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Experimental Investigation of Oscillating Heat Pipes

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Operation requirements of oscillating heat pipes (OHPs) are proposed. Based on the requirements, OHPs with nonflammable fluorocarbon fluids, FC-72 and FC-75, as the working fluid are developed. The OHPs have an inner diameter of 1.75 mm, a total length of 446 mm, and 40 tubing turns. There are two condensers on both outer sides and one evaporator in the middle of the OHPs. Thermal performance tests are conducted at various operating temperatures and heat rates. The working fluid fill ratio is varied. A high-performance OHP with FC-72 has been indicated for the first time. The FC-72 OHP can transport a 2040-W heat rate without dryout. The gravitational acceleration does not have a noticeable influence on the performance of the fluorocarbon OHP. The thermal performances of the fluorocarbon OHPs are compared with the case of an acetone OHP.

Nomenclature

A = outer wall area of tube, m^2
A_i = inner cross-sectional area, m^2
Bo = Bond number, $d_b\sqrt{[g(\rho_l-\rho_g)/\sigma]}$
d = tube diameter, m
g = gravitational acceleration, m s^{-2}
h = heat transfer coefficient, W m^{-2} K^{-1}
h_{fg} = latent heat of vaporization, J kg^{-1}
$j_{g,e}$ = critical vapor superficial velocity, m s^{-1}
j_g^* = dimensionless vapor superficial velocity, $j_{g,e}\rho_g^{0.5}/[gd_i(\rho_l-\rho_g)]^{0.5}$
n = tubing turn number
Q = heat rate (total output heat rate), W
Q_e = onset heat rate for cocurrent two-phase flow, W
Q_1 = heat rate through upper condenser, W
Q_2 = heat rate through lower condenser, W
T = temperature, °C
T_{m3} = operating temperature, °C
ΔT = temperature difference, °C
ρ_g = vapor density, kg m^{-3}
ρ_l = liquid density, kg m^{-3}
σ = surface tension, N m^{-1}
ϕ = fill ratio

Subscripts

$a1$ = upper adiabatic section
$a2$ = lower adiabatic section
b = liquid bridging
$c1$ = upper condenser
$c2$ = lower condenser
e = evaporator
$e1$ = evaporator related to the upper oscillating heat pipe (OHP)
$e2$ = evaporator related to the lower OHP
i = inner
min = minimum

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Introduction

THE application of unconventional heat pipes to high-power electrically driven actuators, for example, electromechanical actuators (EMAs), is important because EMAs possess no inherent means of removing heat. The EMA cooling system should have a passive heat transport feature and compact size, be lightweight, and respond quickly to actuator duty cycles. To meet these requirements, oscillating heat pipes (OHPs) are selected as the heat transfer element for the cooling system.

The OHP is a closed two-phase heat transfer device consisting of a meandering capillary tube. The OHP uses the oscillation movement of two-phase flow to transport energy from the evaporator to the condenser. The OHP operation was experimentally verified for different heat modes relating to OHP orientations.[1,2] Maezawa et al. investigated a chaotic behavior of an OHP with R142b as working fluid by monitoring a tube surface temperature in the adiabatic section. The OHP had a length of 600 mm, an inner diameter of 2 mm, and 40 turns. It was confirmed from an analysis that the temperature oscillation had chaotic characteristics and was governed by the nonperiodic dynamic system of two-phase flow.[2] Overall thermal performances of OHP-based heat exchangers were presented.[2,3] Kiseev and Zolkin tested an OHP on a spin table.[4] The OHP had an inner diameter of 1.1 mm, a length of 420 mm, and 23 turns. Acetone was used as the working fluid, and the fill ratio was 60%. It was exhibited that there was an increase of the evaporator temperature by 30%, with an increase of the acceleration from an adverse acceleration of $-6\ g$ to a favorable acceleration of $12\ g$. Lin et al. recently reported heat transfer characteristics of an acetone OHP with an inner diameter of 1.75 mm (Ref. 5). The fill ratio was varied from 25 to 50%. It was indicated from experimental results that the heat transfer coefficients of evaporator and condenser are noticeably higher for the fill ratio of 38% than those for 50%. The desired fill ratio of the acetone OHP is 38%.

The present investigation deals with OHPs with fluorocarbon FC-72 and FC-75 as the working fluid. The fluorocarbon fluids are selected because of their nonflammability feature, which is desired for the cooling of aircraft EMAs. Heat pipes with FC-72 and FC-75 have not been reported before. The experimental investigation focuses on the thermal performance in the evaporator and condenser. The design of the OHP test setup is aimed at achieving high performances in a wide range of operating temperatures from 25 to 85°C. The fill ratio of the working fluid is varied to find a suitable fill ratio. The input power is adjusted in the range of 140–2100 W. A comparison of OHP heat transfer characteristics between the vertical operation and horizontal operation is made.

Requirements of Operation

An effective operation of the OHP relies on the oscillation movement of the two-phase flow in the OHP. The two-phase flow oscillation, a flow instability, is related to the pressure drops in each channel between two consecutive turns not being in phase with each other. A basic flow pattern in the OHP can be the slug flow caused by liquid bridging or an oscillating cocurrrent two-phase flow, depending on the type of working fluid and the tube diameter. The performance limitation of the OHP is caused by dryout due to an insufficient liquid supply to the evaporator.

The slug flow (liquid bridging) occurs under the condition that the inner tube diameter of the OHP is less than a critical diameter d_b, which is determined using the following relation[1,6]:

$$Bo = 2 \tag{1}$$

where the Bond number Bo is expressed as

$$Bo = d_b \sqrt{g(\rho_l - \rho_g)/\sigma} \tag{2}$$

The cocurrent two-phase flow dominates in case the vapor superficial velocity is greater than a critical value $j_{g,e}$, so that the vapor can drag the liquid along with it. The critical vapor superficial velocity in the case of vertical operation depends on the following equation[7]:

$$j_g^* = 0.89 \tag{3}$$

where the dimensionless vapor superficial velocity j_g^* is given by

$$j_g^* = \frac{j_{g,e}\rho_g^{0.5}}{[g d_i(\rho_l - \rho_g)]^{0.5}} \tag{4}$$

Under the assumption that the vapor and liquid are in the saturated state and the heat rates transported by every channel are the same, the minimum heat rate Q_e, required to generate the critical vapor superficial velocity, is determined as follows:

$$Q_e = (n+1) h_{fg} A_i \rho_g j_{g,e} \tag{5}$$

To maintain the proper two-phase flow pattern (the slug flow or cocurrent two-phase flow), either the condition of $d_i < d_b$ or $Q > Q_e$ should be satisfied. A smaller Q_e is beneficial to the OHP startup.

The passive oscillation movement of the two-phase flow requires that the tubing array of the OHP should have sufficient turns. The minimum tubing turn number for the excited oscillation depends on dynamic balances of mass, momentum, and energy of the two-phase flow in the OHP and heating and cooling boundary conditions. Because constitutive relations of the two-phase flow and heat transfer in the OHP have not been developed, it is difficult to estimate the minimum tubing turn number through numerical simulation. In the present experiment, the tubing turn number of $n = 40$ is selected according to the application background.

There exists the minimum fill ratio of the working fluid in which the OHP cannot operate even at a low heat rate, for example, 10% of its maximum performance. The fill ratio refers to the ratio of the fill liquid volume (at 25°C) to the OHP internal volume. The proper operation of the OHP requires also that the actual fill ratio should be greater than the minimum value, $\phi > \phi_{\min}$. Beyond $\phi_{\min}$, an increase in ϕ can result in an increase in the internal thermal resistance of the OHP. Concerning this, a proper fill ratio is searched for in the present experimental study of the OHP.

Experimental Setup and Procedure

The OHP for the thermal performance test is shown in Fig. 1. The OHP is made of a small copper tubing with an inner diameter of 1.75 mm and outer diameter of 3.18 mm. The inner radius of the OHP tubing turn is 6.35 mm. The tubing is bent into a multitube array having 40 turns and a length of 446 mm. Both ends of the tubing are separately sealed. There are two condensers on the top and bottom of the OHP and one evaporator in the middle of the OHP. Lengths of the evaporator, condenser (on one side), and adiabatic section (on one side) are 125, 108.7, and 51.8 mm, respectively. Because of their nonflammability feature, FC-72 and FC-75 are employed as the OHP working fluid. The thermal performance of the fluorocarbon OHPs is compared with that of an OHP with acetone as the working fluid. The fill ratio of the fluorocarbon fluids is varied from 32 to 50%. In the range of the present OHP operating temperature (from 25 to 85°C), the smallest d_b and the greatest Q_e occur at 85°C. For FC-72 and FC-75, the values of d_b at 85°C are 1.1 and 1.38 mm, smaller than the OHP d_i (1.75 mm). However, the values of Q_e are only 190 W (FC-72) and 110 W (FC-75), small enough to operate

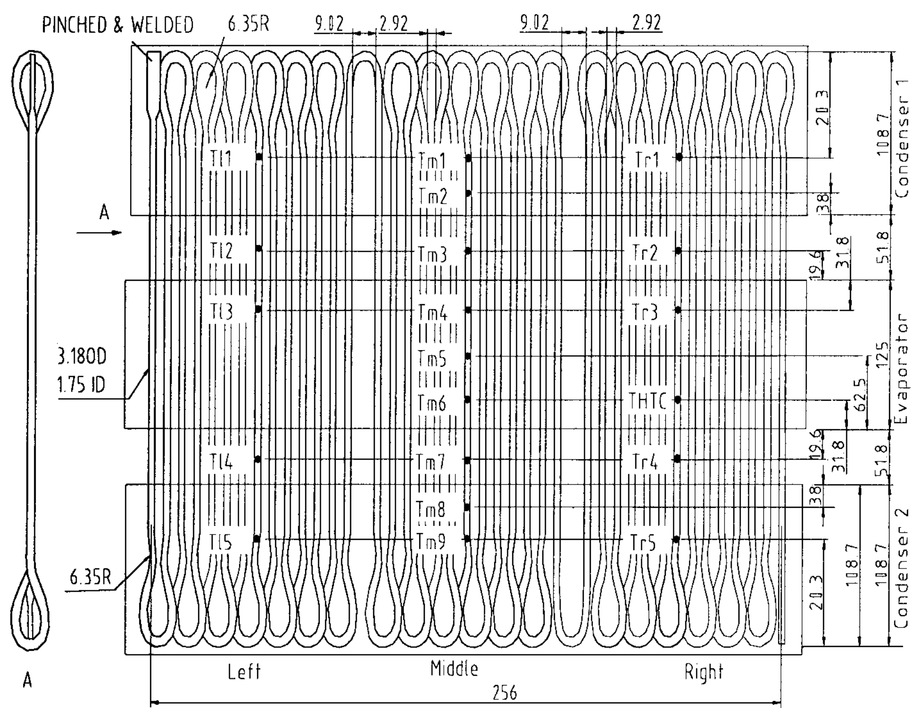

Fig. 1 OHP and thermocouple locations in millimeters.

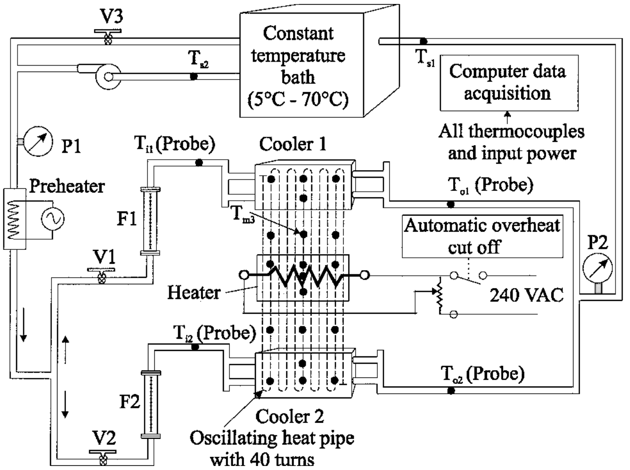

Fig. 2 Schematic of OHP experimental setup: $P1$, $P2$ = pressure gauges; $V1$, $V2$ = flow control valves; $F1$, $F2$ = flow meters; $V3$ = bypass valve; and ● = thermocouples.

the OHP under the condition of the oscillating cocurrent two-phase flow. For acetone, the value of d_b at 85°C is 3.0 mm, greater than the OHP d_i, so that the slug flow exists in the OHP.

The schematic of OHP experimental setup is shown in Fig. 2. The heater contains four aluminum thermal conduction plates, two on one side and the other two on the opposite side of the tube array. The thermal conduction plates are machined on their contact side with the OHP tube array into semicircle grooves having the same radius as the OHP tube. The thermal conduction plates are tightly mounted onto the evaporator of the OHP by mechanical joining. To keep a good contact with the OHP tube, thermal conduction grease is applied between the OHP tube and thermal conduction plate. Two electric resistors are installed in the thermal conduction plates, one on each side of the evaporator, and are connected in parallel. There are two coolers, each connecting with one OHP condenser. The main parts of the cooler are a rectangular tank, a couple of rectangular flanges, a couple of sealing bars, room temperature vulcanization sealing material, and coolant inlet and outlet manifolds. Water is used as the coolant. In addition to the coolers, the coolant circulation system includes two control valves, one bypass valve, two flowmeters (rotameter type, 0.00631–0.0631 l/s), two pressure gauges (0–1.03 bar), one preheater, one water pump, and one constant temperature bath that contains a filter. The coolant flow rate is controlled by the two control valves. The supply pressure ($P1$) can be adjusted using the bypass valve, $V3$. The temperature of the coolant bath can be set in the range of 5–70°C. AC power is supplied to the resistance heater through a variac with which the input heat rate can be adjusted. The input power is digitally displayed by using a MAGTROL power analyzer.

Nineteen type T thermocouples with a diameter of 0.2 mm are placed on the OHP wall, among which five are located in the evaporator, six in the adiabatic sections, and eight in the condensers, as shown in Fig. 1. The thermocouples can be divided into three groups such as left group, middle group, and right group. The middle group is related with the middle tube, the left group with the eighth tube counted from the left side, and the right group with the eighth tube counted from the right side. For each group, the thermocouple locations are indicated in Fig. 1. In the cooling system, four probe thermocouples, T_{i1}, T_{i2}, T_{o1}, and T_{o2} are used to measure the inlet and outlet temperatures of the coolers. Two thermocouples, T_{s1} and T_{s2}, are used to monitor the inlet and outlet temperatures of the coolant bath. The signals of the 19 thermocouples are acquired using a KEITHLEY 500A measurement and control unit supported by a Viewdac data acquisition and control program. In addition, 38 thermocouples are attached to the rest of the tubes at locations close to the top of the evaporator to monitor the temperature of each tube. These thermocouples are connected to a rotary switch from which a thermocouple is led to a FLUKE 2100A digital thermometer. For safe operation, a thermocouple measuring the evaporator temperature is connected to a high-temperature cutout unit. If the temperature is too high, the power supply will automatically cut out.

The tests are started by turning on the coolant circulation system, setting flow rate, and controlling the supply pressure ($P1$) by adjusting the bypass valve. The variac is then adjusted until a desired electric power is displayed on the MAGTROL unit. The OHP operating temperature (temperature level) indicated by the temperature of T_{m3} is maintained to be a constant by adjusting the flow rate and coolant bath temperature for different input powers. After the system attains steady state, the temperature, flow rate, and input power are recorded. The present experimental conditions are as follows: OHP temperature level, 25–85°C with 10°C intervals; input power, 200–2100 W; coolant flow rate, 0.00631–0.025 l/s for each flowmeter; coolant inlet temperature, 2.4–63.6°C; fluid fill ratio, 50% (22.5 ml), 38% (16.9 ml), and 32% (14.24 ml); OHP orientation, vertical (indicated in Fig. 1) and horizontal (with the multitube array in the horizontal plane).

The inlet temperature of the coolant for the top cooler is almost the same as that for the bottom cooler. During the experiment, the flow rates through the top flowmeter and bottom flowmeter are set to be the same.

Measurement Uncertainty

The data acquisition unit and thermocouples are compared to a precision digital resistance temperature device (RTD), with 0.03°C rated accuracy, over the range of interest. The temperature measurement system accuracy values for the 0.2-mm thermocouples (on the OHP wall) and for the four probe thermocouples are 0.2 and 0.1°C, respectively. The accuracy of the thermocouple locations in reference to the top of OHP is 2.0 mm. The flowmeters are calibrated using a measuring glass and stopwatch at temperatures of 10, 25, 40, 50, and 60°C. The accuracy of calibrated flow rate is within 3.2×10^{-4} l/s. The calibration result shows that the coolant temperature has negligible influence on the flow rate greater than 0.0095 l/s. The uncertainty of the electrical power input through the power analyzer is 0.5% of reading.

Heat losses are less than 25% of the input power at input powers greater than 500 W. Overall uncertainties for the heat transfer coefficients of evaporator and condenser are estimated and plotted along with the corresponding data in resultant figures presented in this paper.

Results and Discussion

To estimate the heat transfer characteristic of the OHP, mean temperature differences are calculated as follows:

$$\Delta T_{e1} = T_e - T_{a1} \tag{6}$$

$$\Delta T_{e2} = T_e - T_{a2} \tag{7}$$

$$\Delta T_{c1} = T_{a1} - T_{c1} \tag{8}$$

$$\Delta T_{c2} = T_{a2} - T_{c2} \tag{9}$$

where T_e is the mean temperature of the five thermocouples in the evaporator, T_{a1} and T_{a2} are the mean temperatures of the three thermocouples in the upper adiabatic section and those in the lower adiabatic section, and T_{c1} and T_{c2} are the mean temperatures of the four thermocouples in the upper condenser and those in the lower condenser. The heat transfer coefficients with respect to the evaporator and condensers are defined as follows:

$$h_{e1} = Q_1/(A_e \Delta T_{e1}) \tag{10}$$

$$h_{e2} = Q_2/(A_e \Delta T_{e2}) \tag{11}$$

$$h_{c1} = Q_1/(A_{c1} \Delta T_{c1}) \tag{12}$$

$$h_{c2} = Q_2/(A_{c2} \Delta T_{c2}) \tag{13}$$

where A_e is the outer area of the total tubes in the evaporator, A_{c1} the outer area of the total tubes in the upper condenser, A_{c2} the

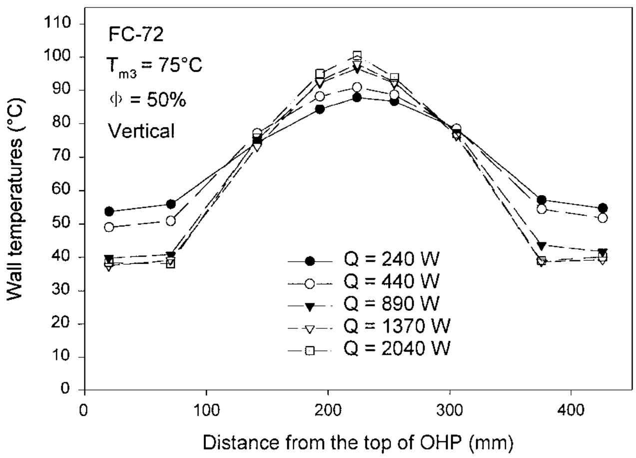

Fig. 3 OHP wall temperature profiles along the middle tube of FC-72 OHP at different total heat rates for T_{m3} = 75°C and ϕ = 50% in vertical operation.

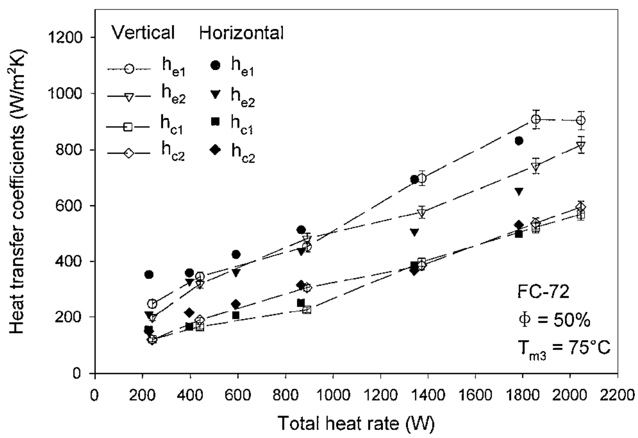

Fig. 4 Heat transfer coefficients of evaporator and condenser vs total heat rate for FC-72 OHP with 50% fill ratio at 75°C operating temperature.

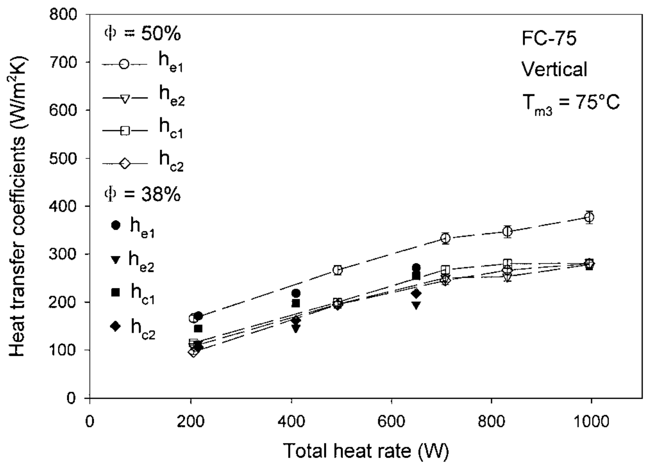

Fig. 5 Heat transfer coefficients of evaporator and condenser vs total heat rate for FC-75 OHP in vertical operation at 75°C operating temperature.

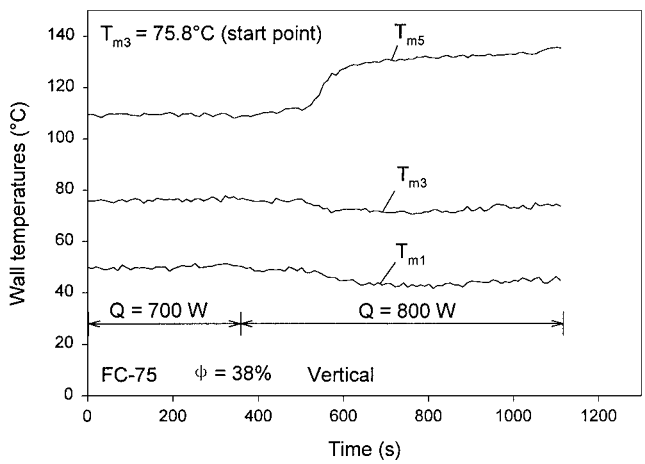

Fig. 6 Excursion of temperatures on FC-75 OHP with 38% fill ratio during dryout.

outer area of the total tubes in the lower condenser, Q_1 the heat rate through the upper condenser, and Q_2 the heat rate through the lower condenser.

For the FC-72 OHP with the fill ratio of 50% vertically operating at a temperature level of 75 and 85°C (T_{m3}), no dryout on any tube wall of the evaporator has been found at total heat rates up to 2040 W. This implies that the performance of the OHP with the 50% fill ratio is high. However, the fluorocarbon OHPs with the 32% fill ratio cannot operate appropriately even at the low input power of 200 W due to the occurrence of the dryout in the evaporator.

Figure 3 shows wall temperature profiles of the FC-72 OHP along the middle tube in vertical operation for $T_{m3} = 75°C$ and $\phi = 50\%$ at different total heat rates. Generally, the wall temperature increases with an increase of the distance up to the evaporator section and then decreases. There are three thermocouples in the evaporator, two in the upper condenser, one in the upper adiabatic section, one in the lower adiabatic section, and two in the lower condenser. The thermocouple indicating the highest temperature is located in the middle part of the evaporator due to a higher heat flux in the middle. The higher the heat rate, the larger the temperature difference is between the evaporator and condenser. The average temperature difference between the evaporator and condensers is 46°C at 2040 W. No temperature excursion has been found at the achievable heat rates up to 2040 W.

Figure 4 gives heat transfer coefficients of the evaporator and condenser as a function of the heat rate for the FC-72 OHP in vertical and horizontal operations for $T_{m3} = 75°C$ and $\phi = 50\%$. The heat transfer coefficients increase with the heat rate. The increase of the heat rate results in an increase of the vapor and liquid velocities in the OHP and intensifies the flow boiling heat transfer and condensation heat transfer. On the other side, the increase of the flow velocities brings about a rise of the flow pressure drop and, consequently, enlarges the mean temperature differences expressed in Eqs. (6–9). In the present case, the former trend is dominant. Figure 4 shows that the differences between h_{e1} and h_{e2} and between h_{c1} and h_{c2} are not significant in vertical operation or horizontal operation. The difference between the results for the vertical operation and those for the horizontal operation is small.

Figure 5 shows heat transfer coefficients of the evaporator and condenser as a function of the heat rate for the FC-75 OHPs with the 50 and 38% fill ratios in vertical operation at the operating temperature of 75°C. The heat transfer coefficients increase with the heat rate. The OHP with the 38% fill ratio has undergone the dryout at input heat rates of 800 W. The heat transfer coefficients of the evaporator are slightly lower with the 38% fill ratio than those with the 50% fill ratio due to insufficient wetting of the evaporator wall. The difference of the heat transfer coefficients of the condenser between $\phi = 38$ and 50% is small. An excursion of temperatures on the OHP with the 38% fill ratio during the dryout is shown in Fig. 6. At the total heat rate of 700 W, the OHP operates appropriately in the steady state. At the total heat rate of 800 W, the temperature in the evaporator T_{m5} starts to continuously increase, and the temperature in the upper condenser begins to drop. The operation of the OHP is deteriorated due to the dryout in the evaporator.

Figure 7 shows heat transfer coefficients of the upper evaporator and condenser vs the operating temperature for the FC-72 OHP with the 50% fill ratio in vertical operation at heat rates of 455 and 1390 W. The heat transfer coefficient of the upper condenser decreases only slightly with the operating temperature. The heat transfer coefficient

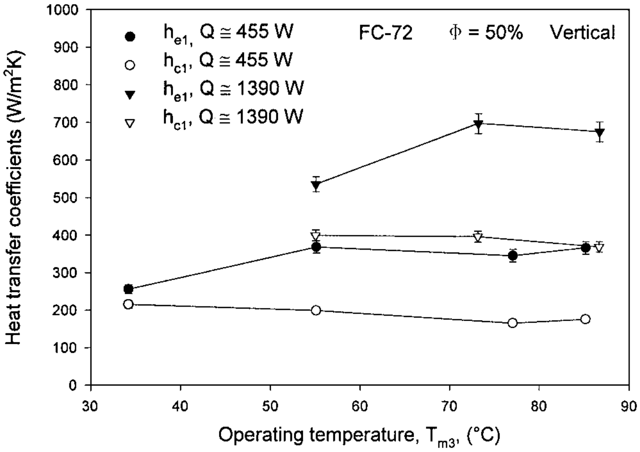

Fig. 7 Heat transfer coefficients of the upper evaporator and condenser vs operating temperature.

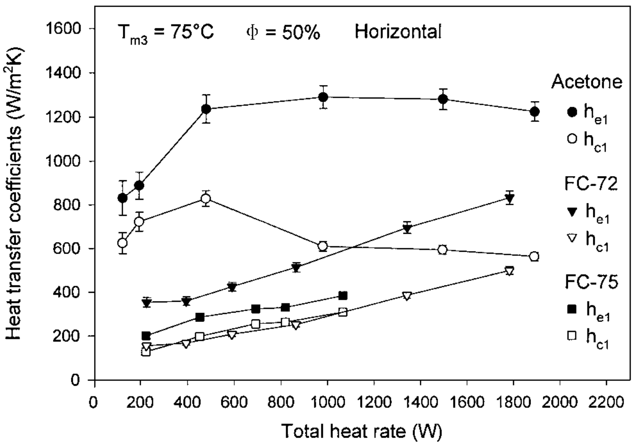

Fig. 8 Comparison of heat transfer coefficients of FC-72 OHP with those of FC-75 OHP and acetone OHP with respect to the upper evaporator and condenser.

of the upper evaporator increases with the operating temperature up to 75°C for $Q = 1390$ W and up to 55°C for $Q = 455$ W. The heat transfer coefficients of the upper evaporator and condenser are much higher for $Q = 1390$ W than those for $Q = 455$ W.

Figure 8 shows the comparison of heat transfer coefficients of the FC-72 OHP with those of the FC-75 OHP and an acetone OHP[5] with respect to the upper evaporator and condenser. The OHPs have the fill ratio of 50% and operate horizontally at $T_{m3} = 75$°C. The heat transfer coefficients of the upper evaporator of the FC-72 OHP are lower than those of the acetone OHP, albeit higher than those of the FC-75 OHP. The heat transfer coefficients of the upper condenser of the FC-72 OHP are lower than those of the acetone OHP and approximately the same as the FC-75 OHP. The difference of the heat transfer coefficients of the evaporator and condenser between the acetone OHP and FC-72 OHP becomes smaller with increasing the heat rate.

Figure 9 shows a comparison between the heat rate through the upper condenser and lower condenser of the FC-72 and FC-75 OHPs with the fill ratio of 50% at various operating temperatures. The heat rate transferred through the upper condenser is approximately the same as through the lower condenser in both vertical and horizontal operations.

Figure 10 shows the comparison of the heat transfer coefficients of the upper evaporator and condenser in vertical position with those in horizontal position for the fluorocarbon OHPs with the 50 and 38% fill ratios at heat rates between 175 and 1850 W and at operating temperatures between 25 and 75°C. The result indicates that there is no significant difference between the heat transfer coefficients of the evaporator and condenser in the horizontal position and those in the

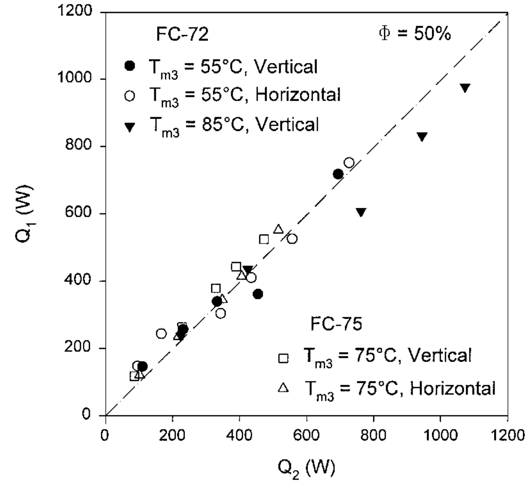

Fig. 9 Comparison between heat rate through the upper condenser (Q_1) and lower condenser (Q_2) of FC-72 and FC-75 OHPs.

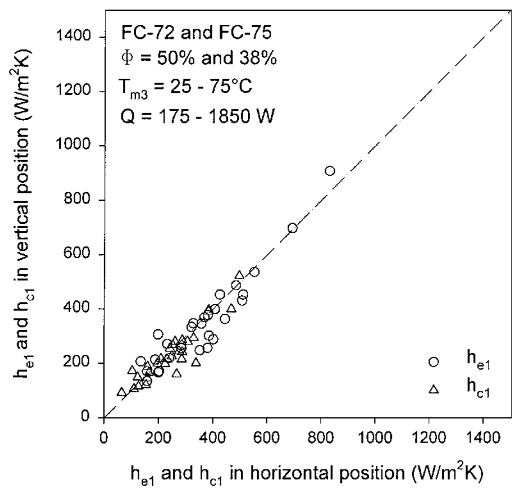

Fig. 10 Comparison of heat transfer coefficients of the upper evaporator and condenser in vertical position with those in horizontal position.

vertical position. The result implies that the thermal performance of the OHPs is insensitive to the gravitational acceleration. The heat transfer coefficients vary widely, from 134 to 907 W/m^2 K for h_{e1} and from 66 to 520 W/m^2 K for h_{c1}, depending on the working fluid and operating condition.

Conclusions

1) The FC-72 OHP with 50% fill ratio has the capability of transporting 2040 W without dryout.

2) The performance limitation of the OHPs is caused by the dryout, which is indicated by the excursion of the temperatures on the OHP wall. The dryout limit for the 50% fill ratio is much higher than that for the 38%. This means that the fill ratio of 38% is not sufficient for the fluorocarbon OHPs. The fluorocarbon OHPs can not operate in the case of the 32% fill ratio, even at the low input power of 200 W.

3) In most cases, the heat transfer coefficients of the fluorocarbon OHPs increase with the heat rate.

4) The heat rate dissipated through the upper condenser is approximately the same as through the lower condenser (in both vertical and horizontal operations). The energy transport of the OHP is attributed to the oscillation of the two-phase flow in the OHP. The tubing turn number of 40 used in the present experiment is sufficiently large to generate the oscillation in the OHP.

5) There is no significant difference between the heat transfer coefficients of evaporator and condenser of the fluorocarbon OHPs in the horizontal position and those in the vertical position.

6) At the same operating temperature, the heat transfer coefficients of evaporator of the FC-72 OHP are lower than those of the acetone OHP and are higher than those of the FC-75 OHP. The heat transfer coefficients of condenser of the FC-72 OHP are lower than those of the acetone OHP and are approximately the same as the FC-75 OHP. The difference of the heat transfer coefficients of the evaporator and condenser between the acetone OHP and FC-72 OHP becomes smaller with increasing the heat rate.

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