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Heat Pump Systems for Enhancement of Heat Rejection from Spacecraft

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ABSTRACT

Temperature boosting of waste heat from spacecraft by means of heat pumps makes it possible, under some conditions, to achieve considerable savings in radiator area and mass. This study considers several possibilities for employing work-actuated and heat-actuated heat pumps (WAHP and HAHP, respectively) for this purpose. In the former case, the spacecraft power source is required to generate extra power to operate the heat pump; in the latter, use is made of the heat rejected from the power source to energize the heat pump. The mass and area savings are calculated for a range of operating parameters, including the temperatures of the waste heat from the power source and payload and that of the effective heat sink. The dimensionless parameters governing the behavior are determined. It is shown that for given operating conditions, a proper choice of the temperature boost and the associated heat pump coefficient of performance leads to an optimum in radiator area and system mass savings. A detailed cycle calculation is presented for an absorption heat pump - a particularly promising HAHP with few moving parts. Design considerations are given for space-based WAHP and HAHP systems.

NOMENCLATURE

- A - Radiator area (m^2)
- COP - Coefficient of performance
- e, e_1 - Efficiency (dimensionless)
- m - Mass (Kg)
- Q - Heat rate (W)
- T - Temperature (K)
- W - Electric Power (W)

Greek Symbols

- β, β_1 - Coefficient of performance (dimensionless)
- ϵ - Emissivity (dimensionless)
- η - Fin efficiency (dimensionless)
- μ - Dimensionless power mass penalty parameter, Eq. (14)
- σ - Stefan-Boltzman coefficient (W/m^2K^4)
- ϕ - Fraction of Carnot COP (dimensionless)

Subscripts

- 0 - Ambient
- 1 - Power source
- 2 - Power radiator
- 3 - Payload
- 4 - Payload radiator

- c - Carnot
- e - Heat engine
- h - Heat pump
- p - Power source
- r - Radiator

1.0 INTRODUCTION

The trend toward increasing spacecraft power has created a growing need for effective methods to reject the waste heat produced by power-generating and power-consuming equipment aboard. In particular, the current SDI and Space Station plans for 10-100 kW of steady electric power represent an order of magnitude increase in waste heat compared to present and past missions. Conventional radiators for such power rates become very large, and revolutionary advances in thermal management may be required [1]. One promising option for enhancing heat rejection is the use of heat pumps to boost the radiator temperature, thereby reducing its area and mass.

Figure 1 describes schematically a typical spacecraft thermal management system [2]. The power source, equipped with a radiator for rejecting its own waste heat at the required temperature, produces electric power W distributed to one or several payloads (e.g., electronic equipment, sensors, etc.). Each of the latter is equipped with a heat acquisition unit (also known as "cold plate") which keeps it at a temperature low enough for its required operation. A heat transport loop, (also known as "thermal bus") transfers the waste heat from the payloads to a common radiator for rejection. The transport loop may employ a forced-circulation fluid or a heat pipe. This method of collective payload heat rejection requires that the transport loop be at a temperature lower than any of the cold plates it serves. The payload radiator must accordingly be kept at as low or lower temperature and therefore requires a large area (and mass) per unit heat rejected, generally more so than the power radiator. Therefore, payload waste heat radiators tend to be the heaviest and bulkiest components in a thermal management system. Since the radiated heat rate is approximately proportional to the fourth power of the absolute temperature, an increase in the radiator temperature would help reduce its size. The key idea in this work is to show that such an increase may be effectively achieved by a heat pump which may be energized by electric power or by waste heat.

One of the first studies on the use of heat pumps in space was performed by Dexter and Haskin [3]. They considered the option of an electrically-driven vapor compression heat pump, using a compressor to transfer a working fluid from an evaporator at the low source temperature to a condenser at a higher (radiator) temperature. They carried out a thorough calculation of the possible mass savings with the heat pump system compared to a pumped fluid loop, for electric power ranging from 0 to 100 kWe. Assumptions were made, based on available experience, of the power mass penalty,

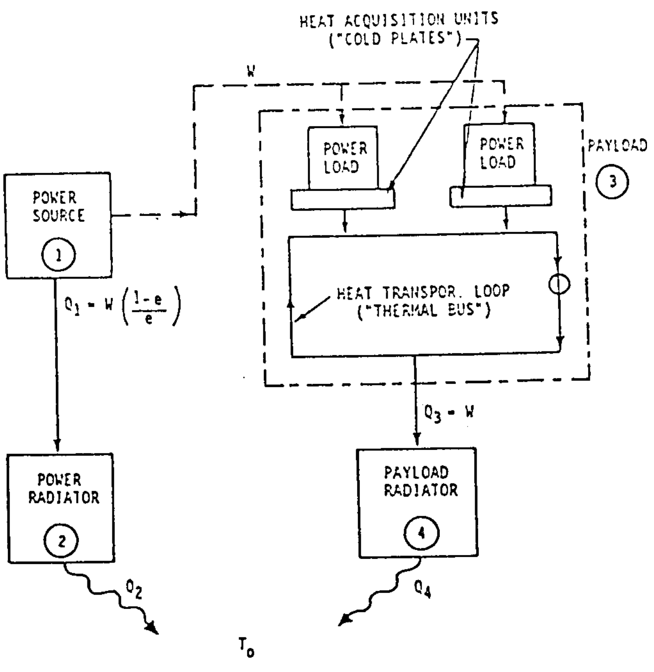

Fig. 1 Schematic description of spacecraft thermal management system

compressor and motor mass, cold plate and fluid loop lines and their variation with the power rating. A sensitivity analysis considered the effects of varying the source and sink temperatures, the coefficient of performance and the extra mass of the power plant. The analysis showed the heat pump system to have lower mass and lower radiator area in all cases, with significant savings under some conditions.

The electrically-driven compressor heat pump is only one of several heat pump options and not necessarily the best one. Its use requires extra electric power from the spacecraft's power source; there are some practical and reliability problems regarding the operation of compressors in space, on which there is little or no experience. A study of wider scope was carried out by Drolen [4], who considered both work-actuated and heat-actuated heat pumps (WAHP and HAHP, respectively), as well as their hybrids. Recognizing that in most cases waste heat is available from both the power source and the payload, with the former at a considerably higher temperature, one can utilize the former as a source of energy for a HAHP to boost the temperature of the latter, thus achieving overall radiator savings. In evaluating different heat pump options, Drolen [4] calculated the available (or affordable) heat pump mass based on radiator area savings and data on specific mass of the radiator and power source. Using this approach, rather than calculating the actual mass savings, made it possible to perform the evaluation in the absence of reliable information on the mass of heat pumps designed for space applications. Drolen [4] carried out an extensive study of the effect of different parameters on the available heat pump mass, including radiator hardening level and geometry, payload rejection temperature, effective space temperature and heat pump efficiency. His optimization was, however, limited. No attempt was made, for example, to find the optimum heat pump COP and temperature boost which would yield maximum radiator savings under given operating conditions. As will be shown later, in the case of a HAHP there are two choice intermediate temperatures (between those of the power and payload waste heats) which affect the overall performance. Also, some of the savings calculated in [4] came from the assumption of different types of radiators for the power and payload, where the heat pump's role amounted basically to the transfer of heat from a heavy to a light radiator. This would result in radiator mass savings even with no heat pumping at all.

Kerrebrock [5] considered the payload radiator area savings achievable by a heat engine-driven heat pump, which is essentially a HAHP. He did calculate an optimum value for radiator temperatures, intermediate between the power and payload rejection temperatures, at which the heat engine and heat pump dispose of their waste heat. Kerrebrock [5] assumed a zero sink temperature which represents radiation into deep space, but is unrealistic for low earth orbits. As a result, he found significant area savings possible only for large temperature ratios of the power to payload waste heats. Merrigan and Reid [6] calculated the mass savings for a similar heat engine/heat pump combination with a non-zero sink temperature, using a common radiator temperature for the heat engine and the heat pump. Neither [5,6] considered the associated savings in power radiator area and mass resulting from the diversion of part of this radiator heat toward the heat engine.

In this study, the potential radiator area and mass savings are calculated using both WAHP and HAHP, without the limitations which had been outlined for the earlier studies. It is shown that reasonably simple, yet descriptive equations may be derived without too many simplifying assumptions. A non-dimensional approach is taken where the characteristic dimensionless parameters governing the behavior are developed. Optimum conditions for operation are calculated.

The main purpose of this study has been to consider some practical approaches to the design of space-based heat pumps. The limitations on reliability of different rotating and reciprocating machinery are well-known. It is shown that an absorption heat pump may be used which combines several favorable features, including high reliability, low mass, few moving parts and good performance.

2.0 AREA AND MASS SAVINGS WITH WORK-ACTUATED HEAT PUMP (WAHP)

The first option we investigate for boosting the temperature of the payload radiator is the use of a work-actuated heat pump of the type most commonly used in terrestrial applications. It employs a vapor compression cycle where the low-temperature heat input causes the evaporation of a working fluid in a low-pressure evaporator; an electrically driven compressor transfers the vapor to a higher-pressure condenser, where it gives up its heat of condensation at a correspondingly higher temperature; the condensate is returned to the evaporator to complete the cycle [7]. The vapor compression cycle had been considered by Dexter and Haskin [3]; we will derive expressions for the area and mass savings and compare them, on the same basis, with a case using a heat-actuated heat pump.

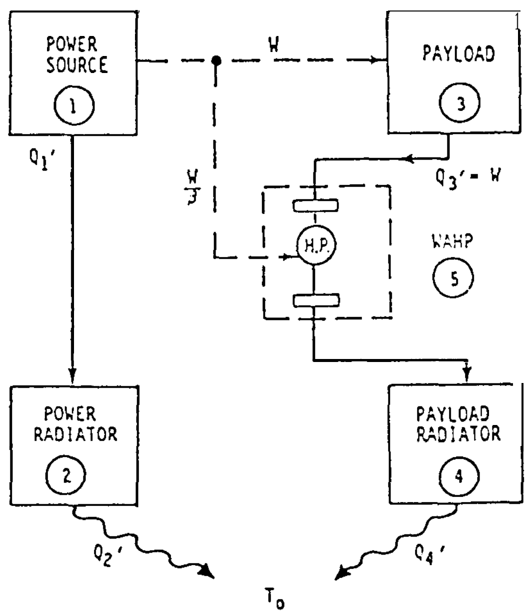

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Fig. 2 Schematic description of work-actuated heat pump (WAHP) incorporated in thermal management system

Figure 2 describes schematically a WAHP incorporated in the thermal management system of the spacecraft. The individual sub-units of the system have been numbered consistently with Figure 1, the base case with no heat pump. Temperatures and heat quantities pertaining to the present case are distinguished from those of the base case by a prime. The power loads, cold plates and heat transport loop shown separately in Figure 1 have been combined into a single box in Figure 2. The payload is assumed to be maintained at the same temperature ($T_3' = T_3$) and to consume the same amount of electric power, W, in both cases. It is further assumed that this entire power is converted into waste heat, so that $Q_3' = Q_3 = W$. the temperature of the power radiator is also the same for both cases ($T_2' = T_2$). Finally, it is assumed that the power and payload radiators are both of the same type, with the maximum efficiency and minimum specific mass allowed by the mission. Assuming otherwise a hardened, one-sided and therefore heavy radiator for one and an unhardened, two sided and light radiator for the other, as in [4], would result *in* the obvious conclusion that considerable mass saving may be achieved by transferring all the heat from the former to the latter.

2.1 Area Saving

The amounts of heat rejected by the power and payload radiators, respectively, for the case with no heat pump (Figure 1) are:

$$Q_2 = \epsilon\eta\sigma A_2\left(T_2^4 - T_o^4\right) \tag{1a}$$

$$Q_4 = \epsilon\eta\sigma A_4\left(T_4^4 - T_o^4\right) \tag{1b}$$

where A2 and A4 are the areas of the two radiators; ϵ - the emissivity of the radiator surface; η - the radiator efficiency (analogous to fin efficiency; expressing the ratio of energy radiated to that if the surface was isothermal); and σ - the Stefan-Boltzman coefficient. The total radiator area is, therefore:

$$A = \frac{1}{\epsilon\eta\sigma}\left[\frac{Q_2}{\left(T_2^4 - T_o^4\right)} + \frac{Q_4}{\left(T_4^4 - T_o^4\right)}\right] \tag{2}$$

Similarly, for the case with WAHP (Figure 2):

$$A' = \frac{1}{\epsilon\eta\sigma}\left[\frac{Q_2'}{\left(T_2^4 - T_o^4\right)} + \frac{Q_4'}{\left(T_4'^4 - T_o^4\right)}\right] \tag{3}$$

The area saved, as a fraction of the original area, may hence be expressed by:

$$\frac{A - A'}{A} = \frac{\dfrac{Q_4}{\left(T_4^4 - T_o^4\right)} - \dfrac{Q_4'}{\left(T_4'^4 - T_o^4\right)} - \dfrac{\left(Q_2' - Q_2\right)}{\left(T_2^4 - T_o^4\right)}}{\dfrac{Q_2}{\left(T_2^4 - T_o^4\right)} + \dfrac{Q_4}{\left(T_4^4 - T_o^4\right)}} \tag{4}$$

For the case with no heat pump, the payload heat equals that of its radiator, $Q_4 = Q_3 = W$. With the heat pump added, the payload radiator must reject extra heat due to the electric power input to the heat pump. The coefficient of performance (COP) of the latter is defined as the ratio of the heat input to the work input. Denoting the coefficient of performance by β, it is easy to show that for a heat input $Q'_3 = W$ the required work input is W/β. Thus, the heat rejected by the payload radiator (Q_4') as well as the total power required from the power source are both equal to $W(1 + \beta)/\beta$. The power source efficiency is defined as the ratio of the electric power output to the input, and the heat rejected from it is the difference between the two. Denoting the power source efficiency by e, we find the reject heat in the case with no heat pump to be

$Q_2 = W(1-e)/e$ and with the heat pump added $Q_2' = W(1+\beta)(1-e)/(e\beta)$. Substitution of all the above in (4) yields:

$$\left(\frac{A - A'}{A}\right) = \frac{1 - \left(\frac{1-e}{e\beta}\right)\left(\frac{T_3^4 - T_o^4}{T_2^4 - T_o^4}\right) - \left(\frac{1+\beta}{\beta}\right)\left(\frac{T_3^4 - T_o^4}{T_4'^4 - T_o^4}\right)}{1 + \left(\frac{1-e}{e}\right)\left(\frac{T_3^4 - T_o^4}{T_2^4 - T_o^4}\right)} \qquad (5)$$

The coefficient of performance of the heat pump β is a function of its operating temperatures at the low and high sides, T_3 and T_4', respectively. The exact temperature relation depends on the working fluid, the quality of the compression and the condensate expansion from the condenser to the evaporator. The more reversible these processes, the larger β for given temperatures; for a perfectly reversible Carnot heat pump, the relation is

$$\beta_c = \frac{T_3}{T_4' - T_3} \qquad (6)$$

In selecting a heat pump for given operating conditions, there is one degree of freedom in choosing the temperature boost. A higher T_4' would tend to reduce the radiator size but would also lower the heat pump COP. An optimum therefore exists which may be found by setting to zero the derivative of $(A - A')/A$ with respect to T_4'. Hence, from (5):

$$\left[1 + \left(\frac{1-e}{e}\right)\left(\frac{T_4'^4 - T_o^4}{T_2^4 - T_o^4}\right)\right]\frac{d\beta}{dT_4'} = -\frac{4\beta(1+\beta)T_4'^3}{\left(T_4'^4 - T_o^4\right)} \qquad (7)$$

from which the optimum values of β and T_4' may be found, given their functional relation. Some of the earlier studies [4,5] have suggested taking a fixed portion of the Carnot COP for a non-ideal, practical heat pump. Others [3] have proposed a formula describing the fraction of Carnot efficiency as a weak function of the operating temperatures. For any given relation, the optimization could be carried out using Eq. (7). Taking the former approach and assuming $\beta = \phi\beta_c$, Equation (7) becomes:

$$\left[1 + \left(\frac{1-e}{e}\right)\left(\frac{T_4'^4 - T_o^4}{T_2^4 - T_o^4}\right)\right] = \frac{4T_4'^3\left[T_4' - (1-\phi)T_3\right]}{\left(T_4'^4 - T_o^4\right)} \qquad (8)$$

In particular, for a Carnot heat pump ($\phi = 1$):

$$\left(\frac{T_4'^4 - T_o^4}{T_2^4 - T_o^4}\right) = \frac{3}{2}\left(\frac{e}{1-e}\right)\left[1 + \sqrt{1 + \frac{16}{9}\left(\frac{1-e}{e}\right)\frac{T_o^4}{\left(T_2^4 - T_o^4\right)}}\right] \qquad (8a)$$

Figure 3a describes the optimal area saving as a function of the space sink to payload temperature ratio, for different values of the power to payload waste heat temperatures. A power source efficiency of $e = 0.3$ was assumed in all cases. Two cases have been considered: A Carnot heat pump ($\phi = 1$) represented by the solid lines, and a non-ideal one with $\phi = 0.75$ represented by the broken lines. The optimum values of T_4' and β do not vary much with T_o/T_3 for the cases calculated. It is evident that the area saving increases with T_2/T_3 and also with T_o/T_3. A singular point exists at $T_o/T_3 = 1.0$ showing 100% savings for all T_2/T_3. At this point,

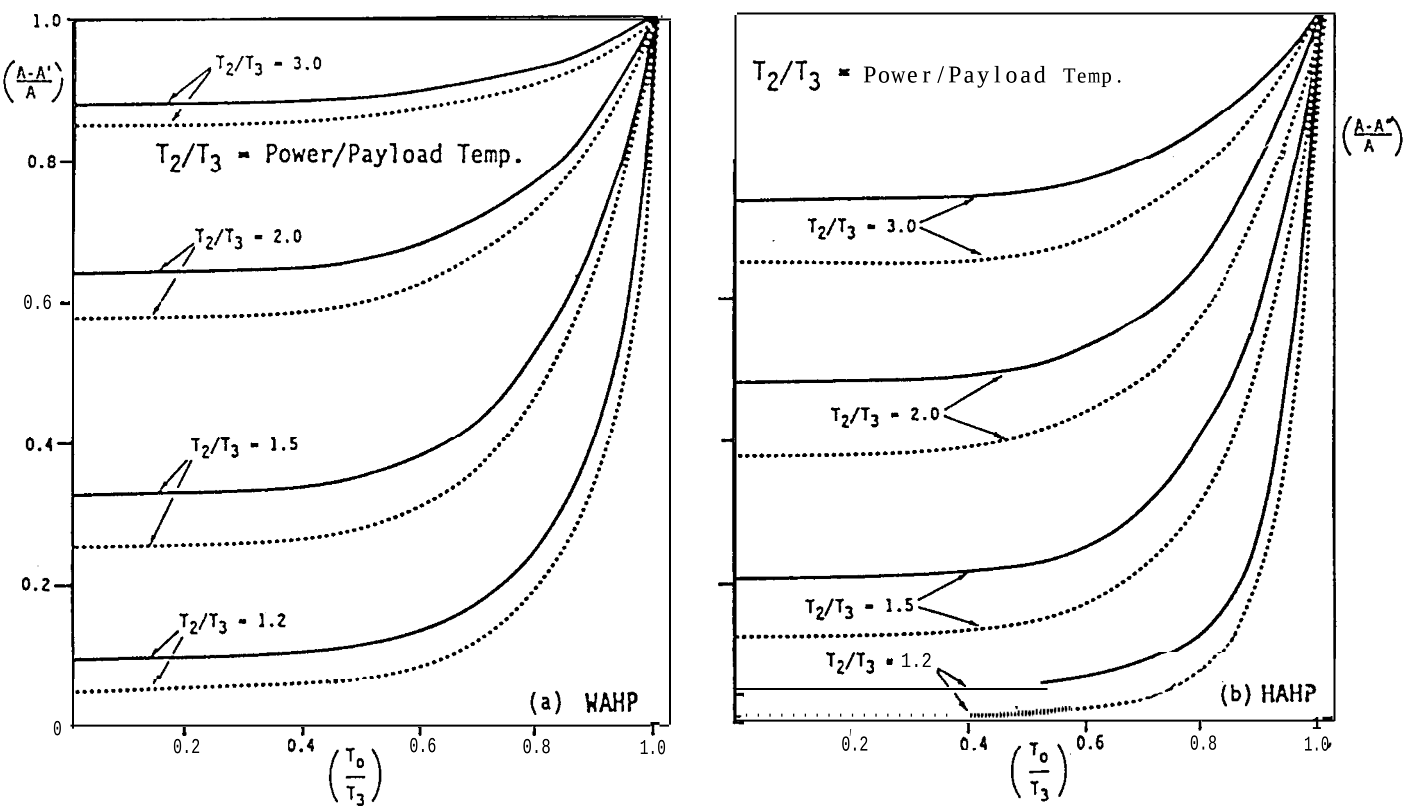

Fig. 3 Fraction of radiator area saved under optimal conditions using heat pumps as a function of operating temperature ratios. $e = 0.3$ in all cases. (a) WAHP, (b) HAHP. Solid lines represent ideal (Carnot) heat pumps; broken lines represent non-ideal case with $\phi = 0.75$ for WAHP and $\phi_e = \phi_\beta = (0.75)^{1/2}$ for HAHP.

with the payload temperature equal to that of the space sink, the only possible mode of operation is with a heat pump. The decrease in area saving with the non-ideal heat pump compared to the ideal one is more pronounced at low values of T_0/T_3 and T_2/T_3. Best results are achieved, in all cases, for applications where the payload temperature T3 is only slightly above that of the available sink, as may be expected.

The heat pump must operate above a certain COP in order to have an effect in area savings. If the value of the boosted temperature T'_4 resulting from Equation (7) or (8) is no greater than the payload temperature T_3, then there is no advantage in using a heat pump. For a given set of operating conditions, the critical value of ϕ below which a WAHP does not contribute to area saving may be found by setting $T_4' = T_3$ in Equation (8):

$$\phi_{crit} = \left(\frac{T_3^4 - T_o^4}{4T_3^4}\right)\left[1 + \left(\frac{1-e}{e}\right)\left(\frac{T_3^4 - T_o^4}{T_2^4 - T_o^4}\right)\right] \quad (9)$$

2.2 Mass Saving

Using a WAHP imposes a power mass penalty due to the extra electric power required to energize the heat pump. Let m_r be the mass of the radiator per unit area and m_p the mass of the power source per unit power produced. The combined mass of the power plant and both radiators (power and payload) is therefore:

$$\text{without heat pump:} \quad m = m_r A + m_p W \quad (10a)$$

$$\text{with heat pump:} \quad m' = m_r A' + m_p W\left(\frac{1+\beta}{\beta}\right) \quad (10b)$$

and therefore, the mass saving due to the WAHP:

$$\Delta m = m_r (A - A') - m_p W/\beta \quad (11)$$

The mass saved per unit power produced, W, (which is also the power rejected from the payload), may be expressed by:

$$\frac{\Delta m}{W} = \frac{m_r A}{W}\left(\frac{A - A'}{A}\right) - \frac{m_p}{\beta}$$

Substituting from (2) for the radiator area:

$$\frac{\Delta m}{W} = \frac{m_r}{\epsilon\eta\sigma}\left(\frac{A - A'}{A}\right)\frac{1}{W}\left[\frac{Q_2}{\left(T_2^4 - T_o^4\right)} + \frac{Q_4}{\left(T_4^4 - T_o^4\right)}\right] - \frac{m_p}{\beta}$$

and substituting the values of Q_2 and Q_4, we finally obtain

$$\frac{\Delta m}{W} = \frac{m_r}{\epsilon\eta\sigma T_3^4}\left[\left(\frac{1-e}{e}\right)\frac{T_3^4}{\left(T_2^4 - T_o^4\right)} + \frac{T_3^4}{\left(T_3^4 - T_o^4\right)}\right]\left(\frac{A - A'}{A}\right)$$

$$- \frac{m_p}{\beta} \quad (12)$$

$(\Delta m/W)$ expresses the available (or affordable) specific mass for the heat pump, following the approach proposed by Drolen [4]. Again, an optimum saving may be found by proper choice of the heat pump COP and temperature boost. This optimum would generally differ from the one for the area saving, and would depend on the relative values of m_p and $(m_r/\epsilon\eta\sigma T_3^4)$, where the latter expression is the mass of the radiator per unit power radiated at the payload temperature T_3. Typical values for m_r, ϵ and η are 5.0 kg/m^2, 0.85 and 0.9, respectively [8]. Typical values of m_p are given in Table 1 for four space power sources under consideration [4]. $(A - A')/A$ and β in Equation (10) are both functions of the choice heat pump outlet temperature T_4'. Setting to zero the derivative of $(\Delta m/W)$ with respect to T_4' will yield the optimum value of T_4':

$$\left[1 + \left(\frac{1-e}{e}\right)\left(\frac{T_4'^4 - T_o^4}{T_2^4 - T_o^4}\right) + \mu\left(\frac{T_4^4 - T_o^4}{T_3^4}\right)\right]\frac{\partial\beta}{\partial T_4'} =$$

$$- \frac{4\beta(1+\beta)T_4'^3}{\left(T_4'^4 - T_o^4\right)} \quad (13)$$

TABLE 1. SPACE POWER SOURCES

	Power output (kW$_e$)	Heat Rejection Temp (°K)	Efficiency e (%)	Specific Mass (kg/KWe)
Nuclear Reactor/ Thermionic Converter (SP-100)	100	994	7	27.0
Solar Dynamic/ Stirling cycle	100	533	33	76.9
Photovoltaic Solar Array	100	350	18	30.3
Dynamic Isotope (DIPS)	10	366	26	77.0

where μ is a dimensionless power mass penalty parameter:

$$\mu = \left(\frac{m_p}{m_r/\epsilon\eta\sigma T_3^4}\right) \quad (14)$$

For a heat pump operating at a fixed fraction of the Carnot COP, Equation (11) becomes:

$$\left[1 + \left(\frac{1-e}{e}\right)\left(\frac{T_4'^4 - T_o^4}{T_2^4 - T_o^4}\right) + \mu\left(\frac{T_4'^4 - T_o^4}{T_3^4}\right)\right] =$$

$$\frac{4T_4'^3\left[T_4' - (1-\phi)\,T_3\right]}{\left(T_4'^4 - T_o^4\right)} \quad (15)$$

In particular, for a Carnot heat pump, $(\phi = 1)$:

$$\left(\frac{T_4'^4 - T_o^4}{T_2^4 - T_o^4}\right) =$$

$$\frac{3\left[1 + \sqrt{1 + \frac{16}{9}\left[\left(\frac{1-e}{e}\right)\frac{T_o^4}{\left(T_2^4 - T_o^4\right)} + \mu\frac{T_o^4}{T_3^4}\right]}\right]}{2\left[\left(\frac{1-e}{e}\right) + \mu\left(\frac{T_2^4 - T_o^4}{T_3^4}\right)\right]} \quad (15a)$$

The critical value of ϕ under which there is no contribution to mass saving from the use of a WAHP is found by setting $T_4' = T_3$ in Equation (15):

$$\phi_{crit} = \left(\frac{T_3^4 - T_o^4}{4T_3^4}\right)\left[1 + \left(\frac{1-e}{e}\right)\left(\frac{T_3^4 - T_o^4}{T_2^4 - T_o^4}\right) + \mu\left(\frac{T_3^4 - T_o^4}{T_3^4}\right)\right] \quad (16)$$

Figure 4a describes the available specific heat pump mass under optimal conditions, normalized with respect to radiator specific mass, as a function of the operating temperatures. As in the case of the area savings, the governing temperature ratios are (T_0/T_3) and (T_2/T_3). A power source efficiency e = 0.3 has been assumed for all cases as well as a typical value for the power mass penalty parameter, $\mu = 2.0$. Solid lines represent an ideal, Carnot heat pump and broken lines are for a non-ideal one with $\phi = 0.75$. The optimum values of T4 and β are generally different from those for maximum area savings. It is evident from Figure 4a that the available heat pump mass increases with T_2/T_3 and also with T_0/T_3, reaching very high values where T_0/T_3 is close to unity. The decrease in available mass with the non-ideal heat pump compared to the ideal one is very significant for low values of T_0/T_3, but becomes small with T_0/T_3 close to unity.

3.0 AREA AND MASS SAVINGS WITH HEAT-ACTUATED HEAT PUMP (HAHP)

The heat-actuated heat pump is an important alternative to the WAHP discussed in the previous section. Figure 5 describes schematically a HAHP incorporated in the thermal management system of a spacecraft. The same sub-unit numbering system has been used consistently with Figure 1, the base case with no heat pump, and with Figure 2, the WAHP case. Heat quantities and temperatures pertaining to this case have been designated by a double prime. Again, the payload is maintained at the base case temperature ($T_3'' = T_3$) and consumes the base amount of electric power, W. All the power is converted into waste heat ($Q_3'' = Q_3 = W$) and the power radiator temperature remains unchanged ($T_2'' = T_2$).

The HAHP is described conceptually as a work-actuated heat pump driven by a heat engine, which uses part of the power source waste heat for its operation. We say 'conceptually' since there are several options for building a HAHP, including some in which no mechanical energy is being transmitted from a "driving" to a "driven" part of the system. One such option is based on the absorption cycle, which will be discussed in detail later. The feature of no moving parts (except for small, auxiliary equipment) makes the absorption heat pump highly

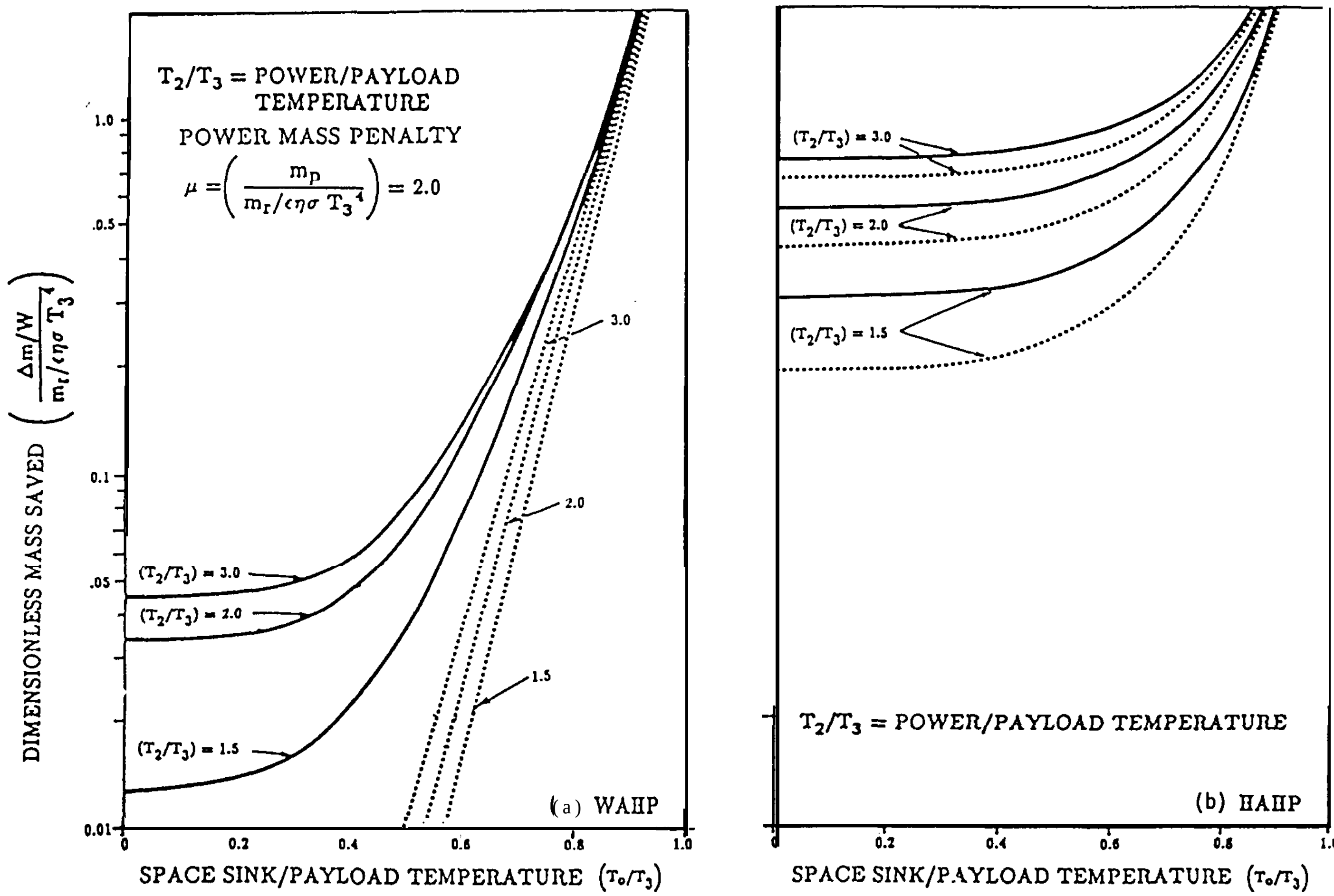

Fig. 4 Available specific mass for WAHP under optimal conditions, as a function of operating temperature ratios. e = 0.3 in all cases. (a) WAHP with $\mu = 2.0$, (b) HAHP. Solid lines represent ideal (Carnot) heat pumps; broken lines represent non-ideal case with $\phi = 0.75$ for WAHP and $\phi_e = \phi_R = (0.75)^{1/2}$ for HAHP.

reliable and therefore particularly attractive for space applications. Other options for a HAHP include a variety of heat-engines driving the compressor of a vapor-compression heat pump, a heat-driven ejector-type compression system, thermoelectric devices and more. The fundamental thermodynamic principle as illustrated in Figure 5 is, however, the same for all of the above.

The HAHP has two radiators for heat rejection, which may operate at two different temperatures, although in many cases they are united and made to work at the same temperature. We designate the temperature of the radiator serving the heat engine portion of the HAHP by T_{4e}'' and that *serving* the heat pump portion by T_{4h}'', as shown in Figure 5.

3.1 Area Saving

The area requirement for the base case with no heat pump was given by Eq. (2). Similarly, for the case with HAHP:

$$A'' = \frac{1}{\epsilon\eta\sigma}\left[\frac{Q_2''}{\left(T_2^4 - T_o^4\right)} + \frac{Q_{4e}''}{\left(T_{4e}''^4 - T_o^4\right)} + \frac{Q_{4h}''}{\left(T_{4h}''^4 - T_o^4\right)}\right] \quad (17)$$

The area saved, as a fraction of the original area may hence be expressed by:

$$\frac{A - A''}{A} = \frac{\dfrac{Q_4}{\left(T_4^4 - T_o^4\right)} - \dfrac{Q_{4e}''}{\left(T_{4e}''^4 - T_o^4\right)} - \dfrac{Q_{4h}''}{\left(T_{4h}''^4 - T_o^4\right)} + \dfrac{\left(Q_2 - Q_2''\right)}{\left(T_2^4 - T_o^4\right)}}{\dfrac{Q_2}{\left(T_2^4 - T_o^4\right)} + \dfrac{Q_4}{\left(T_4^4 - T_o^4\right)}} \quad (18)$$

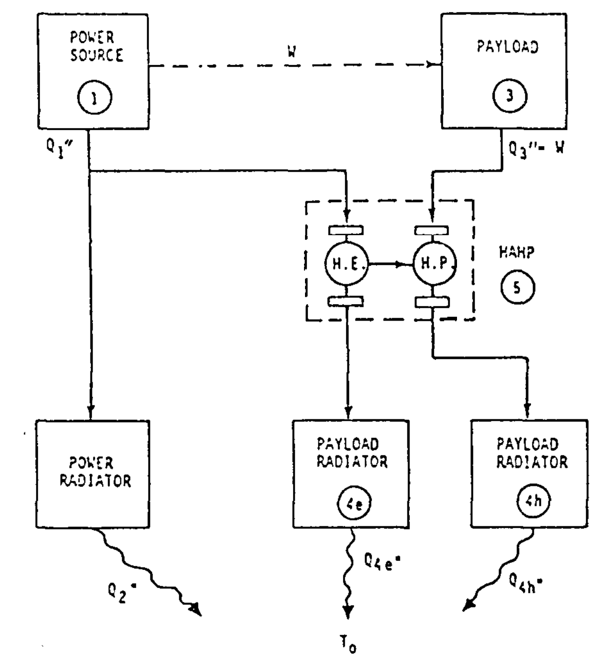

Fig. 5 Schematic description of heat-actuated heat pump (HAHP) incorporated in thermal management system.

The heat quantities in the base case were shown before to be $Q_4 = W$ and $Q_2 = W(1 - e)/e$. The quantity $(Q_2 - Q_2'')$ in the last term of the numerator of Equation (18) is the amount of power source waste heat diverted from the power radiator toward the HAHP. If we denote the COP of the heat pump portion of the HAHP by β_1, it is easy to show that the amount of work transferred internally in the latter from the heat engine to the heat pump is W/β_1. Further, if the efficiency of the heat engine portion of the HAHP is e_1, then $W/\beta_1 = (Q_2 - Q_2'')e_1$. Thus, it is possible to calculate $(Q_2 - Q_2'')$ as $W/\beta_1 e_1$, and to find $Q_{4e}'' = W(1 - e_1)/\beta_1 e_1$ and $Q_{4h}'' = W(1 + \beta_1)/\beta_1$. Substitution in (18) yields:

$$\frac{A - A''}{A} = \frac{1 + \frac{1}{\beta_1 e_1}\left(\frac{T_3^4 - T_o^4}{T_2^4 - T_o^4}\right) - \left(\frac{1 - e_1}{\beta_1 e_1}\right)\left(\frac{T_3^4 - T_o^4}{T_{4e}''^4 - T_o^4}\right) - \left(\frac{1 + \beta_1}{\beta_1}\right)\left(\frac{T_3^4 - T_o^4}{T_{4h}''^4 - T_o^4}\right)}{1 + \left(\frac{1 - e}{e}\right)\left(\frac{T_3^4 - T_o^4}{T_2^4 - T_o^4}\right)} \quad (19)$$

Both of the HAHP performance parameters, e_1 and β_1, are functions of its four operating temperatures. The exact temperature relation depends on the type of cycle used, the working fluids and design details. For reversible (Carnot) systems, e_1 and β_1 are given by

$$e_{1c} = 1 - \frac{T_{4e}''}{T_2}\,; \qquad \beta_{1c} = \frac{T_3}{T_{4h}'' - T_3} \quad (20)$$

For non-ideal systems, fractions of these values may be taken, which are themselves generally temperature-dependent.

As in the case of a WAHP, there is some degree of freedom in selecting a HAHP for given operating conditions. Here, both rejection temperatures T_{4e}'' and T_{4h}'' may be selected to provide maximum area saving. To do this, the functional relation of e_1 and β_1 to the operating temperatures must be known. then. the partial derivatives of $(A - A'')/A$ with respect to T_{4h}'' and T_{4e}'' are set to zero. Hence, from (19):

$$\left[1 - \left(\frac{T_{4e}''^4 - T_o^4}{T_2^4 - T_o^4}\right)\right]\frac{\partial e_1}{\partial T_{4e}''} = -\frac{4\, e_1\left(1 - e_1\right) T_{4e}''^3}{\left(T_{4e}''^4 - T_o^4\right)} \quad (21)$$

$$\left[1 + \left(\frac{1 - e_1}{e_1}\right)\left(\frac{T_{4h}''^4 - T_o^4}{T_{4e}''^4 - T_o^4}\right) - \frac{1}{e_1}\left(\frac{T_{4h}''^4 - T_o^4}{T_2^4 - T_o^4}\right)\right]\frac{\partial \beta_1}{\partial T_{4h}''} = -\frac{4\,\beta_1\left(1 + \beta_1\right) T_{4h}''^3}{\left(T_{4h}''^4 - T_o^4\right)} \quad (22)$$

Equations (21), (22) may be solved for the optimum values of T_{4e}'' and T_{4h}''. For certain types of HAHP, particularly those with Rankine engine-driven vapor compression, it is reasonable to take e_1 and β_1 as fixed fractions, ϕ_e and ϕ_β, respectively, of their corresponding Carnot values. then, from Eq. (20):

$$e_1 = \phi_e \frac{\left(T_2 - T_{4e}''\right)}{T_2} \qquad \beta_1 = \frac{\phi_\beta T_3}{T_{4h}'' - T_3} \tag{23}$$

Substitution in (21) yields:

$$\left[1 - \left(\frac{T_{4e}''^4 - T_o^4}{T_2^4 - T_o^4}\right)\right] = \frac{4T_{4e}''^3 \left(T_2 - T_{4e}''\right)\left[T_2 - \phi_e \left(T_2 - T_{4e}''\right)\right]}{T_2\left(T_{4e}''^4 - T_o^4\right)} \tag{24}$$

The use of a HAHP at the optimum operating point is subject to a limitation on the availability of waste heat from the power source. It is easy to see that a solution of (24), for any ϕ_e, is $T_{4e}'' = T_2$. The optimization drives the heat engine radiator to operate at the highest possible temperature, equal to that of its source; this makes the heat engine very inefficient and requires very large input heat in order to generate the motive power for the heat pump. However, the heat available for this purpose from the power source is limited to $W(1 - e)/e$, which can generate an amount of work $e_1 W(1 - e)/e$. Matching this quantity with the heat pump requirement (W/β_1) yields the maximum heat engine sink temperature:

$$T_{4e}'' = T_2\left[1 - \left(\frac{e}{1 - e}\right)\left(\frac{T_{4h}'' - T_3}{\phi_e \phi_\beta T_3}\right)\right] \tag{25}$$

at this limit, the power radiator is eliminated completely and the entire amount of power source waste heat is used for actuation of the HAHP. Substituting T_{4e}'' from (25) in (19), it is now possible to optimize with respect to T_{4h}'', and find:

$$\frac{3\,T_{4h}''^4 + T_o^4 - 4\,T_{4h}''^3\, T_3\left(1 - \phi_\beta\right)}{3\,T_{4e}''^4 + T_o^4 + 4\,T_{4e}''^3\, T_2\left(1 - \phi_e\right)/\phi_e} = \left(\frac{T_{4h}''^4 - T_o^4}{T_{4e}''^4 - T_o^4}\right)^2 \tag{26}$$

In particular, for a Carnot heat pump $\phi_e = \phi_\beta = 1$ and hence, from (25) and (26):

$$T_{4e}'' = T_{4h}'' = \left[\frac{1 - e}{T_2} + \frac{e}{T_3}\right]^{-1} \tag{27}$$

Finally, a critical value of ϕ_e, ϕ_β below which the HAHP becomes ineffective may be found in a way similar to that used for the WAHP, by setting $T_{4e}'' = T_2$, $T_{4h}'' = T_3$ in (26). To do this, a relation must be assumed between ϕ_e and ϕ_β, or one of them set and the other found. For $\phi_e = \phi_\beta$ we find:

$$\phi_{crit} = \frac{\left(T_3^4 - T_o^4\right)\left(T_2^4 - T_3^4\right)}{8\left(T_2^4 - T_o^4\right)T_3^4}\left[1 + \sqrt{1 + \frac{64\,T_2^4\,T_3^4}{\left(T_2^4 - T_3^4\right)^2}}\right] \tag{28}$$

Figure 3b describes the area savings under optimum conditions as a function of the dimensionless space sink to payload temperature ratio, for different values of the power to payload waste heat temperatures. A power source efficiency e = 0.3 was assumed in all cases, as in the analysis of the WAHP. Again, two cases were considered: A Carnot heat pump (ϕ = 1) represented by the solid lines, and a non-ideal one with $\phi_e = \phi_\beta = (0.75)^{1/2}$ represented by the broken lines. The fractions of Carnot in the latter case were selected for equal-basis comparison with the ϕ = 0.75 WAHP. The optimum values of T_{4e}'', T_{4h}'', e_1 and β_1 do not vary much with T_0/T_3 for the cases calculated. The behavior is very similar to that with the WAHP (Figure 3a). It is evident that the area saving increases with T_2/T_3 and also with T_0/T_3. A singular point exists at $T_0/T_3 = 1.0$ showing 100% savings for all T_2/T_3. At this point, with the payload temperature equal to that of the space sink, the only possible mode of operation is with a heat pump. Maximum area saving is achieved, in all cases, where the payload temperature T_3 is only slightly above that of the available sink, as may be expected. A comparison of Figures 3a and 3b shows that for the same operating temperatures, the savings provided by an optimal WAHP is greater by a few percent than with an optimal HAHP.

3.2 Mass Saving

Unlike the case of the WAHP, there is no power mass penalty associated with the use of a HAHP; no extra electric power is required since the heat pump makes use of the available waste heat. The system mass saving in the case of a HAHP for the system excluding the HAHP is therefore directly proportional to the area saving and given by:

$$\Delta m = m_r \ (A - A'')$$

which can be rewritten as:

$$\frac{\Delta m}{W} = \frac{m_r}{\epsilon\eta\sigma\, T_3^4}\left[\left(\frac{1 - e}{e}\right)\frac{T_3^4}{\left(T_2^4 - T_o^4\right)} + \frac{T_3^4}{\left(T_3^4 - T_o^4\right)}\right]\left(\frac{A - A''}{A}\right) \tag{29}$$

$(\Delta m/W)$ expresses the available specific mass for the heat pump. The optimum in this case coincides with the optimum for area savings, because of the proportionality relation between the two.

Figure 4b describes the available specific heat pump mass under optimal conditions, normalized with respect to radiator specific mass, as a function of the operating temperatures. A power source efficiency e = 0.3 has been assumed for all cases. Solid lines represent an ideal, Carnot heat pump and broken lines are for a non-ideal one with $\phi_e = \phi_\beta = (0.75)^{1/2}$. The behavior is qualitatively similar to that for the WAHP, described in Figure 4a. However, for the same operating temperatures, the available mass for the HAHP is significantly greater than the one for the WAHP. This is due primarily to the power mass penalty associated with the latter - the need to generate extra electric power for its operation. The difference between the HAHP and WAHP is particularly large for the non-ideal heat pumps - greater than an order of magnitude for low sink temperatures. This constitutes a definite advantage of the HAHP over the WAHP.

4.0 THE ABSORPTION HEAT PUMP (AHP)

A particular type of HAHP holding considerable promise for space applications is the absorption heat pump. Its major advantage is in the almost total absence of moving parts, which provides for high reliability. Unlike the heat engine-driven compression cycle discussed in the previous section and by [5,6], there is no mechanical energy involved. The cycle employs two working fluids -- a refrigerant and an absorbent - instead of the single fluid used in the vapor compression cycle. The temperature lift occurring when one substance is absorbed in the other provides the heat pumping action without compression.

Absorption systems may be built with several working fluid combinations and in a variety of cycle configurations; they may be staged to provide higher COP or higher temperature lifts [9]. A complete discussion of the different options is outside the scope of this article. A description of the basic, single-stage cycle may be found in [10]. Unlike the heat engine-driven vapor compression system, the absorption system may not be assumed to operate at a fixed portion of the Carnot efficiency for the same operating temperatures; it has a different operating curve, approaching Carnot for some temperatures and quite different from it for others [11]. To calculate its performance, a complete cycle analysis is required for the given working fluids and operating conditions.

In order to evaluate the capability of the AHP, a specific application was selected in the present study - the rejection of waste heat from steady-state, 100 kWe electronic equipment aboard a spacecraft in low earth orbit (LEO), to be kept at 300°K. A conceptual design of an absorption heat pump for this purpose was carried out and the resulting radiator area and mass savings calculated. The system prior to adding the heat pump is as described schematically by Figure 1, with Q_3 = W = 100 kW, $T_3 = T_4 = 300°K$, T_o = 250°K. The power source assumed for this application was the Solar Dynamic/Stirling (Table 1) with e = 0.33, T_2 = 533°K. Two-sided, unhardened radiators are considered with $m_r = 5.0$ kg/m², $\epsilon = 0.8$, $\eta = 0.9$. Substituting these values in Equation (2), the total radiator area required is 649 m² with a total mass of 3245 Kg, for the total amount of heat rejected ($Q_2 + Q_4$) = 303 kW.

The system with the AHP added is described in Figure 6. The AHP has six main components - absorber, condenser, evaporator, generator, recuperator and precooler - with the first two

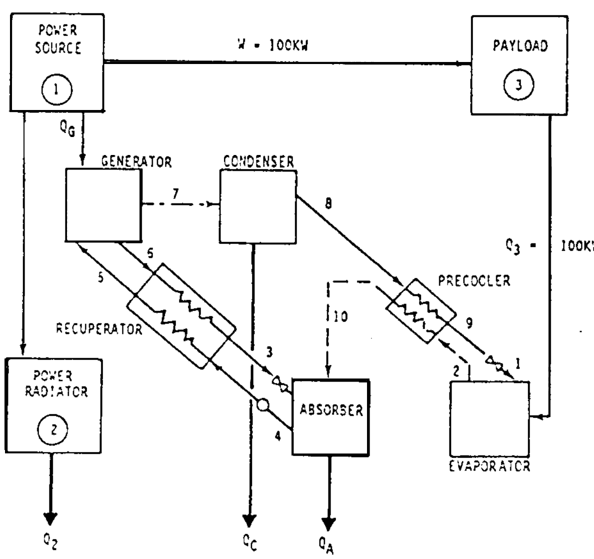

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Fig. 6 Schematic description of absorption heat pump (AHP) incorporated in thermal management system.

serving for heat rejection. An absorbent/refrigerant combination commonly used for absorption cooling in terrestrial applications - lithium bromide/water - has been selected. Other working fluids which may be more suitable for space application are discussed later.

Liquid refrigerant (water) enters the evaporator at state 1 and evaporates at 300°K, thereby removing heat from the payload. The vapor at state 2 passes through through the precooler and enters the absorber where it is absorbed by a highly hygroscopic lithium bromide/water solution, entering concentrated at state 3, and leaving more dilute at state 4. The solution must be cooled during the process to reject the heat of absorption, in order to keep a low vapor pressure and high hygroscopic capability. The rejection temperature is higher than that of the evaporator by the temperature lift of the working fluid pair. The dilute solution at state 4 passes through the recuperative heat exchanger and enters at state 5 into the generator, where waste heat from the power source at 533°K is applied to it. This heat causes the desorption of water from the solution, reconcentrating it to state 6. The reconcentrated solution returns to the absorber via the recuperator. The desorbed water vapor at state 7 condenses to state 8 in the condenser, where the heat of condensation is rejected to space. The condensate exchanges heat with the vapor at state 1 before returning through the expansion valve to the evaporator. A small pumping device is required to transfer the dilute solution from the low pressure absorber to the higher pressure generator.

Cycle calculations were performed to define the operating temperatures and concentrations at the different state points, and the heat quantities in the sub-units. A modular computer program was employed, capable of simulating absorption systems in different cycle configurations and with different working fluids [12]. The rejection temperatures in the absorber and condenser may be optimized, as explained in Section 3. In the present case study with the $LiBr/H_2O$ working fluid, crystallization sets a limit on the high solution concentration which was selected to be 67%. The results of the cycle calculations are given in Table 2. The total radiator area, composed of the absorber, the condenser and the reduced power radiator is found to be:

$$A'' = \frac{1}{\epsilon\eta\sigma}\left[\frac{Q_2}{\left(T_2^{\,4} - T_o^{\,4}\right)} + \frac{Q_c}{\left(T_c^{\,4} - T_o^{\,4}\right)} + \frac{Q_A}{\left(T_A^{\,4} - T_o^{\,4}\right)}\right] \tag{30}$$

$$= 416 \text{ m2}$$

where Q_c and QA are the heat quantities in the condenser and absorber, respectively, and T_c and TA are their temperatures. Note that in the absorber the solution temperature varies, and an average value is taken. Thus, 233 m² of radiator area and a corresponding mass of 1165 kg have been saved. The available mass for the generator, evaporator, recuperator and the auxiliary parts of the AHP is 11.65 kg/kW.

The above example demonstrates the possibility for reducing radiator area and mass. using an absorption heat pump. The values calculated for the savings provide only an indication of what may be achieved in practice. Considerably greater potential exists with higher power radiator temperatures, lower payload temperatures and higher sink temperatures.

TABLE 2. AHP OPERATING CONDITIONS

State Point	Temp. (°C)	LiBr Concentration (Weight %)	Pressure (kPa)	Vapor Fraction	Mass Flow Rate (kg/sec)	Enthalpy (kJ/kg)
1	27.0	0	3.57	0.138	0.0476	446.7
2	27.0	0	3.57	1.000	0.0476	2550.8
3	79.4	67.0	132	0.000	0.5902	227.7
4	i4.4	62.0	3.57	0.000	0.6378	191.5
5	229.8	62.0	732	0.000	0.6378	463.2
6	260.0	67.0	732	0.000	0.5902	542.9
7	260.0	0	732	1.000	0.0476	2973.5
8	166.6	0	732	0.000	0.0476	704.4
9	105.4	0	732	0.000	0.0476	448.7
10	161.6	0	3.57	1.000	0.0416	2806.5

Sub Unit	Evaporator	Absorber	Condenser	Generator	Recuperator	Pre-cooler	Pump
Heat Rate (kW)	100.0	145.8	108.0	153.8	186.0	12.2	0.29

Working fluids other than $LiBr/H_2O$ may yield better results; as mentioned earlier, $LiBr/H_2O$ is limited in its temperature lift due to crystallization at the low temperatures. For given temperatures, operating conditions of the AHP such as the solution circulation rate may be optimized. A system design suitable for space applications must be developed, which would differ in some respects from the common AHP technology. Some practical considerations are discussed next.

5.0 DESIGN CONSIDERATIONS

A great deal of experience is available on the design and operation of heat pumps, both WAHP and HAHP, for terrestrial applications. An extensive literature search has revealed no information on specific designs for space applications. The primary concern for earth systems is usually cost and often physical dimensions, where the heat pump has to be shipped a certain way or fit into a designated space. The two predominant considerations guiding the design of space-based heat pumps are mass and reliability. Volume, once deployed, is of no major concern. Thus, these particular issues must be addressed.

The main problem in the design of a space-based WAHP is the compressor. Usually, rotating or reciprocating machinery operating continuously at many cycles per second poses serious reliability problems. On the other hand, compressors for low pressure, high volume vapors such as water, which are problematic on earth could have an advantage in space. A study of an inflatable/retractable radiator [13] has come up with a design of a system serving the functions of a compressor and a radiator/condenser at the same time, thus comprising the major part of a WAHP. The above system was developed for a different application - handling burst power. It consists of an inflatable fabric bag, which is extended and filled with steam produced by the waste heat released during the short power burst. As the steam condenses, the bag is retracted and the remaining vapor compressed to maintain a fixed saturation pressure. A special drive mechanism is available for this purpose as well as a sponge system for recovering the liquid condensate from the walls of the bag. A slightly modified version of this inflatable/retractable radiator is capable of serving as a compressor for steam. combined with a condenser. The compression process is gentle in comparison with conventional technology; it is also performed under isothermal, rather than adiabatic conditions, thereby eliminating the possible need for intercooling and saving work.

Similar problems related to rotating machinery are involved in HAHP's with heat engine-driven compressors. The absorption heat pump therefore seems more suitable for space applications, as it avoids most of the problems due to moving parts. The only device involving work - the solution pump - is very small compared to other components, as evident from the example in Table 2. Lightweight fabric radiators of the type mentioned earlier may be of advantage in an AHP design. The two sub-units involving heat rejection • the absorber and condenser - may be made out of coated fabric. Under the inflatable radiator study [13] a sponge system was proposed to recover liquid from the walls of the condenser. A similar system may be employed to deliver and recover absorbent solution to and from the walls of a fabric absorber.

Other AHP considerations are associated with working fluids. For terrestrial applications, a refrigerant with sub-atmospheric vapor pressure is usually not desirable, due to the risk of leaks and the need for continuous purging of non-condensables. In space, non-condensables do not constitute a problem and a low vapor pressure refrigerant would be desirable. There is a need for an absorbent/refrigerant pair with high temperature lift. Water, with its high latent heat is a promising refrigerant with a number of possible absorbents.

6.0 CONCLUSIONS

The radiator area and system mass savings achievable through the use of heat pumps have been calculated for WAHP and HAHP. For given power and payload waste heat temperatures and a given space sink temperature, there is generally a degree of freedom in the choice of the temperature boost delivered by the heat pump, traded off against the coefficient of performance. An optimum condition which maximizes the savings may therefore be found. The results have been expressed in dimensionless form and the governing parameters identified. The fraction of radiator area saved under optimum conditions is a function of two temperature ratios: T_0/T_3 (space sink to payload) and T_2/T_3 (power source to payload waste heat). The available (or affordable) heat pump mass per unit power, normalized with respect to radiator mass per unit power, is also a function of these two temperature ratios, but in the case of WAHP it depends additionally on - the dimensionless power mass penalty.

Cases with ideal (Carnot) and non-ideal heat pumps have been analyzed. Both the area and mass savings under optimal conditions increase with T_0/T_3 and with T_2/T_3. Best results are achieved where the payload temperature is close to that of the available sink. For the same operating temperatures, the fraction of area saved is about the same for the WAHP and HAHP, with some advantage to the former. The mass saving is considerably in favor of the HAHP, due to the power mass penalty associated with the WAHP.

The absorption heat pump, a particular form of the HAHP, holds considerable promise for space applications due to few moving parts and possible lightweight design. Further development of the AHP is needed with the specific space application in mind, particularly in working fluids and system design.

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