CONDENSER EFFECT IN THE PERFORMANCE OF THE HIGH SPEED ROTATION HEAT PIPES

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Abstract. *Rotating cylindrical heat pipes have some important applications, in electric engines cooling, turbine thermal control, and other kinds of equipment. It can also be applied in satellites with spin movement. The efect of rotation in its operation becomes important. when the angular velocity is too large, comparared to the evaporator in the heat pipe, recirculations appear, grow and scatter in the vapor phase, and the fluid involved in such recirculation cannot take part in the heat exchange process, which reduces the efficiency of the heat pipe. In some cases, the combination of high rotation and heat flux can obstruct the phase change from one section to another. In this work, is simulated a rotoating cylindrical heat pipe using the finite volume method. The energy and mometun equations are coupled through th flux at the liquid-vapor interface that defines the suction and velocities of injection for the liquid and vapor flow. Several values for angular and linear Reynolds number, as functions of angular velocity an heat flux, respectively, are showed, as well some results are compared with experimental data.*

Keywords: *Rotating heat pipes, Finite volume, Phase change in porous media.*

1. MATHEMATICAL FORMULATION

Considering a cylindrical pipe with constant angular velocity Ω in steady state operation, submitted to a wall heat flux Q in a half section and $-Q$ in the other, represented in a axisymmetric cylindrical coordinate system (Fig. 1).

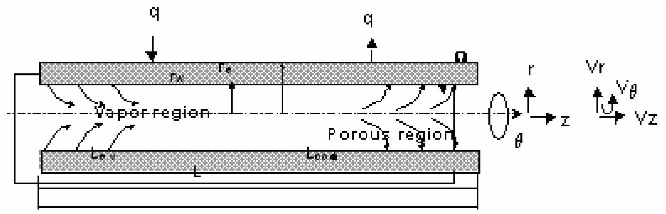

Figure 1. Schematic view of a rotating heat pipe.

Assuming that both phases are uncompressible and have constant physical properties under laminar flow, each one is represented by the following set of equations:

$$\frac{\partial Vr}{\partial r}+\frac{Vr}{r}+\frac{\partial Vz}{\partial z}=0 \tag{1}$$

$$\begin{aligned}\rho\left(Vr\frac{\partial Vr}{\partial r}-\frac{V\theta^2}{r}+Vz\frac{\partial Vr}{\partial z}\right)&=-\varepsilon\frac{\partial P}{\partial r}\\ &+\mu\left(\frac{\partial^2 Vr}{\partial r^2}+\frac{1}{r}\frac{\partial Vr}{\partial r}-\frac{Vr}{r^2}+\frac{\partial^2 Vr}{\partial z^2}\right)-\frac{\mu\varepsilon}{K}Vr\\ &+2\rho\Omega V\theta+\rho\Omega^2 r\end{aligned} \tag{2}$$

$$\begin{aligned}\rho\left(Vr\frac{\partial V\theta}{\partial r}-\frac{V\theta^2}{r}+Vz\frac{\partial V\theta}{\partial z}\right)&=\\ \mu\left(\frac{\partial^2 V\theta}{\partial r^2}+\frac{1}{r}\frac{\partial V\theta}{\partial r}-\frac{VrV\theta}{r}+\frac{\partial^2 V\theta}{\partial z^2}\right)&\\ -\frac{\mu\varepsilon}{K}V\theta+2\rho\Omega Vr&\end{aligned} \tag{3}$$

$$\rho\left(Vr\frac{\partial Vz}{\partial r}+Vz\frac{\partial Vz}{\partial z}\right)=-\varepsilon\frac{\partial P}{\partial z}$$
$$+\mu\left(\frac{\partial^2 Vz}{\partial r^2}+\frac{1}{r}\frac{\partial Vz}{\partial r}+\frac{\partial^2 Vz}{\partial z^2}\right)-\frac{\mu\varepsilon}{K}Vz \tag{4}$$

$$Vr\frac{\partial T}{\partial r}+Vz\frac{\partial T}{\partial z}=\alpha\left(\frac{\partial^2 T}{\partial r^2}+\frac{1}{r}\frac{\partial T}{\partial r}+\frac{\partial^2 T}{\partial z^2}\right) \tag{5}$$

where Vr, $V\theta$ and Vz are the radial, angular and longitudinal velocity components, respectively. In the vapor region, the porosity is considered to be $\varepsilon = 1$ and the permeability $K \to \infty$. The boundary conditions are:

$$Vr = V\theta = Vz = 0\,;\ \frac{\partial T}{\partial z}=0 \text{ at z = 0 , z = L} \tag{6a}$$

$$Vr = V\theta = 0\,;\ \frac{\partial Vz}{\partial r}=0\,;\frac{\partial T}{\partial z}=0 \text{ at r = 0} \tag{6b}$$

$$Vr = Vr_i(z)\,;V\theta = Vz = 0\,;T = T_{sat} \text{ at } r = r_i \tag{6c}$$

at $r = r_w$

$$Vr = V\theta = Vz = 0 \text{ and}$$
$$-\,k\frac{\partial T}{\partial r}=Q \text{ in } z < Lev \tag{6d}$$

In $z > L_{ev}$

$$T = T_w \qquad (1^{st}\text{ kind}) \tag{6e}$$

$$-k\frac{\partial T}{\partial r}=-Q \quad (2^{nd}\text{ kind}) \tag{6f}$$

$$-k\frac{\partial T}{\partial r}=h(T-T_\infty) \quad (3^{rd}\text{ kind}) \tag{6g}$$

where $V_i(z)$ is the local velocity of injection, that couples the vapor and porous regions by:

$$q(z)+k_e\frac{\partial T}{\partial r}=\rho Vr_i(z)h_{lv}$$
(7)

Where:
h_{lv} is the latent heat of vaporization.
$Vr_i(z)$ is found along the solution, via numerical iterations, T_w is the prescribed wall temperature, T_{oo} is the environment bulk temperature and h is the convection heat transfer coefficient.

For a given superficial temperature profile, obtained by the experimental data, it is necessary to estimate the vapor mass injection due to the heat input in function of this temperature distribution. We can write:

$$q(z)=\frac{k_r}{\delta_r}(TP-T_{Sat.}) \tag{8}$$

where:
k_r is the effective conductivity, including the wall and porous thickness, δ_r .

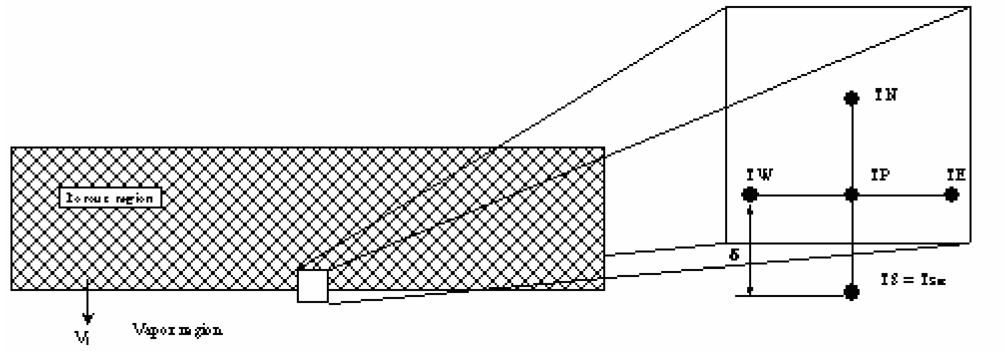

Figure 1a. Schematic position of TP to obtain the fluid injection velocity.

TP is the temperature obtained in an point preceding the first point at the vapor region. This way we can determine the injection velocity by:

$$Vi(z) = \frac{q(z)}{\rho h_{lv}} \tag{9}$$

With the given temperature distribution, obtained trhough experimental data, we can obtain the velocity profile. This way we are able to compare, indirectly, the simulated result with the experimental data, e discuss some high rotation heat pipe behavior.

Ismail and Miranda, 1997 presented a two-dimensional model for a rotating heat pipe with porous structure, with the objective to stabilize the liquid region. Machado and Miranda, 2000 extended the Ismail and Miranda model in order to analyze the effects of increasing heat flux and rotation over the fluid flow and heat exchange into the heat pipe. Machado and Miranda, 2003 using the two-dimensional model for a rotating heat pipe, trying to find a limiting curve of operation for these pipes based on angular and injection Reynolds number.

These simulations were not possible to compare withh experimental data, due to the lack of experimental data in that time, mainly for higher rotations.

2. EXPERIMENTAL WORK

If the rotation is fast enough, phase change will occur only with higher temperature and in this process can appear some recirculation in the vapor region. This behavior was not observed by empirical and analytical models proposed by Daniels and Al-Jumaily,1975, Daniels and Willians, 1978 and Ling and Cao, 2000.

Ponnapan. and He, 1998 presented performance results for stainless steel 316-methanol an stainless steel 316- water high-speed rotating heat pipes (HSRHPs) that were tested at speeds up to 30.000 rpm. The HSRHPs were cooled by air and oil jets in separate tests to compare their relative performances. Interesting temperature profiles and test result was presented for HSRHPs that were heated by an induction heater and tested in the temperature range of 20-25ºC for power inputs from 250 to 1300W.

All of the performance tests on both water and methanol HSRHPs were conducted in horizontal orientation an on-axis rotation mode under steady-state condition of the test parameters. The spread of the present test parameters were as follows:

1. Power input: 250, 500, 750 W (air-cooled mode); and 250, 500, 750, 1000, 1250, and 1300 W (oil-cooled mode);
2. Speed: 5,000-30.000 rpm in steps of 5,000 rpm;
3. Flow rate: 14,87 standard m^3/h air at 25ºC; and 0,87, 1,89, 2,84, and 3,26 lpm oil;
4. Oil temperature: 30-60ºC;
5. Jet cooling nozzle arrangement: three nozzles mode (air and oil); and one nozzle mode (oil only).

An axial temperature profile for Air Cooling Mode when the pipe is subjected to heat load of 750W and rotation of 10.000-30.000 rpm in steps of 5,000 rpm, it showed that the temperatures decreased as the speed increased, which is the expected trend. Three air nozzles kept 120-deg apart around the pipe were used to apply the air cooling. The gradual change of condensate film thickness occurring inside the pipe indicates the temperature gradients from the midsection to the evaporator and condenser ends. The lowest temperature happens at the middle, because is where it had the high fill thickness.

When the oil cooling mode is used, the temperature profiles of the HSRHPs showed that there is an unexpected behavior of increasing temperature with increasing speed. According to Ponnapan and. He, 1998, this anomaly can be attributed to two reasons. One is that the inlet temperature of the oil to the condenser was not constant and the other is the poor contact of the coolant oil with the HSRHPs condenser at higher speeds. As consequence it has that the velocity of the impinging oil-jet is not high enough to overcome the increasing peripheral velocity of the HSRHPs with increasing rotation speed.

In this work it will be shown that the reasons presented for the increase of the temperature profiles, when the oil cooling mode is used, they are not suitable, and that the true reason of the fact, is that the condenser is not properly operating with the evaporator and, maybe rotation will increase the vapor pressure. It is necessary to know the vapor pressure that the heat pipe is operating.

$$p_v = p_{ref} + \rho_l \Omega^2 R_i \,. \tag{10}$$

And

$$T_{Sat} = f(p_v). \tag{11}$$

In order to obtain the vapoor pressure, it is necessary to know the reference pressure, Eq. 10. The reference pressure is that the heat pipe was filled. In this work we tried to determine this pressure by infering the vapor pressure trhough the experimental results of Ponnapan. and He, 1998.

3. RESULTS

As the porous medium stabilizes the liquid film, keeping constant thickness, for a constant heat input and output, the temperature profile is independent of the rotation, as shown in Fig. 2 and 3. The reason is that the rotation affected the inner pressure of the heat pipe, but not in this case the liquid film thickness. However, it is not necessarily true for the case with liquid film, but its thickness is so small to give a great variation in the temperature. But more experimental data are necessary to make an accuracy evaluation that relates the film thickness variation with the rotation.

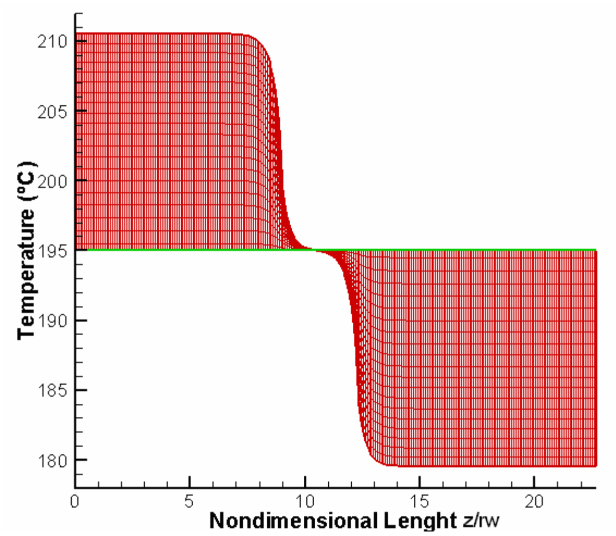

Figure 2. Temperature distribution for constant heat Flux for heat pipe operating at 5,000 rpm

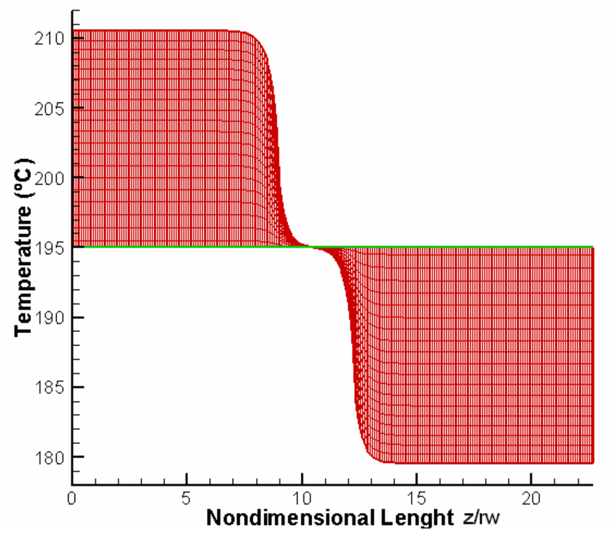

Figure 3. Temperature distribution for Constant heat flux for heat pipe operating at 30,000 rpm

Imposing a temperature profile at the heat pipe surface, operating in accordance with the experimental data given by Ponnapan and He, 1998, operating with water and being cooled by oil and considering the saturation temperature of 195ºC, one can verify that the evaporator region and condenser, are not operating in an adequated form, as it can be seen in Figure 4. This means that the condenser is not removing the waited amount of heat that would be of 750 W.

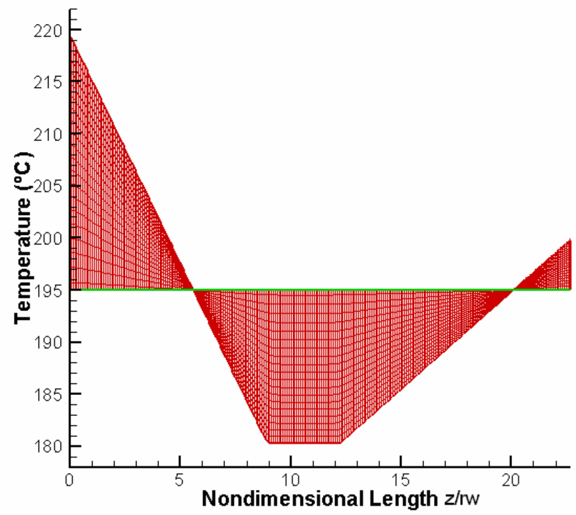

Figure 4. Experimental temperature profile for rotation of 30,000 rpm.

However, the rotation will affect the injection of vapor mass in this pipe. How greater the pipe rotation lesser will be the amount of vapor effective mass, that will not take part in the heat transfer process. Figures 5 and 6 show the streamlines for a rotating heat pipe operating properly, with constant input and output heat flux. Note that some vapor mass is not transferring energy by condensation and evaporation, and this tendency increases with the increase in the rotation.

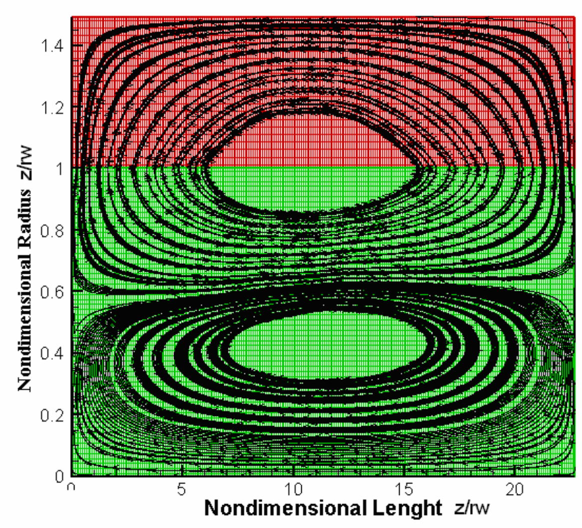

Figure 5. Streamlines for constant heat fluux for rotation of 5,000 rpm

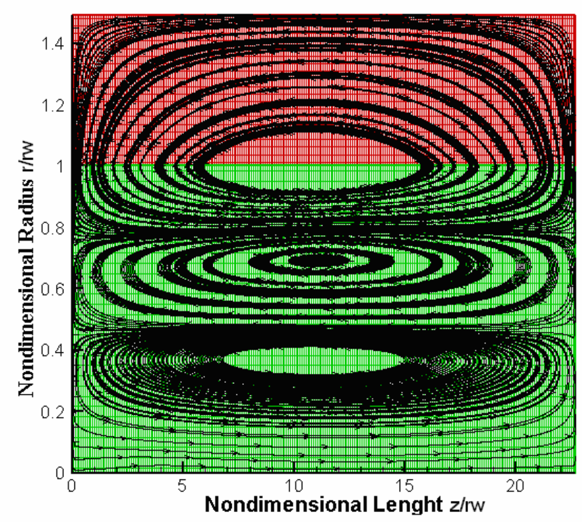

Figure 6. Streamlines for constant heat flux for rotation of 30,0000 rpm

In accordance with the given temperature profile, it is verified that the amount of effective heat mass transferring heat is greatly reduced. This can be verified when we compare the Fig. 6 and 7. We can note that the effective vapor mass transferring energy is even more greatly reduced.

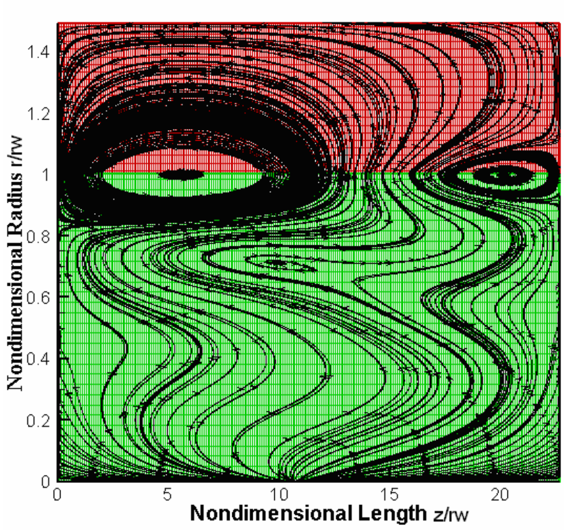

Figure 7. Streamlines profile for the given experimental temperature profile for rotation of 30,0000 rpm

We can assure that the main problem in the decreasing heat transfer process is due to the condenser, that is not operating properly and not due to the liquid film or the effective vapor mass flow. Analyzing the experimental temperature profile and comparing with the theoretical profile, Fig. 4, we can observe that was a heat input decreasing of 58% approximately.

Increasing the heat pipe rotation, and increasing the centrifugal forces, the heat pipe will respond to compensate this increase in pressure, increasing the saturation temperature of the working fluid. It is necessary more experimental data to verify how the pressure vapor core will change with the heat pipe rotation.

Considering that its heat pipe works with a filled reference vapor pressure of 100000 N/m^2 , we can have a saturation temperature varying with rotation as given in the Fig. 8

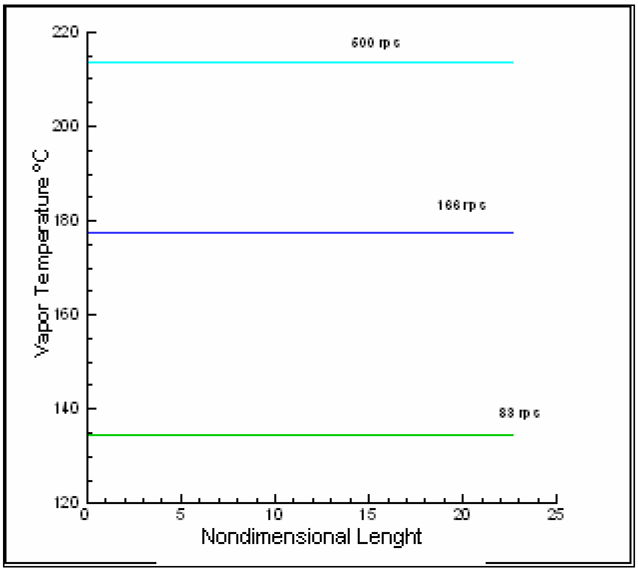

Figure 8. Saturation temperature for different heat pipe rotation.

This model could predict with good accuracy the heat pipe behavior in high speed rotation.

4. CONCLUSIONS

The most serious problem in an adequate rotating heat pipe operation is due to the condenser working properly, that will adjusted the given heat by the evaporator. Higher rotations will diminish the effective vapor mass that will take part in the heat transfer process and will increase the vapor temperature.

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