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AERODYNAMIC AND THERMOTECHNICAL TEST RUN OF AN AIR-GAS REGENERATIVE HEAT EXCHANGER

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The article represented the experimental test bench, methods and results of aerodynamic and thermotechnical investigations of characteristics and operation modes of shell-and-tube direct flow heat exchanger.

The plant (fig.1) consists of a heat source 1, heat exchanger, including calefacient flue 4 of circular section, with the diameter d_g, a boiler air heater of square section with inlet and outlet pipe branches.

For registering temperatures of calefacient and heated carrier and temperatures of surfaces, glass 3, 5, 8 and contact 9, 10, 11, 12 thermometers are set down.

For registering speed of movement and consumptions of calefacient and heated air carrier, vane anemometers 13, 14 are used on the plant. For determining the consumptions of flared gas on the plant, gas meter 15 is used, and micromanometer 16 is used for registering gas pressure.

When kindle gas, using a burner 6, products of gas combustion and induced air form mixture of heating gas, directed to a flue 4 by a convective flow, which has inlet $t_{PG}^{\acute{i}}$ and outlet $t_{GC}^{\hat{e}}$ temperature, registered by a thermometer 5. Mean temperature of the heat-release surface $t_{9,10}^{\text{ср}}$ of a flue 4 is measured by contact thermometers 9, 10 and found from the formula:

$$t_{9,10}^{\text{ср}} = \frac{t_9^{\text{н}} + t_{10}^{\text{н}}}{2}, \qquad (1)$$

where $t_9^{\text{н}}, t_{10}^{\text{н}}$ are respectively inlet and outlet temperatures of the surface of a flue, °C.

Amount of heat Q_{GCP}, extended for warming ventilation air from the surface of flue, will be defined according to the formula:

$$Q_{\text{гсп}} = K_{Г} \cdot F_{\text{ПГ}} \cdot \left(t_{9,10}^{\text{ср}} - t_{8,14}^{\text{ср}}\right), \qquad (2)$$

where $\hat{E}_G$ – is a heat-transfer coefficient through a flue wall 4, Вт/м^2 °C;

F_{PG} – heat-release surface of a flue, sq.m;

$t^{\text{ср}}_{9,10}$ – mean temperature of mixed gas in a flue, °C

$t^{\text{ср}}_{8,14}$ – mean temperature of warmed ventilation air, °C.

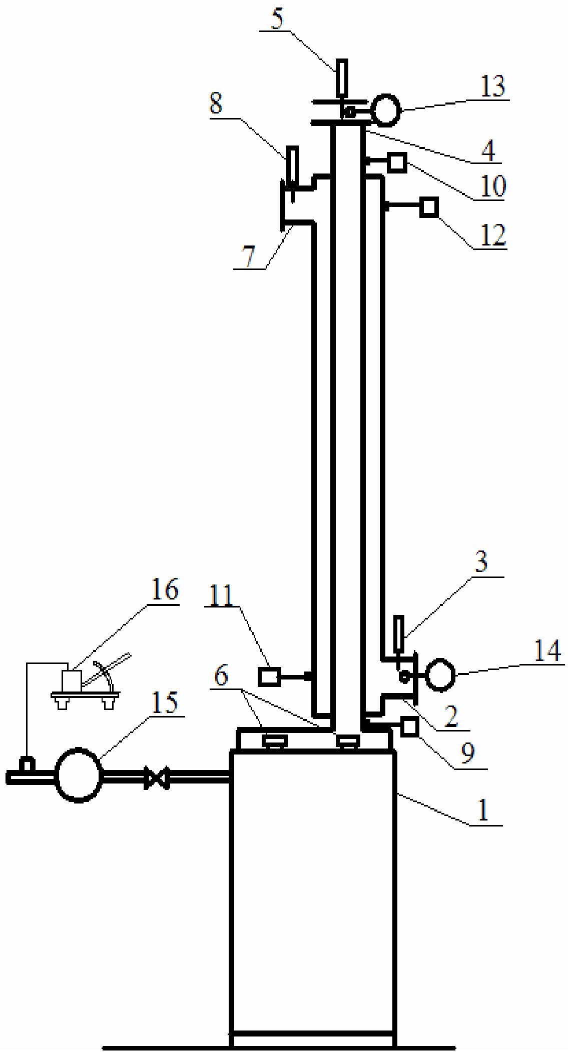

Fig. 1. The plan of the experimental test bench for testing a heat exchanger

a heat-transfer coefficient K is defined by the following formula:

$$K = \frac{1}{\frac{1}{\alpha_{\hat{A}}} + \frac{\delta}{\lambda} + \frac{1}{\alpha_{\acute{i}}}}, \qquad (3)$$

where δ – is wall thickness of a flue, m

λ – is thermal conductivity of the wall material of a flue, W/m of °C;

$\lambda_{\hat{A}}$ – is surface heat exchange coefficient from mixed gas to inner surface of a flue, which is defined by the following formula:

$$\alpha_{\hat{A}} = \frac{q}{\pi \cdot d_{\hat{A}} \cdot (\tau_{\hat{A}} - \tau_{\hat{A}\acute{i}})}, \tag{4}$$

q – heat loss, classified to running meter of the length of an air flue.

d_B – is the bore of an air flue, m

τ_B – is mean temperature of the outside surface of a heat pipeline, °C;;

$\tau_{\hat{A}\acute{i}}$ – is temperature of the inner surface of an air flue wall, °C;

$\alpha_{\acute{i}}$ – is heat-transfer coefficient from the outer surface of a flue, which is defined according to the formula:

$$\alpha_{\acute{i}} = 11,6 + 7\sqrt{V_{BB}}, \tag{5}$$

where V_{BB} – speed of warming ventilation air when moving inside a heat exchanger.

Heat-release surface of a flue is defined according to the formula [1],

$$F_{\ddot{I}\tilde{A}} = \pi \cdot d_g \cdot l_T, \tag{6}$$

where d_g – is the bore of a flue, m

l_T - is length of a heat exchanger, m

Under such conditions, amount of heat Q_{GCP}, transferred through the fuel's wall, is equal to the amount of heat, passing to warmed ventilation air inside the heat exchanger, i.e.

$$Q_{GCP} = Q_{PG}. \tag{7}$$

Q_{PG} is a numerically equal to the amount of abstracted from the surface heat fuel of a fuel, i.e.

$$Q_{PG} = \alpha_{\acute{i}} \cdot F_{PG} \cdot \left(t_{GC}^{\tilde{n}\eth} - t_{\hat{A}\hat{A}}^{\tilde{n}\eth}\right), \tag{8}$$

where all input values are known and defined by computing or experimentally.

Having found the value Q_{PG} from the formula (8), including the formula (7), we will define $t_{GC}^{\tilde{n}\eth}$ according to the formula (2):

$$t_{GC}^{\tilde{n}\eth} = t_{\hat{A}\hat{A}}^{\tilde{n}\eth} + \frac{Q_{GCP}}{k \cdot F_{PG}}, \tag{9}$$

and $t_{GC}^{\acute{i}}$ according to the formula:

$$t_{GC}^{\acute{i}} = 2 \cdot t_{GC}^{\tilde{n}\eth} - t_{G\tilde{N}}^{\hat{e}}, \tag{10}$$

Values of Re for defining $\alpha_{\hat{A}}$, are computed by calculating,

$$\mathrm{Re} = \frac{\upsilon_{G\tilde{N}} \cdot d_g}{\nu}, \tag{11}$$

where d_g – is a flue's bore 4, m

$\upsilon_{G\tilde{N}}$ - speed of mixed gas movement, measured with an anemometer experimentally (fig. 1), m/s;

ν – is a kinematic viscosity coefficient (for mixed gas mean temperature).

Details and results of the research are represented in the tables 1.1 and 1.2.

Table 1.1 – The results of the research

	t_3	t_8	t_5	t_9	t_{10}	t_{11}	t_{12}	$\Delta t_{3,8}$	$t_{9,8}^{\tilde{n}\eth}$	$t_{11,12}^{\tilde{n}\eth}$	n_1	$n_1/60$	n_2	$n_2/60$	υ_1
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
1	20,5	35,3	74	28,8	41,0	28,2	27,2	14,8	34,9	27,7	85	1,4	129	4,3	0,7
2	21,0	42,0	89	31,4	43,0	29,0	28,5	21,0	37,2	28,8	91	1,5	145	4,8	0,75
3	21,0	44,0	98	30,0	45,0	27,0	25,0	23,0	37,5	26,0	75	1,25	135	4,5	0,63
4	22,0	48,0	110	34,0	48,6	33,6	27,6	26,0	41,3	30,6	98	1,6	173	5,7	0,78
5	22,0	50,5	119	37,0	53,5	31,0	29,5	28,5	45,3	30,3	105	1,75	173	5,7	0,84
6	22,5	52,0	129	39,0	54,0	31,5	30,2	29,5	46,5	30,8	113	1,88	184	6,1	0,89
7	22,0	55,5	140	42,0	60,0	32,4	31,6	33,5	51,0	32,0	120	2,0	187	6.2	0,92

Table 1.2 – The results of the research

υ_2	L_1, м^3/с	V_T	$\alpha_í$	Q_{PG}, Вт	Δt_{in}	Δt_{out}	$\ln\frac{\Delta t_{in}}{\Delta t_{out}}$	$\Delta t_{in}-\Delta t_{out}$	$\frac{\Delta t_{in}}{\Delta t_{out}}$	$t^{ô}_{3,8}$	$\Delta t_{ñð}$	К	Re
17	18	19	20	21	22	23	24	25	26	27	28	30	31
2,0	0,0067	0,22	14,9	82,4	8,3	5,7	0,375	2,6	1,456	27,9	6,92	15,7	8070
2,2	0,0071	0,24	15,0	67,5	10,4	1,0	2,340	9,4	10,4	31,5	4,02	21,25	8877
2,1	0,006	0,2	14,7	58,1	9,0	1,0	2,200	8,0	9,0	32,5	3,64	20,2	8473
2,7	0,0074	0,25	15,1	75,2	12,0	0,6	3,000	11,4	20,0	35,0	3,8	25,05	10895
2,7	0,008	0,27	15,2	108,7	15,0	3,0	1,610	12,0	5,0	36,25	7,45	18,5	10895
2,8	0,0085	0,28	15,3	111,8	16,5	2,0	2,110	14,5	8,25	37,25	6,87	20,6	11298
2,9	0,0087	0,29	15,4	149,0	20,0	4,5	1,490	15,5	4,44	38,75	10,4	18,14	11702

According to the data above, we developed the method and obtained the experimental results for updating regularities of changing intensities of a direct flow of heat exchanger productivity for the purpose of using it in the systems of energy-efficient buildings.

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MATHEMATICAL MODELS FOR CHECKING LEAKAGE IN THE COMBUSTION CHAMBER OF AN INTERNAL COMBUSTION ENGINE

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We propose a mathematical model to present changes in the pressure within the above-piston space of an internal combustion engine depending on the temperature within the compression chamber. The model takes into account leakage through the piston ring-liner sealing.

Investigations referring combustion engine failures show that 31 % of combustion engines failed due to faults in the cylinder piston group, whereas 45 % of failures are due to the faults in fuel system [1]. Wear and tear of the engine may cause leakage of the working fluid through the piston rings or intake valves. The main leakage of the working fluid from the cylinder body occurs through the piston ring joints. These findings result from the numerous measurements conducted for the leakage of the working fluid down the piston rings [2]. With successful break-in of unworn set of piston rings, 80 % of leakage is due to expansion gaps in the piston rings [3]. The flow of the working fluid down the piston ring joints is at 250-300 m/sec. Thus, the contact time between the working fluid and piston ring sides is minimal, and the gas flow process may be regarded as adiabatic.

The given mathematical model was developed for internal combustion engines and can be used to simulate the engine performance and engine output modes. Therefore, it is significant for defining mechanical conditions of the cylinder-piston group and monitoring the overall performance of the engine.

Meanwhile, it should be noted that mathematical models proposed by Prof. V.G. Dyachenko are best applied to simulating complex physical processes within the combustion chamber of a diesel engine, and can be utilized for the processes of mass and heat transfer in the above-piston space. In the cases of compression travel, changes in the pressure can be determined using the following formula:

$$dp=\frac{kp}{V}\cdot\left[-\frac{1}{p}\cdot dM_{УТ}+\frac{(k\text{-}1)}{k}\cdot\frac{(\pm dQ_{T})}{p}-dV_{П}\right] \quad (1)$$

where $dM_{УТ}$ = the decrease of the working fluid mass in the above-piston space caused by the leak of the working fluid down the valves, $d\tau$ = piston rings over the computed period, ; k = adiabatic exponent of the real working fluid under temperature values as of the initial computed time interval; dQ_T = loss of heat energy of the piston walls surface to and from the working fluid within the computed time interval; $dV_П$ = volume changes of the above-piston space caused by the piston transfer over the computed time interval; P и V = respectively pressure and volume of the mixture at the start of computed time interval.