COMPUTATIONAL AND EXPERIMENTAL DETERMINATION OF ENERGY LOSS OF THE OPERATING FLUID IN THE INTAKE SYSTEM OF THE AUTOMOBILE PISTON PNEUMATIC ENGINE USING THE EXERGY METHOD

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Abstract. The application exergy method of the thermodynamic analysis for determining the energy loss of the operating fluid in the intake system of the automobile piston pneumatic engine using experimental data of its engine test rigs has been considered. The analysis of the performed exergy calculational and experimental research on the evaluation of energy loss in the intake system of the engine for current modes of its operation has been presented. The sample of a piston pneumatic engine for a combined automobile power unit created at the ICE Department of KhNAYU requires further improving and developing especially such components and systems as those connected to the burn process. A compressed air intake system is one of them. Compressed air is not only some working fluid for a pneumatic engine but the energy source for its intake system. The energy loss factor like reliability and durability is referred to the performance and efficiency factors. The object of the study is a pneumatic engine created by converting a gasoline four-cylinder four-stroke ICE at the ICE Department of KhNAYU. The experimental research was carried out by laboratory engine test rigs of the pneumatic engine with speed characterization provided that two thermodynamic parameters of compressed air at the intake were kept constant for each characteristic: pressure $p_s = \text{idem}$ and temperature $T_s = 293 \text{ K}$. In each test mode all external parameters of the pneumatic engine were recorded and indicator diagrams of the first cylinder were taken. Each speed characteristic consisted of $6\text{–}8$ modes $p_s = \text{idem}$ and $T_s = 293 \text{ K} = \text{idem}$ with changing the rotational rate of the crankshaft $n$, rpm, from minimally stable (about $n=200 \text{ rpm}$) to maximum possible $n = 1000 \pm 50 \text{ rpm}$. It should be noted that there is an operating fluid pressure drop in the intake system chamber due to available inconvertibilities: throttling in the test port, hydraulic loss along the intake port as a result of fluid friction, swirls and other gas–dynamic phenomena. The exergy loss of the operating fluid enthalpy in the intake system $D_{\text{уп}} = E_{\text{уп}} - E_{\text{нап}}$ with the intake pressure $p_s = 0.5 \text{ MPa}$ is $13.3 \text{ kW}$, with the intake pressure $p_s = 0.7 \text{ MPa}$ is $15.0 \text{ kW}$, with the intake pressure $p_s = 0.9 \text{ MPa}$ is $16.9 \text{ kW}$, with the intake pressure $p_s = 1.1 \text{ MPa}$ is $19.7 \text{ kW}$.

Key words: piston pneumatic engine, intake system, energy loss, enthalpy exergy, maximum efficient power modes.

Introduction

The sample of a piston pneumatic engine for a combined automobile power unit created at the ICE Department of KhNAYU requires further improving and developing especially such components and systems as those connected to the burn process. A compressed air intake system is one of them. Compressed air is not only some working fluid for a pneumatic engine but the energy source for its intake system. The energy loss factor like reliability and durability is referred to the performance and efficiency factors.

The method of determining the energy loss of the compressed air in the intake system of the pneumatic engine using the experimental data [1] and well-known exergy method [2, 3] has been considered in the paper.

Purpose and Tasks

The purpose of this computational and experimental study is to develop a technique for applying the well-known exergy method to determine the energy loss of the operating fluid of the piston pneumatic engine in the intake system and the energy loss of operating fluid (compressed air) as its enthalpy exergy loss in the intake system of the piston pneumatic four-cylinder engine with intake operating pressures $p_s = 0.5\text{–}1.1 \text{ MPa}$ at the modes of the maximum possible efficient power $N_e^{\text{max}}$ at given pressure levels $p_s$ according to speed characteristics.
\( p_s = \text{idem with in take constant temperature} \ T_c = 293 \, \text{K}. \)

**Analysis of Publications**

Nowadays there are many publications devoted to the fundamentals of the exergy analysis and its application in various fields of technology. Last decade of the twentieth century the work on creation of a mathematical apparatus of exergy analysis in solving optimization problems in the field of improving equipment and designing new equipment and technologies was completed. The fundamentals of exergy analysis are mostly generalized and fully described in the works of V. M. Brodyansky [2], G. N. Kostenko [3], A. R. Kotin [4], G. D. Rem [5]. The issues of the exergy method application in the field of heat engineering are studied by A. I. Andryushchenko [6 and others].

Almost all the studies on the exergy method published since 1964 are referred to the exergy applications in heat engineering, chemical technology, metallurgy, cryogenic engineering and economics. The discussions about the necessity of this section in thermodynamics and correctness of its main principles were left in past. In the 21\textsuperscript{st} century the exergy method got known to many specialists of various fields of technology. They considered it as a convenient tool for solving their specific tasks. The main advantages of the exergy method of thermodynamic analysis are its simplicity and universality [2]. It is possible to determine both absolute and relative values of energy loss due to inconvertibilities for any process or its part using the exergy balance.

So in the recently published work [7] the simplicity and accuracy of determining the energy loss of the operating fluid in case of its inconvertible leakages using the exergy method were presented. This paper shows determining the dependence of the exergy characteristic – enthalpy exergy loss due to external conditions: pressure and temperature of the supplied compressed air at the intake of the power unit \( p_s \) and \( T_c \) were found using experimental data and theoretical dependences between the burn process parameters.

**Object of Study**

The object of the study is a piston pneumatic engine with specifications presented in Table 1. The considered pneumatic engine was created at the ICE Department of KhN\textsc{A}HU by converting a gasoline four-cylinder four-stroke ICE. The experience of this conversion is described in detail in [1].

In accordance with the above formulated purpose the specific object of the study in the experiment and calculations is the intake system of the pneumatic engine prototype sample. Each cylinder of the engine has the same intake system through which the process of filling the cylinders with the incoming charge of compressed air is carried out.

**Table 1 – Specifications of a piston pneumatic engine with spool-type air distribution – prototype sample designed and created at the ICE Department of KhN\textsc{A}HU in 2008**

| Parameter                  | Notation | Dimensions | Numeric value |
|----------------------------|----------|------------|---------------|
| Cylinder diameter          | \( D \)  | mm         | 76            |
| Piston stroke              | \( S \)  | mm         | 66            |
| Number of cylinders        | \( Z \)  |            | 4             |
| Included angle of cylinder block | \( \gamma \) | degree     | 90            |
| Cylinder displacement      | \( V_h \) | dm\(^3\)   | 0.2994        |
| Volume of clearance space  | \( V_0 \) | dm\(^3\)   | 0.280         |
| Relative clearance volume  | \( E_0 = V_d/V_h \) |            | 0.935         |
| Filling volume             | \( V_1 \) | dm\(^3\)   | 0.0862        |
| Filling rate \( E_1 = V_d/V_h \) |            | -           | 0.288         |
| Volume of reverse compression start | \( V_s \) | dm\(^3\)   | 0.147         |
| Degree of reverse compression \( E_3 = V_d/V_h \) |            | -           | 0.491         |
| Fullflow area at intake line | \( f_{v_1}^{\max} \) | m\(^2\) \( \cdot \) 10\(^{-4} \) | 3.14          |
| Fullflow area at exhaust line | \( f_{v_2}^{\max} \) | m\(^2\) \( \cdot \) 10\(^{-4} \) | 3.14          |
| Adiabatic exponent of dry air | \( k \)  |            | 1.40          |
| Adiabatic exponent of humid air, relative humidity \( q = 100 \% \) [9] | \( k_p \) |            | 1.32          |

This process is not adjustable. It is performed by a structurally specified period of filling the cylinder with the incoming charge of compressed air. The process of filling lasts up to 58 degrees of crankshaft turn: three degrees before-top dead centre (TDC) and 55 degrees after TDC.

The cylinder in take system includes a take port of circular cross-section with a diameter of 20 mm and a length of 800 mm, two
test ports of the same cross-section: the first is on the outer cylindrical rotor surface of the spool-type air distributor and the second is on the inner cylindrical stator surface at the beginning of the intake port. Test ports are partially or fully matched during the filling. Their combined flow area varies from zero to a maximum that is equal to the area of an open port.

The volume of the internal chamber of the intake system along with the object above the piston space at the top dead centre is so-called dead volume that increases the flow density of compressed air \( g_a \), kg / (kW/h).

### Experimental Techniques

The experimental study was carried out by laboratory engine test rigs of the pneumatic engine with speed characterization provided that two thermodynamic parameters of compressed air at the intake were kept constant for each characteristic: pressure \( p_i \) = idem and temperature \( T_i = 293 \text{ K} \). In each test mode all external parameters of the pneumatic engine were recorded and indicator diagrams of the first cylinder were taken. Each speed characteristic consisted of 6–8 modes \( p_i = \) idem and \( T_i = 293 \text{ K} \) = idem with changing the rotational rate of the crankshaft \( n \), rpm, from minimally stable (about \( n = 200 \text{ rpm} \) ) to maximum possible \( n = 1000 \pm 50 \text{ rpm} \).

The method of pneumatic engine testing to take the speed characteristics of the pneumatic engine is described in detail in [1]. The whole experiment comprised four speed characteristics for intake pressures of the compressed air (being of exceptional interest). \( p_i = 0.5; 0.7; 0.9 \text{ and } 1.1 \text{ MPa} \), at constant temperature \( T_i = 293 \text{ K} \).

Then using the obtained experimental data we constructed the dependences graphs of the efficient power \( N_e \), kW, on the rotational speed of the shaf, rpm and determined the maxima of the functions \( N_e (n) \) – the modes of the indicated power for the corresponding pressure level \( p_i \) at the temperature \( T_i \), corresponding to the ambient temperature \( T_{\text{ambient}} = 293 \text{ K} \).

The operating fluid energy loss in the intake system of the pneumatic engine was determined only for these four modes \( N_{e \text{ max}} \) mentioned above (being of exceptional interest).

### Results of the Computational and Experimental Study and Their Analysis Using the Exergy Method

The main results of this experimental and computational study are presented in Table 2 and fig. 1. They are the results of the second stage of the study, the starting points for which are the results of the first stage devoted to the determining the operating fluid energy loss at the very beginning of the working chamber of the pneumatic engine– in the spool-type air distributor due to leakage of incoming compressed air \( D_{\text{leak}} \), kW [7].

Thus the starting point for the second stage of the study is the level of exergy of the operating fluid enthalpy at the beginning of the intake process equal to the enthalpy exergy of \( E_{\text{en}} \), kW, supplied to the engine, minus the energy loss for leakage.

\[
E_{\text{en}} = E_a - D_{\text{leak}}, \text{ kW}.
\] (1)

After that we defined the level of exergy of the operating fluid enthalpy at the end of the intake, i.e. at the end of the filling process, \( E_{\text{en}} \), kW, that was dependant on the thermodynamic parameters of the air condition at that moment, primarily on its pressure \( p_{\text{ambient}} \) and temperature \( T_{\text{ambient}} \).

It should be noted that there is an operating fluid pressure drop in the intake system chamber due to available inconvertibilities: throttling in the test port, hydraulic loss along the intake port as a result of fluid friction, swirls and other gas-dynamic phenomena.

The analysis of the obtained indicator diagrams during the engine test rigs allowed to determine the cylinder pressure values while filling. Table 2 shows the pressure values in the working chamber at various levels of compressed air intake pressure \( p_i \) for maximum power modes according to speed characteristics. The thermodynamic parameters (enthalpy and entropy) were determined according to the level of air pressure and temperature using tables [8]. The performance was calculated as the specific enthalpy exergy \( E_{\text{ex}} \), kJ/kg, and the mass for the gas flow \( E_{\text{man}} \), kW.

It should be noted that changing temperature of the operating fluid while filling provided that the compressed air is supplied into the engine at ambient temperature \( T_i = T_{\text{ambient}} = 293 \text{ K} \) may be neglected the basis of little heat exchange with the environment due to the absence of temperature difference and extremely short filling process (milliseconds), absence of expansion and contraction processes, and present throttling process occurred with a constant gas enthalpy can cause as known [10] a Joule-Thompson effect not exceeding the temperature drop by more than 1 K, i. e., within the accuracy of temperature measuring.
All this allows us to calculate the exergy of the operating fluid enthalpy at the end of the filling process according to its pressure $p_{нмт}$, MPa and temperature $T_{нмт} = T_{о.с.} = 293$ K taking into account known dependences [2 и 5]:

- specific exergy

$$e_{нмт} = (h_{нмт} - h_{о.с.}) - T_{о.с.}(s_{нмт} - s_{о.с.}), \text{kJ/kg} \quad (2)$$

- air flow at the end of the filling process

$$E_{нмт} = G_{номт}e_{нмт}, \text{kW}, \quad (3)$$

where $G_{номт} = G - G_{нмт}$, kg/s – mass second compressed airflowefficiently used; $G$, kg/s – compressed air flow supplied into the pneumatic engine; $G_{нмт}$, kg/s – experimentally found mass flow of compressed air for leakage in the given pneumatic engine, depending on its parameters at the intake; $h_{нмт}$, $h_{о.с.}$, kJ/kg – air specific enthalpy respectively for the given pressure $p_{нмт}$ and temperature $T_{нмт}$, and for ambient parameters $p_{о.с.}$ and $T_{о.с.}$, $K$ – ambient temperature; $S_{нмт}$, $S_{о.с.}$; kJ/(kg K) – air specific entropy respectively in terms of $p_{нмт}$, $T_{нмт}$ and in terms of $p_{о.с.}$ and $T_{о.с.}$

The exergy loss of the operating fluid enthalpy in the intake system is equal to the difference in its exergy levels at the beginning and at the end of the process

$$D_{нмт} = E_{нмт} - E_{нмт}, \text{kW}. \quad (4)$$

Table 2 – Experimental data on the heat loss of the operating fluid in the intake system of the piston pneumatic engine during its operation according to speed characteristics $p_с$ = idem and constant intake temperatature of compressed air $T_4 = 293$ K at maximum power modes $N_с^{max}$, according to [1]

| Name of Value | Notation | Dimension | Pressure $P_с$, MPa | 0.5 | 0.7 | 0.9 | 1.1 |
|---------------|----------|-----------|---------------------|-----|-----|-----|-----|
| Supplied (spent) amount of compressed air | $G$ | kg/s | 0.0339 | 0.0656 | 0.1000 | 0.1583 |
| Supplied (spent) thermal energy of the operating fluid in the form of its enthalpy $H_i = G - h_i$ | $H_i$ | kW | 9.91 | 19.17 | 29.17 | 46.08 |
| | | | % | 100 | 100 | 100 | 100 |
| Supplied (spent) operative heat energy – enthalpy energy | $E_i$ | kW | 4.59 | 10.69 | 18.77 | 31.72 |
| | | | % of $E_i$ | 100 | 100 | 100 | 100 |
| The content of non-operative heat - enthalpy energy in the supplied heat (in the compressed air enthalpy) | $B_i$ | kW | 5.32 | 8.48 | 10.36 | 14.36 |
| | | | % of $H_i$ | 53.7 | 44.2 | 35.5 | 31.2 |
| Efficient consumption of compressed air | $G_{номт}$ | kg/s | 0.0324 | 0.0577 | 0.0860 | 0.1396 |
| Specific enthalpy of compressed air under the conditions at the intake of the pneumatic engine | $h_{о.с.}$ | kJ/kg | 292.3 | 292.4 | 291.3 | 290.0 |
| The specific entropy of compressed air under the conditions at the intake of the pneumatic engine | $S_{о.с.}$ | kJ/kg·K | 6.382 | 6.288 | 6.213 | 6.156 |
| Air specific enthalpy according to environmental conditions | $h_{о.с.}$ | kJ/kg | 293.2 | 293.2 | 293.2 | 293.2 |
| Air specific entropy according to environmental conditions | $S_{о.с.}$ | kJ/kg·K | 6.847 | 6.847 | 6.847 | 6.847 |
| Specific exergy of air enthalpy under the terms of pneumatic engine intake | $e_{сs}$ | kJ/kg | 135.4 | 163.0 | 187.7 | 200.3 |
| The pressure in the working chamber at the end of the filling process (intake) | $P_{нм}$ | MPa | 0.395 | 0.495 | 0.580 | 0.640 |
| The specific exergy of the operating fluid enthalpy at the end of filling | $e_{нм}$ | kJ/kg | 116.5 | 135.3 | 150.7 | 155.6 |
| The exergy of the operating fluid enthalpy at the beginning of the intake $E_{нм} = E_i - D_{нмт}$, kW | $E_{нм}$ | kW | 4.38 | 9.40 | 16.14 | 27.98 |
| | | | % of $E_i$ | 95.4 | 87.9 | 86.0 | 88.2 |
| The exergy of the operating fluid enthalpy at the end of the filling process | $e_{нм}$ | kJ/kg | 116.5 | 135.3 | 150.7 | 155.6 |
| The exergy of the operating fluid enthalpy at the end of filling $E_{нм} = G_{номт}e_{нмт}$, kW | $E_{нм}$ | kW | 3.77 | 7.80 | 21.95 | 21.72 |
| | | | % of $E_i$ | 82.1 | 73.0 | 69.0 | 68.2 |
| Exergy loss of the operating fluid enthalpy in the intake system $D_{нм} = E_{нм} - E_{нмт}$, kW | $D_{нм}$ | kW | 13.3 | 15.0 | 16.9 | 19.7 |
| | | | % of $E_i$ | 116.5 | 135.3 | 150.7 | 155.6 |
The practical and scientific significance of this work is the following: for a particular type of piston automobile pneumatic engine with a spool-type air distribution the level and patterns of changes in the energy loss of the operating fluid in the form of the enthalpy exergy loss in the intake system depending on the compressed air supplied pressure at the intake in the most important operating range of $p_s = 0.5 - 1.1$ MPa at constant air temperature equal to ambient temperature $T_s = T_{a,c} = 293$ K have been determined.

The level of energy loss of the operating fluid (compressed air) in the form of its enthalpy exergy loss in the intake system of a four-cylinder piston pneumatic engine while its operating with various intake pressures are: with intake pressure $p_s = 0.5$ MPa is $13.3$ kW ($116.5$ % of $E_x$), with intake pressure $p_s = 0.7$ MPa is $15.0$ kW ($135.3$ % of $E_x$), with intake pressure $p_s = 0.9$ MPa is $16.9$ kW ($150.7$ % of $E_x$), with intake pressure $p_s = 1.1$ MPa is $19.7$ kW ($155.6$ % of $E_x$).

References
1. Vorkonov A.I., Nikitchenko I.N. Rabochiy process automobilechnogo pnevmomotora: monografiya. – X.: Hnadu, 2015. – 200 c.
2. Brodnyak B.M., Fyattser B., Mihalek K., Brodnyak B.M. Eksergeticheskiy metod i ego prilozheniya / Pod. red. B.M.Brodnyakogo. – M.: Energoatomizdat, 1988. – 288 c.
3. Kostenko G.N. Eksergeticheskiy analiz teplonyykh prossekov i ustanovok: konспект lekcii. – Odessa (OPI), 1964.
4. Kotin A.O. O fizicheskie svoystva poniatija ekstuz // Izvestiya vuzov, Energetika, 1965. № 7. C. 49-57.
5. Ee G.D. Tekhnicheskaya thermodynamika: teoreticheskie osnovy i tehnicheskie proizvedeniya / Pod red. B.M. Brodnyakogo i G.N. Kostenko. – M.: Mir, 1977. – 519 c.
6. Andriyenko A.I. Voprosy thermodynamicheskoj teorii teplofionskix ustanovok / A.I. Andriyenko. – M.: MGI, 1955.
7. Vorkonov A.I. Opredelenie potery ekstuzi otpryaka robachego tela v zolotnikovom vozduhospredelitеле automobilechnogo portshchennogo pnevmomotora.
8. Varagbnik N.B. Spravochnik po teplofizicheskim svoystvam gazov i iizdokov. – M.: Energoizdat, 1972. – 636 c.
9. Borisenko K.S. Pnevmaticheskie dvigateli gornych maschin. – M.: Ugletekhizdat, 1958. – 205 c.
10. Alabovski A.N., Nedujh Ii. Tekhnicheskaya thermodynamika i teploperehod. – K.: Vyscha shk. 1990. – 255 c.
References
1. Voronkov, A.I., Nikitchenko, I.N. (2015). Rabochnyi process avtomobil'nogo pnevmovdis- gatelya [The burn process of the automobile pneumatic engine]: monografiya. Kh.: KhNADU, 200 [in Russian].
2. Brodyanskii V.M., Fratsher V., Mikhail K. (1988). Eksergeticheskiy metod i ego prilozhe- niya [Exergy method and its applications] Pod. red. V.M. Brodyanskogo. Moscow: Energo- atomizdat, 288 [in Russian].
3. Kostenko G.N. (1964). Eksergeticheskiy analiz teplovыхk hyp processov i ustanovok [Exergy analysis of thermal processes and units], Konspekt lektsiy. Odessa (OPi) [in Russian].
4. Kotin A.O. (1965). O fizicheskykh sushchnostях pon-yatiya eksergiy [On the physical essence of the exergy concept], Izvestiya vuzov, Energetika, 7, 49-57 [in Russian].
5. Ber G.D. (1977). Tekhnicheskaya termodina- mika: teoreticheskie osnovy i tekhnicheskii pri- lozheniya [Technical thermodynamics: theoretical fundamentals and technical applications]. Pod red. V.M. Brodyanskogo i G.N. Kostenko. Moscow: Mir. 519 [in Russian].
6. Andryushchenko A.I. (1955). Voprosy termo- dinamicheskoy teorii teplofikatsionnykh ustanovok [Issues of thermodynamic theory of heat installations], Moscow: MEI: 57 [in Russian].
7. Voronkov A.I. Opredelenie poteri eksergii ental- pii rabochnogo tela v zolotnikovom vozdukhos- raspredelitеле AVTOMOBIL'NOGO PORSHNEVogo pnevmovdisgatelya [Determination of exergy loss of the operating fluid enthalpy in the spool-type air distributor of the automobile piston pneumatic engine].
8. Vargaflik N.B. (1972). Spravochnik po teplo- fizicheskam svoystvam gazov i zhidkostey [Handbook of thermophysical properties of gases and liquids]. Moscow: Energoizdat, 636 [in Russian].
9. Borisenko K.S. (1958). Pnevmaticheskie dvigateili gornykh meshin [Pneumatic engines of mining machines]. Moscow: Ugletekhizdat, 205 [in Russian].
10. Alabovskiy A.N., Neduzhiy I. (1990). Tekhnich- eskaya termo﹋namika i teploperedacha [Technical thermodynamics and heat transfer]. Kijv: Vishcha shk., 255 [in Russian].

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Розрауховно-експериментальне визначення втрат енергії робочого тіла в системі впуску автомобільного поршневого пневмодвигуна з використанням ексергетичного методу

Анотація. Розглянуто методику додатка ексергетичної теорії до визначення втрат енергії робочого тіла в системі впуску автомобільного поршневого пневмодвигуна з використанням експериментальних даних його стендових моторних випробувань. Викладено аналіз використання ексергетичного розраухов- во-експериментального дослідження з оцінки ве- личин втрат енергії у системі впуску двигуна для наступних режимів його роботи.

Створений на кафедрі ДВЗ ХНАДУ для комбінованої автомобільної силової установки з розпізнаванням поршневого пневмодвигуна має потребу в удосконаленні й довірчості особливо вузлів і систем, пов'язаних з дійсними режимами роботи двигуна. Однією з таких систем є система впуску, яка має значну ексергетичну втрату енергії робочого тіла в системі впуску двигуна.

Об'єктом дослідження є розраухунок пневмо- двигуна, створений на кафедрі ДВЗ ХНАДУ шля- хом конвертації бензинового чотирициліндрового чотиритактного ДВЗ. Експериментальнє дослідження проводилося шляхом лабораторних стендових моторних випробувань пневмодвигуна зі зважуванням швидкості роботи двигуна. Основними характеристиками двигуна є: котел, поверхня зміни робочого тіла, накинутість та втрати енергії робочого тіла. На кожному режимі випробувань залежно від температур робочого тіла, накинутість та втрати енергії робочого тіла.
След зазначити, що в порожній системи впушки відбувається падіння тиску робочого тіла внаслідок незворотних процесів: дроселювання в контрольному отворі, гідравлічних втрат по довжині впуского каналу внаслідок тертя газу об стінки, завихрень й інших газодинамічних явищ. Втрати ексергії ентальпії робочого тіла в системі впушки $D_{nw} = E_{nw} - E_{nsw}$ становлять для тиску на впуску $p_s = 0,5$ МПа $= 13,3$ кВт, для тиску на впуску $p_s = 0,7$ МПа $= 15,0$ кВт для тиску на впуску $p_s = 0,9$ МПа $= 16,9$ кВт, для тиску на впуску $p_s = 1,1$ МПа $= 19,7$ кВт.

**Ключові слова:** поршневий пневмодвигун, система впушки, енергетичні втрати, ексергія ентальпії, режими максимальної ефективної потужності.

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Расчетно-экспериментальное определение потерь энергии рабочего тела в системе впушки автомобильного поршневого пневмодвигателя с использованием эксергетического метода

Аннотация. Рассмотрена методика приложения эксергетического метода термодинамического анализа для определения потерь энергии рабочего тела в системе впушки автомобильного поршневого пневмодвигателя с использованием экспериментальных данных его стендовых моторных испытаний. Изложен анализ выполненного эксергетического расчетно-экспериментального исследования по оценке величин энергопотерь в системе впушки двигателя для актуальных режимов его работы.

Созданный на кафедре ДВС ХНАДУ для комбинированной автомобильной силовой установки образец поршневого пневмодвигателя нуждается в совершенствовании и доводке, особенно узлов и систем, связанных с осуществлением рабочего процесса. Одной из таких систем является система впушки сжатого воздуха. Для пневмодвигателя сжатый воздух является не только рабочим телом, но и энергоносителем для его системы впушки. К показателям работоспособности и эффективности, кроме надежности и долговечности относится и показатель уровня энергетических потерь.

Объектом исследования рассматриваемый пневмодвигатель, созданный на кафедре ДВС ХНАДУ путем конвертации бензинового четырехцилиндрового четырехтактного ДВС.

Экспериментальное исследование проводилось путем лабораторных стендовых моторных испытаний пневмодвигателя со снятием скоростных характеристик при условии поддержания неизменных для каждой характеристики двух термодинамических параметров сжатого воздуха на входе: давления $p_{in}$ idem и температуры $T_{s} = 293$ К. На каждом режиме испытаний регистрировались все внешние параметры работы пневмодвигателя и снимались индикаторные диаграммы первого цилиндра. Каждая скоростная характеристика состояла из шести-восьми режимов $p_{in}$ idem и $T_{s} = 293$ K = idem с изменением частоты вращения коленчатого вала $n$, об/мин, от минимально устойчивых (около $n=200$ об/мин) до максимально возможных $n = 1000 ± 50$ об/мин.

Следует отметить, что в полости системы впушки происходит падение давления рабочего тела вследствие имеющих место необратимостей: дросселирования в контрольном отверстии, гидравлических потерь по длине впускного канала в результате трения газа о стенки, завихрений и прочих газодинамических явлений. Потери эксергии энтальпії рабочего тела в системе впуску $D_{nw} = E_{nw} - E_{nsw}$ составляют для давления на впуске $p_s = 0,5$ МПа $= 13,3$ кВт, для давления на впуске $p_s = 0,7$ МПа $= 15,0$ кВт, для давления на впуске $p_s = 0,9$ МПа $= 16,9$ кВт, для давления на впуске $p_s = 1,1$ МПа $= 19,7$ кВт.

**Ключевые слова:** поршневой пневмодвигатель, система впушки, энергетические потери, эксергия энтальпії, режимы максимальної эффективної мощности.

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