Heat-Pump-Augmented Spacecraft Heat-Rejection Systems

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Future military space missions will introduce significant new technological needs for spacecraft energy systems. It is generally accepted that spacecraft heat-rejection systems that use heat pumps to boost the radiator temperature will reduce the radiator area. However, these systems must also result in weight savings and high reliability. This paper discusses several heat-pump configurations and discusses the benefits of a combination of the heat-pump system and the thermal management/thermal transport system.

Introduction

VARIOUS technologies need to be evaluated to determine the best technology (in terms of weight, efficiency, radiator area, etc.) for a particular application. Several research papers have illustrated the benefits of heat-pump-augmented spacecraft heat-rejection systems in the 10 k–100 kW range. The results indicate that even modest increases in radiator rejection temperature, made feasible by the use of a heat pump, lowered the radiator size and thus system weight. A heat-pump system is shown to weigh less than the pumped fluid system at all loads, but at higher loads the differences become significant, making the heat pump attractive for high-power cooling.[1-5] In addition, future directed energy applications will have pulsed power requirements for a peak-to-average heat flux ratio as high as 100,000: 1.[5]

Practical Spacecraft Vapor-Compression Heat-Pump Configurations

Designing a heat pump for spacecraft operations would require the use of onboard power to drive the compressor. However, if high-temperature waste heat is available (such as waste heat from a nuclear or solar dynamic power system), then a thermally driven system, rather than an electrically driven system, might be preferred. Clearly, the development of both thermally and electrically driven heat-pump systems are needed to satisfy future applications. In the area of thermally driven systems, the engineer has the choice of several chemical heat-pump systems or the use of a heat engine to power a vapor-compression heat pump. The chemical heat pumps (absorption, metal-hydride, or complex-compound heat pumps) typically have only liquid pumps and therefore tend to be falsely represented as less complex than heat-engine vapor-compression (Rankine-Rankine or Stirling-Rankine) systems. However, the metal-hydride and complex-compound systems are intermittent, or batch, systems requiring several systems operating at different intervals to approximate continuous operation. Besides the control and plumbing problems associated with these designs, chemical heat pumps also tend to have an inherently low coefficient of performance (COP_c) and are larger and heavier than many systems of equal capacity. The absorption systems are continuous systems but also tend to be both inherently low in COP_c and heavy.

A heat engine driving a vapor-compression heat pump typically outperforms an absorption heat-pump system because the heat engine can use the high-temperature heat available (from nuclear power source waste heat, for example), and typically the absorption system cannot (although new absorption-refrigerant pairs could solve that problem). In addition, the absorption systems tend to be larger and heavier because they require a large liquid-vapor surface area to achieve the equilibrium concentrations necessary for optimum refrigerant transport. In terms of system performance, size, and mass, a thermally driven heat pump, composed of a high-temperature heat engine and vapor-compression heat pump, appears superior to both adsorption and absorption alternatives.[1]

Brayton cycles are sometimes used in place of Rankine cycles for power conversion (and reverse Brayton cycles are sometimes used in place of reverse Rankine cycles for heat pumps). Brayton cycles use stable, gaseous fludis over the entire temperature range of the cycle. In spite of this thermal stability, the Brayton cycles have inherent disadvantages when compared to Rankine cycles. The first disadvantage in using Brayton cycles is that they typically have lower cycle efficiencies because only the sensible heating of the working fluid is available to transfer energy in the cycle,[6-8] resulting in larger mass flow rates (compared to the Rankine cycle, which uses the latent heat of vaporization of the fluid to carry a significant amount of the energy). The second disadvantage is that the weight of the Brayton cycle is typically higher than the weight of a comparable Rankine cycle due to the low heat-transfer coefficient of the gaseous working fluid (which results in larger heat exchangers) and higher flow rates (or higher pressures) for the Brayton system. The weight difference between comparable Rankine and Brayton cycles is documented in the literature.[9]

In terms of Stirling-powered heat pumps, the only potentially reliable design would use a free-piston configuration in which the free-piston Stirling engine drives a third compression piston within the hermetically sealed Stirling enclosure. In our adaptation of this approach, the Stirling working fluid and the vapor-compression heat pump working fluid were envisioned to be the same, allowing the system to be hermetically sealed. The working fluid was selected to insure superheated conditions in the Stirling system. Althought the Stirling-powered vapor-compression heat pump has the potential for higher performance due to the higher theoretical performance of the Stirling heat engine (compared to the Rankine heat engine), this performance benefit has never been practically demonstrated in any Stirling heat engine. In addition, the Stirling heat engine will be heavier than the Rankine heat engine, and long-term reliability of the Stirling heat engine has not been demonstrated. Finally (for the free-piston configura-

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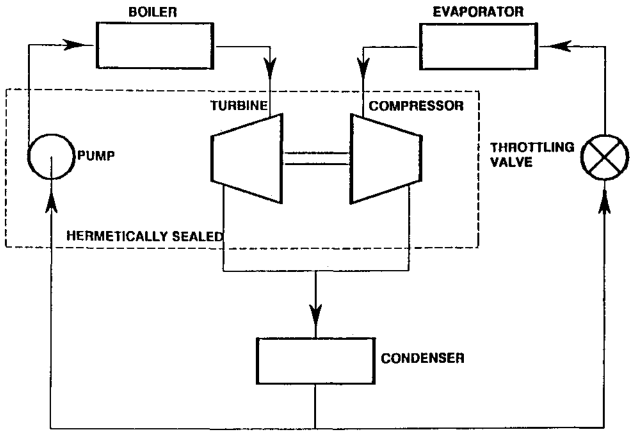

Fig. 1 Thermally driven heat pump.

tion), it is not clear that the effect of the variable compressor load on the motion and frequency of the Stirling displacer is well understood.

Because of the advantages of vapor-compression heat pumps, we have been investigating both electrically driven and Rankine-powered thermally driven vapor-compression heat pumps in great detail.[1-3,8] Our performance improvements to date use a combination of innovative designs and alternative working fluids. For the thermally driven application, when both the power cycle and the heat-pump cycle use the same working fluid, the system can be hermetically sealed, avoiding long-term leakage problems. The typical problem with this concept is that the working fluid for a Rankine heat engine at the higher hot-side heat engine temperature is typically not the appropriate working fluid for the vapor-compression heat pump (especially at the low evaporator temperature).

Figure 1 displays a thermally driven heat pump (Rankine-Rankine system) that uses a common condenser and common refrigerant. This design allows for the compressor, expander, and pump to all be hermetically sealed together. This hermetically sealed design is to be contrasted to a design in which the heat engine uses one working fluid and the heat pump a second working fluid. In the second design, the expander and compressor cannot be hermetically sealed individually because a rotating shaft would have to protrude from each, and this would be a source of leaks. Alternatively, a magnetic coupling would have to be used, adding significant weight to the overall system. A single refrigerant system is preferable (from a minimum leakage standpoint) because the system could be effectively sealed (no protruding rotating shafts).

This basic Rankine-Rankine design has performance and weight advantages over the chemical or absorption heat-pump systems but still has the "perceived" disadvantage of a rotating or reciprocating expander and compressor. Although such devices have been shown to operate for long periods, alternative designs that incorporate no moving parts always appear attractive. We are currently investigating an alternative device to replace the expander and compressor of the Rankine vapor-compression thermally driven heat pump.[2]

New Fluids For High-Temperature Heat-Pump Applications

Typically, three types of fluids would be applicable for a thermally driven vapor-compression heat-pump application: high-molecular-weight organic fluids, liquid metals, and water. High-molecular-weight organic fluids are used in thermodynamic cycles with temperatures up to about 700 K. Liquid metals are currently used as working fluids in nuclear and solar dynamic power cycles with heat sources as high as 1500 K. However, the majority of the liquid metals have melting points above 500 K and are therefore solids, or have extremely low vapor pressures at the expected 300 K evaporator temperature of the heat pump. Water is stable at high temperatures but is typically not used for spacecraft applications because of its low vapor-phase density. Freon-type refrigerants are clearly unstable at the expected heat-source temperatures and are therefore unsuitable for application in the hermetically sealed, thermally driven heat-pump application. Other organic working fluids are currently receiving consideration as working fluids in many spacecraft thermal management applications. However, these fluids suffer from thermal decomposition at temperatures around 700 K.

We have identified a family of new working fluids and are currently performing experiments to verify their thermal transport properties and their thermal stability for operating temperatures as high as 1000 K.

Heat-pump performance improvements have also been directed toward the use of nonazeotropic blends of newly identified working fluids. Nonazeotropic mixtures are those mixtures that exhibit a temperature change as they either condense or evaporate. This is because they are comprised of components with different normal boiling points. Several advantages have been both theoretically calculated and experimentally measured for heat pumps that use nonazeotropic mixtures in residential heat pumps.[1] Among the benefits are improved coefficient of performance (COP_c) (about 10%, as the result of the gliding temperatures in the saturation zone), multitemperature-level evaporators or condensers, and refrigeration composition change, which alters the capacity of the working fluid as load requirements vary. The last of these benefits may make these working fluids ideal for spacecraft applications because there is not only capacity control but also energy storage capability in the nonazeotropic mixture.

Benefits of Heat-Pump-Augmented Radiator Systems

To realistically identify the area and weight savings that are possible by the use of a heat pump, a few representative calculations need to be performed.

Table 1 Projected radiator area and system mass reductions for a thermally driven heat pump

$T_{evap} = 300$ K	$T_{cond} = 400$ K	$T_{boil} = 1000$ K
$P_{max} = 150$ atm	$P_{min} = 0.005$ atm	Cooling load = 10 kW
Pump efficiency = 0.85	Turbine efficiency = 0.85	Compressor efficiency = 0.85
$T_{space} = 227$ K	Radiator mass = 10 kg/m^2	Radiator emissivity = 0.80
Heat pump mass = 11	kg/kW-cooled	

Fluid name	COP_c	Radiator area reduction, %	System mass reduction, %
MEC-79[a]	0.58	36	9.0
MEC-3	0.56	34	7.6
MEC-73	0.55	34	6.9
MEC-66	0.52	31	4.5
Water	0.48	28	0.7
R-11[b]	0.26	−13	−39.7 (increase)
Toluene[b]	0.14	−82	−112.0 (increase)

[a]MEC fluids are Patent Pending Proprietary Mainstream Engineering Corp. Working Fluids.
[b]Fluid unstable over the operating temperature range.

Table 2 Projected radiator area and system mass reductions for a thermally driven heat pump

$T_{evap} = 300$ K	$T_{cond} = 400$ K	$T_{boil} = 1000$ K	
$P_{max} = 150$ atm	$P_{min} = 0.005$ atm	Cooling load = 10 kW	
Pump efficiency = 0.85	Turbine efficiency = 0.85	Compressor efficiency = 0.85	
$T_{space} = 227$ K	Radiator emissivity = 0.80		
HPL system savings = 0.7 kg/kW-cooled			
Heat-pump specific mass kg/kW-cooled	**Radiator specific mass, kg/m²**	**Radiator area reduction, %**	**System mass reduction, %**
11.0	10.0	28.0	2.4
5.5	20.0	28.0	22.0

Table 3 Projected radiator area and system mass reductions for an electrically driven heat pump

$T_{power\ system\ reject} = 1000$ K		$T_{evap} = 300$ K	
Compressor efficiency = 0.85		$T_{cond} = 400$ K	
Power system efficiency = 0.08		$T_{space} = 227$ K	
Heat pump mass = 6 kg/kW-cooled		Radiator mass = 10 kg/m²	
Power system mass = 30 kg/kW-electric		Radiator emissivity = 0.80	
Fluid name	**COP_c**	**Radiator area reduction, %**	**System mass reduction, %**
MEC-10	1.92	61	7.4
MEC-3	1.86	60	5.6
R-10	1.74	59	1.5
Water	1.64	58	−2.0 (increase)
R-11	1.45	55	−10.0 (increase)
Toluene	0.91	43	−53.0 (increase)

Radiation heat transfer in space has been accurately modeled in the past by using an effective space temperature of 227 K.[1–4] Also, by assuming a typical radiator effectiveness of 0.8, the heat rejection from a spacecraft radiator can be calculated from the Stefan-Boltzmann relation, Eq. (1).

$$q = \sigma\epsilon(T^4_{rad} - T^4_s) \tag{1}$$

where

q = heat flux per unit area, W/m²
σ = Stefan-Boltzmann constant, W/m²K⁴
ϵ = emissivity of the radiator surface
T_{rad} = radiator temperature, K
T_s = effective space temperature, 227 K

For radiator surface tempertures of 300, 400, and 1000 K, the radiator heat flux is 247.0, 1040.0, and 45,000 W/m², respectively. Assuming a 10-kW cooling-requirement base-line system that does not use a heat pump and therefore uses a 300 K radiator, the area required is 40.5 m². In addition, waste heat from the electrical power system will also be rejected from a high-temperature (1000 K) radiator.

If a thermally driven heat pump with a COP_c of 0.56 is proposed (MEC-3, Table 1), then for a cooling requirement of 10 kW, the condenser energy rejection would be 28 kW. It is useful to compare the savings in radiator area and total system mass that would result from the use of this heat-pump system.

The heat-pump system would use 18 kW of energy from the 100 K waste heat energy source to power the heat pump. Because this energy is now rejected for a 400 K radiator, the base-line system area requirement should also include the radiator area necessary for the rejection of this amount of energy from the 1000 K radiator. This results in an additional area of 0.4 m². Therefore, the base-line no-heat-pump system requires an area of 40.9 m² to reject 10 kW at 300 K and 18 kW at 1000 K.

The heat-pump system, with a COP_c of 0.56 (MEC-3, Table 1), will reject the 28 kW of energy at 400 K and will therefore require 27 m² of radiator area. Thus, the heat-pump system would result in an area savings of 14 m² for this 10-kW cooling system; that is, a reduction in the area of 34% (14.0/40.9).

Typical radiation-hardened radiators currently weigh approximately 20 kg/m². However, it has been recently argued that 10 kg/m² is a more meaningful long-term projection, and it is this radiator specific mass that will be assumed. The assumption of a 10 kg/m² radiator results in a radiator mass savings of 140 kg (14 m² at 10 kg/m²). However, the savings in radiator area require the use of a heat pump, and the additional system mass due to the heat pump must be considered. A recent study found that electrically driven terrestrial residential and commercial split systems ranged in specific mass from 11–18 kg/kW-cooled.[10] These household and commercial heat-pump systems were never designed to be lightweight and actual spacecraft heat pumps should be considerably lighter. An electrically driven sliding-van rotary compressor environmental control unit (ECU) for the LANTIRN electro-optical pod system was built with a specific mass of 3.8 kg/kW-cooled.[11] Performance predictions for a scroll compressor ECU unit with specific mass estimate of 5.85 kg/kW-cooled have been developed.[12] In addition, for this scroll compressor system estimate, the mass of the electric motor was 0.7 kg/kW-cooled. The mass of a thermally driven Rankine-Rankine system has therefore been estimated as *twice* the specific mass of the electrically driven scroll heat pump (without an electric motor mass). This results in a conservative specific mass of 10.3 kg/kW-cooled.

As an added conservative measure, 11.0 kg/kW-cooled was used as the specific mass in the following comparisons. Using this heat-pump mass, the total system mass savings was 31 kg (141 kg-110 kg). Using these very conservative mass estimates resulted in a mass savings of 7.6% (31/409).

The preceding paragraphs described the calculation procedure for the mass and radiator area savings when using MEC-3 working fluid. These calculations were repeated for various working fluids and are listed in Table 1. Because the projected savings are a function of the power system specific mass, the heat-pump specific mass, and the radiator specific mass, these calculations have been repeated for a variety of component specific masses and the results are presented in Table 2. Similar computations were also performed to exhibit the bene-

Table 4 Projected radiator area and system mass reductions for an electrically driven water heat pump

$T_{evap} = 300$ K	$T_{cond} = 400$ K		$T_{space} = 227$ K	
$P_{max} = 150$ atm	$P_{min} = 0.005$ atm		Cooling load = 10 kW	
Compressor efficiency = 0.85			Radiator emissivity = 0.80	
HPL system savings = 0.7 kg/kW-cooled				

Heat-pump mass, kg/kW-cooled	Power system mass, kg/kW-electric	Radiator mass, kg/m^2	Radiator area reduction, %	System mass reduction, %
6.0	30.0	10.0	58.0	−0.3
6.0	50.0	20.0	58.0	13.0
3.8	50.0	20.0	58.0	17.0

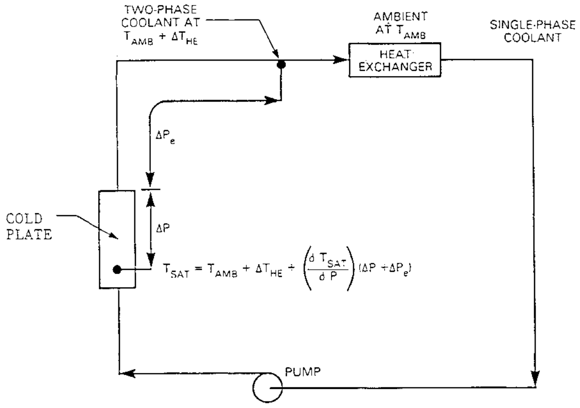

Fig. 2 A simple two-phase flow loop.

fits of electrically driven heat pumps on spacecraft heat-rejection systems. The projected mass and radiator savings for electrically driven systems were presented in Tables 3 and 4. Tables 1–4 clearly indicate that with the proper working fluid, a heat pump can realistically result in significant radiator area reduction and overall system mass savings. Fluids such as Toluene and R-11 are not being considered as potential heat-pump working fluids. These fluids are presented to illustrate that the use of an improper working fluid can seriously affect the area and mass savings. These particular fluids were selected because several heat-pump tradeoff studies have erroneously used these fluids in the past.

Benefits of Using the Heat Pump as the Heat Transport Loop

The current trend for longer on-orbit spacecraft operations has led to a modular approach to satellite subsystem design. The thermal control subsystems must be adaptable to mechanical configurations and high heat-load changes. System modularity requires connectable/disconnectable devices through which thermal energy can be transported. Further complicating the task are the projected heat loads and heat flux densities, which are expected to be several orders of magnitude higher than those of current spacecraft. The intent of this section is to highlight the benefits that would be obtainable by the substitution of heat-pump components into the thermal transport loop. This substitution would reduce the control problems, simplify the connection and disconnection problems, remove the liquid/vapor separation problems, and simplify the lubrication problems.

To date, all potential two-phase thermal transport systems fall into three broad categories: mechanically pumped loop (MPL) systems, capillary pumped loop (CPL) systems, and hybrid (capillary and mechanically pumped) systems. Each of these categories has significant variations depending on the system developer. This is especially true of the mechanical systems.

MPL systems offer the greatest transport distances, can accommodate the largest heat loads and energy densities, and can accommodate the widest variations in duty cycle and operational conditions. The mechanically pumped systems are capable of transporting the heat loads over the distances projected for future spacecraft applications.[13] Pumped systems are sized for specific heat loads and line lengths. As these paramreters change, the system characteristics can change as well. This is most noticeable in the turn-down ratio limits on the minimum heat load that is suitable for normal operations. In general, MPL systems require minimum system heat loads to be greater than 10% of the design load for the system[13,14]. This requirement stems from problems caused by boiling hysteresis and a design inflexibility that is removed in the proposed design configuration.

A further restriction and requirement of the control system is the pump inlet subcooling. All pumped systems require subcooling at the pump inlet. Normally, this subcooling is about 5–7 °C. Figure 2 displays a simple schematic of a two-phase flow loop. The drop in saturation temperature is a function of the system pressure drop and the slope of the saturation curve (for the temperature range used). A large pressure drop has two major disadvantages: 1) energy is wasted and heat is generated in the pumping of fluid throughout this loop; and 2) the condenser, which is rejecting heat to space, is operating at a lower temperature. This means the radiators must be larger because of their lower temperature.

Every two-phase system has a different pressure drop, and every effort is usually made to minimize this drop because it results in lower rejection temperatures. However, for a given heat load and fluid, the latent heat of vaporization dictates the mass flow rate, and therefore the only way to reduce the pressure drop is to use larger pipe diameters. In the regions in which two phases exist in the pipe, the majority of the designs use a fluid management system that separates the liquid and vapor into two streams. This added complexity is performed to insure that two phase flow instabilities do not occur and to provide positive assurance of the flow conditions in the systems. Because two-phase transport systems are already typically lighter and more compact than single-phase systems, the requirement to use complex fluid management systems is tolerated.

Finally, another concern with two-phase or single-phase pumped coolant is the operation lifetime of the liquid pump. The two-phase loops require lower pumping powers and therefore have smaller pumps for equal capacity, but both systems have the same pump reliability problem. When typical Freon-type refrigerants are used, long life is a well-established concern. The problem is that the fluorocarbons are excellent solvents and the lubricants are dissolved into the liquid and difficult to separate. Teflon-bearing surfaces have not adequately solved the problem because the Teflon has shown large wear rates.

The heat-pump loop (HPL) concept is an evolution of the two-phase thermal transport loop and has all of the advantages of a heat pump; namely, elevated condenser temperatures resulting in reductions in radiator sizes. Furthermore, the proposed approach removes or reduces the drawbacks of the pumped coolant systems.

The basic approach is to use a compressor rather than a pump to transport the working fluid around the loop. A schematic of the HPL system is displayed in Fig. 3. Such a configuration transforms the thermal transport loop into a heat pump. Use of a compressor rather than a pump will drastically reduce the concerns of lubrication and potential operational lifetimes that exist with pumped coolant systems. When a compressor instead of a liquid pump is used, the outlet from the compressor will be a superheated vapor with entrained liquid oil. A large fraction of the liquid oil can be easily separated from the vapor. The oil, once separated, can be injected directly into the required frictional surfaces of the compressor.

The increased pressure-drop tolerance in the HPL system means that larger pressure drops for fluid flowing through the modular disconnect devices, and/or larger temperature drops across these devices, can be tolerated. The ability to accommodate larger pressure and temperature drops in the cold plates allows for significantly higher heat fluxes because swirl-induced or turbulent-flow conditions could be tolerated, as well as heat flux spreading within the heat sink.

Fluid management and thermal control in the HPL system under varying operational requirements is more flexible and the control system is more localized. The proposed HPL system would, of course, result in increases in the pumping power requirements when compared to two-phase pumped coolant loops, due to the high specific volume of vapor relative to liquid. However, in situations in which a heat pump in series with the thermal management system results in a reduction in radiator area and system mass (as discussed in the previous section), the HPL system will result in even greater reductions in system mass with very little impact on the total system power requirements. This is because for thermal management systems that incorporate a heat pump in some fashion, the energy required to pump the thermal management coolant is only a small fraction of the energy requirement of the entire system. Thus, even though the fluid transport pumping power has increased, this power requirement is only a small fraction of the total power, and therefore the total power by a less efficient "pumping" method is outweighted by the reductions in system mass and radiator area that are obtainable by combining systems (as well as the other benefits stated above). For example, Braun et al.[13] discuss two-phase pumped coolant systems and state that a reasonable temperature drop between the evaporator and condenser of such systems is on the order of 5–7 °C. Using a 6 °C temperature drop for R-113 (evaporating near 300 K) results in an evaporator-to-condenser pressure drop of only 10.4 kPa. Based on this information, the heat-pump performance calculations were performed with and without an additional 10 kPa pressure drop between the evaporator and condenser. In a R-113 heat-pump system (operating with a 300 K evaporator and 400 K condenser, a compressor efficiency of 66%, a motor efficiency of 80%), the overall COP_c of this system can be calculated to be 1.05. In this system the high-side pressure is 784 kPa and the low-side pressure is 58 kPa. The addition of a 10-kPa evaporator-to-condenser frictional pressure drop would decrease the COP_c to 0.94. In spite of this decrease in COP_c, a heat-pump thermal transport system with a COP_c of 0.94 (rejects 2.06 times the energy at 400 K compared to the energy it would have to reject at 300 K) would result in a heat-pump system radiator area that is 49% of the no-heat-pump (300 K) radiator area (these calculations assume a 227 K effective space temperature). However, the additional energy requirements to operate the heat pump would reduce this radiator area saving to 43% (because of the additional power conversion radiator area required and assuming 10% power conversion efficiency and 1000 K power conditioning radiators). It is also reasonable to assume that the integral heat-pump system will weigh less than a separate thermal transport system and heat pump.

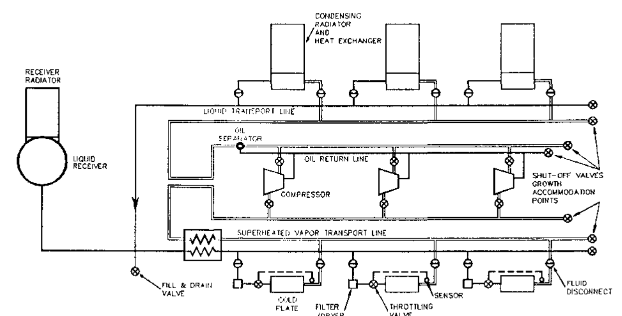

Fig. 3 A simple heat-pump flow loop.

Conclusions and Proposed Future Efforts

The authors believe that thermally driven heat-pump-augmented fluid transport/thermal management systems are superior to single-phase or two – phase pumped coolant designs. Heat-pump spacecraft heat-rejection designs can result in reductions in radiator area, which the other designs cannot, and more importantly should be able to reduce system weight. One design challenge that faces a pumped coolant loop, namely, the long-term reliability of single-phase fluorocarbon refrigerant pumps, is more difficult than the reliability of a fluorocarbon compressor. The authors also believe that the use of heat pumps can provide capacity control, reduction in radiator area, reduction in weight, improved turn-down and turn-up ratios, and thermal storage capability without degrading system performance or reliability.

The completion of this preliminary effort has resulted in the conceptual development of several heat-pump-augmented spacecraft heat-rejection configurations. In addition, we have identified a family of working fluids that will improve the performance of these heat-pump systems. Current efforts are directed toward experimental demonstration of these innovative designs, as well as an experimental demonstration of the benefits of the newly identified familiy of working fluids.

This research has also resulted in the conceptual development of an integrated heat-pump thermal management system. Realistic calculations demonstrated the potential for reductions in radiator area and system mass by using a vapor-compression heat pump as the thermal management loop. Use of the heat pump as the thermal management system will result in even further reductions in system mass. Future efforts will be directed toward experimental demonstration of the mass and area savings as well as the reliability benefits. Further work in the identification of new high-temperature refrigerant-absorbent pairs and high-temperature nonazeotropic refrigerant pairs is planned.

Another result of this research has been the development of spacecraft component models for the SimTool transient fluid/thermal system simulator. SimTool was used in the modeling of the micro-gravity single-phase two-phase, and heat-pump thermal management systems.[15,16]

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