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An Evaluation of
Liquid and Two-Phase Cooling Techniques
for Use in Electrical Machinery

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However, there still exists a paucity of theoretical analyses and experimental data to enable a designer to successfully implement heat pipes for cooling electric motors. Areas requiring attention are discussed.

SUMMARY

The advantages that can be achieved through the use of advanced techniques for cooling of electric machinery are discussed. A brief literature survey is presented on liquid cooling of rotating machinery, where the liquid is in off-axis rotation. The lack of models to adequately predict pressure drop and heat-transfer performance in this situation has been noted.

The use of heat-pipe cooling appears to be superior to liquid cooling techniques, which require rotating seals with questionable reliability. A detailed literature survey is presented to reveal the advancements achieved through the use of heat-pipe cooling of electric motors. Heat pipes have been successfully used to cool electric motors, resulting in lower and nearly uniform temperatures in rotors and stators. Heat-pipe cooling has been demonstrated to increase the power-to-weight ratio of electric motors by 50 percent or more. In addition, up to 10 percent increase in efficiency has been reported in the literature.

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NOMENCLATURE LIST

A	Surface area
C_p	Specific heat at constant pressure
C_v	Specific heat at constant volume
D	Inside diameter of tube
$\bar{D}$	Average D
f	Friction factor
f_0	Friction factor with no rotation
F	Constant defined in equation (3.13)
Gr	Grashof number = $g\ \beta\ \Delta T\ D^3/\nu^2$
h	Heat-transfer coefficient
h_{fg}	Specific enthalpy of vaporization
H	Radius of rotation
j	Velocity
j^*	Non-dimensionalized velocity, equations (3.5) & (3.6)
k	Thermal conductivity
L	Condenser length
$\dot{m}$	Mass flow rate
Nu	Nusselt number
P	Pressure
Pr	Prandtl number
Q	Heat-transfer rate
R	Universal gas constant
Ra	Rayliegh number (Gr.Pr)
Re	Reynolds number ($\rho\ V\ D/\mu$)
Re_{vL}	Two-phase Reynolds number at X = L
Ro	Rossby number ($V/\omega\ D$)
Sh_L	Sherwood number as defined in equation (3.19)
T	Temperature
ΔT	Wall-to fluid temperature drop
V	Velocity
V^*	Sonic velocity
X	Distance along liquid flow direction

Greek Symbols:

α	Inclination angle
β	Coefficient of thermal expansion
γ	Ratio of specific heats (C_p/C_v)
θ	Half-cone angle
μ	Dynamic viscosity
ν	Kinematic viscosity
ρ	Density
σ	Surface tension of liquid
τ	Interfacial shear stress
ϕ	Angle as defined in Figure 12
ω	Angular velocity

Subscripts:

b	Boiling section
c	Condensing section
cr	Critical value
ℓ	Liquid
L	Evaluated at X = L
m	Mean value
max	Maximum value
s	Constant entropy
sat	Value at saturated condition
v	Vapor
w	Value at wall
0	Value with no rotation

1 INTRODUCTION

1.1 Heat Generation in Electric Motors

An electric motor is a device that converts electrical energy into mechanical energy. Unfortunately, as is the case for any real system, complete conversion of electrical energy into mechanical energy is an impossible task. Instead, some portion of this energy is converted into heat energy. This portion can be as high as 40 percent of the supplied energy depending, to a great extent, on the type (synchronous, induction, etc.) and the size of the motor. The factors contributing to this heat generation are:

1. Mechanical friction losses (at bearings),
2. Air friction losses (at windings),
3. Hysterisis and eddy-current losses,
4. Resistance losses (I^2R losses), and
5. Excitor losses (in the case of synchronous machines).

The mechanical frictional losses occur at bearings which are quite small and are accessible to provide adequate cooling. On the other hand, all other four factors contribute to increasing the temperatures in the rotor and the stator of a motor. Among these four factors, the resistance losses (I^2R) contribute the largest portion.

1.2 Heat Dissipation in Conventional Motors

The dissipation of heat generated in conventional motors occurs through all three heat-transfer mechanisms [1].

1. Conduction:
   * stator to casing
   * rotor to stator (through air gaps)
   * rotor to shaft
   * shaft to casing
   * shaft to external device (coupling, reduction gear, etc.).

2. Convection (both forced and natural):
   * casing to ambient
   * stator to internal air
   * rotor to internal air
   * shaft surface to internal air
   * shaft surface to external air

3. Radiation:
   * external surface to sinks
   * internal surface to casing

Obviously, the presence of all these heat-transfer mechanisms makes a theoretical prediction somewhat impossible. However, the presence of each mechanism desirably contributes to the cooling of the motor. Yet, each mechanism contributes considerably less than the electrical engineers would like to achieve. More effective cooling, with lower motor temperatures, therefore remains to be an important challenge for both electrical and mechanical engineers.

The advantages of effective cooling of electrical motors can be studied by considering the I^2R losses. The electrical resistivity of virtually all metals decreases with decreasing temperature. For example, the resistivity of copper decreases by about 2 percent if the temperature is decreased from 150 °C to 100 °C [2]. Similarly, the resistivity of soft steel will decrease by about 3 percent for the same drop in temperature. In this manner, motors will produce smaller I^2R losses at lower operating temperatures. Note that any decrease in losses contributes to a greater motor efficiency.

Further, the lifespan of insulation materials used in electric machines is very sensitive to hot-spot temperatures. For example, if a hot-spot temperature is decreased from 105 °C to 95 °C, the lifespan of Class A insulation can be, at least, doubled [3].

2 LIQUID COOLING OF MOTORS

2.1 Background

Single-phase liquid (water, for example) cooling can be easily adapted for cooling stators of large pieces of electric machinery. The analysis of liquid cooling in such a stationary system is a fairly well understood subject area. Any basic heat-transfer textbook provides appropriate equations for this analysis, depending on if the flow is either laminar or turbulent. Therefore, no further discussion is provided in this report on cooling of non-rotating systems.

On the other hand, liquid cooling of rotors is considerably more difficult to understand due to the rotation that occurs. Figure 1 shows a schematic of a liquid-cooled rotor. As seen in this figure, the liquid must enter and leave the rotor through rotating seals provided on the shaft. The requirement of such rotating seals raises serious questions from a reliability point of view, as thoroughly discussed by Szatkowski [1]. He also discussed the difficulties associated with the analysis of liquid cooling of rotating parts, owing mainly to the lack of understanding of this phenomenon for both laminar and turbulent conditions. The presence of centrifugal accelerations, Coriolis accelerations and entrance effects makes a complete understanding a very difficult task. Indeed, as discussed by Szatkowski, only a very limited amount of work on this general subject is reported in the literature, and contradictory findings are quite common. The lack of satisfactory experimental data and the existence of contradictory findings are mainly attributable to the very difficult nature of instrumentation of rotating equipment. A brief description of the heat-transfer performance for flow in channels in off-axis rotation is presented below for both laminar and turbulent conditions as well as for the case of flow transition.

2.2 Laminar Flow in Channels in off-Axis Rotation

Figure 2 shows a cross-section of a typical heated channel in off-axis rotation. Owing to the very high centrifugal accelerations possible in such situations, the cooler (denser) liquid is thrown outward, while the warmer (less dense) liquid is forced in the direction of the axis of rotation. In this manner, the centrifugal forces may cause secondary flows, contributing to enhanced heat transfer. However, it must be noted that this enhancement mechanism also makes it very difficult to generate a satisfactory model (theoretical or experimental).

For fully-developed laminar-flow heat transfer, Mori and Nakayama [4] and Woods and Morris [5] found correlations that are in reasonable agreement. As shown in Figure 3, both these findings report enhancement over the fully-developed, non-rotating, laminar Nusselt number of 48/11. This enhancement was attributed to the presence of secondary flow inside the rotating heated channel.

2.3 Turbulent Flow in Channels in off-Axis Rotation

For the case of turbulent flow, Stephenson [6] measured the Nusselt number to be lower than the value predicted by the well-known Dittus-Boelter equation. However, Nakayama [7] showed a slight enhancement over the Dittus-Boelter value. Figure 4 shows the comparison of the Nakayama and Stephenson results with the Dittus-Boelter value. According to Stephenson, turbulence is somewhat suppressed in the presence of high centrifugal accelerations, whereas according to Nakayama, the presence of secondary flow enhances the heat-transfer performance. Despite these two opposite observations, however, it is important to note that these two predictive methods are in close agreement with the

Dittus-Boelter equation. It appears then that the use of the Dittus-Boelter equation may be quite adequate.

2.4 Flow Transition in Channels in off-Axis Rotation

Even for the case of non-rotating channels, there exists no precise Reynolds number that will change the flow condition from laminar to turbulent. Instead, transition may occur at any Reynolds number between 2,000 and as high as 10,000. In view of this fact, a satisfactory understanding of flow transition in rotating channels is an extremely difficult task. The reasons for this difficulty are mainly the lack of sufficient experimental results and the existence of further complicating factors, such as centrifugal and Coriolis accelerations.

Mori et al. [8] studied the flow transition when secondary flows are present in a heated, stationary channel using air as the working fluid. They found that the transitional Reynolds number depends strongly on the inlet turbulence level (they achieved various turbulence levels by using a turbulator). With high inlet turbulence levels, the critical Reynolds number was as low as 2,000 and it increased with heating as given by the following equation (note Ra increases with increased heating):

$$Re_{cr} = 128\ (Re\ Ra)^{0.25} \tag{2.1}$$

where Re, Re_{cr} and Ra are Reynolds, critical Reynolds and Rayleigh numbers, respectively, as defined in the Nomenclature list.

When the turbulence level at the inlet was low, the critical Reynolds number was higher (up to 7,000), and heating decreased this value according to the following equation:

$$Re_{cr} = Re_{cr,0} / (1 + 0.14 \times 10^{-5} \, Re \, Ra) \tag{2.2}$$

where $Re_{cr,0}$ is the critical reynolds number with no rotation.

Figure 5 shows their results. Even though the the above equations reveal interesting results, their applicability to liquid-cooled channels in off-axis rotation is highly questionable. It must be noted that, when considering rotating systems, the 'g' in the Rayliegh number Ra must be replaced by $\omega^2 H$. This substitution may make the Rayleigh number several orders of magnitude greater than for the stationary case. In this manner, the use of these equations for the rotating case requires considerable extrapolation beyond the range of the experimental data, leading to additional uncertainty in the results.

2.5 Pressure Drop in Channels in off-Axis Rotation

Szatkowski [1] summarizes the literature with regard to the friction factor, and hence pressure drop, in radial, curved inlet, and axial sections(see Figure 1) of flow through channels in off-axis rotation. Under these circumstances, Nakayama and Fuzioka [9], using water as the working fluid, developed the following correlations:

$$f/f_0 = 2.2 \, Ro^{-0.33} \; \text{(radial)} \tag{2.3}$$

$$f/f_0 = 1.5 \, Ro^{-0.3} \; \text{(curved inlet)} \tag{2.4}$$

$$f/f_0 = 14.0 \, Ro^{-0.8} \; \text{(axial)} \tag{2.5}$$

where f_0 is the friction factor for the stationary channel and Ro is the Rossby number ($V/\omega D$). In this manner, the friction factor increases with increasing rotational speed.

Further experiments performed by Ito and Nanbu [10] are summarized by Szatkowski [1] and no further discussion is provided in this report.

2.6 Summary

As computed by Szatkowski [1], the heat load that must be handled by each cooling channel may be quite small in the case of electric machinery. In fact, he computed a typical heat duty per channel to be about 50 W, where the tube diameter was 4.8 mm and the tube length was 0.82 m. Based on this reasoning, very high flow rates are not needed in these channels. This is especially important to minimize the power needed to pump the liquid through these narrow channels. In this manner, the flow condition through these channels can be even laminar. Therefore, unlike in the case of stationary heat-transfer equipment, where flow condition is almost always turbulent, the flow condition in liquid-cooled rotating machinery can be laminar. Therefore, more accurate means to predict the heat-transfer and pressure-drop mechanisms for both laminar and turbulent conditions require further attention.

2.7 Advantages/Disadvantages of Liquid Cooling of Electric Machinery

Owing to the superior heat-transfer capability, liquid cooling offers an important advantage over the conventional gas cooling in obtaining cooler machine temperatures. On the other hand, however, liquid cooling has two major disadvantages:

1. It requires cumbersome piping arrangements for liquid flow, and

2. It uses rotating seals of questionable reliability to transport the liquid to and from a stationary source to the rotating system.

3 HEAT-PIPE COOLING OF ROTATING EQUIPMENT

3.1 Advanced Cooling with High Reliability

As thoroughly discussed by Szatkowski [1], successful cooling of rotating equipment can be achieved through the use of heat pipes with a high degree of reliability. This method eliminates the requirement of rotating seals that are needed in the case of liquid cooling. A brief discussion of heat pipes and their applicability as well as their advantages/disadvantages will be discussed in this chapter.

3.2 The Heat Pipe

The heat pipe (or thermosyphon) is basically a hollow, evacuated container partially filled with a working fluid that evaporates at one end (evaporator), while the vapor condenses at the other end (condenser). When the evaporator and condenser sections are properly placed in a heat source and in a heat sink, respectively, this device can transfer heat quite successfully without the need of any external work. Since the heat-pipe operation is associated with two of the most effective heat-transfer mechanisms, evaporation and condensation, heat pipes can be remarkably efficient. The successful operation of heat pipes depends, to a considerable extent, on the ability of condensate to return to the evaporator section. Figure 6 shows the basic geometry of a conventional, capillary heat pipe. As originally conceived [11,12], the transport of condensate in a conventional heat pipe is achieved through the use of a capillary wick (see Figure 6). The condensate can also be transported to the evaporator by means of gravity as shown in Figure 7. However, this arrangement may cause start-up problems since the liquid stratifies on the bottom of the heat pipe when it is not in operation.

Despite their remarkable effectiveness, the major development of capillary heat pipes began only in the 1960s with primary emphasis directed toward aerospace requirements [13]. More details of gravity- or capillary-driven heat pipes are given in [14].

3.3 The Rotating Heat Pipe

In 1969, Hoffmann and Fries [15] recognized the applicability of heat pipes to cool electric machinery, and applied for a patent in Germany which was granted in 1973. Quite remarkably, also in 1969, Gray [16] independently invented the concept and proposed the terminology of the rotating heat pipe in the U.S.. He received a U.S patent in 1974. In a recent paper, Marto [17] provides a detailed discussion of rotating heat pipes.

Figure 8(a) shows the simplest form of a rotating heat pipe which contains no capillary wick. At sufficiently high rotational speeds, the liquid forms a thin annulus (depending on the amount of liquid present), covering the inside wall of the heat pipe. Owing to the evaporation and condensation processes that take place in the evaporator and condenser sections respectvely, the thickness of the liquid layer in the evaporator tends to be smaller than in the condenser section. This difference in thicknesses, together with a sufficiently high centrifugal force, can successfully drive the condensate back to the evaporator to complete the cycle. In this manner, for heat pipes with sufficient rotational speeds, capillary wicks are not needed to transport the liquid from the condenser to the evaporator. Since ordinary liquids have very low thermal conductivities, however, the thickness of this layer should not be too excessive. On the other hand, the thickness should also not be too small to cause dryout in the evaporator section. The use of a conical heat pipe as shown in Figure 8(b) results in a very thin layer of liquid in the crucial condenser part. A slight taper of 1 to 3 degrees allows centrifugal acceleration to pump the condensate back to the evaporator. This results in much thinner films and more effective performance at the condenser section compared to cylindrical heat pipes.

Figure 8(c) shows another rotating heat-pipe arrangement with an abrupt increase in diameter at the beginning of the evaporator section, while the condenser section has a taper. This heat pipe is less sensitive to the amount of working fluid used. Vasiliev [18] proposed the use of a porous cylindrical insert in the evaporator as shown in Figure 8(d). In this case, centrifugal acceleration forces the condensate through the pores within the wall of this insert and the condensate sprays onto the evaporator surface in numerous fine droplets, causing high evaporator heat-transfer coefficients.

Also, two proposed heat-pipe arrangements for radial transport of heat are shown in Figures 8(e) and 8(f). The device shown in Figure 8(e) permits heat to flow from a large cylindrical drum-type surface, whereas Figure 8(f) shows a device configured as a rotating disk-type surface.

Further improvements in rotating heat-pipe technology include the use of axial internal (straight or spiral) fins [19] or internal grooves with increasing depth toward the evaporator section [20]. Since high centrifugal forces allow only an extremely thin layer of condensate on the fins (see Figure 9), internally-finned heat pipes have demonstrated far superior performance than straight, smooth-wall heat pipes. For example, Nefesoglu [19] measured a 400 percent improvement in the condensing-side heat-transfer coefficient of a finned heat pipe (22 straight, axial fins) over a smooth-wall heat pipe using water as the test fluid.

Figure 10 shows a schematic of a rotating heat pipe with internal grooves. Unlike the case of internally-finned heat pipes, in this situation, condensation occurs on the circumferential portions and flows into the grooves.

Vasiliev and Khrolenok [20] developed a theoretical model to predict the performance of these grooved heat pipes and obtained satisfactory results.

As shown schematically in Figure 11, rotating heat pipes can be located on-axis and/or off-axis with the axis of rotation. It is clear that in the case of on-axis heat pipes, the heat-pipe fluid forms an annulus, while in off-axis heat pipes, the liquid occupies a circular segment away from the axis. In this manner, off-axis heat pipes experience non-uniform temperature profiles around their periphery as the liquid occupies only a portion of the heat-pipe cross section. On the other hand, owing to the relatively large radius on which the heat pipe is located, sufficient centrifugal accelerations ($\omega^2 H$) may be present at considerably lower rotational speeds in large equipment with off-axis heat pipes compared to equipment with on-axis heat pipes.

3.4 Heat-Pipe Operating Limits

The thermal performance of rotating heat pipes are subjected to several operating limits. Therefore, a complete understanding of these limits is essential in designing heat pipes. These operating limits are thoroughly discussed by Szatkowski [1], and are briefly described below:

1. Sonic Limit. The vapor velocity in a two-phase closed system is limited to the sonic velocity computed at the operating conditions. The sonic limit (also called the "choking" limit) is represented by the following equation:

$$Q_{max} = \dot{m}\, h_{fg} = \rho_v A_v V^* h_{fg} \qquad (3.1)$$

where V^* is the sonic velocity of the vapor. This limiting velocity may be determined experimentally using the relationship:

$$v^* = \sqrt{\left(\frac{\partial P}{\partial \rho}\right)_s} \tag{3.2}$$

or may be computed using the approximate perfect-gas relationship:

$$V^* \approx (\gamma RT_{sat})^{1/2} \tag{3.3}$$

2. Entrainment Limit. In heat pipes with axial heat transfer, the vapor and liquid flow in opposite directions. Since the specific volume of vapor is much greater than that of the liquid, the vapor velocity can be considerably greater than the liquid velocity (note that the mass flow rates of liquid and vapor are identical at a given axial position and at steady-state conditions). If the vapor velocity is high enough, it may retard the flow of liquid from the condenser section to the evaporator section. If this happens, the evaporator can be "starved," drastically reducing the heat-pipe performance.

In 1969, Wallis [21] derived an expression to express the entrainment limit of heat pipes by balancing inertial and hydrostatic forces:

$$j_\ell^{*2} + C_1\, j_v^{*2} = C_2 \tag{3.4}$$

where,

$$j_\ell^* = \frac{4\, \dot{m}_\ell}{\rho_\ell \pi D^2} \sqrt{\frac{\rho_\ell}{g\, D\, (\rho_\ell - \rho_v)}} \tag{3.5}$$

$$j_v^* = \frac{4\, \dot{m}_v}{\rho_v \pi D^2} \sqrt{\frac{\rho_\ell}{g\, D\, (\rho_\ell - \rho_v)}} \tag{3.6}$$

The constants C_1 and C_2 in equation (3.4) depend on the friction factor at the wall/liquid interface and the friction factor at the liquid/vapor interface. Based on experiments with air and water in an adiabatic, counter-current flow arrangement, Wallis developed the following empirical equation:

$$j_v^{*0.5} + C_3 j_\ell^{*0.5} = C_w \qquad (3.7)$$

For turbulent air-water flows, C_3 is equal to unity, while C_w is 0.7-1.0. However, for liquid-vapor flows (this is the situation that exists in heat pipes), C_w can be greater than unity [22]. In 1973, Sakhuja [23] modified Wallis' correlation for vertical heat pipes using C_w=0.725 to yield:

$$Q_{max} = C_w^2 \frac{\pi D^{2.5}}{4} \frac{h_{fg} \sqrt{g \rho_v (\rho_\ell - \rho_v)}}{[1 + (\rho_v / \rho_\ell)^{0.25}]^2} \qquad (3.8)$$

In 1978, Tien and Chung [24] used a similar approach (to that of Wallis) for rotating heat pipes, recognising that the centrifugal force is the driving force, and developed the following equation:

$$\left[\frac{j_\ell \rho_\ell^{1/4}}{(\sigma D\omega^2 \sin\Theta)^{1/4}}\right]^{1/2} + \left[\frac{j_v \rho_v^{1/4}}{(\sigma D\omega^2 \sin\Theta)^{1/4}}\right]^{1/2} = C_k \qquad (3.9)$$

where D is the largest diameter of the truncated heat pipe, and θ is the half-cone angle. They derived the corresponding maximum heat-transfer rate as:

$$Q_{max} = C_k A_v h_{fg} (\sigma D\omega^2 \sin \Theta)^{1/4} (\rho_\ell^{-3/8} + \rho_v^{-3/8})^{-2} \qquad (3.10)$$

They stated the need for finding C_k based on experimental data. It appears that no experimental data exist in the literature to verify the above two equations. Notice that these equations are not suitable for cylindrical, rotating heat pipes. Thus, data are needed for both conical and cylindrical rotating heat pipes.

Nguyen-Chi and Groll [25] present a detailed discussion of other correlations found in the literature primarily for stationary heat pipes or thermosyphons. They extended the above correlations for various inclination angles ($1° \le \alpha \le 80°$) using the following empirical equation based on their experimental data:

$$f(\alpha) = \left[\frac{\alpha}{180} + \sqrt{\sin 2\alpha} \right]^{0.65} \tag{3.11}$$

They recommended the use of $f(\alpha)$ as a multiplier for equations (3.7) and (3.8). Their data agreed quite well with equation (3.7).

Based on the above discussion, the Wallis' correlation (equation (3.4)), together with Nguyen-Chi and Groll correlation (equation (3.11)), can be used for both vertical and inclined stationary heat pipes. It appears that the use of these correlations for the case of rotating heat pipes is highly questionable. As also stated by Nguyen-Chi and Groll, the entrainment limit can be quite low for long, narrow heat pipes. On the other hand, the presence of high centrifugal accelerations may increase the relative velocity, between vapor and liquid, at which the entrainment will take place. This may be especially true in the case of off-axis heat pipes.

3. <u>Boiling Limit</u>. In the case of capillary heat pipes, the boiling limit occurs when nucleate boiling occurs in the evaporator wick section. Since the bubble formation severely disrupts the replenishment of the boiling section with new liquid, the on-set of nucleate boiling must be avoided. In the case of wickless heat pipes, the heat-

transfer capacity of the evaporator section depends on the area wetted by the working liquid and the critical heat flux. The area wetted by the working liquid is most crucial in the case of off-axis heat pipes. Thus, keeping all other variables (such as wall temperature, operating pressure, etc.) the same, a given surface area covered by the working liquid would transfer only a limited amount of heat. Therefore, heat pipes must be provided with the correct amount of working liquid to satisfy the heat-transfer requirement. In this situation, the Zuber-Kutateladze prediction [26] can be used to compute the boiling limit:

$$Q_{max} = 0.13\, A_b\, \rho_v^{1/2}\, h_{fg}\, [\omega^2 H\, (\rho_\ell - \rho_v)\, \sigma]^{1/4} \tag{3.12}$$

This equation shows that Q_{max} increases with increasing boiling surface area, A_b.

4. Condensing Limit. As in the case of the boiling limit, the condensing limit depends on many parameters. Note that in the case of off-axis heat pipes, the liquid stratifies in a circular segment away from the center of rotation, resulting in a very thin film in the other circumferential portion of the tube (see Figure 11). On the other hand, in the case of on-axis heat pipes, the liquid takes an annular form. In this manner, for given fluid and operating conditions, condensation heat transfer depends mainly on the area that is exposed to vapor in the case of off-axis heat pipes, or on the amount of liquid fill in the case of on-axis heat pipes. In other words, the condensing heat-transfer limit decreases with increasing liquid fill. Collier [27] presents a correlation to compute the condensing limit for a stationary horizontal cylinder, where the liquid stratifies on the bottom:

$$Q_{max} = A_c\, \Delta T\, F \left[\frac{\rho_\ell\, (\rho_\ell - \rho_v)\, g\, h_{fg}\, k_\ell^3}{D\, \mu_\ell\, (T_{sat} - T_w)} \right]^{1/4} \tag{3.13}$$

where F depends on ϕ (see Figure 12). A listing of F is

Table 1
Values of F for Laminar-Film, Stratified Condensation in a Horizontal Tube [27]

ϕ (°)	F
0	0.725
10	0.712
20	0.689
30	0.661
40	0.629
50	0.594
60	0.557
70	0.517
80	0.476
90	0.433
100	0.389
110	0.343
120	0.296
130	0.248
140	0.199
150	0.150
160	0.100
170	0.050
180	0.0

provided in Table 1. Note that, with g replaced by $\omega^2 H$, the above equation is valid only for off-axis heat pipes. In the case of on-axis heat pipes, the film thickness should be made as small as possible to maximize condenser performance. This can be achieved in a number of ways: 1. use a truncated cone condenser with a larger diameter at the evaporator end, 2. use a stepped cylinder (with sudden increase in diameter at the beginning of the evaporator), 3. use an internally-finned condenser (with axially-straight or helical fins), or 4. use an internally-grooved condenser. Marto [17] presents a detailed discussion of the appropriate models for the first two cases. For example, for rotating truncated-cone condensers, Ballback [28] performed a Nusselt-type analysis, and obtained the following equation for the average Nusselt number:

$$Nu = \frac{h_m L}{k_\ell} = 0.904 \left[\frac{\omega^2 \; L^2 \; R_o^2}{\nu^2} \; \frac{Pr}{C_p \; (T_{sat} - T_w) \; / \; h_{fg}} \right]^{1/4} G(\psi) \quad (3.14)$$

where,

$$G(\psi) = \frac{\{(1 + \psi)^{8/3} - 1\}^{3/4}}{\sqrt{\psi} \; (2 + \psi)} \quad (3.15)$$

and

$$\psi = \frac{L \; \sin \Theta}{R_o} \quad (3.16)$$

In deriving equation (3.14), Ballback made three major assumptions: 1. isothermal wall condition, 2. negligible interfacial vapor shear, and 3. negligible vapor pressure drop. Later, Daniels and Al-Jumaily [29,30] extended Ballback's analysis. They assumed that $d\delta/dx$ was much smaller than $\tan(\theta)$, and included the effect of vapor drag, while neglecting vapor-pressure drop. Assuming further constant-wall-temperature conditions, they obtained the following expression:

$$Nu = \frac{h_m L}{k_\ell} = \frac{4}{3} \; Sh_L \left(\frac{\delta_L}{L} \right)^3 - \frac{1}{2} \; Pr_L \left(\frac{\delta_L}{L} \right)^2 - \frac{1}{2} \; Re_{vL} \left(\frac{\delta_L}{L} \right) \quad (3.17)$$

where,

$$Pr_L = \frac{\rho_\ell \; \tau_v \; h_{fg} \; \gamma^2 \; \cos \Theta}{\mu_\ell \; k_\ell \; (T_{sat} - T_w)} \quad (3.18)$$

$$Sh_L = \frac{\rho_\ell^2 (\omega^2 \bar{D}/2 - g) h_{fg} X^3 \sin \Theta}{4 \mu_\ell k_\ell (T_{sat} - T_w)} \tag{3.19}$$

$$Re_{vL} = \frac{\rho_\ell V X \cos \Theta}{\mu_\ell} \tag{3.20}$$

If vapor drag is neglected, for small values of θ, Marto [17] has shown that equation (3.17) reduces identically to equation (3.14). The literature appears to lack a model to adequately predict the performance of internally-finned rotating condensers though Nefesoglu [19] reported data taken on various internally-finned rotating condensers. He showed up to 400 percent improvement with 22 straight fins compared to the performance of a smooth condenser. Vasiliev and Khrolenok [20] present a somewhat complicated model (not presented here for simplicity) for the analysis of internally-grooved rotating condensers. They showed very good agreement between their data and the model.

5. Condenser-to-Ambient Heat-Exchanger Limit. If the eventual heat transfer from the condenser of the heat pipe is to the ambient, careful consideration must be given to the design of this heat-transfer process because of the poor heat-transfer performance associated with air. Therefore, the use of external fins in the condenser section is highly desirable to improve performance. Also, if practical, ambient or chilled water cooling is most desirable. The heat-transfer performance of rotating heat pipes with external fins in the condenser section is a poorly-understood subject area. Indeed, literature reveals no correlations to predict the performance of such circular fin arrays undergoing off-axis rotation.

Appropriate correlations for heat transfer from stationary single tubes or multi-tubes to gases in cross flow are presented in [31] for both smooth and externallyfinned tubes. Tube bundles include both in-line and staggered arrangements. Different correlations exist for high fins and low fins. These correlations take fin geometry, such as fin spacing, fin height and fin thickness, into consideration. The correlations provided in [31] are not presented here mainly due to the fact that their validity is highly questionable for rotating finned tubes or fin arrays. Especially, in the case of off-axis heat pipes, each tube is in the wake of another tube.

The use of segmented fins (in both on-axis and off-axis heat pipes) and the use of water cooling (only with on-axis heat pipes) appear to offer effective cooling, thus warranting systematic experimental investigations.

3.5 Summary

Marto [17] provides a qualitative representation of the first four heat-pipe limits as shown in Figure 13. The exact locations of these curves depend on a large number of parameters as described above, such as heat-pipe geometry, type of working fluid, the presence of non-condensing gases in the vapor space, heat-pipe rotational speed, etc.. It is important to note that as these conditions change, the limit curves will shift in relation to one another, resulting in different limitations on thermal performance. In general, however, for ordinary geometries and working fluids, the condenser end appears to limit the overall heat-transfer performance of the heat pipe. Further, if air cooling is used for the condenser-to-ambient heat-exchange process, this may result in the controlling limit.

Using an off-axis heat pipe, Szatkowski [1] estimated the heat-pipe limits (except the fifth one listed above) for a pipe of 4.6 mm in inside diameter and 0.82 m in length located on a radius of 0.4 m. Using water as the heat-pipe fluid, he plotted the heat-pipe limits versus the percent liquid fill at a rotational speed of 3,600. Figure 14 shows these plots. It can be seen that for his example, the dominant limit occurs at the condenser end.

Marto [17] presents a detailed discussion on the available models that can be used to compute the performance of on-axis, rotating evaporators and condensers. Therefore, no discussion is provided in this report. Unfortunately, there exist no adequate models for the analysis of off-axis, rotating heat pipes, and only limited experimental data exist [32,33].

3.6 General Applications of Rotating Heat Pipes

Heat-pipe cooling can be a highly viable candidate for almost all situations where there is a need to transport heat from a rotating component. Gray [16,34] described various applications including a scheme to cool high-speed drills as shown in Figure 15. In this scheme, the working fluid transfers heat axially from the hot cutting end of the drill bit to a region along the shaft, where heat is removed by a coolant flowing through an annular gap. Vasiliev [35] described various energy recovery applications that occur in gas turbine engines, rotating drum-type dryers, and centrifugal systems used for heating granular materials or pasteurizing milk and other food products.

As stated by Marto [17], cooling of electric motors and generators is probably the most promising application of rotating heat pipes. As discussed earlier, effective cooling of electric machines makes it possible to improve the efficiency as well as the power-to-weight ratio, resulting in compact machines. The use of both on-axis and off-axis heat pipes for cooling electric machinery is discussed in detail in the next chapter.

4 HEAT-PIPE COOLING OF ELECTRIC MACHINES

4.1 Literature Review

In 1970, Fries [36] used a hollow-shaft, rotating heat pipe to cool electric motors. He reported that the use of the shaft-mounted heat pipe resulted in a 35-K drop in the rotor temperature. Since then, he and his co-workers have received several patents on this concept in the Federal Republic of Germany [37,38]. Figure 16 shows a schematic of his concept.

In 1970, in the U.S., Corman and McLaughlin [32] reported on the performance of a heat-pipe-cooled (both stator and rotor), 10-HP, induction motor with promising results. They cooled the stator with stationary, capillary-wick-type (fine copper mesh) heat pipes and the rotor with off-axis heat pipes with no wicks. They installed 12 thermocouples in the stator and seven thermistors in the rotor. They varied the rotor RPM from 0 to 12,000 and showed considerable reductions (up to 100 K) in the rotor and stator temperatures when the RPM was increased from about 1,200 to about 12,000. Unfortunately, they did not measure the temperatures without the heat pipes in operation (i.e., by removing the working fluid from the heat pipes) for comparison purposes, nor did they do any testing using on-axis heat pipes. This work appears to reveal the only rotating heat-pipe-cooled motor data that have been reported in the U.S. open literature.

In 1977, Oslejsek and Polasek [39] reported experimental results obtained on an electric motor containing an on-axis heat pipe. Their heat pipe had a conically-shaped condenser section and a stepped evaporator as shown in Figure 17. They provided external fins on the condenser section. They measured rotor temperatures at three axial locations with and without the heat pipe in operation. As can be seen in Figure 17, the use of the heat pipe served two important goals: 1. it lowered the rotor temperatures, and 2. it produced a nearly-uniform temperature profile in the axial direction.

Kukharsky, as stated by Vasiliev et al. [40], obtained a 5- to 10-percent increase in efficiency and an increase in power of 1.5 to 2 times by the use of shaft-mounted heat pipes in motors of 1.5- to 7-kW capacity. Further, they stated that dozens of electric machines with heat-pipe cooling were tested primarily in the USSR and Czechoslovakia, and they cited 9 references (no English translations of these references apparently exist).

Chalmers and Herman [41] fitted a heat pipe in the shaft of an estimated 15-20 kW (authors did not specify this value) induction motor. They measured the outer surface temperature of the heat pipe at three locations: the mid point and the two ends. Figure 18 shows the measured temperatures at two rotor speeds: 1030 and 1470 RPM with heat losses of 800 and 129 W, respectively. This figure shows a considerable temperature drop with the heat pipe in operation. Not only are the measured temperatures up to about 100 K lower, but also the presence of the heat pipes resulted in a uniform temperature profile in the axial direction. Schneller et al. [42] provided heat pipes both in the rotor and in the stator of various electric motors. They showed that winding temperatures decreased by as much as 30 K. By keeping the same hot-spot temperatures (as in the case without heat pipes), a 10-kW motor measured a performance increase of about 25 percent. They cite 13 references dealing with heat-pipe cooling of motors; again, no English translations of these references seem to exist.

Thoren [43] tested two simulated induction motors with heat-pipe cooling. He provided 36 capillary heat pipes with internal threads in the stator. In addition, 28 bars of the squirrel cage were constructed as heat pipes (presumably with no capillary wicks) that were in an off-axis rotation. While he provided external fins on the condenser sections of all stator heat pipes, the rotor heat pipes were unfinned. In order to avoid performance degradations possible with even slight inclinations, he provided the condenser section for each heat pipe only on one end of the motor, but each motor end shared an equal number of condensers. The evaporator sections ran the entire length of the rotor or the stator. He found as much as a 60-K temperature drop in the windings. He reported data for up to 500 RPM, for which the rotor temperatures were nearly constant in the axial direction, revealing successful cooling. Sattler and Thoren [44] tested a 75-kW, totally-enclosed, variable-speed (0-5,000 RPM) induction motor with a 27.5-mm-thick (in the evaporator section) heat pipe in the shaft. They used both water and methanol as the working fluid. While they reported on dryout with methanol, the use of distilled water decreased the rotor temperatures by about 115 K. The presence of a rotor heat pipe decreased even the stator temperatures by as much as 30 K near rated power. Also, by maintaining the maximum allowable insulation temperature, they achieved a 17-percent increase in power. Brost et al. [45] simulated a 138-kW, variable-speed (0 to 2,660 RPM) DC motor using a prototype hollow shaft, which contained a conically-shaped condenser section. They took data with various fill charges and various half-cone angles in the condenser section and found an optimum angle of 2.8 degrees. They successfully avoided steel-water-compatibility problems by copper plating the interior of the hollow steel shaft. Groll et al. [46] also simulated a 150-kW induction motor (0 to 5,000 RPM). They provided successful cooling at all operating conditions.

Furuya [47] provided a shaft-mounted heat pipe in a motor of a vertical machine tool. Owing to the lower temperatures, the temperature deformations (the errors resulting from the thermal expansion) were decreased by about 75 percent in the most crucial directions (i.e., in the directions perpendicular to the cutting tool). As reported by Koizumi [48], the Furukawa Electric Company of Japan manufactures commercial motors (AC and DC) with heat pipes located in the shaft. He emphasized the need of heat-pipe cooling of totally-enclosed motors.

Xianzhi et al., as stated by Tongze and Zhangyan [49], fitted four heat pipes in the stator and a shaft-mounted heat pipe in a 3.2-kW DC servomotor. They measured a 50-percent increase in the motor output torque and an increased output power to 6.5 kW. They also noticed that the motor temperature rise above ambient decreasd from 80 K to 60 K.

4.2 Summary

It appears that the application of heat-pipe cooling in electric motors can result in significant reductions in the size and weight of motors as well as in significant improvements in motor performance. This is especially true in the case of totally-enclosed motors, where conventional cooling is very ineffective, and in the case of variable-speed motors, where the majority of heat generation takes place in the rotor.

4.3 Advantages/Disadvantages of Heat-Pipe Cooling of Electric Motors

Heat-pipe cooling has several important advantages over the use of liquid cooling, which is considerably more efficient than conventional gas cooling:

1. It does not require rotating seals.

2. It is more efficient than convective liquid cooling owing to two of the most efficient heat-transfer mechanisms (condensation and evaporation).

3. The two-phase medium is lighter than the all-liquid

medium used in the case of liquid cooling.

On the other hand, heat-pipe-cooled machines may suffer a small disadvantage over liquid cooled or conventional machines with some start-up problems. This is due to the fact that the liquid in the partially-filled heat pipes will stratify on the lower portion of the heat pipes when the machine is not in operation, thus slightly shifting the center of gravity of the entire rotor in the vertically downward direction. Further, both heat-pipe cooling and liquid cooling require considerable complications added to the rotor and/or stator owing to the presence of heat pipes or liquid-flow passages. This is probably the only disadvantage of these two cooling techniques over conventional cooling.

5 CONCLUSIONS AND RECOMMENDATIONS

5.1 Conclusions

Based on the discussion presented in this report, the following conclusions can be made:

1. Effective cooling will result in significant improvements in power-to-weight ratio and longer insulation lifespan of electric machines.
2. Heat-pipe cooling offers the most effective cooling since it uses two of the most efficient heat-transfer mechanisms (condensation and evaporation).
3. Stators can be cooled by stationary heat pipes, while the rotors can be cooled by rotating heat pipes.
4. On-axis heat pipes appear to be best for small to moderate electric machines (0 to 150 kW).
5. Off-axis heat pipes appear to be best for large machines (≥ 150 kW).
6. Moderate (30-150 kW) machines may be successful with either on-axis or off-axis heat pipes.
7. A need exists to develop reliable models that can be used to design liquid-cooled or heat-pipe-cooled electric machines.
8. Liquid cooling may result in considerable temperature reductions in electric motors.
9. The reliability of rotating seals needed in liquid cooling raises serious questions about reliability.
10. Liquid cooling may be suitable for large, low-speed machines, where rotating seals can be made sufficiently reliable.

5.2 Recommendations

1. Conduct an experimental investigation to model the entrainment limit associated with heat pipes in off-axis rotation.

2. Test various internal geometries (such as internally finned, or grooved) to obtain improved cooling on actual electric motors.

3. Conduct an experimental investigation to model the condenser-to-ambient heat-exchanger performance of fin arrays undergoing off-axis rotation. Test various fin geometries using air as the heat-transfer fluid.

4. Develop a computer code to design heat-pipe-cooled electric machines.

5. Conduct experimental investigations to generate reliable correlations to predict the transition from laminar to turbulent flow in pipes in off-axis rotation.

6. Generate correlations to predict heat transfer and pressure drop in both laminar and turbulent flow in pipes in off-axis rotation.

6 REFERENCES

1. Szatkowski, J. L., "Cooling of High Power Generators and Motors for Electric Propulsion," M.S. Thesis, Naval Postgraduate School, Monterey, California, March 1984.

2. Handbook of Tables for Applied Engineering Science, 2nd Edition, pg. 224.

3. Military Standadization Handbook, Section 2.7.

4. Mori, Y. and Nakayama, W., "Forced Covective Heat Transfer in a Straight Pipe Rotating Around a Parallel Axix (1st Report, Laminar Region)," Int. J. Heat Mass Trnsfer, 1967, V. 10, pp. 1185-1201.

5. Woods, J. L. and Morris, W. D., "An Investigation of Laminar Flow in the Rotor Windings of Directly-Cooled Electrical Machines," Journal of Mech Engrg. Science, 1974, V. 16, No. 6, pg. 408.

6. Stephenson, P. L., "An Experimental Study of Flow and Heat Transfer in a Duct Rotating about a Parallel Axis," ICHMT 14th Symposium, 1982, Dubrovnik, Yugoslavia.

7. Nakayama, W., "Forced Convective Heat Transfer in a Straight Pipe Rotating around a Parallel Axis (2nd Report, Turbulent Region)," Int. J. Heat Mass Transfer, 1968, V. 11, pp. 1185-1201.

8. Mori, Y., Futagami, K., Tokuda, S. and Nakamura, M., "Forced Convective Heat Transfer in Uniformly Heated Horizontal Tubes," Int. J. Heat Mass Transfer, 1966, V. 9, pp. 453-463.

9. Nakayama, W. and Fuzioka, K., "Flow and Heat Transfer in the Water-Cooled Rotor Winding of a Turbine Generator," IEEE Trans. on Power Apparatus and Systems, 1978, V. PAS-97, No. 1, pg.225.

10. Ito, H. and Nanbu, K., "Flow in Rotating Straight Pipes of Circular Cross Section," J. of Basic Engineering, Sept. 1971, pp. 383-394.

11. Gaugler, R. S., "Heat Transfer Device," U.S. Patent 2,350,348, June 6, 1944.

12. Grover, G. M., Cotter, T. P. and Erickson, G. F., "structures of Very High Thermal Conductivity," J. Appl. Phys., V. 35, pp. 1190-1191, 1964.

13. Midwest Research Institute, "Heat Pipes," NASA-CR-2508, 1975.

14. Chi, S. W., THeat Pipe Theory and Practice, Hemisphere, Washington, D.C., 1976.

15. Hoffmann, M. and Fries, P., "Cooling the Rotor of an Elecric Motor," Patent Application No. 1,900,411, Federal Republic of Germany.

16. Gray, V. H., "The Rotating Heat Pipe - A Wickless, Hollow Shaft for Transferring High Heat Fluxes," ASME Paper No. 69-HT-19.

17. Marto, P. J., "Rotating Heat Pipes," Heat and Mass Transfer in Rotating Machinery, Eds.: D. E. Metzger and N. H. Afgan, Hemisphere Publishing Corporation, 1984, pp. 609-632.

18. Vasiliev, L. L., "Heat Exchangers based on Heat Pipes," Nauka i Technika, Minsk, pp. 86-87 (in Russian).

19. Nefesoglu, A., "Heat Transfer Measurements of Internally Finned Rotating Heat Pipes," M.S. Thesis, Naval Postgraduate School, Monterey, California, December 1983.

20. Vasiliev, L. L. and Khrolenok, V. V., "Study of a Heat Transfer Process in the Condenser Zone of Rotating Heat Pipes," J. Heat Recovery Systems, V. 3, No. 3, pp. 281-290, 1983.

21. Wallis, G. B., One-Dimensional Two-Phase Flow, McGraw-Hill, New York, 1969.

22. Shirer, G. L. and Pickering, A. R., "The Flooding Phenomenon in Counter-Current Two-Phase Flow," Proceedings Symposium on Two-Phase Flow, Exerter, 2, pp. B.501-B.538, 1965.

23. Sakhuja, R. S., "Flooding Constraint in Wickless Heat Pipes," ASME Paper 73-WA/HT-7.

24. Tien, C. L. and Chung, K. S., "Entrainment Limits in Heat Pipes," 3rd International Heat Pipe Conference, Palo Alto, California, May 22-24, 1978, pp. 36-40.

25. Nguyen-Chi, H. and Groll, M., "Entrainment or Flooding Limit in a Closed Two-Phase Thermosyphon," Advances in Heat Pipe Technology, 4th International Heat Pipe Conference, Sept 7-10, 1981, London, England, pp. 147-162.

26. Lienhard, J. H., A Heat Transfer Text Book, Prentice-Hall, Inc., p. 406, 1981.

27. Collier, J. G., Convective Boiling and Condensation, McGraw-Hill (UK), 1972.

28. Ballback, L. J., "The Operation of a Ratating, Wickless Heat Pipe," M.S. Thesis, Naval Postgraduate School, Monterey, California, 1969.

29. Daniels, T. C. and Al-Jumaily, F. K., "Investigations of the Factors Affecting the Performance of a Rotating Heat Pipe," International Journal of Heat and Mass Transfer, Vol. 18, pp. 961-973, 1975.

30. Daniels, T. C. and Al-Jumaily, F. K., "Theoretical and Experimental Analysis of a Rotating Wickless Heat Pipe," 1st International Heat Pipe Conference, Stuttgart, Germany, 1973.

31. Heat Exchanger Design Handbook, Hemisphere Publishing Corporation, 1983, section 2.5.3.

32. Corman, J. C. and McLaughlin, M. H., "Thermal Design of Heat-Pipe Cooled A-C Motor," ASME Paper 71-WA/HT-14.

33. Niekawa, J., Matsumoto, K., Koizumi, T., Hasegawa, K., Kaneko, H. and Mizoguchi, Y., "Performance of Revolving Heat Pipes and Application to a Rotary Heat Exchanger," Journal of Heat Recovery Systems, Vol. 1, pp. 331-338, 1981.

34. Gray, V. H., "Methods and Apparatus for Heat Transfer in Rotating Bodies," U.S. Patent No. 3,842,596, 1974.

35. Vasiliev, L. L., "Heat Exchangers based on Heat Pipes," Nauka i Technika, Minsk, pp. 129-136 (in Russian).

36. Fries, P., "Experimental Results with a Wickless Centrifugal Heat Pipe," Int. J. Heat Mass Transfer, V. 13, pp. 1503-1504, 1970 (in German).

37. Fries, P. and Schultze, P., "Arrangement for Cooling Rotating Bodies," German Patent No. 1,928,358 (in German), 1974.

38. Fries, P., "Cooling Arrangement for the Rotor of an Electric Motor with a Centrifugal Heat Pipe," German Patent No. 2,251,841 (in German), 1974.

39. Oslejsek, O. and Polasek, F., "Cooling of Electric Machines by Heat Pipes," Proceedings of the II International Heat Pipe Conference, Bologna, Italy, pp. 503-514, 1976.

40. Vasiliev, L. L. Kukharsky, M. P. and Khrolenok, V. V., "Centrifugal Heat Pipes and Heat Exchangers," Heat and Mass Transfer Process in Porous Media with Phase Transformations, Academy of Science, BSSR, Ed. A. B. Lykov (in Russian).

41. Chalmers, B. and Herman, J., "Induction-Motor Fan Dtive with Unlaminated Rotor and Heat-Pipe Cooling," Proceedings IEE, V. 124, No. 5., May 1977, pp. 449-453.

42. Schneller, J., Pokorny, B. and Polasek, F., "Heat Transfer in Rotating Co-Axial and Parallel Heat Pipes and their Application in Machinery," Heat and Mass Transfer in Rotating Machinery, Eds.: D. E. Metzger and N. H. Afgan, Hemisphere Publishing Corporation, 1984.

43. Thoren, F., "Heat Pipe Cooled Induction Motors," 5th International Heat Pipe Conference, Tsukuba City, Japan, May 1984.

44. Sattler, P. K. and Thoren, F., "Totally Enclosed Heat Pipe Cooled Induction Motor (Theoretical and Experimental Results)," Proceedings Int. Conference on Electric Machines, Laussane, Switzerland, September 1984.

45. Brost, O., Unk, J. and Canders, W. R., "Heat Pipes in Electric Motors," 5th International Heat Pipe Conference, Supplement, Tsukuba City, Japan, May 1984.

46. Groll, M., Krahling, H. and Munzel, W. D., "Heat Pipes for Cooling of an Electric Motor," 3rd International Heat Pipe Conference, Palo Alto, California, 1978, pp. 354-359.

47. Furuya, H., "Heat Pipe Applications to Electronic Devices, Electric Equipment and Machine Tools," The Institute of Space and Astronautical Science Research Note, Tokyo, Japan, April 1983, pp. 18-39.

48. Koizumi, T., "Present Status of Utilization of Heat Pipes," ibid, pp. 62-80.

49. Tongze, M. and Zhangyan, J., "Heat Pipe Research and Development in China," 5th International Heat Pipe Conference, Tsukuba City, Japan, May 1984, pp. 81-92 and 113.

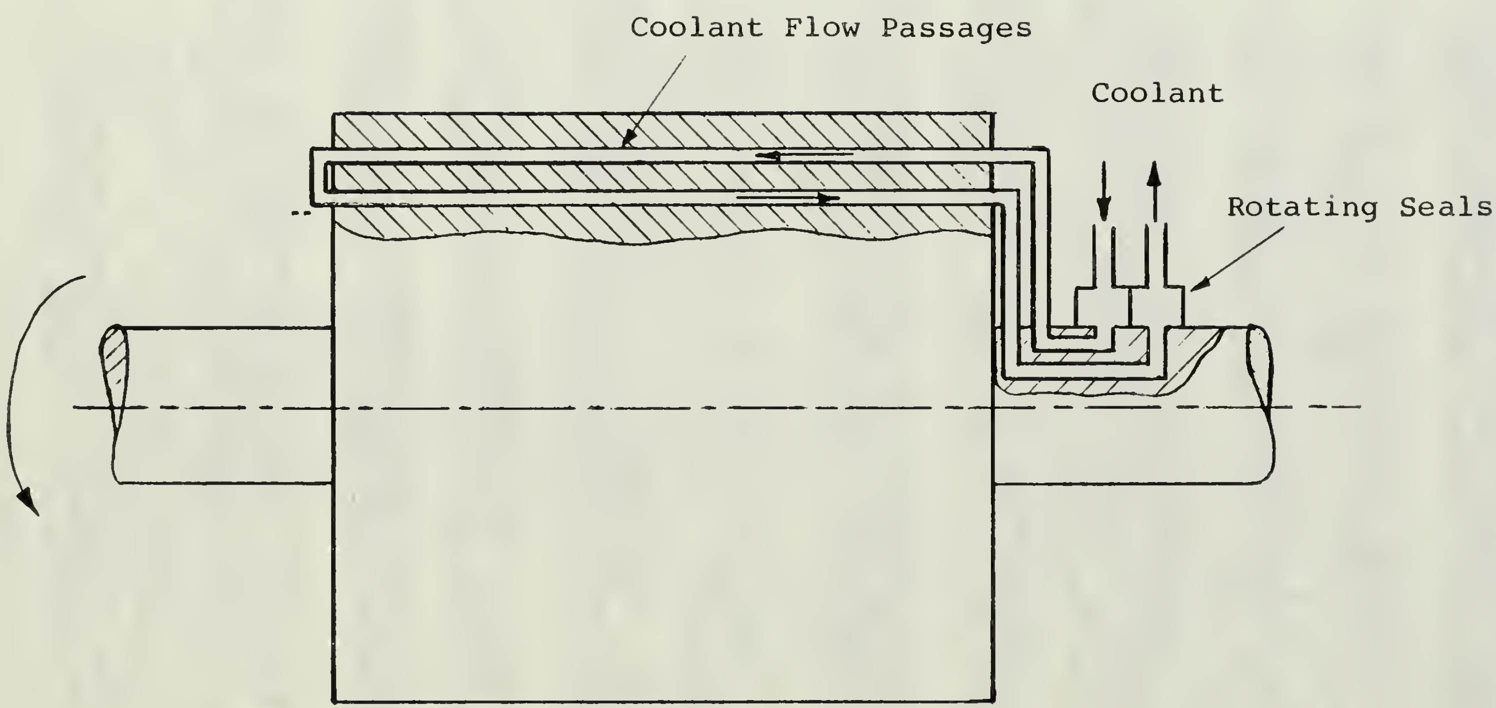

Figure 1 Schematic of a Liquid-Cooled Rotor.

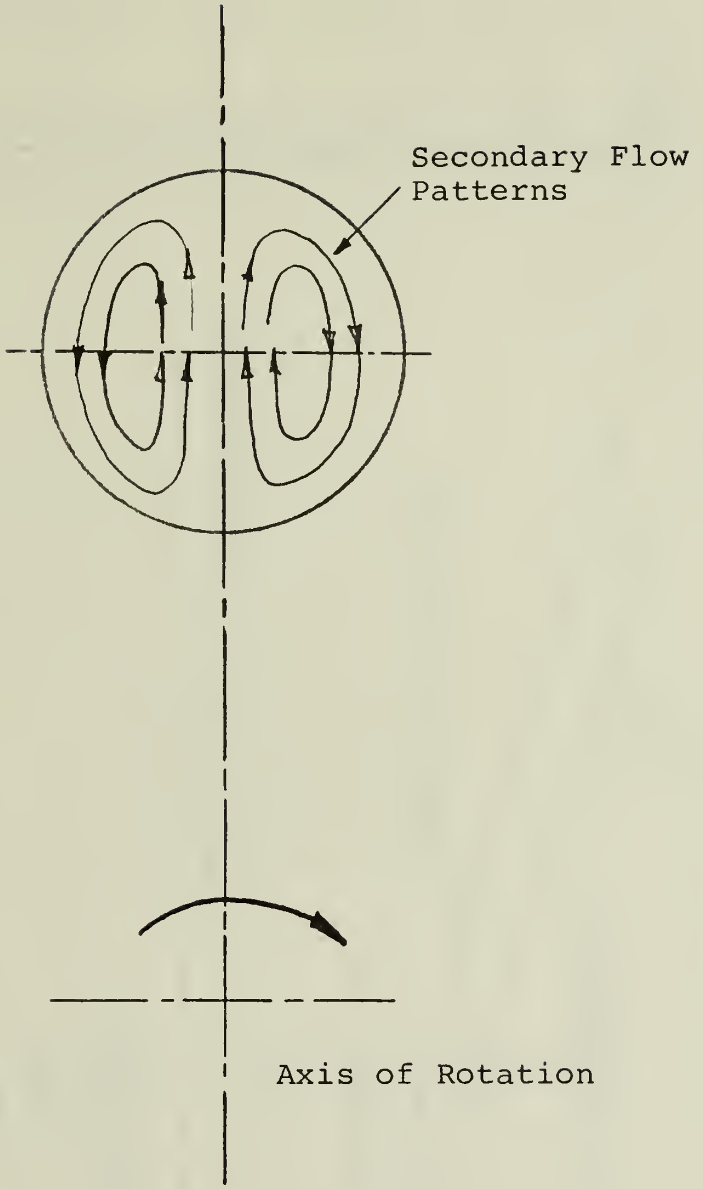

Figure 2 Typical Cross Section of a Heated Channel in off-Axis Rotation.

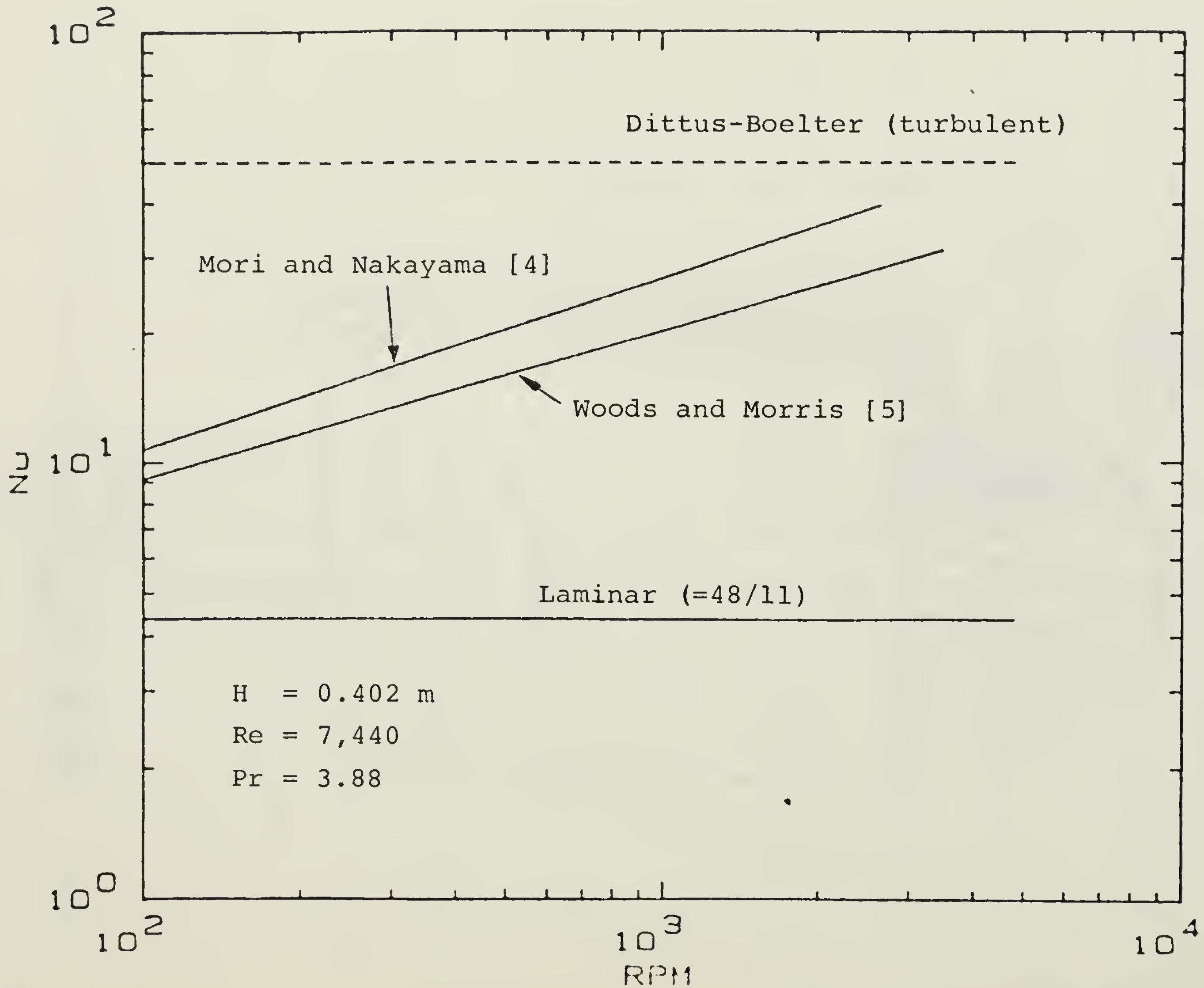

Figure 3 Comparison of Laminar Nusselt Number Correlations [1].

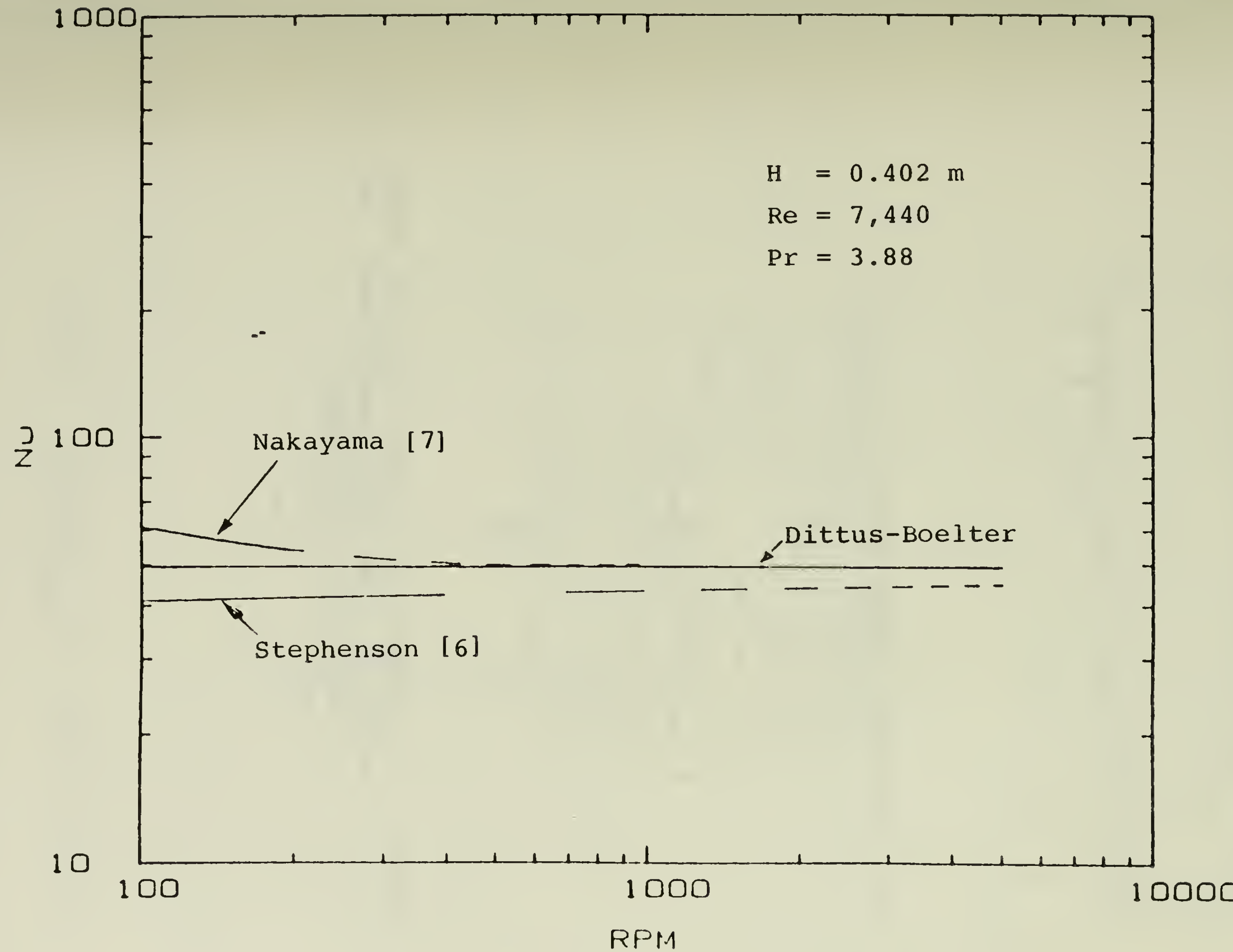

Figure 4 Comparison of Turbulent Nusselt Number Correlations [1].

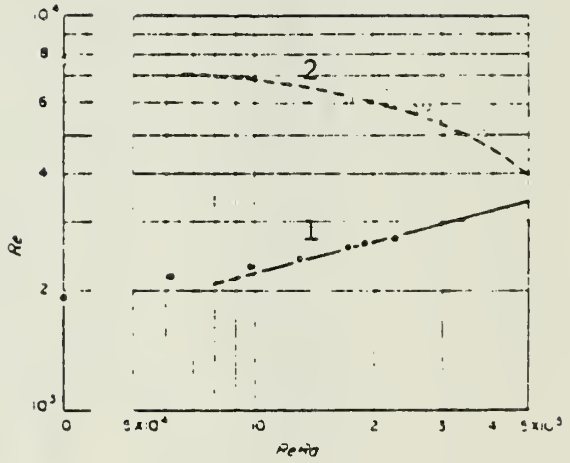

Figure 5 Variation of Critical Reynolds Number [8] with: 1. high inlet turbulence, and 2. low inlet turbulence.

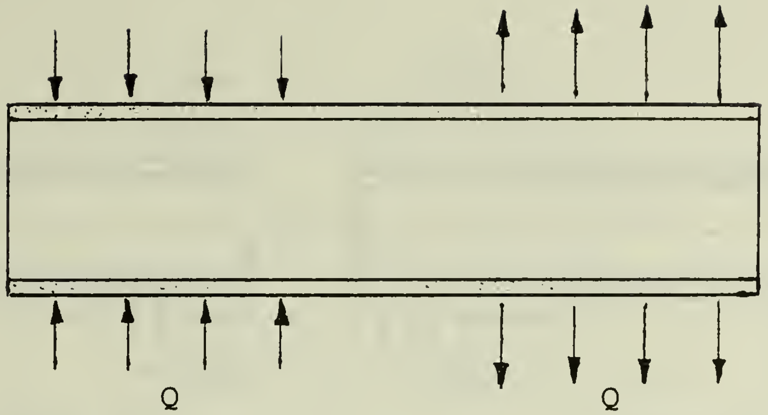

Figure 6 Basic Geometry of a Heat Pipe.

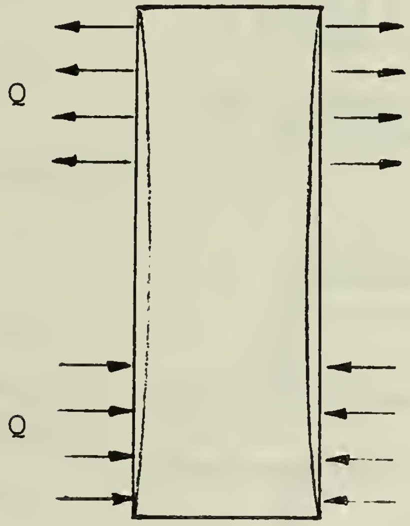

Figure 7 Schematic of a Gravity-Driven Heat Pipe.

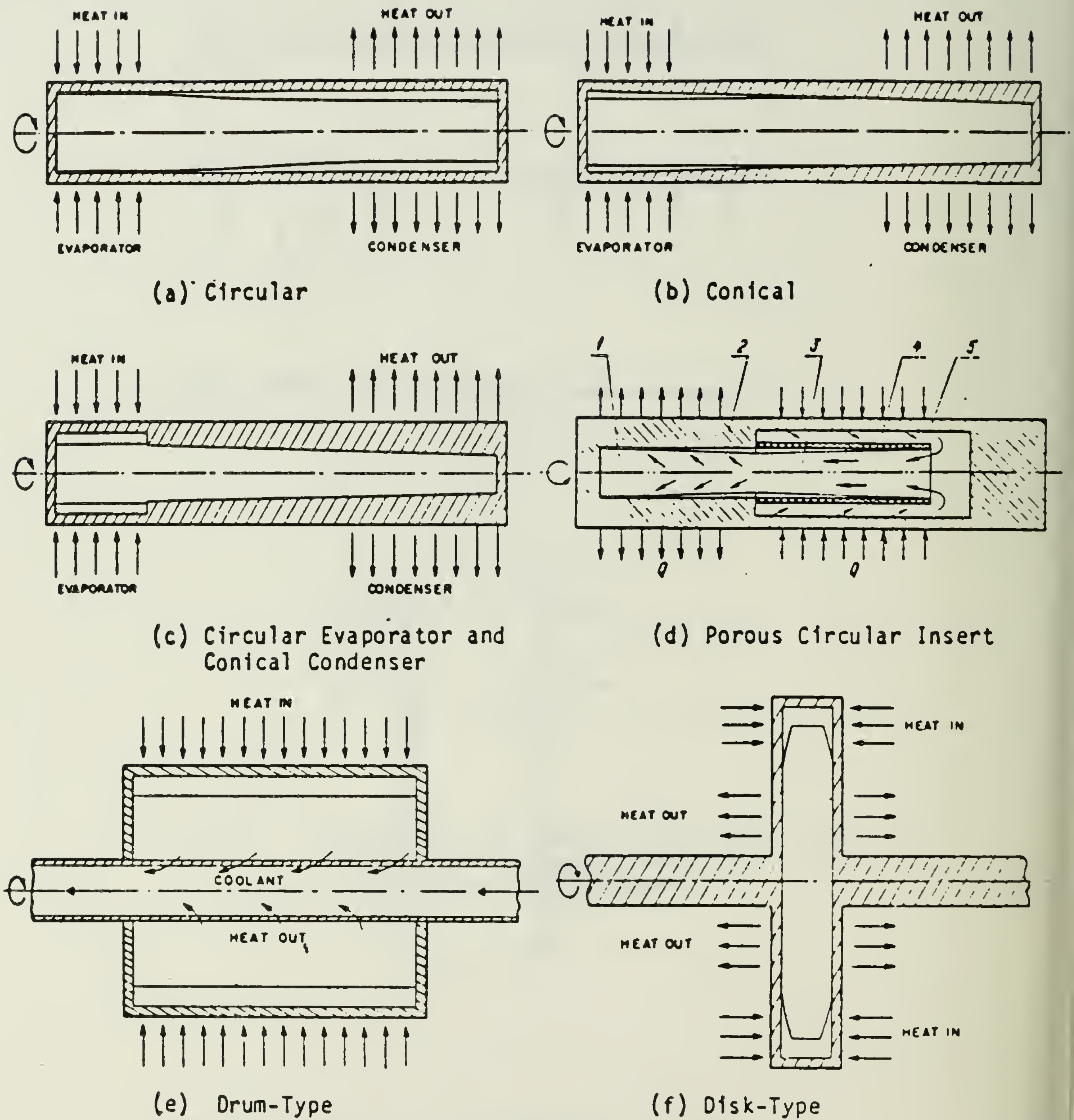

Figure 8 Various Rotating Heat Pipes.

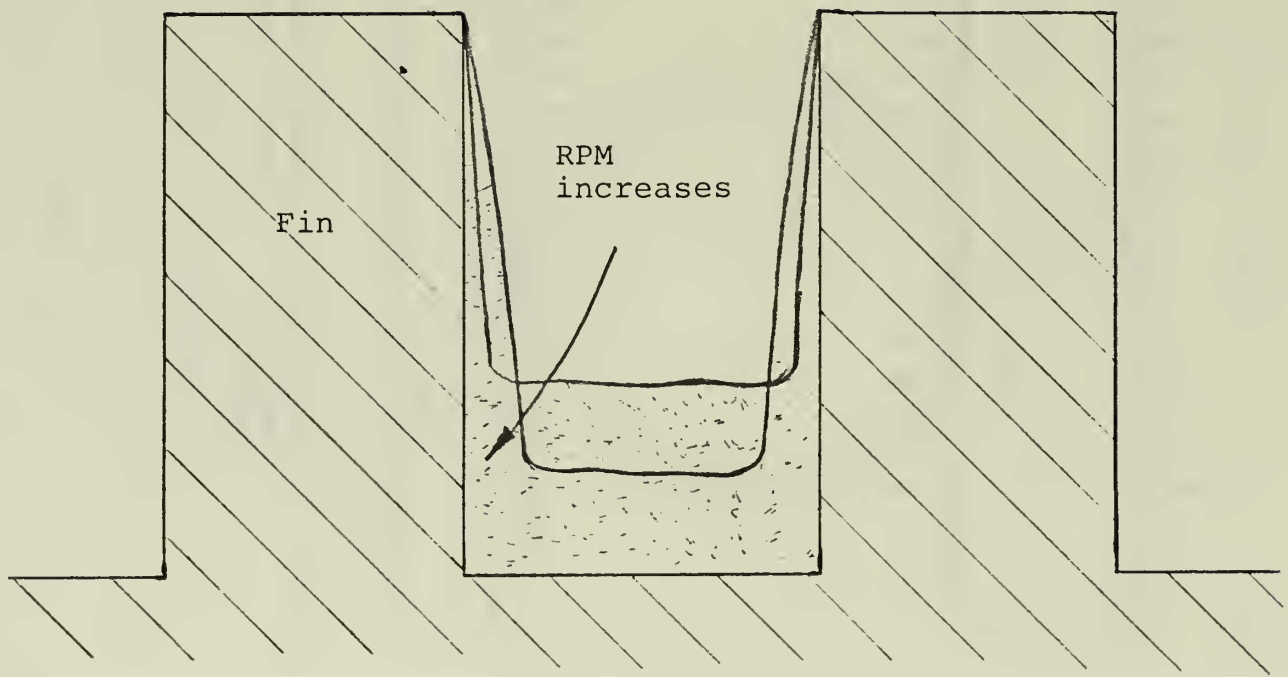

Figure 9 Effect of Rotational Speed on Condensate Film Thickness of an Internally-Finned, Rotating Heat Pipe.

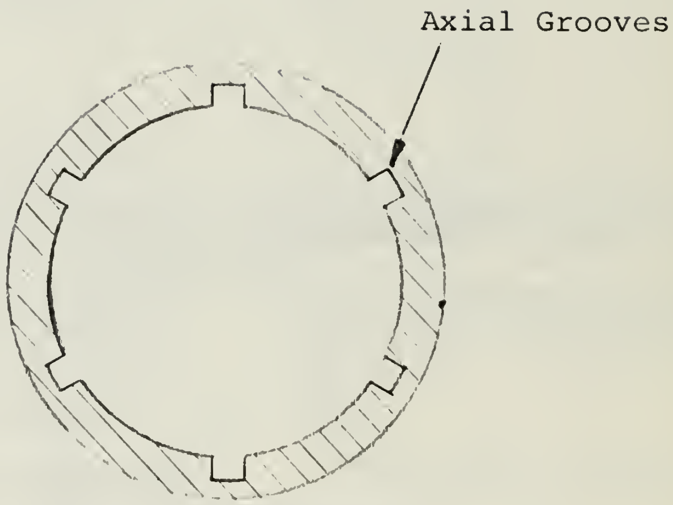

Figure 10 Schematic of a Heat Pipe Cross Section with Axial, Internal Grooves.

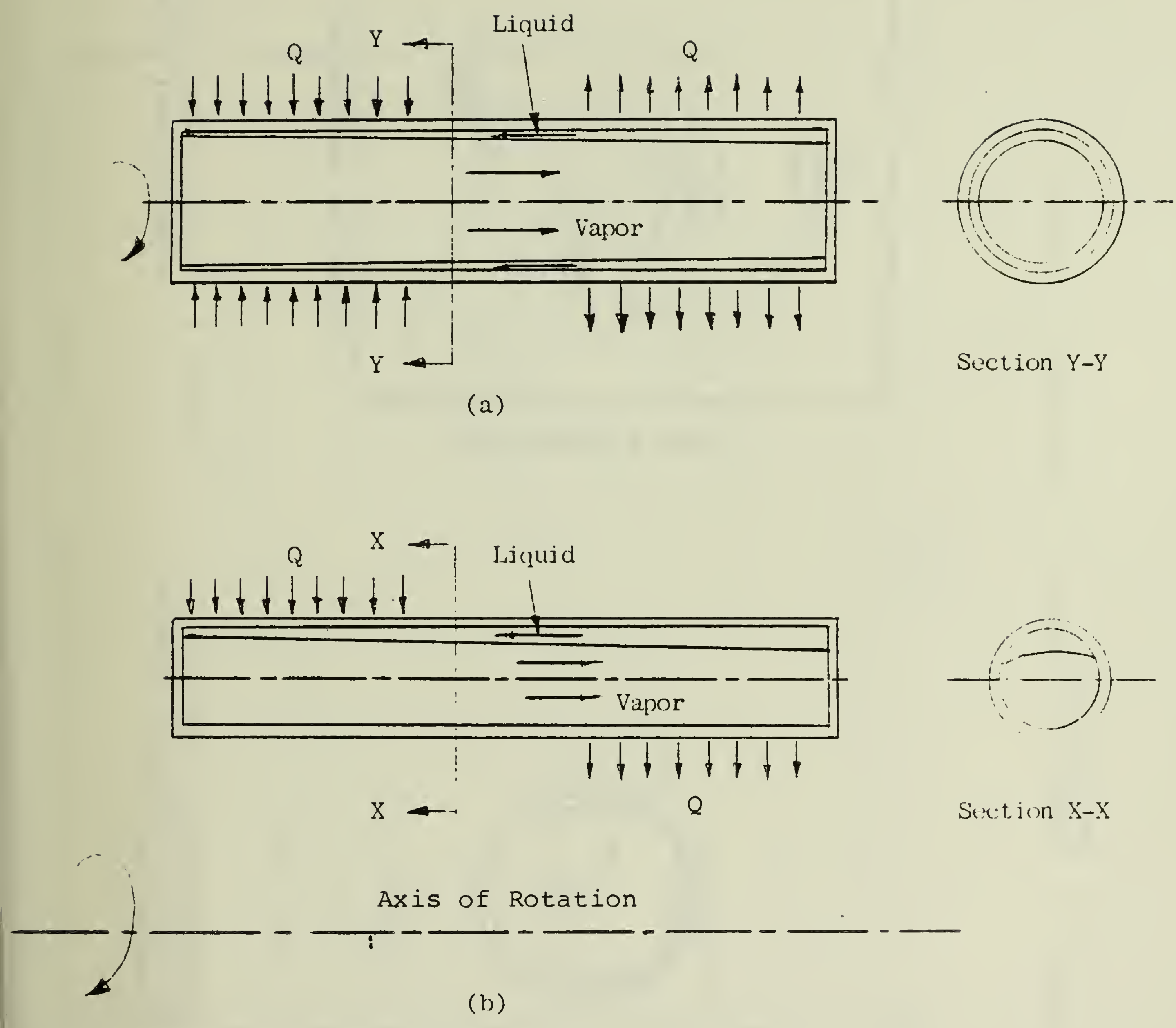

Figure 11 Rotating Heat-Pipe Arrangements: (a) on-axis heat pipe, and (b) off-axis heat pipe.

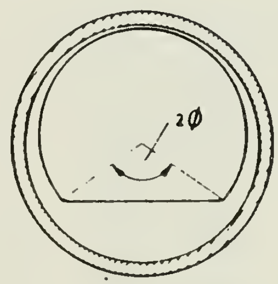

Figure 12 Cross-Sectional View of a Horizontal Tube with Internal Condensation.

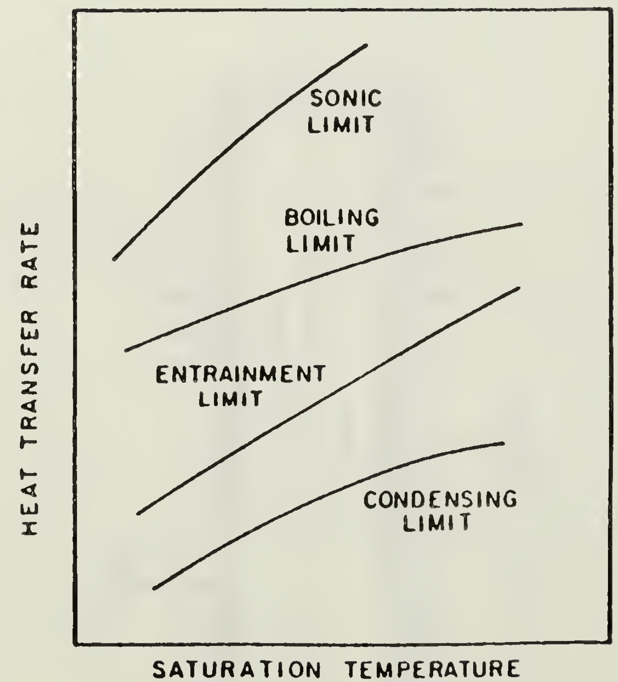

Figure 13 Operating Limits of Rotating Heat Pipes.

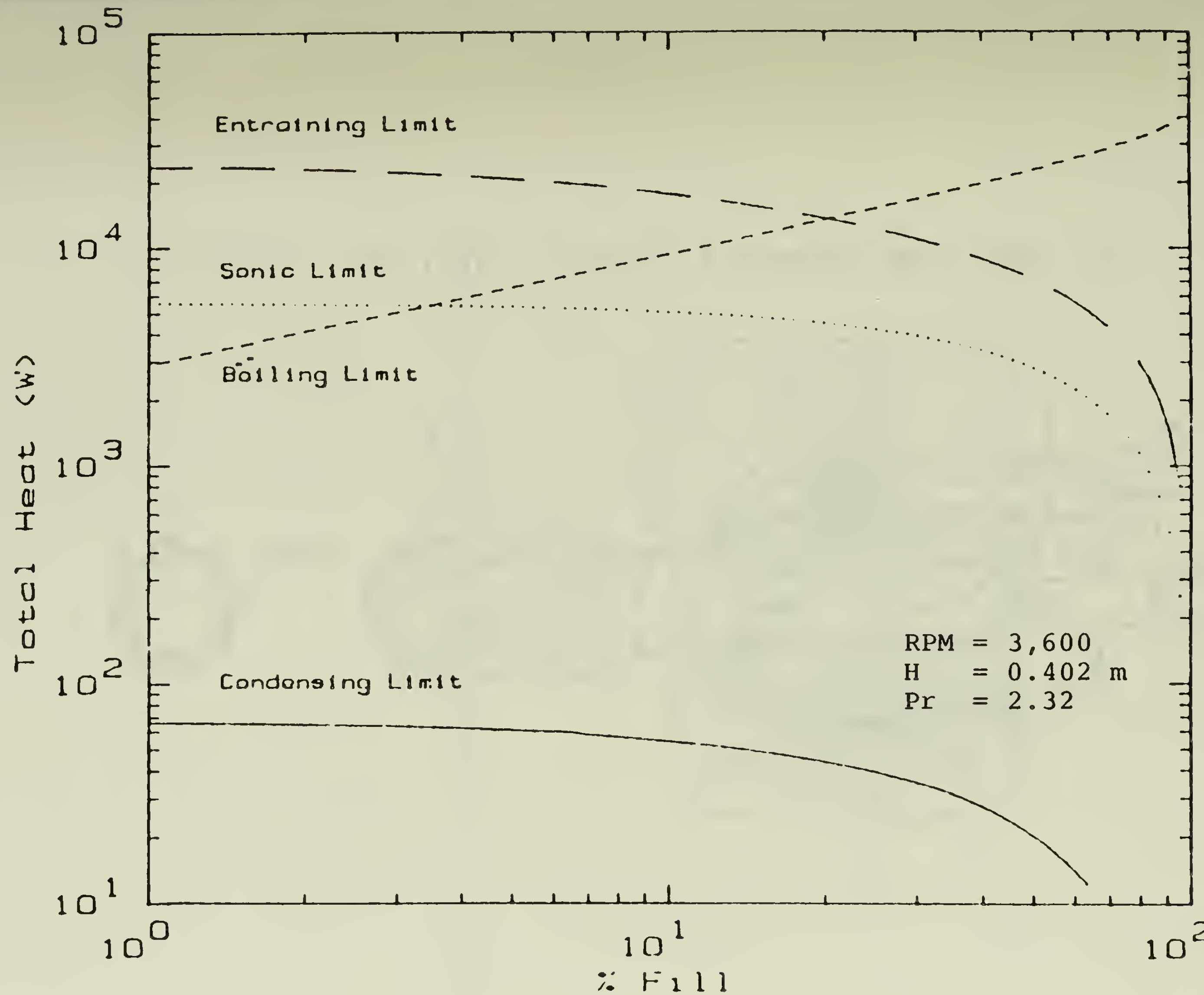

Figure 14 Operating Limits of a Heat Pipe in off-Axis Rotation [1].

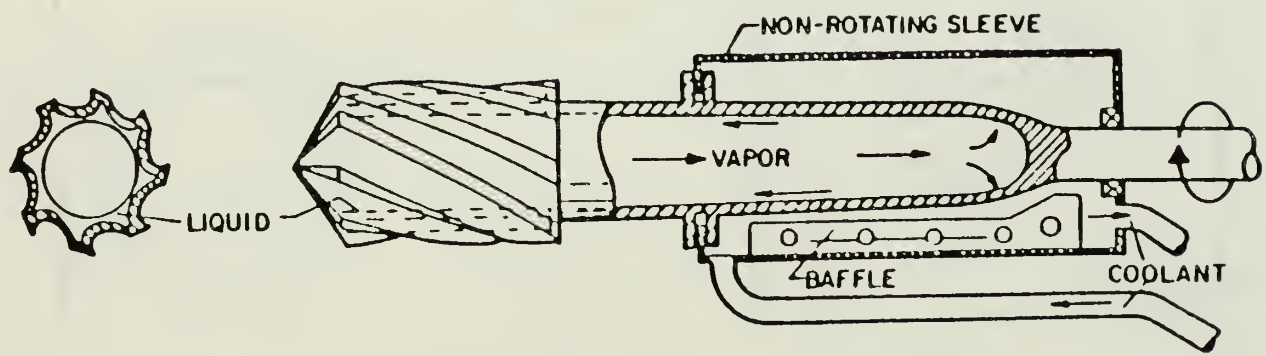

Figure 15 Drill Bit Cooled by a Rotating Heat Pipe [34].

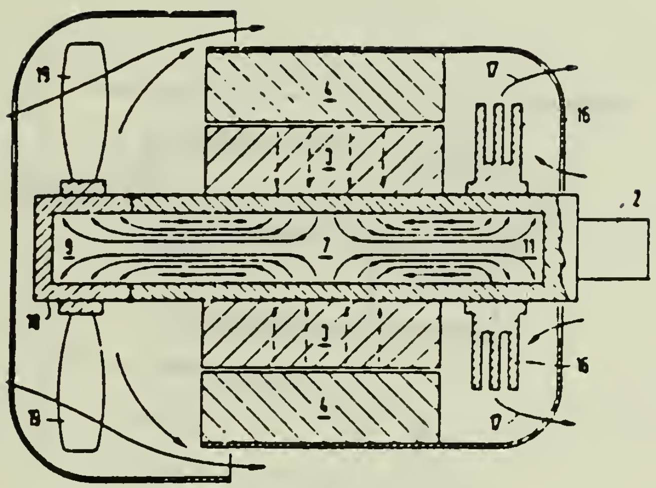

Figure 16 Electric Motor Cooled by a Rotating Heat Pipe [37].

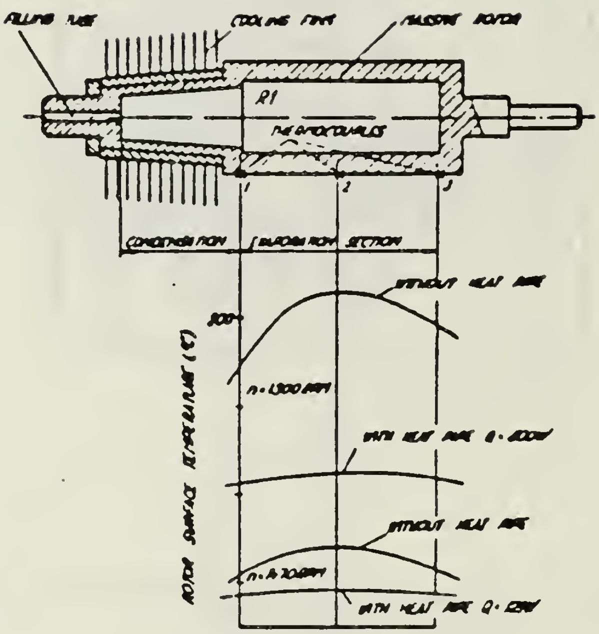

Figure 17 Heat-Pipe-Cooled Induction Motor [39].

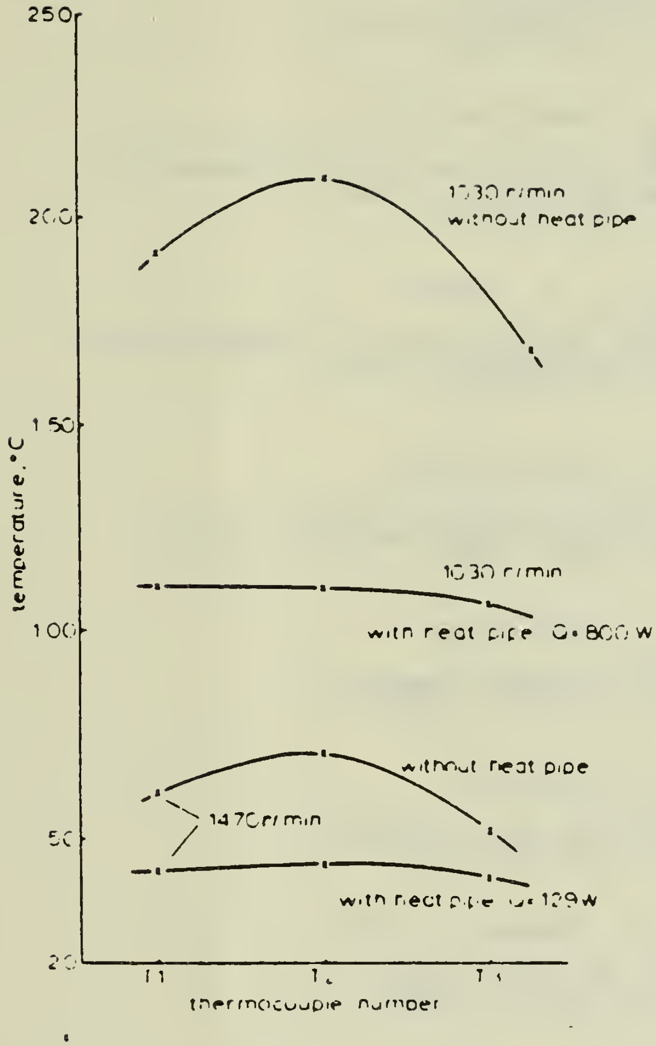

Figure 18 Effect of Heat-Pipe Cooling on Rotor Temperatures of an Induction Motor [41].

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