Proof of Existence of Optimal Point of Test Variables to Detect Noncondensable Gas in Heat Pipe Tests

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Abstract. *This work presents the proof of existence of an optimal point in the test variables to detect noncondensable gas in heat pipe tests. The variables tested in this work are the temperature difference above the ambient temperature and the intensity of condenser cooling. A numerical model of the heat pipe condenser was developed to simulate the steady state temperature distribution under different test conditions and find the quantity of noncondensable gas that is possible to be detected for each combination of these two variables. The model was executed to simulate a aluminium grooved heat pipe charged with acetone or ammonia in the presence of an ideal gas.*

Keywords: Heat Pipe; Heat Transfer; Numerical Modeling; Noncondensable Gas.

Nomenclature:

A_{cs}	Cross-sectional Area $[m^2]$	p_v	Pressure of Vapor $[Pa]$
A_v	Cross-sectional Area of Vapor Core $[m^2]$	P_{int}	Inner Perimeter of Heat Pipe $[m]$
A_w	Cross-sectional Area of Wall $[m^2]$	P_{ext}	Outer Perimeter of Heat Pipe $[m]$
A_{wick}	Cross-sectional Area of Wick $[m^2]$	R_u	Universal Gas Constant $[J/molK]$
h_{int}	Heat Transf. Coef. Wick/Vapor $[W/m^2K]$	R_v	Gas Constant of Vapor $[J/kgK]$
h_f	Heat Trans. Coef. Wall/Amb. $[W/m^2K]$	T	Temperature $[K]$
k	Equivalent Therm. Conductivity $[W/mK]$	T_0	Reference Temperature $[K]$
k_{eff}	Effective Thermal Conductivity $[W/mK]$	T_{amb}	Ambient Temperature$[K]$
k_l	Thermal Conductivity of Liquid $[W/mK]$	T_v	Temperature of Vapor $[K]$
k_w	Thermal Conductivity of Wall $[W/mK]$	T_w	Temperature of Wall $[K]$
L_c	Length of the Condenser Section $[m]$	V_g	Volume of Noncondensable Gas $[m^3]$
L_g	Length of Condenser with NCG $[m]$	w_l	Width of Groove Filled with Liquid $[m]$
n_g	Number of Moles of Noncondensable Gas	w_s	Width of Solid Fin $[m]$
p	Pressure $[Pa]$	x	Coordinate along x axis $[m]$
p_0	Pressure of Reference $[Pa]$	λ	Latent Heat of Evaporation $[J/kg]$
$p_{v,av,b}$	Avg Pres. of Vapor at Blocked Zone $[Pa]$	ΔT_v	Temp. Difference Vapor/Ambient $[°C]$
p_g	Pressure of Noncondensable Gas $[Pa]$	$\Delta T_{NCG,noNCG}$	Temp. Difference between Pipes $[°C]$

1. Introduction

A heat pipe (HP) is a heat transfer device used to transport heat between two solid interfaces. The principle of a HP was first conceived by Gaugler (1944), but its applications and effectiveness was showed later by Grover and his colleagues from Los Alamos Scientific Laboratory (GROVER et al. 1964).

In its simplest form a HP is composed by a metallic tube closed with its inner surfaces lined with a wick structure. The wick structure is saturated with liquid and the remaining volume contains the vapor phase of a working fluid. When a heat load is applied at some section of the heat pipe by an external source, this heat vaporizes the working fluid at that. This additional vapor generated results in a difference in pressure at the core of the pipe driving vapor from evaporator to the condenser zone where the vapor condenses and releases the latent heat of vaporization to a heat sink. When liquid is depleted by evaporation at evaporator zone, a capillary pressure is developed in this point. This capillary pressure pumps the condensed liquid back from condenser to evaporator.

Noncondensable gas (NCG) can be present inside a HP due to chemical reactions between the fluid and container or decomposition of the working fluid (MARCUS, 1972). The presence of NCG is probably the most common indication of heat pipe failure and, as the NCG tend to accumulate in the HP condenser section, which gradually becomes blocked, it is easy to identify because of the sharp temperature drop that exists at the gas-vapor interface (REAY et al, 2014).

The detection of a small quantity of NCG inside the pipe can be very helpful to reduce cost and time, with long periods of time and the demand for physical structures like rooms, benches, data acquisition systems and thermocouples, HP tests are usually very expensive. It is therefore necessary to accelerate the tests because if the results of a test appears quickly, the test can be shortened or new combination of working fluid with material of container and capillary structure can be evaluated in a fastest way.

The detection capability depends strongly of two test parameters, the temperature difference above the ambient temperature and the intensity of condenser cooling. At higher temperature difference, the temperature profile distortion is more visible, while under lower temperature the NCG blockage zone can be higher for the same NCG mass. It can also be affected by factors such a high conductivity of container material or end cap, and the presence of a filling tube.

The typical temperature profile along the HP condenser, without NCG and the same HP with a small amount of NCG is shown in Figure 1. In real applications, even without the presence of NCG, the final section of HP can have a decrease in temperature due to the presence of end cap and filling tube (which is commonly present during tests).

During a test setup, a few temperature sensors are installed along the heat pipe, and at least one sensor is located at the end of condenser section and another sensor is positioned at some distance where the temperature profile is still not distorted by NCG. The NCG is typically detected by a temperature difference measured between the sensor at the end of condenser and other sensor at the unaffected zone. The more this temperature difference, the detection ability is better.

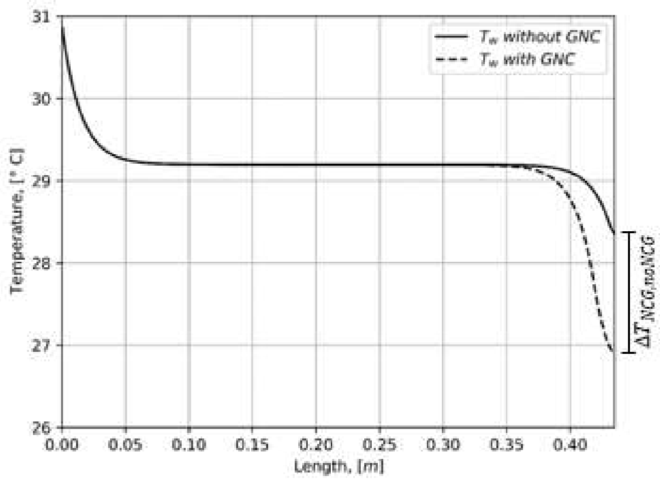

Figure 1. Typical temperature profile of heat pipe condenser.

Even with the importance of this problem, no studies or attempts to find how this two test parameters affect the NCG detecting ability have been published up to the moment.

The objective of this work is to develop a representative model of heat pipe condenser with and without the presence of NCG, test this model with different input parameters show the influence of different input parameters in the minimum NCG quantity which is possible to detect HP tests, for a certain heat pipe.

2. Mathematical and Numerical Modeling

To accomplish the task of sweeping the search space with two variables and solving a numerical model each time, the model needs to be solved for a very large number of times, which drives the necessity to develop a simplified model to simulate the heat pipe but still representative of the effects occurring in the HP.

To accomplish such task, a model based on the analytical model presented by Chi (1976) was developed. The model does not consider the presence of the evaporator section, instead it uses the temperature of the vapor as input to the power source. The model accepts the existence of an adiabatic zone, but such feature was not used in the simulations presented in this article.

The model considers a HP without evaporator and adiabatic zone but with a detailed treatment of condenser-end zone, where the temperature distortion occurs due to superposition of two different effects, an additional cooling from end cap and filling tube, and vapor core blockage with NCG. Both effects can cause the temperature profile distortion by a very similar manner when even a small amount of NCG is present.

The mathematical model domain is the condenser zone of the HP which includes active and inactive portions of the condenser length, the end cap, and the filling tube. The main model definitions are shown in the Figure 2.

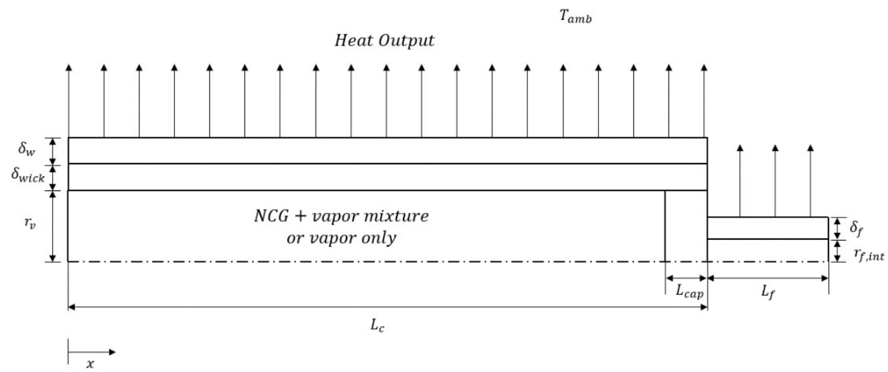

Figure 2. Heat Pipe Model Numerical Domain

Steady-state 1D mathematical model of condenser

The model presented here is a simplified steady-state, one-dimensional model, with a single energy equation for wall and wick. A representation of the model nodes is seen in Figure 3.

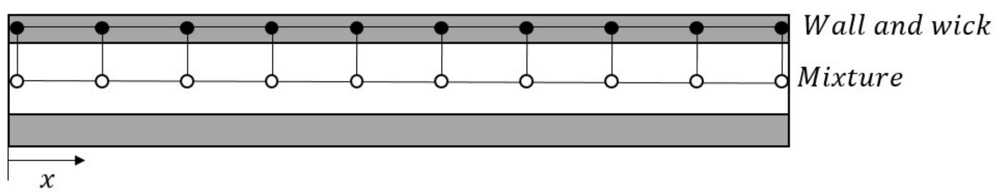

Figure 3. Graphic representation of wall, wick and mixture nodes

Equation (1) shows the one-dimensional steady-state energy conservation equation for the wall-wick combined effective set in its differential form. The energy conservation equation takes into account the heat input-output configuration of the heat pipe, which considers a heat transfer between the wick and vapor, and between the container and the surrounding environment:

$$k\frac{\partial^2 T_w}{\partial x^2} + \frac{h_{int}P_{int}}{A_{cs}}(T_v - T_w) + \frac{h_f P_{ext}}{A_{cs}}(T_{amb} - T_w) = 0, \quad (1)$$

where k is the thermal conductivity of wall-wick set, T is the temperature, x is the coordinate along the axis, h is the heat transfer coefficient, P is the perimeter, A_{cs} is the crossectional area.

The wick is considered saturated with liquid, and h_{int} is set to zero at the filling tube and in the zone blocked by noncondensable gas.

The effective thermal conductivity for a grooved wick, in the condenser section is given by (Chi, 1976):

$$k_{eff} = \frac{w_l k_l + w_s k_w}{w_l + w_s}, \quad (2)$$

where k_l is the thermal conductivity of liquid, k_w is the thermal conductivity of wall, w_l is the width of groove, w_s is the width of groove fin.

The thermal conductivity of the wall and wick combined was used, and is calculated from the area average of wick and container thermal conductivities:

$$k = \frac{A_w k_w + A_{wick} k_{eff}}{A_w + A_{wick}}, \tag{3}$$

where, A_w is the crossectional area of wall and A_{wick} is the crossectional area of wick.

The Clausius-Clapeyron equation relates the temperature and pressure of a saturated fluid:

$$ln\left(\frac{p}{p_0}\right) = \frac{\lambda}{R_v}\left(\frac{1}{T_0} - \frac{1}{T}\right), \tag{4}$$

which when solved for pressure become:

$$p = p_0 e^{\frac{\lambda}{R_v}\left(\frac{1}{T_0} - \frac{1}{T}\right)}, \tag{5}$$

where p is the current pressure of working fluid, p_0 and T_0 are values of a reference point on the real saturation curve, T is the current temperature of fluid, λ is the latent heat of evaporation of fluid and R_v is the gas constant of fluid.

Table 1 presents the parameters and dimensions of the heat pipes container and wick.

Table 1 - Dimensions and Parameters of HPs

Parameter	Value
L_c	435 mm
L_f	25 mm
L_{cap}	5.0 mm
r_v	9.8 mm
δ_w	2.2 mm
δ_{wick}	1.2 mm
$r_{f,int}$	1.5 mm
δ_f	1.0 mm
w_s	1.98 mm
w_l	0.52 mm
k_w	200.0 W/mK

Length of blocked zone

Considering a flat-front model to describe the behavior of the interface formed with vapor and noncondensable gas, the pressure of noncondensable gas is given by the difference in pressure of vapor from the non blocked zone and the average pressure of vapor inside the blocked zone:

$$p_g = p_v - p_{v,av,b}, \tag{6}$$

where, p_g is the pressure of noncondensable gas, p_v is the pressure of vapor at the entrance of the condenser and $p_{v,av,b}$ is the average pressure of vapor at the blocked zone.

The average pressure of vapor on blocked zone is calculated from the average temperature of blocked zone.

The equation of state relates the temperature, pressure, and density of the vapor phase and non-condensable gas:

$$p_g V_g = n_g R_u T, \tag{7}$$

and the length of condenser occupied by the noncondensable gas can be derived from the equation of state:

$$L_g = \frac{n_g R_u T}{p_g A_v}, \tag{8}$$

where, V_g is the volume occupied by the noncondensable gas, n_g is the number of moles of noncondensable gas, R_u is the universal gas constant, L_g is the length of condenser occupied by noncondensable gas, T is the temperature of gas and A_v is the crossectional area of vapor core of the heat pipe.

Boundaries and Fixed Conditions

At the right side, a Neumann boundary condition was applied to the temperature:

$$\frac{\partial T_w}{\partial x} = 0 \text{ at } x = L_c, \tag{9}$$

at the left side, a constant temperature was applied:

$$T_w(x) = T_v \text{ at } x = 0. \tag{10}$$

The vapor temperature is considered constant over the entire length of the pipe, and all cases were simulated with T_{amb} of 20 °C and h_{int} of 3000 W/m²K.

The output result is a given value of n_g which causes a fixed temperature difference between the pipe with and without the NCG. This fixed temperature, was chosen calculating the minimal temperature difference capable to be measured with a T type thermocouple, and is given by:

$$\Delta T_{NCG,noNCG} = 1.470 \pm 0.005 \text{ [°C]}, \tag{11}$$

and the value of n_g that caused such difference in temperature is obtained by interpolation.

To generalize the numerical data and make them comparable with experimental data, the results are presented related with ambient temperature. The ΔT_v is given by the temperature difference between vapor and ambient:

$$\Delta T_v = T_v - T_{amb}. \quad (12)$$

Table 2 presents the thermophysical properties of the working fluids under saturated conditions at the selected temperature.

Table 2 - Thermophysical Properties of Working Fluids.

Working Fluid	R_v[J/kgK]	λ[kJ/kgK]	k_l[W/mK]
Acetone	143.1	540	0.180
Ammonia	488.2	1371	0.417

Numerical procedure

An uniform axial grid with 5500 nodes was used for the heat pipe, end cap and filling tube. The fully implicit discretized energy equations resulted in a tridiagonal system, which were solved using a tridiagonal matrix algorithm (ANDERSON, 1995).

The numerical procedure to calculate the temperature profile along the heat pipe was performed in FORTRAN as follows:

1. Input the model parameters and properties.
2. Input the initial conditions.
3. Initialize the temperature and pressure.
4. Calculate the length of blocked zone and set the heat transfer coefficient.
5. Solve the heat equation for the combined wall-wick set.
6. Update temperature and pressure.
7. Repeat steps 4 to 6 until convergence.

To return the final value of n_g for each combination of h_f and ΔT_v, the steps above are performed first for two very small guessed values of n_g, and then $\Delta T_{NCG,noNCG}$ is measured. If the current value of $\Delta T_{NCG,noNCG}$ satisfies the Equation (11) the value of n_g is registered, otherwise it is interpolated and the model is reevaluated until Equation (11) is satisfied.

3. Results and Discussion

Figures 4 to 6 shows level maps of n_g with variable convection coefficient h_f and variable vapor temperature difference from ambient ΔT_v. The simulations were performed considering an aluminum HP, with a grooved wick, and two different working fluids separately, acetone and ammonia. The NCG was modeled as an ideal gas.

Figure 4 shows the level map of n_g for a grooved aluminum heat pipe charged with acetone with an ambient temperature T_{amb} of 20°C. The level map clearly show the lines pointing to a minimum around h_f of 400 W/m²K and ΔT_v of 10°C.

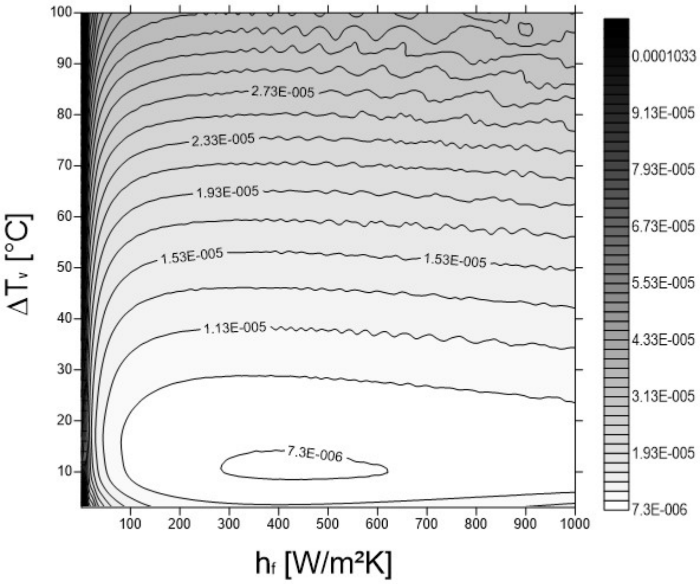

Figure 4. Level map for the omega grooved heat pipe, charged with acetone, aluminum shell and ideal gas.

Figure 5 shows the level map of n_g for the same grooved aluminum heat pipe, but this time charged with ammonia, with an ambient temperature T_{amb} of 20°C. The level map show a small displacement of the minimum in comparison with Figure 4. The n_g levels show that the capability to detect the NCG is drastically reduced due to the high partial pressure of ammonia vapor. A small quantity of NCG is detected more easily on the acetone pipe.

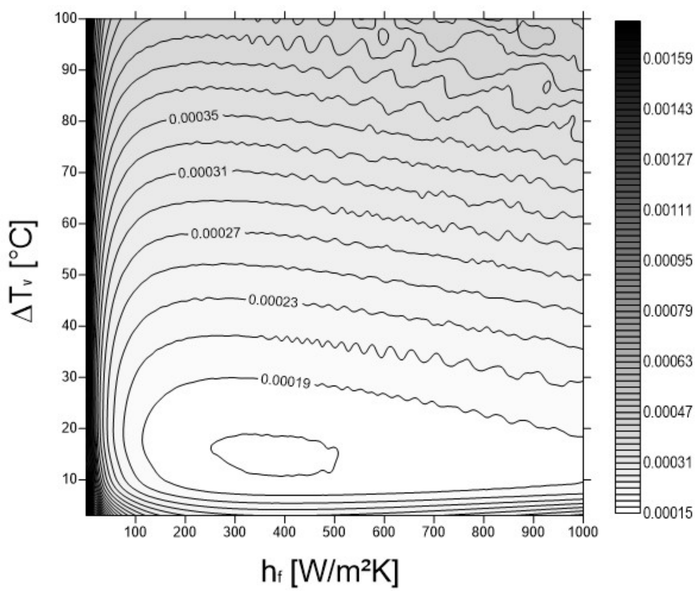

Figure 5. Level map for the omega grooved heat pipe, charged with ammonia, aluminum shell and ideal gas.

Figure 6 shows the level map of n_g for a grooved aluminum heat pipe charged with ammonia and this time with T_{amb} of -50°C for cryogenic tests. The level map show the lines pointing to a minimum above h_f of 250 W/m²K and ΔT_v around of 10°C. The cryogenic test increased significantly the sensitivity of NCG detection.

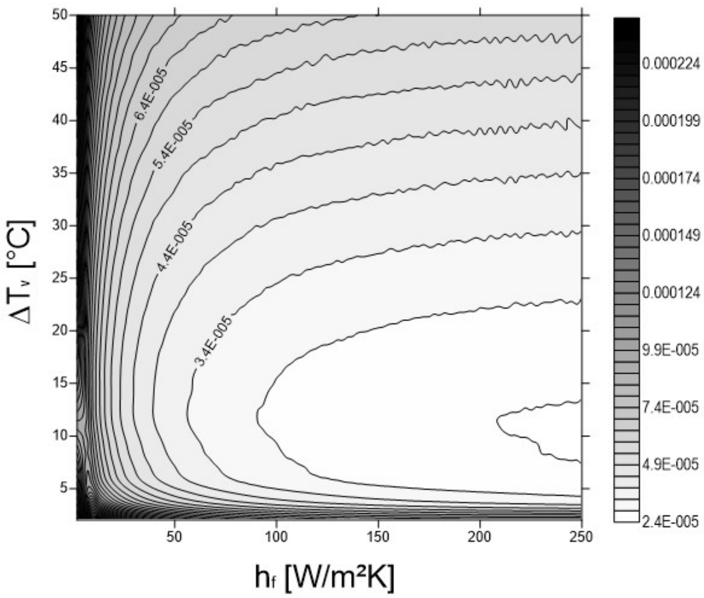

Figure 6. Level map for the omega grooved heat pipe, charged with ammonia, aluminum shell and ideal gas for the cryogenic test.

4. Conclusion

A numerical model capable to simulate the temperature distribution of a heat pipe condenser with and without the presence of noncondensable gas was developed, the model is also capable to return the minimum quantity of noncondensable gas is possible to be detected for a given pair of h_f and ΔT_v values.

Using the numerical model with varying h_f and ΔT_v and different working fluids, a series of level maps were presented for different levels of n_g. From the level maps it was shown, that a minimum detectable n_g exists and a future procedure to find the optimum can be performed. It was also shown that the minimum NCG quantity detectable can be different for different working fluids at the same test conditions.

This novel results have the potential to reduce time and cost of HP life tests, can also improve the search of different pairs of working fluid and container and the numerical model can be used to reverse detect the presence of NCG in heat pipes already in use using temperature measurements.

Other parameters like the ambient temperature, especially in tests being made inside a nitrogen flooded chamber, may be an additional test parameter to be considered when the normal ambient laboratory temperature does not provide the adequate condition for the detection ability of small NCG amount.

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