AXISYMMETRIC BLUNT BODY-TO-GAS FLOW HEAT TRANSFER IN THE PRESENCE OF COOLANT INJECTION VIA ROUND HOLES AND VIBRATIONAL DISTURBANCES

A. N. Golovanov

UDC 553.6.011.6

An experimental study is made of heat transfer between a blunt body and a high-temperature flow in the presence of gas-coolant injection via a series of round holes in the vicinity of the frontal critical point, linear, radial, and tangential vibrations of a wall.

Vibrations of surfaces, turbulent noises, acoustic oscillations and other low energy disturbances almost always accompany operational processes of heat power units. Despite the insignificant value of disturbances, a result of such interaction may essentially distort heat and mass transfer characteristics [1, 2].

In the present work, investigation is made of heat transfer of a blunt body made in the form of a truncated cone with gas injection through round holes in a minor base of the cone in the vicinity of the frontal critical point. Such units find application as heat-protection systems of aircraft members, in heat power engineering, in plasma-chemical reactors, etc. [3].

In terms of generalized variables, the problem under consideration may be written as $\psi = \psi(\mathrm{I}, \mathrm{B})$, where $\psi(q_+ - q_-)/q_-$ is the relative heat transfer function; $\mathrm{I} = 1/2\rho c(2\pi f A)^2$ is the vibration rate; $\mathrm{B} = 4G_w/[\pi d^2 \cdot (\alpha/c_p)_0]$ is the injection parameter.

Experiments were conducted in air jets of an EDP-104A/50 electric-arc plasma generator, an ohmic gas heater, and in a working section of a low-speed small-turbulent wind tunnel of the type MT-324.

Figure 1a is the schematic representation of models manufactured in the form of a truncated cone, in a minor base of which seven round holes 1, 2 are drilled. Arrows 3 and 4 in the figure indicate, respectively, the incoming flow and the injected gas-coolant (air). Different types of vibration-testing units 6 were connected to the models via pipe connection 5.

Vibrations of the end face of the models, longitudinal to the flow and sinusoidal in time, the so-called linear vibrations (the terminology from [4]), were generated by the vibration-testing unit assembled of a motor with gear 7 mounted onto the axis (see Fig. 1b). Amplitude A and vibration frequency f were varied at the expense of the gear geometry and the speed of motor shaft rotation w.

Radial vibrations of the surface (see Fig. 1c) were initiated analogously to linear vibrations; the difference was related to misalignment of the motor shaft and the pipe connection, as well as to the presence of pressure springs 8.

Tangential surface vibrations (Fig. 1d) were generated by transforming the rotational motion of motor shaft 7 into the reciprocating motion of the wall with the help of rocker 9 and rod 10. The frequency and an amplitude of tangential vibrations were regulated by the speed of motor shaft rotation w and rocker length *l*.

The geometric parameters of the models and experimental conditions are given in Table 1.

The frequency and amplitude of tangential vibrations were varied within f = (0-25) Hz, A = $(0.5\text{-}7.0) \cdot 10^{-3}$ m, and A = $(0\text{-}9)^{\circ}$.

The incoming flow parameters and heat and mass transfer characteristics were determined by thermocouples, a thermoanemometer, pneumometric probes, and rotameters from the energy balance condition of the operating plasma generator by using the procedures described in [5-7]. The temperature of air supplied via round holes was insignificantly changed, i.e., from 300 to 310 K. For each experiment, the parameters of the incoming flow and the injected gas were not changed with time.

The wall temperature T_w and the heat flux toward the wall q in the vicinity of the frontal critical point were determined by the known thermoelectric and exponential methods [3-6].

Tomsk State University. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 63, No. 2, pp. 194-198, August, 1992. Original article submitted November 11, 1991.

1062-0125/92/6302-0804$12.50

TABLE 1. Geometric Parameters of the Models and Experimental Conditions

Type of installation	D, 10^{-3} m	d, 10^{-3} m	δ, 10^{-3} m	n	T_∞, K	v_∞, m/sec	Re_D	$(\alpha/c_p)_0$, kg/(m^2·sec)
EDP 104A/50	14—25	4—8	1	4—12	3600	66,1	935	0,2
Ohmic heater	14—25	4—8	1	4—12	370	3,07	25	0,54
MT-324	120	68	12	7	300	4,34	372	—

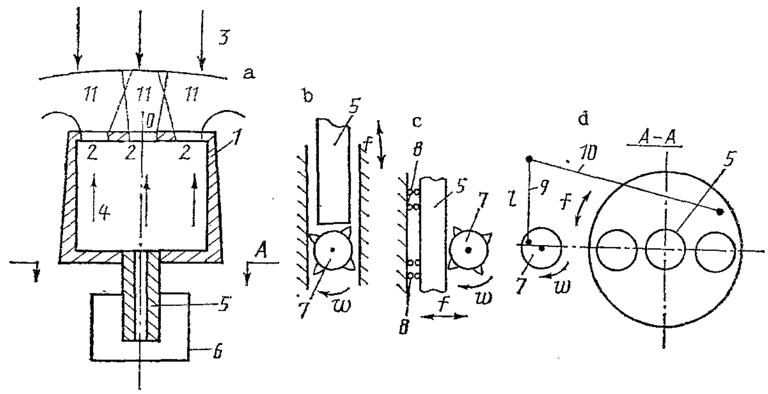


Fig. 1. Schematic of the models and vibration-testing units.

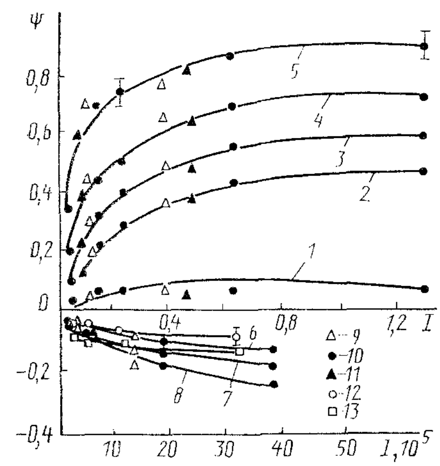


Fig. 2. Relative function of heat transfer vs frequency of surface vibrations. I, kg/sec^3; I, 10^5 kg·deg/(m^2·sec^3).

The total errors in the determination of the parameters did not exceed $\delta T_\infty \le 10\%$, $\delta v_\infty \le 6\%$, $\delta q \le 8.7\%$, $\delta G_w \le 5\%$, $\delta T_w \le 4.1\%$, $\delta f \le 2\%$, $\delta A \le 3.2\%$.

Figure 2 shows the relative function of heat transfer versus frequency for different types of vibrational disturbances. Curves 1-5 are obtained, respectively, for gas-coolant flow rates (2.38, 2.61, 2.97, 3.2, 3.45)$\cdot 10^{-4}$ kg/sec and linear surface vibrations. Points 9-11 are obtained for models with geometric dimensions D = (14, 19, 25)$\cdot 10^{-3}$ m; d = (4, 6, 8)$\cdot 10^{-3}$ m, and n = n = 4, 7, 12. The curves represent calculations by the approximation formula derived in the present work:

$$\Psi = (0{,}046B - 1{,}376)\, I^{0{,}0024B + 0{,}174},$$

at $31.2 \le B \le 49.4$; 0.032 kg/sec$^3 \le 1 \le 1.25$ kg/sec^3.

The approximation error does not exceed 4.3%.

TABLE 2. "Threshold" Vibration Intensities and Injection Parameters

Vibration type	I_{th}	B_{th}
Linear	0,032 kg/sec^3	2,83
Radial	0,050 kg/sec^3	22,46
Tangential	$1.75\cdot10^5$ kg·deg^2/(m^2·sec^3)	2,30

Analysis of curves 1-5 in Fig. 2 shows that the linear surface vibrations of the models within the investigated frequency and coolant flow rate ranges enhance the heat transfer process; an increase of the vibration frequency causes an increase of the relative heat-transfer function.

Curves 6-8 in Fig. 2 (the lower abscissa axis) are obtained, respectively, for the gas-coolant flow rates $(0.3, 0.41, 0.73)\cdot10^{-4}$ kg/sec and tangential surface vibrations of the models. The curves represent calculations by the approximation formula derived in the present work

$$\Psi = 0{,}0523B^{0,95} - 0{,}01\,(0{,}6 + 0{,}5B)\ln I,$$

at $2.83 \le B \le 6.89$; $1.75\cdot10^5$ kg·deg^2/(sec^3·m^2) $\le I \le 39.2\cdot10^5$ kg·deg/(sec^3·m^2) with an approximation error not exceeding 7.5%.

Points 12 in Fig. 2 are obtained for the gas-coolant flow rate $2.38\cdot10^{-4}$ kg/sec, 13 for $2.61\cdot10^{-4}$ kg/sec and radial surface vibrations. The curves represent calculations by the formula obtained in the present work

$$\Psi = 0{,}668 - 0{,}032B + (0{,}007B - 0{,}243)\,I,$$

$$22{,}4 \leqslant B \leqslant 24{,}6;\ 0{,}05\ \text{kg/sec}^3 \leqslant I \leqslant 0{,}65\ \text{kg/sec}^3,$$

with an approximation error not exceeding 5.1%.

Table 2 lists "threshold" vibration intensities I_{th} and injection parameters B_{th}, at the excess of which the vibrations begin to exert an influence on heat transfer. It is seen that tangential and radial surface vibrations cause a decrease of the relative function of heat transfer.

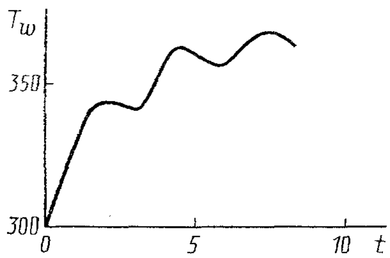
The confidence limits in Fig. 2 are calculated for 3-5 tests with the confidence coefficient 0.95 at the Student coefficient $t_\alpha = 1.96$.

It is worth noting that at tangential and radial surface vibrations and small injection parameters $B < 2.3$, unstable processes may develop which are related with wall temperature fluctuations (Fig. 3). The frequency of such fluctuations is close to that of surface vibrations, thus testifying to the relationship between this event and inertial properties of the system. For low gas-coolant flow rates, surface vibrations are no longer insignificant disturbances. At moments of time corresponding to a change of the sign of a disturbance phase because of inertial forces, coolant density redistribution is likely to occur in the gas curtain region near the wall to be protected. The hot gas from the external flow finds its way to the wall, the temperature of the latter increases and the process periodically recurs.

The results obtained for the vibration effect on heat transfer may be interpreted using the physical model from [7], according to which additional flow turbulization takes place on the basic sections of elementary jets 11 (Fig. 1). In these zones, low-frequency, large-scale gas pulsations are initiated longitudinally to the elementary jet axis, which transport the gas from an incoming high-temperature flow to the wall to be protected and thus enhance the heat transfer process. Linear surface vibrations excite analogous low-frequency gas pulsations and enhance heat transfer as well.

At relatively small coolant flow rates (see curve 1, Fig. 2), the effect of linear vibrations is insignificant since the gas-coolant does not form elementary jets. At small gas flow rates, the models are cooled by a gas curtain mechanism from hole to hole; with increasing coolant flow rate, the elementary gas jets converge, and the effect of heat-transfer enhancement owing to linear surface vibrations manifests itself to a greater extent (see curves 2-5, Fig. 2).

Tangential and radial surface vibrations cause a failure of high-gradient gas flow regions on the main sections of the elementary jets where large-scale, low-frequency gas pulsations occur [6]. Gas velocity profiles near the wall become more smooth, turbulence is suppressed and, as a result, the magnitude of heat flux to the wall decreases.



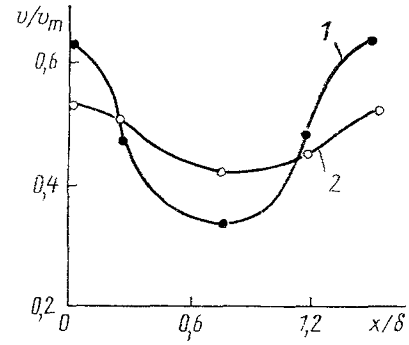


Fig. 3

Fig. 4

Fig. 3. Wall temperature oscillogram recorded in the vicinity of the frontal critical point T_w (K) vs time t (sec).

Fig. 4. Dimensionless gas velocity profiles near the wall; v_m = 4.2 m/sec.

Figure 4 shows dimensionless gas velocity profiles measured in an MT-324 wind tunnel at a distance $y/\delta = 0.5$ from the surface of face dulling of the cone (the coordinate system is shown in Fig. 1). Curve 1 is obtained for models with the flow around them when vibrational disturbances are absent; curve 2 for radial vibrations, at a frequency 5.2 Hz and amplitude $2 \cdot 10^{-3}$ m.

From the figure it is seen that the vibrational disturbances smooth out the velocity profile; the gas flow becomes more uniform, without high-velocity gradients. Such smoothing of the gas velocity profiles is attributed to turbulence suppression [5] and, probably, is a cause of heat transfer deterioration in the case of radial surface vibrations. Analogous curves are obtained for flow past the models subjected to tangential vibrations.

Thus, the experiments conducted have revealed that linear, radial, and tangential vibrations of a perforated wall with injection of a gas-coolant may enhance and deteriorate the gas-to-wall heat-transfer process. Despite a slight intensity of vibrations, the interaction may considerably distort heat-transfer characteristics. The possibility appears of governing gas dynamic parameters of a gas flow near the wall as well as controlling the heat and mass transfer process in such systems.

NOTATION

ν, coefficient of kinematic viscosity; v_∞, mean flow rate of an incoming flow; c, sonic velocity; λ, thermal conductivity; ρ, gas density; G_w, gas-coolant flow rate; d, diameter of a permeable wall section; α, heat transfer coefficient; c_p, coefficient of specific heat at constant pressure; q, specific heat flux; D, diameter of a minor cone base; δ, hole diameter; n, number of holes; T_∞, mean-mass temperature of an incoming flow. Indices: +, −, parameters with and without vibrations; 0, parameters without gas injection.

LITERATURE CITED

1. V. E. Nakoryakov, A. P. Burdukov, A. M. Boldarev, and P. N. Terleev, Heat and Mass Transfer in an Acoustic Field [in Russian], Moscow (1970).
2. A. S. Ginevskii, E. V. Blasov, and A. V. Kolesnikov, Aeroacoustic Interactions [in Russian], Moscow (1978).
3. Yu. V. Polezhaev and F. B. Yurevich, Thermal Protection [in Russian], Moscow (1976).
4. B. V. Raushenbakh, Vibrational Combustion [in Russian], Moscow (1961).
5. A. N. Golovanov, Zh. Prikl. Mekh. Tekh. Fiz., No. 1, 153-158 (1989).
6. A. N. Golovanov, Zh. Prikl. Mekh. Tekh. Fiz., No. 2, 18-23 (1988).
7. A. N. Golovanov, Izv. Akad. Nauk SSSR, Ser. Tekh. Nauk, No. 3, No. 18, 46-51 (1987).