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THE EFFECT OF EXTERNAL BOUNDARY CONDITIONS ON CONDENSATION HEAT TRANSFER IN ROTATING HEAT PIPES

T. C. DANIELS* and R. J. WILLIAMS†

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Abstract—Experimental evidence shows the importance of external boundary conditions on the overall performance of a rotating heat pipe condenser.

Data are presented for the boundary conditions of constant heat flux and constant wall temperature for rotating heat pipes containing either pure vapour or a mixture of vapour and non-condensable gas as working fluid.

1. INTRODUCTION

OVER the past decade many major engineering institutions have devoted considerable time and effort to the study of heat pipes. These studies have mainly been directed towards the stationary heat pipe which relies on capillary action to re-cycle the working fluid. Another type of heat pipe which relies on centrifugal forces to return the condensate has also come under investigation [1, 2]. The rotating heat pipe is a closed hollow shaft with a slight internal taper along its length and contains a fixed amount of working fluid. Heat is transferred from the evaporator to the working fluid causing evaporation. The vapour flows axially towards the condenser where it is condensed on the cooled walls. It is then centrifuged back to the evaporator along the tapered walls. The limitations imposed by the failure of the wick structure are therefore eliminated by the use of the centrifugal force field to return the condensate to the evaporator. This device is very attractive for cooling of components where rotation is inherent in the system, such as heavily loaded bearings or gears, the cooling of electric motors [3] etc.

Detailed theoretical and experimental analysis of rotating heat pipes has been carried out and presented in previous papers [4, 5].

Whilst the importance of external boundary conditions has been noted by previous research workers [6] no systematic investigations have been carried out on its precise effect. The present work thus constitutes an extension of the work of [5] and will rely heavily on this reference for a detailed explanation of the experimental apparatus. The investigation evaluates the performance of the rotating heat pipe under various external boundary conditions, both with and without non-condensable gas present.

*Department of Mechanical Engineering, University College of Swansea, Singleton Park, Swansea, SAZ, 800, England, and

†NASA-Ames Research Center, Moffett Field, California, U.S.A.

2. EXPERIMENTAL RESULTS

The experimental apparatus used for this investigation is shown in Fig. 1 (see Ref. (5) for a detailed description). Briefly the heat pipe consists of a 76 mm O/D × 150 mm copper evaporator with a 225 mm long condenser tapering from 63.7 mm producing a 3° half cone angle. Thermocouples embedded in the evaporator and condenser walls provide wall temperature data while a travelling thermocouple probe enables the internal temperature distribution of the heat pipe to be measured. The condenser has been segmented into six compartments (each with its own individual spray cooler) and provision has been made in each one to record inlet and outlet cooling water temperatures and flow rates. It is therefore possible, by adjusting the individual flow rates to each compartment, to obtain various conditions such as uniform wall temperature or uniform heat flux removal.

3. PURE VAPOUR ONLY

The pure vapour tests were carried out with an overall cooling water flow rate of $300\,\mathrm{cm^3\,min^{-1}}$. However as the surface area of each condenser compartment decreases along the axial length of the condenser, the cooling water flow rate to each compartment was set proportional to the surface area of that compartment so that sink conditions for each compartment would be the same. The working fluid used for these tests was Refrigerant 113 (CCL_2F–$CCLF_2$) and the results obtained in Fig. 2 show the relationship between the local heat transfer rates Q of the condenser and the temperature difference (Θ_s) between the saturation temperature and the condenser wall temperature. The theoretical analysis of [4], which is based on a boundary layer type approach, is shown for comparison. The experimental points are the average of the local heat flux of compartments 2–5 (compartments 1 and 6 have been neglected in order to eliminate end effects). The variation of local heat flux along the condenser was approximately 10%, good repeatability was obtained with the experimental data with approxi-

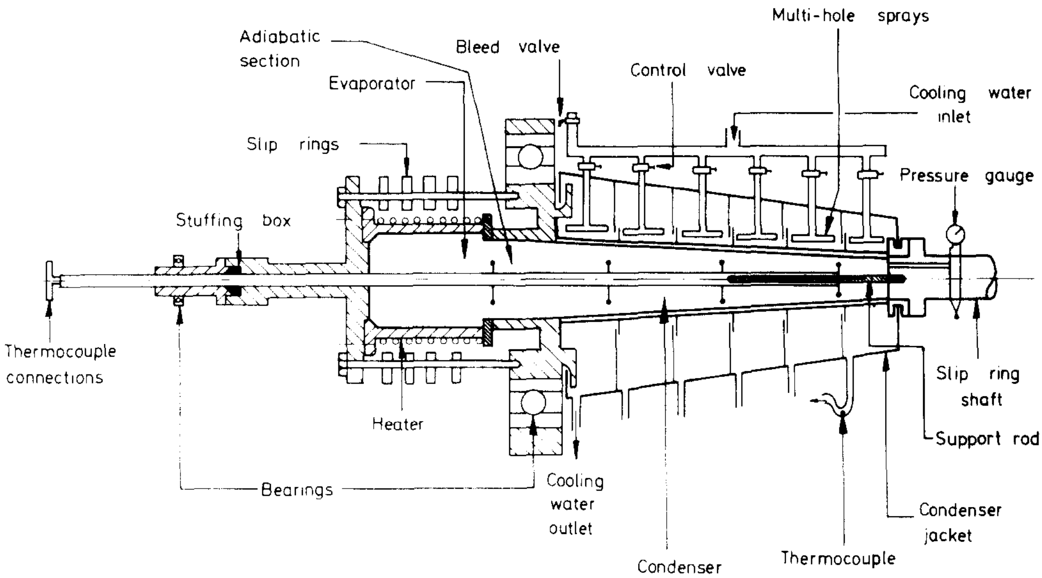

FIG. 1. Schematic diagram of rotating heat pipe.

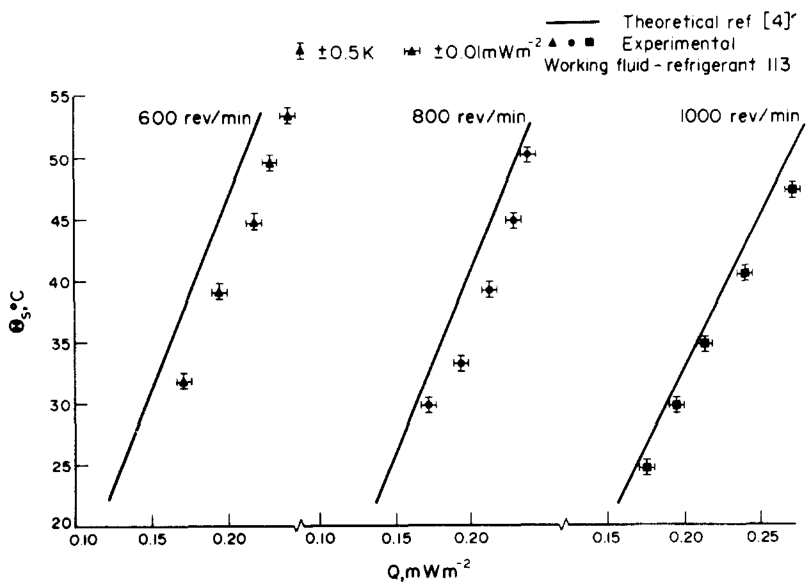

FIG. 2. Comparison of analytically predicted heat transfer rates with experiments.

mately 80% of the heat input to the evaporator being recovered in the condenser cooling water. The remaining 20% was probably lost as the heat leaks through the insulation and by some condensation in the adiabatic section of the heat pipe.

As stated, these data were produced with the water flow proportional to the individual area of each compartment, this procedure producing a uniform wall temperature in the condenser. It was decided at this point to investigate the boundary condition of uniform heat flux removal by adjusting the individual water flows, to ascertain if this had any significant effect on heat pipe operation. The results obtained were found to lie on the same experimental curve as the previous results, however there was a wall temperature variation of approximately 10%.

The flexibility of control of the cooling water flow rates prompted a further investigation. It was decided to increase the overall flow rate in stages, keeping the individual flow rates proportional to area and monitor the heat transfer characteristics of the heat pipe. As the flow rate was increased between $300\,cm^3\,min^{-1}$ and $400\,cm^3\,min^{-1}$, little change occurred in the heat pipe characteristics, i.e. a uniform wall temperature profile was preserved and the results were in good general agreement with those obtained previously with the theoretical results. Between $400\,cm^3\,min^{-1}$ and $500\,cm^3\,min^{-1}$ a non-uniform axial temperature profile was set up in the condenser, a temperature gradient existing from the large to the small end of the condenser accompanied by an increased variation of the local

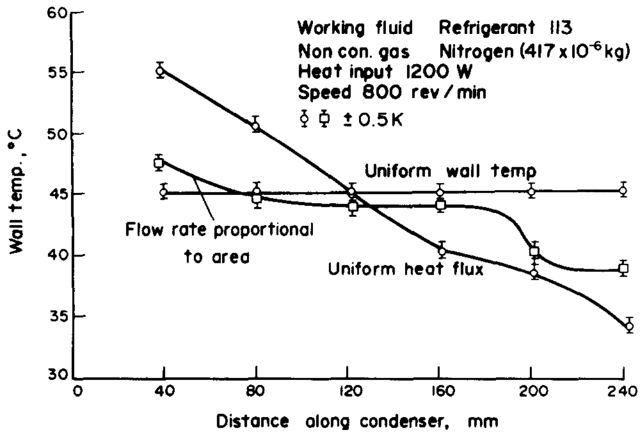

FIG. 3. Uniform wall temperature and uniform heat flux profiles.

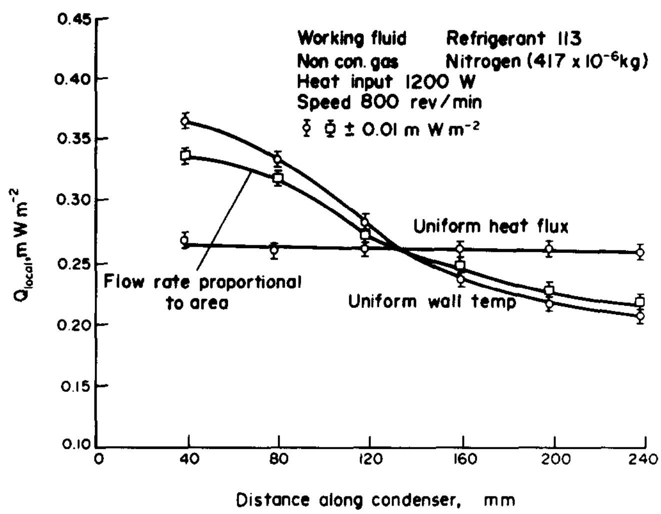

FIG. 4. Uniform wall temperature and uniform heat flux profiles.

rates of heat transfer. This situation gradually deteriorated until at approximately $600\,cm^3\,m^{-1}$ most of the heat (i.e. approximately 85% of that being transferred) was being removed in the first two compartments where the heat flux was $5.6\,W\,cm^{-2}$. This was estimated from the fact that about 80% of the heat input 1200 W was transferred to the condenser and that 85% of this is transferred in the first two compartments of area $145\,cm^2$. The heat transferred to the cooling water in compartments five and six was of the order of 10–15 W which was probably due to just the conduction down the wall of the condenser. Agreement between the experimental results for compartments one and two and theoretical values was poor with experimental temperature differences higher than the theoretical values over the range of heat transfer rates, i.e. lower heat transfer coefficients than those predicted by theory. Physically this may be interpreted in the following manner: the large cooling water flow rate employed produces a very efficient "cold sink" and this sink enables the heat to be removed over the first few centimeters of the condenser by virtue of its high external heat transfer coefficient. On entering this relatively cold region the dryness fraction of the vapour may change, and this wet vapour would then be thrown outwards on to the condenser walls by centrifugal action and very little vapour would venture up to the far end of the condenser. Similar conditions to this may occur during start up since, initially, condensation takes place only over the entry section of the condenser; as more vapour is liberated by the boiling process more of the condenser will come into use. If sufficient heat flux were available from the evaporator then more of the condenser would have to be used for steady state conditions. Unfortunately the present system is limited to a heat input of 2 kW due to the high wall temperatures in the evaporator and cooling problems of the main bearing.

4. NON-CONDENSABLE GAS

In order to explore the effects of various boundary conditions on a rotating heat pipe containing non-

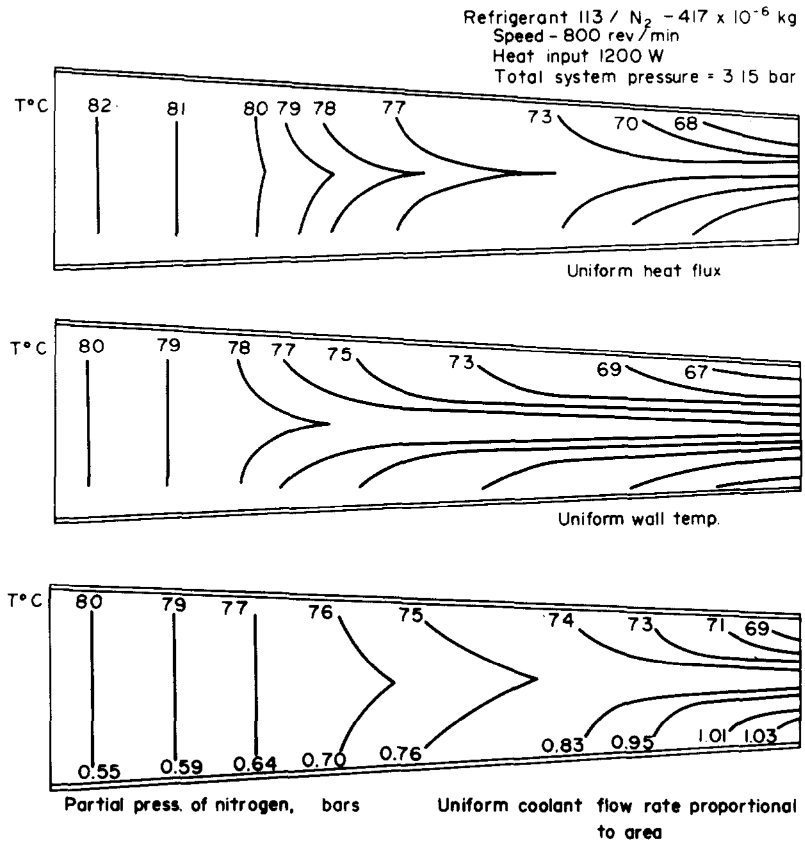

FIG. 5. Comparison between uniform heat flux and uniform wall temperature, and uniform coolant flow rate proportional to area.

condensable gas 417×10^{-6} kg of nitrogen was introduced into the system. It has already been shown [5] that in many ways the effect of non-condensable gas on a rotating heat pipe is similar to its effect on a stationary heat pipe and that the ratio of molecular weights (working fluid to non-condensable gas) has an important effect. With most stationary gas-loaded heat pipes the condenser boundary conditions are not rigidly controlled. However two distinct temperature regions exist, a high temperature region indicative of an active condenser and a lower temperature region indicative of an inactive gas blocked condenser.

Three separate tests were conducted, the first with cooling water flow rates proportional to area, the second with the boundary conditions of uniform heat flux and the third with the boundary condition of uniform wall temperature. The overall water flow rate was kept constant for these tests and the individual sprays adjusted to give the required conditions. The total system pressure remained approximately constant for these three tests at 3.15 bar. The wall temperature and heat flux profiles from these tests are shown in Figs. 3 and 4. The flow rate proportional to area case produces a temperature profile which is similar to those encountered in stationary gas loaded heat pipes. However, as expected, the uniform heat flux case produces a gradually decaying wall temperature profile while the uniform wall temperature case produces a gradually decaying heat flux profile. By traversing the thermocouple probe along the inside of the condenser, the internal isotherms of the vapour/non-condensable gas mixture can be mapped (in Fig. 5). The temperature of the vapour gas isotherms is given along the top of each diagram while the pressure along the bottom of the last diagram gives an indication of the partial pressure of the non-condensable gas. The figure shows that while the external conditions of the heat pipe may have changed, the internal behaviour of the heat pipe is not significantly altered. A more complete discussion of the internal isotherms obtained by this method is given in [5].

CONCLUSIONS

The external boundary conditions play an important role in the behaviour of a rotating heat pipe. By maintaining an efficient spray cooling mechanism which wets the entire condenser area, higher heat transfer coefficients can be obtained. However there appears to be an upper cooling water flow rate above which dry saturated conditions cease to exist over the entire condenser length.

Finally, there appears to be little difference in the internal behaviour of the rotating heat pipe when subjected to the external boundary conditions of constant heat flux, constant wall temperature or flow rate proportional to area.

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EFFET DES CONDITIONS AUX LIMITES EXTERNES SUR LE TRANSFERT THERMIQUE AVEC CONDENSATION DANS LES CALODUCS ROTATIFS

Résumé—Une évidence expérimentale montre l'importance des conditions aux limites externes sur la performance globale d'un condenseur de caloduc tournant. Des résultats sont présentés pour les conditions de flux constant et de température pariétale constante pour des caloducs rotatifs qui contiennent soit une vapeur pure, soit un mélange de vapeur et de gaz non-condensable comme fluide actif.

DER EINFLUSS ÄUSSERER RANDBEDINGUNGEN AUF DEN KONDENSATIONS-WÄRMEÜBERGANG IN ROTIERENDEN WÄRMEROHREN

Zusammenfassung—Das Experiment zeigt den bedeutenden Einfluß äußerer Randbedingungen auf die Gesamtleistung eines rotierenden Wärmerohr-Kondensators. Für die Randbedingungen konstante Wärmestromdichte und konstante Wandtemperatur werden für rotierende Wärmerohre Versuchswerte angegeben; Arbeitsmittel ist sowohl reiner Dampf wie auch ein Dampf-Inertgas-Gemisch.

ВЛИЯНИЕ ВНЕШНИХ ГРАНИЧНЫХ УСЛОВИЙ НА ПЕРЕНОС ТЕПЛА ПРИ КОНДЕНСАЦИИ ВО ВРАЩАЮЩИХСЯ ТЕПЛОВЫХ ТРУБАХ

Аннотация — Результаты экспериментов свидетельствуют о влиянии внешних граничных условий на эффективность конденсатора вращающейся тепловой трубы. Представлены данные для двух видов граничных условий: постоянной плотности теплового потока и постоянной температуры стенки, для вращающихся тепловых труб, в которых в качестве рабочей жидкости используется или чистый пар, или смесь пара с неконденсирующимся газом.