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Augmenting the Condenser Heat-Transfer Performance of Rotating Heat Pipes

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A rotating heat pipe assembly was tested at rotational speeds of 700, 1400, and 2800 rpm using distilled water as the working fluid. Tests were made during film condensation on several copper condensers, including smooth-walled cylinders, an internally finned cylinder, and a truncated cone. The truncated cone surface was also promoted for dropwise condensation using n-octadecyl mercaptan in octanoic acid. Heat-transfer performance improved with increasing rotational speed. The internally finned cylinder and the truncated cone showed a 100% improvement over the equivalent smooth-walled cylinder. Dropwise condensation showed substantial improvement over film condensation, primarily at low rotational speeds.

Nomenclature

c_p = specific heat
g_c = gravitational constant
$\dot{m}$ = mass rate of flow
Q = heat-transfer rate
R = radius
ΔT = cooling water temperature difference
ρ = density
σ = surface tension
ω = angular velocity

Introduction

SINCE the inception of the rotating heat pipe concept in 1969,[1] various investigations have been carried out to understand and predict the performance characteristics of this device.[2-6] In particular, such design variables as condenser wall material and taper angle as well as the choice of working fluid have been shown to have a significant effect upon heat-transfer performance. Operating variables such as rotational speed, and noncondensible gas content, also have been shown to be extremely important.

To date, rotating heat pipe condensers have been conical in shape, necessitating a difficult fabrication process. The purpose of this study was to explore the use of less costly condenser sections, and to examine ways to improve upon condenser heat-transfer performance in order to increase the attractiveness of using this device in applications involving rotating machinery.

Experimental Equipment

The apparatus used for this study was similar to the one used in earlier studies[2,5] with several modifications.[7,8] A cross-sectional drawing of the rotating heat pipe is shown in Fig. 1. Figure 2 shows a schematic diagram of the overall experimental equipment, and Fig. 3 is a photograph of the experimental assembly. The heat pipe was rotated using a variable speed motor and a V-belt drive and was supported by two bearings as shown in Fig. 1. The entire heat pipe apparatus was bolted to a steel bed-plate which could be oriented from the horizontal to the vertical position (Fig. 3).

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Index categories: Heat Pipes; Multiphase Flows; Rotating Machinery.

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Evaporator Section

Great care was taken in fabricating the evaporator to insure good dynamic balance as well as a vacuum-tight seal at high rotational speeds. This was particularly crucial since the high heat loads expected in the heat pipe require operation at elevated temperatures.

The evaporator was machined from oxygen-free, high conductivity copper. It was cylindrical in shape with an internal diameter of 74 mm, a length of 91 mm, and a wall thickness of 3.7 mm. As shown in Fig. 1, the inside diameter was stepped at each end to allow room for Viton O-ring seals. Stainless steel end flanges were brazed to this copper section in a hydrogen furnance at 1040°C, using a 35% gold/65% copper brazing alloy. Following this brazing operation, a brazing alloy foil composed of 82% gold/18% nickel was wrapped around the evaporator cylinder. The evaporator was then helically wound with an 11-gage Chromel-A heater wire encased in a 4.8 mm outside diameter Inconel sheath. These heater coils were then brazed to the evaporator surface in a hydrogen furnace at 980°C. The heater was insulated with several layers of Sauereisen cement, and a magnesium-oxide-filled quilted blanket. Electrical power to the heater coil was passed by graphite brushes through bronze collector rings, and was controlled by a solid-state phase amplifier power controller. Glass end windows were used initially to view the boiling, condensing action during rotation, but because of difficulties with repeated cracking, they were replaced with polycarbonate plastic pieces. When using the plastic end windows, 6.4 mm diam copper tubing was threaded into the plastic to serve as the fill line.

Condenser Sections

Four copper condensers were tested during this investigation. Each condenser was about 250 mm long, and was flanged on each end to facilitate attachment to the evaporator and to a cylindrical end plug which was threaded internally to mate with the drive shaft.

In addition to a standard truncated cone with a 2-deg taper angle, such as used in earlier studies,[2,4,5] two smooth cylinders with inside diameters of 25 and 37 mm were chosen, as well as a 25 mm diam, internally finned cylinder which is described below. The truncated cone had a 25 mm long cylindrical section on its small end so that the end plug could be inserted. The inside diameter of this section was 37 mm. This dimension set the size of the larger cylindrical condenser. The dimensions of the smaller cylindrical condenser were chosen to match the size of the internally finned condenser. Condenser wall thickness was 1.6 mm for the cone and large

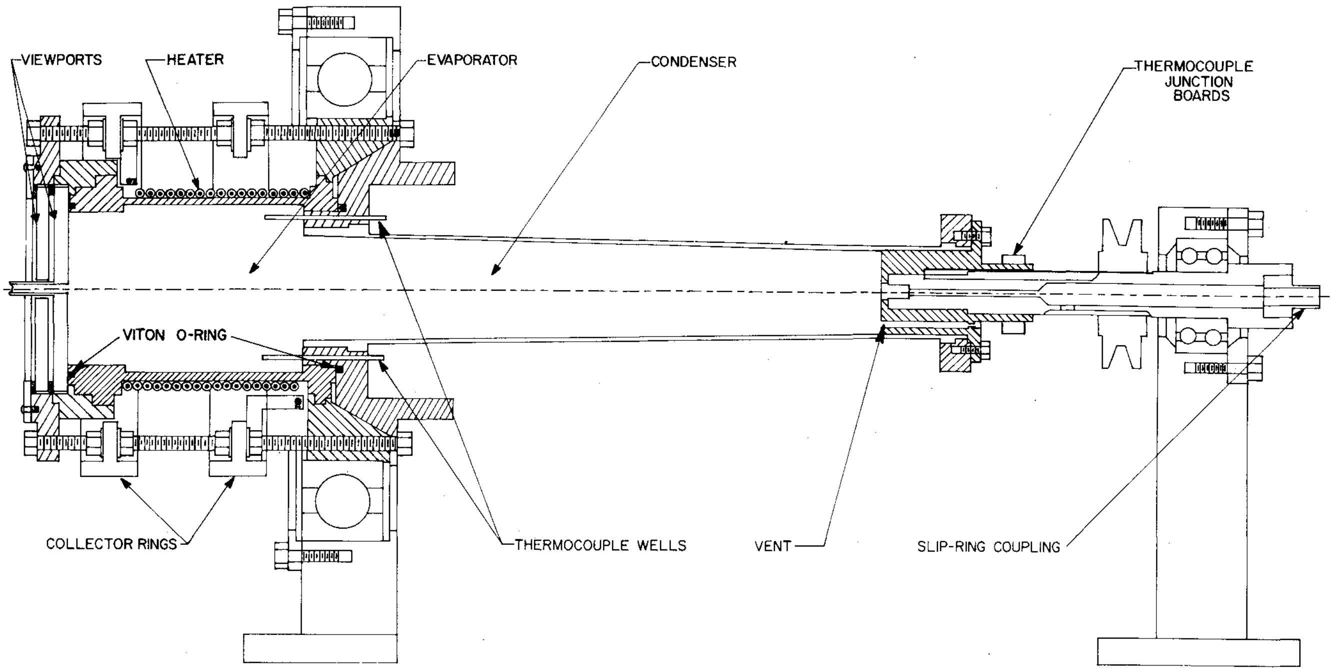


Fig. 1 Cross-sectional drawing of the rotating heat pipe.

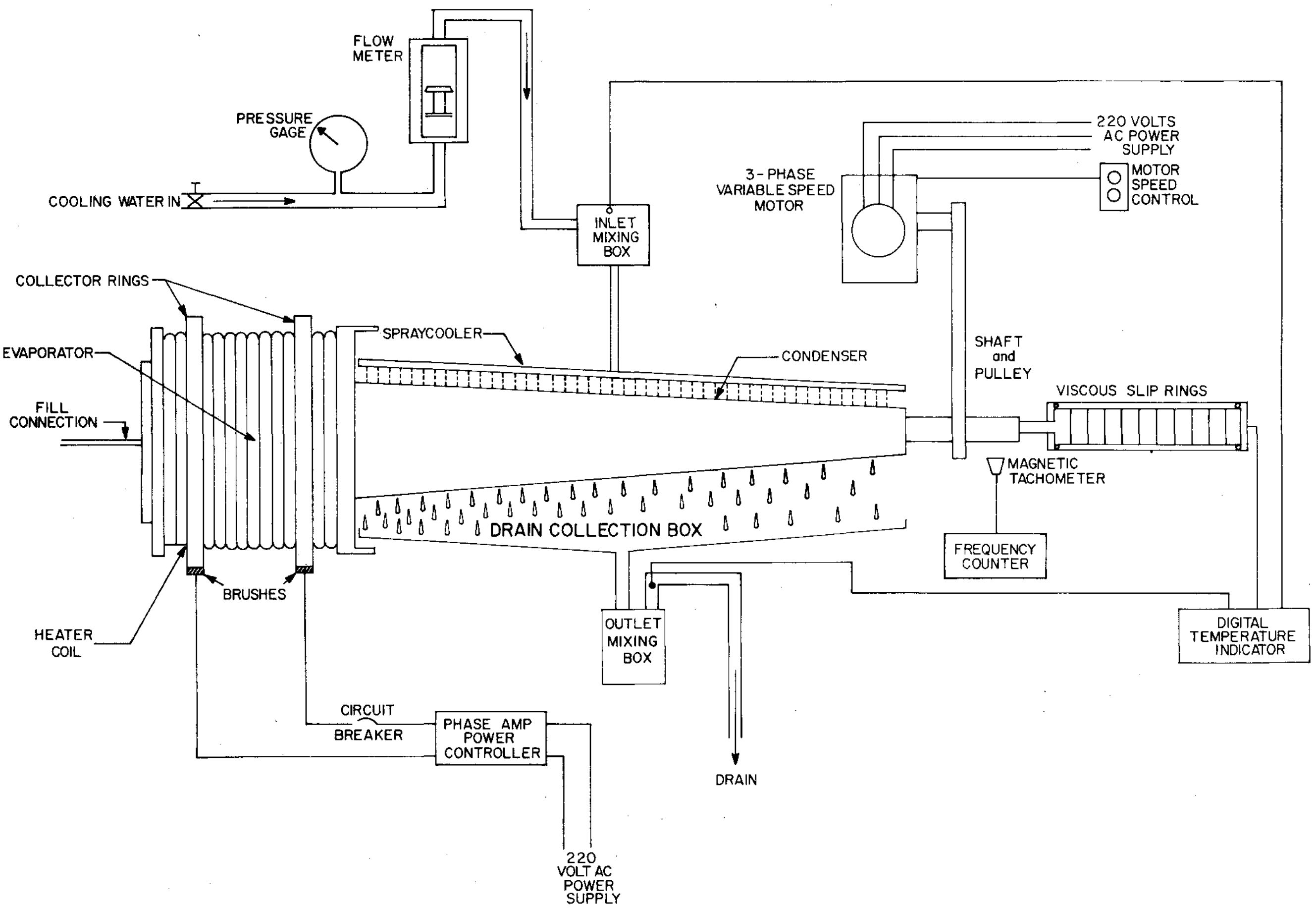


Fig. 2 Schematic diagram of the heat pipe experimental equipment.

cylinder and 0.8 mm for the small cylinder and finned cylinder.

The internally finned condenser was fabricated from a tube furnished by Noranda Metal Industries, Inc. This tubing has been tested in both single-phase forced convection[9] and condensation[10] applications. Tube specifications are given in Table 1, and Fig. 4 is a photograph of a section of the tube, showing the internal fin geometry.

Cooling System

In an effort to flatten the axial wall temperature profile in the condenser section, the spray cooling technique used

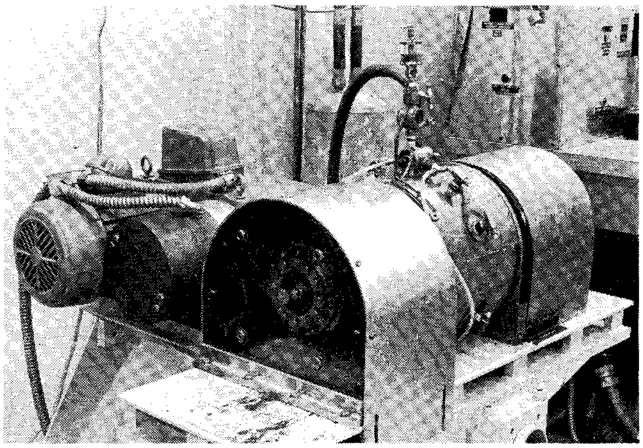


Fig. 3 Photograph of the heat pipe experimental assembly.

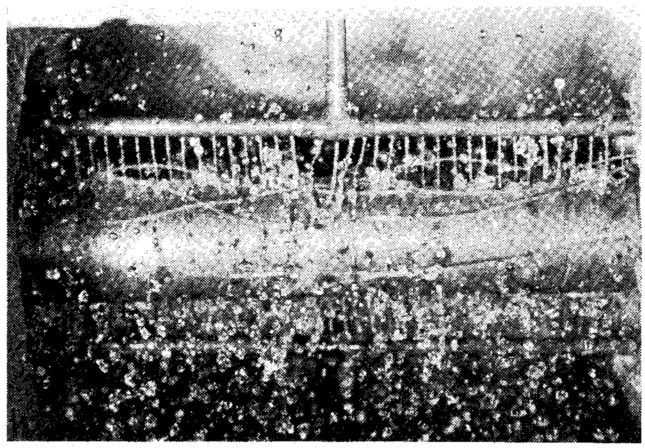


Fig. 5 Photograph of cooling system with heat pipe rotating at 700 rpm.

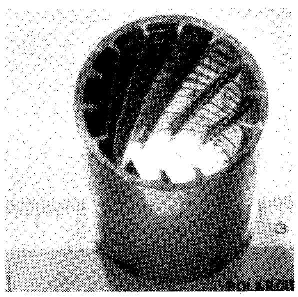
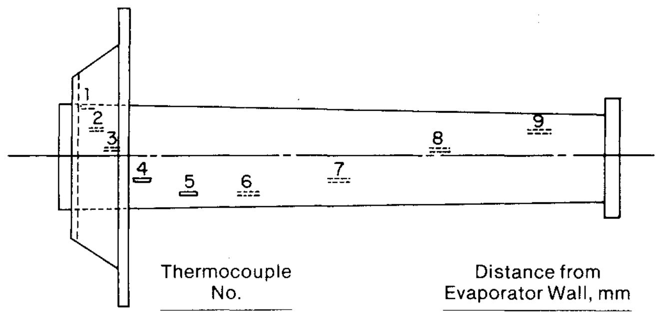


Fig. 4 Photograph of a section of the internally finned condenser.



Thermocouple No.	Distance from Evaporator Wall, mm
1	5.0
2	8.9
3	15.2
4	25.4
5	50.8
6	76.2
7	127.0
8	178.0
9	229.0

Fig. 6 Schematic view of thermocouple positions on the condenser wall.

earlier[5] was replaced with a jet impingement cooling system. This system consisted of four 6.4 mm diam copper tubes with 0.8 mm holes drilled every 5.1 mm along their axes. The supply tubes ran the length of the condenser, parallel to the axis, and were spaced 90-deg apart. Filtered and "softened" tap water was used as the cooling fluid. The cooling system was contained in a sealed and insulated box, as shown in Fig. 3 (with the outer insulation removed). The condenser was rotated inside this box using Garlock spring-loaded rubber seals. Figure 5 shows a photograph of the jet impingement system in operation with the heat pipe rotating at 700 rpm. The top and bottom supply tubes can be seen clearly along with the water jets. These jets breakup into droplets because of the high impingement velocity, the rotational speed of the condenser, and the presence of the thermocouple leads which are running longitudinally along the condenser surface. The third supply tube is hidden from view by the heat pipe, while the fourth supply tube has been removed in order to allow photographs to be taken through the cooling box window.

Instrumentation

All of the temperature measurements on the heat pipe were made with 30-gage, teflon-insulated, copper-constantan thermocouples. Condenser wall temperature measurements were made by placing nine of these thermocouples in small axial grooves along the condenser surface. The locations of these grooves are sketched in Fig. 6. The groove depth was 0.25 mm for the thin-walled condensers and 0.64 mm for the thick-walled condensers. The welded thermocouple beads were soldered into the grooves with a standard lead-tin low-temperature solder. The leads were secured to the condenser by banding with twisted steel wire. Two additional thermocouples were sheathed in 1.6 mm stainless steel tubing. The sheaths protruded 50 mm into the evaporator to measure the vapor temperature and were soldered closed at the tip to maintain the system vacuum. All of the heat pipe thermocouple leads were fed through the small condenser flange, through the drive shaft, to a mercury slip-ring unit. The inlet and outlet temperatures of the cooling water were monitored by Kapton-insulated copper-constantan thermocouples. The outlet thermocouple was actually five separate thermocouples wired in parallel and inserted into the outlet mixing box discharge line (Fig. 2). All of the thermocouples were recorded using a Hewlett Packard data acquisition system, and were calibrated to an accuracy of $\pm 0.5°C$ in a constant temperature bath with a platinum resistance thermometer. Cooling water flow rate was measured with a calibrated rotameter. Accurate regulation of the flow rate was accomplished by adjusting the pressure regulator installed in the cooling line. This regulator, when cleaned, provided flow rates easily maintained to within $\pm \frac{1}{2}\%$ of full scale. Rotational speed was measured using a 60-tooth geared

Table 1 Specifications of the internally finned tube

Number of fins	16
Outside diameter	26.7 mm
Inside diameter	25.3 mm
Fin height	2.1 mm
Unfinned wall area	0.046 m^2/m length
Finned area	0.085 m^2/m length
Fin spacing	3.6 mm
Fin pitch	152.3 mm (left-hand twist)
Total inside surface area/nominal area	1.64

flywheel and a variable reluctance transducer whose output frequency was then counted. An electronic strobe light was also used to provide an independent measurement of rpm.

Experimental Procedures

Preparation of the Condensers

Each condenser surface was treated chemically to insure either filmwise, or dropwise condensation conditions. To obtain reliable wetting of copper by water, a procedure used earlier by Tucker[11] was followed with a few minor modifications.[8] The most essential step of this procedure was the use of a solution of equal parts of ethyl alcohol and a 50% solution of sodium hydroxide warmed to 80°C. Following a scrubbing of the copper condenser surface with this solution, the surface was well wet by water, which formed a continuous film.

The truncated-cone condenser was prepared for dropwise condensation by saturating a swab with n-octadecyl mercaptan ($C_{18}H_{37}SH$), one percent by weight in octanoic acid, and thoroughly swabbing the condenser wall. This promoter was selected because it has been found to be an excellent promoter.[12-14] The condenser was rinsed with distilled water after application of the dropwise promoter, and the surface showed no tendency for wetting.

Filling and Venting Procedures

Prior to filling the heat pipe apparatus with water, the system was evacuated to a pressure of 2.0×10^{-5} Torr. The vacuum pump was then isolated from the heat pipe to insure that a vacuum could be maintained in the assembly. In each run, 250 ml of distilled, degassed water was used as the working fluid. Following the filling procedure, the copper fill tube was crimped twice with vise grips and then sealed with a fast-drying epoxy. No noticeable loss of vacuum was observed during this sealing procedure.

Prior to operation, the heat pipe was vented to the atmosphere to drive off any noncondensible gases. With the heat pipe stationary and tilted at a 30-deg angle (evaporator end depressed), power to the heater was set at 250 W. When the saturation temperature in the evaporator was at least 104°C, the vent was opened. The heat pipe was vented for a minimum of 10 min to allow escaping steam to drive off any air trapped within the system. The pressure in the heat pipe was always maintained above atmospheric pressure by regulating the power to the heater, and adjusting the position of the vent screw.

Data Reduction

During steady-state operation, all of the thermocouples were recorded, along with the cooling water flow rate, and tachometer reading. The condenser heat-transfer rate was determined using the measured values of the cooling water flow rate and overall temperature difference between outlet and inlet:

$$Q = \dot{m} c_p \Delta T \tag{1}$$

A correction was made to this result, however, to take into account frictional heating effects of the bearings and seals, and viscous dissipation effects within the cooling water during rotation.[8]

Results and Discussion

Each of the test condensers was operated at rotational speeds of 700, 1400, and 2800 rpm. Performance curves were obtained wherein the condenser heat-transfer rate was plotted vs measured vapor, or saturation, temperature. Since the condenser heat-transfer rate depends on the overall temperature difference between the vapor and the cooling water, care was taken in all of the experiments to insure that the cooling water inlet temperature was kept at 23 ± 1°C.

Truncated-Cone Condenser

Film Condensation

Figure 7 shows the results of the runs made with the conical condenser and film condensation. Performance obviously improves with increasing rpm. This improvement is due to the increased centrifugal force which increases the axial force component on the condensate, thereby increasing the condensate velocity and flow rate. The increase in centrifugal force also flattens the condensate film, reducing the overall thermal resistance.

Dropwise Condensation

Figure 8 shows the results of the runs made with the conical condenser prepared to promote dropwise condensation.

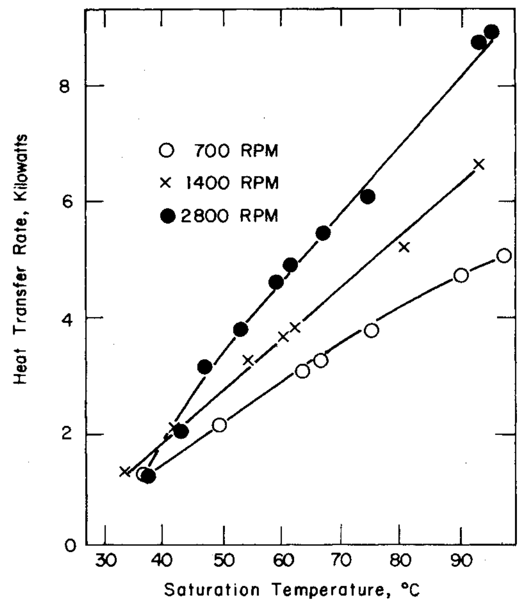


Fig. 7 Heat-transfer rate vs saturation temperature for truncated-cone condenser with film condensation.

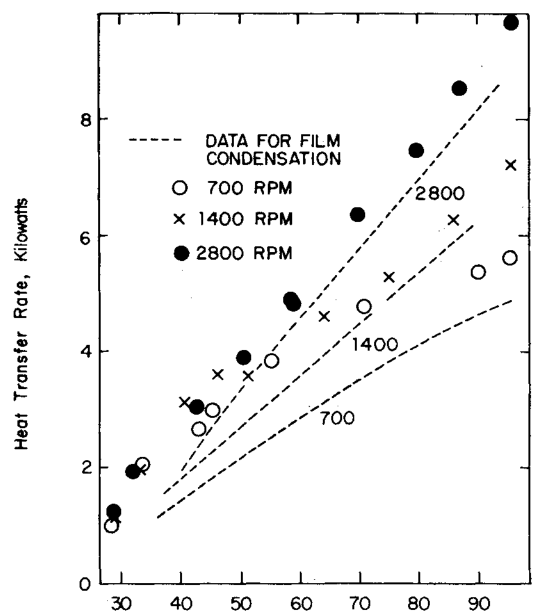


Fig. 8 Heat-transfer rate vs saturation temperature for truncated-cone condenser with dropwise condensation.

Performance obviously improves in comparison to the runs with film condensation, with the greatest improvement occurring at 700 rpm and low saturation temperatures.

It is well known that during dropwise condensation, heat-transfer rates can be increased substantially over those during film condensation.[12] This is because of the growth of thousands of microscopic-sized drops on the surface, causing high heat-transfer. These small drops grow, coalesce with one another, and eventually form into very large drops (~ 1.0 mm in diameter) which are swept off the surface by gravitational forces. The key to enhanced heat transfer during this mode of condensation is to remove these large-diameter drops as quickly as possible, otherwise they will sit on the surface, providing an insulating effect. In the rotating heat pipe, the centrifugal force will sweep the drops along the condenser wall, but because of the small taper angle of only 2 deg, more of the centrifugal force is normal to the condenser wall than tangential to it. Thus, the drops tend to flatten out, spreading out further along the surface, tending to cover the surface with a film. At high rotational speeds such as 2800 rpm, this centrifugal force is 16 times the force at 700 rpm, causing an increased flattening of the droplets, and a tendency toward complete wetting of the wall.

At high saturation temperatures, corresponding to high pressures in the vapor space, the dropwise results appear to approach the filmwise data. This may be because at higher pressures the drops grow faster than at low pressures.[12] It can be postulated, therefore, that in the rotating heat pipe, the drops experience a growth time and a surface sweep time. If the drops grow slowly and are swept along the condenser surface while still small, then heat-transfer is high. On the other hand, if the drops grow quickly in comparison to the surface sweep time, then the drops coalesce into larger drops which tend to flatten out, forming a film.

A further question about the use of dropwise promoters in rotating heat pipes involves the effective lifetime of the promoter itself. A difficulty with all chemical promoters is that they tend to wash off the surface with time as the drops grow and sweep along the surface. The promoter used in this study was chosen because of its ability to adhere to the surface for hundreds of hours.[13-15] Tests during this investigation were carried out only for several hours, so it remains to be seen what long-term performance with this promoter would be like. It might be desirable in the future to operate a condenser with a more permanent promoter such as an ultrathin layer of teflon, or other low surface energy organic coating.

Smooth-Walled Cylindrical Condensers

Obviously, smooth cylindrical condensers are the simplest type of geometry which can be used in a rotating heat pipe, and two such cylinders were tested during this investigation with diameters of 25 mm and 37 mm, respectively. Figures 9 and 10 show the performance curves of these two condensers. The characteristic improvement of heat-transfer rate with rotational speed is clearly evident. The performance of the 37 mm diam cylinder is significantly less than the truncated cone as shown earlier in Fig. 7. This is to be expected because, even though they both have the same small end diameter of 37 mm, in the conical condenser the condenser wall is sloped toward the evaporator, and centrifugal forces accelerate the condensate back to the evaporator section. In the rotating cylinder, however, the condensate flow is induced by a hydrostatic pressure gradient which is established in the condensate as a result of its variable film thickness along the condenser axis. The resulting condensate film thickness is larger than in the case of the conical surface, but respectable heat-transfer rates are still attainable as predicted by Nimmo and Leppert,[16] and Marto.[17] General agreement exists between the measured results and the prediction of Nimmo and Leppert,[16] and is discussed further by Wagenseil.[8]

Results for the 25 mm diam cylinder are less than the 37 mm diam case, as expected. At high vapor saturation temperatures, however, the data appear to merge and may even cross over. According to Nimmo and Leppert,[16] the condensation heat-transfer coefficient is proportional to cylinder diameter to the 1/5 power, so that in going from 25 to 37 mm, the heat-transfer coefficient should increase by 8%. However, in our experimental apparatus, other variables may be affected in going from the smaller to larger diameter cylinders. For example, the cooling water jets may interact differently with the condenser surfaces due to different path lengths from the supply tubes to the actual heat pipe surface. Also, several investigators[18,19] have shown that different fluid flow regimes exist in the liquid film on the surface of a rotating cylinder. These regimes vary with the Weber number, given by

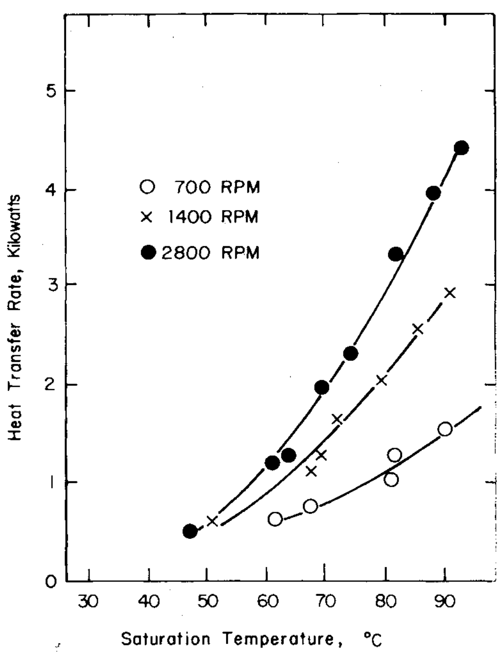


Fig. 9 Heat-transfer rate vs saturation temperature for 25 mm diam cylindrical condenser.

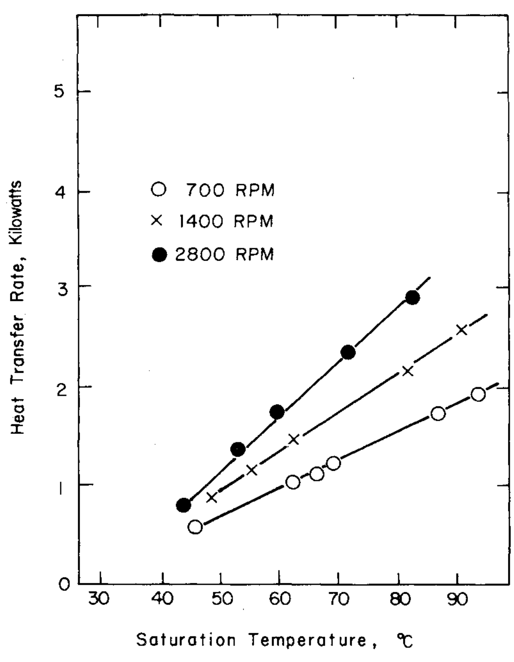


Fig. 10 Heat-transfer rate vs saturation temperature for 37 mm diam cylindrical condenser.

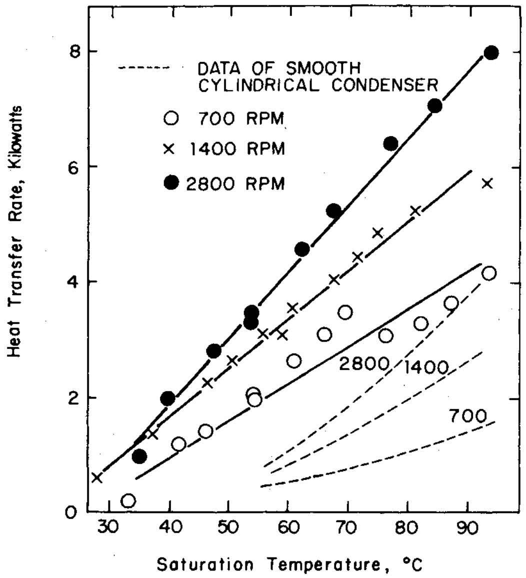


Fig. 11 Heat-transfer rate vs saturation temperature for 25 mm diam internally finned condenser.

$\rho\omega^2 R^3/\sigma g_c$. Clearly then, as cylinder diameter or radius changes, the Weber number changes accordingly, leading perhaps to differences in fluid flow phenomena at the point of contact of the impingement jets. Finally, it may also be that the data taken with the 37 mm diam cylinder were affected by noncondensibles to a slight extent. Further testing with different size cylinders should continue.

Because a hydrostatic pressure gradient is essential in returning the condensate in the case of smooth cylinders, drop promotion was not attempted. It was reasoned that any drops formed on the cylindrical surface would flatten out due to centrifugal force, and would have to coalesce to form a film before the condensate could be returned to the evaporator.

Internally Finned Cylindrical Condenser

Figure 11 compares the results of the 25 mm diam internally finned cylinder to the corresponding smooth cylinder. It is apparent that at high saturation temperatures, an improvement of 100% is possible using the internal fins, and at lower saturation temperatures larger improvements are evident. When these data are compared to the truncated cone data of Fig. 7, it is seen that this internally finned condenser performs as well as the larger-diameter, and certainly more costly, conical surface.

In examining the geometrical properties of this finned tube (Table 1), notice that it has an area ratio of total inside area to nominal area of 1.64. Hence, the heat-transfer improvement results are due to more than just an increase of area. In fact, the spirally finned tube, with a counterclockwise spiral, acts as a condensate pump when the shaft is rotated in a clockwise fashion. The fins therefore act not only as additional condenser surfaces, but also as impellers to force the condensate back to the evaporator. This internally finned cylinder is therefore an excellent surface to be used in rotating heat pipes.

Conclusions

Several clear conclusions can be made based on the experimental results described.

1) In a rotating heat pipe, the use of a conical condenser surface gives superior results over a smooth-walled cylindrical condenser of approximately corresponding size. In particular, the conical surface with a 2-deg taper angle, as used in these tests, has twice the heat-transfer capability of the cylinder.

2) Dropwise condensation improves performance in the conical condenser section. This improvement is most pronounced at low rotational speeds and diminishes at higher vapor saturation temperatures.

3) The internally finned cylindrical condenser shows dramatic improvement over the smooth-walled cylinder of equal diameter. Heat-transfer rates as much as 100 to 200% greater than with the cylinder are possible with the finned condenser.

4) Smooth cylindrical condensers exhibit relatively poor performance when compared to the other geometrical shapes tested. However, their relatively low cost makes their use attractive in rotating heat pipe applications.

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