# EXPERIMENTAL STUDY OF THE EFFECT OF ACOUSTIC OSCILLATIONS ON HEAT TRANSFER IN A GAS FLOW

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The influence of acoustic oscillations on heat transfer in a high-temperature turbulent flow is determined. It is shown how the heat transfer at resonance frequencies differs at nodes and antinodes of the standing acoustic wave.

It is a well-known fact that acoustic (sonic and ultrasonic) oscillations in a gas flow can significantly affect the heat-transfer process [1-3]. The most detailed investigations of heat transfer in a pulsating flow have been directed at laminar flow in a boundary layer. Far fewer studies have been devoted to the influence of oscillations on heat transfer in turbulent flows, despite the patently greater practical importance of the latter. We should point out that the research results depend strongly on the type of flow perturbed by acoustic oscillations (laminar or turbulent), the intensity of the oscillations, and the site at which the heat flux is measured (for oscillations at resonance frequencies).

The objective of the present study is to determine how the heat transfer in a pulsating high-temperature turbulent gas flow as well as at resonance-frequency nodes and antinodes differs from the heat transfer in the nonpulsating case over a wide range of oscillation frequencies and freestream Reynolds numbers.

We have designed and built a special apparatus (Fig. 1) to accomplish the stated objective. It consists of a gas-jet generator 1 in the form of a tube, in which a gas flow is created from the combustion products of burning solid matter and in which the heat flux and flow parameters are measured; a dc motor 2; a clutch 3; a variable-ratio reducer 4; and a perforated-disk counterflow siren 5 for the generation of oscillations in the tube. The solid matter 6 lines the side surface and burns at one end. The mean pressure level in the tube is determined by the area of the burning solid surface, the composition of the solid, and the effective critical cross section of the nozzle. The flow velocity of the combustion products is dictated by the tube diameter and pressure (the burning rate of solid matter is pressure-dependent). The natural frequency of longitudinal modes is determined by the length of the tube. The solid matter is inserted in a holder, which can be moved along the gas-jet generator tube to vary the acoustical properties of the latter. The siren, which is mounted on the reducer output shaft and is driven by the motor, periodically closes off the nozzle orifice, introducing pressure oscillations (longitudinal waves) into the tube interior. The apparatus permits smooth variation of the generated oscillation frequency from 50 to 5000 Hz. The following parameters are measured in the course of the experiment:

1) pressure pulsation amplitude and frequency, by the cooled piezoelectric pressure pulsation gauge 7 (maximum errors of determination of the pressure amplitude and frequency: 15% and 2%, respectively);

2) mean pressure in the gas-jet generator, by the tensometric pressure gauge 9 (maximum error: 3%);

3) rpm of the electric motor, by the tachometer-generator 10 and an induction angular-velocity gauge (maximum error: 1%);

4) temperature in the calorimetric unit 8, by a Chromel—Alumel thermocouple 0.1 mm in diameter (maximum error: 5%).

The thermodynamic temperature of the combustion products is 2300°K, and the investigated pressure range is from 20 to 80 bars. These parameters make it possible to cover a range of Reynolds numbers from $8.5 \cdot 10^3$ to $20 \cdot 10^3$.

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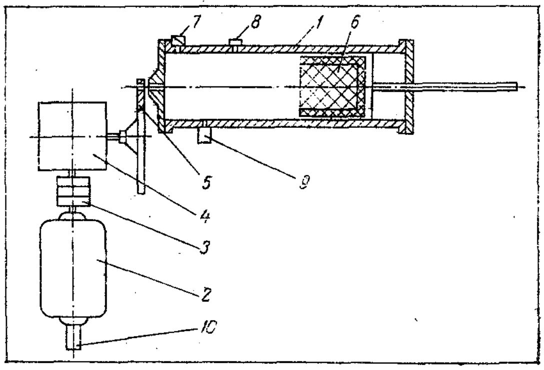


Fig. 1. Schematic of experimental apparatus. 1) Gas-jet generator; 2) dc electric motor; 3) clutch; 4) reducer; 5) perforated-disk siren; 6) solid matter; 7) pressure pulsation gauge; 8) heat-flowmeter; 9) mean-pressure gauge; 10) tachometer-generator.

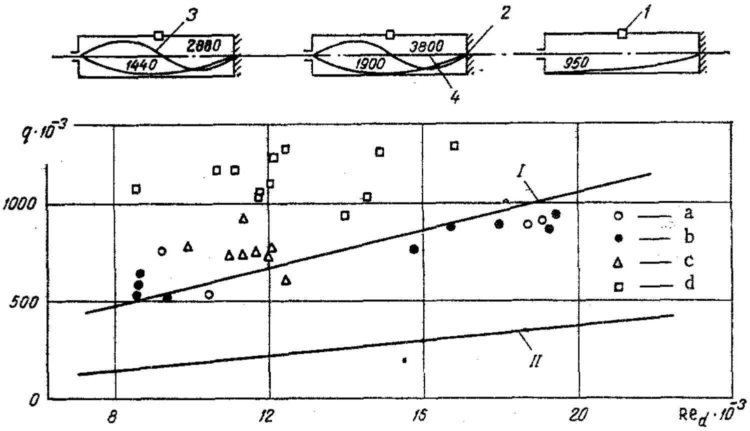


Fig. 2. Acoustic field patterns and experimental results on heat transfer in an acoustic field. 1) Heat-flowmeter; 2) burning surface of solid matter; 3) acoustic velocity distribution in standing wave; 4) calculated resonance frequency, Hz; a) experimental results without oscillations; b) with nonresonance oscillations; c) with the heat-flowmeter at a velocity node of the standing wave; d) with the heat-flowmeter at a velocity antinode of the standing wave; q in $W/m^2$; $q_\Sigma = q_c + q_r$ (curve I); $q_c$ (II).

The test sequence is as follows: 1) without pressure oscillations; 2) with pressure oscillations at nonresonance frequencies from 100 to 500 Hz (the tube length is less than a quarter-wavelength); 3) with pressure oscillations at resonance frequencies from 1000 to 4000 Hz (an integral number of quarter-waves fits the length of the tube). The resonance frequencies for longitudinal modes are calculated from the equation

$$f = \frac{c}{4l}\, n.$$

The velocity of sound in the given case is determined from the condition for the inception of maximum oscillation amplitude at the instant of resonance at the natural frequency of the tube and the frequency of the forced oscillations (the temperature computed from this condition turns out to be somewhat lower than the calculated thermodynamic temperature). At resonance frequencies the oscillation nodes and antinodes have fixed positions, and a velocity node always occurs at the combustion surface. In experiments where the frequency of the forced oscillations is equal to the natural frequency of the gas column in the tube with the latter closed at both ends, the pressure pulsation gauge is set up near the nozzle orifice plate. The steady flow velocity in the tube is calculated by means of the gasdynamic functions, and the acoustic perturbation velocities are calculated from the equation

$$\Delta V = \frac{1}{k} \cdot \frac{c \cdot \Delta P}{P}.$$

The heat flux is measured precisely at ten seconds of operation of the gas-jet generator. The relative pressure amplitude $\Delta P/P$ when the heat flux is measured at a pressure antinode does not exceed 1%. The relative acoustic velocity amplitude when the heat flux is measured at a velocity antinode is greater than 1.0. The relative pressure amplitude for nonresonance frequencies does not exceed 1%. The burning rate of the solid

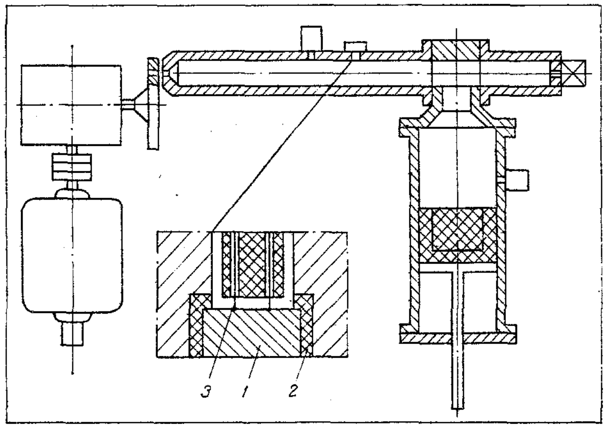


Fig. 3. Schematic of modified arrangement. 1) Pure iron cylinder; 2) insulation; 3) Chromel–Alumel thermocouple.

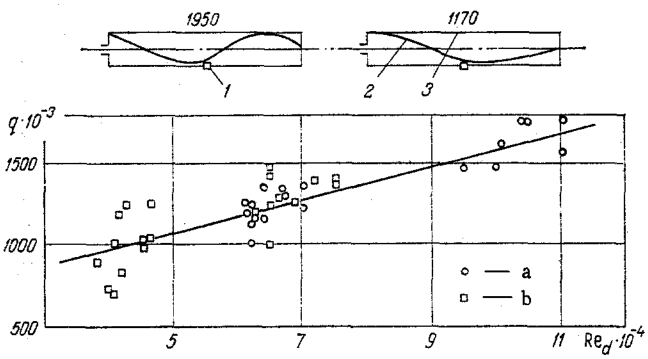


Fig. 4. Acoustic field patterns and experimental results on heat transfer in an acoustic field in the modified apparatus. a) Experimental results without oscillations; b) with resonance oscillations; 1) heat-flowmeter; 2) acoustic velocity distribution in standing wave; 3) calculated resonance frequency, Hz.

matter during all oscillations does not deviate from the nonoscillatory burning rate. The experimental results and acoustic field patterns are shown in Fig. 2. It is important to note that the frequencies of the injected oscillations do not exactly correspond to the calculated natural frequencies of the combustion products in the tube.

Calculations according to the convective heat-transfer equation [5]

$$\mathrm{Nu}_d = 0.036\,\mathrm{Re}_d^{0.8}\,\mathrm{Pr}^{0.4}\left(\frac{x}{d}\right)^{-0.2}\left(\frac{T}{T_w}\right)^{0.18},$$

in which

$$\mathrm{Nu}_d = \frac{q_c d}{(T - T_w)\,\lambda_g},$$

show (Fig. 2) that for the attendant velocities of the combustion products in the tube (they do not exceed 2 m/sec) the heat fluxes are not determined solely by convection. The flow in the tube without oscillations is assumed to be turbulent, because the actual combustion of condensed matter introduces considerable turbulence into the generated combustion products. We therefore estimate the radiative heat transfer according to the expression [5]

$$q_r = \varepsilon_g \cdot \frac{\varepsilon_w + 1}{2} \cdot 4.96\left[\left(\frac{T}{100}\right)^4 - \left(\frac{T_w}{100}\right)^4\right].$$

The values so obtained agree with the estimates of [5]. The dependence of the calculated total heat flux on the flow parameters of the combustion products is consistent with the experimental results obtained here without the generation of acoustic oscillations in the flow (Fig. 2). A certain discrepancy at small Reynolds numbers may be attributed to the presence of products of incomplete combustion of the condensed matter at low pressures corresponding to $Re_d = 10^4$ or less.

An analysis of the experimental results (Fig. 2) indicates the following.

1. The heat transfer associated with the generation of nonresonance oscillations in the flow of combustion products does not differ from the heat transfer without oscillations. The additional flow turbulence produced by the introduction of oscillations is probably small at the investigated levels of oscillation intensity and flow parameters of the combustion products, so that they do not tend to increase the heat transfer [1]. Other authors [3] have also reported that an increase in the heat flux with oscillations over the heat flux without oscillations is not observed when the excited frequency is far from resonance (90 ±45 Hz).

2. The heat transfer at a velocity node of the standing acoustic wave when resonance oscillations are generated in the flow does not differ significantly from the nonoscillatory heat transfer. This result does not concur with the results of [2, 4], in which a substantial decrease in the heat flux was observed at the given node, and is explained by the much higher temperature and pressure of the gas flow (by comparison with [2, 4]) and the lower oscillation level relative to these parameters.

3. The heat flux at a velocity antinode of the standing acoustic wave for resonance oscillations in the flow is appreciably greater than the heat flux without oscillations. This result is consistent with the results of [2-4].

To expand the range of the investigated flow parameters in the tube and the intensity level of the injected oscillations we have conducted experiments on a slightly modified apparatus (Fig. 3). The tests are carried out with and without the introduction of oscillations at resonance frequencies corresponding to the natural frequencies of the combistion products for the tube with one end open. A piezoelectric pressure pulsation gauge is set up at the closed end of the tube. The selected composition of the solid matter in the gas-jet generator and the selected tube diameter make it possible to carry out investigations in the range of pressures from 8 to 30 bar and Reynolds numbers from $4 \cdot 10^4$ to $11 \cdot 10^4$. The frequencies of the injected oscillations are between 800 and 2000 Hz. The calorimetric units are not positioned at velocity nodes and antinodes of the standing wave. The heat flux is measured precisely at three seconds of operation of the gas-jet generator. The relative amplitude of the pressure oscillations at the heat-flux measurement stations does not exceed 20% ($\Delta P/P < 0.2$). The maximum relative velocity amplitude $\Delta V/V$ is less than 1.0. The test results and acoustic field patterns in the tube are shown in Fig. 4.

It is seen that the heat flux with oscillations present does not differ from the nonoscillatory heat flux.

The mechanism of the action of acoustic oscillations on turbulent flow clearly involves nothing more than the production of additional turbulence or the creation of a system of wall vortices. In the given investigations the additional flow turbulence is probably small and does not significantly affect the heat transfer (at the investigated oscillation intensity levels); this conjecture is supported by the results of experiments with nonresonance (see Fig. 2) and resonance frequencies (see Fig. 4).

For the case of nonresonance oscillations with a relative velocity amplitude $\Delta V/V > 1$ a wall vortex system is formed in the flow, where it alters the heat-transfer conditions between the flow and the wall at a velocity antinode of the standing wave (Fig. 2). For resonance oscillations with a relative velocity amplitude $\Delta V/V < 1$ the wall vortex system is not formed, and the heat-transfer conditions between the flow and the wall remain inchanged. The acoustic frequency was not observed to have any influence on the heat-transfer process in the given investigation.

## NOTATION

$f$, oscillation frequency, Hz; c, velocity of sound, m/sec; $l$, length of tube, m; n, number of quarter-wavelengths fitting the length of the tube; $\Delta V$, acoustic velocity, m/sec; k, adiabatic exponent; $\Delta P$, acoustic pressure, bars; P, mean pressure, bars; Nu, Nusselt number; Re, Reynolds number; Pr, Prandtl number; d, tube diameter, m; x, distance from beginning of gas flow to point of heat-flux measurement, m; T, temperature of combustion products, °K; $T_w$, wall temperature, °K; $q_c$, convective heat flux, W/m$^2$; $\lambda_g$, thermal conductivity of gas, W/m°K; $q_r$, radiative heat flux, W/m$^2$; $\varepsilon_g$, emissivity of combustion products; $\varepsilon_w$, emissivity of wall.

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# EXPERIMENTAL STUDY OF HEAT TRANSFER IN THE FLOW OF ANOMALOUSLY VISCOUS LIQUIDS IN CIRCULAR AND ELLIPTICAL PIPES


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Results are presented of an experimental study of heat transfer in the laminar flow of anomalously viscous liquids in circular and elliptical pipes.


The study of heat transfer in the flow of anomalously viscous liquids in pipes having cross sections of various shapes is of great theoretical and practical interest for a number of branches of industry. Unfortunately, most of the work on this problem has been devoted to a study of heat transfer in the flow of anomalously viscous liquids in circular and plane-parallel channels and pipes.

The purpose of the present work is to determine experimentally the laws of heat transfer in the laminar flow of anomalously viscous liquids in circular and elliptical pipes. The studies were performed on an experimental arrangement shown schematically in Fig. 1. The working liquid was drawn from the preliminary temperature control tank 1 by the pump 2 through a closed circuit consisting of the pressurized tank 3 with an overflow device, the buffer tank 4, the heat-transfer and damping chambers 5 and 6, the working element 7, and the mixing chamber 11. The flow rate of the liquid was controlled by adjusting the pump speed and the valve 12. The pipe wall of the working portion 7 was kept at a constant temperature by sectional cascade electric heaters. The wall temperature was measured with a set of Chromel—Copel thermocouples 8 made of 0.2 mm diameter wire, and the potentiometer 10. The removable working portions of the arrangement were made of copper and brass pipes 1500 mm long with inner surface roughness corresponding to the 8th class of surface finish. The circular pipes had diameters of 13.6 and 19.8 mm and the semiaxes of the elliptical pipes were $6.3 \times 2.4$ and $3 \times 1$ mm. The experiments were performed under steady thermal and hydrodynamic conditions.

Preliminary work with the experimental setup was performed with transformer oil. The working liquids were 5 and 7.5% aqueous solutions of sodium carboxymethylcellulose (CMC) and 3 and 8% aqueous solutions of polyvinyl alcohol (PVA). The rheological characteristics of the solutions were determined on a Rheotest rotary viscometer. The results of the viscometric measurements in the 20 to 80°C temperature range are shown in Fig. 2. In the range of shear rates investigated the rheological behavior of the solutions is well described by the equation [1, 2]


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