EXPERIMENTAL INVESTIGATION OF HEAT EXCHANGE BETWEEN A TURBULENT AIRFLOW AND A SHORT ROTATING CYLINDRICAL TUBE

V. M. Buznik, G. A. Artemov,
V. N. Bandura, and A. M. Fedorovskii

UDC 536.244

The experimental measuring equipment and procedure are described. It is found that the local and mean heat transfer between the wall of a totating tube and a turbulent airflow decreases with increasing rpm. Generalized equations are derived in dimensionless form.

Owing to the operating conditions, the heat transfer surfaces of numerous elements of power units (hollow shafts of electrical machines, turbine rotors, rolling-mill rollers, etc.) participate in the rotating motion. Whereas the external heat transfer problem for rotating surfaces has been reasonably well investigated [1], the data available for the internal problem are sparse, contradictory, and do not allow the influence of rotation on heat transfer to be taken into account in design calculations.

In the present paper, the heat transfer between a short rotating tube and a turbulent airflow is studied with experimental apparatus which uses a 1Kh18N10T stainless-steel rotary calorimeter (inner diameter of 97.6 mm, wall thickness of 3.5 mm, and a length of 3000 mm) as the sensor. The end faces of the calorimeter were insulated by Textolite disks and the inner surface was elaborately polished. The tube was dc-heated by means of a constantan band uniformly wound about the tube. Through a fixed lemniscate inlet tube, the air was conducted into the sensor, and from there, via a hollow current-collector shaft, a gland, and an air-duct system, to the intake of an exhaust fan. The air-flow meter was placed between the sensor and the ventilator. A honeycomb and a flow laminating convergent nozzle were placed in front of the flow-meter inlet. The flow rate was regulated by electrically driven slides. Airtight sealing of the sensor was achieved with the aid of a packing gland.

The tube was rotated by a dc motor, using a conical belt drive. Bearings and belt drive were designed for providing smooth and stable rotation over the entire rpm range employed.

The wall temperature was measured by copper-constantan thermocouples placed at the following relative distances (in tube diameters) from the heat source: 0.26, 0.77, 1.3, 2.0, 3.1, 4.6, 6.1, 7.8, 8.8, 9.8, 11.9, 13.9, 15.2, 16.2, 17.3, 19.3, 21.4, 22.6, 25.7, and 28.8. The hot junctions of the thermocouples were welded to the tube wall flush with the inner surface. From the point of location of the junction, the thermocouple leads were first directed along the isotherms (along the circumference of the tube cross section) and then along a longitudinal groove to the intermediate terminal. The rotating thermocouples were connected to the measuring instruments by a brush-type current collector of a design described in [2]. After preparation, all thermocouples were calibrated for temperatures ranging from 0 to 300°C. During calibration, the thermocouple emf was measured with a PPTN-1 potentiometer, and during the tests with a KP-59 potentiometer of class 0.05. Measurements during the tests were performed under steady operational conditions, the latter being determined on the basis of recordings made by two ÉPP-09M potentiometers. In order to determine the heat losses to the ambient medium, the installation was calibrated without throughput of air. The discrepancy between the power calculated from the electric current with allowance for losses to the ambient medium and the power calculated from the airflow rate did not exceed 5 to 6%. The tests were performed in series, each at a specific airflow rate and a specific heat flux at the wall. The test conditions (8 to 10 in each series) differed in the calorimeter rpm. The rpm employed in the tests ranged from 0 to 1170. The heat flux varied between 900 and 5800 W/m^2, and the airflow rate between 0.05 and 0.4 kg/sec.

S. O. Makarov Shipbuilding Institute, Nikolaev. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 15, No. 5, pp. 832-835, November, 1968. Original article submitted February 13, 1968.

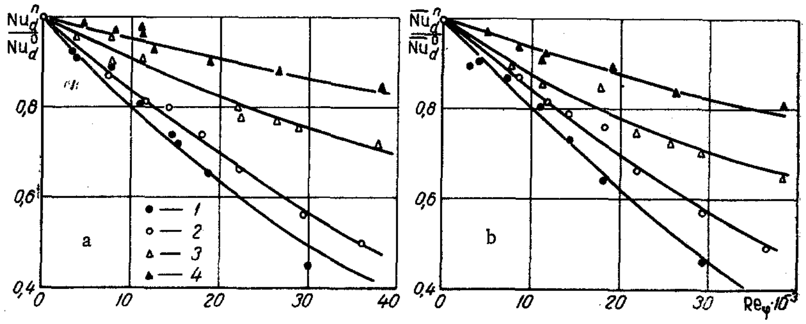

Fig. 1. Influence of longitudinal flow velocity on local (a) and mean (b) heat transfer in a rotating tube: 1) $\overline{Re}_d$ = 37,700; 2) 53,500; 3) 110,000; 4) 223,000.

Heat transfer calculations were based on the amount of heat released in the electric heater with allowance for ambient losses. The local and mean heat transfer coefficients were calculated for the tube cross sections at which the thermocouples were welded to the wall. The variation of the local coefficients along the tube length showed that for $x/d \sim 3$ at the tube wall, there exists a boundary layer transition. This effect was observed up to axial-velocity Reynolds numbers of $2.3 \cdot 10^5$. Data for $x/d > 3$ were used for processing, i.e., for a fully developed turbulent flow region in the boundary layer.

The local heat transfer coefficients for a fixed tube were compared with data [3] for a turbulent gas flow in a straight circular tube, and were found to agree with them to within 3 to 4%. The mean heat transfer coefficients obtained from stationary tests were compared with those calculated for the known formula proposed by Mikheev [4]. The increase in mean heat transfer in the inlet section of the tube was accounted for with the aid of Alad'ev's [5] corrections. The discrepancy between our static tests and Mikheev's formula did not exceed ±5%.

The experiments revealed an almost constant influence of rotation on the local and mean heat transfer along the length of a short tube. For turbulent airflow in a rotating tube, the decrease in the heat transfer coefficients was the same at the inlet and exit section of the calorimeter. Figure 1 shows the test results in the form of the heat transfer ratio for a rotating and stationary calorimeter as a function of the rotational Reynolds number Re_φ. It can be seen from Fig. 1 that an increase in the peripheral angular velocity of the tube wall at a constant airflow rate leads to an almost proportional decrease in heat transfer. A very weak nonlinear dependence of heat transfer on the speed of rotation was observed over the entire range investigated.

The generalized data made it possible to obtain final computational relations in dimensionless form. The following formula

$$Nu_d^n = 0.022\, Re_d^{0,8}\, Pr^{0,43}\, \frac{1.38}{\left(\dfrac{x}{d}\right)^{0,12}} \left(1 - \mathrm{th}\left[1 + 0.001335 \frac{Re_\varphi}{Re_d^{7/8}} - \exp\left(-0.23 \frac{Re_\varphi}{Re_d^{7/8}}\right)\right]\right) \quad (1)$$

is proposed for determining local heat transfer in a short rotating tube, and formula

$$\overline{Nu}_d^n = 0,021\, \overline{Re}_d^{0,8}\, \overline{Pr}^{0,43}\, \bar{\varepsilon}_e \left(1 - \mathrm{th}\left[1 + 0.000175 \frac{Re_\varphi}{\overline{Re}_d^{0,685}} - \exp\left(-0.03 \frac{Re_\varphi}{\overline{Re}_d^{0,685}}\right)\right]\right) \quad (2)$$

for the mean heat transfer in such a tube.

It proved possible to compare the test results obtained with the experimental data in [6]. For a tube with a 3.25 in inner diameter rotating about its axis at 6000 rpm, the mean heat transfer coefficient of a heated wall at a mass flow rate of 20,000 lb/h of water was seven times smaller than in the case of a stationary tube. For these data, the decrease in heat transfer from formula (2) is 6.8 times.

NOTATION

x is the distance from the heat source;
d is the inner diameter of the tube;

Nu_d^n	is the local value of the Nusselt number for a rotating rube;
Re_d, Pr	are the local values of the Reynolds number and Prandtl number as a function of the relative distance x/d;
$\overline{Nu}_d^n$, $\overline{Re}_d$, $\overline{Pr}$	are mean values of Nusselt number, Reynolds number, and Prandtl number between the tube inlet and cross section x;
Nu_d^0, $\overline{Nu}_d^0$	are local and mean values of the Nusselt number between the tube inlet and cross section x of a stationary tube;
Re	is the rotary Reynolds number calculated from the peripheral velocity and the inner diameter of the tube.

LITERATURE CITED

1. V. M. Buznik, Heat Transfer in Power Units of Ships [in Russian], Izd-vo Sudostroenie (1967).
2. V. M. Buznik, G. A. Artemov, V. N. Bandura, Yu. D. Kardashev, and A. M. Fedorovskii, Sudostroenie, No. 9 (1965).
3. V. A. Mukhin, A. S. Sukomel, and V. I. Velichko, Inzhen.-Fiz. Zh., 5, No. 11 (1962).
4. M. A. Mikheev, Izv. Akad. Nauk SSSR, Otd. Tekh. Nauk, No. 10 (1952).
5. I. T. Alad'ev, Izv. Akad. Nauk SSSR, Otd. Tekh. Nauk, No. 11 (1951).
6. Discussion of the paper by R. F. Pattenden, J. Mech. Engn. Sci., 6, No. 3, 309-310 (1964).