HEAT EXCHANGE IN THERMOSIPHONS IN THE FIELD OF INFLUENCE OF CENTRIFUGAL AND CORIOLIS FORCES

L. M. Zysina-Molozhen and G. M. Tkachenko

UDC 536.24

Results are given of experimental investigation of the mechanism of movement and heat exchange in closed rotating channels, which form models of the cooling channels of the working blades of a gas turbine installation.

Liquid cooling using closed channels, the upper end of which is heated in the vane of the blade, and the lower end of which is cooled in the root, is considered to be one of the progressive methods of cooling the blades of high temperature gas turbines at the present time. As a result of the difference of the densities, which is determined by the presence of longitudinal pressure and temperature gradients, an intensive convective movement of the liquid occurs in the channel, which creates favorable conditions for heat exchange.

In order to carry out an analytical solution of the problem, it is necessary to have an idea of the nature of the movement of the liquid inside the channel. Owing to the extraordinary difficulty of setting up such experiments a number of hypotheses were proposed in the literature concerning the nature of the movement, based on observations of the movement of liquid in stationary channels [1-3].

The present work gives the results of an experimental investigation of the mechanisms of movement of liquid in a blind channel in conditions of rotation. The solution of the stated problem was preceded by

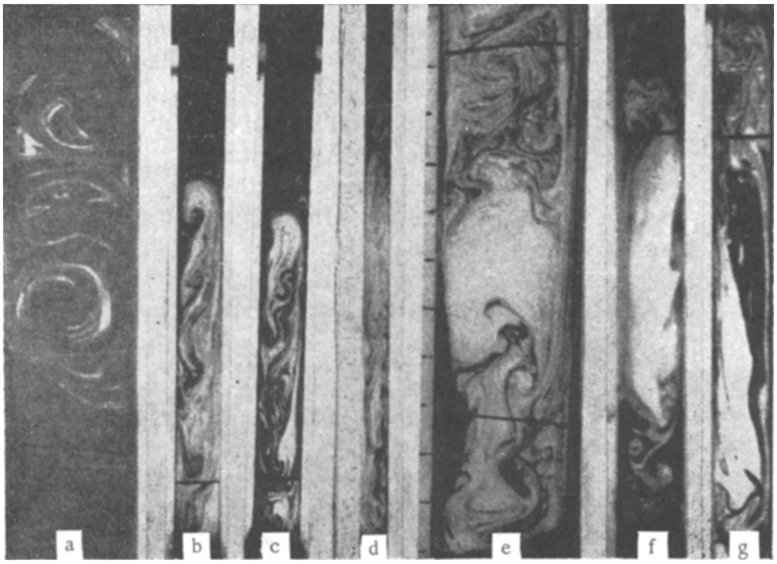

Fig. 1. Pictures of the movement of liquid in rotating thermosiphons: a) $\overline{l}$ = 4.3; n = 100 rpm; b, c) $\overline{l}$ = 12, n = 200 rpm; d) 30 and 250 respectively; e) 4.3 and 360; f) 12 and 500; g) 12 and 390.

I. I. Polzunov Boiler Turbine Institute, Leningrad. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 20, No. 1, pp. 157-160, January, 1971. Original article submitted May 15,1970.

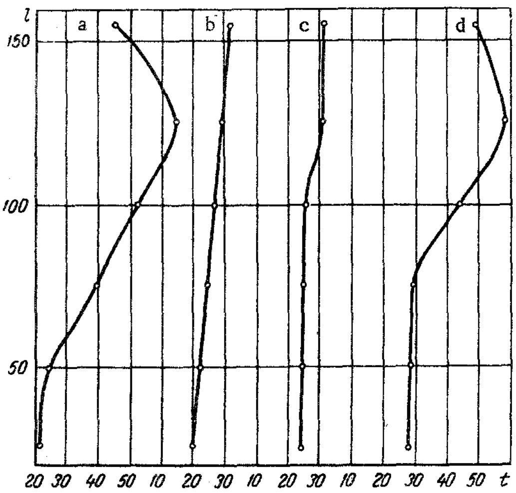

Fig. 2. Temperature distribution over the height of the channels: a) air, $\bar{l} = 18$; n = 390 rpm; b) water, $\bar{l} = 18$, n = 500 rpm; c) water, $\bar{l} = 18$, n = 920 rpm; d) air, $\bar{l} = 30$, n = 100 rpm.

great deal of work in developing the method of flow visualization and in creating an experimental installation and equipment. The method of electrolytic coloring of the liquid using tellurium hydrosols based on the electrophoresis phenomenon was used as the visualization method. The experiments were carried out with different lengths and diameters of the channels, so that their relative lengths $\bar{l} = l/d$ varied in a range $\bar{l} = 3.9$–30 (hence 150 mm $\leq l \leq$ 180 mm; 3 mm $\leq d \leq$ 30 mm).

The rotatory test rig included a model of the cooling channel which was radially fastened to the shaft of an electric motor. Electric heaters were located in the "upper" part of the channel over the length of the heating section l_n = 70 mm, the "lower" part was cooled by a stream of air. During rotation of the working parts recording of the picture obtained was carried out by using synchronous illumination by pulse lamps of the type IFK-200 with the cine camera AKS-2 or by conventional photo equipment. The temperature of the liquid in the channel, the temperature of the walls and the amount of heat supplied were recorded during the experiments. Curves for the distribution of temperatures and heat flows were plotted from these data, and the values of the parameters Nu, Nu*, λ_{eff} etc. were determined.

Figure 1, a–g gives typical pictures of the flow of liquid in thermosiphons of this kind at different speeds of rotation for channels of different diameters. By examining these it is possible to establish two characteristic systems of flow:

1) intensive helical flow in the center of the flow over the whole height of the channel with well defined flow in the boundary layer at the walls (Fig. 1, a, b, c, d). In this case, as can be seen from curves a and b in Fig. 2, the temperature varies continuously over the whole length;

2) the movement, which characteristics the occurrence of a closed region of rotation in the region of the section "zero feed" of heat between the heated and cooled sections of the thermosiphon and closed sections of lateral flows at the ends of the channel (Fig. 1, e, f, g). The characteristic temperature distribution (curves c and d, Fig. 2) corresponds to this system when the whole temperature variation takes place only at the section "zero feed."

Changeover to the second system of flow takes place when the speed of rotation of the channel is increased; the greater the length of the channel $\bar{l}$, the more intensive this changeover for higher rotating speeds. The second system obviously corresponds with the considerable predominance of the influence of centrifugal and Coriolis forces over the influence of gravitational forces.

Treatment of the experimental data for heat exchange also enabled two characteristic systems of heat exchange to be established which correspond with the two abovementioned mechanisms of movement of the liquid in rotating thermosiphons. For the first system of flow all the experimental points can be approximated by the relationship

$$\lambda_l = 3.6\cdot 10^{-2}\,(\mathrm{GrPr})^{0,4}\,\bar{l}^{-0,35}. \qquad (1)$$

This relationship is similar in structure to the known formulas [4-6]. Obviously all these relationships correspond to the first flow system.

The intensity of the longitudinal heat elimination for the given flow system is quite great. It is approximately double the average intensity of heat exchange on the section "zero feed" and it is expressed by the relationship,

$$\lambda_{\text{eff}} = \frac{2{,}25}{\bar{l}} (\text{GrPr})^{0{,}25}. \tag{2}$$

In the case of changeover to the second system of flow the picture of the heat exchange in the thermosiphon varies considerably. Heat transfer directly in the longitudinal direction takes place here only on the section "zero feed," in the remaining sections the temperature of the flow does not vary over their length. The intensity of the heat exchange on the section "zero feed" for this system appears greater than in the case of the first system, and it is approximated by the relationship

$$\text{Nu}^* = 0.98\,(\text{GrPr})^{0{,}29}\bar{l}^{-0{,}35}. \tag{3}$$

Heat exchange between the liquid and the side walls of the thermosiphon also appears different for the first and second system and it is expressed respectively:

for the first system

$$\text{Nu} = 3\bar{l}^{-0{,}7}(\text{GrPr})^{0{,}25}, \tag{4}$$

for the second system

$$\text{Nu} = 0.47\bar{l}^{-0{,}7}(\text{GrPr})^{0{,}29}. \tag{5}$$

Comparison of the formulas (1)-(5) indicates that the intensity and mechanism of heat exchange for the first and second systems of flow are different. The use of formulas obtained from the hypothesis regarding the existence of flows in the channel, observed in nonstationary thermosiphons for calculations, is proportional only for the first system of flow, which corresponds to the low speed rotation $\approx$ (440-700) rpm and to the range of $\bar{l}$ = 4-30 respectively. The second system of flow occurs after this limit. The heat exchange can be evaluated according to formulas (3) and (5) for this system.

LITERATURE CITED

1. M. J. Lighthill, "Theoretical considerations on free convection in tubes," Quart. Journ. of Mech. Appl. Math., 6, No. 4 (1958).
2. A. G. Romanov, Izv. Akad. Nauk SSSR, Otdel. Tekh. Nauk, No. 6 (1956).
3. F. Baylay and G. S. H. Lock, "Heat transfer characteristics of the closed thermosiphon," Trans. ASME, Ser.C, 1, (1965).
4. M. I. Khomyakov, Trans. No. 348, P. I. Baranov Institute (1959).
5. V. L. Ivanov and Yu. D. Lapin, Teploénergetika, No. 6 (1966).
6. J. P. Fraser, and D. J. Oakley, "Turbulent free convection, heat transfer rates in a horizontal pipe," Trans. ASME 7, 79 No. 5 (1957).