6. V. M. Kovalenko, "Some characteristics of the wall region of a boundary layer at a rough plate," Izv. Sib. Otd. Akad. Nauk SSSR, No. 13, Issue 3 (1973).
7. A. I. Yakushev, Interchangeability, Standardization, and Technical Measurements [in Russian], Mashinostroenie, Moscow (1974).
8. K. K. Fedyaevskii and N. N. Fomina, "Investigation of the effect of roughness on the drag and the boundary-layer state," Tr. TsAGI, No. 441 (1939).
9. V. F. Droblenkov, "Turbulent boundary layer at a rough curvilinear surface," Izv. Akad. Nauk SSSR, Otd. Tekh. Nauk, No. 8 (1955).
10. V. P. Mugalev, "Heat transfer and flow at a rough porous surface with blowing," in: Heat and Mass Transfer [in Russian], Vol. 1, Part 2 (1972).
11. K. Bammert and H. Sonnenschein, "Der Einfluss verdickter und verdunnter Turbinenschaufeln auf die Gittereigenschaften," Arch. Eisenhuttenwes., 38, No. 4 (1967).
12. V. M. Epifanov, "Aerodynamic investigation of turbine lattices with thickened outer edges," in: High-Temperature Cooled Gas Turbines [in Russian], Mashinostroenie, Moscow (1971).

EXPERIMENTAL STUDY OF HEAT EXCHANGE IN HELIUM FLOW WITHIN A METAL–CERAMIC TUBE

L. L. Vasil'ev, G. I. Bobrova, and L. A. Stasevich

UDC 536.242

Results are presented of an experimental study of the heat-transfer process in helium flow within a sintered porous bronze tube. The effect of flow suction rate through the porous wall upon temperature distribution and Nusselt number is demonstrated.

The problem of developing effective cooling methods is of great practical importance at the present time. A large number of studies has been presented on intensification of heat exchange, in particular, an intensification by suction of a portion of a flow through a porous channel wall.

A review of the available literature shows that heat exchange for liquid flow in permeable channels of various geometries has been studied to determine whether the heat-exchange process can be intensified by suctioning a portion of the flow through the permeable wall. The major portion of the works published analyze this process theoretically, usually assuming a laminar flow regime. This is apparently because theoretical analysis of a laminar flow is simpler and more concrete than treatment of a turbulent regime. Only a few experimental studies are available.

Analysis of heat exchange for a steady-state laminar flow proceeds from the system of equations of motion, continuity, and energy. Approximately 30 years ago Sellars [1] and Berman [2] demonstrated that the equations of motions for a laminar, completely developed flow with extraction or injection of a portion of the flow through a permeable porous wall may be reduced to a fourth-order nonlinear differential equation.

These studies were then expanded and refined in [3, 4].

Since suction through the wall intensifies heat exchange in flows in porous channels, investigators have concentrated their efforts not only on solving the hydrodynamic problem, but also on determining the effect of suction on temperature distribution over the radius and length of the porous tube.

Thus, in [3] the authors considered the effect of low draft velocities through a permeable tube wall into the main liquid flow within the tube upon the tube wall temperature distribution. In [5] temperature profiles were calculated for a flow moving within a porous tube, with the wall temperature of the tube maintained constant. Raithby [6] studied hydrodynamics and heat exchange for constant wall temperature and thermal flux, and analyzed the effect of individual parameters such as draft, suction, channel geometry, etc. on the temperature and velocity profiles.

A. V. Lykov Institute of Heat and Mass Exchange, Academy of Sciences of the Belorussian SSR, Minsk.
Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 37, No. 3, pp. 419–423, September, 1979. Original article submitted December 16, 1978.

0022-0841/79/3703-1029$07.50

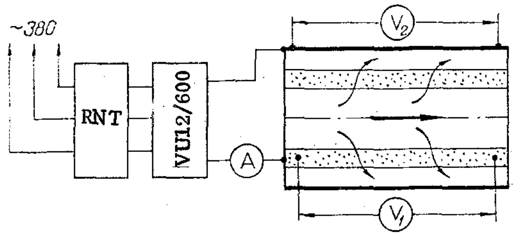

Fig. 1. Diagram of experimental apparatus.

A complete study of the problem of heat exchange in motion of a liquid flow in a permeable tube, i.e., with either suction or draft, is quite complex, and has not been performed fully even for a laminar flow. For the case of turbulent flow, it is still more complex. Suction or draft also has a significant effect on hydrodynamics and heat exchange in turbulent flows.

In particular, [7] examined the effect of surface suction on the characteristics of heat and mass exchange in motion of turbulent flows in tubes. It was shown that suction has a significant effect on Nusselt number, friction coefficient, and the velocity profile.

At the present time, porous heat exchangers are being used ever more widely in electrotechnical and cryogenic applications. Heat-exchange coefficients must be evaluated in their design and construction. Therefore, an experimental verification of the analytically obtained relationships characterizing heat exchange in motion of a turbulent flow in tubes with suction or draft by liquefied gases is of great importance.

There are few experimental studies available to confirm or complement theoretical results, and the total knowledge in this area is insufficient to define a simple and correct method for estimating the effectiveness of heat exchange in porous heat exchangers. The authors of the present study have made an attempt to complement the available experimental data by studying the effect on the heat-exchange process of suction through the wall of a portion of a helium flow passing through a porous metal–ceramic tube.

Figure 1 shows a diagram of the experimental apparatus, a conventional horizontal cryostat, described in [8]. The working volume is 650 mm long, with a diameter of 40 mm, and is formed with a thin-walled stainless-steel tubing. A vacuum layer acting as insulation separates the tube from a liquid nitrogen screen space. Residual gas pressure in the vacuum layer is maintained on the order of 10^{-5} mm Hg.

The nitrogen screen and the working space are located within a 120-mm-diameter tube at room temperature, with the outer tube being insulated from the nitrogen bath by a second vacuum layer. The working volume (Fig. 1) of the 40-mm-diameter stainless-steel tube holds the porous tube under study axially. The latter is heated by an electric current. At a distance of 5 mm from the outer surface of the tube an adiabatic screen is located, the temperature of which is maintained within (1-2)°K of the temperature of the porous tube wall. To compensate for temperature deformations, all tubes are connected to end bushings by bellows. The helium flow under a low pressure head from a 40-liter Dewar flask passes along the porous tube, first passing through the thermal insulating segment in order to hydrodynamically stabilize the flow. In its motion through the tube, a portion of the flow passes through the permeable wall into the peripheral channel. The flows through the central and peripheral channels can be regulated by valves located in the "hot" zone of the exhaust system for these flows. After the valves, passing into the "hot zone" through separate heat exchangers and being raised to a temperature of 20°C, the gaseous flows are passed through RS-5 rotameters and RG-40 gas counters. After the gas counters, the gaseous helium is returned to the liquefaction system. The rotameters are used basically to regulate the flows, i.e., a deviation of the float from its equilibrium position was used as a signal to adjust flow rates at the porous tube input system, but was not used for exact flow rate measurements, which were made by the gas counters. Preliminary measurements of the pressure drop along the entire length of the tube showed low values of 0.01-0.1 atm, so that for simplification of the experimental apparatus, pressure changes were recorded in the hot zone at the input to the apparatus and at the output in the central and peripheral channels, using type MO reference manometers, with scale gradation of 0.005 atm.

The porous tubes used in the experiments were prepared from type BrOF10-1 spherical bronze powder, particle size 0.2-0.315 mm, by sintering. Wall porosity varied from 28 to 40% with a permeability coefficient of $7 \cdot 10^{-8}$ cm^2, maximum pore dimension 190 μm, mean pore size 60 μm. Tube dimensions were as follows:

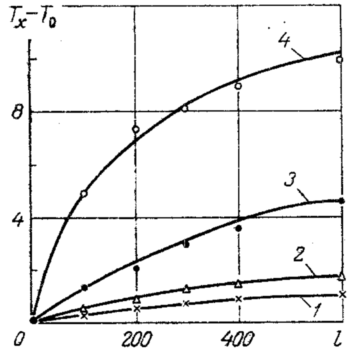

Fig. 2

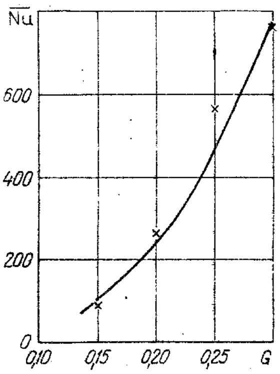

Fig. 3

Fig. 2. Temperature distribution along length of porous tube for various suctions through the tube wall and Re_{in} = $2.4 \cdot 10^4$; 1) G = 0.3 g/sec; 2) 0.25; 3) 0.2; 4) 0.15. $T_x - T_0$, °K; l, mm.

Fig. 3. Nusselt number averaged over length vs G, g/sec.

length, 600 mm; inner diameter, 12 mm; outer diameter, 17 mm. The initial tube section, with a ratio l/d of 300, had the same diameter, 12 mm. The tube was heated by passage of a dc current from a VU 12/600 rectifier.

Copper–copper, iron thermocouples with electrode thickness of 0.3 mm were used for temperature measurements. They were caulked into the tube wall at 100-mm intervals, with the tip of every thermocouple being located at the inner surface of the tube.

During the experiments, measurements were made of porous tube wall temperature, temperature of the surrounding screen, flow temperature at the porous tube input and output, flow rates of nitrogen and of helium passing along the tube axis and withdrawn through the wall into the peripheral channel. Current and voltage supplied to the porous tube and screen were also measured. During an experiment, the temperatures on the inner surface of the tube and the screen were maintained constant by adjusting the voltage supplied to the experimental tube and screen. Thus, there was no radial temperature change across the gap surrounding the experimental tube. This was done to simplify thermal flux density evaluations.

When an electric current is passed through the porous tube, a power IU is dissipated in the tube, part of which is removed by the helium flow passing along the axis, with another portion being removed by the portion of the flow which is suctioned through the tube wall. This latter quantity can be defined as $q = M_{suc} \cdot c_p \Delta T_{suc}$, where M_{suc} is the mass of the gas suctioned; c_p is specific heat, and ΔT_{suc} is the difference in the temperature of the gas suctioned before and after entry into the porous wall, which is practically impossible to measure.

At $\Delta T_{suc} = 0$, as in our case, we define the thermal flux density as the power IU per unit exterior surface of the porous tube.

Figure 2 presents experimental data on temperature distribution over porous tube length for various suction values and one and the same helium flow rate at the tube input, corresponding to a Reynolds number of $2.4 \cdot 10^4$. With increase in suction through the wall the temperature level along the tube decreases, such a temperature distribution being characteristic for the entire Reynolds number range studied $(1.6\text{–}4.5) \cdot 10^4$. The actual value of tube porosity had little effect on temperature over the range studied.

The temperature distributions obtained served as a basis for estimating heat-exchange coefficients averaged over length and Nusselt numbers. Figure 3 shows Nusselt numbers averaged over length as a function of suction through the wall.

The temperature distributions and Nusselt numbers obtained were compared with the analogous data of Aggarwal and Hollingsworth [9]. The latter were the results of experiments on air flow in a porous tube, and those researchers found that at a fixed Reynolds number at the tube input the mean values of Nusselt number increase with increasing suction.

The character of the temperature distribution and Nusselt number curves is similar to that of the curves shown in Figs. 2 and 3, i.e., the effect of suction on temperature distribution and heat-exchange coefficient is the same for air and helium flow.

NOTATION

G is the helium flow suctioned through tube wall, g/sec;
T is the temperature, °K;
Nu is the Nusselt number;
Re is the Reynolds number;
l is the length of porous tube, m;
T_x is the temperature of inner surface of porous tube, °K;
T_0 is the temperature of inner surface of porous tube at input section, °K;
q is the thermal flux density, W/m^2;
U is the voltage, V;
I is the current, A.

LITERATURE CITED

1. J. R. Sellars, J. Appl. Phys., 26, 489 (1955).
2. A. S. Berman, J. Appl. Phys., 24, No. 9 (1953).
3. S. W. Yuan, Trans. Am. Soc. Mech. Eng., 78, No. 4 (1956).
4. R. Kinney, Int. J. Heat Mass Transfer., 11, No. 9 (1969).
5. R. J. Pederson and R. Kinney, Int. J. Heat Mass Transfer, 14, No. 1 (1971).
6. G. Raithby, Int. J. Heat Mass Transfer, 14, No. 1 (1971).
7. R. Kinin and E. Sparrow, Teploperedacha, No. 2 (1970).
8. G. I. Bobrova and V. A. Morgun, in: Heat and Mass Transfer of Cryogenic Liquids in Porous Heat Exchangers [in Russian], ITMO Akad. Nauk BSSR, Minsk (1974).
9. J. K. Aggarwal and M. A. Hollingsworth, Int. J. Heat Mass Transfer, 16, 591 (1973).

CALCULATION OF CIRCULATION CHARACTERISTICS OF A TWO-PHASE THERMOSYPHON

A. G. Beinusov and V. B. Utkin

UDC 536.24

A new method is described for calculating the circulation characteristics of a thermosyphon with separate vapor and condensate channels. Calculation and experiment are compared.

At the present time, two-phase thermosyphons with separate vapor and condensate channels are widely used for heat transfer purposes [1]. The efficiency of thermosyphons of this sort is largely dependent on the circulation characteristics of the closed hydraulic circuit. The circulation of a boiling liquid in closed hydraulic circuits can be estimated in various ways [1-3].

In the present paper we propose a new method for solving this problem as applied to thermosyphons with separate vapor and condensate channels.

A schematic diagram of the circulation circuit is shown in Fig. 1. Here 1 and 2 are respectively the down- and up-pipes; 3 is the condenser. The vaporizer is located in the up-pipe. In the down-pipe there is only liquid; in the up-pipe there is a mixture of vapor and liquid. The equation of motion of the liquid and vapor in a closed circulation circuit (ignoring the compressibility of the components, energy losses on changing the interphase surface, and oscillations of vapor bubbles) can be brought to the form [4]:

$$g(\rho' - \rho'') L\varphi = \sum (\Delta P_{fr} + \Delta P_{acc}) \sin \beta. \tag{1}$$

Novosibirsk Electrotechnical Institute. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 37, No. 3, pp. 424-428, September, 1979. Original article submitted November 9, 1978.

0022-0841/79/3703-1032$07.50