Numerical and experimental investigations of a grooved heat pipe in microgravity-like conditions

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Abstract (version française en appendice)

In collaboration with Euro Heat Pipes (EHP), we continuously improve a one dimensional thermohydraulic model able to predict the heat transport capacity of grooved heat pipes for microgravity and gravity assisted applications. The code is used through a Java user-friendly interface easing the heat pipe design of the space industries dealing with this technology. After a short description of the code, this paper proposes a comparison between numerical results and experimental measurements obtained on a re-entrant grooved heat pipe in microgravity-like conditions. To reproduce the 0-g environment, a smart setup, based on the rotation of the heat pipe, was built up. By investigating the evolution of the maximum power versus the heat pipe load, we found both good qualitative and quantitative agreement between the code and the experiment.

1. Introduction

Since several decades, the heat pipe technology has proven its efficiency in the thermal control of highly dissipative equipments such as electronic components of satellites. Among conventional heat pipes, grooved heat pipes are often used for spacecraft thermal regulation as they offer a high capillary pumping action, allowing for an easy motion of the fluid between the radiative panels and the dissipative components of the central unit. As power increase and size reduction are constantly asked in the space industry, geometric optimisation of the designed product appears as a real challenge for industrials involved with this technology. Therefore, many industries active in the space domain show an interest to develop numerical tools allowing for the sizing, the characterisation or the risk failure prediction of the heat pipes and avoiding a too large amount of experiments.

Moreover, in case of space applications, microgravity experiments are rare and expensive, justifying the use of numerical prediction tools. At the same time, many industrial efforts are done to match as well as possible a microgravity environment to test the heat pipes on earth. EHP develops a smart and low-cost experimental setup, reproducing in a good way the 0-g conditions.

The paper describes a numerical model able to investigate the grooved heat pipe working. The code is used through a graphical user interface developed in Java and both code and interface are dedicated to an industrial use. They are described in the second section of this paper. The third section deals with the experimental setup developed by EHP to investigate the grooved heat pipe performances in a microgravity-like environment. Finally, we show experimental results and compare them with the code predictions in the last section.

2. The grooved heat pipe modelling approach

The maximum heat transport capacity of the heat pipe is estimated by a stationary semi-analytical hydraulic model. Since the heat pipe operation mainly depends on the axial capillary action, allowing for the motion of both phases in a passive way, and since the grooved heat pipe length is often one or two orders of magnitude larger than its diameter, we made the choice of a one-dimensional modeling approach. Similar models have already been suggested. Ref. [1] gives a detailed mathematical model similar to the one we propose.

A partial derivative form of the one-dimensional momentum and mass conservation equations in a single groove by a Runge-Kutta method. The physical model relies on the equilibrium between the friction losses induced by the liquid and the vapor motions, the external forces such as the gravity and the capillary pressure developed in the groove:

$$\frac{dp_v}{dx} - \frac{dp_l}{dx} + \rho g = \frac{\sigma}{{r_c}^2}\frac{dr_c}{dx} \tag{1}$$

As a solution, the code gives the evolution of the capillary radius r_c inside of the considered groove along the pipe *x*-axis, which is divided by around 500 discretization steps. To solve equation (1), an initial condition must be defined. We impose a given initial capillary radius at the condenser beginning $r_{c,ini}$, which is similar to impose a given fluid volume in the heat pipe. A convergence criterion must also be added. As we plan to calculate the maximum heat transport capacity, we imposed the minimum capillary radius is reached at the end of the evaporator.

In that case, any power increase would lead to the dry out of the grooves and the failure of the heat pipe operation [1]. An iterative power loop is then implemented in order to find the maximum heat transport. Assuming the power (or the mass flow), we solve the equation (1) by a Runge-Kutta method, check if the convergence criterion is verified and, if not, tune the power value until the criterion is satisfied.

Capillary pressure is described by the Young Laplace equation with the assumption of an infinite curvature radius in the axial direction. A zero contact angle is assumed which is a fairly good approximation for fluids such as ammonia. Friction losses are defined by one dimensional correlations based on a fully-developed flow [2,3,4]. Laminar regime is considered in the liquid phase and both laminar and turbulent regimes are supposed for the vapour flow as the Reynolds number can reach values up to 3000 at high temperatures. In addition to the viscous effects in both phases, the code considers an inertial contribution in the vapour phase loss term due to the mass addition or removal in the evaporator and the condenser [5,6]. This effect is neglected in the liquid phase due to the lower velocity inside of the liquid. The code also considers a shear stress at the liquid/vapour interface due to the counter flow of the liquid and the vapour.

Most of the one dimensional heat pipe models use correlation formula established in the literature for circular closed channels. They define an equivalent diameter in order to take into account their specific geometry. Nevertheless, it can be demonstrated this leads to errors that we cannot neglect. In order to improve the friction loss correlations inside of the grooves, we have implemented a method allowing for the estimate of the friction losses for any kind of duct shapes, opened or not [1,7]. More mathematical details can be found in Ref. [1].

The code execution is fast (no more than 5 minutes on PC at 1 GHz). It is therefore suitable for a geometric optimisation and an industrial design which require a quick result and numerous calls to the code. These criteria limit the use of full computational fluid dynamic calculations.

3. Experimental setup for microgravity-like

EHP designed a system dedicated to the measurement of the maximum heat transport capacity of the heat pipe with respect to its operating temperature and the evaporator and condenser heat transfers.

Under standard static gravity tests, the appearance of an excess of liquid in the bottom of the pipe is due to the gravity field applied on the grooves located on the top of the profile. This excess, known as a puddle, has a key role in the power transport, which renders a poor comparison with the microgravity behaviour, in terms of performances and friction losses. Therefore, in order to reduce the differences with the liquid distribution in 0-g, we applied an axial rotation to the heat pipe. After different tests on a reference heat pipe, a velocity of 18 turns every minute was chosen. This guarantees a negligible influence of the centrifugal effects, appearing above 60 turns every minute and masking the correct working of the pipe.

The bench is 3.5 m long and is able to test heat pipes until 3 m long and 40 mm wide. The heat pipe is positioned horizontally by the way of supports into a larger rigid tube with an absolute error on the horizontality lower than 0.5 mm. In order to decrease as much as possible the heat losses to environment, the tube is filled with insulation material and heated to be at the heat pipe operating temperature. A rotating fluid connector is placed at the heat pipe condenser side. This connector transfers the cooling fluid to the heat pipe heat exchanger. A PID controller regulates the secondary cooling loop temperature in order to obtain the desired heat pipe operating temperature.

In order to measure the heat pipe performance, a heater is located at one side over a 400 mm distance and a heat exchanger cools over the same distance the opposite side of the 3 m long heat pipe. Many temperature sensors are placed along the heat pipe to assess its behaviour with respect to heat load. The data recorder is in rotation with the heat pipe and the data are transferred by a slip ring electrical connector to the computer. The maximum heat load performance of the heat pipe is assessed according to ESA PSS49 [8] when the plot of the mean evaporator temperature minus the adiabatic section temperature versus the increasing evaporator heat load sensibly departs from a straight line, meaning a clear degradation of the evaporator heat transfer.

4. Results

Grooved heat pipes are commonly used in the spacecraft thermal control. For example, they are positioned under the satellite platform in order to bring the heat load from the platform to the radiative panels. In such applications, they can be several meters long and their external diameter has an order of magnitude of 10 millimetres. This is the case of the heat pipe investigated in this paper. EHP manufactured a re-entrant grooved heat pipe, named AG190. In the re-entrant configuration, the grooves are composed of a slot, which ensures a good capillary action, and a hydraulic duct attached at its end, which reduces the friction losses. More the slot is narrow, more the capillary pressure increases. Nevertheless a reasonable opening must be kept in order to allow for a bubble escape from the groove to the vapour core.

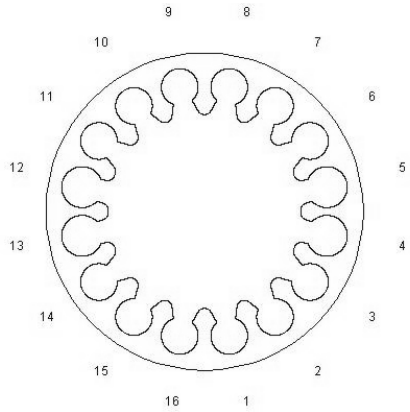

Figure 1: *Section of the AG190 re-entrant heat pipe, obtained with the Java interface.*

The AG190 heat pipe is composed of two types of grooves. Its section, drawn by the Java interface, is shown in Figure 1. Twelve grooves have a narrow slot opening, thus ensuring a good capillary pumping action. Four larger grooves are also added (indexes 4,5,12 and 13 in Figure 1) in order to leave the liquid to fill the smaller grooves in the case of a great liquid contraction at low temperatures. At high temperatures or large filling rates, a part of the liquid excess is stored inside of the largest grooves. It therefore reduces the slug size and allows for a better heat pipe working over a larger temperature range. According to an imposed criterion, the heat pipe can be filled with a maximum excess of liquid corresponding to a 25 mm maximum liquid slug length at 75°C, for a 1 m long heat pipe.

Figure 2 illustrates the experimental data obtained for this heat pipe for a working temperature of 75°C (see the hollow circle curve). It shows the evolution of the maximum heat power versus the fluid volume introduced in the heat pipe. It is called the load curve. In the same figure, we present two curves obtained with the code for two different sets of ammonia thermohydraulic parameters, coming from two different sources, EHP (see the dotted curve) and A. Faghri [2] (see the dashed curve). These parameters are summarised in Table 1. The EHP fluid parameters are the results of a compilation between different sources from ESA, NIST, VDI Faghri [2] and Dunn et al. [4].

	EHP	A. Faghri
Latent heat (10^3 J/kg)	906	895
Liquid density (kg/m^3)	515	512
Vapour density (kg/m^3)	29.8	31.3
Liquid viscosity (10^{-6} Ns/m^2)	81.9	83.2
Vapour viscosity (10^{-6} Ns/m^2)	11.8	13.7
Surface tension (10^{-3} N/m)	8.66	9.60

Table 1: *Thermohydraulic properties of ammonia at 75°C, according to two different sources.*

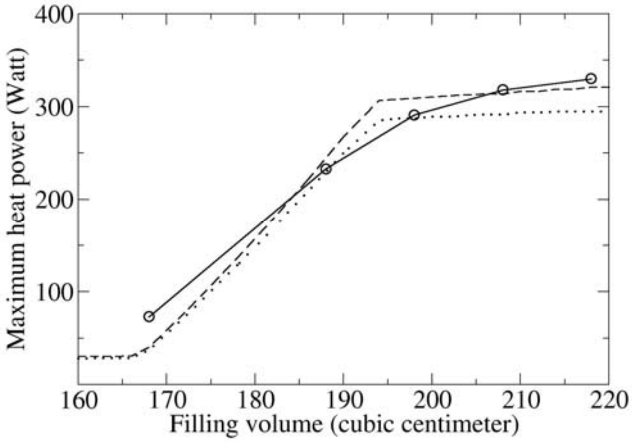

Figure 2: *Load curve: maximum heat power versus the ammonia volume introduced in the heat pipe (hollow circle curve: experimental data; dotted curve: numerical results for the EHP ammonia parameters; dashed curve: numerical results for the A. Faghri ammonia parameters).*

We find a very good agreement between the numerical results and the experimental data, proving the accuracy of the physical model implemented in the one dimensional code. Indeed, both the general behaviour of the experimental curve and the values of the maximum heat powers are reproduced with the code.

In one hand, the general behaviour is well reproduced by the code. Below 165 cm^3, the heat pipe load is too small to develop a high power. The grooves are not fully filled at the end of the condenser and the capillary pressure gradient developed between the evaporator and the condenser ends is too small to balance important friction losses, proportional to the power. This is no more the case in the range [165-195] cm^3 in which the power increases from 30 W to 300 W. This important variation is well reproduced by the code. In that range, the grooves are able to develop a better capillary pressure along the pipe axis. Progressively, a liquid excess appears in the condenser as a slug. Beyond 195 cm^3, the slug becomes more and more important, reducing the condenser length and heat transfer capability. The code shows a power saturation in that range. Nevertheless, the experimental curve presents a remaining small power increase. Since the heat pipe rotation velocity is low, the large excess of liquid can not be stored in a perfectly shaped slug as in microgravity. The gravity action induces a partial localization of the fluid in the bottom part of the pipe. The gravity action on this puddle is favourable, explaining the slight power increase above 195 cm^3.

In another hand, the calculated power values are also in agreement with the experimental data, proving the code accuracy despite its one dimensional characteristics. Two arguments can explain the differences between the numerical and the experimental data. First of all, we must mention the great sensitivity of the results to the thermohydraulic parameters used to describe the ammonia. By using two referenced parameter sets (see Table 1), we found important differences in the code results. This is obviously not a limitation of the one dimensional model. Therefore, as we do not know the properties of the introduced fluid, this renders difficult the comparison between the numerical and the experimental data in terms of relative error percentage for example. Moreover, in the [165-195] cm^3 range, the result is extremely sensitive. In fact a relative error of 1 per cent on the volume of fluid introduced in the heat pipe can lead to a relative error close to ten per cent on the power estimate. This might for example explain the lack of accuracy at the 168 cm^3 experimental point.

5. Conclusion

In this paper, we described a one dimensional hydraulic model able to predict the maximum heat transport capacity of grooved heat pipes for microgravity applications. The code relies on the equilibrium between the friction losses and the capillary pressure developed inside of the grooves. It is used through a graphical Java interface that we developed in order to ease the heat pipe design in an industrial framework. We described an experimental setup, based on the heat pipe rotation, able to reproduce microgravity-like conditions. We proposed a comparison between numerical results and experimental measurements performed on a re-entrant grooved heat pipe with this setup. Taking into account the one dimensional character of the code, we found a good qualitative and quantitative agreement between both kinds of results, which justify the use of the code in an industrial design purpose.

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APPENDIX : version française du résumé

En collaboration avec Euro Heat Pipes, nous développons un modèle thermohydraulique unidimensionnel capable d'estimer la puissance maximale transportée par un caloduc à rainures en microgravité et pour des applications terrestres. Après une description du code, cet article propose une comparaison entre les résultats numériques et les mesures expérimentales obtenus sur un caloduc à rainures réentrantes en condition de pseudo-microgravité. Pour reproduire l'environnement 0-g, une installation inventive, basée sur la rotation du caloduc, a été mise au point par EHP. En étudiant l'évolution de la puissance maximale vis-à-vis de la charge du caloduc, nous montrerons qu'il existe un bon accord qualitatif et quantitatif entre le code et l'expérience.