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Heat transfer from the working fluid in an air motor with a spool air distribution mechanism

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Abstract. Problem. *Heat transfer processes from the working fluid to the cylinder walls in heat engines play an important role. First, the thermal state of engine components, which determines operational reliability, depends on the parameters of the working fluid and on the intensity of heat transfer. Second, heat transfer from the working fluid represents a loss of thermal energy, which always reduces the conversion of the working fluid energy into useful work. Therefore, during engine design as well as during its improvement, reliable methods for calculating heat transfer from the working fluid to the cylinder walls are required.* ***Purpose.*** *The aim of this study is to improve the method for calculating heat transfer from the working fluid to the walls, taking into account the design features of a pneumatic motor, and to determine the amount of heat loss that could otherwise be converted into useful work.* ***Methodology.*** *It is proposed to study heat transfer separately in the pneumatic motor cylinder and in the channel connecting the spool valve with the cylinder. This approach makes it possible to account for significant heat losses in the channels and to confirm them using Joule's law.* ***Results.*** *Processing of experimental p-V diagrams of the pneumatic motor made it possible to obtain the magnitude of heat losses in the channels connecting the spool valve with the cylinder and to identify ways to reduce them. An evaluation of the relationships governing heat transfer in the pneumatic motor cylinder was carried out.* ***Originality.*** *The proposed approach enables reliable separate determination of the heat transferred from the working fluid in the pneumatic motor cylinder and in its channels.* ***Practical value.*** *The obtained results make it possible to identify ways to improve the pneumatic motor design. To reduce heat losses, the channel length should be minimized, and when using a valve-type air distribution system, the channels should be eliminated altogether.*

Keywords: *convective heat transfer, heat transfer coefficient, pneumatic motor, heat losses, compressed air, Joule's law, thermal balance.*

Introduction

The object of the study is convective heat transfer from the working fluid (compressed air) of an automotive pneumatic motor of the KhNAHU design to the cylinder walls. A distinctive feature of this pneumatic motor design is that the spool valve is installed in the V-angle of the V-type configuration, and the spool rotor is driven by a chain transmission.

The stationary part of the spool valve is connected to the cylinders by fixed channels (tubes). This design feature must be taken into account when determining the parameters of the pneumatic motor workflow process (p, V, T, M). In the present study, experimental p-V diagrams recorded at various operating modes of the KhNAHU-designed pneumatic motor were used [1].

Analysis of publications

When studying convective heat transfer from the working fluid in engines cylinder, the author [2] recommends using the Annand correlation in the following form, W/(m^2·K)

$$h = C_1 \cdot p^{0,8} \cdot T_g^{-0,53} \cdot v^{0,8} + C_2 \cdot \frac{\dot{m}_f \cdot \Delta H_c}{A \cdot (T_g - T_w)}, \quad (1)$$

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where C_1 = 0,035; C_2 = 0,2; p – the working fluid pressure; T_g – the working fluid temperature; m_f – the mass of burned fuel; ΔH_c – the lower heating value of the fuel; A – the surface area of the combustion chamber; T_w – the temperature of the combustion chamber walls.

The vast majority of researchers recommend the H. Woschni α-correlation for calculating heat transfer from the working fluid to the cylinder walls.

Thus, in [3] it is recommended to calculate the heat transfer coefficient, W/(m²·K)

$$\alpha = 128 \cdot D^{-0,2} \cdot (10 \cdot p)^{0,8} \cdot T^{-0,53} \times$$
$$\times [C_1 \cdot C_m + C_2 \cdot \frac{T}{p \cdot V} \cdot V_h \cdot [p - p_m]^{0,8}, \quad (2)$$

where D – the cylinder diameter, m; p – the working fluid pressure, MPa; T – the working fluid temperature, K; V – the cylinder volume, m³; V_h – the cylinder displacement (swept volume), m³; C_m – the mean piston speed, m/s; p_m – the cylinder pressure during motoring mode (engine cranking without fuel supply), MPa.

During the gas exchange process

$$C_1 \cdot 6{,}18 + 0{,}417 \cdot C_T / C_m, \quad (3)$$

during the compression and expansion strokes

$$C_1 \cdot 2{,}28 + 0{,}308 \cdot C_T / C_m, \quad (4)$$

where C_T – the tangential component of the working fluid velocity in the cylinder's clearance volume above the piston, m/s.

The authors of [1], for calculating the working fluid parameters of a pneumatic motor using the Newton–Richmann law, propose the H. Woschni formula to determine the surface-averaged heat transfer coefficient, α_i, W/(m²·K)

$$\alpha_i = C_0 \cdot D^{-0,2} \cdot p_i^{0,8} \cdot T_i^{-0,53} \cdot (C_1 \cdot C_m)^{0,8}, \quad (5)$$

where C_0, C_1 – proportionality coefficients (C_0 = 100…128); C_1 = 2,28 – during compression and expansion strokes; C_1 = 6,18 – during intake and exhaust strokes; D – the cylinder bore, m; p_i – the working fluid pressure, MPa; T_i – the working fluid temperature, K.

The search for a high-efficiency automotive powertrain has led to the development of a hybrid pneumatic motor [4].

When studying convective heat transfer in the hybrid pneumatic engine, the authors also recommend using the H. Woschni formula in the following form.

$$h = C \cdot d^{m-1} \cdot p^m \cdot w^m \cdot T^{0,75-1,62 \cdot m}, \quad (6)$$

where h – the heat transfer coefficient, W/(m²·K); C – the empirical coefficient, determined experimentally; d – the cylinder diameter, m; p – the working fluid pressure, MPa; V – the cylinder volume, m³; T – the working fluid temperature, K;

$$w = C_1 \cdot \bar{V} + C_2 \cdot \frac{V_d \cdot T_g}{p_g \cdot V_g} \cdot (p - p_m), \quad (7)$$

where w – the cylinder gas velocity; C_1 – the tunable coefficient; C_1 – the Woschni coefficient; V_d – the displaced volume; p – the instantaneous cylinder pressure; p_m – the motored cylinder pressure at same Crank Angle as p; p_g, V_g, T_g – respectively the pressure, volume and temperature at a reference point (i.e., intake valve closing).

The calculation of the heat transferred to the cylinder walls according to the Newton-Richmann law was verified by the amount of heat going into the cylinder walls according to the first law of thermodynamics for an open system (Joule's law)

$$dU = \delta W + \delta Q + dH, \quad (8)$$

where $dU = Mi \cdot C_v \cdot \Delta T_i$ – change in the internal energy of the working fluid, J; $\delta W = p_i \cdot 10^6$ V_i – the infinitesimal work done, J; dQ – the infinitesimal heat transfer, J; $dH = \Delta M_i \cdot C_p \cdot T_{mean}$ – change in enthalpy, J.

For simplification of calculations, the authors recommend a correlation for calculating the heat, derived from the following relationship (9)

$$\frac{\delta Q}{dt} = \frac{1}{\gamma - 1} \times$$
$$\times [\gamma \cdot p \cdot \frac{dV}{dt} + V \cdot \frac{dp}{dt}] - C_p \cdot T_i \cdot \frac{dm_i}{dt}, \quad (9)$$

where γ – the adiabatic index.

The object of study in [5] is hybrid pneumatic engines with regenerative braking and compressed-air starting.

For modeling the urban driving cycle, calculations of the workflow process were performed. The cylinder pressure of the engine was determined using the following formula

$$\frac{dp}{d\varphi}=\frac{k-1}{V}\cdot\left(\frac{-k}{k-1}\cdot p\cdot\frac{dV}{d\varphi}+\frac{dQ_w}{d\varphi}+\right.$$
$$\left.+h_i\cdot\frac{dm_i}{d\varphi}-h_e\cdot\frac{dm_e}{d\varphi}\right), \quad (10)$$

where p – cylinder pressure, MPa; V – the cylinder volume, m^3; Q_w – the heat transferred to the cylinder walls, J; m_i, h_i – the mass and enthalpy of the gas entering the cylinder, kg, J; m_e, h_e – the mass and enthalpy of the gas leaving the cylinder, kg, J; k – the adiabatic index.

For calculating the heat transfer to the cylinder walls according to the Newton-Richmann law, the heat transfer coefficient was determined using the H. Woschni correlation

$$h_x=0{,}1129\cdot p^{0{,}8}\cdot\overline{u}^{0{,}8}\cdot D^{-0{,}2}\cdot T^{-0{,}594}, \quad (11)$$

where $\overline{u}$ – the effective gas velocity (in this study, taken as the mean piston speed C_m), m/s; D – the cylinder bore, m; T – the gas temperature, K.

The authors [6], when modeling the thermal state of a cylinder with a finned outer surface of a pneumatic motor, arbitrarily assigned an average heat transfer coefficient of 25 W/(m²·K), claiming that the amount of heat transferred to the cylinder walls is very small.

Based on the conducted literature analysis, the aim and objectives of the study can be formulated.

Purpose and Objectives of the Study

The purpose of the study is to improve the calculation method by taking into account the heat transfer in the channels connecting the spool valve with the cylinder of a pneumatic motor, and based on this improvement, to perform calculations of heat transfer from the compressed air to the cylinder walls.

Research objectives:

1) Based on experimental p-V diagrams, determine the parameters of the working fluid in the pneumatic motor under different operating modes (p_i, V_i, M_i, T_i);

2) Investigate heat transfer from the compressed air in the channels connecting the pneumatic motor's spool valve to the cylinder;

3) Conduct a computational study of heat transfer from the compressed air to the cylinder walls of the pneumatic motor using the correlations of V. Annand and H. Woschni;

4) Analyze the results of the computational heat transfer study based on Joule's law

Investigation of flow and heat transfer processes of the working fluid in the channels

A distinctive feature of the KhNAHU pneumatic motor design is that the spool valve (6) is connected to the cylinder (2) via a channel (4) (Fig. 1).

Considering the location of the sensor (3) for measuring the cylinder pressure, it became necessary to investigate the air flow processes in the channels.

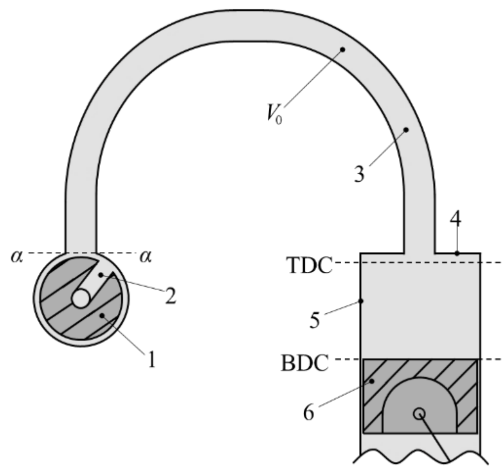

Fig. 1. Diagram of the connection between the pneumatic motor cylinder and the spool valve mechanism: 1 – piston; 2 – cylinder; 3 – location for installing the pressure sensor; 4 – channel connecting the cylinder to the spool valve mechanism; 5 – spool valve intake/exhaust port; 6 – spool valve mechanism.

For forced air flow in the channel, the Reynolds number is

$$\mathrm{Re}=\overline{W}\cdot d/\nu, \quad (12)$$

where $\overline{W}$ – the mean air velocity, m/s; d – the channel diameter, m; ν – the kinematic viscosity of air, m^2/s.

For turbulent flow in a smooth channel, the following formula can be used [7]

$$Nu = 0{,}022 \cdot \mathrm{Re}^{0{,}8} \cdot \mathrm{Pr}^{0{,}43} \cdot \varepsilon_l, \qquad (13)$$

where Nu = ($\alpha_c \cdot d/\lambda$) – the Nusselt number; α_c – the heat transfer coefficient, W/(m^2·K); λ – the thermal conductivity of the air boundary layer W/(m·K); Pr = ν/a – the Prandtl number (Pr = 0,703); a – thermal diffusivity of air, m^2/s; ε_l – the coefficient accounting for the variation of heat transfer along the channel length (l/d = 40, ε_l = 1,02).

By calculating the Nusselt number using formula (14), the heat transfer coefficient in the channel can be determined

$$\alpha_c = Nu \cdot \lambda / d, \qquad (14)$$

Knowing the heat transfer coefficient in the channel, the temperature drop along its length can be determined [7, 8]

$$T_b - T_s = (T_{in} - T_s) \cdot \exp(-\beta x), \qquad (15)$$

where T_h – the fluid temperature at a certain distance from the channel inlet, K; T_s – the constant wall temperature of the channel, K; T_{in} – the fluid temperature at the channel inlet, K; x – the coordinate along the channel from the inlet, m; $\beta = h \cdot P/\dot{m} \cdot C_p$; h – the heat transfer coefficient between the wall and the fluid, W/(m^2·K); P – the perimeter of the channel in contact with the fluid, m; $\dot{m}$ – the mass flow rate of the working fluid, kg/s; C_p – the specific heat at constant pressure, J/(kg·K).

To determine the pressure loss along the length of the channel under turbulent flow, Darcy's law can be used [7]. According to this law, the pressure drop along the channel length is given by

$$\Delta p = \xi \cdot \frac{l}{d} \cdot \frac{\rho \cdot \overline{W}^2}{2}, \qquad (16)$$

where ξ – the hydraulic friction factor; ρ – the air density, kg/m^3.

To determine ξ in equation (16), the following correlation is used [7]

$$\xi = 0{,}184 \cdot \mathrm{Re}^{-0{,}2}. \qquad (17)$$

Air pressure at the cylinder inlet

$$p_{in} = p_s - \Delta p. \qquad (18)$$

At the maximum air velocity $\overline{W}$ = 309,33 m/s, the Reynolds number is Re = 412 450,2, and the heat transfer coefficient is α_c = 775,8 W/(m²·K). Considering the large surface area of the channel, heat losses reach 201…236 J at a mode of n = 206 min^{-1}.

Determination of working fluid parameters in the pneumatic motor cylinder

The p-V diagrams were processed presented in Table 1 [1].

Table 1. Operating modes of the pneumatic motor

Mode	Rotational speed n, min^{-1}	Intake pressure p_s, MPa	Exhaust pressure p_T, MPa
A1	206	0.7	0.1
A2	206	0.9	0.1
A3	1009	0.7	0.12
A4	1009	0.9	0.12

To determine the workflow process parameters, the formula for calculating the change in working fluid pressure in the cylinder during the pneumatic motor's operating cycle was used [1], Pa

$$\Delta p_i = \frac{k_s \cdot p_i}{V_i} \cdot \left[\frac{1}{\rho_i} \cdot \left(\Delta M_{si} \cdot \frac{T_s}{T_i} - \Delta M_{ti} \right) + \right. \\ \left. + \frac{k_s - 1}{k_s} \cdot \frac{\Delta Q_i}{p_i} - \Delta V_i \right], \qquad (19)$$

where p_i, V_i, M_i, T_i – pressure, volume, density, and temperature of the working fluid in the cylinder at the beginning of the calculation step, Pa, m³, kg/m³, K; k_s – the adiabatic index for the filling and expansion processes of the working fluid – wet air with 100% relative humidity; ΔM_{si} – the mass of working fluid entering to the cylinder during the calculation step, kg; ΔM_{ti}– the mass of working fluid leaving the cylinder and entering to the exhaust channel during the calculation step, kg; ρ_i – the density of the working fluid at the control section of the spool valve system during filling and compression or exhaust and compression, kg/m³; ΔQ_i – the heat added to or removed from the working fluid during the calculation step, J; ΔV_i – the change in the cylinder clearance volume during the calculation step, m³.

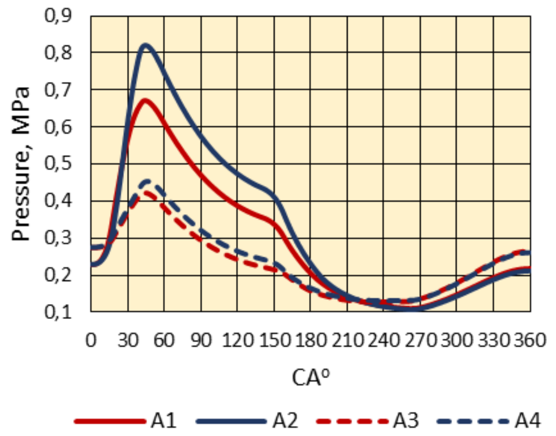

Fig. 2. Working fluid pressure in the pneumatic motor cylinder for operating modes A1, A2, A3, and A4

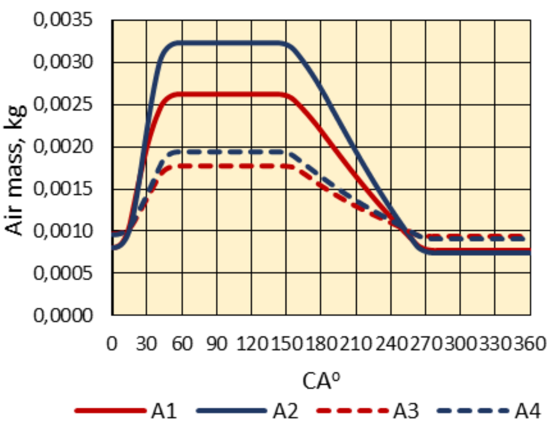

Fig. 3. Mass of the working fluid in the pneumatic motor cylinder for operating modes A1, A2, A3, and A4

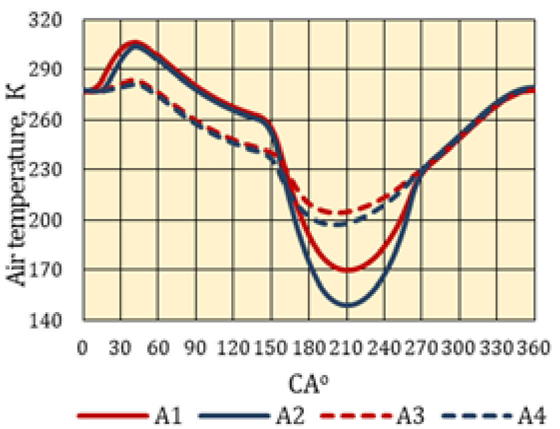

Fig. 4. Temperature of the working fluid in the pneumatic motor cylinder for operating modes A1, A2, A3, and A4

Figures 2, 3, and 4 show the curves of the pneumatic motor working fluid parameters as a function of crankshaft rotation angle ($\Delta\varphi$ = 1° CA). These curves served as the basis for calculating heat transfer from the working fluid in the channel and the cylinder of the pneumatic motor.

Calculation of heat transfer in the pneumatic motor cylinder

The amount of heat transferred to the cylinder walls during the operation of the pneumatic motor was calculated using the Newton-Richmann formula, with the convective heat transfer coefficient determined based on the α-correlations of V. Annand and H. Woschni (per 1° of crankshaft rotation):

– V. Annand α-correlation for the convective component of heat transfer coefficient [7, 8]

$$\alpha_A = 0,26 \cdot \lambda \cdot \frac{\rho^{0,7} C_m^{0,7}}{\eta^{0,7} \cdot D^{0,3}}, \qquad (20)$$

where $\lambda = 0,000361 \cdot T_{mean}^{0,75}$ – the thermal conductivity of the gas boundary layer at the average temperature $T_{mean} = (T_g + T_{mean})/2$, W/(m·K); $\rho = 3,49 \cdot 10^{-3} \cdot p_g/T_g$ – the density of the working fluid, kg/m³; $\eta = 0,56 \cdot 10^{-6} \cdot T_{mean}^{0,62}$ – the dynamic viscosity of the working fluid at temperature T_{mean}, Pa·s; C_m – the mean piston speed, m/s; D – the cylinder diameter, m;

– H. Woschni α-correlation according to formula (11)

$$\alpha_W = 0,1129 \cdot D^{-0,2} \cdot p_i^{0,8} \cdot T_i^{-0,594} \cdot C_m^{0,8}, \qquad (21)$$

where D – the cylinder diameter, m; p_i – the working fluid pressure in the cylinder, bar.

The results of the heat transfer calculation over the entire cycle in the pneumatic motor cylinder using the α-correlations are presented in Table 2.

Table 2. Heat transferred during one operating cycle

Total heat Q_Σ according to the α-correlation:	Mode			
	A1	A2	A3	A4
	J			
V. Annand	3.44	4.51	1.91	2.18
H. Woschni	7.17	9.68	4.55	4.24

Total Heat Losses in the Pneumatic Motor

The results of the heat transfer calculations in the channel and cylinder are shown in Fig. 5. It can be seen that the total amount of heat in the channel and cylinder of the pneumatic motor differs from the heat calculated using Joule's law for one operating cycle of the pneumatic motor.

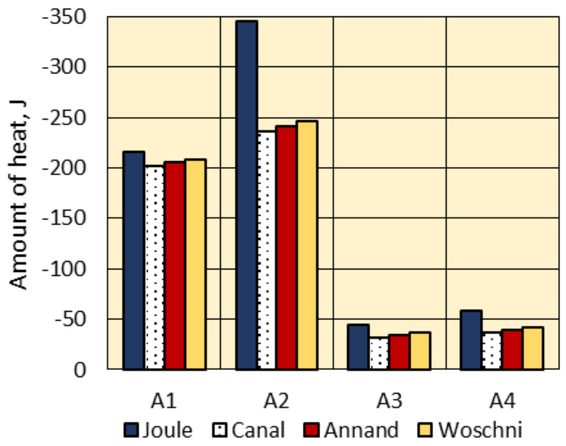

Fig. 5. Heat losses of the pneumatic motor at different operating modes

Processing of the *p-V* diagrams made it possible to determine the cyclic air consumption G_c and the indicated work per cycle L_i at different operating modes. The calculation results are presented in Table 3. Additionally, Table 3 shows: ΔQ – amount of heat according to Joule's law; l_{ad} – adiabatic work performed by the compressed working fluid used for air expansion; l_{ad}^{av} – available adiabatic expansion work.

Table 3. Calculation Results

Mode	A1	A2	A3	A4
δQ, J	215.32	345.46	44.96	58.98
G_c, kg	0.00257	0.00321	0.00177	0.00205
l_{ad}, J	125.52	137.22	125.52	137.22
l_{ad}^{av}, J	323.28	440.60	222.26	281.45
L_i, J	102.32	131.83	43.11	50.46
η_i, %	31.6	29.9	19.3	17.9
Q_w, %	66.6	78.4	20.2	20.9

From Table 3, it can be seen that the indicated efficiency η_i decreases with increasing crankshaft rotational speed.

Heat losses to the pneumatic motor walls Q_w reach up to 78% of the adiabatic work; therefore, to reduce these losses, it is advisable to shorten the channel lengths and, ideally, switch to a valve-type air distribution system.

Conclusions

The results of the computational study of heat transfer from the working fluid to the cylinder walls of a pneumatic motor with a long channel from the spool valve to the cylinder allow the following conclusions:

1. Based on experimental *p-V* diagrams, the workflow process parameters of the pneumatic motor were determined for different operating modes.
2. For the study of heat transfer from the working fluid, heat transfer was considered separately in the channel and directly in the cylinder.
3. The total amount of heat transferred from the working fluid was verified using the thermal balance according to Joule's law. The error ranged from 3% to 34%.
4. The amount of heat transferred in the channel accounts for 87–96% of the total heat losses.
5. To reduce heat losses from the working fluid to the cylinder walls, it is advisable to shorten the channels connecting the spool valve and the cylinders (ideally, eliminate the channels entirely).
6. The most promising solution is to switch to a valve-type air distribution system.

Conflict of interests

The authors declare that there is no conflict of interests regarding the publication of this paper.

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Тепловіддача від робочого тіла у пневмодвигуні із золотниковим механізмом розподілу повітря

Анотація. Проблема. *Процеси тепловіддачі від робочого тіла до стінок циліндра у теплових двигунів грають важливу роль. По-перше, температурний стан деталей двигуна, що визначає його надійність роботи, залежить від параметрів робочого тіла і від інтенсивності тепловіддачі. По-друге, тепловіддача від робочого тіла це втрата теплоти, яка завжди зменшує перетворення енергії робочого тіла в корисну роботу. Тому під час проектування двигуна, а також, під час його вдосконалення, необхідні надійні методики розрахунку тепловіддачі від робочого тіла до стінок циліндра.* ***Мета****. Метою дослідження є вдосконалення методики розрахунку тепловіддачі від робочого тіла у стінки, враховуючи особливості конструкції пневмодвигуна і визначення кількості втраченої теплоти, яку можна було б перетворити в корисну роботу.* ***Методологія****. Запропоновано окремо дослідити тепловіддачу в циліндрі пневмодвигуна і каналі, що з'єднує золотник із циліндром. Це дозволило врахувати велику втрату теплоти в каналах і підтвердити це за допомогою закону Джоуля.* ***Результати****. Оброблені експериментальні індикаторні діаграми пневмодвигуна дозволили одержати величину втрат теплоти в каналах, що з'єднують золотник із циліндром і визначити шлях до їх зменшення. Проведена оцінка залежностей, що визначають тепловіддачу в циліндрі пневмодвигуна.* ***Оригінальність****. Запропонований підхід дозволив надійно визначити окремо теплоту, що віддається від робочого тіла в циліндрі пневмодвигуна і в його впускних каналах.* ***Практичне значення****. Отримані результати дослідження дали можливість визначити шляхи удосконалення конструкції пневмодвигуна. Для зменшення втрат теплоти необхідно зменшувати довжину каналів, а при застосуванні клапанного розподілу повітря зовсім відмовитися від них.*

Ключові слова: *конвективна тепловіддача, коефіцієнт тепловіддачі, пневмодвигун, втрати теплоти, стиснене повітря, закон Джоуля, тепловий баланс*

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