-FIBER MODULE COMPONENT WEIGHTS

FIGURE 39
OSMOTIC PUMP MODULE WEIGHTS

T (°C)	MODULE SIZE (kW)	MODULE WEIGHT (KG) AMMONIA*	WATER
4	1	3.17	-
	5	14.0	-
	12.5	36.1	-
	25	73.9	-
	50	154	-
20	1	3.28	-
	5	14.5	-
	12.5	37.3	-
	25	76.5	-
	50	159	-
40	1	3.50	2.09
	5	15.4	9.3
	12.5	39.7	22.4
	25	81.4	43.9
	50	170	86.9
120	1	-	2.23
	5	-	9.86
	12.5	-	23.8
	25	-	46.5
	50	-	92.0

*CONTAINER WEIGHT BASED ON VAPOR PRESSURE AT 40°C

CASE 1 - AMMONIA

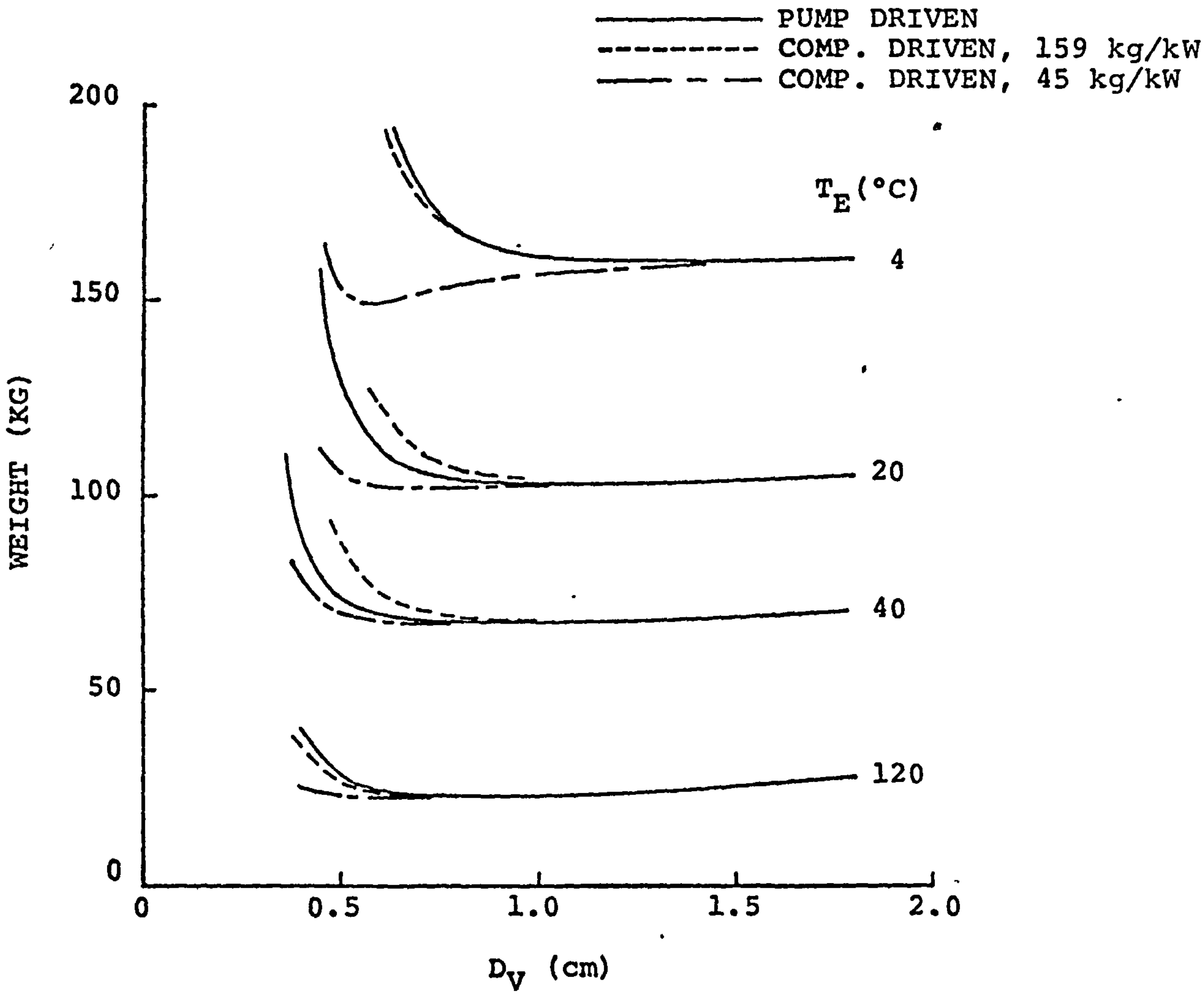


FIGURE 40
COMPARISON OF PUMP AND COMPRESSOR DRIVEN CONCEPTS

CASE 1 - FREON 11

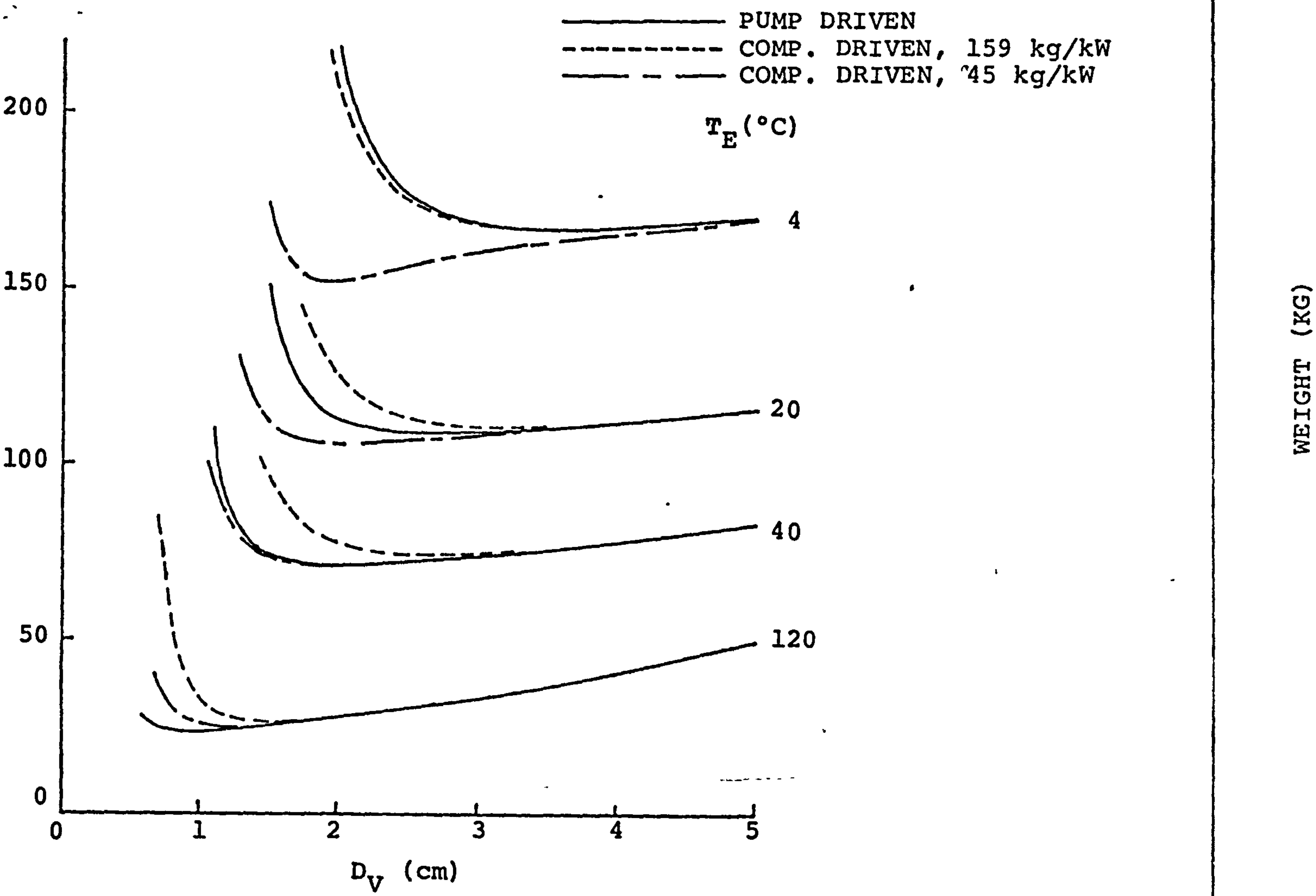


FIGURE 41

COMPARISON OF PUMP AND COMPRESSOR DRIVEN CONCEPTS

CASE 1 - WATER

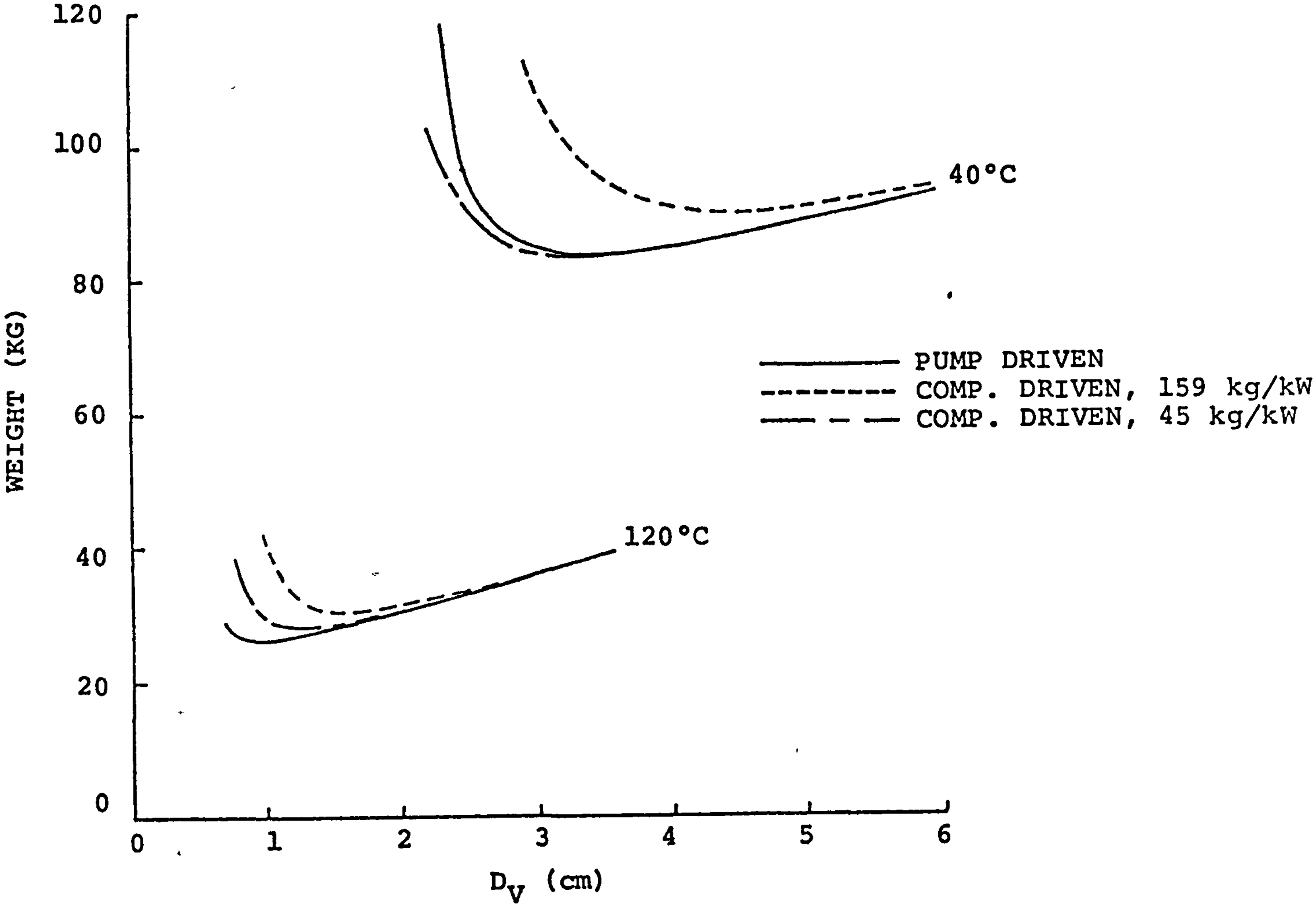


FIGURE 42
COMPARISON OF PUMP AND COMPRESSOR DRIVEN CONCEPTS

For the 4°C thermal bus, as diameter is decreased the reduced radiator weight more than offsets the increased power penalty weight up to a point. The result is a smaller optimum line diameter but much higher power requirements.

5.2 RELIABILITY, REDUNDANCY AND MAINTENANCE TRADE STUDY

A trade study was conducted to determine the maintenance and redundancy approach to achieve the reliability goals delineated in Requirements and Guidelines (Figure 2). The reliability goal was a 0.99 probability of no failure for a 10 year life.

The components of each of the four concepts were identified and failure rates assigned based on previous studies (References 7 and 8). A listing of the components for the four concepts with the corresponding failure rates are presented in Figures 43 through 46. The number of components for the four cases considered, i.e. heat loads of 5, 25, 150 and 350 kW, are also given with the summation of failure rates for each case. Failure rates are also given for selected redundant components. The redundant components were chosen based on the criteria of both having high failure rates and being logically feasible to place fully redundant components into the configuration. For Concept 1 the pump and accumulator were considered redundant, for Concept 2 the accumulator only, for Concept 3 the osmotic pumps and the accumulator and for Concept 4 the compressor, accumulator and bypass valve.

Figures 47 through 49 show the effect of maintenance on the system reliability. Selected components were considered for replacement at intervals of 5 years (1 replacement), 3-1/3 years (2 replacements), 2 years (4 replacements) and 1 year (9 replacements). The effect on reliability for single system/single component approaches are shown. In no case did replacement result in reliability near the 0.99 goal.

Figure 50 presents the 10 year life reliability for the four concepts and four cases. Reliabilities are given for: single system - single components, single system - redundant components, and redundant system - redundant components. These reliabilities were calculated using the failure rate data given in Figures 43 through 46. The single system - single component reliability is calculated by:

$$R = e^{-\Sigma\lambda t} \quad (1)$$

R = probability of no failure for period of time, t
t = design life
λ = failure rate, No. of failures/time

The reliability of redundant components is found from:

$$R = e^{-\lambda t} + \frac{\lambda}{\lambda_s} [e^{-\lambda t} - e^{-(\lambda + \lambda_s)t}] \quad (2)$$

R = probability of no failure
λ = component failure rate
λ_s = failure rate of failure detection and switch system
t = design life

FIGURE 43 FAILURE RATES FOR CONCEPT 1 - PUMP DRIVEN

			SINGLE COMPONENT		SYSTEM WITH REDUNDANT COMPONENT	
COMPONENT	CASE	QTY	FAILURE RATE λ x 10^{-6} HRS	Σλ	FAILURE RATE λ x 10^{-6} HRS	Σλ
PUMP	ALL	1	2.9	2.9	.327*	.327
ACCUMULATOR	ALL	1	0.22	0.22	.009	.009
EVAPORATOR HX	5 kW	5	0.20	1.0		
	25 kW	5	0.20	1.0		
	150 kW	8	0.20	1.6		
	350 kW	12	0.20	2.4		
LINES	ALL	100 M	0.05	0.05		
CONDENSER HX	ALL	1	0.20	0.20		
REGENERATIVE HX	ALL	1	0.20	0.20		
TOTALS	5 kW			4.57		1.786
	25 kW			4.57		1.786
	150 kW			5.17		2.386
	350 kW			5.97		3.186

*SWITCH TO REDUNDANT PUMP PROBABILITY OF SUCCESS - .99

FIGURE 44 FAILURE RATES FOR CONCEPT 2 - CAPILLARY PUMPED

			SINGLE COMPONENT		REDUNDANT COMPONENT	
COMPONENT	CASE	QTY	FAILURE RATE λ x 10^{-6} HRS	Σλ	FAILURE RATE λ x 10^{-6} HRS	Σλ
CAPILLARY PUMPS	5 kW	5	0.50	2.50		
	25 kW	5	0.50	2.50		
	150 kW	8	0.50	4.00		
	350 kW	12	0.50	6.00		
ACCUMULATOR	ALL	1	0.22	0.22	0.009	0.009
CONDENSER HX	ALL	1	0.20	0:20		
REGENERATIVE HX	ALL	1	0.20	0.20		
LINES	ALL	100 M	0.05	0.05		
TOTALS	5 kW			3.17		2.959
	25 kW			3.17		2.959
	150 kW			4.67		4.459
	350 kW			6.67		6.459

FIGURE 45 FAILURE RATES FOR CONCEPT 3 - OSMOTIC PUMP

COMPONENT	CASE	QTY	SINGLE COMPONENT: FAILURE RATE λ x 10^{-6} HRS	SINGLE COMPONENT: Σλ	REDUNDANT COMPONENT: FAILURE RATE λ x 10^{-6} HRS	REDUNDANT COMPONENT: Σλ
OSMOTIC PUMP (MEMBRANE ASSY)	5 kW	5	0.50	2.50	.021	.105
	25 kW	5	0.50	2.50	.021	.105
	150 kW	8	0.50	4.00	.021	.168
	350 kW	12	0.50	6.00	.021	.252
EVAPORATOR HX	5 kW	5	0.20	1.00		
	25 kW	5	0.20	1.00		
	150 kW	8	0.20	1.60		
	350 kW	12	0.20	2.40		
CONDENSER HX	ALL	1	0.20	0.20		
ACCUMULATOR HX	ALL	1	0.22	0.22	.009	.009
REGENERATIVE HX	ALL	1	0.20	0.20		
LINES	ALL	100 M	0.05	0.05		
TOTALS	5 kW			4.17		1.564
	25 kW			4.17		1.564
	150 kW			6.27		2.443
	350 kW			9.07		3.327

FIGURE 46 FAILURE RATES FOR CONCEPT 4 - COMPRESSOR DRIVEN

COMPONENT	CASE	QTY	SINGLE COMPONENT FAILURE RATE λ x 10^{-6} HRS	SINGLE COMPONENT Σλ	REDUNDANT COMPONENT FAILURE RATE λ x 10^{-6} HRS	REDUNDANT COMPONENT Σλ
COMPRESSOR	ALL	1	2.90	2.90	0.327	0.327
ACCUMULATOR	ALL	1	0.22	0.22	0.009	0.009
CONDENSER HX	ALL	1	0.20	0.20		
EVAPORATOR HX	5 kW	5	0.20	1.00		
	25 kW	5	0.20	1.00		
	150 kW	8	0.20	1.60		
	350 kW	12	0.20	2.40		
LINES	ALL	100 M	0.05	0.05		
BYPASS VALVE	ALL	1	0.43	0.43	0.0100	0.0100
REGENERATIVE HX	ALL	1	0.20	0.20		
TOTALS	5 kW			5.00		1.795
	25 kW			5.00		1.795
	150 kW			5.60		2.396
	350 kW			6.60		3.196

FIGURE 47
MAINTENANCE EFFECT ON RELIABILITY
CONCEPT 1 - PUMP DRIVEN

REPLACED PART	CASE	PROBABILITY OF NO FAILURE: SINGLE SYS. SINGLE COMP.	REPLACE ONCE	REPLACE TWICE	REPLACE 4 TIMES	REPLACE 9 TIMES
PUMP PACKAGE (PUMP & ACCUMULATOR)	5 kW	0.670	0.761	0.794	0.821	0.842
	25 kW	0.670	0.761	0.794	0.821	0.842
	150 kW	0.636	0.722	0.753	0.779	0.799
	350 kW	0.593	0.673	0.702	0.726	0.745
EVAPORATOR HX	5 kW	0.670	0.700	0.710	0.719	0.725
	25 kW	0.670	0.700	0.710	0.719	0.725
	150 kW	0.636	0.682	0.698	0.711	0.721
	350 kW	0.593	0.658	0.682	0.701	0.716
REPLACE PUMP PKG AND EVAP HX	5 kW	0.670	0.795	0.841	0.881	0.911
	25 kW	0.670	0.795	0.841	0.881	0.911
	150 kW	0.636	0.774	0.827	0.827	0.906
	350 kW	0.593	0.748	0.808	0.859	0.900

FIGURE 48

MAINTENANCE EFFECT ON RELIABILITY

CONCEPT 2 - CAPILLARY PUMPED

REPLACED PART	CASE	PROBABILITY OF NO FAILURE				
		SINGLE SYS. SINGLE COMP.	REPLACE ONCE	REPLACE TWICE	REPLACE 4 TIMES	REPLACE 9 TIMES
CAPILLARY PUMPS	5 kW	0.758	0.845	0.877	0.902	0.922
	25 kW	0.758	0.845	0.877	0.902	0.922
	150 kW	0.664	0.791	0.839	0.879	0.910
	350 kW	0.558	0.725	0.791	0.849	0.895

FIGURE 49
MAINTENANCE EFFECT ON RELIABILITY

CONCEPT 3 - OSMOTIC PUMP

		REPLACE 9 TIMES
OSMOTIC PUMP AND EVAPORATOR HX	5 kW	0.914
	25 kW	0.914
	150 kW	0.867
	350 kW	0.876

CONCEPT 4 - COMPRESSOR DRIVEN

REPLACED PART	CASE	PROBABILITY OF NO FAILURE REPLACE 9 TIMES
COMPRESSOR, BYPASS VALVE, AND EVAPORATOR HX	5 kW	0.908
	25 kW	0.908
	150 kW	0.903
	350 kW	0.897

FIGURE 50
RELIABILITY COMPARISON - 10 YEAR LIFE

CONCEPT	CASE	PROBABILITY OF NO FAILURE: SINGLE SYSTEM SINGLE COMPONENTS	SINGLE SYSTEM* REDUNDANT COMPONENTS	REDUNDANT SYSTEM* REDUNDANT COMPONENTS
1-PUMP DRIVEN	5 kW	0.670	0.855	0.984
	25 kW	0.670	0.855	0.984
	150 kW	0.636	0.780	0.971
	350 kW	0.593	0.756	0.949
2-CAPILLARY PUMPED	5 kW	0.758	0.772	0.946
	25 kW	0.758	0.772	0.946
	150 kW	0.664	0.677	0.892
	350 kW	0.558	0.568	0.809
3-OSMOTIC PUMP	5 kW	0.694	0.872	0.982
	25 kW	0.694	0.872	0.982
	150 kW	0.577	0.807	0.961
	350 kW	0.452	0.748	0.934
4-COMPRESSOR DRIVEN	5 kW	0.645	0.854	0.984
	25 kW	0.645	0.854	0.984
	150 kW	0.612	0.811	0.970
	350 kW	0.571	0.756	0.948

*PROBABILITY OF SUCCESSFUL SWITCH = 0.99

And the reliability of redundant systems from:

$$R = R_A R_D \tag{3}$$

R_A = probability of no failure for active system
R_D = probability of no failure for dormant system

R_D is determined for dormant components from Equation (2) and R_A for active components from:

$$R_A = 3e^{-\Sigma\lambda_A t} + e^{-\lambda_s t} - e^{-(\Sigma\lambda_A + \lambda_s)t} - e^{-2\Sigma\lambda_A t} - 1 \tag{4}$$

λ_A = failure rate of active components

The only components considered dormant in this analysis are those not subject to primary modes of failure when not active such as pumps, compressors, valves, etc. Other components such as lines, heat exchangers and heat pipe type components such as the osmotic and capillary pumps are considered active at all times since they are subject to primary failure modes whether or not they are in an active system.

The analysis results shown in Figure 50 indicate some difficulty may be encountered in achieving the 0.99 reliability goal, especially for the larger size systems. The redundant system with selected redundant components approach will result in reliabilities close to the goal for Concepts 1, 3 and 4 at the 5 and 25 kW size. Care in component selection and an aggressive quality and testing program could result in lowering the expected failure rates enough to achieve the 0.99 goal. Failure rate data used in this analysis represent the nominal values in the cases where a range of values were available. For the capillary pumped system, Concept 2, the low reliabilities are a result of the larger number of components and the fact that the characteristics of the capillary pumps prohibit effective use of redundant components since these devices are subject to the same failure modes whether fully active or not. A program of replacement combined with a redundant system approach is necessary to bring the reliability to the goal. Replacement of the capillary pumps every two years for the 5 and 25 kW cases and every year for the 150 and 350 kW cases would be required to achieve the 0.99 goal at the assumed failure rates. If a failure rate of 0.1×10^{-6} could be achieved for the capillary pumps (0.5×10^{-6} was assumed from Reference 8 values for high capacity heat pipes) the 0.99 goal could be met with the redundant component/redundant systems approach.

A program of replacement of the evaporator heat exchangers at 3.3 year intervals for the 150 kW case and 2 year intervals for the 350 kW case combined with a redundant component/redundant system approach will achieve the 0.99 goal for Concepts 1 and 4. A similar approach with two and one year replacements of the evaporator heat exchangers for the 150 kW and 350 kW cases respectively will achieve the goal. If the failure rates of the evaporator heat exchanger could be reduced to the equivalent for lines (0.05×10^{-6}), the 0.99 could be achieved in all Concept 1, 3 and 4 cases, except the 350 kW case for Concept 3, with a redundant component/redundant system approach.

A redundant thermal bus is inevitability required to achieve the design reliability goal unless dramatic reductions in the failure rates from those assumed can be achieved.

5.3. CONCEPT COMPARISON

The four concepts studied were evaluated to select the most promising concept for future development. The trade criteria were grouped under the following major categories: Performance, Reliability and Life, Development Considerations, and Operational Considerations.

The trade matrix presented in Figure 51 shows the evaluation of each concept. The performance evaluations were made at heat loads of 5, 25, and 350 kW based on a nominal operating temperature of 20°C (68°F) and assuming ammonia as the working fluid. The concept comparison for each category are described below.

Performance

The concepts were compared for four key performance items: weight, power, radiator area, and controllability. The mechanical pump concept is the lowest weight overall and the osmotic pump concept is the heaviest. All of the weights however are relatively close and weight probably would not be a leciding factor. The capillary and osmotic concepts have an advantage in requiring no power, although the power requirement for the mechanical pump :oncept is low (1 to 12 watts). As expected, the power requirements for the :ompressor concept are relatively high. Radiator area differences between the :oncepts are insignificant. The capillary pump concept is rated best in ontrollability since it is self regulating in flow control to the vaporators. The osmotic concept controllability is unknown since a proven eans for regulating flow was not identified. Controllability for the echanical pump and compressor concepts were judged equal and both were given good rating.

Reliability and Life

This category includes complexity and number of pieces, component life, ıd projected 10-year system reliability. The capillary pump concept is ınsidered superior in this category even though its 10-year reliability is ightly lower than the others. It requires the least number of pieces and mponent life should be high because of its passive nature. It's reliability lower due to the relatively high failure rate assumed for the capillary ıp/evaporators. The mechanical pump and compressor concepts follow closely this category with the mechanical pump given a slight edge over the ıpressor in complexity. The osmotic concept is given the lowest rating in ıplexity and component life based on the current state of development of the ıotic system.

Development Considerations

The concepts were compared in four areas: date of technology readiness, elopment risk, potential for success, and potential for growth. The ıanical pump concept is considered superior overall in this category ıuse of the developed nature of its components. The compressor concept is ed a close second with the only difference being longer lead time for

FIGURE 51 CONCEPT COMPARISON TRADE MATRIX

	CONCEPT				
RANKING CATEGORY	MECH PUMP	CAPILLARY PUMP	OSMOTIC PUMP	COMPRESSOR 159 kg/kW	COMPRESSOR 45 kg/kW
PERFORMANCE[1]					
. WEIGHT (KG)					
– 5 kW	111	110	125	112	114
– 25 kW	555	566	624	–[2]	–[2]
– 350 kW	7578	7620	8700	7607	7485
. POWER (W)					
– 5 kW	1	0.0	0.0	7.68	90.86
– 25 kW	6	0.0	0.0	–[2]	–[2]
– 350 kW	12	0.0	0.0	571	5499
. RADIATOR AREA (M^2)					
– 5 kW	17	17	17	17	16
– 25 kW	86	85	86	–[2]	–[2]
– 350 kW	1192	1189	1192	1183	1153
. CONTROLABILITY	GOOD	EXCELLENT (SEMI-PASSIVE)	UNKNOWN	GOOD	GOOD
RELIABILITY & LIFE					
. COMPLEXITY & NO. OF PIECES	GOOD	EXCELLENT	POOR[3]	FAIR	FAIR
. COMPONENT LIFE	FAIR	GOOD	POOR[3]	FAIR	FAIR
. PROJECTED 10 YEAR[4] SYSTEM RELIABILITY	0.98	0.95	0.98	0.98	0.98

[1] FLUID : AMMONIA
BUS TEMPERATURE : 20°C

[2] VALUES NOT CALCULATED

[3] WITH CURRENT STATE OF DEVELOPMENT

[4] WITH FULLY REDUNDANT THERMAL BUS

FIGURE 51 (CONT'D)

RANKING CATEGORY	CONCEPT: MECH PUMP	CAPILLARY PUMP	OSMOTIC PUMP	COMPRESSOR 159 kg/kW	COMPRESSOR 45 kg/kW
<u>DEVELOPMENT CONSIDERATIONS</u>					
. DATE OF TECHNOLOGY READINESS	1984	1987	1993	1985	1985
. DEVELOPMENT RISK	LOW	MEDIUM[3]	HIGH[3]	LOW	LOW
. POTENTIAL FOR SUCCESS	GOOD	FAIR[3]	POOR[3]	GOOD	GOOD
. POTENTIAL FOR GROWTH	GOOD	FAIR	GOOD	GOOD	GOOD
<u>OPERATIONAL CONSIDERATIONS</u>					
. FLEXIBILITY FOR LOCALIZED HEAT REMOVAL	GOOD	FAIR	GOOD	GOOD	GOOD
. AUTONOMOUS OPERATION	POOR	EXCELLENT	GOOD	POOR	POOR
. EASE OF STARTUP	GOOD	FAIR	FAIR	GOOD	GOOD
. FLEXIBILITY FOR RECONFIGURATION	GOOD	GOOD	GOOD	GOOD	GOOD

[3] WITH CURRENT STATE OF DEVELOPMENT

compressor development. Because of development required for the capillary pump, this concept was rated as a medium risk. The osmotic concept is considered the highest risk concept because of the development required in the areas of membranes, solvents, solutes, zero-g operation, and control technique.

Operational Considerations

This category includes flexibility, autonomous operation, and ease of startup. Flexibility for localized heat removal includes considerations such as the ability to locally add heat loads (heat exchangers) or increase the heat load at a given heat exchanger. The capillary pump concept was only given a fair rating in flexibility since the addition of heat loads could exceed the relatively low pumping limits of the capillary pumps unless the pumps are oversized to begin with. The other concepts were rated good because of their high pumping potential. The capillary pump and osmotic concepts are superior in autonomous operation. Both require zero power but the osmotic concept could require batteries if the electrostatic control technique is used; therefore it was rated slightly lower than the capillary pump concept. In ease of startup, the mechanical pump and compressor concepts were rated superior and in flexibility for reconfiguration all concepts were rated equal.

6.0 CONCLUSIONS

A number of conclusions were reached as a result of this study. The following conclusions were reached on the concepts considered:

(1) The mechanical pump concept has the lowest development risk and shortest lead time. This concept offers good performance, simplicity, and reliability.

(2) The capillary pump concept is promising because of its passive nature and its long life potential. The development of a capillary pump results in a longer lead time and more development risk than the mechanical pump.

(3) The osmotic pump, while having many attractive features, has many unanswered questions in the areas of heat load control techniques and availability of membrane materials with suitable performance and life characteristics. These unanswered questions cause excessive risks with selection of this concept.

(4) The compressor driven concept offers no significant advantage over the other concepts.

Other conclusions reached from this study are:

(1) Ammonia is the superior working fluid in terms of weight and performance over the range of conditions considered in this study. No suitable fluid was identified for use in a manned cabin at temperatures below 40°C (104°F).

(2) The capillary pump concept's vapor and liquid flow lines require approximately twice the flow area as the mechanical pump for most cases.

(3) Power penalty has a small effect on the total system weight for the compressor driven concept. However the optimum system configuration is significantly impacted. Compared to the nominal power penalty (159 kg/kW), the low penalty (45 kg/kW) resulted in reduced line diameter and radiator area, but significantly increased power requirements.

(4) Fully redundant systems are required for the thermal bus to obtain reliabilities approaching the goal of 0.99 probability of success for 10 years. Scheduled maintenance without the redundant bus will not achieve the desired goal.

7.0 RECOMMENDATIONS

Based on this study, the following recommendations are made for development of a thermal bus to meet the needs of thermal management on large, long life space platforms projected for the 1990's.

(1) The mechanical pump driven concept should be developed initially. A prototype thermal bus should be built and tested to develop the heat rejection and temperature control techniques, the pump, and the two-phase heat exchangers.

(2) The capillary pump should be developed separately to establish its characteristics and limitations.

(3) Laboratory study should be continued on the osmotic pump to address the unresolved questions of material selection and control technique.

A set of abbreviated specifications for the thermal bus development are provided below.

THERMAL BUS SPECIFICATION

GENERAL

- The thermal bus shall provide the transport of heat between the collection interfaces and the rejection interfaces under near isothermal conditions.
- The bus must operate in the space environment of low earth orbit.

PERFORMANCE

• Total Heat Transport	:	25 to 100 kW
• Bus Temperatures	:	Controllable from 4°C to 40°C (39°F to 104°F)
• Isothermal Character	:	5°C (9°F) Band at Control Temperature
• Individual Heat Loads	:	1 to 50 kW

PHYSICAL CHARACTERISTICS

- Centralized System Length - 15 m to 50 m (49 ft to 164 ft)
- Capable of interfacing multiple payload heat loads
- Must interface with the space constructable radiator
- Minimum weight and volume

DESIRED OPERATIONAL CHARACTERISTICS AND TIMING

- On-orbit deployment, start-up, and shut-down capability
- Modular growth capability to 200 kW

- On-orbit reconfiguration capability
- Capability for simple make and break of interfaces with equipment
- Minimize monitoring and control required
- Minimize on-orbit maintenance
- 1990 technology readiness
- Shall be operational for quiescent periods or full up periods of up to 6 months without Orbiter re-supply
- A method for detecting, locating, isolating and repairing fluid leaks within the system on-orbit

ENVIRONMENTS

- Unpressurized area
- Assume thermal control and micrometeoroid protection is provided by surrounding structure
- Launch vibroacoustic and acceleration per Spacelab User's Handbook (pressurized module and pallet)

INTERFACES

- Heat load interface to the Thermal Bus shall be through two types of devices
  - (1) liquid-to-thermal bus evaporator heat exchangers
  - (2) thermal bus evaporating coldplates

- Heat rejection interface to the Thermal Bus shall be through space constructable radiator contact heat exchangers

CONTROL

- The Thermal Bus must provide uniform temperature (within the performance requirement) with varying individual and total heat loads. Individual heat loads can vary from 0% to 100% of rated load; total heat load can vary from 5% to 100% of total rated load

- The Thermal Bus must provide the heat rejection system control

'RADE PENALTIES

- Power : 159 kg/kW (350 lb/kW)
- Launch Costs : \$1540/kg (\$700/lb)
- Heat Rejection : 5.9 kg/m^2 (1.2 lb/ft^2), from constructable radiator concepts in NAS3-22270 at -40°C environmental sink

\FETY

- No toxic or flammable fluids in pressurized compartments
- Fluid toxicities compatible with practical ground handling for bus in unpressurized areas
- No contact temperatures above 45°C (113°F)
- General guidelines from Rockwell Phase B Modular Space Station

THERMAL BUS SPECIFICATION (CONTINUED)

RELIABILITY

- 10 year life design goal at 0.99 probability
- Redundancy and minimal maintenance to achieve life goal
- Indefinite life with further maintenance
- Minimize moving parts
- Minimize complexity

OTHER

- Minimize life cycle cost
- Minimize vibration
- Minimize EMI generation
- Minimize contamination threat to payloads
- Minimize power
- Minimize heat rejection surface area
- Minimize subcontract launch site support

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APPENDIX

Included in this Appendix are detailed breakdowns of component weights, pump power, radiator area, and line sizes for each design point evaluated. They are arranged in order of fluid, case number, temperature, and concept number.

PHYSICAL CHARACTERISTICS SUMMARY

CASE 1 CONCEPT 1

AMMONIA 4°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 m	2.73	19 M LIQUID LINE @ 0.50 CM ID 19 M VAPOR LINE @ 1.27 CM ID
HEAT EXCHANGERS (WET)	6	6.0	
ACCUMULATOR (WET)	1	0.83	
RADIATOR PENALTY	26.6 m^2	156.8	
POWER PENALTY	0.49 W	0.08	
PUMP	2	1.81	
TOTAL		168.3	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 1 CONCEPT 2

AMMONIA 4°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 m	3.75	19 M LIQUID LINE @ 0.60 CM ID 19 M VAPOR LINE @ 1.52 CM ID
HEAT EXCHANGERS (WET)	6	6.0	
ACCUMULATOR (WET)	1	0.16	
RADIATOR PENALTY	26.5 m^2	156.3	
POWER PENALTY	0	0	
PUMP	0	0	
TOTAL		166.2	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 1 CONCEPT 3

AMMONIA 4°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	2.73	19 M LIQUID LINE @ 0.50 CM ID 19 M VAPOR LINE @ 1.27 CM ID
HEAT EXCHANGERS (WET)	6	6.0	
ACCUMULATOR (WET)	1	0.15	
RADIATOR PENALTY	26.6 M^2	156.8	
POWER PENALTY	0 W	0	
PUMP	5	15.85	
TOTAL		181.6	

PHYSICAL CHARACTERISTICS SUMMARY

CASE I CONCEPT 4A

AMMONIA 4°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	2.84	19 M LIQUID LINE @ 0.52 CM ID 19 M VAPOR LINE @ 1.30 CM ID
HEAT EXCHANGERS (WET)	6	6.0	
ACCUMULATOR (WET)	1	0.15	
RADIATOR PENALTY	26.3 M^2	154.9	
POWER PENALTY	13.1 W	2.09	159 KG/KW
PUMP	2	2.9	
TOTAL		168.9	

PHYSICAL CHARACTERISTICS SUMMARY

CASE I CONCEPT 4B

AMMONIA 4°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	1.19	19 M LIQUID LINE @ 0.24 CM ID 19 M VAPOR LINE @ 0.60 CM ID
HEAT EXCHANGERS (WET)	6	6.0	
ACCUMULATOR (WET)	1	0.12	
RADIATOR PENALTY	21.6 M^2	127.5	
POWER PENALTY	490.4 W	22.1	45 KG/KW
PUMP	2	11.8	
TOTAL		168.7	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 1 CONCEPT 1
AMMONIA 20°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	2.05	19 M LIQUID LINE @ 0.45 CM ID 19 M VAPOR LINE @ 1.02 CM ID
HEAT EXCHANGERS (WET)	6	6.0	
ACCUMULATOR (WET)	1	0.51	
RADIATOR PENALTY	17.1 M^2	100.7	
POWER PENALTY	0.99 W	0.16	
PUMP	2	1.81	
TOTAL		111.2	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 1 CONCEPT 2
AMMONIA 20°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	10.64	19 M LIQUID LINE @ 1.71 CM ID 19 M VAPOR LINE @ 2.10 CM ID
HEAT EXCHANGERS (WET)	6	6.0	
ACCUMULATOR (WET)	1	0.27	
RADIATOR PENALTY	3.38 M^2	19.96	
POWER PENALTY	0 W	0	
PUMP	0	0	
TOTAL		36.9	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 1 CONCEPT 3
AMMONIA 20°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	2.05	19 M LIQUID LINE @ 0.45 CM ID 19 M VAPOR LINE @ 1.02 CM ID
HEAT EXCHANGERS (WET)	6	6.0	
ACCUMULATOR (WET)	1	0.13	
RADIATOR PENALTY	17.1 M^2	100.7	
POWER PENALTY	0 W	0	
PUMP	5	16.4	
TOTAL		125.3	

PHYSICAL CHARACTERISTICS SUMMARY

CASE I CONCEPT 4A

AMMONIA 20°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	2.59	19 M LIQUID LINE @ 0.53 CM ID 19 M VAPOR LINE @ 1.20 CM ID
HEAT EXCHANGERS (WET)	6	6.0	
ACCUMULATOR (WET)	1	0.14	
RADIATOR PENALTY	16.9 M^2	99.8	
POWER PENALTY	7.7 W	1.22	159 KG/KW
PUMP	2	2.4	
TOTAL		112.2	

PHYSICAL CHARACTERISTICS SUMMARY

CASE I CONCEPT 4B

AMMONIA 20°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	1.42	19 M LIQUID LINE @ 0.31 CM ID 19 M VAPOR LINE @ 0.70 CM ID
HEAT EXCHANGERS (WET)	6	6.0	
ACCUMULATOR (WET)	1	0.12	
RADIATOR PENALTY	16.4 M^2	96.69	
POWER PENALTY	90.9 W	4.09	45 KG/KW
PUMP	2	6.0	
TOTAL		114.3	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 1 CONCEPT 1

AMMONIA 40°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	1.90	19 M LIQUID LINE @ 0.45 CM ID 19 M VAPOR LINE @ 0.89 CM ID
HEAT EXCHANGERS (WET)	6	6.0	
ACCUMULATOR (WET)	1	0.37	
RADIATOR PENALTY	11.1 M^2	65.29	
POWER PENALTY	1.50 W	0.24	
PUMP	2	1.81	
TOTAL		75.6	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 1 CONCEPT 2

AMMONIA 40°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	3.48	19 M LIQUID LINE @ 0.69 CM ID 19 M VAPOR LINE @ 1.38 CM ID
HEAT EXCHANGERS (WET)	6	6.0	
ACCUMULATOR (WET)	1	0.16	
RADIATOR PENALTY	11.03 M^2	65.1	
POWER PENALTY	0 W	0	
PUMP	0	0	
TOTAL		74.7	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 1 CONCEPT 3

AMMONIA 40°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	1.90	19 M LIQUID LINE @ 0.45 CM ID 19 M VAPOR LINE @ 0.89 CM ID
HEAT EXCHANGERS (WET)	6	6.0	
ACCUMULATOR (WET)	1	0.13	
RADIATOR PENALTY	11.07 M^2	65.3	
POWER PENALTY	0 W	0	
PUMP	5	17.5	
TOTAL		90.8	

PHYSICAL CHARACTERISTICS SUMMARY

CASE I CONCEPT 4A

AMMONIA 40°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	2.37	19 M LIQUID LINE @ 0.55 CM ID 19 M VAPOR LINE @ 1.10 CM ID
HEAT EXCHANGERS (WET)	6	6.0	
ACCUMULATOR (WET)	1	0.14	
RADIATOR PENALTY	11.01 M^2	65.0	
POWER PENALTY	4.36 W	0.69	159 KG/KW
PUMP	2	2.0	
TOTAL		76.2	

PHYSICAL CHARACTERISTICS SUMMARY

CASE I CONCEPT 4B

AMMONIA 40°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	1.70	19 M LIQUID LINE @ 0.40 CM ID 19 M VAPOR LINE @ 0.80 CM ID
HEAT EXCHANGERS (WET)	6	6.0	
ACCUMULATOR (WET)	1	0.12	
RADIATOR PENALTY	10.95 M^2	64.6	
POWER PENALTY	19.2 W	0.86	45 KG/KW
PUMP	2	3.4	
TOTAL		76.7	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 1 CONCEPT 1
AMMONIA 120°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	2.48	19 M LIQUID LINE @ 0.73 CM ID 19 M VAPOR LINE @ 0.90 CM ID
HEAT EXCHANGERS (WET)	6	6.0	
ACCUMULATOR (WET)	1	0.24	
RADIATOR PENALTY	3.38 M^2	19.96	
POWER PENALTY	2.78 W	0.44	
PUMP	2	2.72	
TOTAL		31.8	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 1 CONCEPT 2
AMMONIA 120°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	10.64	19 M LIQUID LINE @ 1.71 CM ID 19 M VAPOR LINE @ 2.11 CM ID
HEAT EXCHANGERS (WET)	6	6.0	
ACCUMULATOR (WET)	1	0.27	
RADIATOR PENALTY	3.38 M^2	19.96	
POWER PENALTY	0 W	0	
PUMP	0	0	
TOTAL		36.9	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 1 CONCEPT 4A
AMMONIA 120°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	2.18	19 M LIQUID LINE @ 0.65 CM ID 19 M VAPOR LINE @ 0.80 CM ID
HEAT EXCHANGERS (WET)	6	6.0	
ACCUMULATOR (WET)	1	0.10	
RADIATOR PENALTY	3.38 M^2	19.94	
POWER PENALTY	3.73 W	0.59	159 KG/KW
PUMP	2	1.80	
TOTAL		30.6	

PHYSICAL CHARACTERISTICS SUMMARY

CASE I CONCEPT 4B

AMMONIA 120°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	1.61	19 M LIQUID LINE @ 0.49 CM ID 19 M VAPOR LINE @ 0.60 CM ID
HEAT EXCHANGERS (WET)	6	6.0	
ACCUMULATOR (WET)	1	0.08	
RADIATOR PENALTY	3.37 M^2	19.91	
POWER PENALTY	14.5 W	0.65	45 KG/KW
PUMP	2	3.0	
TOTAL		31.3	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 2 CONCEPT 1

AMMONIA 4°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	24.3	50 M LIQUID LINE @ 0.99 CM ID 50 M VAPOR LINE @ 2.50 CM ID
HEAT EXCHANGERS (WET)	6	24.4	
ACCUMULATOR (WET)	1	8.5	
RADIATOR PENALTY	133.4 M^2	787.1	
POWER PENALTY	2.99 W	0.48	
PUMP	2	2.7	
TOTAL		847.6	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 2 CONCEPT 2

AMMONIA 4°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	41.86	50 M LIQUID LINE @ 1.31 CM ID 50 M VAPOR LINE @ 3.30 CM ID
HEAT EXCHANGERS (WET)	6	24.4	
ACCUMULATOR (WET)	1	1.01	
RADIATOR PENALTY	132.4 M^2	781.2	
POWER PENALTY	0 W	0	
PUMP	0	0	
TOTAL		848.5	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 2 CONCEPT 3

AMMONIA 4°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	24.3	50 M LIQUID LINE @ 0.99 CM ID 50 M VAPOR LINE @ 2.50 CM ID
HEAT EXCHANGERS (WET)	6	24.4	
ACCUMULATOR (WET)	1	0.78	
RADIATOR PENALTY	133.4 M^2	787.1	
POWER PENALTY	0 W	0	
PUMP	5	74.5	
TOTAL		911.1	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 2 CONCEPT 1

AMMONIA 20°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	16.8	50 M LIQUID LINE @ 0.89 CM ID 50 M VAPOR LINE @ 2.00 CM ID
HEAT EXCHANGERS (WET)	6	24.5	
ACCUMULATOR (WET)	1	5.2	
RADIATOR PENALTY	85.7 M^2	505.4	
POWER PENALTY	5.72 W	0.9	
PUMP	2	2.8	
TOTAL		555.5	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 2 CONCEPT 2

AMMONIA 20°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	39.4	50 M LIQUID LINE @ 1.38 CM ID 50 M VAPOR LINE @ 3.12 CM ID
HEAT EXCHANGERS (WET)	6	24.5	
ACCUMULATOR (WET)	1	1.0	
RADIATOR PENALTY	84.9 M^2	501.1	
POWER PENALTY	0 W	0	
PUMP	0	0	
TOTAL		566.0	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 2 CONCEPT 3

AMMONIA 20°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	16.8	50 M LIQUID LINE @ 0.89 CM ID 50 M VAPOR LINE @ 2.00 CM ID
HEAT EXCHANGERS (WET)	6	24.5	
ACCUMULATOR (WET)	1	0.70	
RADIATOR PENALTY	85.7 M^2	505.4	
POWER PENALTY	0 W	0	
PUMP	5	77.0	
TOTAL		624.3	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 2 CONCEPT 1
AMMONIA 40°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	10.6	50 M LIQUID LINE @ 0.75 CM ID 50 M VAPOR LINE @ 1.50 CM ID
HEAT EXCHANGERS (WET)	6	24.5	
ACCUMULATOR (WET)	1	2.8	
RADIATOR PENALTY	55.8 M^2	329.2	
POWER PENALTY	15.3 W	2.4	
PUMP	2	2.9	
TOTAL		372.4	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 2 CONCEPT 2
AMMONIA 40°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	38.5	50 M LIQUID LINE @ 1.50 CM ID 50 M VAPOR LINE @ 2.98 CM ID
HEAT EXCHANGERS (WET)	6	24.5	
ACCUMULATOR (WET)	1	1.08	
RADIATOR PENALTY	55.1 M^2	325.2	
POWER PENALTY	0 W	0	
PUMP	0	0	
TOTAL		389.3	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 2 CONCEPT 3
AMMONIA 40°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	10.6	50 M LIQUID LINE @ 0.75 CM ID 50 M VAPOR LINE @ 1.50 CM ID
HEAT EXCHANGERS (WET)	6	24.5	
ACCUMULATOR (WET)	1	0.60	
RADIATOR PENALTY	55.8 M^2	329.2	
POWER PENALTY	0 W	0	
PUMP	5	82.0	
TOTAL		446.8	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 3 CONCEPT 1

AMMONIA 4°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	96.1	50 M LIQUID LINE @ 1.98 CM ID 50 M VAPOR LINE @ 5.00 CM ID
HEAT EXCHANGERS (WET)	9	145.9	
ACCUMULATOR (WET)	1	34.0	
RADIATOR PENALTY	799.5 M^2	4716.8	
POWER PENALTY	7.39 W	1.2	
PUMP	2	4.6	
TOTAL		4999	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 3 CONCEPT 2

AMMONIA 4°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	162.4	50 M LIQUID LINE @ 2.58 CM ID 50 M VAPOR LINE @ 6.50 CM ID
HEAT EXCHANGERS (WET)	9	145.9	
ACCUMULATOR (WET)	1	4.9	
RADIATOR PENALTY	794.5 M^2	4687	
POWER PENALTY	0 W	0	
PUMP	0	0	
TOTAL		5001	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 3 CONCEPT 3

AMMONIA 4°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	96.1	50 M LIQUID LINE @ 1.98 CM ID 50 M VAPOR LINE @ 5.00 CM ID
HEAT EXCHANGERS (WET)	9	145.9	
ACCUMULATOR (WET)	1	4.1	
RADIATOR PENALTY	799.5 M^2	4717	
POWER PENALTY	0 W	0	
PUMP	8	477.5	
TOTAL		5440	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 3 CONCEPT 1
AMMONIA 20°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	64.7	50 M LIQUID LINE @ 1.77 CM ID 50 M VAPOR LINE @ 4.00 CM ID
HEAT EXCHANGERS (WET)	9	146.5	
ACCUMULATOR (WET)	1	20.6	
RADIATOR PENALTY	513.4 M^2	3029.3	
POWER PENALTY	14.2 W	2.3	
PUMP	2	4.8	
TOTAL		3268	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 3 CONCEPT 2
AMMONIA 20°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	153.0	50 M LIQUID LINE @ 2.73 CM ID 50 M VAPOR LINE @ 6.15 CM ID
HEAT EXCHANGERS (WET)	9	146.5	
ACCUMULATOR (WET)	1	5.0	
RADIATOR PENALTY	509.5 M^2	3006	
POWER PENALTY	0 W	0	
PUMP	0	0	
TOTAL		3311	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 3 CONCEPT 3
AMMONIA 20°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	96.1	50 M LIQUID LINE @ 1.77 CM ID 50 M VAPOR LINE @ 4.00 CM ID
HEAT EXCHANGERS (WET)	9	146.5	
ACCUMULATOR (WET)	1	3.7	
RADIATOR PENALTY	513.4 M^2	3029	
POWER PENALTY	0 W	0	
PUMP	8	494.6	
TOTAL		3770	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 3 CONCEPT 1

AMMONIA 40°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	44.4	50 M LIQUID LINE @ 1.61 CM ID 50 M VAPOR LINE @ 3.20 CM ID
HEAT EXCHANGERS (WET)	9	147.0	
ACCUMULATOR (WET)	1	12.6	
RADIATOR PENALTY	333.5 M^2	1967.7	
POWER PENALTY	29.1 W	4.6	
PUMP	2	5.0	
TOTAL		2181	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 3 CONCEPT 2

AMMONIA 40°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	149.7	50 M LIQUID LINE @ 2.96 CM ID 50 M VAPOR LINE @ 5.88 CM ID
HEAT EXCHANGERS (WET)	9	147.0	
ACCUMULATOR (WET)	1	5.1	
RADIATOR PENALTY	330.7 M^2	1951	
POWER PENALTY	0 W	0	
PUMP	0	0	
TOTAL		2253	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 3 CONCEPT 3

AMMONIA 40°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	44.4	50 M LIQUID LINE @ 1.61 CM ID 50 M VAPOR LINE @ 3.20 CM ID
HEAT EXCHANGERS (WET)	9	147.0	
ACCUMULATOR (WET)	1	3.4	
RADIATOR PENALTY	333.5 M^2	1968	
POWER PENALTY	0 W		
PUMP	8	527.9	
TOTAL		2690	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 4 CONCEPT 1
AMMONIA 4°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100M	188.3	50 M LIQUID LINE a 2.78 CM ID 50 M VAPOR LINE a 7.00 CM ID
HEAT EXCHANGERS (WET)	13	341.2	
ACCUMULATOR (WET)	1	66.6	
RADIATOR PENALTY	1861 M^2	10978	
POWER PENALTY	8.6 W	1.4	
PUMP	2	6.1	
TOTAL		11582	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 4 CONCEPT 2
AMMONIA 4°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	268.0	50 M LIQUID LINE a 3.31 CM ID 50 M VAPOR LINE a 8.35 CM ID
HEAT EXCHANGERS (WET)	13	341.2	
ACCUMULATOR (WET)	1	10.1	
RADIATOR PENALTY	1854 M^2	10937	
POWER PENALTY	0 W	0	
PUMP	0	0	
TOTAL		11557	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 4 CONCEPT 3
AMMONIA 4°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	188.3	50 M LIQUID LINE a 2.78 CM ID 50 M VAPOR LINE a 7.00 CM ID
HEAT EXCHANGERS (WET)	13	341.2	
ACCUMULATOR (WET)	1	9.0	
RADIATOR PENALTY	1861 M^2	10978	
POWER PENALTY	0 W	0	
PUMP	12	1126	
TOTAL		12643	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 4 CONCEPT 4A
AMMONIA 4°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	188.3	50 M LIQUID LINE @ 2.78 CM ID 50 M VAPOR LINE @ 7.00 CM ID
HEAT EXCHANGERS (WET)	13	341.2	
ACCUMULATOR (WET)	1	9.0	
RADIATOR PENALTY	1835 M^2	10829	
POWER PENALTY	1002 W	159.3	159 KG/KW
PUMP	2	16.3	
TOTAL		11543	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 4 CONCEPT 4B
AMMONIA 4°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	34.6	50 M LIQUID LINE @ 1.19 CM ID 50 M VAPOR LINE @ 3.00 CM ID
HEAT EXCHANGERS (WET)	13	341.2	
ACCUMULATOR (WET)	1	7.1	
RADIATOR PENALTY	1385 M^2	8170	
POWER PENALTY	53427 W	2404	45 KG/KW
PUMP	2	62.6	
TOTAL		11020	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 4 CONCEPT 1
AMMONIA 20°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	145.6	50 M LIQUID LINE @ 2.66 CM ID 50 M VAPOR LINE @ 6.00 CM ID
HEAT EXCHANGERS (WET)	13	342.6	
ACCUMULATOR (WET)	1	46.8	
RADIATOR PENALTY	1192 M^2	7035	
POWER PENALTY	12.4 W	2.0	
PUMP	2	6.3	
TOTAL		7578	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 4 CONCEPT 2
AMMONIA 20°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	252.5	50 M LIQUID LINE @ 3.50 CM ID 50 M VAPOR LINE @ 7.90 CM ID
HEAT EXCHANGERS (WET)	13	342.6	
ACCUMULATOR (WET)	1	10.1	
RADIATOR PENALTY	1189 M^2	7015	
POWER PENALTY	0 W	0	
PUMP	0	0	
TOTAL		7620	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 4 CONCEPT 3
AMMONIA 20°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	145.6	50 M LIQUID LINE @ 2.66 CM ID 50 M VAPOR LINE @ 6.00 CM ID
HEAT EXCHANGERS (WET)	13	342.6	
ACCUMULATOR (WET)	1	8.6	
RADIATOR PENALTY	1192 M^2	7034.8	
POWER PENALTY	0 W	0	
PUMP	12	1168.8	
TOTAL		8700	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 4 CONCEPT 4B
AMMONIA 20°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	64.7	50 M LIQUID LINE @ 1.77 CM ID 50 M VAPOR LINE @ 4.00 CM ID
HEAT EXCHANGERS (WET)	13	342.6	
ACCUMULATOR (WET)	1	7.4	
RADIATOR PENALTY	1153 M^2	6800	
POWER PENALTY	5500 W	247.5	45 KG/KW
PUMP	2	22.7	
TOTAL		7485	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 4 CONCEPT 4A
AMMONIA 20°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	170.9	50 M LIQUID LINE @ 2.88 CM ID 50 M VAPOR LINE @ 6.50 CM ID
HEAT EXCHANGERS (WET)	13	342.6	
ACCUMULATOR (WET)	1	8.9	
RADIATOR PENALTY	1183 M^2	6981.5	
POWER PENALTY	571 W	90.7	159 KG/KW
PUMP	2	12.7	
TOTAL		7607	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 4 CONCEPT 1

AMMONIA 40°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	69.4	50 M LIQUID LINE @ 2.01 CM ID 50 M VAPOR LINE @ 4.00 CM ID
HEAT EXCHANGERS (WET)	13	343.6	
ACCUMULATOR (WET)	1	19.6	
RADIATOR PENALTY	779 M^2	4598	
POWER PENALTY	59.6 W	9.5	
PUMP	2	6.5	
TOTAL		5047	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 4 CONCEPT 2

AMMONIA 40°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	247.1	50 M LIQUID LINE @ 3.79 CM ID 50 M VAPOR LINE @ 7.55 CM ID
HEAT EXCHANGERS (WET)	13	343.6	
ACCUMULATOR (WET)	1	10.2	
RADIATOR PENALTY	771.7 M^2	4553	
POWER PENALTY	0 W	0	
PUMP	0	0	
TOTAL		5154	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 4 CONCEPT 3

AMMONIA 40°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	69.4	50 M LIQUID LINE @ 2.01 CM ID 50 M VAPOR LINE @ 4.00 CM ID
HEAT EXCHANGERS (WET)	13	343.6	
ACCUMULATOR (WET)	1	7.3	
RADIATOR PENALTY	779.4 M^2	4598	
POWER PENALTY	0 W	0	
PUMP	12	1247	
TOTAL		6266	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 4 CONCEPT 4A

AMMONIA 40°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	131.2	50 M LIQUID LINE @ 2.76 CM ID 50 M VAPOR LINE @ 5.50 CM ID
HEAT EXCHANGERS (WET)	13	343.6	
ACCUMULATOR (WET)	1	8.3	
RADIATOR PENALTY	770 M^2	4543	
POWER PENALTY	470 W	74.7	
PUMP	2	11.8	
TOTAL		5113	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 4 CONCEPT 4B

AMMONIA 40°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	87.8	50 M LIQUID LINE @ 2.26 CM ID 50 M VAPOR LINE @ 4.50 CM ID
HEAT EXCHANGERS (WET)	13	343.6	
ACCUMULATOR (WET)	1	7.6	
RADIATOR PENALTY	767.1 M^2	4526	
POWER PENALTY	1200 W	540	
PUMP	2	17.2	
TOTAL		5036	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 4 CONCEPT 1
AMMONIA 120°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	101.5	50 M LIQUID LINE @ 3.26 CM ID 50 M VAPOR LINE @ 4.00 CM ID
HEAT EXCHANGERS (WET)	13	343.6	
ACCUMULATOR (WET)	1	12.6	
RADIATOR PENALTY	237.1 M^2	1399	
POWER PENALTY	146.0 W	23.2	
PUMP	2	10.0	
TOTAL		1890	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 4 CONCEPT 2
AMMONIA 120°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	824.6	50 M LIQUID LINE @ 9.30 CM ID 50 M VAPOR LINE @ 11.4 CM ID
HEAT EXCHANGERS (WET)	13	343.6	
ACCUMULATOR (WET)	1	19.7	
RADIATOR PENALTY	236.8 M^2	1397.4	
POWER PENALTY	0 W	0	
PUMP	0	0	
TOTAL		2585	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 4 CONCEPT 4A
AMMONIA 120°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	101.5	50 M LIQUID LINE @ 3.26 CM ID 50 M VAPOR LINE @ 4.00 CM ID
HEAT EXCHANGERS (WET)	13	343.6	
ACCUMULATOR (WET)	1	5.9	
RADIATOR PENALTY	236.5 M^2	1396	
POWER PENALTY	406 W	64.6	
PUMP	2	10.9	
TOTAL		1922	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 4 CONCEPT 4B
AMMONIA 120°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	77.8	50 M LIQUID LINE @ 2.86 CM ID 50 M VAPOR LINE @ 3.50 CM ID
HEAT EXCHANGERS (WET)	13	343.6	
ACCUMULATOR (WET)	1	5.4	
RADIATOR PENALTY	236.3 M^2	1394	
POWER PENALTY	763 W	34.3	
PUMP	2	15.4	
TOTAL		1871	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 1 CONCEPT 1

FREON 11 4°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	8.69	19 M LIQUID LINE @ 1.16 CM ID 19 M VAPOR LINE @ 3.50 CM ID
HEAT EXCHANGERS (WET)	6	11.5	
ACCUMULATOR (WET)	1	15.2	
RADIATOR PENALTY	26.9 M^2	158.4	
POWER PENALTY	0.54 W	0.09	
PUMP	2	2.3	
TOTAL		196.2	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 1 CONCEPT 2

FREON 11 4°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	8.69	19 M LIQUID LINE @ 1.16 CM ID 19 M VAPOR LINE @ 3.50 CM ID
HEAT EXCHANGERS (WET)	6	17.9	
ACCUMULATOR (WET)	1	1.64	
RADIATOR PENALTY	26.9 M^2	158.4	
POWER PENALTY	0 W	0	
PUMP	0	0	
TOTAL		186.7	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 1 CONCEPT 4A

FREON 11 4 ℃

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	8.69	19 M LIQUID LINE @ 1.16 CM ID 19 M VAPOR LINE @ 3.50 CM ID
HEAT EXCHANGERS (WET)	6	11.5	
ACCUMULATOR (WET)	1	1.24	
RADIATOR PENALTY	26.0 M^2	153.2	
POWER PENALTY	47.6 W	7.56	
PUMP	2	4.7	
TOTAL		186.9	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 1 CONCEPT 4B

FREON 11 4°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	4.26	19 M LIQUID LINE @ 0.66 CM ID 19 M VAPOR LINE @ 2.00 CM ID
HEAT EXCHANGERS (WET)	6	11.5	
ACCUMULATOR (WET)	1	0.98	
RADIATOR PENALTY	21.5 M^2	126.8	
POWER PENALTY	526 W	23.7	
PUMP	2	12.7	
TOTAL		179.9	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 1 CONCEPT 1

FREON 11 20°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	7.70	19 M LIQUID LINE @ 1.11 CM ID 19 M VAPOR LINE @ 3.00 CM ID
HEAT EXCHANGERS (WET)	6	12.2	
ACCUMULATOR (WET)	1	10.9	
RADIATOR PENALTY	17.1 M^2	101.1	
POWER PENALTY	0.65 W	0.10	
PUMP	2	2.3	
TOTAL		134.3	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 1 CONCEPT 2

FREON 11 20°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	8.05	19 M LIQUID LINE @ 1.14 CM ID 19 M VAPOR LINE @ 3.10 CM ID
HEAT EXCHANGERS (WET)	6	18.6	
ACCUMULATOR (WET)	1	1.64	
RADIATOR PENALTY	17.1 M^2	101.0	
POWER PENALTY	0 W	0	
PUMP	0	0	
TOTAL		129.3	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 1 CONCEPT 4A
FREON 11 20°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	7.70	19 M LIQUID LINE @ 1.11 CM ID 19 M VAPOR LINE @ 3.00 CM ID
HEAT EXCHANGERS (WET)	6	12.2	
ACCUMULATOR (WET)	1	1.23	
RADIATOR PENALTY	16.8 M^2	99.1	
POWER PENALTY	31.3 W	4.97	
PUMP	2	4.0	
TOTAL		129.2	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 1 CONCEPT 4B
FREON 11 20°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	4.57	19 M LIQUID LINE @ 0.74 CM ID 19 M VAPOR LINE @ 2.00 CM ID
HEAT EXCHANGERS (WET)	6	12.2	
ACCUMULATOR (WET)	1	1.04	
RADIATOR PENALTY	15.9 M^2	93.6	
POWER PENALTY	187 W	8.4	
PUMP	2	7.9	
TOTAL		127.7	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 1 CONCEPT 1
FREON 11 40°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	5.01	19 M LIQUID LINE @ 0.84 CM ID 19 M VAPOR LINE @ 2.00 CM ID
HEAT EXCHANGERS (WET)	6	12.7	
ACCUMULATOR (WET)	1	4.68	
RADIATOR PENALTY	11.24 M^2	66.3	
POWER PENALTY	2.49 W	0.40	
PUMP	2	2.4	
TOTAL		91.5	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 1 CONCEPT 2
FREON 11 40°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	8.02	19 M LIQUID LINE @ 1.19 CM ID 19 M VAPOR LINE @ 2.85 CM ID
HEAT EXCHANGERS (WET)	6	19.1	
ACCUMULATOR (WET)	1	1.69	
RADIATOR PENALTY	11.06 M^2	65.2	
POWER PENALTY	0 W	0	
PUMP	0	0	
TOTAL		94.1	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 1 CONCEPT 4A

FREON 11 40°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	6.71	19 M LIQUID LINE @ 1.05 CM ID 19 M VAPOR LINE @ 2.50 CM ID
HEAT EXCHANGERS (WET)	6	12.7	
ACCUMULATOR (WET)	1	1.22	
RADIATOR PENALTY	10.95 M^2	64.6	
POWER PENALTY	22.5 W	3.58	
PUMP	2	3.5	
TOTAL		92.3	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 1 CONCEPT 4B

FREON 11 40°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	5.01	19 M LIQUID LINE @ 0.84 CM ID 19 M VAPOR LINE @ 2.00 CM ID
HEAT EXCHANGERS (WET)	6	12.7	
ACCUMULATOR (WET)	1	1.11	
RADIATOR PENALTY	10.81 M^2	63.8	
POWER PENALTY	61.4 W	2.76	
PUMP	2	5.3	
TOTAL		90.7	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 1 CONCEPT 1
FREON 11 120°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	4.78	19 M LIQUID LINE @ 0.93 CM ID 19 M VAPOR LINE @ 1.50 CM ID
HEAT EXCHANGERS (WET)	6	13.8	
ACCUMULATOR (WET)	1	2.22	
RADIATOR PENALTY	3.39 M^2	20.0	
POWER PENALTY	2.60 W	0.41	
PUMP	2	2.6	
TOTAL		43.8	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 1 CONCEPT 2
FREON 11 120°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	10.2	19 M LIQUID LINE @ 1.59 CM ID 19 M VAPOR LINE @ 2.56 CM ID
HEAT EXCHANGERS (WET)	6	20.2	
ACCUMULATOR (WET)	1	1.89	
RADIATOR PENALTY	3.38 M^2	20.0	
POWER PENALTY	0 W	0	
PUMP	0	0	
TOTAL		52.3	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 1 CONCEPT 4A
FREON 11 120°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	4.78	19 M LIQUID LINE @ 0.93 CM ID 19 M VAPOR LINE @ 1.50 CM ID
HEAT EXCHANGERS (WET)	6	13.8	
ACCUMULATOR (WET)	1	1.14	
RADIATOR PENALTY	3.37 M^2	19.9	
POWER PENALTY	11.2 W	1.78	
PUMP	2	2.7	
TOTAL		44.1	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 1 CONCEPT 4B
FREON 11 120°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	3.76	19 M LIQUID LINE @ 0.78 CM ID 19 M VAPOR LINE @ 1.25 CM ID
HEAT EXCHANGERS (WET)	6	13.8	
ACCUMULATOR (WET)	1	1.07	
RADIATOR PENALTY	3.36 M^2	19.8	
POWER PENALTY	26.4 W	1.19	
PUMP	2	3.7	
TOTAL		43.4	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 4 CONCEPT 1

FREON 11 4°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	443.0	50 M LIQUID LINE @ 6.64 CM ID 50 M VAPOR LINE @ 20.0 CM ID
HEAT EXCHANGERS (WET)	13	838.0	
ACCUMULATOR (WET)	1	1305	
RADIATOR PENALTY	1883 M^2	11110	
POWER PENALTY	6.6 W	1.1	
PUMP	2	8.3	
TOTAL		13705	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 4 CONCEPT 2

FREON 11 4°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	443.0	50 M LIQUID LINE @ 6.64 CM ID 50 M VAPOR LINE @ 20.0 CM ID
HEAT EXCHANGERS (WET)	13	1323	
ACCUMULATOR (WET)	1	124.7	
RADIATOR PENALTY	1875 M^2	11061	
POWER PENALTY	0 W	0	
PUMP	0	0	
TOTAL		12952	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 4 CONCEPT 4A
FREON 11 4°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	443.0	50 M LIQUID LINE @ 6.64 CM ID 50 M VAPOR LINE @ 20.0 CM ID
HEAT EXCHANGERS (WET)	13	838.0	
ACCUMULATOR (WET)	1	92.6	
RADIATOR PENALTY	1822 M²	10749	
POWER PENALTY	2906 W	462	
PUMP	2	20.0	
TOTAL		12605	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 4 CONCEPT 4B
FREON 11 4°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	127.3	50 M LIQUID LINE @ 3.49 CM ID 50 M VAPOR LINE @ 10.5 CM ID
HEAT EXCHANGERS (WET)	13	838	
ACCUMULATOR (WET)	1	68.6	
RADIATOR PENALTY	1446 M²	8530	
POWER PENALTY	45444 W	2045	
PUMP	2	54.4	
TOTAL		11664	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 4 CONCEPT 1
FREON 11 20°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	287.1	50 M LIQUID LINE @ 5.53 CM ID 50 M VAPOR LINE @ 15.0 CM ID
HEAT EXCHANGERS (WET)	13	881.2	
ACCUMULATOR (WET)	1	715.6	
RADIATOR PENALTY	1207 M^2	7123	
POWER PENALTY	13.5 W	2.7	
PUMP	2	8.4	
TOTAL		9018	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 4 CONCEPT 2
FREON 11 20°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	303.8	50 M LIQUID LINE @ 6.24 CM ID 50 M VAPOR LINE @ 16.9 CM ID
HEAT EXCHANGERS (WET)	13	1367	
ACCUMULATOR (WET)	1	122.3	
RADIATOR PENALTY	1198 M^2	7068	
POWER PENALTY	0 W	0	
PUMP	0	0	
TOTAL		8861	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 4 CONCEPT 4A

FREON 11 20°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	409.3	50 M LIQUID LINE @ 6.64 CM ID 50 M VAPOR LINE @ 18.0 CM ID
HEAT EXCHANGERS (WET)	13	881.2	
ACCUMULATOR (WET)	1	96.6	
RADIATOR PENALTY	1179 M^2	6955	
POWER PENALTY	1554 W	247.0	
PUMP	2	18.1	
TOTAL		8607	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 4 CONCEPT 4B

FREON 11 20°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	186.5	50 M LIQUID LINE @ 4.43 CM ID 50 M VAPOR LINE @ 12.0 CM ID
HEAT EXCHANGERS (WET)	13	881.2	
ACCUMULATOR (WET)	1	78.7	
RADIATOR PENALTY	1136 M^2	6700	
POWER PENALTY	8712 W	392	
PUMP	2	24.5	
TOTAL		8263	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 4 CONCEPT 1

FREON 11 40°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	185.3	50 M LIQUID LINE @ 4.61 CM ID 50 M VAPOR LINE @ 11.0 CM ID
HEAT EXCHANGERS (WET)	13	915.5	
ACCUMULATOR (WET)	1	372.2	
RADIATOR PENALTY	786.5 M^2	4640	
POWER PENALTY	343 W	5.5	
PUMP	2	8.6	
TOTAL		6128	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 4 CONCEPT 2

FREON 11 40°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	322.1	50 M LIQUID LINE @ 6.58 CM ID 50 M VAPOR LINE @ 15.7 CM ID
HEAT EXCHANGERS (WET)	13	1401.0	
ACCUMULATOR (WET)	1	125.0	
RADIATOR PENALTY	774.1 M^2	4567	
POWER PENALTY	0 W	0	
PUMP	0	0	
TOTAL		6415	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 4 CONCEPT 4A

FREON 11 40°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	337.5	50 M LIQUID LINE @ 6.28 CM ID 50 M VAPOR LINE @ 15.0 CM ID
HEAT EXCHANGERS (WET)	13	915.5	
ACCUMULATOR (WET)	1	93.8	
RADIATOR PENALTY	768.3 M^2	4533	
POWER PENALTY	1075 W	171.0	
PUMP	2	17.2	
TOTAL		6068	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 4 CONCEPT 4B

FREON 11 40°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	185.3	50 M LIQUID LINE @ 4.61 CM ID 50 M VAPOR LINE @ 11.0 CM ID
HEAT EXCHANGERS (WET)	13	915.5	
ACCUMULATOR (WET)	1	80.8	
RADIATOR PENALTY	756.8 M^2	4465	
POWER PENALTY	4233 W	190.5	
PUMP	2	20.9	
TOTAL		5858	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 4 CONCEPT 1
FREON 11 120°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	138.1	50 M LIQUID LINE @ 4.36 CM ID 50 M VAPOR LINE @ 7.00 CM ID
HEAT EXCHANGERS (WET)	13	992.6	
ACCUMULATOR (WET)	1	126.9	
RADIATOR PENALTY	238.2 M^2	1405	
POWER PENALTY	110 W	17.5	
PUMP	2	10.0	
TOTAL		2690	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 4 CONCEPT 2
FREON 11 120°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	481.7	50 M LIQUID LINE @ 8.72 CM ID 50 M VAPOR LINE @ 14.0 CM ID
HEAT EXCHANGERS (WET)	13	1478	
ACCUMULATOR (WET)	1	135.2	
RADIATOR PENALTY	236.8 M^2	1397	
POWER PENALTY	0 W	0	
PUMP	0	0	
TOTAL		3492	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 4 CONCEPT 4A
FREON 11 120°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	218.8	50 M LIQUID LINE @ 5.60 CM ID 50 M VAPOR LINE @ 9.00 CM ID
HEAT EXCHANGERS (WET)	13	992.6	
ACCUMULATOR (WET)	1	84.4	
RADIATOR PENALTY	236.4 M^2	1395	
POWER PENALTY	517 W	82.2	
PUMP	2	12.7	
TOTAL		2786	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 4 CONCEPT 4B
FREON 11 120°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	138.1	50 M LIQUID LINE @ 4.36 CM ID 50 M VAPOR LINE @ 7.00 CM ID
HEAT EXCHANGERS (WET)	13	992.6	
ACCUMULATOR (WET)	1	77.0	
RADIATOR PENALTY	235.4 M^2	1389	
POWER PENALTY	1686 W	75.9	
PUMP	2	18.1	
TOTAL		2691	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 1 CONCEPT 1

WATER 40°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	16.3	19 M LIQUID LINE @ 0.55 CM ID 19 M VAPOR LINE @ 3.50 CM ID
HEAT EXCHANGERS (WET)	6	11.2	
ACCUMULATOR (WET)	1	9.9	
RADIATOR PENALTY	11.64 M^2	68.7	
POWER PENALTY	0.06 W	0.01	
PUMP	2	1.6	
TOTAL		107.7	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 1 CONCEPT 2

WATER 40°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	16.3	19 M LIQUID LINE @ 0.55 CM ID 19 M VAPOR LINE @ 3.50 CM ID
HEAT EXCHANGERS (WET)	6	11.2	
ACCUMULATOR (WET)	1	0.25	
RADIATOR PENALTY	11.64 M^2	68.7	
POWER PENALTY	0 W	0	
PUMP	0	0	
TOTAL		96.5	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 1 CONCEPT 3

WATER 40°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	16.3	19 M LIQUID LINE @ 0.55 CM ID 19 M VAPOR LINE @ 3.50 CM ID
HEAT EXCHANGERS (WET)	6	11.2	
ACCUMULATOR (WET)	1	0.25	
RADIATOR PENALTY	11.64 M^2	68.7	
POWER PENALTY	0 W	0	
PUMP	5	10.5	
TOTAL		106.9	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 1 CONCEPT 4A

WATER 40°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	21.0	19 M LIQUID LINE @ 0.70 CM ID 19 M VAPOR LINE @ 4.50 CM ID
HEAT EXCHANGERS (WET)	6	11.2	
ACCUMULATOR (WET)	1	0.29	
RADIATOR PENALTY	10.90 M^2	64.3	
POWER PENALTY	110 W	17.4	
PUMP	2	6.4	
TOTAL		120.6	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 1 CONCEPT 4B

WATER 40°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	16.3	19 M LIQUID LINE @ 0.55 CM ID 19 M VAPOR LINE @ 3.50 CM ID
HEAT EXCHANGERS (WET)	6	11.2	
ACCUMULATOR (WET)	1	0.25	
RADIATOR PENALTY	10.64 M^2	62.8	
POWER PENALTY	186 W	8.4	
PUMP	2	7.9	
TOTAL		106.9	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 1 CONCEPT 1
WATER 120°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	5.51	19 M LIQUID LINE @ 0.28 CM ID 19 M VAPOR LINE @ 1.00 CM ID
HEAT EXCHANGERS (WET)	6	11.2	
ACCUMULATOR (WET)	1	0.77	
RADIATOR PENALTY	3.49 M^2	20.6	
POWER PENALTY	0.85 W	0.13	
PUMP	2	1.63	
TOTAL		39.8	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 1 CONCEPT 2
WATER 120°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	7.5	19 M LIQUID LINE @ 0.39 CM ID 19 M VAPOR LINE @ 1.38 CM ID
HEAT EXCHANGERS (WET)	6	11.2	
ACCUMULATOR (WET)	1	0.21	
RADIATOR PENALTY	3.41 M^2	20.1	
POWER PENALTY	0 W	0	
PUMP	0	0	
TOTAL		39.0	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 1 CONCEPT 3
WATER 120°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	5.51	19 M LIQUID LINE @ 0.28 CM ID 19 M VAPOR LINE @ 1.00 CM ID
HEAT EXCHANGERS (WET)	6	11.2	
ACCUMULATOR (WET)	1	0.20	
RADIATOR PENALTY	3.49 M^2	20.6	
POWER PENALTY	0 W	0	
PUMP	5	11.15	
TOTAL		48.7	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 1 CONCEPT 4A

WATER 120°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	8.1	19 M LIQUID LINE @ 0.42 CM ID 19 M VAPOR LINE @ 1.50 CM ID
HEAT EXCHANGERS (WET)	6	11.2	
ACCUMULATOR (WET)	1	0.22	
RADIATOR PENALTY	3.37 M^2	19.9	
POWER PENALTY	20.0 W	3.17	
PUMP	2	3.4	
TOTAL		46.0	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 1 CONCEPT 4B

WATER 120°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	38 M	6.8	19 M LIQUID LINE @ 0.35 CM ID 19 M VAPOR LINE @ 1.25 CM ID
HEAT EXCHANGERS (WET)	6	11.2	
ACCUMULATOR (WET)	1	0.21	
RADIATOR PENALTY	3.35 M^2	19.8	
POWER PENALTY	43.4 W	1.95	
PUMP	2	4.5	
TOTAL		44.4	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 4 CONCEPT 1
WATER 40°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	332.8	50 M LIQUID LINE @ 3.74 CM ID 50 M VAPOR LINE @ 24.0 CM ID
HEAT EXCHANGERS (WET)	13	631.5	
ACCUMULATOR (WET)	1	1218	
RADIATOR PENALTY	803.7 M^2	4742	
POWER PENALTY	0.82 W	0.1	
PUMP	2	4.3	
TOTAL		6929	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 4 CONCEPT 2
WATER 40°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	332.8	50 M LIQUID LINE @ 3.74 CM ID 50 M VAPOR LINE @ 24.0 CM ID
HEAT EXCHANGERS (WET)	13	631.5	
ACCUMULATOR (WET)	1	17.7	
RADIATOR PENALTY	803.7 M^2	4742	
POWER PENALTY	0 W	0	
PUMP	0	0	
TOTAL		5724	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 4 CONCEPT 3
WATER 40°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	332.8	50 M LIQUID LINE @ 3.74 CM ID 50 M VAPOR LINE @ 24.0 CM ID
HEAT EXCHANGERS (WET)	13	631.5	
ACCUMULATOR (WET)	1	17.7	
RADIATOR PENALTY	803.7 M^2	4742	
POWER PENALTY	0 W	0	
PUMP	12	613.0	
TOTAL		6337	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 4 CONCEPT 4A

WATER 40°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	432.7	50 M LIQUID LINE @ 4.67 CM ID 50 M VAPOR LINE @ 30.0 CM ID
HEAT EXCHANGERS (WET)	13	631.5	
ACCUMULATOR (WET)	1	21.5	
RADIATOR PENALTY	768.3 M^2	4533	
POWER PENALTY	6150 W	977.8	
PUMP	2	22.7	
TOTAL		6619	

PHYSICAL CHARACTERISTICS SUMMARY

CASE 4 CONCEPT 4B

WATER 40°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	308.9	50 M LIQUID LINE @ 3.50 CM ID 50 M VAPOR LINE @ 22.5 CM ID
HEAT EXCHANGERS (WET)	13	631.5	
ACCUMULATOR (WET)	1	16.9	
RADIATOR PENALTY	758.5 M^2	4475	
POWER PENALTY	8980 W	404.1	
PUMP	2	24.5	
TOTAL		5861	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 4 CONCEPT 1
WATER 120°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	89.0	50 M LIQUID LINE @ 1.69 CM ID 50 M VAPOR LINE @ 6.00 CM ID
HEAT EXCHANGERS (WET)	13	631.5	
ACCUMULATOR (WET)	1	72.7	
RADIATOR PENALTY	241.6 M^2	1426	
POWER PENALTY	12.7 W	2.1	
PUMP	2	4.4	
TOTAL		2225	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 4 CONCEPT 2
WATER 120°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	115.1	50 M LIQUID LINE @ 2.13 CM ID 50 M VAPOR LINE @ 7.55 CM ID
HEAT EXCHANGERS (WET)	13	631.5	
ACCUMULATOR (WET)	1	12.5	
RADIATOR PENALTY	238.4 M^2	1406	
POWER PENALTY	0 W	0	
PUMP	0	0	
TOTAL		2166	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 4 CONCEPT 3
WATER 120°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	89.0	50 M LIQUID LINE @ 1.69 CM ID 50 M VAPOR LINE @ 6.00 CM ID
HEAT EXCHANGERS (WET)	13	631.5	
ACCUMULATOR (WET)	1	11.7	
RADIATOR PENALTY	241.6 M^2	1426	
POWER PENALTY	0 W	0	
PUMP	12	649.3	
TOTAL		2807	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 4 CONCEPT 4A
WATER 120°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	198.1	50 M LIQUID LINE @ 3.38 CM ID 50 M VAPOR LINE @ 12.0 CM ID
HEAT EXCHANGERS (WET)	13	631.5	
ACCUMULATOR (WET)	1	15.7	
RADIATOR PENALTY	236.6 M^2	1396	
POWER PENALTY	407 W	64.7	
PUMP	2	10.9	
TOTAL		2317	

PHYSICAL CHARACTERISTICS SUMMARY
CASE 4 CONCEPT 4B
WATER 120°C

COMPONENT	QTY	WEIGHT (KG)	COMMENTS
LINES (WET)	100 M	140.8	50 M LIQUID LINE @ 2.54 CM ID 50 M VAPOR LINE @ 9.0 CM ID
HEAT EXCHANGERS (WET)	13	631.5	
ACCUMULATOR (WET)	1	13.4	
RADIATOR PENALTY	236.1 M^2	1393	
POWER PENALTY	990 W	44.5	
PUMP	2	16.3	
TOTAL		2239	

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