Conceptual Fluid-Dynamic Heat Rejection System for Space Station Application

Yasunori Kobayashi*
University of Tsukuba, Sakura, Japan

A new type of fluid dynamic heat rejecting system employing heat pump heat rejectors is proposed for a future large scale space station in which more than a hundred kilowatts of heat energy will be dissipated in space. One of the significant features of this new system is that it works both as a radiator and as a refrigerator depending on the thermal environment of the spacecraft. Another feature is that the system can operate at relatively low pressure ranges (less than 1 MPa) by selecting a suitable working fluid, since its thermodynamic cycle consists of two adiabatic and two constant-volume changes under a vapor-liquid coexistent condition. This will lead to a high coefficient of performance and a simplicity of structural design. To verify the validity of this concept, a simple laboratory model of this system was manufactured and its cooling capability demonstrated.

Introduction

THE development of an effective heat rejecting system is one of the critical engineering problems that must be resolved for designing large scale space stations. Since the power consumption on such stations is anticipated to be on the order of more than 100 kW, which is large compared with those that have been required on conventional spacecraft, the station will need an innovative concept for the thermal control system. Currently proposed and developed methods[1,2] all employ active fluid circulation with or without phase change. However, it is not yet clear which system is compatible with that of the overall spacecraft when mission analysis, orbital trajectory, and attitude control are taken into consideration as well as its compatibility with structural configuration and restraints. Further, there remain several fundamental design problems to be clarified for future heat rejecting systems. The main concern is the choice of the type of fluid circulation loop (single-phase or two-phase), since it may drastically change the whole thermal control system. Although a current stage of space technology may not be sufficiently advanced to use two-phase loop systems and/or other advanced concepts,[3,4] evaluation of the possibility of these systems should be continued from the viewpoint of system growth capability.

This paper presents a concept of an entirely new type of heat rejecting system, the heat pump heat rejector (HPHR), which also has an ability for thermal control. This system takes advantage of the author's previous proposal—heat pipe thermodynamic cycle in association with a heat pipe engine and power generator.[5-7] As an application of the HPHR in space, a conceptual design of fluid dynamic loop combined with this new heat rejecting system is developed for a space station with a presumed heat rejecting rate of 100 kW, and its feasibility is discussed after some parametric study on the fundamental elements of such loop systems. In order to identify the HPHR character when evaluating these systems, the loop is assumed to be a single-phase liquid of distilled water with other simplified flow and thermal conditions.

Principle of the Ideal HPHR

Ideal Thermodynamic Cycle Process

A basic idea of the new heat rejecting system originates in the thermodynamic sequence of the heat pipe engine.[5,7] In effect, this system has a reversed thermodynamic cycle process like that of the heat pipe engine. Its ideal thermodynamic cycle consists of two constant-volume and two adiabatic changes in the liquid-vapor coexistent region. This cycle corresponds to that covered by the saturated-liquid and saturated-vapor curves in the thermodynamic chart, as schematically shown in Fig. 1. P, V, X, T, and S in the figure represent pressure, specific volume, quality (or mass ratio of vapor to total fluid), temperature, and entropy of the working fluid, respectively. The cycle is proceeded from a thermal state 1 to 1 via 4, 3, and 2, as directed by arrows in the figure. During this ideal cycle, heat energy is absorbed from an external heat source in the process of changing the state from 4 to 3, and it is rejected to the other heat source in the process from 2 to 1. Between these two constant-volume processes, the fluid is adiabatically expanded to be cooled down (from 1 to 4), and compressed to be heated up (from 3 to 2).

In a practical process, the temperature of the state 3 (T_3) represents a heat absorbing place or spacecraft platform where various electronic boxes and equipment are mounted. The temperature of the state 1 (T_1) represents a heat rejecting place or radiator panel outside the spacecraft. Once the two constant-volume curves in the thermodynamic chart (V_1 and V_3) are given, temperatures of T_1, T_2, T_3, and T_4 can be arbitrarily chosen for them and, as consequence, numerous HPHR cycle processes become available for design. This situation can be seen more clearly in Fig. 1b. For example, all cycles taking processes of $T_1'' - T_4'' - T_3'' - T_2'' - T_1''$, $T_1 - T_4 - T_3 - T_2 - T_1$, and $T_1' - T_4' - T_3' - T_2' - T_1'$ are possible by choosing T_1 at a different point on the V_1 constant-volume curve. (The same effect occurs when T_3 is changed on the V_3 constant-volume curve.) Among these processes, heat is transported from higher to lower temperature energy sources when the relation of $T_3 > T_1$ holds. When $T_3 < T_1$ holds, heat is transported from lower to higher temperature energy sources. Even in the condition of $T_3 = T_1$, this cycle can transport heat energy by taking a cycle process of $T_1 - T_4 - T_3 - T_2 - T_1$, as illustrated in the figure. These facts indicate that the HPHR system utilizing this thermodynamic cycle will work both as a radiator and as a refrigerator in the same cycle process, depending on the relative thermal condition of the states 1 and 3. Consequently, the HPHR is able to maintain the temperature level of T_3 (or that of the spacecraft) by automatically adjusting the heat rejecting rate, regardless of the spacecraft's thermal environment (such as the amount of heat to be rejected or the change of the radiator surface temperature due to solar energy insolation). Likewise, the HPHR could keep the radiator temperature relatively constant by adjusting the heat rejecting rate. These features are exclusively characteristic of the HPHR system, and are realized by changing its piston cycle frequency, which is controlled by

Presented as Paper 85-1051 at the AIAA 20th Thermophysics Conference, Williamsburg, VA, June 19-21, 1985; received Aug. 9, 1985; revision received Feb. 24, 1986.

*Associate Professor, Institute of Engineering Mechanics. Member AIAA.

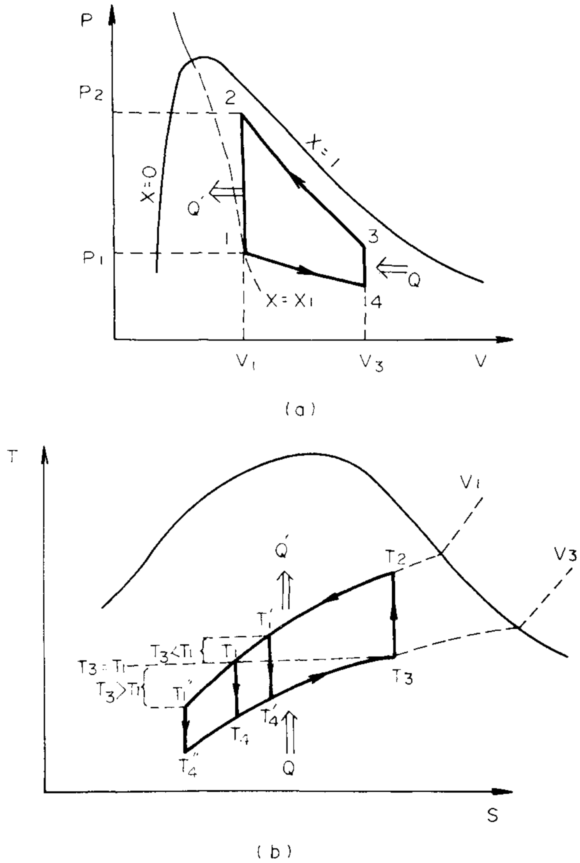


Fig. 1 Schematic illustration of ideal thermodynamic cycle of the HPHR: a) schematic P-V chart; b) schematic T-S chart.

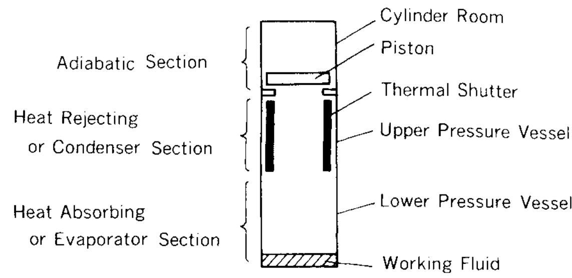


Fig. 2 Schematic illustration of the HPHR ideal structure and its main components.

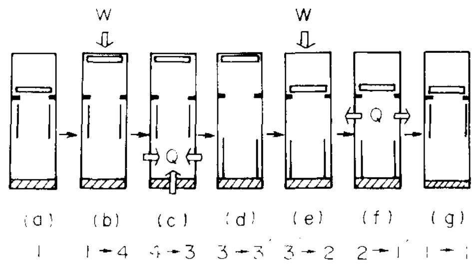


Fig. 3 Ideal HPHR cycle process illustrated with positions of the piston and the thermal shutter.

the externally equipped HPHR frequency monitor. It should also be noted that the system can be operated in a low pressure region, as it does not yield Joule-Thomson effects under high vapor pressure. This will lead to less severe requirements concerning the choice of materials and their thermophyiscal properties, which will in turn make the structural design and manufacturing of the HPHR easier.

Ideal Structural Configuration

The ideal mechanical configuration of an HPHR, which is basically identical to that of a heat pipe engine, is schematically illustrated in Fig. 2. The entire vessel is regarded as a heat pipe or an upright thermosyphon, and is thermally configured into three sections: adiabatic condenser, and evaporator. Functionally, it is divided into two segments of cylindrical chambers, one on the top and another (the pressure vessel) on the bottom, A device called a "thermal shutter" is installed in the pressure vessel. It is a simple hollow cylindrical structure made of thermally non-conductive material without a top or bottom surface. This shutter moves up and down in the pressure vessel (ideally with no friction). The role of this device is to periodically cover and expose the condenser wall so that the vapor flow may alternately condense and not condense on the wall. The ideal thermodynamic cycle process of this system is realized by the following sequential motion, as schematically illustrated in Fig. 3.

1) The piston is driven up from the initial configuration by the pressure resulting from the adiabatic expansion and evaporation of the working fluid (Figs. 3a and 3b).

2) The external heat is absorbed into the pressure vessel through the lower wall (evaporator section) which is cooled by the expanding vapor and liquid evaporation (Fig. 3c).

3) The thermal shutter moves downward to cover the evaporator section, and stop heat absorption (Fig. 3d).

4) The piston is driven down to compress and heat the vapor adiabatically (Fig. 3e).

5) Heat is rejected to the outside through the upper pressure vessel (condenser section) (Fig. 3f).

6)The thermal shutter goes back to the top position and the system returns to the original configuration (Figs. 3a and 3g). Thus, one cycle of the HPHR is completed with transporting heat energy from one heat source to the other. Numeric signs and arrows in the figure indicate the thermal states and directions of change that correspond to those in Fig. 1. It is important to note that the movement of the thermal shutter itself does not generate any effect on the prescribed thermodynamic cycle processes. Practically, its movement will need only a small amount of power since it moves with little friction with the vessel wall.

Typical HPHR Performances

When the coefficient of performance (COP) of the HPHR is defined as a ratio of transported heat energy to input work, the relation of COP with regard to the enthalpy and to the pressure of the working fluid is expressed in a simple form as

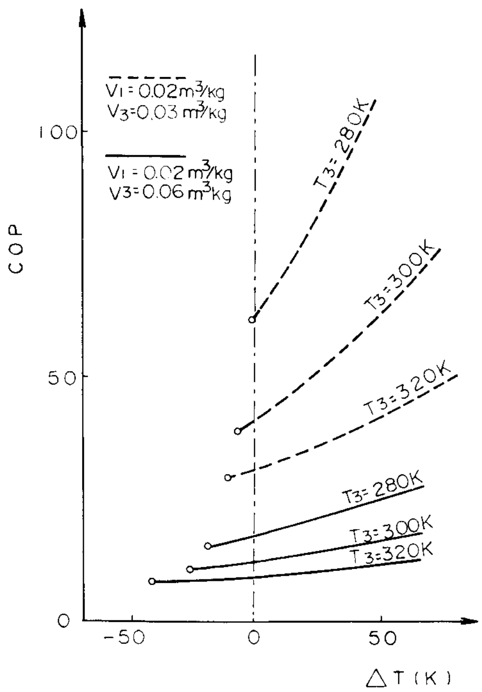
$$\mathrm{COP} = \frac{(h_3 - h_4) - (P_3 - P_4)\cdot V_3}{(h_2 - h_1) - (h_3 - h_4) - (P_2 - P_1) - V_1 + (P_3 - P_4)\cdot V_3}$$

where h, P, and V represent the enthalpy, pressure, and specific volume of the working fluid, respectively, and the numeric suffixes stand for the thermal states corresponding to those in Fig. 1.[7]

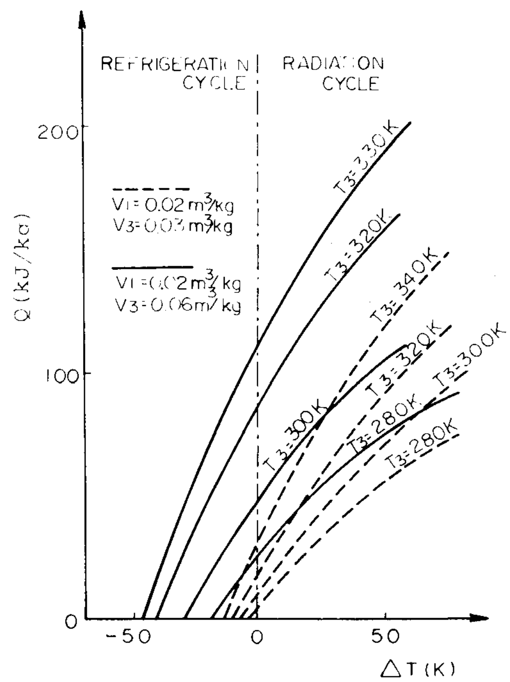
Figure 4 shows typical HPHR performances calculated for Freon 11 and ammonia for two cases of volume changes, from 0.02 to 0.03 and 0.06 m^3/kg, which are obtained by using the empirical state equations with real gas effects in terms of enthalpy vs pressure for individual fluids.[8,9] Figures 4a and 4c show COP values of the HPHR with respect to the temperature difference ΔT between those of the heat absorbing and rejecting sources in the cycle (or $\Delta T = T_3 - T_1$ in Fig. 1b). Figures 4b and 4d show the transported heat energy Q that a unit mass of working fluid can carry per cycle of piston movement in the cylinder room. As discussed in the previous section, the left half of the figure where $\Delta T < 0$ indicates the HPHR performance as a refrigerator, while the right half indicates its function as a radiator. The open circles of curves at the left end in Figs. 4a and 4c indicate the limitations on ΔT range beyond which the HPHR will not be able to function as a refrigerator. Large COP values but relatively small transfers of heat energy Q are obtained from these results when the small constant-volume changes are selected. This indicates that a thermodynamic cycle with large constant-volume change must be incorporated so that the HPHR can be operated over a larger ΔT range. It should be emphasized that these are just two examples with T_3 at near room temperatures. They represent general tendencies of the HPHR function and typical performances, but they may not necessarily produce an optimum performance.

Conceptual Thermal Design of 100-kW HPHR

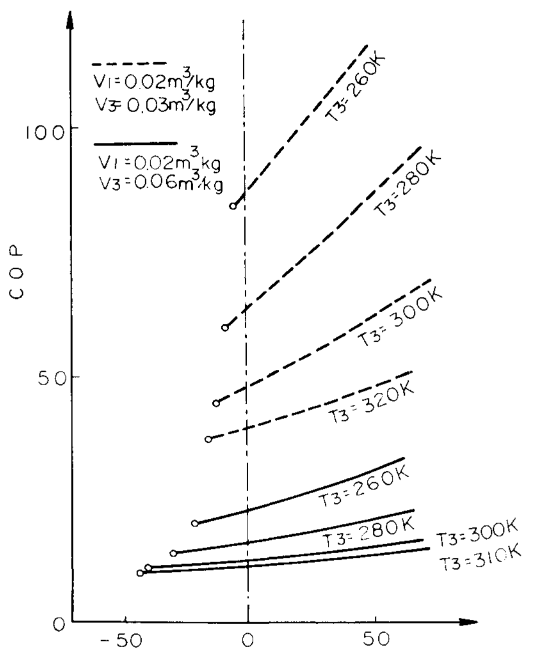
Let us consider a conceptual design of a heat rejecting system for a space station which will dissipate 100 kW of power in space. Temperatures of the spacecraft platforms and structures (T_3) on which various heat dissipating components and equipment will be mounted and those of the radiator panel (T_1) are assumed to be uniform. Further, the HPHR is assumed to be operated by combining two families of coolant loops, i.e., a heat collecting or hot liquid line and a heat dissipating or cold liquid line. A schematic block diagram of such a heat rejecting system for a 100-kW space station is



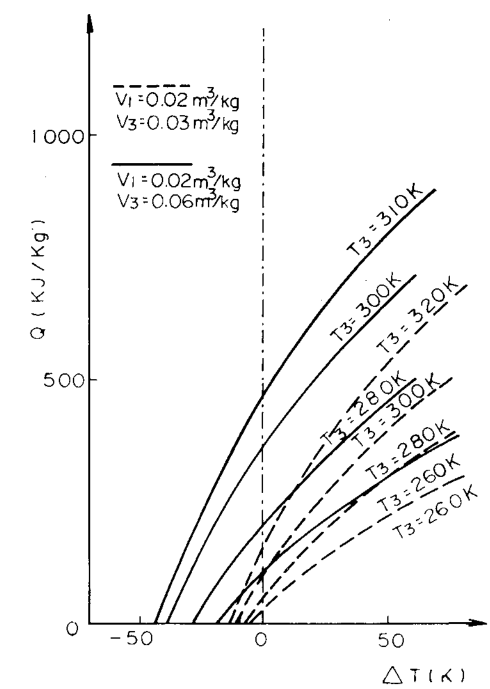
a) COP values for Freon 11.



b) Reject heat Q per cycle of the piston for Freon 11.



c) COP values for ammonia.



d) Reject heat Q per cycle of the piston for ammonia.

Fig. 4 A typical HPHR performance calculated with heat absorbing temperatures as a parameter.

shown in Fig. 5, where four units of HPHRs, each capable of a maximum 25 kW of heat rejection, are connected in series. The hot liquid line will circulate inside the space station and the cold liquid line will circulate through the radiator panel outside in space. A number of fixed-conductance heat pipes (FCHP) buried in the structural membranes, platforms, and radiator panel will serve as an effective heat collector and/or distributor at various local heat exchanging components, reducing thermal resistance between the coolant fluid and these components.

A practical configuration and size of an HPHR for space use, for instance, of a 25-kW heat rejecting capability, will be realized, as illustrated in Fig. 6, where the scale is in millimeters and the cylinder room and the thermal shutter are arranged in one component. The on-off valve will be operated in synchrony with the piston movement to let the working liquid evaporate from and make it return to the hot pressure vessel (or evaporator section) where the working liquid will absorb heat from the hot liquid line. Although this practical configuration looks a little different from that of Fig. 2, it is apparent that the basic function and cycle processes are identical for both cases.

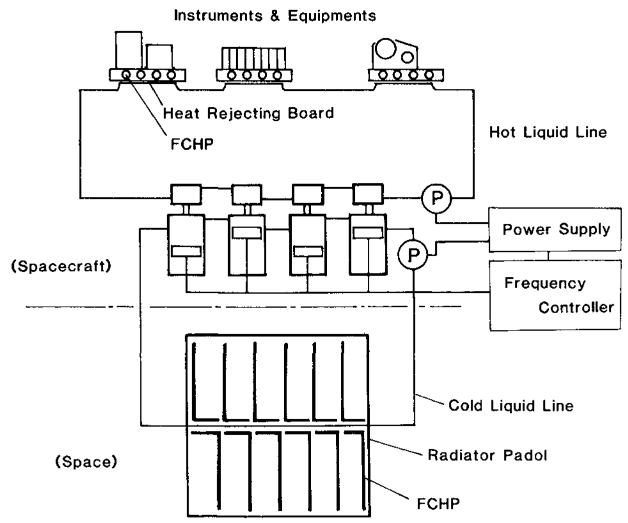


Fig. 5 Block diagram of a 100-kW heat rejecting system using the HPHR for space use.

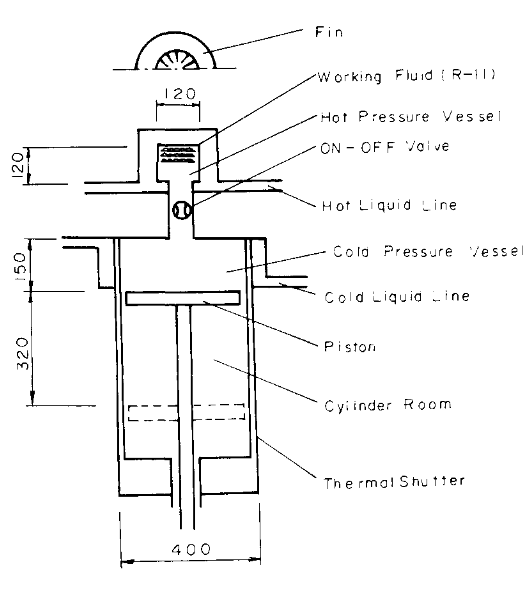


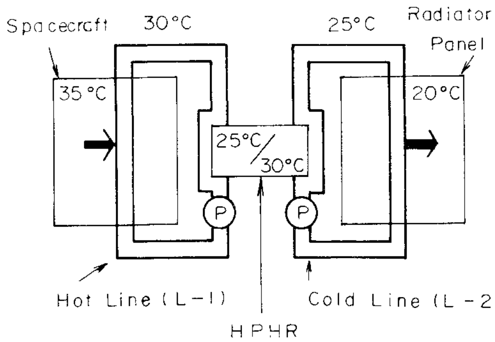
Fig. 6 Schematic HPHR configuration of a 25-kW heat rejecting capability for space use.

A trade-off study of fluid lines for a 100-kW heat transporting capability has also been conducted for single- and two-phase fluid loops. However, an evaluation of the HPHR is made only in combination with a single-phase loop of distilled water, since there are many uncertain design parameters for a two-phase loop, such as the boiling heat transfer coefficient, dynamic flow pattern of a fluid in a microgravity condition, and dry-out limit of a fluid in the loop, as well as the possible occurrence of flow instability in the tube. Table 1 shows part of a typical performance of a cooling system with an HPHR for two cases where the average radiator surface temperatures were 20°C and 35°C, using the loop tube diameter as a parameter. Two kinds of fluids, Freon 11 and ammonia, are considered for the HPHR working fluid. The following assumptions are made for the calculation: 1) the heat load is uniformly distributed over the line; 2) the line is a circular aluminum tube of 20 m in total length, and 15 m in effective heat exchanging length; 3) the total heat loss of the line is assumed to be compensated for by a pumping power of 80% efficiency; and 4) the temperature differences between the contacting components are assumed to hold to 5°C everywhere, with a fixed spacecraft temperature of 35°C. The total power in the right column of the table signifies the summation of a net amount of 100-kW reject heat plus the necessary power for the operation of the HPHR system and the two liquid pumps. This table also shows that the HPHR cycle frequency and operating pressure ranges will be different with different working fluids. For instance, an HPHR with Freon 11 may be operated in lower pressure regions but needs a higher piston cycle frequency than one with ammonia. As the HPHR COP value used in these cases is about 10, the required power for the HPHR operation for a 100-kW heat rejection is about 10 kW, which is rather large. However, we must remember that the fluid loops without an HPHR will usually need a kind of thermal control or protection subsystem which may require additional power and elaborate radiator design.

In order to clarify this point, a comparison is made between 100-kW heat-rejection loop systems with and without an HPHR, as summarized in Table 2. The left half of this table shows the length of cooling liquid line required for a 100-kW heat rejection with respect to different radiator surface temperatures, assuming the average spacecraft temperature to be 30°C and the inner diameter of the loop to be 0.03 m. The simple loop system without an HPHR requires a longer cooling liquid line as the radiator surface temperature rises. It will not be able to reject appropriate heat energy at temperatures above 25°C due to an insufficient temperature difference between coolant liquid and radiator surface. On the other hand, the loop with an HPHR will work, even when the radiator surface temperature becomes higher than that of the spacecraft, by HPHR operation with almost the same input power of about 8 kW and with changing piston cycle frequency.

Typical conceptual design performances and specifications of an HPHR system of 100-kW heat rejecting capability with Freon 11 as the working fluid are summarized in Table 3. Here, in addition to the assumptions used for Table 1, the following are also assumed: the average spacecraft temperature is 30°C; the inner diameter of the loop is 0.03 m; the total weight of the HPHR is estimated based on a construction entirely from aluminum; and the HPHR is operated at the design point of a 20°/30°C refrigerator (see Table 2) to accomplish the required performance. The amount of an 8-kW power penalty for the HPHR operation shall be evaluated by considering the potential advantages for the overall system design of spacecraft. As discussed previously with regard to Table 2, it is clear that the thermal condition of spacecraft with a simple loop heat rejecting system is strongly affected by the radiator temperature and size. Roughly estimated power and weight penalties allocated to thermal control equipment and subsystems such as radiator articulation and orientation, electric heaters, accumulators, etc., will

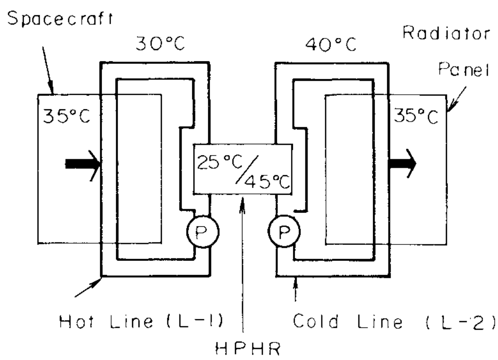
Table 1a Typical results of parametric study for HPHR system, functioning as a 25°/30°C refrigerator



Diameter	Mass flow rate, kg/s L-1	Mass flow rate, kg/s L-2	Pumping power, hW L-1	Pumping power, hW L-2	HPHR cycle[a] freq., rpm Freon	HPHR cycle[a] freq., rpm Ammonia	Total power, kW
0.03	7.8	9.8	5.5	10.4	45	4.5	126.4
0.04	9.0	10.8	2.0	3.4	44	4.4	115.6
0.05	10.1	12.1	1.0	1.3	43	4.3	112.7

[a]HPHR operating pressure range: Freon 0.08–0.36 MPa, Ammonia 0.48–3.4 MPa.

Table 1b Typical results of parametric study for HPHR system, functions as a 25°/45°C refrigerator



Diameter	Mass flow rate, kg/s L-1	Mass flow rate, kg/s L-2	Pumping power, hW L-1	Pumping power, hW L-2	HPHR cycle[a] freq., rpm Freon	HPHR cycle[a] freq., rpm Ammonia	Total power, kW
0.03	7.8	8.8	5.5	7.3	88	9.3	126.3
0.04	9.0	9.5	2.3	2.3	85	9.0	114.4
0.05	10.1	10.5	1.0	1.0	84	8.9	112.1

[a]HPHR operating pressure range: Freon 0.04–0.36 MPa, Ammonia 0.38–3.4 MPa.

amount to approximately 3 kW and 200 kg. The impact of the HPHR system on the radiator weight becomes larger, as it can function with a radiator up to 25% smaller in surface area than that of the simple loop system. Whether the HPHR system has an advantage in weight over other cycle systems, such as Rankine and Stirling refrigerators, has not been thoroughly investigated because little reliable information is available to the public. However, it is reasonable to state the following: 1) The proper radiator size will be determined mostly by the refrigerating performance of the system used; 2) As far as the theoretical thermal performance is concerned, an HPHR does not have any inferior characteristics when compared with other candidate systems; and 3) The structural configuration of an HPHR system will be lighter and smaller than the other systems which employ a process of high pressure gas phase in their cycle and are equipped with auxiliary components. From these, the HPHR system appears to have an equal or larger opportunity to be realized in the future.

There remain a number of technical problems that should be resolved in order to employ the HPHR in space. Among them the following are typical: 1) sealing of the HPHR main body; 2) effective boiling heat transfer method and/or augmentation device in the pressure vessel; 3) minimization of the thermal conduction loss along the wall between the upper and lower pressure vessels; 4) return of the liquid and its confinement in the hot pressure vessel under microgravity conditions; 5) possible unfavorable vibration effect of the system on the station. Of these, the first three items seem to be the most serious and need further investigation and development. But none of these have insurmountable difficulties and the solution will be found in the existing technological base developed in various engineering field, by the automobile, aeronautical,

Table 2 Comparison of heat-rejecting capability of the system with and without an HPHR[a]

Radiator surface temperature, °C	Loop without HPHR: Line length, m	Loop without HPHR: Pumping power, kW	Loop with HPHR (4 units of 25 kW HPHRs): Line length, m	Pumping power, kW	Operating power, kW	Operating temperature, °C	Cycle frequency, rpm
20	5.5	2.4	5.0	1.0	8.0	20/30	60
22	8.4	3.7					
24	17.8	7.7					
25	–	–	5.0	1.0	8.0	20/35	75
30	–	–	5.0	1.0	8.0	20/40	150

[a]The HPHR works as a refrigerator or a heat pump transporting heat energy from 20 to 30°C heat sources.

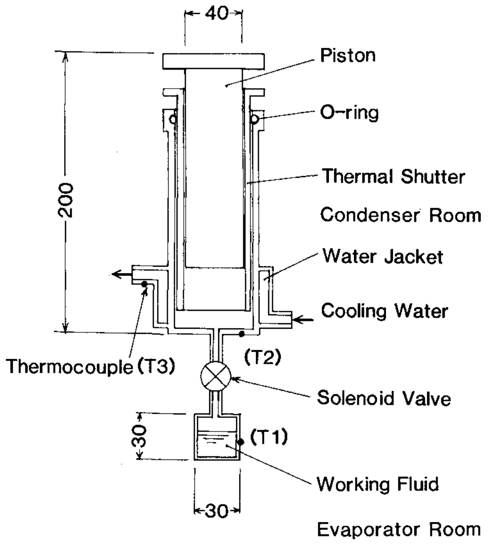


Fig. 7 Laboratory model of an HPHR manufactured for the purpose of preliminary experiment.

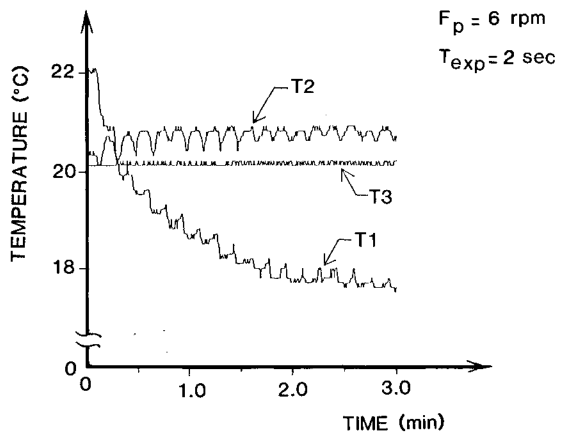


Fig. 8 Typical cooling result obtained by the HPHR laboratory model.

space, and other similar industries and research institutions. Some technology, such as that of heat pipes or Stirling engines, will be directly applicable to the development of this system.

Preliminary Experiment of the HPHR

An HPHR laboratory model of similar configuration to that in Fig. 6 and illustrated in Fig. 7 was manufactured and tested to demonstrate its cooling function as a radiator and as a refrigerator. The piston and the thermal shutter were made of glass and the rest was constructed of copper. The electromagnetic valve was installed between the hot and cold

Table 3 Conceptual design specifications of the spacecraft using a 100-kW HPHR system

Heat collecting board	Augmented by FCHP[a]
Heat pump heat rejector (HPHR)	
Working fluid	Freon 11 (43 kg)
Design temperature	
Hot pressure vessel	25°C
Cold pressure vessel	30°C
Maximum operating pressure	0.36 MPa
Piston cycle frequency	40 rpm (normal)
Required power	8 kW
Layout	25 kW × 4 units
Weight	116 kg
Radiator panel	
Surface area	170 m^2
Attitude and orientation	Fixed mode
Surface coating	SSM[b] ($\alpha = 0.07$, $\epsilon = 0.80$)
Surface temperature	20°C
Main heat path	Augmented by FCHP
Weight	1830 kg
Liquid circulation line and pump	
Hot liquid line	Distilled water
Cold liquid line	Distilled water
Pump power	2.3 kW
Weight including fluid	540 kg

[a]FCHP = Fixed conductance heat pipe. [b]SSM = Second surface mirror.

pressure vessels. A total of 15 g of Freon 11 was put in the hot pressure vessel after evacuating it. The cold pressure vessel was cooled by running water through the water jacket. The entire model was covered by several layers of thermal insulation. Several values of piston frequencies and valve opening timings were tested to examine the effects of these two important parameters.

One of the typical results of this HPHR model is shown in Fig. 8. In this case, the piston was moved at the constant frequency F_p of 6 rpm and the valve opening duration per piston cycle T_{exp} was 2 s. The abscissa represents the elapsed time after the HPHR cycle started and the ordinate represents the temperature variations of the hot pressure vessel (T_1 in Fig. 7), cold pressure vessel (T_2), and exit cooling water (T_3). This figure indicates that in the first 30 s, until T_1 became equal to T_3, the HPHR worked as a radiator. Later, when T_1 became lower than T_3, it worked as a refrigerator until a final steady state temperature (of about 18°C in this case) was reached. The calculated value of COP as a refrigerator under a thermally steady-state condition in this case was about 4.8, which was not as large as the ideal prediction. This discrepancy resulted mainly from relatively large heat dissipation from the electromagnetic valve installed in the model and from insufficient thermal insulation of the cylinder room. Figure 9 shows a similar cooling performance of this model for three different valve opening durations under the same piston cycle frequency of 3 rpm. These data strongly suggest that there must be an optimum valve opening timing and duration for maximizing HPHR heat transporting capability. Although the experiment was primitive, it did demonstrate the HPHR character

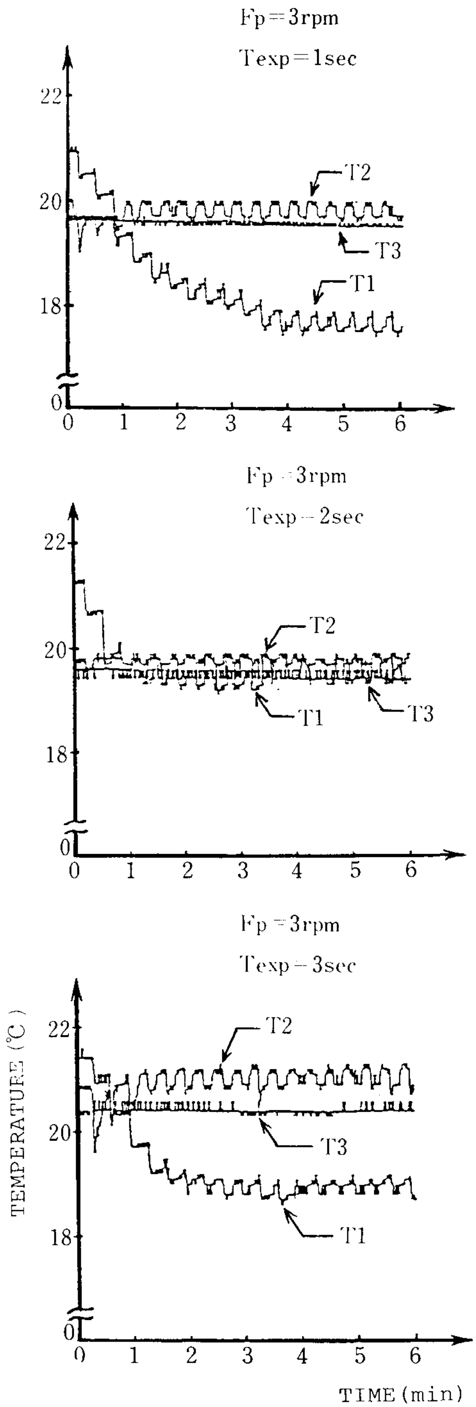


Fig. 9 Typical results indicating the effect of valve opening durations on HPHR cooling performance.

qualitatively and the purpose of verifying of the concept was fulfilled. Currently, a more sophisticated developmental test, which uses a more realistic HPHR model combined with coolant loops, pumps, and a cold plate, is under preparation.

Conclusions

The concept of the heat pump heat rejector (HPHR) is proposed as a new heat rejecting system with thermal control capability that can be used as a main facility and/or auxiliary subsystem for a future large scale space station. The principal feature of this HPHR system is that it works both as a radiator and as a refrigerator, depending on the thermal condition of the space station, by automatically adjusting its heat rejecting rates. The system can keep the spacecraft temperature relatively constant, regardless of the thermal condition of the radiator panel, by changing the piston cycle frequency which is in turn controlled by the external cycle frequency monitor. Essentially, no additional thermal control and protection devices or subsystems are required in this system.

The major technical problems of the HPHR system are the sealing of the vessel and thermal decoupling along the vessel wall. (The sealing of the vessel at the moving portion of the piston and thermal shutter will affect the reliability of the system.) These are, however, surmountable difficulties that will be eventually overcome by applying the existing technology accumulated in various engineering fields.

References

[1]Ollendorf, S., "Recent and Planned Developments at the Goddard Space Flight Center in Thermal Control Technology," *Proceedings of the International Symposium on Environmental and Thermal Systems for Space Vehicles,* ESA SP-200, Dec. 1983, pp. 45–51.

[2]Laux, U. and Weydandt, J., "System Aspects of the EURECA Thermal Control Subsystem," *Proceedings of the International Symposium on Environmental and Thermal Systems for Space Vehicles,* ESA-SP-200, Dec. 1983, pp. 307–315.

[3]Marchis, V. and Nervegna, N., "Adaptive Control of Thermal Fluid Loops for Space Applications," *Proceedings of the International Symposium on Environmental and Thermal Systems for Space Vehicles,* ESA SP-200, Dec. 1983, pp. 319–327.

[4]Hwangbo, H. and McEver, W. S., "High Thermal Capacity Cold Plate/Hot Plate," AIAA Paper 85-0921, 1985.

[5]Kobayashi, Y., "A Concept of Heat Pipe Engine," *Proceedings of the 17th Intersociety Energy Conversion Engineering Conference,* IEEE, 1982, pp. 1128–1133.

[6]Kobayashi, Y.,"Development of a Phase-Change Power Generator for Space Use," *Proceedings of the 13th International Symposium on Space Technology and Science,* Agune Shobo Inc., Tokyo, 1982, pp. 607–612.

[7]Kobayashi, Y., "Heat Pipe Thermodynamic Cycle and its Applications," *Journal of Solar Energy Engineering, Transactions of ASME,* Vol. 107, No. 2, 1985, pp. 153–159.

[8]Reynolds, W. C., "Thermodynamic Properties in SI," Department of Mechanical Engineering, Stanford University, Stanford, CA, 1979.

[9]*The Properties of Gases and Liquids,* edited by R.C. Rein, J.M. Prausnitz, and T.K. Sherwood, 3rd edition, McGraw-Hill, New York, 1977, pp. 181–222.