EFFECT OF THERMAL INSTABILITY ON THE RATE OF CONVECTIVE HEAT TRANSFER

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Results are shown of an experimental study concerning the transient convective heat transfer in the housing clearance in a turbine model.

Relations which have been derived for the transient heat transfer show that the process rate in this case is affected not only by such parameters as the Reynolds number, the Prandtl number, and the hydraulic diameter (which characterize the steady-state heat transfer) but also by the thermophysical properties of the material, by its thickness, and its initial temperature, i.e., by factors which determine the transient heat conduction.

The authors have studied the effects of transient heat conduction through the flanges of a horizontal coupling, in a model of a turbine housing, on the rate of convective heat transfer in the housing clearance of the high-pressure cylinder.

The boundary conditions in an analysis of transient thermal processes are unknown. A direct measurements of thermal fluxes is in this case rather difficult, but it is much simpler here to measure the temperature at some point on the surface in the gas stream and, thus, to reduce the problem to that of determining the thermal flux produced by such a mode of heating. It is necessary, therefore, to solve the temperature problem in reverse, namely to determine the boundary conditions from the known temperature at some point of the body whose shape, thermophysical characteristics, and initial temperature distribution are all given. In our case these basic data had been obtained on the model of a K-300 turbine housing one fifth the original size.

The preliminary measurements included the temperature of the ambient heating medium and the temperature of the model, as a function of time, at various points across the thickness of the housing components.

In order to evaluate the test data by the most appropriate method, we first examined the temperature field of the flange coupling within the test zone of the high-pressure cylinder. An analysis revealed that the one-dimensional temperature field of the flange coupling was almost identical to the temperature field of some equivalent plate with a thickness R_V greater than the nominal flange thickness R. This conclusion agreed with the results in [1], where flange couplings of turbines had also been studied.

These results, then, justify solving the reverse problem of heat conduction as a one-dimensional problem. In view of this, we will first consider the heating of an infinitely large plate with boundary conditions of the first kind. The measured temperature characteristics of the metallic flange surface may serve as the boundary functions. According to the test data, the temperature-time characteristics of heat transfer surfaces in the turbine housing may, for a definite initial period of time, be approximated by the equation

$$t(R, \tau) = t_m - (t_m - t_0)\exp(-k\tau). \qquad (1)$$

The temperature of the unheated flange side follows the characteristic of an adiabatic insulated surface.

If one considers the problem, i.e., assumes the thermophysical properties to be constant over the test range of temperatures, then the mathematical problem can be formulated as follows:

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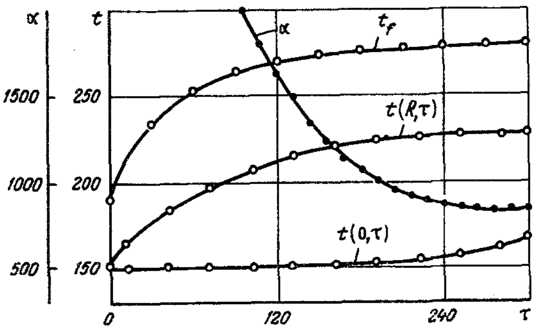

Fig. 1. Temperature and heat transfer coefficient as functions of time.

$$\frac{\partial t(x,\ \tau)}{\partial \tau} = a\,\frac{\partial^2 t(x,\ \tau)}{\partial x^2} \quad (\tau > 0;\ 0 < x < R),$$
$$t(x,\ 0) = t_0 = \text{const},$$
$$t(R,\ \tau) = t_m - (t_m - t_0)\exp(-k\tau), \tag{2}$$
$$\frac{\partial t(0,\ \tau)}{\partial x} = 0.$$

The solution to the problem stated in [2] yields the thermal flux density at the heat transfer surface

$$q = -\lambda \frac{\partial t}{\partial x}\bigg|_{x=R} = -\lambda\left\{(t_m - t_0)\left[\frac{\exp(-\mathrm{PdFo})\sqrt{\mathrm{Pd}}}{R\cos\sqrt{\mathrm{Pd}}}\sin\sqrt{\mathrm{Pd}}\right.\right.$$
$$\left.\left. + \sum_{n=1}^{\infty}\frac{A_n\mu_n\exp\left(-\mu_n^2\mathrm{Fo}\right)}{R\left(1 - \dfrac{\mu_n^2}{\mathrm{Pd}}\right)}\sin\mu_n\right]\right\}. \tag{3}$$

From the values of thermal flux found by the solution of Eq. (3) and from the ambient temperatures at the surface, one then determines the heat transfer coefficients according to the Newton–Richman equation.

The temperature of the flange surface, the temperature of the vapor, and the heat transfer coefficient at the characteristic section of the high-pressure cylinder are shown in Fig. 1 as functions of time. The points on the curves represent the measured values of the temperature on both flange sides under a constant rate of vapor flow. The solid line, which represents the temperature variation at the heated surface, has been plotted according to Eq. (1). The temperature curve for the insulated surface is based on the solution to the equation of heat conduction for a plate with boundary conditions of the first kind (1). The discrepancy between calculated and measured temperatures does not exceed here 0.5%.

According to Fig. 1, the heat transfer coefficient is a function of time. Its maximum value occurs at the start of the heating process. It then decreases with time down to a constant level. The time to reach a quasisteady mode of heat transfer is 200-350 sec, which corresponds to a Fourier number within the 0.125-0.220 range. Such a trend is in accord with the basic hypotheses concerning transient heat transfer which have been formulated and then verified experimentally in [3-5].

If one assumes a quasisteady effect of the Reynolds number, of the Prandtl number, and of the hydraulic diameter on the heat transfer, then the dimensionless heat transfer coefficient during transient heating will be a function of the transient number:

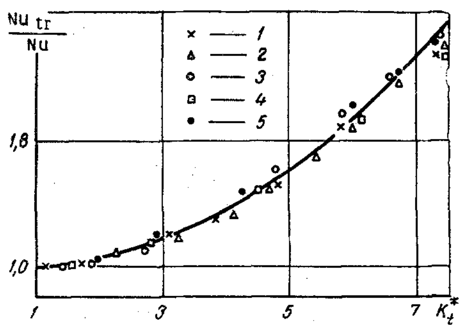

Fig. 2. Ratio of dimensionless transient heat transfer coefficient to its respective quasisteady value $\mathrm{Nu_{tr}/Nu}$, as a function of the transient number K_{tr}^*: Re = 24,000 (1), 25,000 (2), 130,000 (3), 11,000 (4), 90,000 (5).

$$\mathrm{Nu}_{tr} = \mathrm{Nu} f(K_{tr}), \tag{4}$$

where $\mathrm{Nu} = f_0(\mathrm{Re}, \mathrm{Pr}, d)$ is the Nusselt number for quasisteady heat transfer and $K_{tr} = dt_\omega/d\tau \cdot d^2/(t_f - t_\omega)a_f$ is the transient number [6], which characterizes the rate of change of the relative surface temperature.

The tests were performed with vapor flow rates resulting in values of the Reynolds number within the 11,000-130,000 range at the test sections. As the characteristic dimension for the Nusselt number and the Reynolds number we selected the gap width between inner and outer cylinder. The operating vapor characteristics were varied during the tests over the following ranges: $p = 4.5$-7.0 bars and $t_f = 175$-330°C.

When the surface temperature varies exponentially according to Eq. (1), then the K_{tr} number can be defined as follows:

$$K_{tr} = \frac{k(t_m - t_0)\exp(-k\tau)\, d^2}{(t_f - t_\omega)\, a_f} \tag{5}$$

The $\mathrm{Nu}_{tr}/\mathrm{Nu} = f(K_{tr})$ relation has split into different curves corresponding to different tests. With the introduction of a corrective temperature factor, it became possible to narrow the scatter of test points down to a universal curve shown in Fig. 2 with the abscissas

$$K_{tr}^* = K_{tr}\frac{t_f}{t_\omega}.$$

As K_{tr}^* increases, the difference between the rates of transient and quasisteady heat transfer decreases. The trend of the relation between the Nusselt number ($\mathrm{Nu}_{tr}/\mathrm{Nu}$) and the transient number (K_{tr}) seems to follow approximately a parabola. Thus, Eq. (4) becomes

$$\frac{\mathrm{Nu}_{tr}}{\mathrm{Nu}} = 1 + 0.015\,(K_{tr}^*)^{2.3}. \tag{6}$$

The maximum scatter of test points about the approximating curve is 9%.

In comparing the empirical relation (6) for the rate of transient heat transfer with the theoretical analysis of transient heat transfer in [4], it must be noted that qualitatively similar results have been obtained by the two different approaches to the problem.

With the established dependence of the heat transfer rate on the transient number, it is possible now to define the boundaries between the quasisteady and the transient regions. It follows from Fig. 2 that the transient effect becomes significant when $K_{tr}^* < 1.7$. In this case the process can be described by the equations of steady-state heat transfer with instantaneous values of the parameters.

The results of the experimental study show that transient and quasi-steady convective heat transfer differ significantly and that this difference widens as the rate of change of temperature at the heated surface increases.

The transient effect on the heat transfer rate within the test range of parameter variations is accurately enough accounted for by the first time derivative of the temperature.

NOTATION

t_0 is the initial temperature of component, °C;
t_ω is the temperature of the heated surface, °C;
t_m is the maximum temperature of heated surface, °C;
t_f is the temperature of ambient heating medium, °C;
τ is the time;
R_v is the thickness of equivalent plate;
d is the hydraulic diameter;
q is the thermal flux density, W/m^2;
α is the heat transfer coefficient, $W/m^2 \cdot °C$;
a is the thermal diffusivity, m^2/h;
λ is the thermal conductivity, $W/m \cdot °C$;
p is the pressure of ambient heating medium, bars;

$\mathrm{Fo} = \frac{a\tau}{R_v^2}$; $\mathrm{Re} = \frac{\omega d}{\nu_f}$; $\mathrm{Nu} = \frac{\alpha d}{\lambda_f}$.

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