THERMAL-MECHANICAL CHARACTERISTICS OF STATIONARY AND PULSATING GAS-FLOWS IN A GAS-DYNAMIC SYSTEM
In Relation the Exhaust System of an Engine

by

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Original scientific paper
https://doi.org/10.2298/TSCI201029171P

It is a relevant objective in thermal physics and in building reciprocating internal combustion engines (RICE) to obtain new information about the thermal-mechanical characteristics of both stationary and pulsating gas-flows in a complex gas-dynamic system. The article discusses the physical features of the gas dynamics and heat transfer of flows along the length of a gas-dynamic system typical for RICE exhaust systems. Both an experimental set-up and experimental techniques are described. An indirect method for determining the local heat transfer coefficient of gas-flows in pipe-lines with a constant temperature hot-wire anemometer is proposed. The regularities of changes in the instantaneous values of the flow rate and the local heat transfer coefficient in time for stationary and pulsating gas-flows in different elements of the gas-dynamic system are obtained. The regularities of the change in the turbulence number of stationary and pulsating gas-flows along the length of reciprocating internal combustion engines gas-dynamic systems are established (it is shown that the turbulence number for a pulsating gas-flow is 1.3-2.1 times higher than for a stationary flow). The regularities of changes in the heat transfer coefficient along the length of the engine's gas-dynamic system for stationary and pulsating gas-flows were identified (it was established that the heat transfer coefficient for a stationary flow is 1.05-1.4 times higher than for a pulsating flow). Empirical equations are obtained to determine the turbulence number and heat transfer coefficient along the length of the gas-dynamic system.

Key words*: reciprocating engine, exhaust system, stationary and pulsating flows, turbulence number, local heat transfer, empirical equations*

Introduction

It is a relevant objective in thermal physics to obtain new data on the thermo mechanical characteristics of streams in both stationary and pulsating flows in complex gas-dynamic systems. It is known that the pulsations of the induced gas-flow, due to gas-dynamic non-stationarity, can significantly change the flow structure, its gas dynamics and heat transfer characteristics. This significantly affects functioning of technical devices in a medium moved by a pulsating mode [1]. For example, the processes in the gas exchange systems of RICE are non-stationary [2, 3]. It should be noted that even today, some scientists and specialists study gas dynamics and heat transfer of flows in both exhaust and output systems of engines in a stationary mode of gas-flow [4, 5]. It is evident that this approach can lead to unjustifiably large

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errors in obtaining certain thermomechanical or operational characteristics in comparison with real RICE indicators.

The currently available fundamental works on pulsating gas-flows describe in some detail the effect of induced pulsations on velocity profiles, shear stress, laminar-turbulent transition, wave structure of the flow, heat transfer*, etc.* [6-8]. However, the results obtained often have a limited area of application and are valid only for some special cases due to the multifactorial physical nature of the problems in that area. Therefore, the data of different authors may be contradictory. For example, some research on heat transfer in pulsating modes can demonstrate both increased and decreased heat transfer. Let us take a closer look at publications on this topic.

A number of works have shown that the intensity of gas-flow heat exchange under non-stationary conditions can decrease by 1.2-2.2 times in comparison with the stationary case. [9-11]. At the same time, some articles give other results, which indicate that gas-dynamic unsteadiness either does not significantly affect the intensity of heat transfer in pipes, or conversely, leads to an increase by 30%-40% compared to a steady flow. So, in the studies of Wang and Zhang [12] with numerical simulation, it was found that the local heat transfer coefficient increases by an average of 30% with a pulsating gas-flow. Similar results, although based on experimental research, were obtained by Davletshin *et al.* [13]. Finally, similar results can also be found in Park *et al.* [14] and Chung and Tucker [15], in which it is shown that gas-dynamic unsteadiness has a weak effect on the heat exchange characteristics of gas-flows and causes a change in the heat transfer coefficient within 15%.

The intensity of heat transfer in pulsating flows is influenced by a large number of factors, including the velocity defect law, the acoustic properties of the medium, the geometry of the gas-dynamic system, the location of the pulsator, and the frequency and amplitude of pulsations. Therefore, in order to better understand the physical mechanism of the influence of the gas-dynamic non-stationarity of flows on heat transfer, it is necessary to follow up the research in various areas under different boundary conditions in order to replenish the knowledge base about those processes. One of the applied aspects in this area of research is the RICE exhaust system. New data on the gas dynamics and heat transfer of unstable processes in the exhaust system can be useful for refining valve strength calculation methods [16], improving the environmental performance of the engine [17], reducing exhaust noise [18], creating systems for the conversion of the thermal energy of exhaust gases [19, 20] and updating the design of exhaust systems [21, 22].

Thus, based on the literature review, the following tasks were formulated:

- To develop experimental installations and methods for conducting experiments to study the thermal-mechanical characteristics of stationary and pulsating gas-flows in a RICE exhaust system.
- To carry out a comparative analysis of the turbulence number values in different elements of the exhaust system for stationary and pulsating gas-flows.
- To compare the intensity of heat transfer between stationary and pulsating gas-flows in the RICE exhaust system.
- To obtain empirical equations for calculating the turbulence number and the heat transfer coefficient in different elements of the RICE exhaust system.

Features of thermal-mechanical processes and experimental set-up

This article presents the outcomes of experimental research on the gas dynamics and heat transfer of both stationary and pulsating flows in a gas-dynamic system with the main elements and boundary conditions typical of RICE exhaust systems.

First, the physical features of gas-dynamic processes in the RICE exhaust system will be considered. It should be noted that the driving factor of the flows in the system under examination is the excess pressure in the cylinder, *i.e.* after opening the valve, gas-flows from the cylinder through the exhaust system's elements, with their subsequent discharge into the atmosphere. As shown in the overview, at present there are not many experimental studies on gas dynamics and heat transfer in gas-dynamic systems characteristic of RICE exhaust systems under the gas-dynamic non-stationarity conditions. This is partly due to the complexity of the research target, since gas-flows in engine exhaust systems pulsate (the valve assembly operates at frequencies from 10-100 Hz and higher). Pulsation processes are associated with the periodic nature of the working cycle of the RICE system and are determined by the crankshaft rotation frequency value (engine speed). One of the key features of the pulsations of gas-flows in the exhaust system is that during the engine operating cycle, the valve is closed most of the time (about 60%-65%) and, therefore, gas is not supplied from the cylinder to the system. The cycle time, in turn, depends on the crankshaft speed; in this research, it ranged from 0.04 second (at n_{cs} = 1500 rpm) to 0.2 second (at n_{cs} = 300 rpm).

Another important feature of pulsating flows in the exhaust system that determines the gas-dynamic conditions of heat exchange with the channel walls is the presence of return flows. This phenomenon is explained by the fact that the gas-flow moving out of the cylinder is reflected from the turbine wheel blades, muffler or atmosphere and begins to move in the opposite direction. After returning to the valve assembly, the flow is reflected again, which again forces it to exit [23]. The studied reverse return flows in the exhaust system are possible precisely due to the fact that the valve is open for only 35%-40% of the entire engine operating cycle.

To study pulsating flows in a gas-dynamic system, an experimental set-up, (shown in fig. 1(a), was developed.

The experiment sequence was as follows. The tank (with a volume of approximately 50 L) was supplied with compressed air from an external source. The pressure values at the tank inlet and tank outlet were determined with pressure gauges. When the required pressure was reached, compressed air was supplied to the cylinder. The overpressure value in the cylinder was 0.05-0.2 MPa. The air temperature was from 25-33 °C. Then it passed through the exhaust valve, the channel in the cylinder head and the exhaust pipe. Then it was discharged into the atmosphere. A pulsating flow was created by the cyclic operation of the outlet valve. To drive it, an electric motor connected to a camshaft on the cylinder head was used. The camshaft rotation speed, n_{cs}, varied from 300-1500 rpm (respectively, the valve operated at a frequency of 5-25 Hz). It should be noted that the gas temperature can reach 700 °C in the exhaust system of a real engine. Therefore, the results of research on laboratory stands should be clarified during operational tests of RICE.

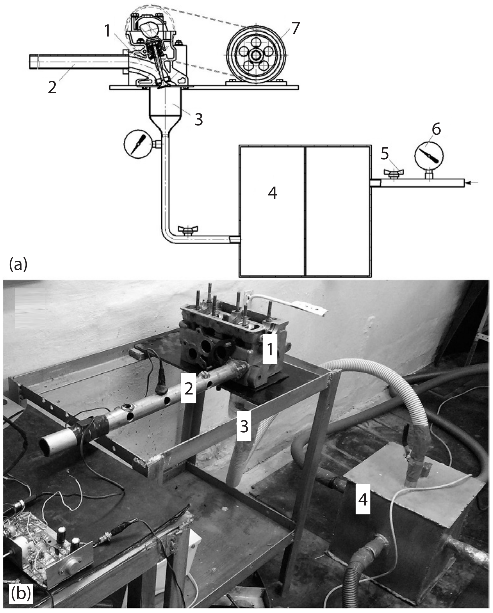

Figure 1. Set-up (a) and photograph (b) of the experimental stand for stationary and dynamic charging of the piston engine exhaust system; *1 – cylinder head with a valve mechanism, 2 – exhaust pipe, 3 – piston chamber, 4 – flow stabilization tank, 5 – control valve, and 6 – pressure control gauges*

When studying stationary gas-flows, the outlet valve was constantly in the open position, fig. 1(b). At the same time, the air, before being fed into the cylinder, also first entered the tank (where it was stabilised). After that, it entered the exhaust pipe from the cylinder through the open valve and, when passing through it, was discharged into the atmosphere.

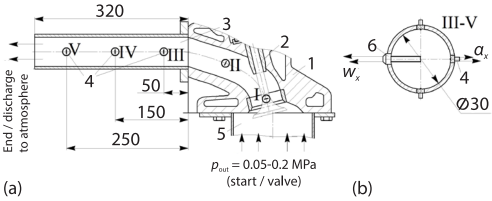

Figure 2. Configuration of the exhaust system with an indication of the installation locations of the sensors (a) and a cross-section of the control section (b); *1 – cylinder head, 2 – exhaust valve, 3 – exhaust pipe, 4 – anemometer sensors for determining the local heat transfer coefficient, 5 – piston chamber, 6 – anemometer sensors for determining the instantaneous values of flow rate, and I-V – control sections*

The configuration of the exhaust system and the sensor location are shown in fig. 2.

In this research, a straight exhaust pipe with a length of 320 mm and a standard cylinder head from a vehicle engine (with a cylinder diameter of 82 mm) were used. The start of the gas-dynamic system was the valve seat, while the outlet was the end of the exhaust pipe-line. In the system, there were five control sections, at distances of 15 mm (Section I), 85 mm (II), 200 mm (III), 300 mm (IV), and 400 mm (V). In each control section, sensors to measure the instantaneous values of the flow velocity, w_x, and local heat transfer coefficients, α_x, were installed. The methods for determining the basic physical quantities are presented in the following section of the article. In the first control section, the gas-flow rate was not measured due to design restrictions (the presence of the valve). In each control section, the local heat transfer coefficient was sequentially measured in four places separated from each other at an angle of 90°. Then those values were averaged.

Instrumentation and research methods

Determination of gas-dynamic characteristics of flows in channels

To analyse the gas-dynamic characteristics of flows in the systems under research, the functions of instantaneous values of the air-flow velocity $w_x = f(\tau)$ and the turbulence number , Tu, of the flows were chosen as indicators.

The instantaneous velocity (averaged by the channel cross-section) of the gas-flow in the pipe-line's control section was measured with a constant temperature hot-wire anemometer [24]. The relative standard uncertainty in determining the instantaneous values of the gas-flow rate was 4.9%.

The turbulence number was determined as the ratio of the root-mean-square pulsation velocity component to the average velocity of the flow:

$$Tu = \frac{\sqrt{w'^2}}{\overline{w}} \tag{1}$$

where w' is the pulse-coupled components of velocity projections and $\overline{w}$ – the average gas-flow rate.

The difference of calculation $\overline{w}$ from that for stationary and pulsating flows consisted of the following. If in the stationary case $\overline{w}$ was determined as the expected value of the function $w_x = f(\tau)$, then, in the case of a pulsating gas-flow, the average flow rate was found with phase averaging by the complete engine operating cycle:

$$\overline{w}(\tau) = \frac{\sum_{n=1}^{N} w_x(\tau + nT)}{N} \tag{2}$$

where N is the number of used engine operating cycles during averaging (normally N = 5-7) and T [s] – the time per working cycle.

As a result, a function of the average local air-flow rate in time was obtained $\bar{w} = f(\tau)$ for a pulsating air-flow in gas exchange systems, relative to which the pulsation components were determined in a non-stationary mode of gas-flow.

An indirect method for determining the local heat transfer coefficient of gas-flows in pipe-lines

In this paper, an indirect method for determining the heat transfer coefficient with a constant temperature hot-wire anemometer and a thread sensor is proposed. This makes it possible to compare the intensity of heat transfer in gas-dynamic systems with different configurations and under different boundary conditions.

The proposed measuring system consists of two main elements, a hot-wire anemometer and a sensor. In this method, a hot-wire anemometer at a constant temperature containing a resistive temperature-sensitive element (hot-wire anemometer sensor) connected to a bridge circuit (Wheatstone bridge) and a feedback amplifier connected to a measuring bridge were used. The hot-wire anemometer sensor had an original design, fig. 3(a). Its basic element was a fluoroplastic substrate with a thermal conductivity coefficient of 0.07 W/mK. A nichrome thread with a diameter of 5 μm and a length of 5 mm, which is the sensitive element of the hot-wire anemometer sensor, was passed over the surface of the substrate with slight tension.

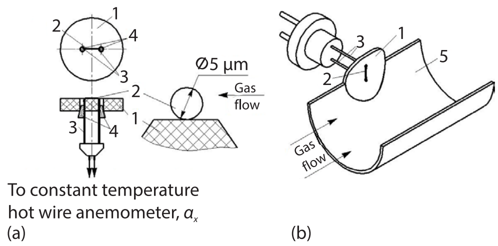

Figure 3. The design of the hot-wire anemometer sensor (a) and the diagram for installing the sensor in the pipe-line (b); *1 – fluoroplastic substrate, 2 – sensitive element of the sensor, 3 – current-conducting rods, 4 – wedges (wood), and 5 – pipe-line*

The hot-wire anemometer's thread sensor measures the local friction stresses, f_x, on the channel surface, on the basis of which it is possible to determine the heat flux density, and subsequently the local heat transfer coefficient, if the physical properties of the gas and other flow parameters are known. The method for installing the proposed sensor in the pipe-line is shown in fig. 3(b).

In this method, the determination of the local coefficient at the wall boundary and the gas-flow is based on the effect of the hydrodynamic analogy of heat transfer (Reynolds analogy). It implies the unity of the impulse and heat transfer processes in a turbulent flow and establishes an unambiguous relationship between heat transfer and hydraulic resistance, *i.e.* friction on the heat exchange surface and heat transfer through the surface are inter-related [25].

In the proposed method, the determination of the local heat transfer coefficient can be performed with indirect calibration according to known empirical relationships. The method is based on the basic local heat transfer index of the well-studied fiducial process, for which stationary heat transfer in a long straight pipe was chosen ($l/d \geq 50$). Thus, calibration consisted of correlating the calculated heat transfer coefficient, α [Wm^{-2}K^{-1}], for a long straight pipe and the signal value from the hot-wire anemometer sensor, U. A calibration curve was obtained in the form of a functional relationship between the voltage at the output of the hot-wire anemometer, U, and the local heat transfer coefficient, α_x.

The applicability of this approach for determining the local heat transfer coefficient for various heat transfer purposes is based on the Kutateladze-Leontiev method (friction and heat transfer laws) [26]. They proved that it is possible to obtain the friction and heat transfer laws of a certain standard (reference) process, and then extend them to more complex cases, *i.e.* those laws are conservative to changes in border-line conditions. The relative standard uncertainty in determining the local heat transfer coefficient with this method was 10.5%. The proposed method has been successfully applied to the analysis of thermomechanical processes in gas-dynamic systems typical of piston engines [27, 28].

Gas dynamics and heat exchange of flows in the exhaust system of the engine in different gas-flow modes

During this research, the gas-dynamic and heat-exchange characteristics of both stationary and pulsating flows along the length of a gas-dynamic system typical of a RICE exhaust system were examined. First, let us consider the primary data for stationary gas-flows in the control sections in the cylinder head and at the end of the exhaust pipe-line (numbered II and V in fig. 2, respectively), which are shown in fig. 4.

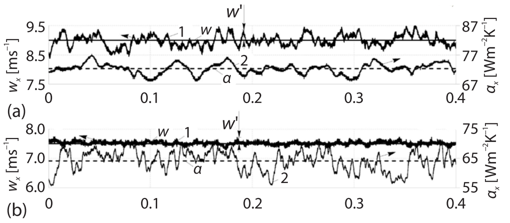

Figure 4. Dependences of the instantaneous values of the flow rate – 1, w_x and the local heat transfer coefficient, α_x, in time, τ, for a stationary gas-flow mode in different elements of the exhaust system; (a) Section II, w = 9.0 m/s and (b) Section V, w = 7.5 m/s

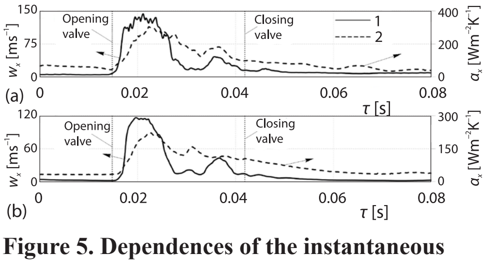

Figure 5. Dependences of the instantaneous values of the flow rate – 1, w_x, and the local heat transfer coefficient, α_x, in time, τ, for the pulsating mode of gas-flow in different elements of the exhaust system; (a) Section II, p_{out} = 0.2 MPa and (b) Section V, p_{out} = 0.2 MPa

Figure 4 shows that in control Section II, the amplitudes of the velocity pulsations relative to the mean velocity value are significantly higher than those for control Section V. The high pulsation components of the velocity in Section II can be explained by the close location of the valve units, which creates turbulence for the flow. The flow movement along the exhaust system somewhat calms (stabilises) the flow. In the last control section (before the gas is discharged into the atmosphere), the pulsation components of the velocity are low. In turn, the maximum values of the local heat transfer coefficient α_x in control Section II are 10%-12% higher than in Section V, fig. 4. In this case, the amplitude value of the pulsations α_x relative to the average value of the heat transfer coefficient, α, remains at the level of 5 W/m^{2}K along the entire exhaust system.

Primary data for a pulsating gas-flow along the exhaust system are shown in fig. 5.

Figure 5 shows that the maximum velocity values in control Section II (the channel in the cylinder head) are approximately 20% higher than in Section V. There are noticeable fluctuations in velocity when the valve is open in Section II compared to Section V. These fluctuations in the gas-flow are also a consequence of the flow passing through the exhaust valve, [29]. Also, after closing the exhaust valve, the flow rate does not equal zero: there are small surges (fluctuations). This indicates the presence of reverse gas-flows in the gas-dynamic system. They are of an identical nature (this process does not develop), and the following exhaust process begins with zero values of the gas-flow rate (a static state in the system). In turn, the maximum values of the local heat transfer coefficient are approximately 30% higher in Section II compared to the values of α_x in Section V, in

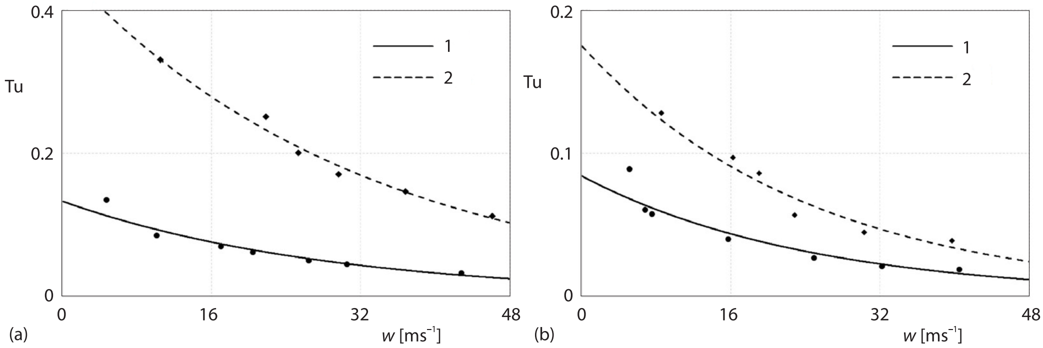

Figure 6. Dependences of the turbulence number on the average flow rate, *w*, in the channel at 1 – stationary and 2 – pulsating gas-flow modes in different elements of the exhaust system; (a) Section II and (b) Section V

fig. 7. Thus, the heat transfer decrease rate along an exhaust system with a pulsating gas-flow is 2-3 times higher than that with a steady flow.

To quantitatively evaluate the pulsation component of the gas-flow velocity along the exhaust system, the turbulence number was determined for all control sections for both stationary and pulsating gas-flow modes, fig. 6.

Figure 6 shows that in Section II, the values of the turbulence number for a pulsating flow are 1.3-2.1 times higher than those for a stationary gas-flow. This indicates that flow pulsation is a more significant factor in the gas-dynamic characteristics of gas-flow than the geometric configuration of the exhaust system. At the same time, at flow velocities, *w*, from 5-20 m/s, the differences in turbulence number values between the pulsating and stationary flows are 1.7-2.1 times; with an increased speed *w*, the difference in turbulence number decreases to 1.3.

It was found that in control Section V, the differences in the turbulence number values for both stationary and pulsating flows slightly decrease (compared with Section II) and are 1.1-1.9 times. At the same time, it remains the case that turbulence number for a pulsating flow has higher values than for a stationary flow. At flow rates, *w*, up to 20 m/s, the differences in turbulence number values between pulsating and stationary flows are 1.4-1.9 times; with an increase in the speed *w*, the difference in turbulence number values decreases to 1.1.

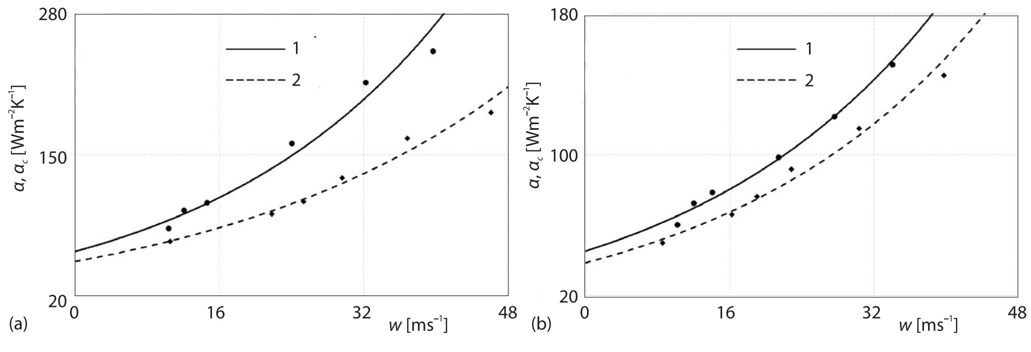

Figure 7. Dependences of heat transfer coefficients α (α_c) on the average flow rate, *w*, in the channel in – 1 stationary and – 2 pulsating gas-flow modes in different elements of the exhaust system; (a) Section II and (b) Section V

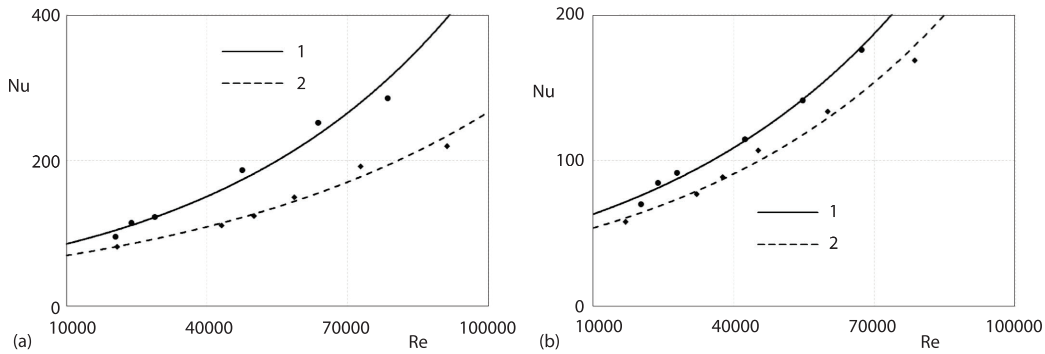

Figure 8. Dependences of the Nusselt number on the Reynolds number in the channel at 1 – stationary and 2 – pulsating gas-flow modes in different elements of the exhaust system; (a) Section II and (b) Section V

It is possible to trace the influence of various gas-dynamic conditions on the intensity of heat transfer in different parts of the exhaust system with fig. 7. The figure shows the averaged values of the local heat transfer coefficients, α_x, in different control sections. It should be remembered that in each control section, α_x was determined at four places spaced 90° from each other. After this, the obtained values were averaged. This procedure was carried out both for a stationary flow and a pulsating one. At the same time, α_x for the pulsating mode of gas-flow was averaged for a period of time equal to a full RICE working cycle. As a result, the heat transfer coefficient values were obtained for the stationary α and pulsating gas-flow α_c per cycle, fig. 7.

Figure 7 shows that in control Section II, the heat transfer coefficient for the stationary flow α is 1.05-1.4 times higher than the heat transfer coefficient for the pulsating flow α_c. Moreover, with an increased air-flow rate, the difference in the α and α_c values increases. It can be concluded that the gas-dynamic non-stationarity of the flow causes a decrease in the heat transfer intensity up to 40% in comparison with the stationary flow. A similar pattern is maintained for control Section V, in which the heat transfer coefficient α is 1.05-1.15 times higher than the heat transfer coefficient for the pulsating flow, α_c. This tendency persists in the entire gas-flow velocity mode under research. The data on the intensity of heat transfer in dimensionless form, Nu = f(Re), confirmed the aforementioned analysis, fig. 8.

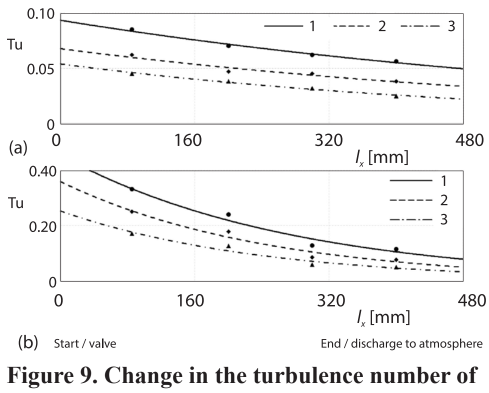

Figure 9. Change in the turbulence number of the flow along the length of the exhaust system at stationary (a) and pulsating (b) gas-flow modes for different *w*: *1 – w = 10 m/s, 2 – w = 20 m/s, and 3 – w = 30 m/s*

Additionally, data on the turbulence number turbulence number for both stationary and pulsating flows along the entire length of the exhaust system for different average speeds, w, were obtained, fig. 9.

Figure 9(a) shows that at a steady flow, turbulence number tends to monotonically decrease as the flow moves along the length of the exhaust system. So, at a flow rate w = 10 m/s, the turbulence number decreases linearly from 0.1-0.05. Similar patterns of turbulence number change along the length, l_x, of the RICE exhaust system are also established for other flow rates. The obtained patterns can be reasonably (with an error of 10%) described:

$$\mathrm{Tu} = -8\cdot 10^{-5} l_x + \frac{0.309}{\sqrt{w}} \tag{3}$$

where l_x is the linear dimension of the exhaust system (l_x from 0 to 500 mm) and w – the average flow rate (w from 5-45 m/s).

From fig. 9(b), it can be seen that with a pulsating flow, turbulence number also tends to decrease as the flow moves along the length of the exhaust system. This is typical of all flow rates, w, under research. The regularity of turbulence number change along the length of the exhaust system is well described (with an error of no more than 5%):

$$\mathrm{Tu} = \frac{1.52\mathrm{e}^{-0.004 l_x}}{\sqrt{w}} \tag{4}$$

where l_x is the linear dimension of the exhaust system (l_x from 0 to 500 mm) and w – average flow rate (w from 5 to 45 m/s).

The heat transfer coefficient measurement regularities along the length of the RICE exhaust system are shown in fig. 10. The data on the intensity of heat transfer in the form of the function $\mathrm{Nu} = f(l_x)$ are shown in fig. 11.

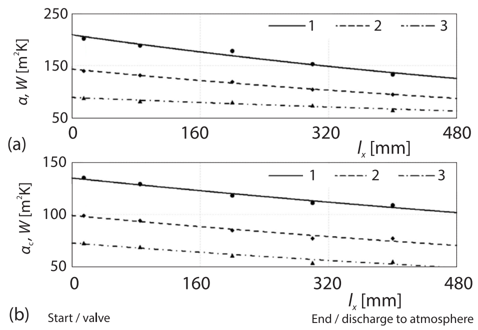

Figure 10. Change in the heat transfer coefficient α (α_c) along the length of the exhaust system in both stationary (a) and pulsating (b) gas-flow modes for different values of the average flow rate: *1 – w = 10 m/s, 2 – w = 20 m/s, and 3 – w = 30 m/s*

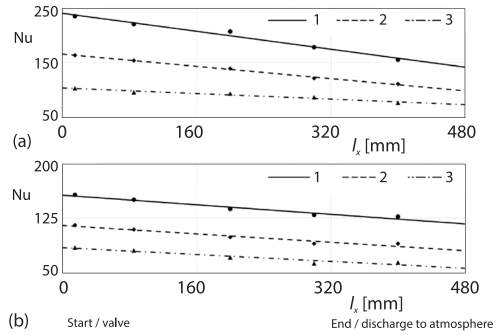

Figure 11. Change in the Nusselt number along the length of the exhaust system at stationary (a) and pulsating (b) gas-flow modes for different values of the Reynolds number: *1 – Re = 10000, 2 – Re = 20000, and 3 – Re = 30000*

From figs. 10 and 11, it can be seen that in a stationary flow, the heat transfer coefficient tends to monotonically decrease as the flow moves along the length of the exhaust system. This pattern can be traced for all flow rates, w, under research. The change in the heat transfer coefficient, α, along the length, l_x, of the exhaust system in a stationary flow mode is reasonably described (with an error of no more than 11.5%):

$$\alpha = -0.117 l_x + 15.2 w^{0.759} \tag{5}$$

where l_x is the linear dimension of the exhaust system (l_x from 0-500 mm) and w – the average flow rate (w from 5-45 m/s).

From fig. 10(b), it can be seen that with a pulsating flow, the heat transfer coefficient also tends to gradually decrease as the flow moves along the length of the RICE exhaust system. A change in the heat transfer coefficient, α_c, along the length, l_x, of the exhaust system in a pulsating flow mode is well described (with an error of no more than 7.6%):

$$\alpha_c = -0.0612 l_x + 20 w^{0.551} \tag{6}$$

where l_x is the linear dimension of the exhaust system (l_x from 0-500 mm)and w – the average flow rate (w from 5-45 m/s).

In the applied aspect, the data on the turbulence number and heat transfer intensity in different elements of the exhaust system can be useful for improving the design quality of gas exchange systems for piston engines with and without turbocharging, for improving the accuracy of calculating temperature stresses in parts and units of gas exchange systems and for clarifying mathematical models of gas exchange processes.

Conclusions

Based on the conducted research, the following main conclusions were reached.

- Experimental installations, a measuring system and experimental methods were developed and tested. These allow for research on both stationary and non-stationary thermo mechanical processes in a gas-dynamic system typical of the piston engine's exhaust system.
- Regularities of changes in the instantaneous values of the flow velocity and the local heat transfer coefficient in time for both stationary and pulsating gas-flows along the length of the gas-dynamic system were obtained.
- It was found that in the engine exhaust system, the turbulence number for a pulsating gas-flow mode is 1.3-2.1 times higher than for a stationary mode.
- It was shown that the turbulence number value tends to decrease as the flow moves along the length of the exhaust system, which is typical of the stationary and pulsating gas-flow modes
- Empirical equations based on empirical functions that describe the change in the turbulence number value along the length of the exhaust system for both stationary and pulsating gas-flow modes were obtained.
- It was found that the heat transfer coefficient for a stationary flow was 1.05-1.4 times higher than for a pulsating flow.
- It was found that the heat transfer coefficient tends to monotonically decrease when the flow moves along the length of the exhaust system, which is typical of both stationary and pulsating gas-flow.
- Empirical equations on the basis of empirical dependencies that describe the change in the heat transfer coefficient along the length of the exhaust system were obtained for both stationary and pulsating gas-flow modes.
- The obtained data have practical application in piston engine building:
  – prediction of the local heat transfer in exhaust systems of engines with heat recovery of exhaust gases which will increase their effectiveness,
  – determination of the dynamics of distribution of temperature stresses in the parts and modules of gas exchange systems and, accordingly, finding the thermal stresses in them, and
  – development of the perspective exhaust systems for the modernisation of existing engines or the development of new RICE.

Further research plans are to develop methods for controlling the gas dynamics and heat exchange of pulsating flows in the exhaust system in order to increase the efficiency of the RICE.

Acknowledgment

The work has been supported by the Russian Science Foundation (Grant No. 18-79-10003).

Nomenclature

d – pipe-line diameter, [mm]
l – linear dimension (measured from the seat of the exhaust valve), [mm]
n – shaft speed, [rpm]
p – static flow pressure, [MPa]
Tu – turbulence number, [–]
w – flow rate, [ms^{-1}]

Greek symbols

α – heat transfer coefficient, [$Wm^{-2}K^{-1}$]
τ – time, [s]

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Paper submitted: October 29, 2020
Paper revised: March 15, 2021
Paper accepted: March 27, 2021