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RESEARCH OF THE DIESEL ENGINE FAILURE DEPENDENCY ON ENGINE OPERATING CONDITIONS

ИССЛЕДОВАНИЯ ЗАВИСИМОСТИ ОТКАЗОВ ДИЗЕЛЬНЫХ ДВИГАТЕЛЕЙ ОТ УСЛОВИЙ ЭКСПЛУАТАЦИИ

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Abstract: the paper deals with the analysis of the factors and aspects covered by the Diesel Engine Reliability Theory and relevant for the discussion of engine reliability. The paper suggests and examines indicators for the assessment of engine reliability, and quantitative function for the evaluation of engine technical condition. The integrated effect of mountainous dessert on automotive engines is also examined although it has received little research attention so far, whereas findings of quantitative criteria analysis intended for reliability assessment under difficult conditions have not been publicly reported as they have been accomplished by manufacturers of military vehicles.

Engine operating conditions refer to the manner of engine use, intensity of engine operation, different types of engine loads and their sequence, as well as culture and qualification of drivers, servicemen and maintenance personnel, provision of repair and diagnostics equipment, etc.

Research of the case under consideration has been accomplished in mountainous desert of Afghanistan showing complicated climatic conditions. For the purpose of collecting precise and accurate data, vehicles used by the Provincial Reconstruction Team situated in Ghor province (Afghanistan) were selected for the research. Their operation course, failures, and repairs can be tracked and captured more precisely than those of civilians as their data are recorded in data logs.

KEYWORDS: ENGINE FAILURES, AFGHANISTAN, ENVIRONMENTAL INTERACTION, CLIMATIC FACTOR.

1. Introduction

Automotive internal combustion engines (hereinafter – ICEs) are usually designed for operation under normal conditions consequently, their adaptation for use in vehicles operated under difficult conditions faces some problems. These issues are caused by the engine operation in locations showing difficult climatic and geographic conditions: surroundings of mountains, deserts and torrential rivers, increased airborne dust levels, diurnal temperature variation, and solar radiation.

Most of the engine operation researches are carried out at the locations having a good road network while maintaining and repairing engines according to the requirements of their manufacturers.

Difficult engine operating conditions normally include low air temperatures, deep snow coverage, frequent snowstorms and snow banks on roads; also regions with hot desert climate with poorly developed network of automobile roads, lack of afforestation and water, increased airborne dust levels, considerable diurnal temperature variation; mountainous terrains with thin air and lower atmospheric pressure; highly variable terrain profiles; off-roads and high-scale road aggravation. Difficult engine operating conditions are also considered to include regions with high humidity levels.

Object of the research – diesel engines operating under difficult conditions.

Research material. The article analyzes performance of twenty nine diesel engines operating under difficult conditions: sixteen vehicles Toyota Land Cruiser 100 (LC) with diesel engines 1HD-FTE, and thirteen high-mobility multipurpose wheeled vehicles M998A2 (HMMWV) with diesel engines GM6.5L. All the vehicles were used under conditions of mountainous desert from the very beginning, i.e., starting with summer of 2005 till December 31, 2011.

The objective of the study – to determine the influence of the system “ICE-environment” on reliability of engines operating under difficult conditions.

Tasks:

- to identify the key factors determining operational reliability of ICEs operating under difficult conditions;
- to reveal adversity of weather and climate in the region under consideration for ICE operation;
- to suggest a dependency for the determination of number of failures of ICEs operated under difficult conditions.

As it was mentioned before, the research was carried out in mountainous dessert of Afghanistan which is located in Central Asia and has borders with Tajikistan, Uzbekistan, Turkmenistan, Iran, Pakistan, and China (Fig. 1).

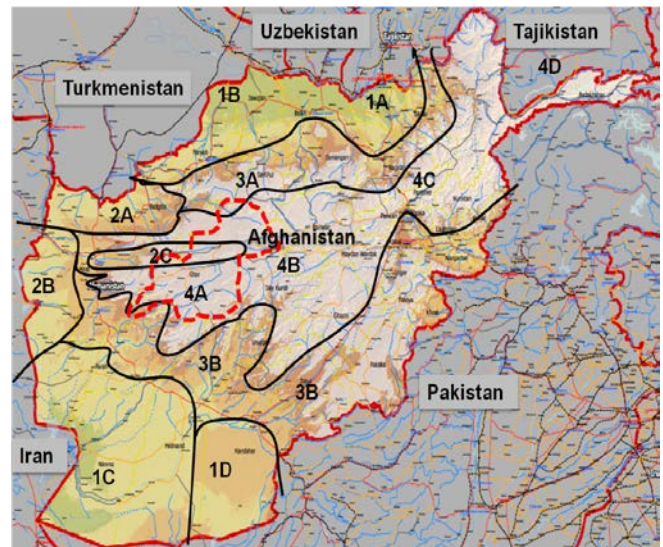


Fig. 1. Map of geographical location of Afghanistan. Regions of deserts: Oxus basin Amu Darja (1A), Kara-Kum Garagum, Badghyz and Karabil (1B). Clay deserts: Seistan, Dasht-e-Margo and Dasht-i-Arhu South, South-West Afghanistan desert (1C), Registan, South-East Afghanistan Sandy Desert (1D). Regions of semi-deserts: Khorassan (2A), part of Persian Iran) desert (2B), valleys of semi-deserts of the Afghan Mountains (2C). Regions of hills, semi-deserts and lower mountains that can be passed through: regions of Safedhok and Circum Oxiani-Hill (3A), and Siahkok (3B). Mountainous regions: Chorat (4A), Paropamiz and Hazarajat Mountains (4B), alpine plain surface Hindu Kush (4C), Pamir high-mountainous region (4H). (Michael A. Mares 1999, 2003).

Mountains are attributed to areas that make vehicle operation difficult and determine specific conditions for their operation. For this reason, making the data base starts with the geographic location of the region where automotive engines are to be operated. Operation of the engines under research took place under realistic

conditions in Ghor province of Afghanistan which is located in semi-desert valleys between the Afghan Mountains (2C) and Chorat (4A), the mountainous region of Afghanistan.

Engine failures tend to occur one after another at particular time intervals, thus ICE failure flow is indicated as the totality of variation of process' factors. The ratio of engine motor hours to time, as well as variation of operating conditions in time are not constant, consequently engine performance indicators and engine condition will highly vary in time in response to the changing intensity and conditions of ICE operation.

2. Investigation of number of failures of ICEs operating under difficult conditions

Assurance of ICEs reliability should never be based on approximate awareness of the average conditions; it requires for precise and accurate data on real operating conditions. Accordingly, the following factors are distinguished: mechanical effect; climatic and other environmental external factors that are dependent on the geographic location, season and time of a day, etc.; biological factors; radiation factors; external factors of electromagnetic fields; specific factors that influence environment externally; thermal factors.

Estimation of climatic and other natural factors influencing exterior of vehicles operated in Ghor province of Afghanistan, and their diurnal variations, used meteorological data recorded by the Air Traffic Control Centre of the Provincial Reconstruction Team of Ghor province using weather data recording equipment *Vaisala TacMet Tactical Meteorological Observation System MAWS 201M*, and processed using software *MIDAS IV Tacmet*. This equipment is used to detect and record meteorological data at one minute intervals 24 hours a day.

The most significant factors potentially leading to destabilization of performance of engines operating under conditions of mountainous desert are heat and cold that are characterized by temperature, relative humidity of air, precipitation, dust and sand, and solar radiation (insolation).

Variety of mountains in the region influences wind regimes. Wind has a negative effect like any other wind-related phenomena (such as snowstorms, sandstorms, hurricanes and tornadoes): a high wind speed puts an extra load on engine due to the increased force of vehicle resistance to movement. In the region under consideration, average wind speed ranges 5–6 m/s, whereas windflaws – as much as 30 m/s; at night a cold wind blows from mountains, whereas at the day time wind blows towards the mountains.

Atmospheric air pressure and air density varies in a course of year and tends to decrease when moving up the mountains. With these factors decreasing the effect of solar radiation and air ionization keeps increasing.

The studies of reference literature sources [7-15] show a negative effect of climatic factors on automotive ICEs to exist, therefore climatic factors and phenomena must be taken into consideration when dealing with the issue of ICE reliability. It's worth noting that several climatic factors tend to act simultaneously leading to the combined effect thereof which can further enhance this negative effect on automotive engines.

The weather adversity of Afghanistan's hot and cold climate for the vehicle operation shows only spotted assessment of the effect of climatic factors during three coldest or hottest months. Consequently, adversity of the climate for vehicle operation proves to be a general characteristic of climatic factors' effect. Afghanistan weather adversity for the vehicle operation is dependent upon the same climatic factors while taking into consideration negative or positive temperature over the period of exposure.

Climatic adversity for ICE operation was determined using the climate evaluation technique suggested by B. Kozhevnikov and based on the entropic method (according to formulas 1–6), however not entire year was taken for the evaluation at once, as the author suggested, but principles adapted from Koch and ir Shevcov were

followed, i. e., weather adversity for ICE operation during each month was assessed in order to show the monthly variation of weather adversity over the year.

Entropy is an extensive variable that specifies the state of thermodynamic system and expresses irreversibility of the isolated system. It varies at almost each process.¹

The amount of thermodynamic entropy is understood as a measure of irreversible spread, provided this energy is treated as an internal energy of the article contained inside its structure, whereas spontaneous energy spread influenced by the climatic factors, in course of time, makes integrity links existent inside the article to weaken, change or break. The resulting outcome – ageing and wearing of an object (Kozhevnikov 2010).

Based on the information theory, originally developed by C. Shannon (1963), the amount of disinformation occurring in result of the climatic factor effect, is marked by the minus sign, thus amount of change in entropy is calculated using formula (1) (Kozhevnikov 2010):

$$(-Q_{Fi}) = A_{Fi}^q \cdot t; \quad (1)$$

$$A_{Fi}^q = \left| \frac{(-q_F)}{(X_F - X_F^N)} \right|; \quad (2)$$

$$(-q_F) = \frac{(k \cdot \ln N)}{\Delta t}; \quad (3)$$

$$k = \frac{1}{\ln 2} = 1,4427; \quad (4)$$

$$(-Q_{KM}) = (-Q_A) + (-Q_B) + \dots + (-Q_M), \quad (5)$$

where: $(-Q_{Fi})$ – change in entropy over time t ; A_{Fi}^q – coefficient of linear regression angle slope; $(-q_F)$ – information speed of the climatic factor, bps; X_F^N – value of the climatic factor under normal conditions; X_F – current value of the climatic factor; N – number of intervals with the value ΔX that falls in range (XX_F^N) with the current value X_F ; $(-Q_{KM})$ – the integrated effect of climatic factors on object; $(-Q_A)$, $(-Q_B)$, ..., $(-Q_M)$ – change in entropy of climatic factors (phenomena) $A, B, \dots, M$ over time t .

Adversity of the climatic factor K_N for vehicle operation is calculated as a ratio of the integrated change in climate's entropy $(-Q_{KM})$ over a specific time t to the change in climate's entropy $(-Q_F^N)$ under normal conditions over the same time t :

$$K_N = \frac{(-Q_{KM})}{(-Q_F^N)}. \quad (6)$$

Any time interval can be selected: a week, a month, a quarter, a year, etc. In the case under consideration time intervals of a month and a year are the most appropriate, as such a selection allows for calculation of weather variation during the year while taking into consideration a specific region of vehicle operation.

The maximum value of the adversity of the climatic factor for engine operation belongs to the cold season, and is followed by relative humidity in the second place by significance, whereas the third place is shared between monthly mean air temperature, solar energy irradiation and mean air temperature variation amplitude, with minor error, then closely followed by the scattered solar energy irradiation; wind speed, air pressure, and average air temperature during three hottest months of the warm season were found to be less important. Other factors – duration of the warm

¹ Zavadskas, E. ir kt. *Technikos enciklopedija*, t. I, Vilnius, 2000, p. 572.

season, fluctuations in air temperature through 0 °C, time period of mist coverage – do not determine adversity of the climate to ICE operation in the region, and such factors as amount of precipitation and time period of precipitation and wind direction have no influence on engine performance at whatsoever.

Fig. 2 offers the graph of climatic adversity for ICE operation obtained based on calculation results.

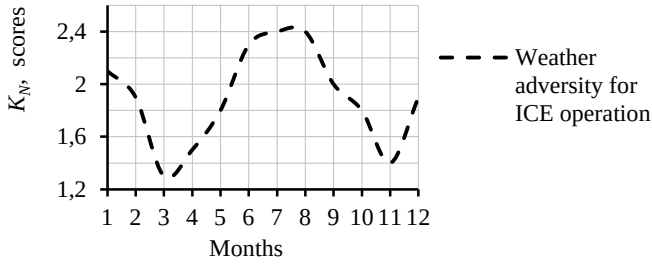


Fig. 2. The climatic adversity for ICE operation monthly for the period of one year in 2005–2011

Geographical location of Afghanistan determines the climate of its regions, which is described as an extremely dry continental climate with characteristic huge amplitude of fluctuations in air temperature. The characteristics of the climate and other natural factors prevailing in Afghanistan region were identified having influence on the exterior of internal combustion engines.

Not only ICE but a technical personnel is also influenced by the mountainous terrain of Ghor province and its climatic factors. Summary table of the adversity of climatic factors for ICE operation based on calculation results reveals that the climatic factor ranges from 1.3 to 2.4 over the year while achieving its maximum values in winter and summer times, and lower values – in autumn and spring. Values of the effect of climatic factors were established, and their priority sequence was formed by applying the calculation methodology.

Seasonal variation in climate changes conditions on roads, and in Afghanistan, vehicles are mainly operated on unpaved roads. Varying humidity and temperature of such roads causes rolling resistance of a vehicle to change significantly, respectively increasing the loading on a vehicle's transmission and engine, all of which results in increased number of failures, too. Variation of road conditions over a single year was expressed as the rolling resistance coefficient of a vehicle. Data were found experimentally in accordance with the methodology suggested in this thesis.

Actual values of the vehicle rolling resistance coefficient were found experimentally using methodology offered by R. Lahno (Fig. 3).

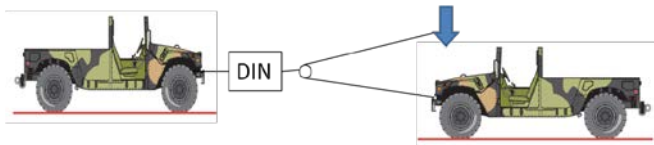


Fig. 3. Principal scheme for finding the vehicle rolling resistance coefficient.

The vehicle rolling resistance coefficient (Fig. 4) is calculated as a sum of ground resistance to motion of a rolling tire and longitudinal angle of uphill (or downhill). Given the fact that longitudinal angle of uphill or downhill remains constant in the same road section, the rolling resistance coefficient is calculated as follows:

$$f = \frac{f_1 l_1 + f_2 l_2 + \dots + f_n l_n}{l_1 + l_2 + \dots + l_n}, \quad (7)$$

where: $f_1, f_2, \dots, f_n$ – actual values of the vehicle rolling resistance coefficient in the road section 1,2,...,n; $l_1, l_2, \dots, l_n$ –

actual value of a vehicle rolling distance in the road section 1,2,...,n.

The rolling resistance coefficient is as follows:

$$f_r = 0.073 + 0.035 \cos[30(t_i - 2.5)]. \quad (8)$$

The rolling resistance coefficient of a vehicle was established defining potentially increased ICE loading having the dependency of seasonal features. The rolling resistance coefficient of a vehicle was found to vary all year round: it was observed to achieve its maximum values in spring $f_r = 0.11$, and minimum values – in summertime $f_r = 0.043$.

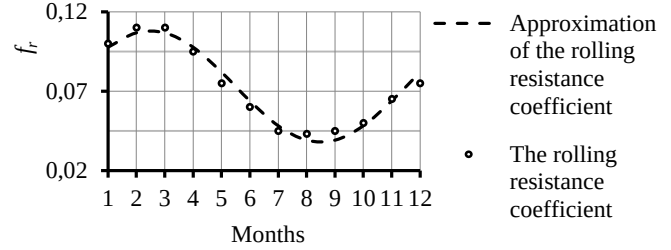


Fig. 4. Diagrams of the rolling resistance coefficient and its approximation for each month of the year 2010.

Engine operating intensity is found from vehicle operation records and logs, and using the same principle as in case of finding operating conditions, ICE operating intensity is then defined as follows:

$$l_{ei} = l_0 + \sum_{k=1}^{g_q} l_{yk} \cos[m(kt_i - t_{0k})] + l_p, \quad (9)$$

where l_0 – constant component of the operating intensity factor which is the mean value of X per cycle; k – harmonic number; g_q – number of harmonics under investigation; l_{yk} – fluctuation wave of the amplitude at the harmonic side k of the operating intensity factor, m – interval between t_k and t_{k+1} in degrees; t_{0k} – fluctuation phase of the initial wave in degrees; l_p – incidental part of the climatic factor corresponding to time (t).

$$l_0 = \frac{1}{n_d} \sum_{i=1}^{n_d} \frac{L_{i+1} - L_i}{t_{i+1} - t_i}, \quad (10)$$

where n_d – number of days in a month under investigation; n_a – number of vehicles under investigation in the group; $L_{i+1} - L_i$ – ICE operating intensity in the interval between the month t_{k+1} and t_i , in km; $t_{i+1} - t_i$ – the interval of a month under investigation between the month t_{k+1} and t_i .

The intensity of operation l_{ei} of diesel engines GM 6.5L of M998A2 cars under investigation was defined through the following dependencies:

$$l_{ei} = 36.5 + 9.3 \cos[30(t_i - 8.0)] + 2.3 \cos[30(2t_i - 1.0)]. \quad (11)$$

The intensity of operation l_{ei} of diesel engines 1HD-FTE of Toyota Land Cruiser 100 cars under investigation was defined through the following dependencies:

$$l_{ei} = 71.2 + 17.1 \cos[30(t_i - 7.3)] + 1.9 \cos[30(2t_i - 3.0)] + 4.1 \cos[30(3t_i - 3.9)]. \quad (12)$$

Engine operating intensity and rolling resistance vary all year round. However, it's worth noting that engine operating intensity is by 1.5 times higher in summertime than in late winter.

Each vehicle has its specific operating intensity over the same period of time. For the purpose of calculations, mean operating intensity value was selected depending on the same group and type of cars.

The approximation equation of the average monthly run L_i for the high-mobility multipurpose wheeled vehicles M998A2 (HMMWV) with diesel engines GM6.5L is defined through the following dependency:

$$L_i = 143 + 42 \cos[30(t_i - 8,2)] + 5,7 \cos[30(2t_i - 10,7)] + 12,6 \cos[30(3t_i - 3,9)]. \quad (13)$$

The approximation equation of the average monthly run L_i for the Toyota Land Cruiser 100 with diesel engines 1HD-FTE is defined through the following dependency:

$$L_i = 216,6 + 54,9 \cos[30(t_i - 7,3)] + 5,7 \cos[30(2t_i - 3,0)] + 12,6 \cos[30(3t_i - 3,9)]. \quad (14)$$

Regression model is suitable as determination coefficient $R^2=0.98$.

The error of approximation of statistical data of vehicle operating intensity and rolling resistance is below 5 %, and determination coefficients are in range of $R^2=0.92 \div 0.96$, which serves as the evidence of the suitability of mathematical expression.

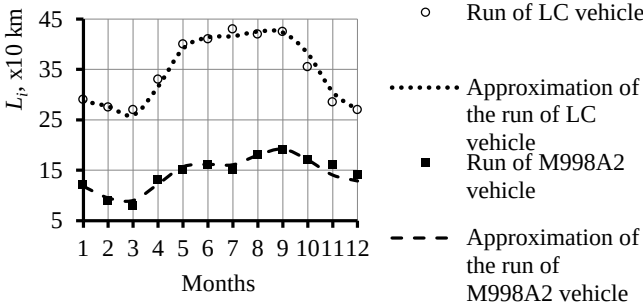


Fig. 5. Dependency of the run L_i of M998A2 and LC vehicles for each month in the period of 2005–2011.

Experience gained through vehicles operation in Afghanistan revealed the following three main reasons behind inoperability of vehicles: combat damages; scheduled repairs after the established amount of motor hours or kilometres run by engine; car accidents and failures in a course of vehicle operation (operating failures) (Dobrzinski, Markshaitis 2009). Failures of engines used in vehicles are also grouped into the following three groups: catastrophic, parametric and combined failures. Taking into account high numbers of ICE failures, engine as a repairable part of a vehicle is described using a particular law of distribution (Lukinskij, 1982). ICE failure flow must be known when setting ICE reliability criteria based on values of which the adjustment of periodic maintenance intervals, need for repairs, and amounts of spare parts stored and available for repairs can be accomplished. When studying ICE operating reliability, the distribution law of ICE failures must be determined VDV.

The collected data on operating failures of diesel engines 1HD-FTE and GM 6.5L used in vehicles Toyota Land Cruiser 100 and M998A2 respectively, operated in Ghor province for 6 years, allowed for constructing dependencies of engine failure flow parameter for each month in the period of 2005–2011 (Fig. 6).

Research shows the failure flow of ICEs operated in Afghanistan to be more intensive in winter and summer times. Engine failure is obviously dependent in the weather adversity for vehicle operation in the region. Engine GM6.5L used in vehicle M998A2 was found to be more dependent on weather adversity when compared to 1HD-FTE used in Toyota Land Cruiser 100. It proves that engine 1HD-FTE has better adaptation potential to such operating conditions. Linear correlations by engine failure flow and

weather adversity for operation can be calculated from the dependency equation. It was selected due to the fact that scores of engine failure flow and weather adversity for vehicle operation are random values, whereas a link between them is of correlative nature, thus it should be examined using correlation and regression methods (Koch 1981).

In case of the research under consideration, a need emerged to evaluate multiple factors simultaneously, therefore a hypothesis was made about using a multi-factor adaptive model identifying operational and road conditions as well as ICE operating intensity:

$$\omega = \omega_0 + a_k K_N^2 + c f + b l_{ei}, \quad (15)$$

where ω_0 – the least parameter of ICE failure intensity, corresponding to the most favourable conditions of engine operation; a_k – the sensitivity of the parameter of ICE failure intensity to the adversity of engine operation; c – the sensitivity coefficient of the ICE failure intensity parameter to road conditions; b – the sensitivity coefficient of the ICE failure intensity parameter to engine operating intensity; K_N – current adversity of ICE operation; f – rolling resistance coefficient; l_{ei} – the ICE operating intensity.

In case under consideration, the failure flow parameter depends on the following three factors: climatic adversity, road conditions, and operating intensity. Given research findings and the equation (15) of proposed hypothesis, a system of linear equations can be constructed which is further expressed in the matrix form.

To find out coefficients of polynomial approximation, the system of linear equations was solved in the matrix form. When solving the system of equations, the coefficients of mathematical simulation of the approximation are calculated such as relationship between the engine failure flow, weather adversity for ICE operation, rolling resistance coefficient, and operating intensity, which results in the following regression function.

The engine failure flow ω parameter of diesel engines GM 6.5L of M998A2 cars under investigation was defined through the following dependencies:

$$\omega = 0.01 + 0.28 K_N^2 + 2.4 f_r + 0.005 l_{ei}. \quad (16)$$

The engine failure flow parameter ω of diesel engines 1HD-FTE of Toyota Land Cruiser 100 cars under investigation was defined through the following dependencies:

$$\omega = 0.026 + 0.06 K_N^2 + 0.6 f_r + 0.002 l_{ei}, \quad (17)$$

Calculations of the approximation of the failure flow parameter that was defined through the dependence (16 and 17) for each ICE model respectively.

Increase of number of kilometres of engine run with time is the process which is dependent on the amount of engine kilometres run per single time unit, or put it otherwise, on the operating intensity. For this reason, number of kilometres run by engine over time t is expressed as follows:

$$\Delta L_i = \int_0^t l_{ei}(t) dt. \quad (18)$$

Taking into account equation (5) results in the following engine kilometres run over time period t :

$$\Delta L_i = \int_0^t \left(l_0 + \sum_{k=1}^{g_g} l_{Yk} \cos[m(kt_i - t_k)] + l_p \right) dt. \quad (19)$$

The engine failure flow is defined as a mean number of failures (Fig. 6) over time t , which is calculated from the following equation:

$$\Omega = \omega \Delta L_i. \quad (20)$$

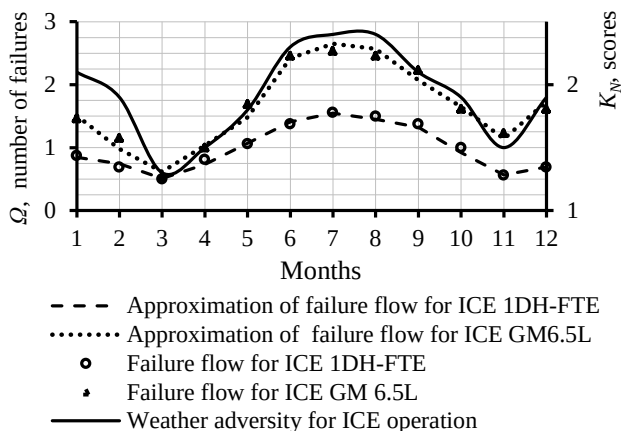


Fig. 6. Distribution the failure flow of engines operating in Ghor province of Afghanistan in the year, depicted monthly

As research shows, the failure flow parameter is variable, not constant, whereas the reliability function differs from the exponent. Assessing engine reliability requires taking into account deviation of the distribution of number of failures from Poisson distribution.

3. CONCLUSIONS

This paper offers analysis of the climate in Afghanistan regions and other characteristics of natural factors having effect on ICE exterior, as well as road conditions and intensity of ICE use under realistic conditions of diesel engine operation. In summary:

1. Geographical location of Afghanistan determines the climate of its regions, which might be described as extremely dry continental climate with characteristic huge amplitude of fluctuations in air temperature. The characteristics of the climate and other natural factors prevailing in Afghanistan region were identified having influence on the exterior of internal combustion engines.

2. Summary table of the adversity of climatic factors for ICE operation based on calculation results reveals that the climatic factors range from 1.3 to 2.4 scores during the year while total climatic adversity for ICE operation scores 1.9. Maximum values are achieved in winter and summer times, and lower values – in autumn and spring. Values of the effect of climatic factors were established, and their priority sequence was formed by applying the calculation methodology.

3. ICE operation intensity was expressed as the vehicle operation intensity which keeps varying all year round. Engine operating intensity was found to be by 1.5 times higher in summertime than in late winter.

4. The rolling resistance coefficient of a vehicle was established defining potentially increased ICE loading having the dependency of seasonal features. The rolling resistance coefficient of a vehicle was found to vary all year round: it was observed to achieve its maximum values in spring $f_r = 0.11$, and minimum values – in summertime $f_r = 0.043$.

5. ICE reliability indicators of diesel engines 1HD-FTE and GM 6.5L used in vehicles Toyota Land Cruiser 100 and M998A2 respectively, operated under conditions of Afghanistan mountainous desert in Ghor province, were found to be dependent on weather adversity for ICE operation, road conditions, and operating intensity, and were constructed based on the adaptive three-component equation which allowed for calculation of ICE failure flow parameter and failure flow.

6. Major factors that determine reliability of ICEs operated difficult conditions are territorial, climatic, biological, human factors, and operating intensity.

7. The failure flow parameter for Toyota Land Cruiser 100 with diesel engine 1HD-FTE and for M998A2 with diesel engine GM6.5L, operated under difficult conditions has been examined. ICE failure flow parameter varies depending on the engine type: for 1HD-FTE – in range of 0.5 to 1.5 failure per month, and for GM6.5L – in range of 0.51 to 2.53 failure per month; generally, ICEs were found to fail less frequently in autumn and spring. In winter and summer-times ICE failure fluctuation amplitude was twice as large when compared to periods of spring and autumn. In other words, it is dependent on the season,

8. ICEs of different structure when operated under difficult conditions show different potential of adaptation: vehicles Toyota Land Cruiser 100 with diesel engines 1HD-FTE, operated under difficult conditions, are twice as durable when compared to vehicles M998A2 with diesel engines GM6.5L.

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RETROFIT OF SHIPS MAIN ENGINE REMOTE CONTROL SYSTEMS

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Abstract: The current paper will present the methods, ways and concrete schemes for retrofit of ships main engine remote control (RC) systems. The drawing solutions from realized projects will be discussed for different engines. Analysis for the projects is done. Modern remote control systems of ships diesel engines are mentioned in this paper.

KEY WORDS: SHIPS ENGINES, CONTROL, SYSTEMS RETROFIT AND UPGRADE

I. Introduction

Nowadays, electronic equipment development in ship industry created a good conditions for increasing of accuracy and reliability of RC system for main engines (ME). The most of new-built vessels are equipped with electronic RC system. But into the world ocean are sailing and older ships with earlier versions of RC operated with mechanical-pneumatic manner. Many ship owners want to increase reliability and working hours of their units but, how it looks like, the spare parts are missing already, so only one solution is possible – renewing of RC system. At the moment a few projects are done for non-reversible engines and one for reversible slow speed engine MAN B&W type. All new systems are approved by Register's classification inspection with attached documentation.

II. Projecting and creating of new RC systems:

After job confirmation latter is received the work plan is accomplished:

- Familiarization with engine type, all operating curves, work modes, limits and safety algorithm.
- Projecting of control algorithm, including sequence of RUN, STOP, REVERSE(if any) modes, clutch operation and propeller pitch, safety and limits programs.
- From the algorithm done the new elements are extracted like type and numbers.
- Delivery, fit on place and sea trails.
- New drawings and papers are prepared.
- Giving over with ship crew and the Register.

Pneumatic and Electric Components

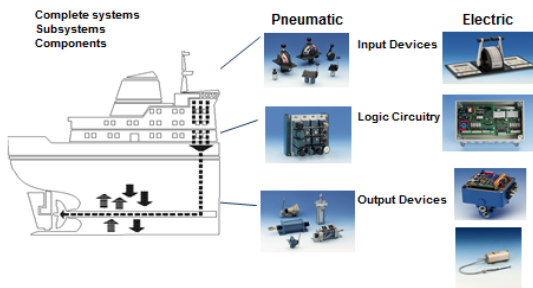


Fig.1

Fig.1 shows the devices which are replaced each other. Speed setting comes through analogue signal 4-20mA, instead of governor with air control pressure 0,5-4,5 bar. The logic algorithm is programmed into controller, instead of pneumatic box with 40-60 valves. The final units are transmitters current/pressure or electro-motors. They replaced pneumo- and hydrocylinders and valves also. At the projects done ME's are mostly SKL, MVM and Deutz type. The mode change is provided by engine rpm and the propeller is with fixed pitch. Engine reversing become by means of electrically controlled clutch (Fig.2.) This is an example for pneumatically system consisted of distributors, valves, air service station and control handle connected with air reducing valve for speed setting. All elements required periodical maintenance and repair kit replacing from manufacturer made. Unfortunately for mostly old vessels the spare parts price is too high or the manufacturing is already closed.

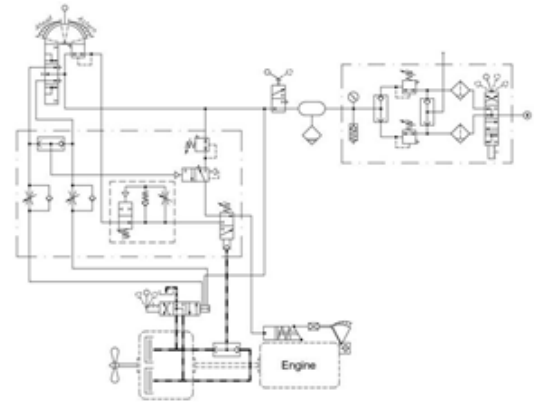


Fig.2

The scheme used for control of hydraulic distributor on reverse clutch and mechanical control of ME speed setting is shown on Fig.3.

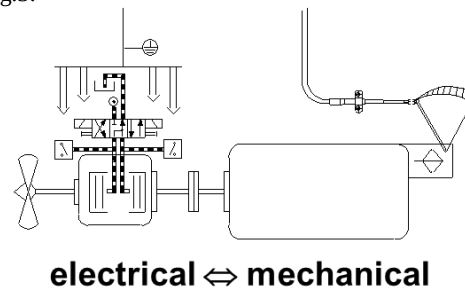


Fig.3

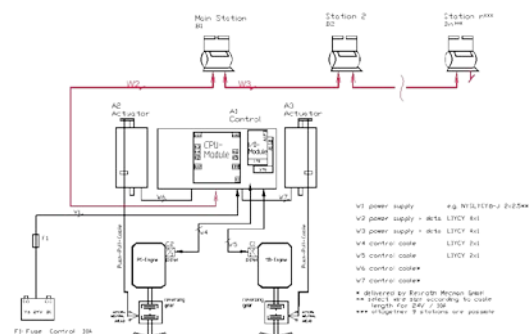


Fig.4

As presented (Fig.4) three control stations for ME are connected with CAN bus in between and all are linked with main controller which calculated output signals to rpm governor and reverse clutch. In this case ship power plant contains two ME's operating with two fixed pitch propellers.

Main functions realized with this solution:

- Speed setting and clutch control.
- Propeller pitch control(option).
- Modes: normal, reverse and emergency reverse.
- Monitoring and safety systems.

Some of projects are made with original mounted sensors and safety switches (like oil pressure switch) and the rest projects used only fully replaced measuring and safety systems.

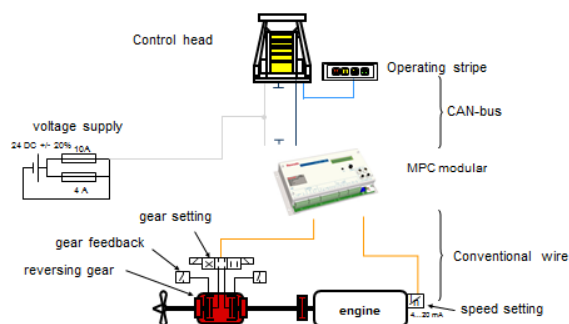


Fig.5

The principal diagram of ME RC system is shown on Fig.5. Here is presented a system with one control station and a panel with the following signals emergency RUN, STOP, and LIMITS CANCEL. Independent supply is ensured by batteries 24 VDC which gives reliable control and alarms.

- M/v Sredna Gora-replacement of the remote control, safety and alarm system .



Fig.6

- M/v G. Mamarchev, m/v V. Aprilov- replacement of the remote control systems together with the governors. Engine type-SKL reversible.



Fig.7

The main components of river ship RC systems are displayed. On Fig.6 monitoring system includes three ME's equipped with separate control and programmable module.

Especially for these purposes the mimic alarm station is developed. It is adapted by types and numbers of alarms depending of engine model and original curves. The "freeze" function is very important and suitable during engine shut down since keeping the "picture" from the condition before stop mode.

- Features:
 - own microprocessor and memory
 - digital and analogue inputs
 - Possible connection to PC
 - Function -freeze at engine stop

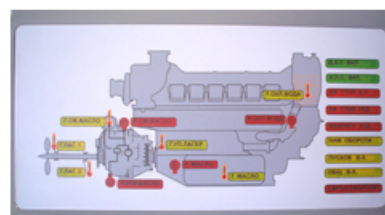


Fig.8

The list with base functional CPU parameters is applied below. The adjustments depend from engine type, power plant configuration and operational modes.

Parameter representation of the I/O modules

Parameter	menu display	values	default	min	max	meaning
2 version of the table of parameters	-	-	11	11	11	
3 start-up state of relay R2	-	0	0	2	2	0: relay is turned off
4 start-up state of relay R3	-	0	0	2	2	1: relay is turned on
5 start-up state of relay R4	-	0	0	2	2	2: relay remains unchanged
6 configuration of the analog input 1 (board 346 067 716 2, clamp)	Input1	0	0	4	4	0: current 4-20mA (R _{in} 200Ω)
7 configuration of the analog input 2 (board 346 067 716 2, clamp)	Input2	0	0	4	4	1: current 0-20mA (R _{in} 200Ω)
8 configuration of the analog input 3 (board 346 067 716 2, clamp)	Input3	0	0	4	4	2: voltage 2-10V (R _{in} 10kΩ)
9 function of the voltage output (board 346 067 716 2, clamp 57)	-	0	0	1	1	3: voltage 0-10V (R _{in} 10kΩ)
10 configuration of the analog output 1 (motherboard, clamp 16/17)	-	0	0	1	1	4: resistor (0-10V) (R _{in} 2.5MΩ)
11 initialization value of the analog output 1	-	0	0	32760	32760	0: 10V reference output
12 configuration of the analog output 2 (board 346 067 716 2, clamp 58/60)	-	0	0	1	1	1: 0-10V (adjustable via PDO)
13 initialization value of the analog output 2	-	0	0	32760	32760	0: 4-20mA
14 Life time factor	LFT	2	0	255	255	1: 0-20mA
15 Guarding time	GT	1000	0	65535	65535	0: 0mA
16 Basis-Id for Guarding	-	0x7002	0x6e0	0x7de	0x7de	32760: 20mA
17 module-Id of the I/O-nodes	Id	5	1	100	100	0: 0mA
18 selection of the CANopen Boot-up	qBoot	0	0	1	1	32760: 20mA
19 display of the parameters and errors in the menu	sym-bols	1	0	1	1	0: Boot-up after CANopen. The node assumes after power-on at first the state preoperational and only after a NMT-Message from the Master has been received, it moves to the state operational. 1: as opposed to CANopen the node moves directly to the state operational (is only used if operating older master nodes) Note: only in the state operational the inputs and outputs are available via PDOs

Fig.9

Signal flow

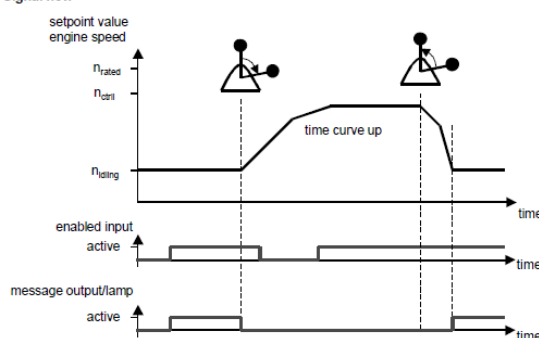


Fig.10

Fig.10 shows speed setting program curve for one of the projects for SKL engine.

IV. Results analysis

The key results from new systems introducing are higher reliability, higher accuracy with speed setting and easier readjusting of parameters according staff requirements. At one of the vessels the old RC system is saved and new one works separately from it. Since three years the original one have used never. Also good scope is a

light maintenance which no needs yearly based prophylactic nor an air compressors with air service system.

V. Conclusion

ME RC system retrofit is a complicated and accountable engineering task. The discussed projects are made thanks to:

- Good team work of all involved persons.
- Trusty equipment and software approved with Register's admissions.
- Field experience and skills with ship automation systems.

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A TESTING STATION FOR STUDYING THE COMBUSTION OF FUEL WITH MECHANICAL CENTRIFUGAL AND ROTATING CUP BURNERS.

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Abstract: The burner of the ship's steam boilers is a device in which air is mixed with fuel with the purpose of burning and supplying heat power. A continuous combustion process of gas and liquid fuels is ensured by means of burners.

KEYWORDS: BURNER, COMBUSTION OF FUEL IN BOILERS, ROTATING CUP OIL BURNER, PRIMARY ATOMIZING AIR STREAM, STEAM BLAST JET TYPE FUEL OIL BURNER.

1. Introduction

A burner is a device in which fuel and air are mixed in order to provide efficient combustion and to generate heat power. A continuous combustion process of gas and liquid fuels is ensured by means of burners. Modern fuel oil burners are complex devices consisting of many components. These burners comprise different types of control and adjustment systems. Fuel oil burners are manufactured as separate modules that can be fitted to different types of burners.

According to the type of the fuel used burners are classified into:

- fuel oil burners;
- gas burners;
- dual fuel burners

According to how the combustion mixture is formed, the fuel oil burners are classified into:

- mechanical;
- injection;
- rotary cup.

2. Problem discussion

The present paper deals with the issue of bringing the testing station into an operational/ simulation condition in order to exhibit the performance and control of the rotary type burner on the one hand and the two-stage mechanical burner on the other hand. Both types of burners are prevalent in the product range of ship's heat equipment manufacturers due to a number of advantages: the scope of burner control has been considerably increased, the burners are not very sensitive to the fuel viscosity, they are easily atomized, different types of fuel can be used.

3. Objective and research methodologies

A lot of parameters affect the quality of the combustion process and the resulting emissions. These include burner adjustment, geometry of the furnace, fuel pressure, maximum values of the temperature and the pressure in the combustion chamber (furnace space), values of the air temperature and pressure [1]. The fuel atomization can be optimized further in the operational process through analysis of the burner flame by means of the testing station.

Flame combustion is accomplished by means of burners. It is widely spread. This is the oldest method for fuel burning. It is used in boilers, furnaces, technological equipment and engines. With this type of combustion the flame is a stable flame in a regular shape. The shape of the flame depends on;

- type (form and design) of the burner;
- the degree of mixing of fuel and oxidant;
- the stream of the burning mixture.

The front of the flame is the boundary between the core and the ignition area. The core is an area where the main ignition parameters are formed. The shape and the size are determined by the intensity of the mixing of the fuel and the oxidant.

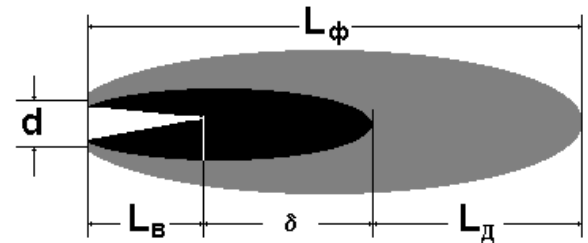


Fig.1 A scheme of flame combustion

L_B - ignition area

D - combustion front, has thickness less than L_B and L_ϕ

L_L - forcing area; has an elliptical shape

L_ϕ - length of flame as a whole; has an elliptical shape

The flame is externally seen as one dark cone with a light elliptical area. The efficiency of combustion is determined by the total length of the flame. The testing station gives the opportunity of controlling the shape of the flame so that optimum results could be achieved for the specific mode of operation.

The testing station for training purposes is made on a mobile platform and its aim is to illustrate the operation of two types of burners used in shipbuilding. At the one end of the platform there is a rotating cup burner with its service and control systems whereas at the other end there is a mechanical burner type Monarch ("Meteor") manufactured by MP "Spartak" (Burgas) along with its control panel. Both burners are positioned opposite to each other in a "furnace", the upper half is made of transparent Plexiglas to provide for good visibility. When operating the burners use oil from a common tank/ reservoir. Oil is selected as the burning fluid due to safety considerations. The oil drained at the bottom is delivered back to the reservoir by an oil pump. Both burners have an autonomous power supply and they can operate independently – see Fig.2.



Fig. 2 Testing station for combustion

The injection burners type „Monarch“ are fully automated. Their design has been considered down to the smallest detail and is being kept up-to-date. They meet all requirements for safety, reliability and low cost. The burners have the following characteristics:

- Automatic sequence of operations;
- Stable fan characteristics – good combustion process;

- Air damper closed on burner shutdown;
- Quiet operation;
- Burner casing hinged to the boiler;
- The design of the burner makes installation, adjustment and servicing easy.

The design, control and operation of the mechanical burner on a testing station are shown in Figures 3, 4 and 5.



Fig. 3 Burner type „Monarch“

- 1.Hinges securing the burner to the furnace;
2. Flame observation port;
3. Asynchronous three-phase motor;
4. Air damper;
5. Pressure gauge for the pump delivery pressure;
6. Pump inlet;
7. High pressure pump;
- 8.Electromagnetic valves controlling the fuel stream to the burner nozzles

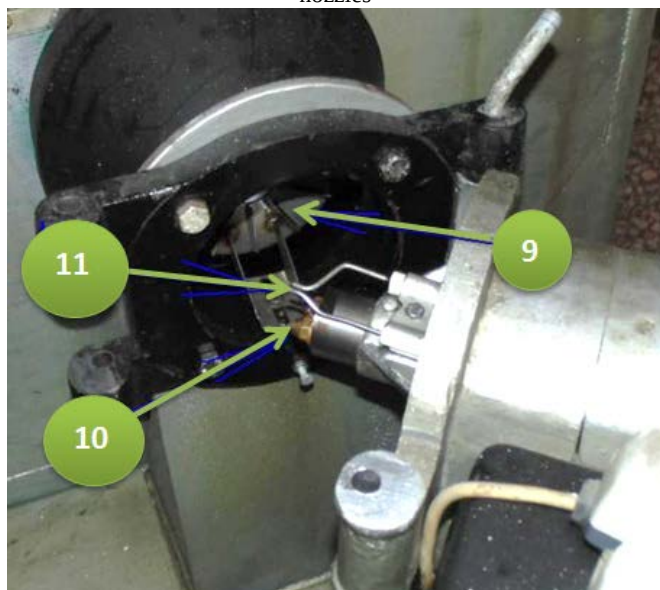


Fig.4 Burner type „Monarch“

9. Diffuser;
10. Nozzles;
11. Ignition electrodes



Fig.5 Burner control panel

switch 1 – breaking the electrical circuit to the electrical switchboard;

switch 2 –the fluid for the burner to operate on (HFO or diesel oil) is determined;

switch 3 and 4 – start / stop

The electrical switchboard for operation and control of the burner type “Meteor” is fully automated. The switch “1” serves to break the electrical circuit to the electrical switchboard. Switch “2” is used to determine the fluid that the burner will operate on – HFO or diesel oil; this choice is important for the operation of the heater i.e. the electrical heater does not work if the burner operates on diesel oil. When the switch is in position “HFO”, the burner control algorithm activates an electrical heater whose function is to maintain a certain viscosity of the fuel. By switches „3“ and „4“ the burner is started or stopped. Above the control push buttons there are light signals located from left to right as follows:

- Green light „testing station power supply“;
- Green light „cold HFO“- it indicates low temperature of the fuel used;
- Red light „failure“;
- Green light „1st stage“, it indicates that the burner is operating at first stage;
- Green light „2nd stage“, it indicates that the burner is operating at second stage;
- Green light „Heater“, indicates that the electrical heater is operating;
- Red light „overheated HFO“.

When switch “1” is set in position „on“, voltage is supplied to the switchboard and the green light comes on. Voltage passes through three of the fuses and is supplied to a step-down transformer 380V-220V; then to the windings of the seven contactors as well as to the Programmable Time Relay (PTR) and the Motor Time Relay (MTR). After pushing the “Start” button the PTR is activated; as a result of that the asynchronous three-phase electrical motor is activated. As the fan and the fuel pump are connected to the electrical motor axis, both delivery of fuel from the service tank and purging of the furnace start. The time for the furnace purging is measured by the Motor Time Relay (MTR). After the time set expires, the Programmable Time Relay (PTR) supplies voltage to the electromagnetic valve of the first nozzle. Voltage is simultaneously supplied to the ignition electrodes. When the photo relay (PR) registers the presence of a flame, the supply of voltage to the electrodes is suspended and the photo relay starts to measure the presence of the flame. If a satisfactory flame fails to be generated at the first attempt or the flame fails to settle down, the red signal “failure” appears and an alarm is sounded.

Rotating cup burners have been developed as a result of the efforts to eliminate the usage of steam (air) when atomizing HFO and to ensure efficient performance for all modes of operation. With the rotating cup burners the fuel is atomized centrifugally. Various designs of this type of burners exist [2].

Rotating cup burners are the most sophisticated type of burners at present. Atomization is accomplished by means of a

conical cup rotating at a speed of 4500÷8000 rev/min. They have the following characteristics:

- reliable, not sensitive to the degree of purity of fuel due to the lack of narrow channels;
- there are no high-pressure and high temperature pipelines that involve risks and dangers;
- depth adjustment $d=18\pm 20$;
- fully automated;
- complex design; hard to repair;
- efficiency depends on the change in the orifice section of the fuel channel;
- the primary atomizing air stream can be supplied by the main fan but more often it comes from a fan mounted on a common axis with the rotary cup and moving along with it [3].

Ignition is done by means of a pilot burner on light fuel oil which is automatically extracted from the furnace with certain models of boilers – see Fig.6 and Fig.7.

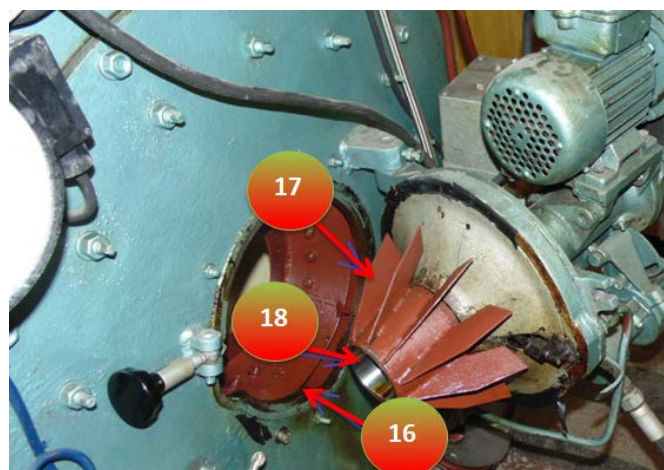


Fig.7 Rotating cup oil burner

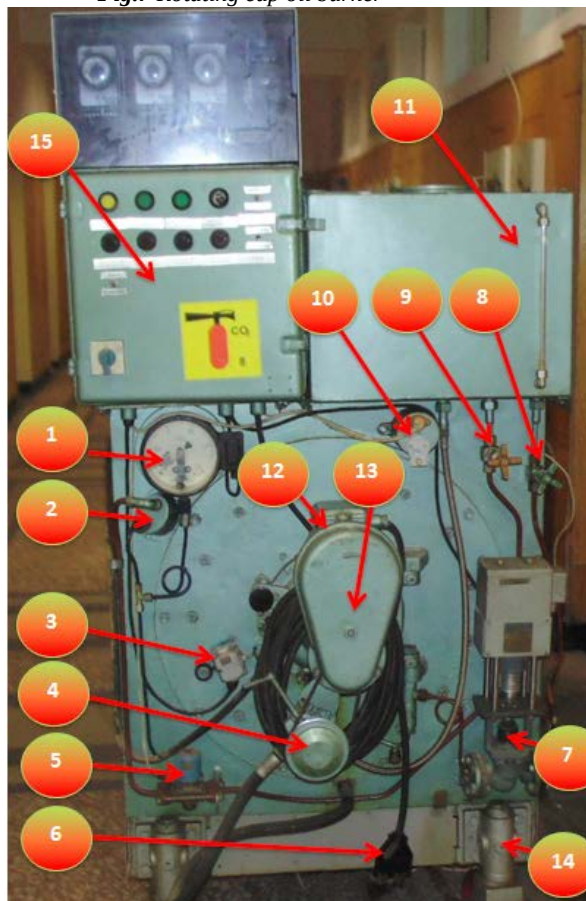


Fig.6 Rotating cup oil burner

1. Pressure gauge „U“ indicating the fuel pressure in the system after the ignition burner;
2. Ignition torch;
3. End switch “B5” of the main torch;
4. Oil pump;
5. Solenoid valve „S1“ for cutting off the fuel to the ignition torch;
6. Power cable;
7. Solenoid valve „S2“ for cutting off the fuel to the main burner;
8. Manual shut off valve for the fuel supply of the main rotating burner;
9. Manual shut off valve for the fuel supply of the ignition burner;
10. Lamp imitating the flame „L1“ and photo relay measuring the flame;
11. Reservoir;
12. Three- phase asynchronous motor;
13. Rotating cup burner;
14. Wheels;
15. Electrical switchboard;
16. Furnace casing nozzle;
17. Burner casing nozzle;
18. Swirling (rotating) cup;

This part of the testing station for the rotating cup burner is controlled by the electrical switchboard which comprises light signals and push buttons.

4. Conclusion

The method of atomization of the fuel in rotating cup burners and injection (mechanical) burners is an advantage which makes them widely used. They are equally efficient for good quality light and heavy fuel oils which is a particularly important feature for the ship's burners. Some research has proved their high reliability which results in fault-free operation.

The methods and equipment for fuel oil combustion in ship's boilers have been presented in the paper. General description and design of the rotating cup and mechanical burners have been discussed. The electrical circuits for the testing station studying the flame shape and combustion with mechanical and rotating cup burners have been restored and made.

The existing testing station can be used for conducting different experiments in connection with the flame shape as well as with the size of the atomized combustion particles. The modern statutory requirements for environmental protection include strict restrictions in relation with sulphur admixtures in different types of fuels such as F.O., light fuel oil, heavy fuel oil, and diesel oil. These sulphur limits are already in force for the sea areas defined as SECA (Sulfur Emission Control Area) [4] and they comprise the North Sea, the Baltic Sea, North America and Canada as well as all European Union ports. The present day rotating cup and mechanical burners meet the modern requirements for full automation, potential for unification, high technology, maximum energy efficiency and good degree of environmental protection.

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A STUDY OF CHARGE STRATIFICATION IN THE TWO-STROKE ENGINE WITH GASOLINE INJECTION

ИССЛЕДОВАНИЕ ПОСЛОЙНОГО СМЕСЕОБРАЗОВАНИЯ В ДВУХТАКТНОМ ДВИГАТЕЛЕ С ВПРЫСКОМ БЕНЗИНА

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Abstract: *Charge stratification of the combustible mixture through separate input of air and gasoline in the two-stroke gasoline engine is studied. Clean air enters the upper part of the vent channels through the check valve plate and then goes into the crank camera. The mixing occurs in the crank chamber and is controlled by the time and duration of the fuel injection nozzle. Before blowing cylinder volumes of the purge channels are filled mostly with clean air. The volume of the crank chamber is filled with a mixture of gasoline and air. Blowing the working volume of the cylinder is carried out mostly by clean air volumes from the purge channels. The process of purging is completed by filling the cylinder with fresh mixture from the crank chamber. Layer-by-layer purging of the cylinder greatly reduces the direct losses of fuel into the exhaust system. This leads to increased fuel efficiency of the two-stroke engine and reduces environmental pollution. Maximum power is achieved for mixtures with air ratio $\alpha = 1.0$, the best efficiency on the poorer mixtures with $\alpha = 1.2-1.25$. In this case the reduction of fuel consumption is up to 20 %.*

KEYWORDS: TWO-STROKE ENGINE, GASOLINE INJECTION, CHARGE STRATIFICATION, SCAVENGING OF THE CYLINDER, AIR RATIO.

1. Introduction

Internal combustion engines running on gasoline, are currently the main source of energy for vehicles, various types of agricultural machinery, motorcycles and many other types of equipment.

The foundation of any engine using hydrocarbon fuels, is the process of converting thermal energy of the fuel during its combustion into mechanical work. In most engines this process is carried out on the four-stroke cycle, but only one measure - the third - is working. For the other cycles the input of mechanical energy for compression of the combustible mixture and the gas exchange in the cylinder is required. Therefore, the vast majority of four-stroke engines have four or more cylinders. The engines provide high power per unit working volume of the cylinder, a fairly good economy, affecting fuel consumption and low exhaust emissions, which reduce the means of neutralization.

However, many types of small vehicles - motorcycles, compact tractors, motor grass-mowing machines, snowmobiles, small boats and other types - use the energy source of one - and two-cylinder engines operating on the two stroke cycle. The main advantage of the two-stroke cycle is higher liter capacity because the stroke in such engines occurs twice as often in comparison to the four-stroke ones for the same frequency of rotation of the crankshaft. The lack of valve gear determines the ease of construction and maintenance of the engine, as well as small dimensions. But the main disadvantages of these engines, limiting their widespread distribution, are low efficiency of their work and higher emissions. To address this problem the present study is aimed.

2. Prerequisites and means for solving the problems.

The principal difference between the two-stroke engines and the four-stroke ones is the lack of intake and exhaust valves. Gas exchange in the cylinder is carried out through the intake, exhaust and inlet ports located on the surface of the cylinder. For the process of gas exchange the volume of the crank chamber, which is filled with a mixture of gasoline and air through the inlet port when the engine works, is also used. The processes of exhausting through the outlet port and filling the cylinder with a fresh mixture from the crank chamber via the scavenging ports occur simultaneously. With the high gas exchange a portion of the fuel gets into the exhaust system and further into the atmosphere, which explains the high

fuel consumption and high emissions. If you consider that the main modes of two-stroke gasoline engine run rich mixtures with excess air ratio $\alpha = 0.7$ to 0.9, the loss of fuel due to this are even greater.

The most effective way of solving this problem is the use of direct injection of gasoline with electronic control that allows you to bring the performance of two-stroke engines in terms of efficiency and toxicity closer to four-stroke ones [1, 2].

But direct injection leads to significant complexity of the design and increases the cost of the engine, which is unacceptable for engines intended for light motor cycles and also small machines. The most promising is the improvement of the two-stroke engine with the crank chamber scavenging, providing substantial or complete exclusion of the direct loss of fuel by blowing. In this case, there is one of the main advantages of these engines - simple design.

One of the recognized and effective ways to substantially reduce the loss of fuel in the exhaust system in the two-stroke engine is the stratification of the fuel-air charge in the intake and gas exchange [3, 4].

3. The problem solution

Our research is aimed at improving the two-stroke gasoline engine, which is implemented by the layer-by-layer mixing processes due to separate admission of air and gas at inlet and purge.

At the first stage of work, when we used the carburetor for mixing air and gasoline, we investigated the scheme of the stratified mixture formation, shown in (Fig. 1,a). Fuel 1 is mixed with the air 2 in the lower chamber of the carburetor and enriched air-fuel mixture 7 fills the crank chamber. In addition, the air through the upper chamber of the carburetor, a lateral recess in the bottom of the recess of the piston 3 at the same time fills the volume of purge channels. By the end of the inlet when the piston is at TDC, the volume of purge channels is filled with clean air, and the subpiston space and the crank chamber are filled with enriched mixture.

With the motion of the piston down to BDC (Fig. 1,b) compression of stratified charge occurs. After opening of the exhaust port, the exhaust gases 4 rush into the exhaust system. Since the opening of the inlet ports the displacement of the exhaust gas is carried out by a flow of clean air 5 and poorer mixture 6. At the final stage of the purge the cylinder is filled with a rich mixture 7 from the crank chamber.

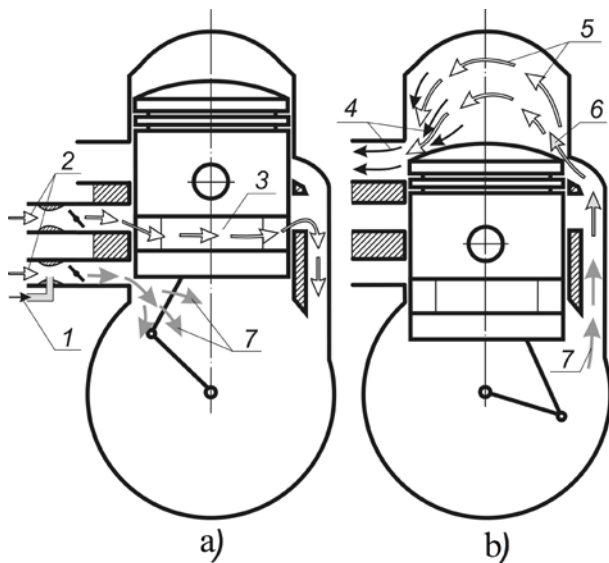


Fig. 1. Diagram of layered entering fresh charge:
1 – admission fuel; 2 – admission air; 3 – the recess in the piston;
4 – exhaust gases; 5 – clean air;
6 – poorer mixture; 7 – rich mixture

By physical modeling of the process of mixture formation with application of the "tracer" gas CO_2 instead of fuel we estimated the distribution of mixture in the purge volume of channels on the excess air ratio in the local points of each channel. The results of the experiments (Fig. 2) confirm the significant stratification of the fresh charge by the beginning of the purge.

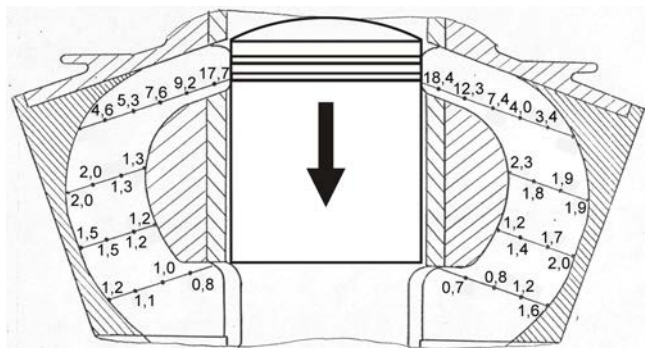


Fig. 2. The stratification of the mixture in the venting channels:
0,8; 1,2; 5,3 etc. – the excess air ratio
at the local point of the scavenging channel

At the top of the channels adjacent to the vent windows the poorer mixture with $\alpha = 12-18$ units is concentrated. The middle part of the channel is filled with a poor mix with $\alpha = 2,0-2,3$. In the lower part of the channel mix is also poor with $\alpha = 1,2$ to $1,5$. And only at certain points of the portion of the channel immediately adjacent to the crank chamber, the mixture is somewhat enriched.

The data obtained in the first stage of this work allowed for us to make a conclusion about the prospects of this direction of improving the two-stroke engine and proceed to the second stage.

At this stage, the study of the efficiency of the stratification of the fuel-air charge in two-stroke engine continues with the use of phased fuel injection using electromagnetic injectors. In this study we look at two different schemes of engine operation with a stratified mixture formation and injection.

One of the schemes of the two-stroke engine with a bundle of fresh charge is characterized by the presence of cumulative chambers 1, electromagnetic injectors 2 and window 4 in the piston (Fig.3).

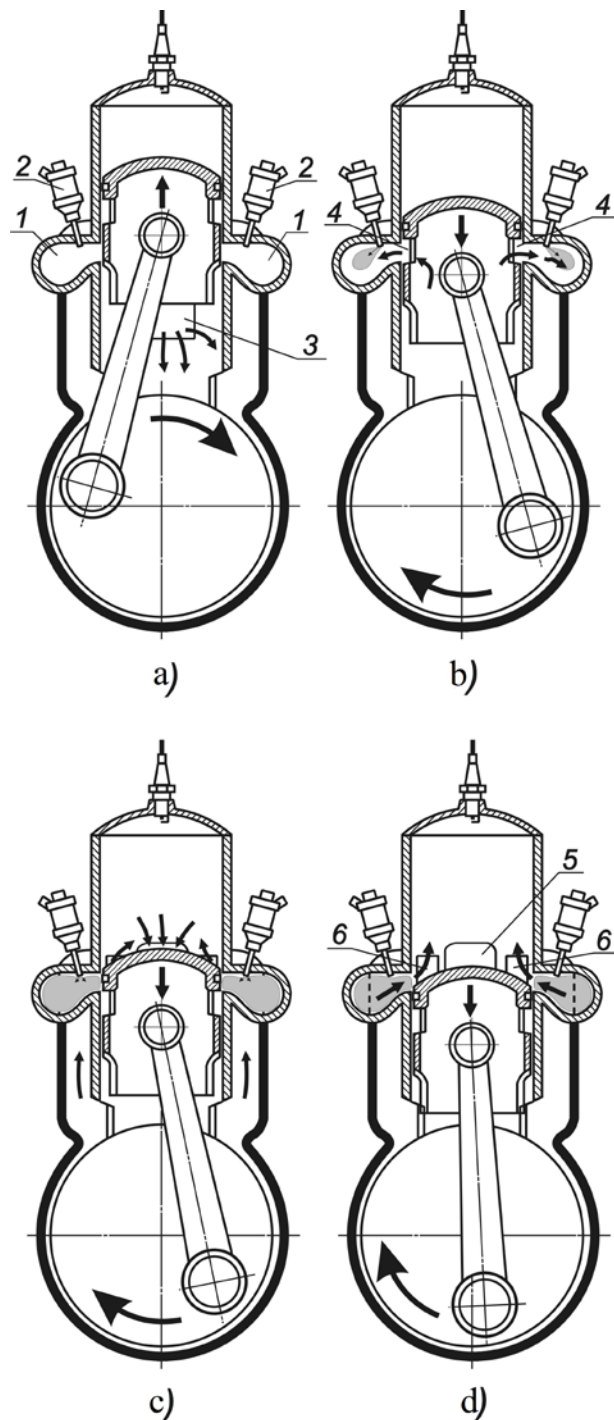


Fig. 3. The scheme of work of the engine with the stratification of the fresh charge and the cumulative chambers:
1 – cumulative chambers; 2 – electromagnetic nozzles;
3 – the inlet port clean air; 4 – ports in piston;
5 – the outlet port; 6 – vent windows

The cumulative chambers are filled with clean air with the simultaneous injection of the volume of gasoline from the two electromagnetic nozzles, which leads to the formation of the enriched fuel-air mixture in the cells in the process of gas exchange. Purging is carried out with clean air, and at the time it is completed the air-fuel mixture is supplied to the cylinder from the cumulative chambers.

The diagram of the engine is shown in (Fig. 3). When the piston moves up, the clean air fills the crank chamber through the open inlet port 3. At the same time the fuel-air mixture compresses above the piston (Fig. 3,a). At the end of the compression, the electric spark ignites the mixture and the piston moves down during

the power stroke and it closes the inlet port and compresses the air in the crank chamber, and the window in the piston opens the inlet to the chamber and pre-compressed air is supplied (Fig. 3,b). At the time of the opening of the inlet the fuel supply is started by the electromagnetic injectors 2. The further downward movement of the piston closes the inlet openings of the cumulative cameras, opens the outlet port 5, and then the scavenging ports 6. Graduation occurs, purging and filling the cylinder with clean air (Fig. 3,c). If necessary, the fuel supply may continue until the opening of the piston upper edges of the cell Windows. At the time of completion of the purge enriched mixture from the shaped chambers under pressure exceeding the pressure in the cylinder, rapidly fills the cylinder (Fig. 3,d). Thus, the cleaning of the cylinder from the exhaust gas is carried out by clean air, but at the end the gas cylinder is filled with the enriched air-fuel mixture that, when mixed with clean air, forms a working air mixture suitable for combustion. With proper selection of the geometric shape and dimensions of the shaped chambers, as well as the geometry of the connecting channels between the chambers and the cylinder of the engine, it is possible to significantly reduce the loss of fuel into the exhaust system by blowing.

The proposed scheme of gas exchange with the use of cumulative cameras was implemented on the experimental engine (Fig. 4), which shows the placement of electromagnetic injectors in jet chambers.



Fig. 4. Two-stroke engine with stratified mixture formation and the cumulative chambers:

- 1 – pressure-sensitive detector; 2 – injector rail;
- 3 – electromagnetic nozzles;

Preliminary results of the bench tests confirmed the efficiency of the proposed scheme of gas exchange and experimental design of the engine.

In another diagram (Fig. 5) a supply of clean air in the crank chamber 8 is carried out via the purge channels 2 connected to the outside of the cylinder with two air inlet channels in its upper part 1. In the cylinder there is no inlet mixing box.

The process of mixing is carried out in the crank chamber 8 of the fuel supply via the electromagnetic nozzles 5 and is governed by the time and duration of injection.

At the inlet (Fig. 5,a), when the piston 6 moves upwards, since the closing of the cylinder inlet ports 7 in the crank chamber 8 creates a negative pressure transmitted through the purge 2 channels in the intake air channel 1. The air from the atmosphere rushes into the general air channel 4, through the throttle valve and the 3 in the air conduit 1 and through the purge 2 channels fills the crank chamber 8.

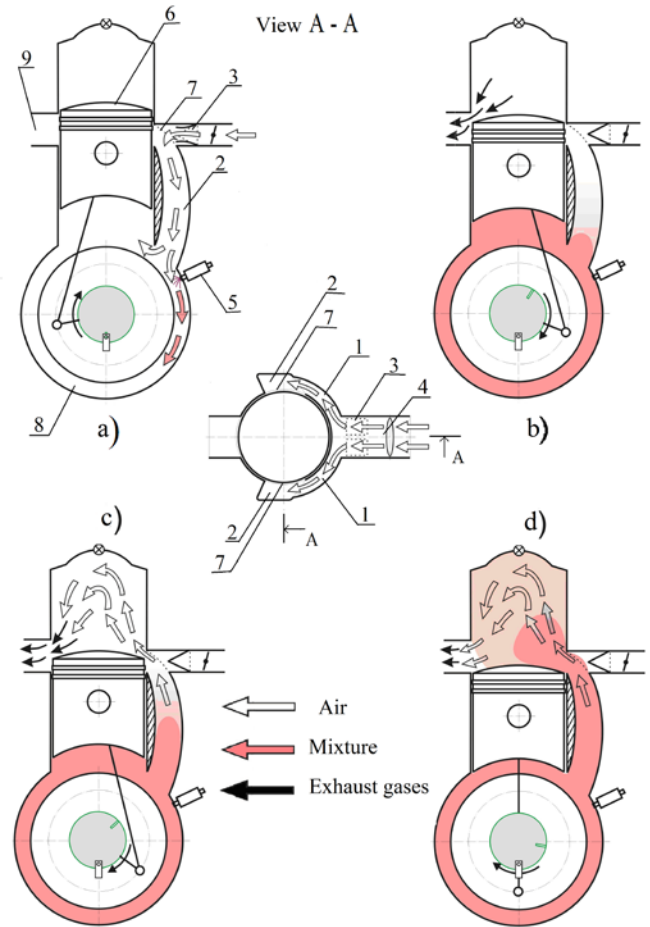


Fig. 5. The scheme of the layer-by-layer mixing and venting in the two-stroke engine with fuel injection:

- 1 – air intake channels; 2 – the purge channels;
- 3 – return plate valve; 4 – air channel;
- 5 – nozzle; 6 – piston; 7 – admission port;
- 8 – crankcase; 9 – exhaust port

When the piston moves down (Fig. 5,b) the crank chamber is pressurized, which locks the reverse valve plate 3. At the beginning of the purge (Fig. 5c), after opening of the inlet ports 7, the displacement of the exhaust gas from the cylinder is carried out by a flow of clean air from the purge channel 2 to the outlet port 9. When the piston moves closer to the bottom dead point, the fuel injected by the injector 5 in the crank chamber 8 is mixed with the purge air and coming through the purge 2 channels into the cylinder, it completes the process of purging (Fig. 5,d). Purging is carried out mostly by clean air, which significantly reduces the direct loss of fuel. This leads to the increased fuel efficiency of the two-stroke engine and reduces environmental pollution. To assess the stratification of the mixture at the inlet and purge the physical modeling of stratified mixture formation is conducted (Fig. 6). In this case, the nozzle emits carbon dioxide instead of gasoline.

Special gas samplers perform stroboscopic sampling of the gas-air mixture from the crank chamber, from different points of the volumes of purge channels and the combustion chamber. Gas analysis of the samples allows to determine the percentage of carbon dioxide in the mixture with air and translate this content calculated by the nomograph in the excess air ratio α . Thus the distribution and qualitative composition of the mixture by gas exchange is determined and is estimated to reduce the losses of fuel into the exhaust system by blowing.



Fig. 6. Two-stroke engine with stratified mixture formation and injection gasoline

1 – air intake channels; 2 – the purge channels;
3 – return plate valve; 4 – throttle;
5 – nozzle; 6 – connecting pipe gas sampler;

4. Results and discussion

Bench testing of the two-stroke engine with a carburetor to mix fuel and air shows high efficiency of the stratified mixture formation. The stratification of the fresh charge intake has a positive effect on engine performance parameters and reduction of the toxicity of exhaust gases (Fig 7.). In the stratified mixture formation in an economical and power state the total composition of the mixture at the inlet is more depleted compared with the conventional scheme of gas exchange. The difference is 0.15 to 0.20 units on the excess air ratio. In this way, the efficiency is improved by 10-15%. The engine in the power mode is accompanied by fuel economy by 15-20%.

This effect is mainly a confirmation of the fact that by blowing, the direct losses of fresh charge are the clean air and a portion of the lean mixture. The composition of the combustion mixture in the cylinder is more enriched than the calculated one at the intake by the expenditure of fuel and air. Therefore, the maximum power and lowest fuel consumption in the engine with stratified mixture formation are achieved with leaner mixtures.

Reduction in the direct losses of the fuel mixture by blowing is reflected in the reduction of emissions of unburned hydrocarbons CH from the exhaust gases by 25-30%. The content of carbon monoxide CO on the power mode is reduced by 1.5 times. The indexing of the experienced engine showed the best stability of the workflow, assessed by the degree of irregularity of cycles δ , the value of which at the economical mode is 5-6%, while the production engine irregularity amounts to 12-13%, ceteris paribus.

Thus, layer-by-layer mixing in the two-stroke gasoline engines is an effective way to improve performance of their work.

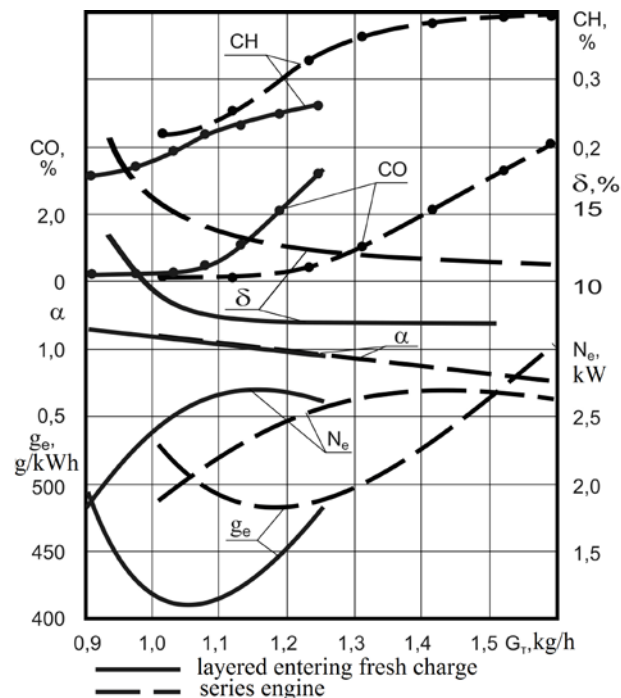


Fig 7. The characteristic of the fuel supply:
 $n=3400 \text{ min}^{-1}$, $\phi_0=0,40$

5. Conclusion

The main direction of improvement of the two-stroke gasoline engines is a significant reduction in direct losses of fuel during scavenging. The rapid development of modern electronic system fuel injection allows you to create different schemes of work for the two-stroke engine with stratified mixture formation, in which the scavenging is carried out with clean air. The efficiency of such engines by the values of fuel consumption approaches the efficiency of the widespread four-stroke engines. This retains the main advantages of the two-stroke engines - simple design, high power density, small size.

6. Literature

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EXPERIMENTAL DETERMINATION OF DISTILLATION CURVES OF ALCOHOLS/GASOLINE BLENDS AS BIO-FUEL FOR SI ENGINES

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Abstract: The use of oxygenates as a gasoline blend is one of production methods for high octane gasoline and reduction of harmful compounds amount in exhaust gas. Distillation curves are presented for single monohydroxy alcohols and gasoline mixtures containing 3-15% by volume of ethanol, 1-butanol and i-butanol (2-methyl-1-propanol) which can be obtained from renewable sources. The influence of various concentrations of these alcohols on the distillation characteristics of base gasoline were investigated. Most alcohols form mixtures with gasoline exhibiting near-azeotropic behavior near its boiling point that significantly affect the shape of the distillation curves. In addition, distillation curves for a variety of dual-alcohol blends are presented, containing 10-20% by volume of two alcohols.

Key words: DISTILLATION, GASOLINE, BLENDS, ETHANOL, 1-BUTANOL, I-BUTANOL

1. Introduction

Constantly increasing fossil fuel prices led humanity to the search for alternative biofuels. Biofuel is a type of fuel which is derived from biological sources. The use of alcohols as blends of gasoline for ignition engines is very actual problem which is applied in the fields of fuel production and its applications. By the late 1990s ethanol had emerged as the predominant alcohol for blending with gasoline and flexible fuel vehicles were designed instead for ethanol blends. Currently ethanol is the most commonly used biofuel worldwide [1–4], however there is an increased interest in other alcoholic fuels especially 1-butanol and isobutanol [5, 6]. The fuel mixtures with these oxygenates in some cases have better properties than the regular gasoline. As oxygenates ethanol and 1-butanol are very usable mainly because they easily can be produced from biomass [7–9]. In addition to that they also meet the essential requirements for oxygenates which can be described as a high octane number, low toxicity and good performance [10]. Currently there is a great attention for the iso-butanol biosynthesis from the biomass of as well as the applications of this alcohol as a gasoline blend [11–14]. Moreover, the use of bio-based products in fuels is a strategic government resolution in most European countries.

Alcohols consists from hydrogen, carbon and oxygen - these compounds are oxygenates. Various researches have shown that the use of oxygenates not only increases combustion efficiency but also reduces the amount of CO, NO_x and unburned fuel in the exhaust gas by about 40% thus reducing smog [15, 16]. The first-ranked monohydroxylic alcohols have high octane number which leads to the fact that they not only improve the combustion of gasoline, but also enhance the octane properties of its mixture. In the EU and US gasoline blends with 10% ethanol by volume (E10) are common, however one drawback ethanol has is that it is a very hygroscopic liquid thus it quite easily absorbs water from the surroundings. To minimize the risk of water-induced phase separation in the ethanol-gasoline blends, anhydrous ethanol is blended into gasoline at the terminal prior to shipping to retail gas stations.

With the development of second-generation biofuels it is possible that alcohols other than ethanol will enter the market. The usage of gasoline and 1-butanol mixtures in the spark ignition engines increase the detonation resistance, achieve higher gas compression and higher thermodynamic efficiency compared with the pure gasoline suggesting that 1-butanol as a higher quality oxygenate is more efficient than ethanol [17–21]. In addition, there is an increasing interest in gasoline that contains alcohol concentrations different from currently applied worldwide.

Prior to such an introduction, a detailed knowledge of the physical and chemical properties of alcohol-gasoline blends is required and their performance in engines and impact on emissions needs to be tested. It was found that gasoline with 10% ethanol blends reduce air polluting substances about 28%. The

concentration of the CO emissions decreases as the ethanol content increase at the fuel blend due to the oxygen presence in the fuel molecules [22]. The investigation of a gasoline and 10% ethanol blend by volume confirm that a mixture can be used without engine modifications. One of the most important characteristic of gasoline it's volatility which is assessed by fractional composition of the distillation. When ethanol and methanol are blended with gasoline a positive azeotropic mixture is formed [23–26]. The ASTM D86 (EN: ISO3405) distillation curve represents the temperature of the fuel vapor versus the volumetric fraction of the fuel sample distilled. In a mixture, an azeotrope would manifest itself as a flat portion of the distillation curve at the boiling temperature of the azeotrope [23, 24]. While this is a necessary condition to verify that an azeotrope is present it should also be noted that a flattening of the curve can also be observed for azeotropic mixture dominated by one component. The fractional distillation composition is expressed in two ways: when the monitored temperature (°C) of distilled gasoline fractions and percentage (%) of the total gasoline up to temperature (E70, E100, E150) according to the standard ASTM D86 (EN: ISO3405) BS EN 228: 2013.

While knowing the temperature t_{10} (°C) of analyzed gasoline's 10% distillation, an estimation can be done to calculate the lowest ambient temperature t_p (°C) at which cold engine can be started [27]. This is calculated by using formula 1:

$$t_p = 0,5t_{10} - 50,5 \quad (1)$$

The lowest ambient temperature t_g (°C) at which the vapor jams can be formed is calculated according to the formula 2:

$$t_g = 2t_{10} - 93 \quad (2)$$

The distillation curve provides insight into the boiling range of the fuel and can be used to predict its operation in engines. The low temperature region of the curve (up to 70 °C) E70 can be related to the startup of the engine, engine warm-up, evaporative emissions, and vapor lock (for carbureted vehicles). The middle range of the curve (70-100°C) E100 can be related to warm-up, acceleration, and cold-weather performance, while the top range of the curve (above 150°C) E150 relates to propensity for combustion deposits and oil dilution [28]. Although there is substantial interest in the potential use of different alcohols in blends with gasoline, the published database concerning distillation curves for alcohol-gasoline mixtures is incomplete. Some distillation curves for the mixtures of ethanol [29–30] and butanol blends with gasoline have been reported. The distillation curves of small quantity isobutanol blends have not been reported and only about 30%, 50% and 70% concentration of isobutanol mixtures with gasoline are reported [31]. Also, there is little data available concerning the behavior of mixtures of two or more alcohols (e.g., ethanol and butanol) with gasoline. In order to have a better understanding of the volatility of alcohol-gasoline blends and their utility as motor vehicle fuels,

distillation curves for gasoline and blends of 3-15% ethanol, 1-butanol and i-butanol with gasoline were determined (Table1). In addition, we present distillation curves for dual-alcohol blends containing 10-20% by volume of two alcohols ethanol and 1-butanol, or ethanol and i-butanol, as well as 1-butanol and i-butanol in gasoline (Table 2). The applied of monohydroxylic alcohols, ethanol, 1-butanol and i-butanol (2-methyl-1-propanol) and their compositions in gasolines production were studied in this work. The influence of various concentrations of these alcohols on the main characteristics of base gasoline were investigated.

2. Prerequisites and means for solving the problem

The fractional distillation composition is expressed in two ways: when the monitored temperature (°C) distilled of 10, 50, 90, 97% share of gasoline fractions and percentage (%) of the total gasoline, up to 70, 100, 150°C temperature (E70, E100, E150). According to the requirements of ASTM D86 (EN:ISO 3405) BS EN 228:2013 standard, distillation characteristics must comply with the E70 - 22-50% by volume; E100 - 46 to 71% by volume, E150 - not less than 75% by volume. In this study the following stocks were used:

1. Base gasoline from ORLEN Lietuva (non-commercial, oxygenate-free, without additives)
2. Dry ethanol from Biofuture and "MG Baltic"
3. 1-Butanol and i-butanol from Sigma-Aldrich.

Distillation points of gasoline with alcohols were measured by a computer-controlled „OptiDist” apparatus (PAC) (Figure 1).



Fig. 1 „OptiDist” apparatus.

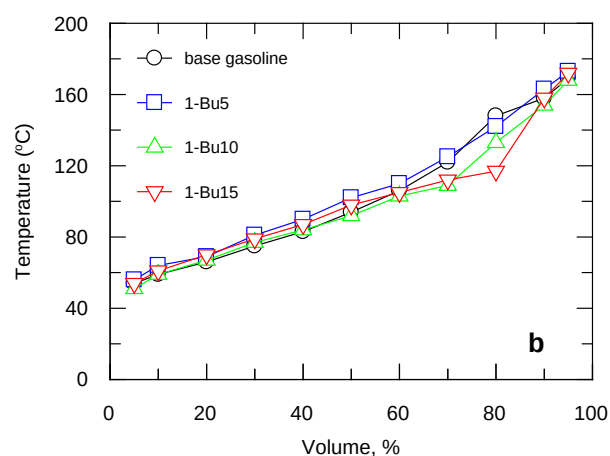
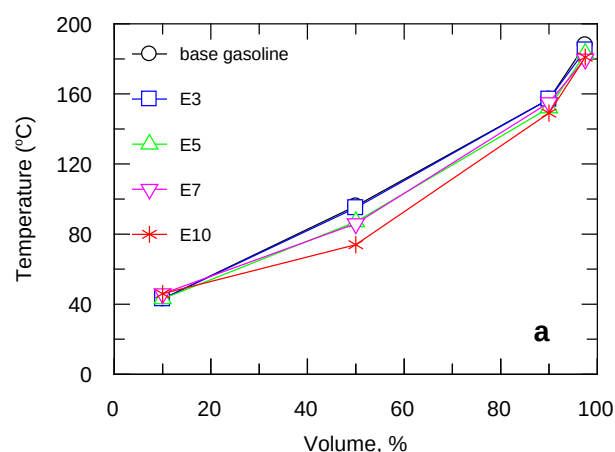
3. Results and discussion

The fuel distillation curves reveal information on its opportunities to be deployed in the ignition engines. Various mixtures of base gasoline with single-alcohol blends and dual-alcohol blends of bio-fuels were investigated. The alcohol content in gasoline was constantly adjusted based on the oxygen quantity in the fuel, therefore mixtures with small quantities of alcohol were selected. The composition of all the mixtures is summarized, respectively in Table 1 and in Table 2.

Table 1: Single-alcohol blends with base gasoline.

Mixture	Gasoline (% by vol.)	Ethanol (% by vol.)	1-Butanol (% by vol.)	i-Butanol (% by vol.)
E3	97	3		
E5	95	5		
E7	93	7		
E10	90	10		
1-Bu5	95		5	
1-Bu10	90		10	
1-Bu15	85		15	
i-Bu3	97			3
i-Bu5	95			5
i-Bu7	93			7
i-Bu10	90			10

The distillation curves of oxygenate-free gasoline A 92 with different volumes of ethanol (0%, 3%, 5%, 7% and 10%) are shown in Figure 2a; mixtures with butanol-1 (0% 5%, 10%, 15%) are shown in Figure 2b and mixtures with i-butanol (0%, 3%, 5%, 7%, 10%) are shown in Figure 2c.



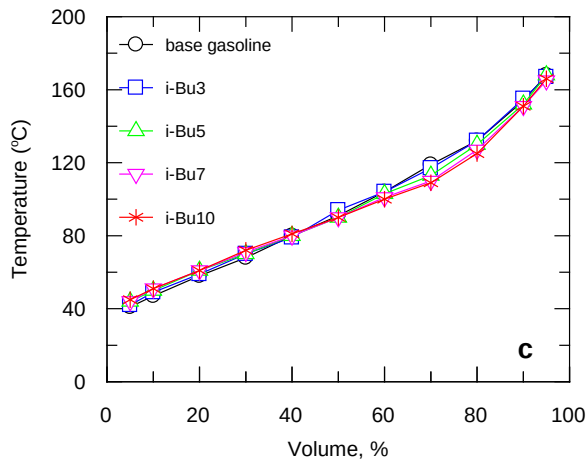


Fig. 2 Distillation curves of base gasoline mixtures with ethanol (a), 1-butanol (b) and i-butanol (c).

From the figures above it is noted that distillation temperatures of gasoline mixtures with ethanol or 1-butanol increase more slowly than the base gasoline from T20 to T70 and T100 to T160. The distillation curves for i-butanol blends are very similar to the ones of base gasoline. These observations indicate that ethanol and 1-butanol can form an azeotropic mixtures with the gasoline hydrocarbons. Investigation of (t_{10}) temperature trend shows that cold engine starts in a temperature of -55°C . From (t_{50}) temperature it is seen that engine has a better warming up properties as well as a reduced fuel consumption. This improves engine dynamism, e.g. facilitation of the transition from one mode to another. Temperatures t_{90} , and $t_{97,5}$ affect the economy of the engine and friction. The higher the temperature (t_{90}), the higher fuel quantity is consumed by the engine [27]. It was found that higher ethanol blend temperatures (t_{10}) and (t_{50}) slightly decrease (shown in the mixtures Fig. 2a). The increased concentration of 1-butanol (Fig. 2b) shows that the temperatures (t_{10}) and (t_{50}) are slightly increasing, but for i-butanol Fig. 2c the temperatures (t_{10}) and (t_{50}) are very similar to the trends of base gasoline. During the test distillation curves, the results showed that the increased oxygenate concentration in the mixtures of different alcohols decreases the temperatures (t_{90}), and ($t_{97,5}$).

The different composition of ethanol, 1-butanol and i-butanol were used as a dual-alcohol blends for gasoline. Full data of gasoline blends presented in Table2.

Table 2: Dual-alcohol blends with base gasoline.

Mixture	Gasoline (% by vol.)	Ethanol (% by vol.)	1-Butanol (% by vol.)	i-Butanol (% by vol.)
E5, 1-Bu10	85	5	10	
E10, 1-Bu5	85	10	5	
E10, 1-Bu10	80	10	10	
E3, i-Bu7	90	3		7
E5, i-Bu5	90	5		5
E7, i-Bu3	90	7		3
1Bu10, i-Bu5	85		10	5
1Bu10, i-Bu7	83		10	7
1Bu10, i-Bu10	80		10	10

The results showed (Figure 3a) that the distillation curve of ethanol–1-butanol–gasoline from T15 to T60 is higher than gasoline, and from T140 is below. Only the ending distillation has a very similar look compared to the basic gasoline distillation curve, and t_{10} , t_{90} , $t_{97,5}$ are the same as those of gasoline, while t_{50} is lower. The distillation of mixtures ethanol–i-butanol–gasoline from T40 to T140 takes place at lower temperatures than gasoline, but distillation from the beginning to the T40 and from T140 is same again (Figure 3b). In conclusion t_{10} , t_{90} , $t_{97,5}$ are the same as those of gasoline, only t_{50} is lower.

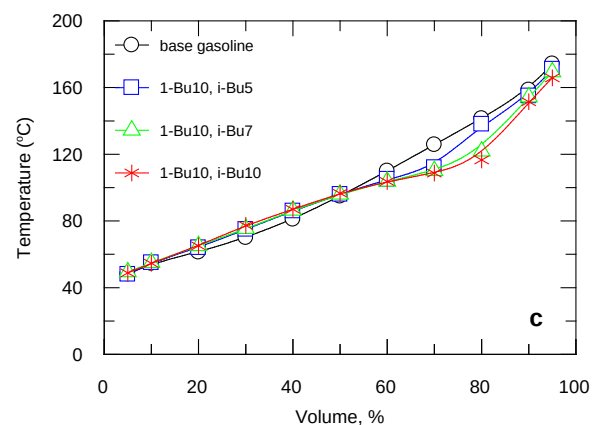
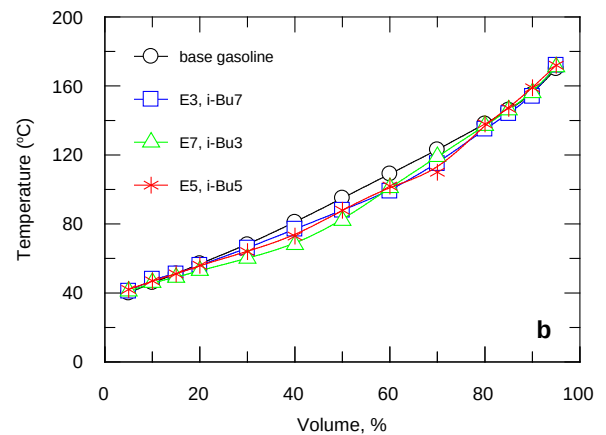
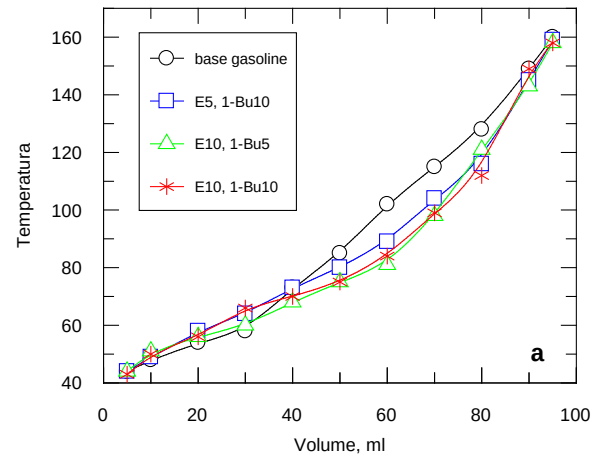


Fig. 3 Distillation curves of base gasoline blends with ethanol ÷ 1-butanol (a), ethanol ÷ i-butanol (b) and 1-butanol ÷ i-butanol (c).

For the gasoline mixtures shown in Figure 3c distillation curves from T50 to T90 are located higher than base gasoline distillation curve. Based on T90 we can note that the mixtures boil at a lower temperature than the base gasoline and the temperature of distillation depends from the concentration of alcohols. In

conclusion the mixtures in comparison with base gasoline have temperatures of t_{10} , t_{90} , $t_{97.5}$, while t_{50} temperatures are the same.

4. Conclusion

As conclusions of the experiments performed and described above the following points are made:

1. Experimental results showed that gasoline mixtures with both ethanol and butanol have the same tendency bend at the points of their blended alcohol boiling temperature – this is the feature of azeotropic mixtures.
2. The results showed that a gasoline blend with two different alcohols provides better influence towards engine operability than a gasoline blend containing higher concentration of one alcohol (a gasoline blend with 15% butanol versus a gasoline blend with 10% ethanol and 5% butanol). This is based on distillation trend in a region of T20-T70 results.
3. Currently worldwide ethanol is the most commonly used oxygenate, however positive experiment results with blends of two different alcohols or higher-chain alcohols such as butanol or i-butanol draw a positive view for the future as these alcohols could be used more advantageously than only ethanol itself.

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FUNCTIONALITY OF THE STIRLING ENGINE WITH NONCONVENTIONAL MECHANISM FIK

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Abstract: At the design of heat engines like Stirling engines are, it is possible to use not only classical crank mechanisms, but also non-conventional mechanisms. Engines with non-conventional mechanisms may have several advantages when used in practice, but the design calculation model is often more difficult [6]. This paper deals with the design of the measuring system for the measurement and diagnostic of basic parameters of the thermal cycles in Stirling engine with nonconventional FIK mechanism, which was solved in the project VEGA: Nonconventional engine FIK – Stirling. The measuring system concept is applied to a special type of piston machines with swinging mechanism that is used in this instance of a Stirling engine type. Stirling engines are currently usually used as a drives of machines for production of the electricity [2]. There are several types of mechanisms, which are suitable by design for use in a Stirling heat engine. FIK mechanism is a swinging system, which is characterized by circular motion of the central point of the swinging plate during the rotation of the shaft. Proposed measuring systems allows to confirm the functionality of the structural design of Stirling engine with a swinging plate and examine the thermodynamic phenomena conducted in the engine cylinders.

Keywords: FIK MECHANISM, SENSOR, DESIGN, MEASUREMENT

1. Introduction

Stirling engines are currently used in the production of electricity. There are several types of mechanisms, which by design are suitable for use in a Stirling heat engine [1]. In the case of prototype devices of Stirling engines, which verifies the functionality of the machine is necessary to determine the progress of thermodynamic phenomena taking place at the premises of the rollers and therefore need to create and implemented on a machine measuring system [3] [4].

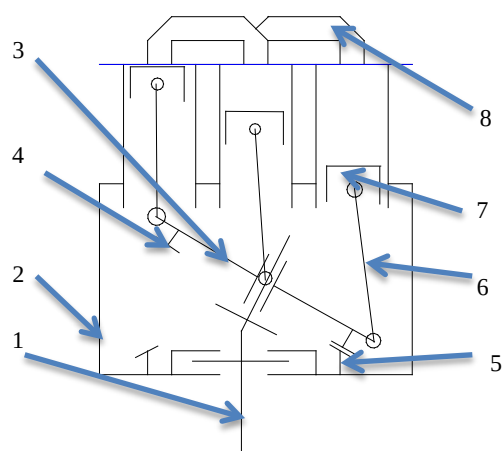


Fig.1 Schematic model of the nonconventional design FIK:
1- Crankshaft, 2- crankcase, 3- swinging plate, 4- bevel wheel (part of swinging plate), 5 bevel wheel (part of crankcase), 6- ball joint segment, 7- piston, 8- head cylinder and regenerator pipe

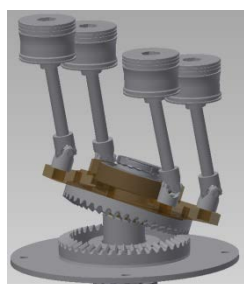


Fig.2 Mechanism FIK virtual model without cylinders and head valves

2 Measuring system

The measuring system (Fig. 3, Fig. 4) consists of pressure and temperature sensors, RPM sensor and the cooling cylinder is provided through non-return valve inlet pressure [5].

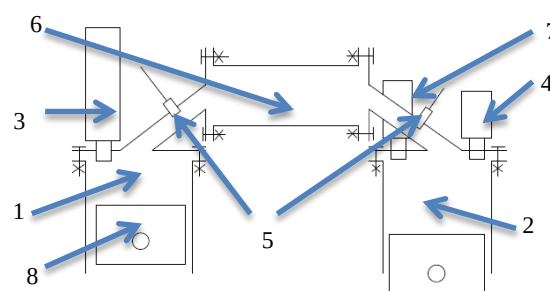


Fig.3 Schematic representation of the measurement system in head cylinder: 1- Heat cylinder, 2- cooled cylinder, 3- Heat cylinder pressure sensor, 4- Cooled cylinder pressure sensor, 5- Temperature sensor, 6- regenerator, 7- Non-return valve, 8- Piston

RPM sensor (Fig.4) is connected with a rubber coupling on the output shaft. This sensor is located at the bottom of the flywheel [5].

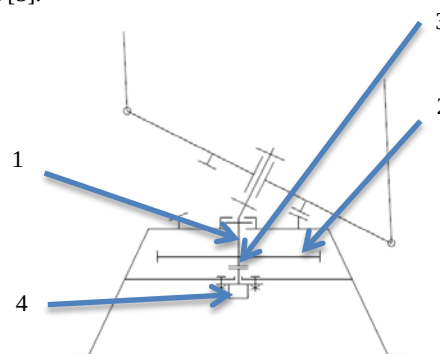


Fig.4 RPM sensor: 1- Output shaft (crankshaft), 2- flywheel, 3- rubber coupling, 4- RPM sensor

3 Virtual and real models

The exact location (Fig. 5) of each sensor was designed in CAD software. In this case, the pressure sensors are located in the cylinder heads and the thermocouples are attached with connections screws in the interconnecting pipes. Supercharging system (Fig. 6, 7) consists of non-return valve and pneumatic distributor [5].

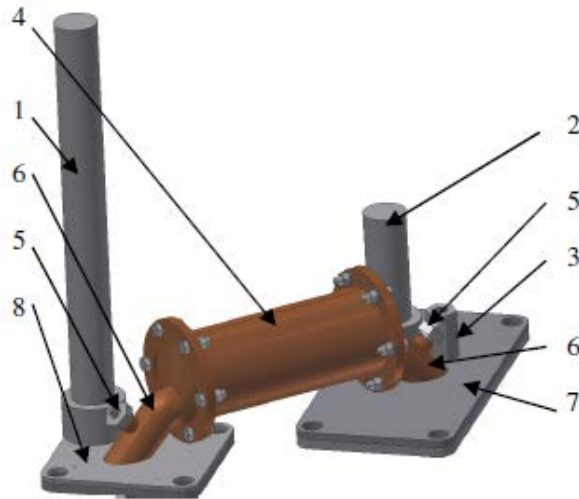


Fig.5 Virtual model of head valves, interconnection pipes (one pair) and regenerator pipe with sensors: 1- Heat cylinder pressure sensor, 2- Cooled cylinder pressure sensor, 3- Non- return valve (Supercharging system), 4- Regenerator pipe, 5- Temperature sensor connection 6- Interconnection pipe, 7- Cooled cylinder head valve, 8- Heat cylinder head valve

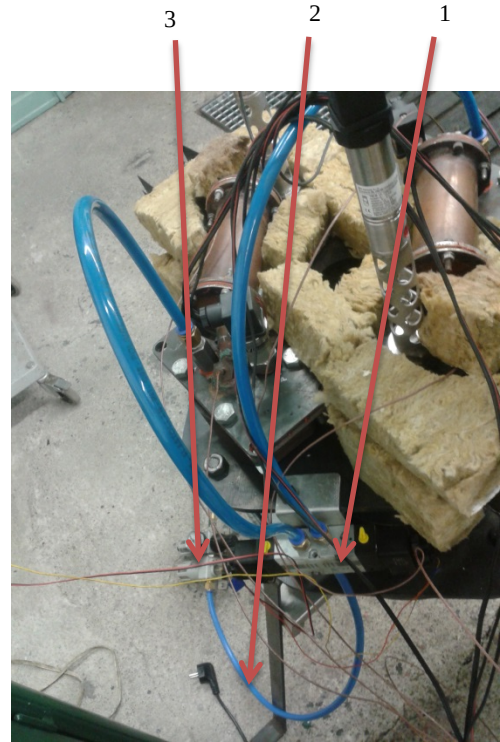


Fig.7 Part of supercharging system: 1- Pneumatic distributor, 2- Rubber hoses (outlet from the compressor), 3- Water separator- oil lubricator

RPM sensor (Fig. 8) is attached to the aluminum console that is fixed to the engine block.

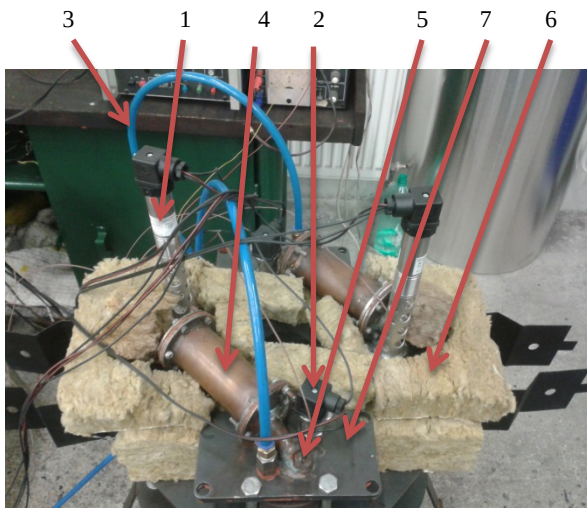


Fig.6 Real model of head valves interconnection pipes and regenerator pipe with sensors: 1- Heat cylinder pressure sensor, 2- Cooled cylinder pressure sensor, 3- Pneumatic rubber hose (Supercharging system), 4- Regenerator pipe, 5- Temperature sensor (thermocouple) connection 6- Thermal isolation, 7- Cooled cylinder head valve

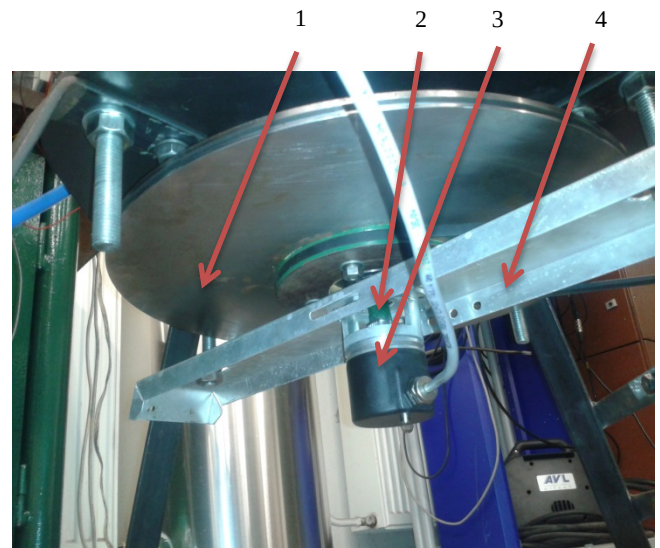


Fig.8 Design of RPM sensor connection: 1- Flywheel, 2- Rubber coupling, 3- RPM sensor, 4- Aluminium console

4 Static pressure test

Initial test, which was performed on the engine was tightness test. In this case, the tightness was tested by pressurizing of one pair of interconnection pipe to 3.4 bar and it was measured time for which the pressure drops to atmospheric pressure. The pressure was measured simultaneously by two sensors, one to be heated cylinder heads, on the other side of the cooled cylinder. Turbocharging system is integrated into the head of cooled cylinder and therefore the measured values between the sensors slightly delayed onset of pressure in the space heating of the cylinder (Fig. 9). For measurements we used a measurement card NI USB 6211. The measured values are processed in LabVIEW environment.

Initial conditions:

- time range 30 second
- pressure range 0- 10 bar
- output voltage signal 0-5 Volts

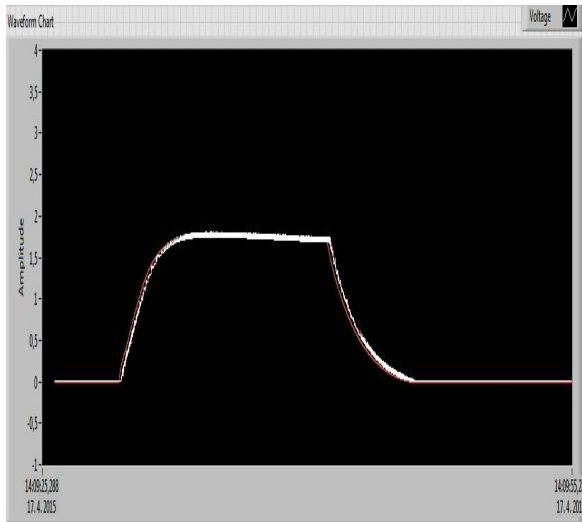


Fig.9 Progress of pressure: cooled cylindersensor- red curve, heat cylinder white curve

To measure the of pressures was created a simple enviroment (measuring system) in LabVIEW (Fig. 10).

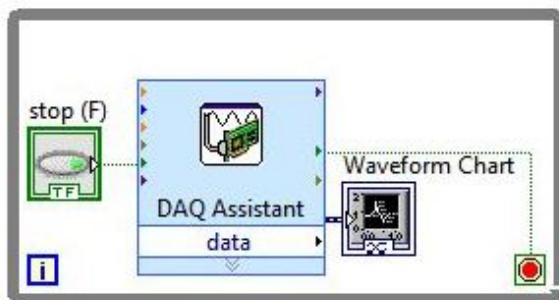


Fig.10 Virtual enviroment in LabVIEW

5 Results

The measured values were exported to Microsoft Excel. As can be seen in Figure 10 the system cannot keep the pressure even at still supercharging.

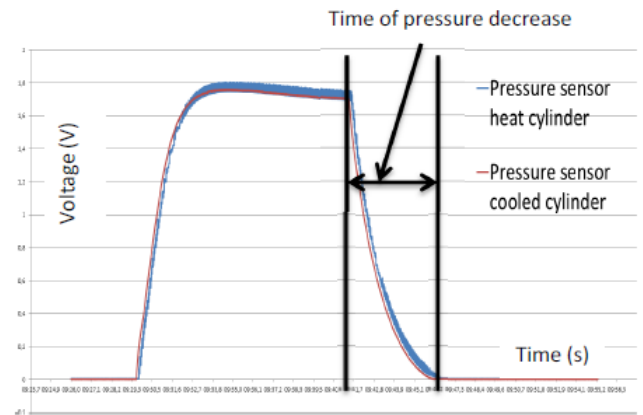


Fig. 11 Progress of pressure in excel

The time of pressure decrease is about 4.5 second. Measurement was done several times on both pairs of interconnection pipes. Waveforms of pressure drop in the two pairs did not differ significantly and their values when calculating of the average values of time decrease were negligible difference.

6 Conclusion

Problems of Tightness in engine FIK can be divided into two parts:

- tightness of Solids engine (this is a leaks of air between the connected parts mainly cylinder heads, cylinders and interconnecting pipelines, regenerator and interconnecting pipelines and integrated sensors),
- tightness of the moving parts of the engine (this is a leaks of air between the cylinder and piston).

The tightness of brazed joints (it is a test using soapy water) showed a small leak incurred in the manufacturing process. The tightness of brazed joints (it is a test using soapy water) showed a small leak (Fig. 12) incurred in the manufacturing process. This problem can be solved by re-melting the solder in the assembled state of the engine.

The tightness between cylinders and pistons can be improved by using suitable piston rings.

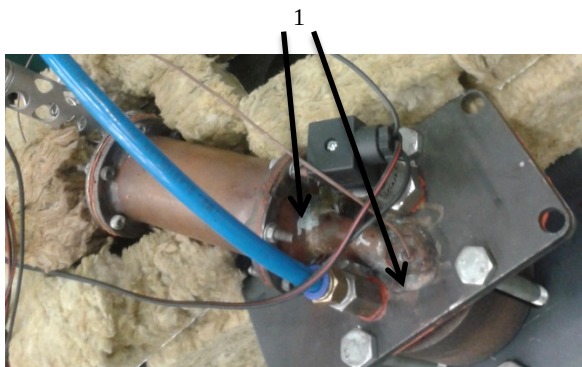


Fig.12 Head valve with interconnection pipe system: 1- soldered joints

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TESTING THE EFFICIENCY OF INTERNAL COMBUSTION CATALYSTS POSITION MODEL.

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Abstract: In order to determine the effectiveness of the catalytic coating applied to the engine valve, a study model. For this purpose built test stand made of elements of the real engine. The catalyst temperature was controlled by electric heating. The position of the working gas fumes were collected with a spark-ignition engine, idling. Performed a simultaneous measurement of gas composition at the inlet to the model and the output from the system. The study was conducted at a constant flow rate and variable temperature exhaust gas catalyst system. The results of the study are presented in the form of comparative charts. It has been found highly effective activity of the catalyst for carbon monoxide and relatively high efficiency for hydrocarbons. Measurements of nitric oxide levels showed no significant changes in concentration in a range of temperatures.

KEYWORDS: THE INTERNAL CATALYST; POSITION MODELS;

1. Introduction

In the present paper under the notion of internal catalyst there is understood the substance of catalytic features which is applied fragmentally or entirely on the walls creating the combustion space of the internal combustion engine or a part of this space- e.g. valve, heat plug which were previously covered with ceramic coat setting the media for this substance.

The ceramic layer fulfills two important functions:

- weakly conducts heat and sets thermal barrier, which enable temperature raising in the place of active factor application
- sets the media for catalytic layer (and thanks to spongy structure it is possible to get bigger contact surface between active layer and gas factor.)

Between external surface of engine element (bearer) and the ceramic layer there has been applied an intermediate layer which aim is to prevent both corrosion and detaching of the ceramic layer due to heat expandability between materials (Fig. 1.1)

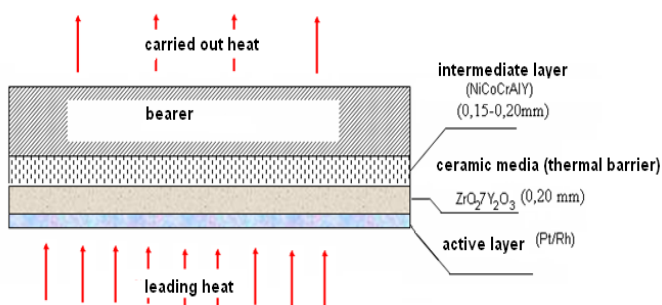


Fig.1.1 Ideological scheme of inner combustion catalyst structure.

Depending on the kind, size and placing of the catalyst as well as on load and engine rotational speed, the effectiveness of catalyst working in the particular stages may change.

In the considered case the active layer has been made on the surface of valve mushrooms. Making the active coat on the engine valves

may cause certain limitation of catalyst influence on the combustion process connected with placement interactions of valve- piston, which in the GMP area cause limitation of the contact of the catalytic coat with the factor inside cylinder. However, in the spark ignition engine, where valves are placed at an angle to the piston surface, the problem of interaction limitation is low.

From the Zeldowicz's formula it follows that the velocity constant of chemical reaction is directly transferred into spread velocity of flame front:

$$U \approx \left(\sigma V e^{-\frac{E}{RT}} \right)^{\frac{1}{2}} \sim \frac{1}{\tau^{1/2}} \quad (1)$$

which is essentially crucial concerning the fact that, as it has been mentioned in the present chapter, the catalyst contributes to the increase of chemical reaction velocity (speeds up the reaction).

As an effect, the heat source power is decreasing in the temperature field, but with retaining carrying out of heat at the foregoing level, which influences the load transformation of the introduced fuel as well as the decrease of the local temperature gradients. The maximum and average temperature over piston are both reduced. On the pointer diagrams then, there can be noticed "soothing" of engine working thus the decrease of an amplitude.

The system aims at balance condition. According to the law of conservation of energy and spin in the process there ensues extra "residual energy" which levels remaining radical processes. Assuming that in the first phase there comes the effect of so called heat explosion, then the second, final phase of ignition caused by residual energy will have character of light pressure increase. Although the pointer diagrams indicate the overall loss pointer efficiency, the mechanical efficiency raises and thus the engine working effectiveness increases.

On the basis of the abovementioned analysis, there can be stated that making use of factor of catalytic qualities in the engine combustion space has an impact on the improvement of engine working.

One can conclude that the catalyst placed inside the combustion space may influence several stages of the fuel mixture combustion process in the engine combustion space [3]:

- preparing of combustible mixture phase- injected fuel cracking processes
- pre flaming phase- shortening of ignition delay
- combustion phase- combustion velocity increase
- hydrocarbon in the wall layer and CO finish burning phase.

In Poland, the first information about internal catalyst made on ceramic coat appeared in the works of Merkisz and Walkowiak [3].

2. Model position

For the purposes of the present work, as the active factor platinum and rhodium have been chosen- the precious metals commonly used in catalytic transducers destined for purifying exhaust gases from internal combustion engines.

Extracting literature data, there have been worked out the procedure of depositing the catalytic coats (platinum, rhodium) on the cylinders used in trials on the model researches seats and on the engine construction elements which have direct contact with air-fuel mixture in the combustion space.

The coverage of the method of depositing the active factor on particular engine elements can be divided into three stages:

- Stage I. The choice of way of depositing precious metals on the ceramic layer.
- Stage II. Implementation of catalytic coat on the proper seat elements for model and actual researches .
- Stage III. Physicochemical researches of coats with catalyst.

In order to determine the effectiveness of the working of catalyst applied on the zircon coat made on engine valve, there have been built the seat for model researches. It has been decided to make the seat based on real engine elements. For this purpose, there has been applied the fragment of engine head with valves and piston which was placed in the cylinder made of eutectic alpacks, on which the spiral electric heater with power of about 2kW has been placed. The heater was supplied by autotransformer system which allows to regulate heating power. The model has been toolled up in the regulating system of piston and valves position. In the space over piston there have been placed 5 thermocouples enabling the temperature changes observation. As working gas, the fumes out of working ZI engine were used. During the researches the composition of fumes leading and carrying out of model has been analyzed. After initial trials there has been assumed that the measurements will be taken out with the catalyst temperature of about 250°C by the fumes flow at about 30l/min. The scheme of model position is presented in the fig. 2.1.

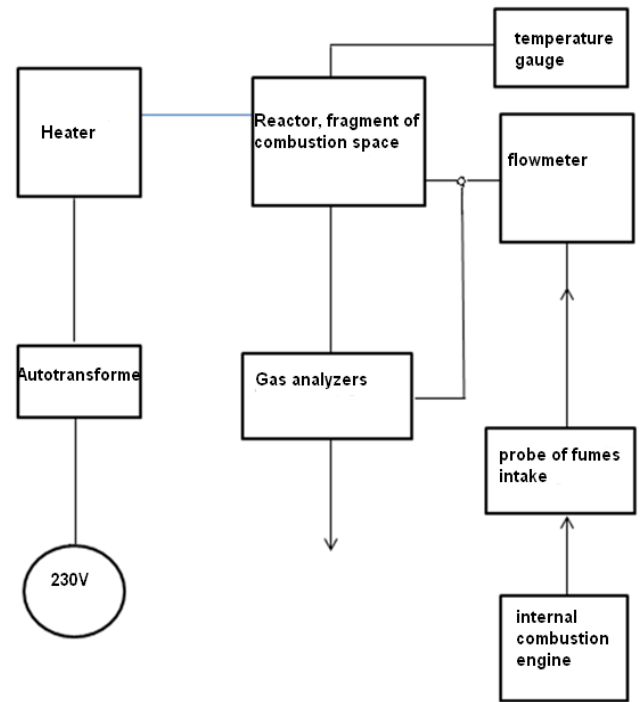


Fig. 2.1. The scheme of model position for effectiveness of the internal catalyst researches.

Assembled model position has been placed in the cubicycle of the roller performance tester near the engine which was used as the fumes generator. The view of the position is presented in the fig. 2.2



Fig. 2.2. The view of model position.

3. Research Methodology

As an operating factor in model systems the fumes from spark ignition engine working on idle running were used. The fumes composition in the range of CO and NO was measured by analyzer from the Hartmann- Braun company, which was working on the basis of nondispersive absorption in infra red, CO₂ and HC analyzers of Junkalor company, which were working on the same basis.

The temperature of fumes and system elements were measured by iron-constantan thermal couples with electric compression and electronic digital display.

The fumes flow strength was measured by the flowmeter with flow support of electric flow pump.

Measurements were made by heating of the reactor system by the use of electric heater which power was regulated by autotransformer with the system stabilizing the temperature.

4. Results of model researches

The researches were made with the use of engine valves with the plasma jet priming layer of zircon ceramic and also applied according to methodology presented in chapter 4 the platinum- rhodium catalytic layer. The valves were placed in the fragment of head in the open position, which enables the flow of the provided factor to the seat through the flexible cord from the fumes intake probe from the engine exhaust system. The system was electrically heated acquiring demanded temperature values of the factor. There have been carried out series of measurements for three temperature values. The measurements results are presented in the following consolidation (table 4.1)

Table 4.1. Researches results in the combustion space model.

Lp	CO %	CO ₂ %	HC ppm	NO %	Temp. °C
1	3,23	3,44	70	0,20	150
2	3,15	3,50	62	0,20	200
3	3,0	3,58	58	0,22	250

And in the fig. 4.1 where the process of CO and HC concentration changes are presented.

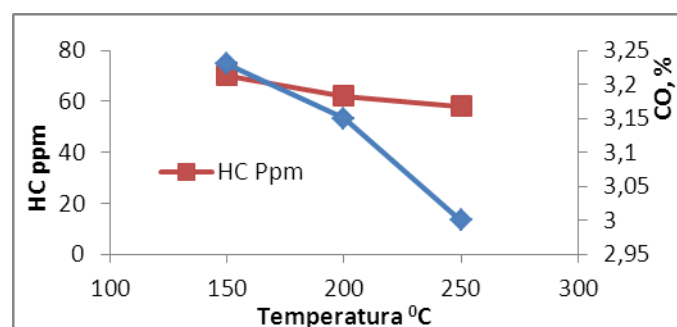


Fig. 4.1 . Measurements results of the effectiveness of catalytic coats in the model studies.

5. Conclusions

- There have been carried out the model researches of solutions series of active coats estimating the influence of internal combustion catalysts on the concentration of fumes toxic components and obtaining their essential reduction.
- The model researches reaffirmed the effectiveness in internal combustion catalysts in CO and HC reduction. There has been affirmed the CO and HC concentration loss of about 10% with the temperature change from 150 to 250 °C.
- On the case of hydrocarbons there has been affirmed the group composition change of HC compound. The changes of concentration proportions for particular groups of components were essential just for the temperature of 200°C. This phenomenon has been decided to be explained in the further researches.

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COMBUSTION ENGINE INTAKE PORT DESIGN ANALYSIS

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Abstract: Engine “breath” air through the intake ports. It is important part of every engine which has influence to the final characteristic. In this article is shown from where the design of intake port result. Article shows books theory with compare to 1D CFD analysis.

KEYWORDS: COMBUSTION ENGINE, ENGINE HEAD, ENGINE PORT, FLOWS, ENGINE MODIFICATIONS.

1. Introduction

Important factor, especially in racing engines is power and torque. Torque is taken from the engine dynamometers measurements or could be calculated from brake mean effective pressure (BMEP). Power is a value which results from torque and rotational speed of the engine.

Another important factor is delivery ratio (DR). It is talking about how many air is taken by the engine. It is different than volumetric efficiency because it takes into account the air density which for example varies with altitude. Increase of DR gives increase of BMEP, torque and power.

$$DR = \frac{M_{cycle}^{air}}{\rho_{ref} V_e} = \frac{\frac{60}{N} \int_{tdc}^{bdc} C_d \rho C A_t d\theta}{\frac{60}{N} \int_{tdc}^{bdc} A_{stp} d\theta} \sim \frac{\int_{tdc}^{bdc} C_d \rho C A_t d\theta}{6 N V_e} \quad (1)$$

where:

ρ_{ref} – air density
 C_d – discharge coefficient
 C – particle velocity
 A_t – flow area
 A_{sta} – specific time area
 $d\theta$ – crank angle

It have to be explained that A_t is a side area of cone in geometry meaning (fig. 1). It is characterized by valve lift, valve diameter, valve and valve seating shape. Flow area vary with crank angle.

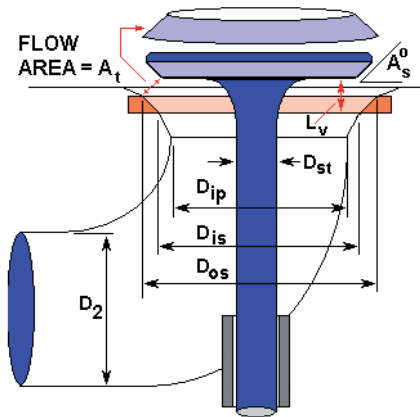


Fig. 1 Valve and port geometry [1].

Engine “breath air” through the intake ports. They should:

- maintain the lowest possible air flow resistance,
- be designed with accordance to the wave and dynamic theory,
- be smoothly connected with intake manifold and design

- should take into consideration valves (valve profile, valve seat and valve guide).

Area of the duct cannot be too high or too low. Lower area provide higher flow resistance but gives better dynamic boost and it is better on lower revolution velocity. Fig. 2 shows how change volumetric efficiency with different parameters of K_1 and K_2 (Brandstetter critic numbers), where:

$$K_1 = \frac{L_D \cdot n}{120 \cdot a} \quad (2)$$

$$K_2 = \frac{L_D}{s} \left(\frac{d_d}{D} \right)^2 \quad (3)$$

where:

L_D – intake duct length
 n – RPM
 a – local sound velocity
 s – engine stroke
 d_d – port diameter
 D – cylinder diameter [3]

Many scientists assumed that duct area should give velocity of air between 40 and 70m/s [2][4][3].

However document [1] says that the best is velocity near to 0,5 mach.

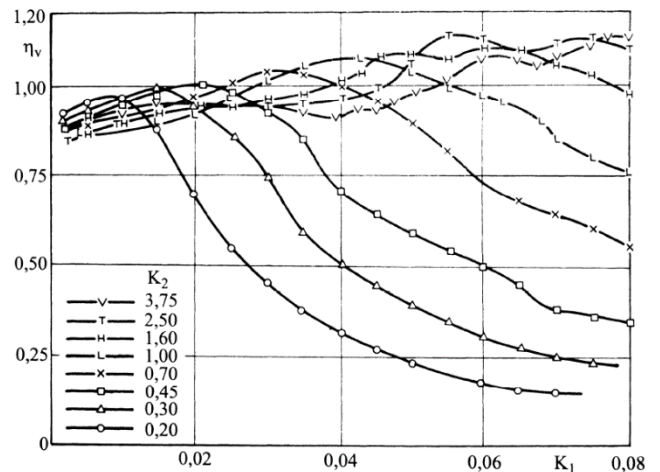


Fig. 2 Volumetric Efficiency versus critic numbers K_1 and K_2 [3].

Optimal ratio of manifold area to port area is described in [1] by K_m coefficient.

$$K_m = \frac{A_{manifold}}{A_{port}} = \frac{D_2^2}{n_v \times (D_{ip}^2 - D_{st}^2)} \quad (4)$$

K_m in case of exhaust should be in range 1.2 to 1.4

K_m in case of intake should be in range 0.9 to 1.0

Diameters are described on fig. 1.

As it was mentioned previously, because of dynamic charging length of duct is strongly connected with engine volumetric efficiency (fig. 3).

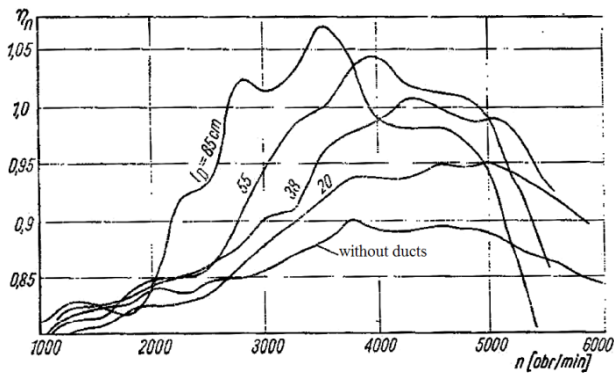


Fig. 3 Volumetric efficiency versus intake length [2].

Most popular formulas to calculation of intake length values are Chrysler formula:

$$L_D = \frac{6a}{n} \pm 0,075 [m] \quad (5)$$

and French brands formula:

$$L_D = \frac{2574}{n} [m] \quad (6)$$

where:

L_D -intake duct length,
 n - RPM in which works dynamic charging,
 a -local sound velocity [4].

2. Investigated Engine

Engine under investigation is GM 500. That is 500 ccm, one cylinder engine which was built in 1983 and is still one of the best engine in speedway competition. Engine is also commonly used in karts. Can be also used in Formula Student competition. It has 4 valve engine head with two camshaft inside. It produce brake power about 50 kW in 10000 rpm. Those engines has many configuration and many tuners are trying to improve them.

Engine has been simulated as a 1 dimensional CFD model, which takes into consideration wave, dynamic engine flow phenomena and flow resistances. It allows to choose the best intake, combustion and exhaust system configuration without expensive real measurements.

External characteristic of the engine model correlate with real parameters. Fig. 5 and fig. 6 shows computed power and torque characteristic.



Fig. 4 Investigated engine head inlet port.

Table. 1 GM 500 Engine parameters.

No. Of cylinders	1
Strokes per cycle	4
Engine Type	Spark Ignition
Power	52 kW/10000 rpm
Torque	55 Nm/8200 rpm
Bore	90 mm
Stroke	78,5 mm
Displacement	496 ccm
Connecting rod lenght	166,5 mm
Compression Ratio	17
Engine head	OHC / 4 valves

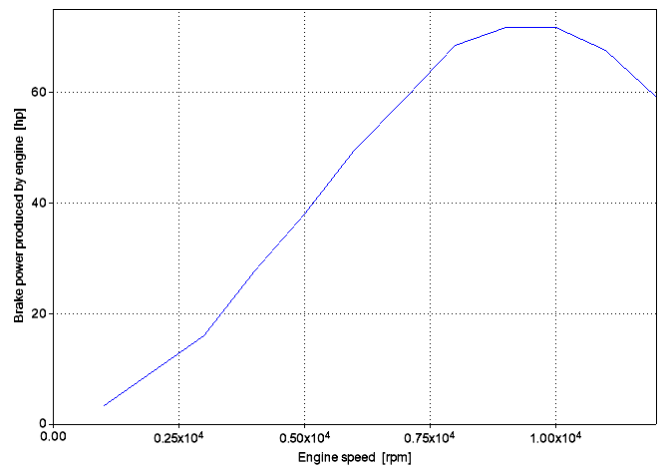


Fig. 5 Computed characteristic of power.

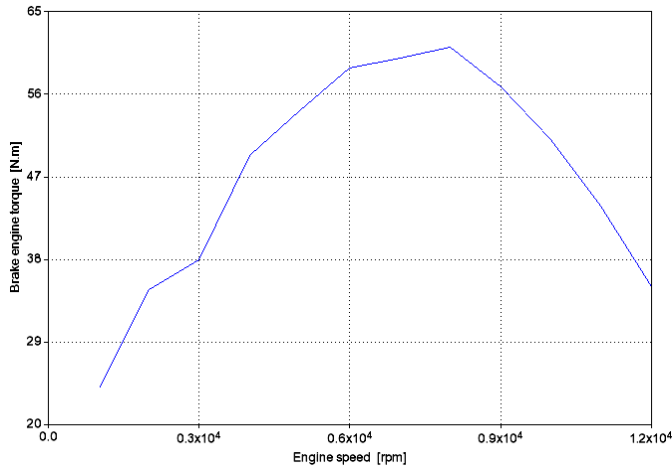


Fig. 6 Computed characteristic of torque.

Engine ports has been scanned in 3D. The result was cloud of points. Fig. 7 shows model which was prepared based on scan. On the left side is exhaust port, on the right side is intake port.

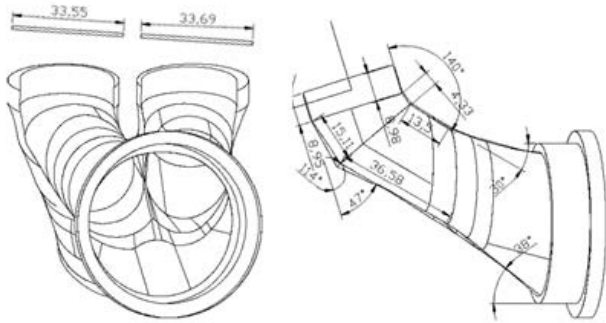


Fig. 7 Inlet port model.

3. Analysis of the problem

Fig. 8 shows the schema of the ports. Elliptical cross section in CFD model was adopted to rounded cross section, where the diameter is the average value of both elliptical axes. Intake port is going narrow in the center part. Main idea was to make this area wider together with individual ports for each valve

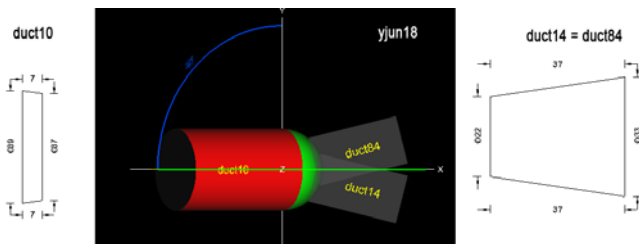


Fig. 8 Inlet port schema.

Investigated variants were:

- Standard design (yjun18 Ø38) (fig. 8),
- duct10 Ø39 mm to Ø42 mm; yjun18 Ø42 mm and duct14/84 Ø25 mm to Ø33 mm,
- duct10 Ø39 mm to Ø44 mm; yjun18 Ø44 mm and duct14/84 Ø26 mm to Ø33 mm.

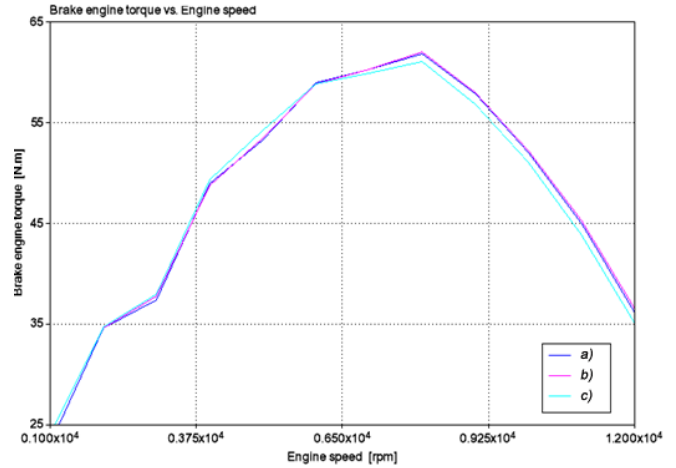


Fig. 9 Computed torque characteristic before and after modification of inlet port.

Standard port design gives better torque (and power) about 5000 rpm. Based on that it can be sad that increase in port area makes increase in engine parameters. Increasing of port area destroy characteristic of the engine. Options b) and c) gives higher power output but worst torque about 5000 rpm (fig. 9).

Lowering exhaust ports give smaller torque in high engine speed and higher in low engine speed. However low engine speed is assumed as less important in motorsport engine.

4. Summary

Summarizing modelling of engine was successful. Final results presented by the model and from dynamometer are related enough. Phenomena effect of geometry changing in theory and in simulated engine are similar. Shortening and increasing area of ducts makes high torque on high engine speed.

Final improvement gives higher power, up to 1,5 hp and about 1 Nm higher torque in wide range of engine speed. The maximum power could be higher but then engine torque would be worst.

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COMPARISON OF CHARACTERISTICS OF SPARK PLUG ENGINES FSI, TSI/TFSI TYPE OF VOLKSWAGEN COMPANY

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Abstract: The use of direct injection in spark ignition engines, significantly facilitated the use of chargers in these engines. This resulted lately in the significant popularization of direct injection engines, initially freely sucking and in final result turbocharged. The greatest popularity on the market gained engines of Volkswagen company, named FSI and TFSI / TSI. Application of Common Rail systems allowed not only to improve the characteristics of the engine by increasing the accuracy in dispensing fuel into individual cylinders. The most important gain is the possibility of second injection of the fuel to the cylinder after the intake valve is closed. On the one hand it allows better control of the load in the cylinder, at first with the piston crown, and now with shaping the injection by the injector.

KEYWORDS: TRANSPORT, COMBUSTION ENGINES, FUEL INJECTION, STRATIFIED INJECTION, CHARGE ENGINES

1. Introduction

Petrol engines recently lost competitiveness against turbocharged diesel engines. Previously used indirect injection technology, was a restriction in supercharging those engines, so that the most effective way of raising the torque of the engine was increasing its capacity, which resulted in the increase of fuel consumption, having negative repercussions on the profitability of the purchase. Application of common rail which injects fuel directly into the combustion chamber, has facilitated the use of turbocharging. This gave birth to so-called current trend of downsizing engines, which is replacing engines with large capacities, by low capacity units, reaching similar power thanks to turbo-charger. This resulted in the recovery of the market for gasoline engines, at the expense of market share of diesel engines. According to IBRM Samar in 2009, sales of new diesel vehicles was about 5% lower than a year earlier. FSI engines on which are based the TSI units, breathed new life into sales of gasoline engines. These are engines with interesting technical solutions and are often awarded in various competitions. Engine 1.4 TSI Twincharger two years in a row (2009 and 2010) has been recognized as the best engine of the year by journalists from around the world.

2. Comparison of the characteristics of the engines naturally aspirated and supercharged

At the beginning it is worth to compare directly the turbo-charged engine which replaces the naturally aspirated unit in the manufacturers offer.

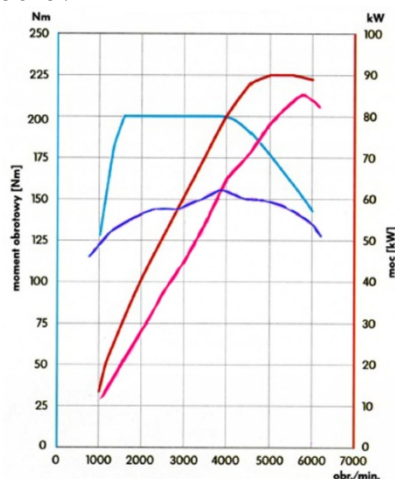


Fig. 2.1 Comparison of characteristics of the 1.6 FSI engine with the 1.4 TSI

Light-red color indicates the characteristics of power of the 1.6 FSI engine, and the purple its torque. Dark-red color indicates the engine power of 1.4 TSI and blue, its torque. It is clear that the curve under the turbo-charged engine is steeper and more quickly reaches its maximum. Faster gaining power by the engine with an increase of its rpm is a positive result. It allows to quickly cruise the vehicle at lower engine speeds, thereby saving the engine and fuel. But the most significant and distinctive difference is the torque curve and its maximum value. In addition to a noticeable jump of the maximum value (about 45Nm), it is kept constant at the high engine speed range (from about 1500 to 4000 rpm). The turbo-charged engine shows a very steep growth of the torque curve in the initial engine speed range (1000-1500 rpm). This is achieved by using a small, high-speed turbocharger. It quickly reaches maximum performance, strongly improving the degree of filling cylinders. Such torque curve allows dynamic drive and smooth overtaking a car, even at low engine speeds. In relation to the naturally aspirated unit there's no need to reduce a gear to achieve a good acceleration. This has a very positive impact, on both the durability of the engine, and its fuel consumption. Such a large range of flat torque curve gives even better operating characteristics then liked by some drivers, turbocharged compression ignition engines.

It's also worth to consider the impact of supercharging the engine, without changing its displacement.

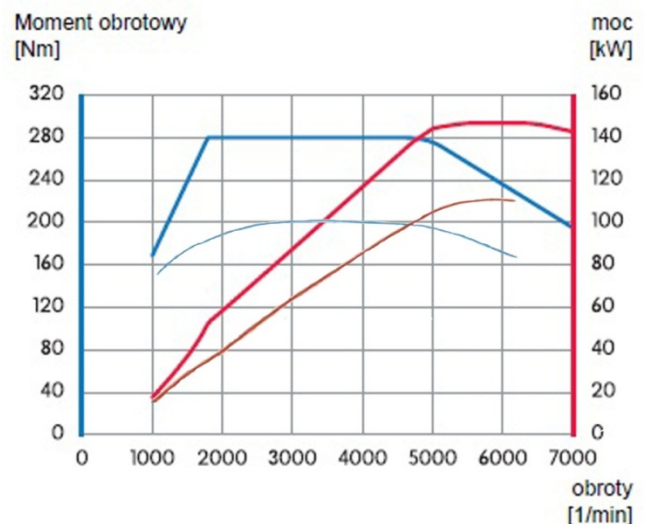


Fig. 2.2 Comparison of characteristics of the 2.0 FSI engine with the 2.0 TFSI

In the graph, the thicker lines are marked characteristics of the power (red) and torque (blue) of the turbo-charged engine, and the thinner lines shows characteristics of the same capacity engine, but without the turbocharger. As in the case of comparing engines with different capacities, turbo-charged engine has a steeper power characteristics, and it reaches the maximum value much earlier. Naturally aspirated engine develops its maximum power for a moment before the end of its range of revving speed, while the turbo-charged one is able to maintain maximum power over some range of rotation. A similar situation exists in the case of the torque curve. By turbocharging the engine reaches full torque in a large range, from 1800 to nearly 5,000 rpm. In the absence of supercharge, the maximum torque is reached at 3,500 rpm, but values close to its maximum value are maintained in the range 2,500 - 4,500 rpm. As with the smaller capacities, supercharge significantly increases the range of maximum torque, improving ride comfort and flexibility of the engine.

The manufacturer claims fuel usage in the city for a naturally aspirated version at 11.3 l/100 km, while for the 11 turbo-charged version of 11 l/100 km. Reports gathered from users by the service autocentrum.pl not show large discrepancies with these claims. According to the users of this service, the maximum average fuel consumption, they managed to get for the version naturally aspirated is 11.4 l/100 km, which is a value coinciding with the manufacturers claim. In the case of turbo-charged version, the report shows the maximum average fuel consumption of 12.3 l/100 km. It's value little different from the manufacturer's data, but nevertheless the difference in fuel consumption between these engines is less than 1 l/100 km. Probably, this involves not only improving the efficiency of the engine thanks to turbocharging, but also increase the range of accessible, high values of torque. This eliminates the need to maintain high engine speed while overtaking, which may have a positive impact on fuel consumption.

3. Comparison of characteristics of the manufacturer with the dyno charts and conclusion

It is worthwhile to compare the characteristics provided by the manufacturer, to the actual plots made on the engine dyno. This allows to verify if the engine manufacturers claims, have representation in the case of units produced and sold on a massive scale. This can also show if differences between engines and profit of the engine downsizing remains at the same level, or undergoes some changes.

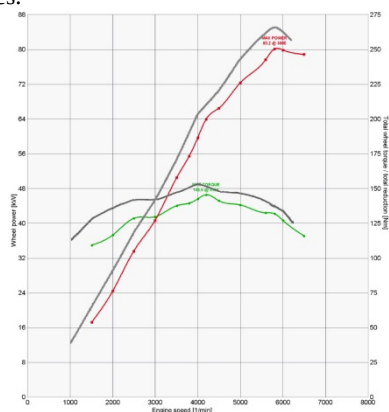


Fig. 3.1 Comparison of the characteristics of the 1.6 FSI green - the actual curve of torque, red - the actual curve of power, curves marked in gray are provided by the manufacturer

It's good to start comparison from naturally aspirated engine, which may be a reference to the supercharged engines with smaller capacities. It is easily seen that the shape of curves is similar to the data declared by the manufacturer. The difference, however, appears in the case of values. In fact, the engine has parameters lower than declared. Maximum power is lower of 5kW, and the

torque of 10Nm. The differences are approximately constant along the curves, additionally can be noted that, in fact, the engine develops its maximum torque at higher revs than declared. Deviation from the declared value may be due to fuel the engine used in the measurement, because the manufacturer recommends the use petrol with octane value 98, and the measured engine used fuel with octane number 95. Changing fuel could affect actual curves by slightly increasing them, and close them to their declared values. This shows the high requirements of naturally aspirated engine with direct injection, on the quality of the fuel mixture, what increases cost of using it.

The manufacturer in the 1.4 TSI Twincharger, recommends the use of fuel 98 octane numbers. In this measurement is used fuel with octane value of 98.

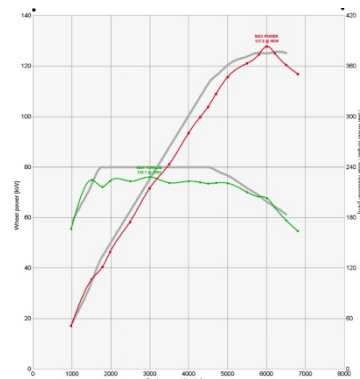


Fig. 3.2 Comparison of the characteristics of the 1.4 TSI twincharger green - the actual curve of torque, red - the actual curve of power, curves marked in gray are provided by the manufacturer

It is easily seen that the 1.4 TSI twincharger engine develops slightly higher maximum power than declared by the manufacturer. However, it can be clearly seen that the difference in favor of actual performance is only at the maximum value. In most parts of the graph of power the curve from the measurement reaches a lower value then declared. A more noticeable difference occurs in the torque curve. Comparison of the maximum torque values are lower in actual unit. Reached values are lower of 15 Nm. Curve of the torque on a substantial part of engine speeds is lower in the measured engine. Chart shows a decrease of torque at engine speed around 1,800 rpm.

This shows that the combination of two types of supercharge, not only raises the value of maximum power and torque, but also improves the course of the curves, especially of the characteristics of torque at low engine speeds. We should see how it looks for a larger engine cylinder capacity and performance.

As an example, it may serve a 2.0 TFSI engine, with claimed 147 kW and 280 Nm of torque.

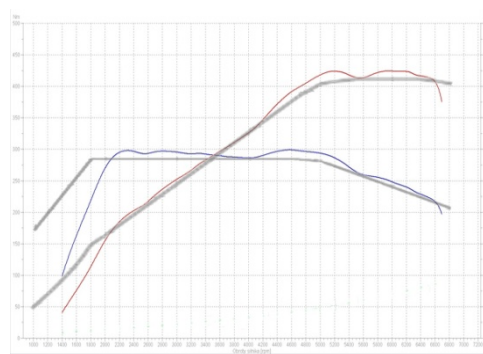


Fig. 3.3 Comparison of the characteristics of the 2.0 TFSI blue - the actual curve of torque, red - the actual the actual curve of power, curves marked in gray are provided by the manufacturer

As is clear from the measurements, 2.0 TFSI engine develops maximum power of 10 kW in more compared to the manufacturer's declaration. Also, the maximum torque is higher by 19 Nm. Measurement is started at a slightly higher engine speed, but the power graph is gaining rapidly on values and at level of 2000 rpm equates to the manufacturer's chart. At medium engine speeds the power curve coincides with the declared course, and at high engine speed, above 4200 rpm is gaining an advantage, and shows slightly higher value than declared. In case of the torque curve, the graph is gaining rapidly and achieves the declared value, and even a bit higher than that at 2200 rpm. Torque curve, maintains a fairly equal and continuous shape, similar to the stated by the manufacturer, but in almost the entire range above 2000 rpm is slightly ahead of his values.

From a comparison of actual characteristics it can be postulated that the better effect can be achieved by adding a small supercharge to the engine with more capacity than the big supercharger to a small engine.

Comparing the supercharged engines with naturally aspirated ones, it is worth considering what is the impact of charging on the work done by the engine. The easiest way is to show on the closed indicator diagram, which will show work done by the engine, in the form of the changes of pressure in the cylinder as a function of capacity. The graph of this type consists of two isentropes (compression and expansion), two isochores (delivering and carrying away of heat), and filling and exhaust factor lines from the cylinder (exchange factor). In the case of an engine naturally aspirated exchange factor process, limited by lines of carrying away and delivering, work is negative. In turbocharged engine, due to the pressure in the system, this work is positive.

These curves are determined, for two Volkswagen engines. These are engines of same capacity, the 2.0 FSI and the 2.0 TFSI.

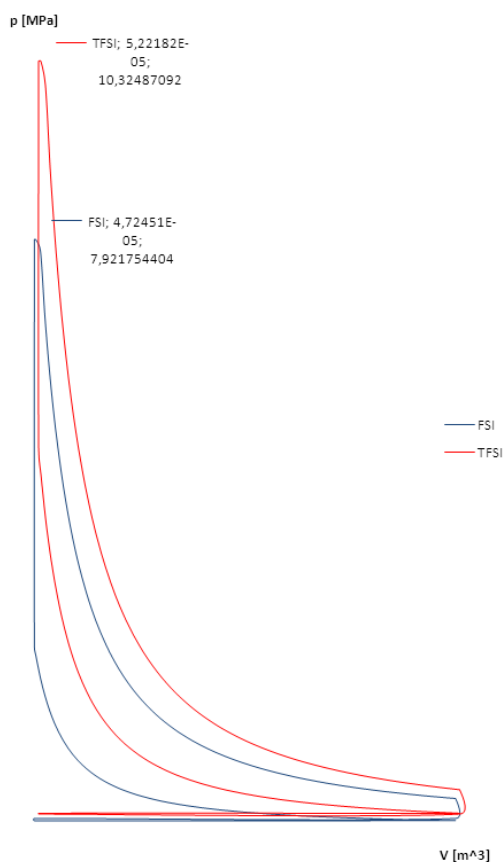


Fig. 3.4 Comparison of closed indicator diagram of 2.0 FSI engine with 2.0 TFSI

Comparing these two graphs, we can see clearly higher pressures achieved by the turbo-charged engine. This implies with a greater quantity of the factor in the combustion chamber, which is being more compressed. Slightly lower minimum volume of the

combustion chamber in a turbo-charged engine, is associated with a lower compression level. Despite of the decrease in level of compression, thanks to turbocharger it reaches higher pressure levels. The entire turbo-charged engine diagram is slightly lifted in relation to the graph of naturally aspirated engine. This shows a positive impact of the turbocharger on the engine.

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ENERGY SAVING IN THE HYDRAULIC CIRCUIT FOR AGRICULTURAL TRACTORS: FOCUS ON THE POWER SUPPLY GROUP.

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Abstract: This work aims to analyze a complete hydraulic system of a medium size agricultural tractor, in order to perform an energy dissipation analysis and to suggest possibly alternative configurations and solutions. The fuel consumption and energy dissipation in off road vehicles have in fact become a key feature, given the great attention devoted to the need of reducing pollutant emissions, in order to satisfy the future emission limits. In this work the focus is on the fluid power supply group and the standard architecture of this unit has been compared with an alternative solution, called variable pump margin strategy and compared on the basis of the power consumption to perform the same duty cycle, showing that a relevant percentage of energy may be saved with simple modifications in the hydraulic power generator group.

Keywords: Agricultural Tractor, Simulation, Power Saving, Duty Cycle

1. Introduction

This paper is intended to describe part of the work done as a research program whose final aim is to test and verify the effectiveness of some high efficiency hydraulic architectures in agriculture tractors. Hydraulic circuit in mid-size tractors has different key roles: feeding the auxiliary utilities, the hitch valve, the trailer brake, the steering, the driveline and PTO clutches, the lubrication line and the pilot lines to control the electro-hydraulic valves. Traditional configuration of this circuit involves the use of separate pumps to feed the auxiliary utilities line, the hitch valve and the steering on one side and lubrication on another side and finally the clutches and pilot lines.

The first step of the research is the analysis and the evaluation of the energy dissipation of the auxiliary utilities of a standard circuit, which is a typical load sensing multi-actuators system. The load sensing concept has been introduced in the second half of the last century and it is still used in hydraulic applications because it has proved to be more efficient than the hydraulic power system which entrusted power management to dissipative components (as the traditional pressure relief valve). At the same time, hydraulic load sensing systems are robust and reliable (see for instance [1], [2], [3] and [4]). Nevertheless, the need to limit pollutant emissions of agricultural machines has become more and more pressing in the last years. Many researchers and manufacturers have focused their efforts on reducing the energy consumption in the system without compromising its functionality and its performance. From this point of view, the combined use of simulation tools and experimental testing represents the most promising way to develop alternative solutions characterized by lower energy consumption (see for example works from [5] to [11], which proposes different solutions spanning from independent metering valves to displacement controlled actuators).

In this kind of system, possibilities to achieve more energy saving lay on the fact that high dissipative distributors are used to manage the flow to the actuators and to maintain control of multiple loads eventually working together and that the fluid power group must work under a fixed pressure margin that is higher than the maximum working pressure generally of about 2-3 MPa; this margin of pressure has to be sufficient to overcome all the pressure losses at the maximum flow rate that the flow come across from the pump to the actuators.

Possibility to reduce power losses lay hence both on the power generation side and in the power utilisation side. In this paper the focus is more on the power generation group improvement. It's noteworthy to underline that hydraulic circuit in a tractor can serve a wide range of actuators, depending on the equipment connected at the moment on the tractor, for example a seeder, or a loader or a harrow. The actuators can be both motors and cylinders

and standardized duty cycles for these kind of circuit do not exist at the moment. This requires more efforts in the analysis to be devoted at the experimental measurement of the main hydraulic variables during a working cycle of the tractor with different equipment. On the basis of previous analysis ([12], [13]), here it was chosen to adopt a standard duty cycle created analysing the whole life and average use of a mid-size power tractor. The alternative solutions evaluated and compared with the standard architecture are built through the adoption of electronically controlled variable displacement pumps. In the future, the logical following step of this work will be to select the best solutions from the simulation results and to realize them on the vehicle, experimentally testing on vehicle the ability of new solutions of maintaining the vehicle performances beside the introduction of energy saving.

2 The Standard Load Sensing Hydraulic System and its Modelling

Given the complexity of the hydraulic circuit of the agricultural tractor, it is convenient to use a top-down approach that divides the entire circuit in different operational sections.

The hydraulic circuit is composed by a fluid power supply group, and a complex whole of hydraulic actuators (high and low pressure users) and control valves.

The power supply unit represents the heart of the tractor. This is composed of two pumps (a charge pump of the gerotor type and an axial piston pump) that generate the operating flow rate required by the users. The power supply unit is a typical load sensing open-circuit group that involves the use of local pressure and flow compensators. Advantages and critical aspects of these kind of systems have been thoughtfully analyzed in literature, for example in [14] and [15]. When speaking of the energy dissipation of these systems, the critical aspect is that they work at the pump delivery with an higher pressure level than the one required by the load. These pressure difference (pump margin) is regulated in order to be able to guarantee the maximum flow rate request at the actuators and it strictly depends on the losses that flow encounters in the passage from the pump delivery to the user. In the standard hydraulic pump, this pressure difference is fixed, even if it could be reduced when the flow request is not at its maximum level, thus reducing dissipation.

A priority valve is placed after the hydraulic power supply unit; this valve has the task of ensuring the proper operation of the three main users of the agricultural tractor: steering system, trailer brake and agricultural equipment whose actuators are controlled with auxiliary distributors.

The priority valve, consisting of two sections that work in parallel, sets limits of operating pressure and in emergency conditions guarantee a level of flow and pressure to the users with a specific priority.

In connection with the priority valve are the following utilities: steering, trailer brake valve and auxiliary distributor, with their respective lines of load pressure sensing. Depending on the load detected, the valves in the priority block are used to adjust the flow rate and the pressure coming from the power supply unit, ensuring a priority order in the functions of power steering, trailer brake and auxiliary utilities.

The trailer brake valve has the task of managing the pressure signal generated by the brake pedals in the cab and the pressure to be provided for the parking brake of the machine.

The auxiliary actuators may work together hence local pressure compensators are introduced on each section of the auxiliary distributor to guarantee the correct flow sharing. These elements, because of the geometry complexity introduced on the distributor section and of the multiple oil passages needed to perform the control and to guarantee the flow, are critical for the dissipation introduced ([16]).

To complete the hydraulic circuit of the agricultural tractor, the main accessory components are the filter and the heat exchanger.

The hydraulic circuit of the tractor in its standard configuration is represented in Figure 1, by means of a functional representation in which the blocks described before are recognizable. These blocks contain the mathematical models that define the components behaviour, developed in LMS Imagine.Lab AMESim and AMESet, following a lumped parameters approach (Figure 1).

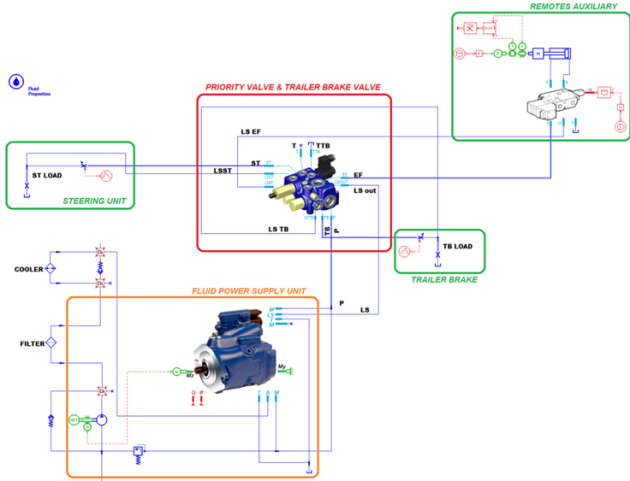


Figure 1: Standard Hydraulic Circuit of a medium sized tractor

The lumped parameters model is divided in three main sections. The first one (orange block in the Figure 1 named FLUID POWER SUPPLY UNIT), includes the detailed model of the axial piston pump of the tractor, together with the flow and pressure compensators, and the model of the charge pump. The instantaneous pressure at the axial piston pump delivery and suction, the flow rate and the torque transients are typical output variables of this block.

The second section (red block), named Priority & Trailer Brake Valve, includes the detailed model of the priority valve and a “functional model” for the trailer brake valve. These models ensure the correct pressure settings and flows-sharing towards the users.

The green blocks contain the hydraulic models of the main users of the agricultural tractor. The steering user and trailer user are modelled by means of variable orifices, suitably controlled and calibrated.

A detailed hydraulic model is instead realized for the auxiliary distributors, since they have shown from previous analysis ([12]) to be quite dissipative elements.

The hydraulic model involve also the filter and lubrication section not shown in Figure 1.

Focussing on the fluid power group, in the following a detailed description is given of the main pump, the variable displacement swashplate axial pistons pump. The hydraulic model realized in LMS Imagine. Lab AMESim environment is an evolution of the model presented by the authors in [17], consisting mainly of three main sections:

- rotating group of axial piston pump, including the pistons moving within the cylinder block, the valve plate, the swashplate

- actuator that perform the adjustment of the displacement of the machine

- flow and pressure compensators: it implements the modelling of the 3 way valves that contribute in the determination of the control pressure within the actuator chamber.

The model allows the determination of the pressure transients in the piston chambers, the instantaneous forces exerted between the movable parts of the pump, the torques at the swashplate and at the shaft and the instantaneous flow. The basic module is represented by the piston chamber which is a typical control volume that communicates with high and low pressure ports through variable orifices, opportunely controlled in order to replicate the opening and closing of the flow areas due to the cylinder blocks and valve plate interposition.

For each control volume continuity equation can be written under the hypothesis of an incompressible fluid and considering isothermal flow:

$$\begin{aligned} \frac{dp_i}{dt} &= \frac{B}{V} \cdot \left(\sum_k Q_k - \frac{dV_i}{dt} \right), \quad i = 1 \dots N \\ \frac{dp_i}{d\theta} &= \frac{B}{V \cdot \omega} \cdot \left(\sum_k Q_k - \omega \cdot \frac{dV_i}{d\theta} \right) \quad \omega = \frac{d\theta}{dt} \end{aligned} \quad (1)$$

In equation (1), B is the Bulk Modulus of the fluid, Q_k represents the generic flow coming in or out the piston chamber. The piston chamber volume is defined as a function of the angular position of the machine shaft θ and of the swashplate β , as in equation (2) where V_0 is the dead volume of the chamber, R_{cp} the cylinder block radius, β the swashplate angle, D_p piston diameter, θ_i the angular position of the piston and ω the angular speed.

$$\begin{aligned} V &= V_0 + R_{cp} \cdot (1 - \cos \theta_i) \cdot \tan \beta \cdot \frac{\pi}{4} D_p^2 \\ \frac{dV}{d\theta} &= \frac{R_{cp}}{\omega} \cdot \frac{\pi}{4} \cdot D_p^2 \left(\sin \theta_i \cdot \tan \beta \cdot \omega + \frac{1 - \cos \theta_i}{\cos^2(\beta)} \cdot \dot{\beta} \right) \end{aligned} \quad (2)$$

The flow through the valve plate ports is determined using the equation of flow through an orifice having geometrical passage equal to A:

$$Q = C_d \cdot A \cdot \sqrt{\frac{2 \cdot |\Delta p|}{\rho}} \cdot \text{sign}(\Delta p) \quad (3)$$

In equation (3) C_d represents the discharge coefficient, variable as a function of the Reynolds number as described in equation (4), where D_h hydraulic diameter of the orifice, ν kinematic viscosity e ρ the fluid density; R_{ec} the value assumed by the Reynolds number where the transition between laminar and turbulent flow happens; C_∞ is the value of the discharge coefficient in fully turbulent flow condition across the orifice.

$$C_d = C_\infty \cdot \tanh(2 \cdot \text{Re} / \text{Re}_c), \quad \text{Re} = \frac{D_h}{\nu} \cdot \sqrt{\frac{2 \cdot |\Delta p|}{\rho}} \quad (4)$$

Particular care has also been devoted to the description of the leakages through the lubricating gaps whose geometry (height) is constant and has been defined on the basis of experimental and numerical comparison of data regarding the average flow, swashplate torque and shaft torque for different operating conditions. Equation (5) is used to calculate the leakages across the slipper bearings-swashplate gap; R_{ext} and R_{int} represents the external and internal slipper radii, μ the fluid absolute viscosity, p_{int} and p_{ext} the pressure values at the external and internal radius of the slipper.

$$q = \frac{\pi \cdot h^3}{6 \cdot \mu} \cdot \frac{(p_{\text{int}} - p_{\text{ext}})}{\ln\left(\frac{R_{\text{ext}}}{R_{\text{int}}}\right)} \quad (5)$$

A similar expression is used for the leakage across the piston ball joint-slipper gap, considering axially-symmetric geometry and again constant gap height.

Equation (6) describe the leakage across the cylinder block-valve plate gap, where leakage may flow both to drain and to the adjacent pistons chambers.

Equivalent geometric dimensions have been defined to describe the gap, the length l and the width b_{cy} , and again a constant height h_{cy} (distance between the cylinder block and the valve plate) has been considered.

$$q = \frac{b_{\text{cy}} \cdot h_{\text{cy}}^3}{12 \cdot \mu \cdot l} \cdot (p - p_D) \quad (6)$$

Finally, equation (7) expresses the flow across the gap between the pistons and the cylinder block, considering a zero eccentricity piston position. In equation (7) h_{pi} is as usual the gap height, l_e the length of the gap and $\dot{s}_P$ the piston speed:

$$q = \frac{\pi \cdot D_p \cdot h_{pi}^3}{12 \cdot \mu \cdot l_e} \cdot (p - p_D) \pm \frac{\pi \cdot D_p}{2} \cdot h_{pi} \cdot \dot{s}_P \quad (7)$$

The piston-slipper assembly force, due to the chamber pressure, inertia, viscous friction, determines one of the contribution of the torques acting on the swashplate; the swashplate, modelled as a rotary inertia, receives also the contribution of the spring and of the control actuator. The variation of the swashplate position is instantaneously conveyed to the piston elements and all the cinematic variables are updated as a consequence.

The main geometrical parameters describing the valve plate, the cylinder block, the pistons are defined through a pre-processor module.

Equations (8) shows the displacement s_p , speed and acceleration of the piston: they are function of the shaft angular position ϑ_i and of the swashplate angular position β , calculated for a certain constant angular speed ω .

$$\begin{aligned} s_{p,i} &= R_{cp} \cdot (1 - \cos \theta_i) \cdot \tan \beta \\ \dot{s}_{p,i} &= R_{cp} \cdot (\sin \theta_i) \cdot \tan \beta \cdot \omega + \frac{R_{cp} \cdot (1 - \cos \theta_i) \cdot \dot{\beta}}{\cos^2(\beta)} \\ \ddot{s}_{p,i} &= R_{cp} \cdot \omega^2 \cdot \cos \theta_i \cdot \tan \beta + \\ &+ \frac{2R_{cp} \cdot \sin \theta_i \cdot \dot{\beta} \cdot \omega + R_{cp} \cdot (1 - \cos \theta_i) \cdot \ddot{\beta} + 2R_{cp} \cdot (1 - \cos \theta_i) \cdot \dot{\beta}^2 \cdot \tan \beta}{\cos^2(\beta)} \end{aligned} \quad (8)$$

Piston-slipper assembly undergoes the action exerted from the pressure within the chamber, $F_{p,i}$ from the inertia contribution $F_{m,i}$, from the viscous friction that arises in the piston-cylinder block gap $F_{v,i}$, from the swashplate reaction $R_{z,i}$.

$$\begin{aligned} F_{p,i} &= p \cdot \frac{\pi}{4} D_p^2 \\ F_{m,i} &= -m \cdot \ddot{s}_{p,i} \\ F_{v,i} &= -\frac{\pi \mu D_p l_e}{h_{pi}} \dot{s}_{p,i} \end{aligned}$$

$$\Rightarrow R_{z,i} + F_{p,i} + F_{m,i} + F_{v,i} = 0 \quad (9)$$

Equation (10) describes the swashplate dynamic motion: I the moment of inertia, T_c is the torque due to the actuator action and spring, T_{sw} is the torque of the piston slipper assemblies on the swashplate, as shown in equation (11), T_v the viscous friction torque, as expressed in equation (12) where c is the viscous friction coefficient.

$$I \cdot \ddot{\beta} = T_c + T_{sw} + T_v \quad (10)$$

$$\begin{aligned} T_{sw} &= \sum_{i=1}^N T_{sw,i} = \sum_{i=1}^N F_{sl-sw,i} \cdot l_{sw,i} \\ l_{sw,i} &= \frac{R_{cp} \cdot \cos \theta_i}{\cos \beta} \\ F_{sl-sw,i} &= -R_{z,i} / \cos \beta \end{aligned} \quad (11)$$

$$T_v = -c \cdot \dot{\beta} \quad (12)$$

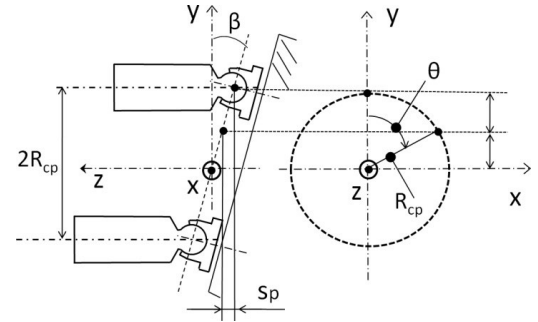


Figure 2 Frame of reference and schematic view of an axial piston swashplate pump.

Finally, detailed modelling of the flow and pressure compensators and of the control and bias actuators are added to the model of pump in order to be able to describe the dynamic variation of the machine displacement during pump operation.

3 Alternative Architectures

Variable Pump Margin (VPM)

The load-sensing variable displacement pump is a very known and studied component, but in recent years it is undergoing substantial changes for the introduction of advanced electronic for the control of the hydraulic power generation.

The electronic control allows an infinite range of adjustments and controls, but a good integration with fluid power components has to be studied and developed in order to guarantee the performances and robustness of the system.

In a typical LS power supply unit, the pump operates at a fixed pressure level higher than the pressure actually required by the user.

This additional pressure, also called Pump Margin, is always fixed by the spring preload of the flow compensator, giving rise to energy losses, depending on the duty cycle or the operation carried out.

Here, the nominal value of 2.5 MPa assigned to Pump Margin is the minimum possible value of system pressure in the standard configuration in order to overcome the losses caused by valves and the actuators connected to it when the flow rate is at its maximum value.

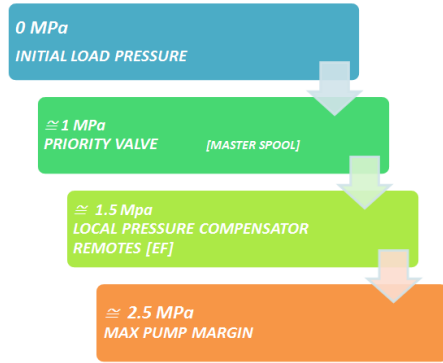


Figure 3: Contribution to build up the pump margin

The estimation of the Pump Margin, as shown in the Figure 3, is due to two fundamental components of the hydraulic circuit: the priority valve and the local pressure compensator of the auxiliary distributors.

The resulting energy losses are due to an high pressure drop across the two valves priority & auxiliary distributors.

The VPM strategy is based on the reduction of the pressure difference between the delivery pressure of the pump and the actual pressure of the generic actuator, by adjusting the spring preload of the flow compensator that equips the axial piston pump, using a solenoid valve and an opportune strategy on the ECU (Electronic Control Unit) to control it.

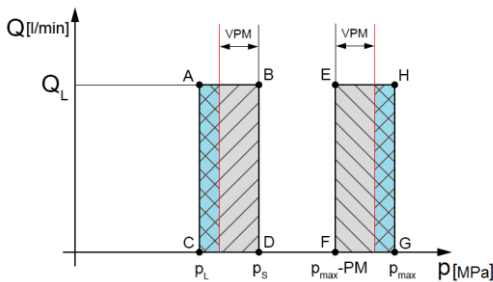


Figure 4 Variable Pump Margin Concept

Let's consider a request of flow rate Q_L corresponding to a pressure p_L , if the regulation of the spring preload of the pump flow compensator is equal to p_M pressure, the main pump is working at pressure p_S given by the sum of p_L and p_M .

The power dissipated in standard working conditions, is represented by the area ABCD between the levels of pressure p_S and p_L . Under pressure saturation condition and considering the maximum pressure p_{max} allowed in the system, the power loss is represented by the area EFGH, between pressure levels p_{max} and $p_{max} - PM$.

For the implementation of the VPM control strategy, it was decided to use a commercial valve, called EFx (EFx is a commercial valve that replace the flow compensator for the axial piston pump. For related and detailed note refer to <http://www.boschrexroth.com>), which replaces the hydraulic compensator with an electro-hydraulically flow compensator controlled by a specific control unit.

The pressure margin can vary between 2.5 MPa to 1.5 MPa due to the action of this valve.

The VPM control strategy provides two types of control that can be implemented on the ECU: fixed point strategy and dynamic variation strategy.

The fixed point strategy is based on the setting of a single value of reduced pump margin for the pump to be regulated when possible and without any possibility of dynamic adjustment.

This strategy is possible only in the case of standby condition of the machine: in fact, in a generic operating condition the lowering of the pressure imposed by the VPM fixed point strategy to the minimum level, will not be able to meet the flow requests coming from the auxiliary and other users connected.

A second control strategy for the VPM is introduced: through the use of pressure sensors (dislocated on the discharge line of the pump and on the main working lines) and reading via CAN net the control signals at the auxiliary directional valves, it is possible to create an operating map for the control unit (ECU).

For a first analysis, a linear dependence has been used between the total required capacity (easily obtainable through the Electro Hydraulic signals of the auxiliary distributors available via CAN network) and the current that has to be provided to the solenoid for the dynamic variation of the Pump Margin.

The VPM dynamic strategy, allows to obtain a continuous variation of the Pump Margin as a function of the required flow and pressure from by the users.

The dynamic variation of the pressure, allows to increase the energy saving of the system in all the duty cycle stages, immediately adapting the Pump Margin to the new conditions of load.

Despite this solution seems quite promising, the minimum level of Pump Margin assignable in dynamic conditions, is limited by the presence of the priority valve. Hence a combined optimization process will have to be developed to obtain the maximum energy saving contribution from this solution.

Electronic Displacement Control - EDC

A further alternative solution related to fluid power supply group, is based on the use of bigger pumps coupled with the internal combustion engine, in order to being able to deliver the same flow at lower speed, thus optimizing the pump-engine coupling and having the chance of a more flexible management of the power and torque characteristics of the engine.

To do this, an electro-hydraulic valve is introduced, able to set a limit of the maximum swash plate angle of the axial piston pump. The valve is screwed directly into the seat of the actuator control of the swashplate and through the regulation current allows to vary the maximum value of the stroke of the swash plate, thereby adjusting the maximum flow supplied by the main pump.

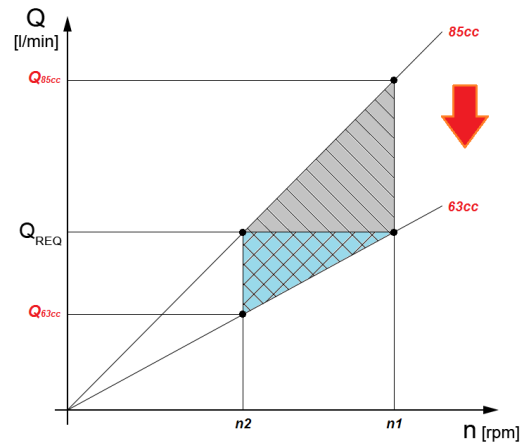


Figure 5 Electronic Displacement Control Concept

Using this concept, the EDC solution allows to obtain the same flow, delivered by the power supply unit, at different speeds of rotation of the combustion engine connected to the power supply.

Setting the level of flow in the standard conditions, a rotational speed n_1 of the pump is obtained. Following the linear characteristics of the machines, the required flow Q_{REQ} can be achieved at a lower engine speed, equal to n_2 , using the machine with higher displacement and the EDC valve.

This may for example allow to switch from the maximum power to the maximum torque operating condition.

4 Duty

Thanks to the collaboration between the University of Modena and Reggio Emilia and the manufacturer CNH Industrial S.p.a, collection of data from the field has led to the definition of a duty cycle that describes the usage of the tractor during its average life, identifying four main phases of work: transport, standby, work in low speed & work at high speed. With reference to the tractor life, the duty cycle defines four main phases of work that occupy the whole lifetime of the tractor as described in the following: 25% transport, 20% standby, 25% work with low engine speed and 30% work with high engine speed. The following subsections will describe the details of how each work phase has been defined and then simulated with the model.

Transport

In this phase the portions of the hydraulic circuit that are working, beside the fluid power group, are the steering system and eventually the trailer system; the value of pressure and flow rate taken as reference in each working phase are non-dimensioned with respect to the maximum pressure and flow of the hydraulic system:

- steering phase: the driver acts continuously on the steering to make corrections of the vehicle trajectory while driving. For this reason, the steering is quite stressed and it is considered that its usage covers the whole transport phase (100%). The operating conditions in terms of flow rate and pressure level at the steering are: flow rate 14 % Q_{max} ; pressure 24 % p_{max} .

- trailer system: during transport, a trailer may be attached to the agricultural tractor. From experimental data however the usage of the trailer is quite limited and covers only 1.25% of the transport phase under operating conditions equal to: flow rate 21 % Q_{max} ; pressure level 87 % p_{max} .

The tractor speed during transport is considered varying between 12 and 40 km/h.

Standby

In this work phase actually the system is not working with the hydraulic actuators, the axial piston pump is at its minimum displacement, delivering only the flow needed to compensate leakages and to allow regulation of the flow and pressure compensators. The operating conditions in terms of flow rate and pressure level are: flow rate 4% Q_{max} , Pressure 12% p_{max} . The engine speed is at its minimum value.

Work at high engine speed

The work phase at high engine speed involves the simultaneous use of the hydraulic actuators controlled by the auxiliary distributors and the steering system:

- steering: in this case the tractor is working on field, hence the contact between the tires and the ground is significantly different from the transport phase. As a consequence, the duty cycle involves higher pressure level and flow rate as operating conditions: flow rate 14% Q_{max} ; pressure 38% p_{max} . The usage of the steering system occupies the 50% of the phase of work at high engine speed.

- auxiliary users: the auxiliary distributors are ones of the most dissipative components of the system and are used for approximately the 7.5% of the phase of work at high engine speed, also considering the simultaneous working of more than one

actuator. The operating conditions are: flow rate 100% Q_{max} ; pressure 48% p_{max} .

The tractor speed during this phase is varying between 5 and 14 km/h.

Work at low engine speed

Also in this phase, a simultaneous usage of auxiliary distributors and steering is considered but the operating conditions are changed.

- steering: the use of the steering system covers the 20% of the phase under the following operating conditions: flow rate 14% Q_{max} ; pressure 48% p_{max} .

- auxiliary users: the auxiliary distributors work occupies the 7.5% of this phase, the operating conditions are equal to the previous ones: flow rate 14% Q_{max} ; pressure 48% p_{max} .

The tractor speed during this phase is varying between 0.05 and 5 km/h.

The Figure 6 shows the duty cycle subdivision just described.

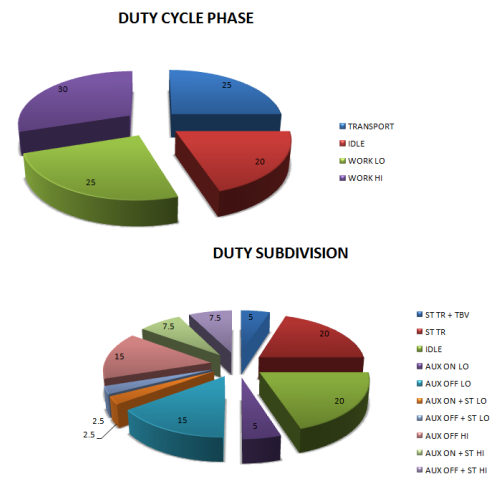


Figure 6 Duty Cycle specification

5 Results

The alternative configurations introduced in the previous sections have been modelled in LMS Imagine.Lab AMESim environment and compared with the standard configuration. The comparison has been done using the duty cycle previously described, chosen accurately and representative of the average life cycle of an agricultural tractor.

Results are presented making reference to the power consumption at the fluid power generator group shaft in percentage, viewed as the power consumption of the alternative architecture with respect to the power consumption of the standard architecture (100%); the power saving is the difference between the percentage power consumption of the standard version and the percentage power consumption of the alternative architectures.

Variable Pump Margin - VPM

The power saving achieved by the VPM fixed point strategy in the standby phase is very low and does not justify the installation on the vehicle.

As can be seen from the Figure 7 the power saving achievable by this configuration is about 0.13%, which is irrelevant for the chosen duty cycle.

This is due to the fact that in the standby phase the flow discharged by the pump is low and the incidence of lowering the pump margin is very little.

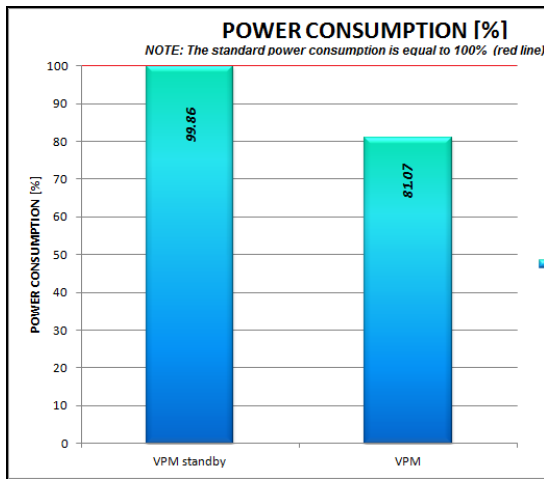


Figure 7 Power consumption: VPM fixed point strategy

In order to increase the amount of power saved, the best solution relies on using the control strategy with dynamic variable pump margin. To show this, a duty cycle where only the auxiliary distributors are working has been supposed. The auxiliary distributors have been proportionally opened in order to gradually rise the flow request till the maximum value. the Figure 8 shows the correspondent percentage of power saving as function of the flow rate.

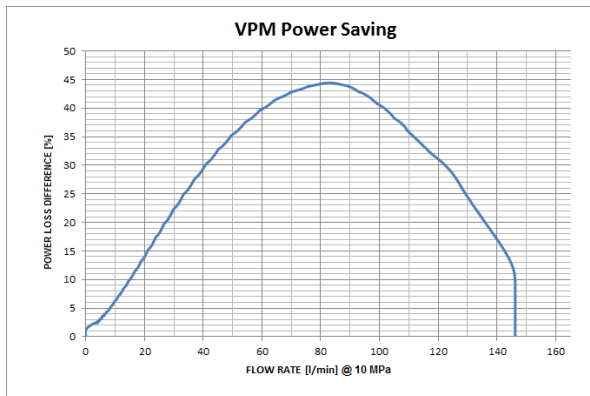


Figure 8 VPM Dynamic Solution

Using the linear mapping performed in the control unit (ECU), the maximum available power saving is achieved at intermediate flow rate request.

For high flow rates all the available pump margin pressure is needed and therefore there is no possibility of obtaining a benefit with respect to the standard configuration.

With reference to the duty cycle defined previously, Figure 9 shows there is no appreciable difference between the control strategy with VPM fixed point (VPM standby) and the standard version, while obtaining a power saving of 19% in the case of using a dynamic VPM adjustment.

During the transport phase, the pressure and flow rate requests are intermediate, so it is possible to achieve the maximum power saving, as shown in Figure 9.

In all other work phases, pressure and flow rate levels are such as to minimize the power saved adopting of the VPM strategy.

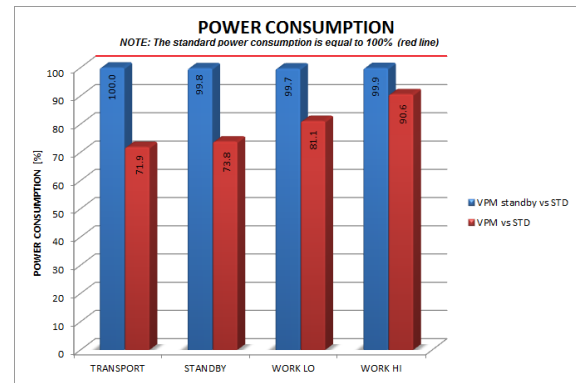


Figure 9 Power consumption during tractor lifetime: VPM Solution

The EDC strategy allows to introduce in the system an additional degree of freedom in order to provide the efficiency of the internal combustion engine-pump coupling. The user may decide to limit the displacement of the main pump in order to get the same flow rate of working fluid at different engine speeds. Generally, the introduction of a larger pump which rotates at a lower engine speed, to keep fixed the flow, involves a reduction in the efficiency of the same pump. The lowering of efficiency of the main pump can be considered relatively low compared to the possible benefits coming from an optimized management of the engine speed. However, with this particular duty cycle the power saving was about 1-2%.

6 Conclusions And Future Work

The motivation of this work derives from the need to reduce energy consumption in hydraulic circuit of agricultural tractors. To do this an intense research work to evaluate the actual situation has been carried out and from the results obtained suggestions for the improvement of the hydraulic energy production and usage in the vehicle are expected. In the target vehicle analyzed the hydraulic circuit has a typical hydraulic load sensing multi-users architecture where hydraulic power is provided by a group of fixed and variable displacement pumps and in this paper the attention has been focused on the variable displacement pump. A detailed lumped parameter model has been developed for the whole hydraulic circuit of the tractor, focusing in this paper in particular on the analysis of the pumps together with the flow compensator. The study of the standard hydraulic layout highlighted that many critical issues are represented by the excessive pressure drop across the valves and an improper management of the pressure levels between the main pump and the principal loads. Some alternative solutions are hence suggested. The standard architecture of the fluid power supply group in the tractor has been compared in terms of power consumption with some simple alternatives which involve the implementation of an electronic control of the variable displacement pump. In relation to the fluid power supply unit, the alternative solution proposed is based on the analysis of the pump margin pressure of the axial piston pump. The “intelligent management” of this additional pressure between pump and load, leads to an energetic improvement of the entire system: adopting a dynamic variation of the pump margin as function of the requested flow rate by means of an electronic flow compensator, it is possible to achieve the maximum energy saving.

With the aim of defining a term of comparison of the standard and alternative architectures, an actual duty cycle has been defined considering the entire life of the vehicle.

From the comparison, depending on the type of the flow rate request and following a linear characteristic within the ECU, it is possible to reach the maximum energy saving for the intermediate pressure and flow rate levels by using the Dynamic VPM strategy (19%).

In case of use of the Dynamic VPM strategy a critical aspect is the difficulty in some operating conditions to ensure the correct flow rate levels: to solve this problem a new architecture, involving a flexible priority valve, able to maintain the correct pressure levels and flow rate levels request by the users, is under investigation.

These alternative solution of the fluid power supply unit will be experimentally tested on a prototype vehicle thanks to the manufacturer CNH Industrial SpA.

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USING OF WASTE HEAT OF INTERNAL COMBUSTION ENGINES AND DRAFT OF EXHAUST GAS HEAT EXCHANGER.

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Abstract: Article discusses about the use of heat exchangers for stationary combustion engines and cogeneration units. The paper is dedicated to the problem of unused thermal energy in stationary engines. It analyses possibilities of accumulation of heat energy and its possible application in various fields. The paper deals with the classification of heat exchangers and with the subsequent description of design solutions of heat exchangers types used in given field. Resolves draft of exhaust gas heat exchanger according to the required parameters of internal combustion engine and subsequent simulation in the simulation program Comsol multiphysics to verify the correctness of the design and the construction solution of the exhaust gas heat exchanger for stationary combustion engines.

Keywords: HEAT EXCHANGERS, COGENERATION UNIT, WASTE HEAT, COMBUSTION ENGINES, DRAFT, SIMULATION

1. Introduction

Nowadays if we omit alternating economic crisis we can talk about ecological time. Political thinking towards just environmental but also economical, gives new insight into the lifestyle and comfort of man. A great impact just on these aspects has energetic. It is due to the increasing energy demands of human society, on which depends in no small measure the environmental burden and efficiency of energy use.

Possibility how to reduce energy consumption, is the way of savings. Reduction in fuel consumption can be utilized in a direction, which deals with the production of several types of energy, and possibly also of the products from the primary source at the same time. To this category can include cogeneration, trigeneration and polygeneration. Find a use for the heat is not as easy as in the case of electrical energy. But nevertheless is being offered several options, such as use of heat for hot water or direct water heating and its subsequent use for houses or large objects, depending on the performance of the cogeneration unit itself. Another option would be to use the absorption unit to transform heat to cold, making it possible to extend services to the production of cold water, for example for supply of air conditioners.

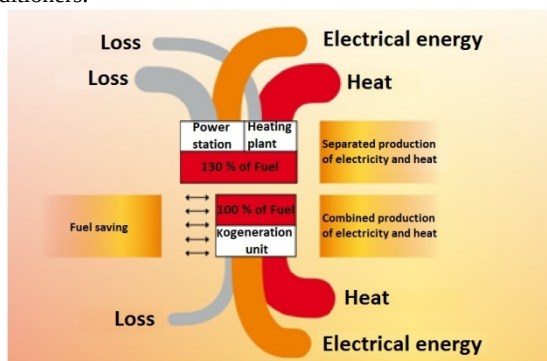


Fig. 1 Cogeneration principle

For all these systems the energy transformation is decisive the method how to submit it. For this intention in case of heat is used inseparable part of most of the systems which is called the thermal coupling node. Thereby may be various types of heat exchangers, coolers, condensers etc. The most common devices nowadays belong heat exchangers. In this case, for the generation of thermal energy from the exhaust gas and its subsequent use in other applications.

2. Use of heat exchangers in cogeneration units and stationary internal combustion engines.

For use of stationary internal combustion engine to generate electricity, or in other applications, arises a waste heat [1]. In most cases, this heat is not used in any way, but today's time more and more forcing producers and consumers to invest in technology that can leverage the potential of unused energy and contribute to cost saving. To this end has started to use exhaust gas heat exchangers. An exhaust heat exchanger is positioned on the exhaust pipe, removing heat flue gases, which could then be used for various applications.

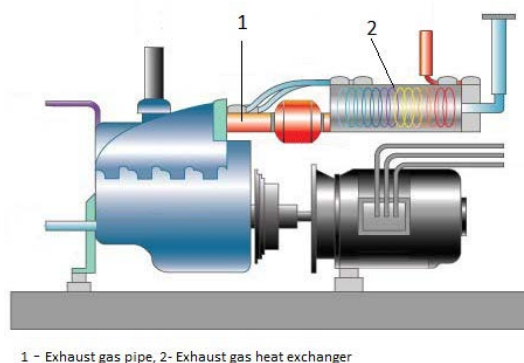


Fig. 2 Position of exhaust gas exchanger in stationary combustion engine

Exhaust gas temperature at the start of the exhaust pipe is in the range 500-700 °C. This means that the exhaust gases offer a great potential for utilization of waste heat. The exhaust gases in the most of cases heat up liquid, which can subsequently be used in several ways.

3. Heat exchanger

Device used for targeted transfer heat energy from the one heat medium to another one, according to the second law of thermodynamics, is called a heat exchanger [2]. These facilities include a large group and can be found in many sorts of systems without us realizing it. According to the purpose and primarily according to the action, which takes place in the heat exchanger can be divided into the condensers, evaporators, coolers, regenerative heat exchangers etc. Another division is quite normal according to the method of heat transfer, i.e. whether there is contact between the media etc.

Heat exchangers are divided into:

- Recuperative - media are separated by a solid impermeable wall and not coming into contact
- Regenerative - occurs periodically substituted flow heating and cooling media in the defined area.
- Contact - media come together for some time in contact without chemical reaction, and then are separated.
- Mixing - media are in a certain place mixed and continuing as a mixture.

The most commonly used type of heat exchanger is recuperative. This group primarily include tubular and plate heat exchanger. From the point of view flow is the most common counter-flow design, which results in better heat distribution than in parallel flow design.

Design with shaped tubes represent different tube axis arranged in the shape of a helix, spiral etc., located in the shell.

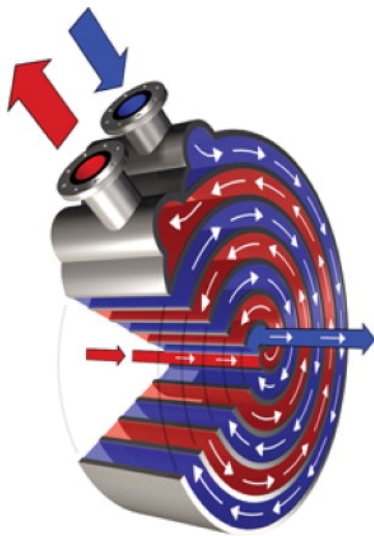


Fig. 3 Spiral tube heat exchanger

Exchangers in design tube in tube are among the simplest device in the above category. It's occurred like dismountable and non-dismountable which are exclusively for pure thermal media.



Fig. 4 Shell & tube heat exchanger

In general the tube exchanger with jacket is the most commonly used heat exchanger, where the main structure consists of a tube bundle placed in the shell of a cylindrical shape. These exchangers are manufactured at many different versions, depending on the configuration of inlet and outlet orifices, pipes, construction attachment of different thermal dilatation of tubes and plastics etc. This type of heat exchanger typically includes partitions that perform two basic functions. Arrestment of tubes resulting in a reduction of bending and vibration and also primarily direct the flow of media that is purposely altered to the cross-flow that increase the intensity of

3.1 Tube heat exchanger

In this type of exchanger, heat exchange takes place between the tube and the tubular-space. Tubular space normally consists of pipes or tubes of circular cross section, but we can meet with cross-sections of other shapes such as oval, square etc. To reduce the dimensional parameters of tubular exchangers can use all sorts of ways to increase the area of the pipe from the side of the pipe as well as the tubular space. For this purpose are used, for example ribs.

Tube heat exchangers can be divided according to the construction on:

- With shaped tubes
 - Tube in tube
 - Tube in the shell

heat transfer. The system also has the disadvantage that, with the inclusion of partitions create higher pressure drop.

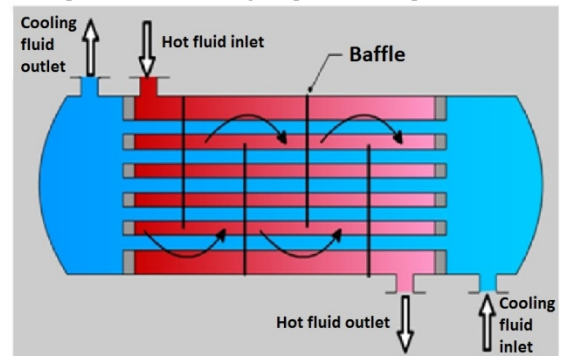


Fig. 5 Tube heat exchanger with baffles

Tube heat exchangers are characterized by good heat resistance and affordable price. However, their disadvantages are small compactness and high weight. The case of the pipes with small diameter, in which is the aqueous medium dirty it's expected gradual decrease of the cross-section pipe up to its complete clogging.



Fig. 6 Real construction of tube heat exchanger with baffles

3.2 Plate heat exchanger

Plate heat exchangers are based on a patent that has already been registered in 1878 by German inventor Albrecht Dracke. This principle, when one liquid cooling another liquid and liquids are flowing on both sides of group thin metal plates, became the basis for the construction of the heat exchanger - commercial plate pasteurizer Alfa Laval.

For more than 130 years was plate heat exchangers developed and structurally modified to devices that are used in thousands of different applications in all industries. Plate heat exchanger was previously designed for heating and cooling of the milk, but now

is commonly used for heating and cooling in industrial processes and it is the basis of air-conditioning in buildings or it provides heating of hot water for hundreds of millions of people.

This type of heat exchanger is characterized with a row lying plates which bear shaped reinforcements create turbulence of heat transfer medium and enlarge the heat-conveying surface. The heat transfer medium, as shown in the figure flows between the slabs of small thickness, whereby the heat is transmitted between substances mainly convective. Plate heat exchangers can be sorted into dismountable and non-dismountable. Non-dismountable exchangers are usually occur in the brazing or welding design, which can also be used in case of the aggressive heat transfer medium. For plate heat exchangers is a clear advantage of their higher performance per unit area, therefore low weight and small size which is for the same performance about 5 times smaller than in tubular heat exchangers. However, the benefits are offset by higher prices and demanding production technology.

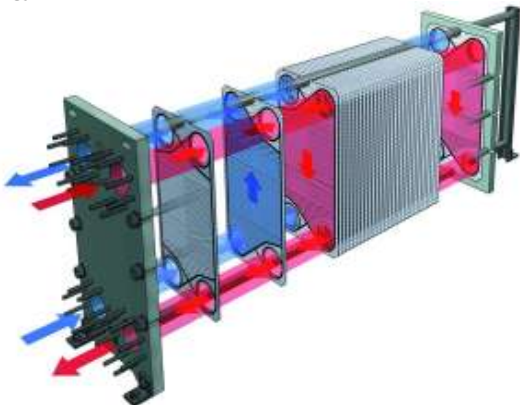


Fig. 7: Plate heat exchanger

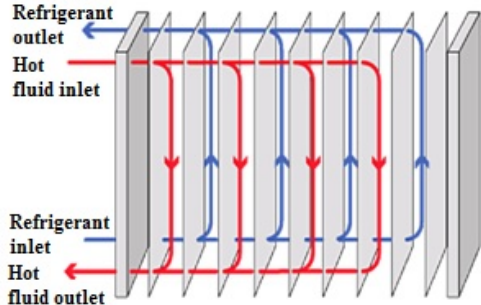


Fig. 8 Fluid flow in plate heat exchanger

4. Draft of exhaust gas heat exchanger

Draft of exhaust gas heat exchanger was realized according to these input parameters

Table 1: Input parameters for draft of heat exchanger

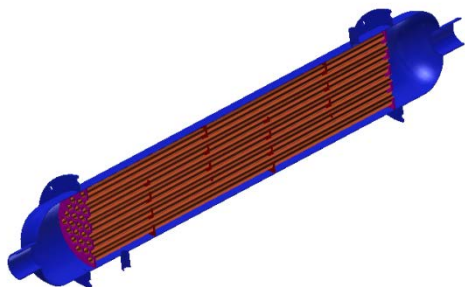
Required power	4,4 kW	Exhaust gas side		Cooling liquid side	
		Flow	82,5 m³/h	Flow	0,69 m³/h
		Inlet temp.	584 °C	Inlet temp.	89,5 °C
		Outlet temp.	≈115 °C	Outlet temp.	≈95,6 °C

Based on the input values I calculated the parameters of the flue gas heat exchanger which are as follows:

Table 2: Calculated parameters of the exhaust gas heat exchanger

Structural design	
Calculated power	4,8 kW
Number of pipes	48
Length of heat exchanger	1000 mm
Volume of heat exchange area	2,03 m²

Based on the calculated parameters I suggested and modelled flue gas heat exchanger.



9: 3D model of designed exhaust gas exchanger

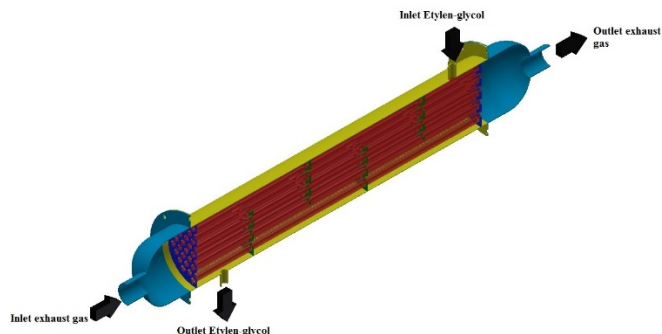


Fig. 10: Stream direction of working substances cross the heat exchanger
Stream direction of working substance cross the heat exchanger was chosen like a counter-flow how you can see in fig 10.

On the proposed exhaust gas heat exchanger was made simulations of flow of exhaust gases and heat transfer in a 3D simulation program COMSOL Multiphysics.

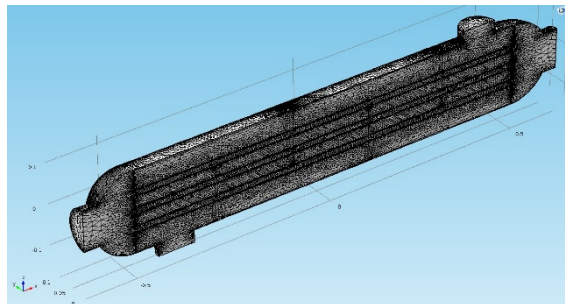


Fig. 11: Layering of the mesh in the model

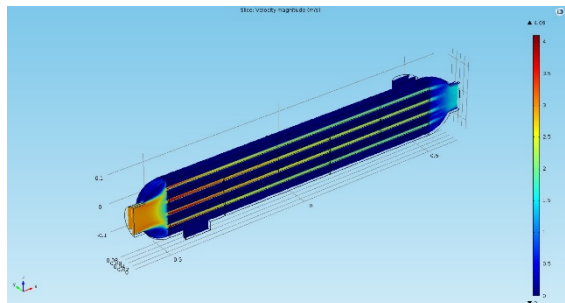


Fig. 12: Simulation of velocity magnitude of exhaust gas heat exchanger in m/s

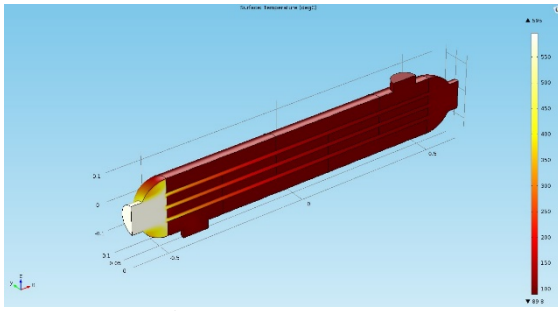


Fig. 13: Temperature of exhaust gas heat exchanger in degC

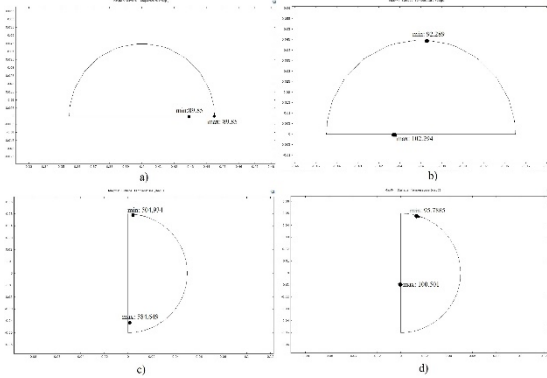


Fig. 14: Comparison of temperatures between inlet and outlet working substances

In Fig 13 we can see Comparison between inlet a) and outlet b) temperature of cooling liquid. Inlet temperature is 363 K (89.85 degC) and mean outlet temperature is 370 K (97 degC). Comparison between inlet temperature c) and outlet temperature d) of exhaust gas. Inlet temperature is 857 K (584 degC) and mean outlet temperature is 373.3 K (98.2 degC). The simulation shows that the draft is correct, because outlet temperatures of working substances are very similar to the required parameters. Cooling liquid side required outlet temperature ≈ 96.5 degC vs outlet temperature from simulation = 97 degC. Exhaust gas side required outlet temperature ≈ 115 degC vs outlet temperature from simulation = 98.2 degC.

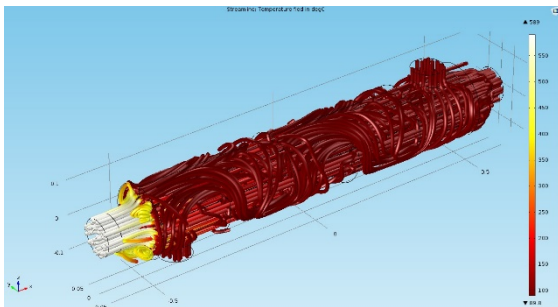


Fig. 15: Streamline in exhaust gas heat exchanger

5. Conclusion

The aim of my work was based on the input parameters to calculate and design a 3D model of the exhaust gas heat exchanger and create a simulation of the proposed model in COMSOL Multiphysics. The simulations were performed under conditions of the internal combustion engine warmed up to operating temperature. In the next plan will be made simulation under the condition of a cold combustion engine. After verifying the correctness of the draft by created simulations will be made the real model of exhaust gas heat exchanger. Subsequently will be performed the measurement of heat exchanger connected to the real combustion engine for the purpose of validate the correctness of the draft.

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MONITORING TECHNICAL CONDITION OF ENGINE BASED ON THE RESULTS OF DIAGNOSTICS TRIBOTECHNICAL

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Abstract: The article deals with monitoring of chemical elements during lifetime of military vehicles engine oil. Tribotechnical diagnosis is non-destructive and non-dismantling diagnostic method used lubricating oil as a source of information about the processes and changes in the mechanical systems to which it is applied. The paper deals with mathematical processing, tracking and analyzing data obtained from measurement tribo - diagnostics. For the analysis were used following methods AES and FTIR to monitor chemical elements, additives and contamination in engine oil. The increase in the metals and reduce the amount of additives were specially observed. The results were modelled using mathematical models to the value of the correlation coefficient R. The models can be used to predict the engine oil in the same or similar aggregates petrol engines.

Keywords: TRIBODIAGNOSTIC, ENGINE OIL, WEAR, SPECTROMETRY, LIFETIME

1. Introduction

Traditional reputable methods and special tribodiagnostic methods are used for monitoring the operating degradation of oils. Determining wear of mechanical systems lubricated by oil is based on knowledge, that oil shows a certain percentage of foreign matter after some operating time. Especially metallic abrasive wear (wear particles), that are dispersed in oil and that after the quantification by some suitable method allows indirect monitoring of mechanical changes in the system that uses the oil. It is then possible to draw specific conclusions based on the determined amount of metallic abrasive wear, growth rate, shape, morphology, size and material composition. If the growth and other parameters are in accordance with nominal values stated for the given mechanical system, it is possible to legitimately deduce normal wear progress without an increased risk of system failure. An abnormal or sudden growth of metallic particles, or their size, indicates an exceptional process. From the size, shape, velocity of growth and other parameters it is possible to deduce the failure severity and necessity to take remedial measures. Knowing the material of all system parts lubricated and rinsed with oil, then according to the metallic abrasive wear type, it is possible to determine and localize the frictional pair with the rapid increase of degrading wear. If it is not possible to localize the position of the increased abrasive wear by this method, then some other technical diagnostics suitable for failure localization should be applied.

The effective tool to monitoring engine condition is analysing of engine oil. With increasing number of taken samples of engine oil increase the accuracy of analyses. The main emphasis is placed to the sampling apparatus – extraction injection, containers, etc. Selection of sampling point and interval of sampling is very important too. This problematic is describes in publication [13].

Extended lifetime of engine oils and downsizing volume of cylinders (with increasing power) are a new trend in these times. The lifetime extension is tempting for customer but for tribotechnic there are the problems. There are no problems for kinematic viscosity, density, and shear stress of engine oils but in used engine oils there are many of metal particles and other contaminants, which pile up in engine oil during lifetime. It can cause malfunction of the engine or even crash the entire system. All of the friction places are made from several metals. The most of them there are treated iron containing other metals, aluminium and copper components, some portion coated with a surface layer of another metal in order to increase the surface hardness and improve sliding properties, improved corrosion protection, etc. Therefore, we must also interested in other metals (except iron only), such as aluminium, copper, chromium, lead, tin, nickel, silver, etc. [3,5].

The other chemical elements, which we must monitoring in engine oil, these are chemical elements contained in oil additives.

These are detergents, dispersants, friction modifiers, viscosity modifiers, anti-freeze agents, antioxidants, substances affecting the freezing point, anti-abrasion additives, lubricating ingredients and some others. In most cases these are the elements molybdenum, phosphorus, boron, calcium, zinc, magnesium, etc. Similar theses have been stated in publications [8,9,12].

Oil analysis methods:

Atomic emission spectrometry (AES) is a method that uses arc or spark sources to get the oil sample into the gaseous state and atomize it. As a result of atomic collisions or energy quantum absorption, the electrons of individual atoms are transiting from the ground state to the excited state. During the transition back to the ground state, atoms emit energy that equals the proportion of the energy levels in question in the form of luminous energy. The wavelength of light value is specific for each element [4].

Fourier transform infrared spectroscopy (FTIR) is an analytical method designed primarily for identification and structural characterization of particularly organic compounds. Infrared spectrometry measures the absorption of infrared radiation of different wavelength by the analyzed material. The analytical output is the infrared-spectrum that is the graphic representation of energy dependence functions on wavelength of the incident radiation. During the Fourier transform infrared spectroscopy the interferometrically acquired signal is transformed by the Fourier transformation to the infra-red spectrum [4].

2. Objects of diagnostics and diagnostics methods

The passenger car Renault Scenic I with gasoline engine has been used. This car have gasoline engine with 1,600 cm³ cylinders capacity and the power 79 kW. The year of made is 1999.

Table 1: Intervals of collect samples of engine used oil

Number of sample	Date of sample taken	Raid of oil, km	Raid of car, km
1	3. 3. 2012	0	171,790
2	4. 3. 2012	20	171,810
3	16. 3. 2012	1,737	173,527
4	28. 3. 2012	3,097	174,887
5	7. 4. 2012	4,462	176,252
6	23. 4. 2012	6,053	177,843
7	9. 5. 2012	7,550	179,340
8	28. 5. 2012	9,104	180,894
9	13. 6. 2012	11,027	182,817
10	27. 6. 2012	12,079	183,869
11	12. 7. 2012	13,815	185,605
12	25. 7. 2012	15,108	186,898

Commercially available automotive engine oil Castrol Magnatec 10W-40 (ACEA A3/B3) has been used. The lifetime interval of this engine oil is 15,000 km (according to the manufacturer's recommendations). The samples of used engine oil have been always taken after 1/10 of lifetime interval of engine oil – thus means 1,500 km. The season was spring and summer. The exact intervals of samples taken are reported in Table. 1.

For a more accurate evaluation of the chemical composition measurements have been samples of used engine oils always compared with the results of samples of new (unused) engine oil.

Diagnostics methods

With the chemical analyses it has been founding metals and additives in the new and used engine oils.

Determining the chemical composition of oils has been measured using Spectro Q100, which is a completely solid state spectrometer, specifically designed for the analysis of oil samples. With this spectrometer we can measure trace levels of elements dissolved or deposited as fine particles in mineral or synthetic oil-based products using long established and reliable technique with rotating disk electrode. The device meets the requirements of ASTM D6595 standard method for the determination of wear metals and contaminants in used lubricating oils and hydraulic mixtures. Increase metals content and decrease additives content have been modeled using linear function as in publication [6].

Determining the oil degradation and contamination of oils has been measured using Spectro FT-IR Oil Alpha Analyzer is specifically designed for the molecular analysis of lubricating oil to determine oil degradation and contamination. It is used for predictive maintenance programs according to JOAP, ASTM E2412 and DIN standards for the rapid determination of oxidation, nitration, sulfation, water, coolant, fuel dilution, soot, wear additive depletion and synthetic lubricant breakdown in used lubricating oils.

3. Oil field data assessment

By the spectrometry have been determined the chemical composition of samples of the new and used engine oil. The samples of used engine oil have been compared with samples of new engine oil same specification. It is important to the finding of the on-going state of degradation [10].

Determining the engine oil degradation oils has been measured using FTIR methods. It is used for determination wear additive depletion in used lubricating oils. On the Fig. 1 are showed the difference between new and used motor oil. The difference is evident in the decline belt wavenumber of $1025 - 960 \text{ cm}^{-1}$, which displays anti-wear additive (ZDDP).

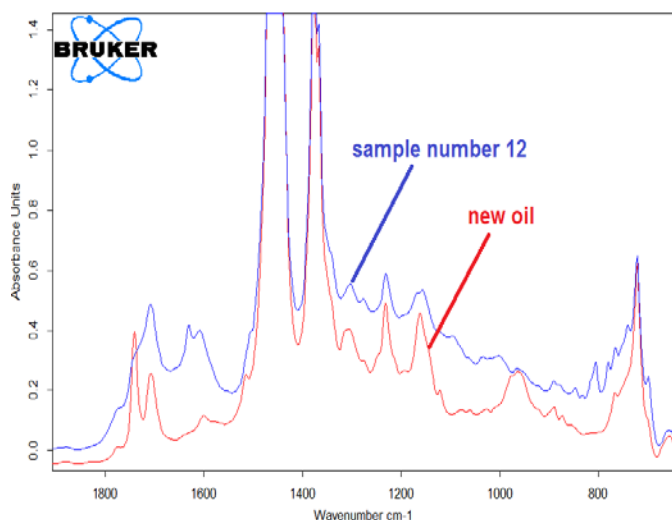


Fig. 1 The resulting infrared spectra for new and used oil

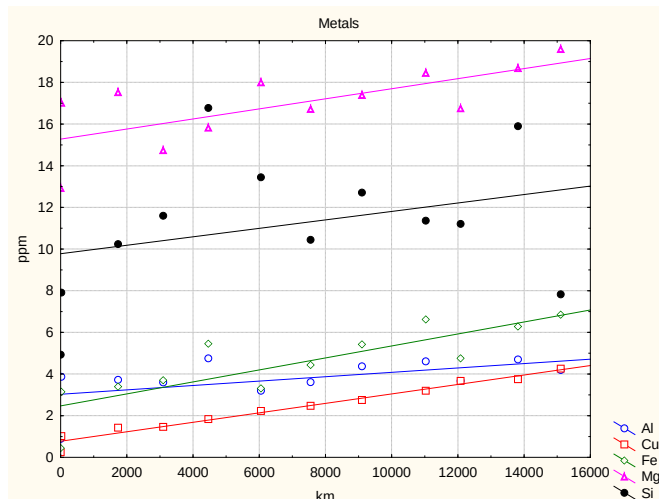


Fig. 2 Increase contents of metals in engine oil

On the Fig. 3 are showed decreases of some chemical elements using as oil additives. Manganic (Mn) and molybdenum (Mo) decrease slowly with raid but boron (B) decreases quickly with raid. Boron contains in anticorrosion additives.

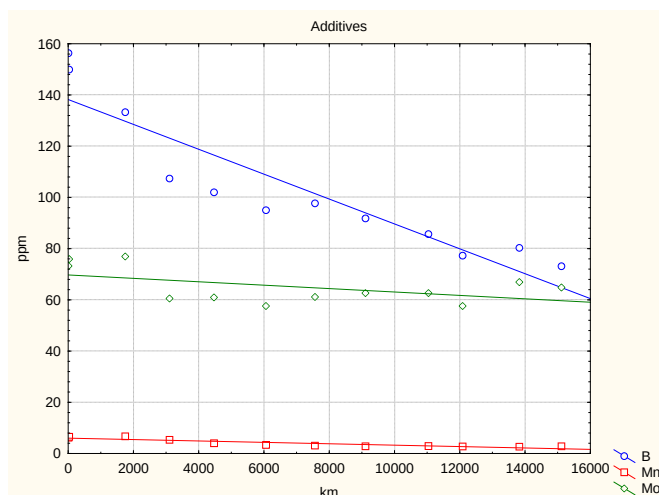


Fig. 3 Decrease contents of additives in engine oil

On the Fig. 4 are showed elements phosphorus (P) and zinc (Zn). Phosphorus is usually contained in oil additives and zinc is usually used as construction material of engine.

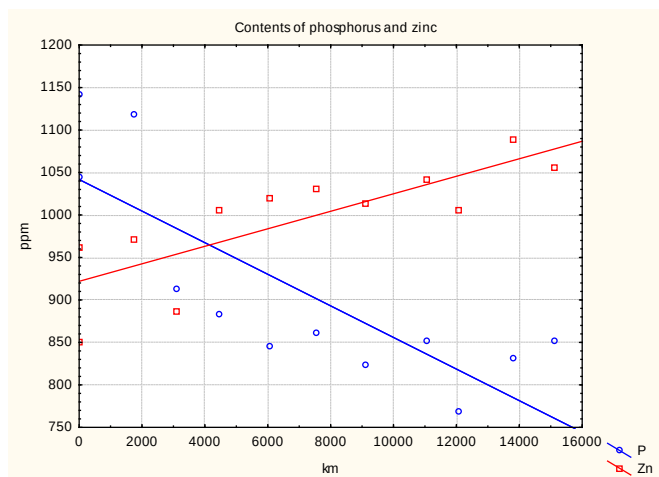


Fig. 4 Contents of P and Zn

To the modelling increasing and decreasing of metal and chemical elements has been used linear function with general formula:

$$y(x) = A + k \cdot x, \quad (1)$$

to calculate for example iron (Fe) state:

$$\text{contain of Fe} = 2,4714 + 0,0003 \cdot \text{raid [ppm; km]} \quad (2)$$

where we must respect the lifetime of engine oil 15,000 km, respective and maximal 30,000 km.

Correlation coefficients R achieve high values from the 0.79 to the 0.98. Similar values of correlation coefficient achieve authors in publications [1,11].

In the Tab. 2 are showed constant values A and k for all of used chemical elements.

Table 2: Constant values A and k

Chemical elements	Constant A	Constant k
Al	3.0318	0.0001
Cu	0.7738	0.0002
Fe	2.4714	0.0003
Mg	15.2793	0.0002
Si	9.7779	0.0002
B	138.0036	-0.0049
Mn	5.8424	-0.0003
Mo	69.5518	-0.0007
P	1041.6257	-0.0186
Zn	922.0322	0.0103

5. Conclusions

In this part of the paper we can state that use method (spectrometry) is as one of the basic oil analyses very suitable. The suitability rises when the samples of used engine oil are compared with the sample of new (unused) engine oil with same specification. Using spectrometry we have determined chemical composition of all samples of engine oil. We have also quantified individual chemical elements in the new and used engine oil. With graphs we can see increasing trends polluting elements (metals) and decreasing trends of oil additives (detergents, dispersants, friction modifiers, viscosity modifiers, anti-freeze agents, antioxidants, substances affecting the freezing point, anti-abrasion additives, lubricating additives and some other). All trends have been modeled using linear function. Linear function is basic mathematical model, which we can use. But linear function is very accurate, if we use it properly. Coefficients of correlation R have achieved high values – from the 0.79 to the 0.98. Created mathematical models can be used to prediction changes in lifetime of engine oil fill in same or similar petrol engine.

The aim of the paper is to shed light on the area of tribodiagnostics including the methods which are applicable and suitable for oil analysis. The data regarding lubrication fluid which is available due to performed analyses is a good source of information when considering the cost savings provided the oil is changed systematically. It would be also good to see the results of the analysis in a broader context as an interesting reflection of an actual state of a technical object from where the oil was taken. When taking into account the results of the tribological analysis, the cost savings might be manifested as the extension of oil changes

time and relating maintenance costs and downtime resulting from object unavailability by extraneous causes. Since there is a wide spectrum of suitable methods while analyzing an immediate state and prognosis (PHM – Prognostics and Health Monitoring), and because the area falls very deeply into interdisciplinary studies, the specification of relevant dependencies of the analysis results on a real technical state is not at all an easy task to do.

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THE GAS TURBINE ENGINES OPERATIONAL PERFORMANCES MATHEMATIC SIMULATION METHOD

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Abstract: The gas turbine engine operational performances definition method based on blade-to-blade mathematic simulation of multistage axial compressor and high temperature cooling transonic multistage turbine using flow paths and middle radius blades geometric parameters is shown. The method allows to take into account air bypass and bleeding from compressor for the cooling and its blowing into turbine and their influence on the engine parameters.

As example, attained by calculation way the results of turbo-shaft engine operational performances and air by pass bleeding from compressor influence on the engine specific parameters, efficiency and stability margin are represented.

Keywords: AVIATION GAS TURBINE ENGINE, OPERATION PERFORMANCE, COMPRESSOR AND TURBINE GEOMETRICAL PARAMETERS, AIR BY-PASS.

1. Introduction

In modern practical designing it is paid a lot of attention to gas turbine engine aviation, marine and stationary applications mathematical simulation, definition their gas and thermo dynamic parameters and operational performances [1-4]. The methods for engines working processes investigation will be considered in this article.

The most simple per-unit engine models which are founded on the "black box" processes description level are widely used.

In these simulations the main units (compressor, turbine, combustion chamber, etc.) performances are set as external data using experimental results or generalized dependences. They don't allow to take into account the influence of many significant for engine performances factors. This article deals with the developing of gas turbine engines stationary process simulations and performances. Three types of engine models will be considered and compared.

2. Theoretical model

Schematic representation of dual-rotor turbo-shaft gas turbine engine in per-unit modeling level is shown on Fig.1a. Such type of models and according calculation techniques are widely used in gas turbine engines design bureaus.

Their application allows to definite the engines operation performances such as thrust (or power), specific fuel consumption, efficiency and other parameters as the dependences of the flight high and speed, type of fuel, control system and so on. However, this approach doesn't make possible to analyze of the geometrical parameters modifying influence, air pulling from compressor flow path and it's blowing into turbine stage for cooling, opening bypass valves and so forth.

To increase the methods accuracy and effectiveness the simulation should have higher level detailed of thermo and gas dynamic processes in engine main units, which will allow to take into account their geometrical parameters, air bleeding and blowing.

The air bleeding from compressor path for different needs or its bypassing from the valves at low rotation speeds plays the significant role in engine operation and influence its performances.

So the developing of the first difficulty level engine simulation Model A (Fig. 1a) and its modernization to the simulation founded by multistage compressor blade-to-blade description made possible to propose Model B.

This simulation is shown on Fig.1b. It considers the compressor flow path and middle radius blades geometry.

So Model B allows:

- to estimate the blades and flow path geometrical parameters changing influence on the engine operational performances;
- to check optimal values of variable guide vanes stagger angles for designing and modernization engine control system;
- to consider by-pass flow rate from the compressor flow channel and its influence on the engine parameters more accurate than if the Model A would be used;

- to estimate the compressor surge margin and identificate the stage, which is the source of the stall phenomenon on various rotational speeds.

The Model C is further development of calculation technique, which includes modernization of the multistage high temperature cooled turbine description accounting flow path and middle radius blades geometrical parameters. Fig. 1c illustrates Model C graphically.

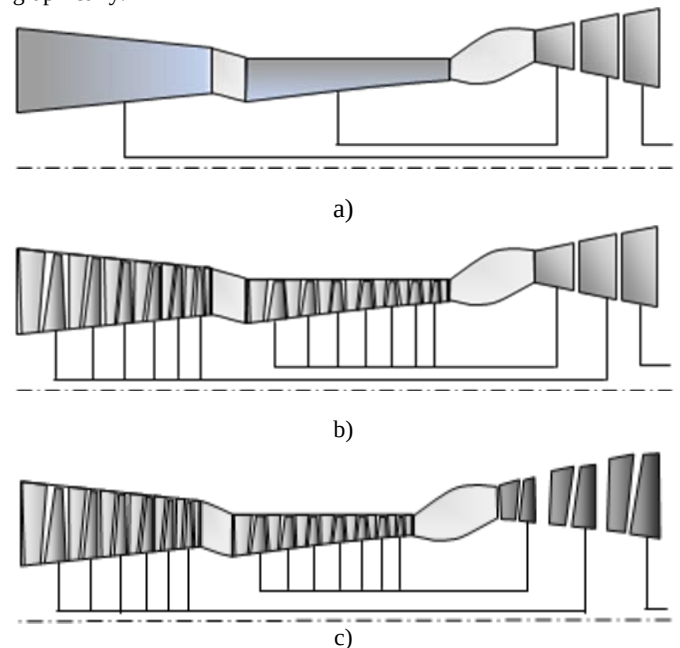


Fig. 1 Gas turbine engine simulation schematically representation
a) Model A; b) Model B; c) Model C

In addition to simulations possibilities previous Model C allows:

- to estimate blowing from the cooling system in turbine gas path air parameters and their influence the operation performances of GTE, especially its mass flow rate, temperature and other;
- to harmonize more accurately air flow rate taking away from compressor path for cooling of engine hot elements and air blowing in turbine stage on various regimes;
- to decrease the number of correction coefficients for identification mathematic model.

In fact, Model C has more possibilities then it was named above.

The developed calculation technique has the modular structure, which is opened to stocking. The modern version of software includes the second order difficulty compressor and turbine modules, which accounted their geometry and the first order simulation of combustion chamber, inlet diffuser and exit nozzle modules.

For stationary operation regimes the equations system includes the engine units working conditions with taking into account blade-to blade multistage turbine and compressor discretization and allows to calculate flow parameters in the engine flow path. The engine

parameters cooling system, control program and environment conditions are set also.

Equations system combines mass flow rate and power balances for engine main units, turbine and compressor stages with accounting of air bypassing, bleeding and blowing.

Flow parameters are calculated by using system of turbo machines gas dynamic equations in one-dimensional statement. To accounting viscosity effects are used the published generalized dependences, which were tested in our laboratory.

In addition to data, which are setting for per-unit engine simulations, Model C requests compressor and turbine flow path and middle radius blades geometric parameters.

3. Practical Application

As the developed method practical application example, let us consider twin spool turbo shaft engine performances in wide range of operation conditions including the cases, when by-pass valves are opened. These regimes take place in gas turbine power plants at low level of rotating speeds.

Low pressure compressor performances attained by calculation by way as the pressure ratio and isentropic efficiency dependences from inlet air mass flow rate and rotational speed are shown on Fig. 2.

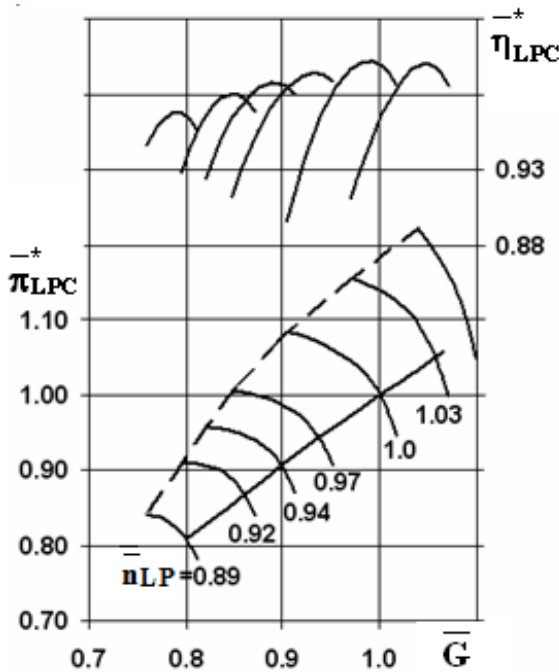


Fig. 2 Low pressure compressor pressure ratio and efficiency dependences by inlet flow rate and relative rotational speed

All parameters are represented in dimensionless form and divided by design point magnitudes. This map comparison with experimental data gave a good agreement. Fig. 3, 4, 5 illustrate high pressure turbine performances in the form of dependences mass flow rate parameter, isentropic efficiency and turbine absolute exit flow angle from turbine stage pressure decreasing ratio π_{ST}^* and rotational speed λ_{u1} with accounting the air blowing in turbine stage from cooling system.

These turbine and compressor characteristics as the curves aren't used in the engine performance calculations, and when the software calls corresponding module, only one point according to the operation conditions is calculated taking into account mass flow rate, rotational speed, flow angle and so on.

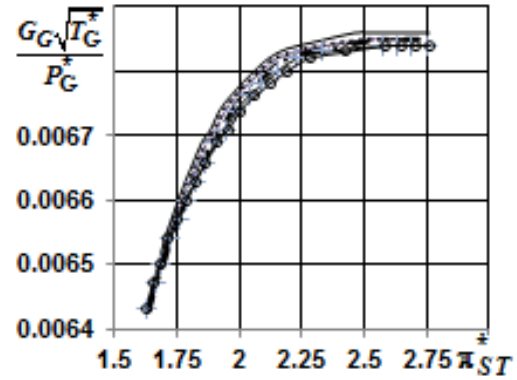


Fig. 3 Gas flow rate parameter dependences by turbine pressure decreasing ratio and rotational speed

--- $\lambda_{u1}=0,42$; — $\lambda_{u1}=0,50$; -□- $\lambda_{u1}=0,54$;
--- $\lambda_{u1}=0,58$; $\lambda_{u1}=0,62$; —+— $\lambda_{u1}=0,66$;
—○— $\lambda_{u1}=0,70$

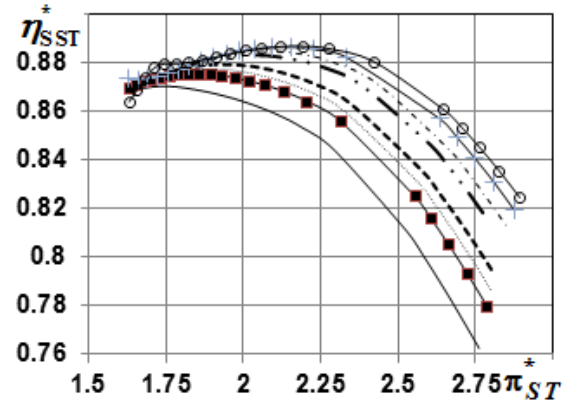


Fig. 4 Turbine isentropic efficiency dependence by pressure decreasing ratio

— $\lambda_{u1}=0,51$; —■— $\lambda_{u1}=0,54$; — $\lambda_{u1}=0,55$;
--- $\lambda_{u1}=0,56$; -...- $\lambda_{u1}=0,58$; —+— $\lambda_{u1}=0,62$;
—○— $\lambda_{u1}=0,65$

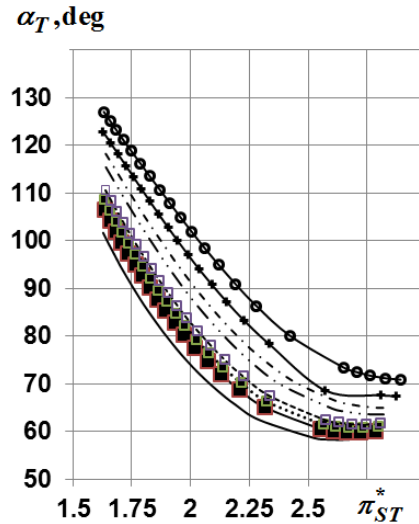


Fig. 5 Turbine stage absolute velocity exit angle dependence by pressure ratio decreasing

— $\lambda_{u1}=0,51$; —■— $\lambda_{u1}=0,54$; — $\lambda_{u1}=0,55$;
--- $\lambda_{u1}=0,56$; -...- $\lambda_{u1}=0,58$; —+— $\lambda_{u1}=0,62$;
—○— $\lambda_{u1}=0,65$

It is seen from Fig 5, that turbine operation conditions changing leads to varying gas flow parameters in wide limits, which is affected the next stage working regime. Only blade-to-blade turbo machines flow description gives the possibility to account its influence.

The stages reciprocal effects might be significant. As example, let us consider the turbine nozzle vanes incidence angle impact.

Fig 6 demonstrates the isentropic efficiency varying for different values of incidence angle. As follows from this figure turbine stage efficiency changes by 1-2% and that factor influence engine operational performances significantly.

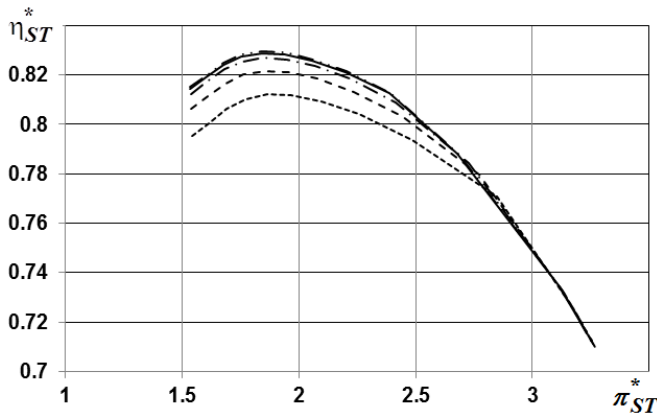


Fig. 6 The turbine stage nozzle vanes incidence angle influence isentropic efficiency

— — — — — $i = +40$ deg; — — — — — $i = +20$ deg; ······ $i = 0$ deg;
— · — · — $i = -20$ deg; — — — — — $i = -40$ deg;

The blade-to-blade discretization of gas dynamic processes in multistage turbine is the new part of presented gas turbine engine simulation. It allows to take into account not only the parameters of main gas flow, but also blowing from cooling system in gas path air parameters. Fig. 7 graphically illustrates isentropic efficiency by turbine pressure ratio decreasing dependences for different values of blowing air relative temperature. The air temperature magnitudes are divided by the incoming the turbine stage main stream gas temperature.

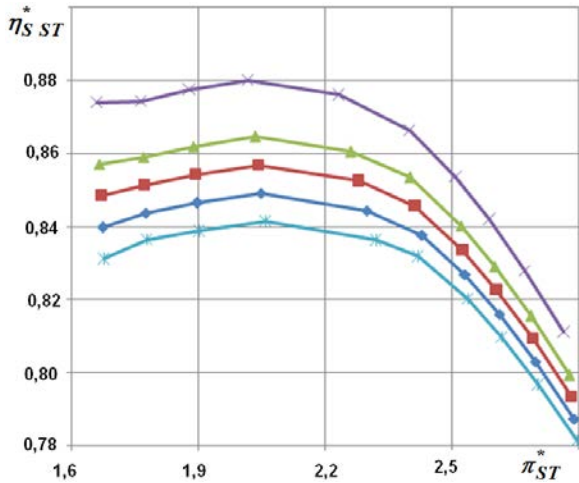


Fig. 7 Dependences the turbine stage isentropic efficiency by the relative temperature of blowing in the path cooling air

—*— $\frac{T_A^*}{T_G^*} = 0,3645$; —◆— $\frac{T_A^*}{T_G^*} = 0,4645$;
—■— $\frac{T_A^*}{T_G^*} = 0,5645$; —△— $\frac{T_A^*}{T_G^*} = 0,6645$;
—×— $\frac{T_A^*}{T_G^*} = 0,8645$

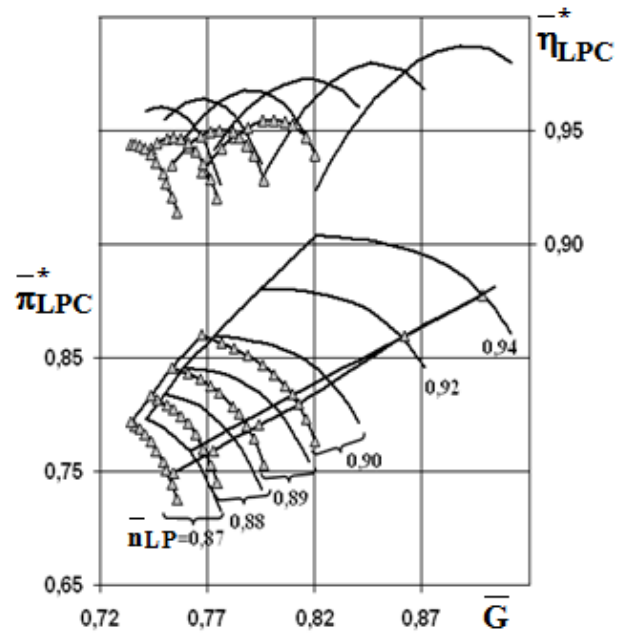


Fig. 9 The flow bypassing from compressor channel influence compressor performance

— — — — — valves are closed
—△— valves are opened

From Fig. 7 one can see, that blowing air temperature increasing leads to the isentropic efficiency value growth. In this case the thermo dynamic losses are decreased and efficiency of cooled turbine stage is raised. However, the effectiveness of blade metal cooling is diminished. That is the traditional question for designers.

Named above and many others turbine and compressor flow features influenced the gas turbine engine performances may be calculated with assistance of presented method and correspondent software.

Fig. 8 shows the engine performances as dependences of effective power and specific fuel consumption from the fuel consumption getting in calculation way in dimensionless form. If $\bar{G}_T \geq 0,7$ the bypass valves are closed. The most advantage of considering calculation method may be received from investigation of changing compressor and turbine blades geometrical parameters, stagger angles of variable guide vanes, bypass mass flow rate influence on engine operational performances, which was presented in [5, 6].

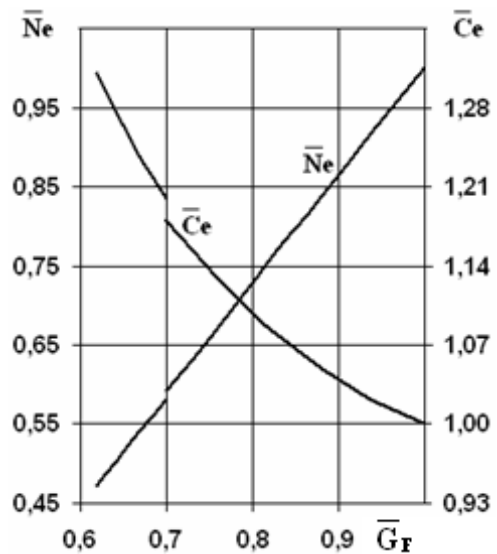


Fig. 8 Effective power and specifically fuel consumption dependences by fuel consumption

Further let us consider the opening of low pressure compressor bypass valves located above third stage influence the compressor map and engine performances. The relative mass flow rate bleeding from the compressor channel through the valves divided by the inlet mass flow rate equals 5 %.

As Fig. 9 illustrates, the valves opening leads to moving the performance curves in lower flow rate level and essential reducing efficiency of compressor. From Fig.10 one can see the blade incidence angle values for different stages of low pressure compressor.

When the valves are closed the first stages have the big values of incidence angles. The opening of the valves allows to reduce the angles and decrease the blade loading.

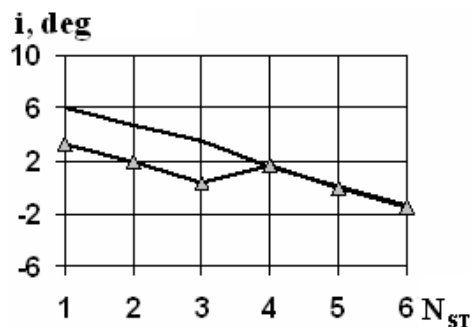


Fig. 10 The blades incidence angles for different stages of low pressure compressor, $\bar{n} = 0,9$ near the working line

— valves are closed
—△— valves are opened

As a result, Fig 11 illustrates the surge margin of compressor calculated by using of the described above technique. So, the bypass valves opening gives the possibility to increase the surge margin of compressor at low rotational speeds.

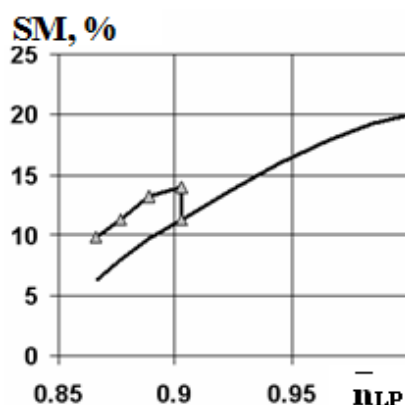


Fig. 11 Low pressure compressor surge margin dependences by its relative rotational speed

— valves are closed
—△— valves are opened

The surge margin growth decreases the engine integral parameters. From Fig 8 it could be quantity estimated the increasing of specific fuel consumption and reducing of power, while the bypass valves are opened.

4. Conclusions

The gas turbine engine operational performance investigation method founded on mathematic models of units and blade-to-blade description of the process in multistage axial compressor and high temperature cooled transonic turbine was presented. Its advantages and reasonable application for practical designing were demonstrated.

As it was shown, the developed method and software allow to estimate the influence of bypass mass flow rate on the engine operation performances and its value optimize for various operation conditions at low rotational speeds.

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METHODS OF AVIATION GAS TURBINE ENGINES COMPRESSORS COMPUTATIONAL INVESTIGATIONS

МЕТОДЫ РАСЧЕТНОГО ИССЛЕДОВАНИЯ КОМПРЕССОРОВ АВИАЦИОННЫХ ГАЗОТУРБИННЫХ ДВИГАТЕЛЕЙ

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Abstract: The prediction methods for calculating of different compressor objects developed at the department of aircraft engines theory of National Aerospace University named after N. E. Zhukovsky "Kharkov Aviation Institute" are presented. Results of computational studies for various compressor objects such as high-pressure fan stage, multistage axial compressors of GTE in a wide range of flow rates and rate of rotation modes and high pressure centrifugal stage with taking into account the air - bleeding and bypass from compressor setting are shown. The Mach number flow isolines near the stability boundary, on the line of operating modes and near the compressor locking are demonstrated. Numerical investigation results of sub- and transonic flow fields and summary performances in high pressure centrifugal stage are presented and the flow features in its elements and its modification results are examined.

KEY WORDS: NUMERICAL INVESTIGATION, PREDICTION CALCULATION METHOD, AXIAL COMPRESSOR STAGE, MULTISTAGE AXIAL COMPRESSOR, HIGH PRESSURE CENTRIFUGAL STAGE, FLOW STRUCTURE, OVERALL PERFORMANCES, BLADES MODERNIZATION

1. Introduction

At present, the study of gas-dynamic processes occurring in aviation gas turbine engines (GTE) compressors is impossible without using of modern flows calculation methods implemented in the respective software systems. The methods may have a different level of complexity: from methods for calculate a one-dimensional formulation flows with weight average parameters (1D) to methods for analyzing spatial viscous flow (3D).

For design and finishing of aircraft engines compressors and compressor control systems methods of flow testing calculation are widely used.

Checking calculations allow you to define the flow parameters and the total compressor characteristics as a whole and its individual stages, as well as to analyze stages work in multistage axial and axy-centrifugal compressors. Depending on the task designers may select and apply appropriate methods.

The most of compressor flow calculations are performed with using of 2D approaches with sufficient practice performance providing, which is especially important when necessary to determine the compressor characteristics or to conduct multi variant calculations.

The most adjusted and verified flow calculation packets are preferable for compressor designers/

Quasi-three-dimensional and two-dimensional approaches developed abroad and in the USSR [1 ... 8 et al.] are widely known. Now small number of organizations deal with new 2D programs creation and improvement, while 2D approaches are widely used for compressor design and are an integral part of specialized software systems (see eg, Concepts NREC et al.).

In Problem Research Laboratory of Gas Turbine Engines and Installations of the National Aerospace University named after N. Zhukovsky "Kharkov Aviation Institute" developed a programs complex for 2D analyze of flow structure and the total characteristics of axial and centrifugal high-pressure stages, as well as multi-stage compressors with air-bleeding and bypass from the air flow passage.

The numerical methods, which are the members of the complex, and some of the results of the study subsonic and transonic flows in compressors of aircraft engines are presented in the report.

2. 2D calculation of sub- and transonic flows

2.1 GTE compressors sub- and transonic flow calculation

The program complex for GTE compressors flows numerical simulation is based on a solution of the stationary Euler equations system using finite - difference method of the second order of accuracy [8]. The generalized coordinate system using allows to simulate the flow in the compressor flow path and axisymmetric channels of arbitrary shape. With the concepts of artificial compressibility the transition through the speed of sound stability increases. Averaged in a circumferential direction relative motion Mach number does not exceed $M_w < 1.4$, which in many cases is sufficient for practical compressor flow calculations. Averaging of obtained calculation flow parameters at different radii with using the laws of conservation of mass, momentum and entropy in the averaged and averaging flows provides the integral values of the flow parameters behind the blade rows, stages and compressor as a whole. Real flow properties are taking into account by using the generalized dependencies to determine the deviation angles and losses (Liblein, Swan, Al-Daini, et al.). The initial data includes geometric parameters of flow path and blade rows, operating parameters (flow rate, rotating speed), as well as the flow parameters radial distributions at the inlet of the computational domain (P_{H^*} , T_{H^*} , α - prerotation of flow in absolute motion). Geometric parameters of blade rows specified in the input data are translated to the estimated all the nodes of the grid.

2.1 2D flow multistage axial compressors and high-pressure axial compressor stages simulation

The calculation method is implemented in the software package AxSym, for calculating sub- and transonic flows in multistage axial compressors of aircraft engines and their components. The compressor gas channel (see Fig. 1 a) is divided by fine mesh, which is converted on a rectangle area (shown in enlargement in Fig. 1 b), which allows to describe the flow in the arbitrary shape parts, it is enough to accurately simulate the surface of rotor blades (RB) and the guide vanes (GV), to determine the flow pattern in interscapular channels and obtain the contours of the flow parameters.

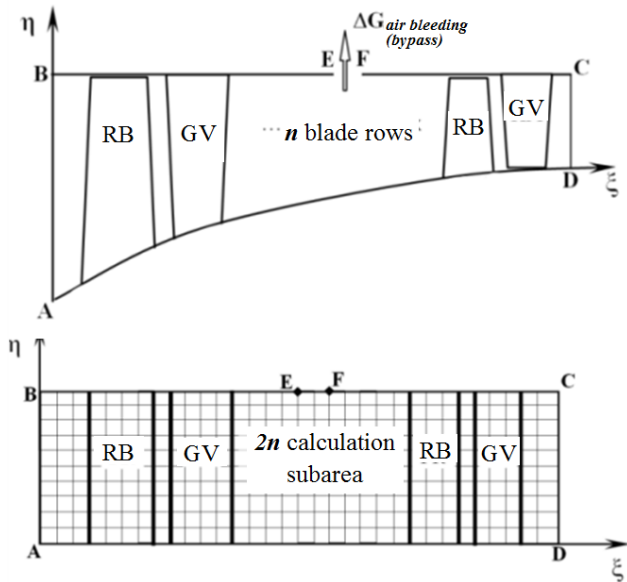


Fig. 1. Scheme of the computational domain

As an example Fig. 2 shows the total characteristic of multi-stage axial compressor of modern aircraft engine obtained with the original software package (SP) AxSym (solid lines), in comparison with the experimental data (markers).

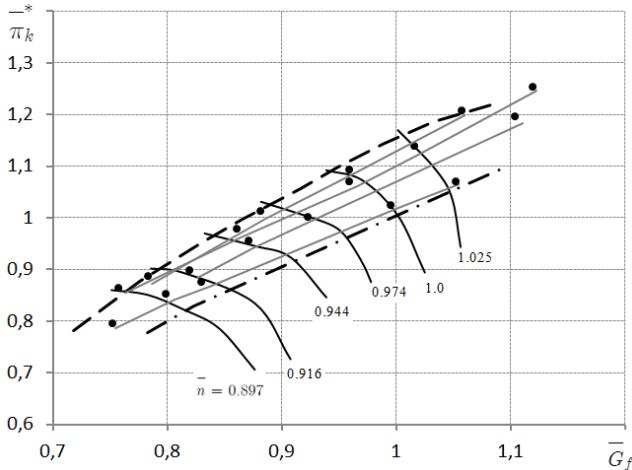


Fig. 2. The total characteristic of multi-stage axial compressor

The SP AxSym verification process for various GTE types demonstrated a satisfactory agreement between the results of calculation and experimental data [9, 10 et al.].

For the high-pressure axial compressor stage flow calculations the numerical method was supplemented by inter-blade rotor flow multi-shocks model with the relevant dependencies for the shock losses determination and the ability to use the profiles with arbitrary midline shape. All changes are implemented in the SP AxSym_M.

As an example of the SP AxSym_M verification fig. 3 shows the results of numerical simulation of total characteristics and flow structure in a high pressure ROTOR 37 [11, 12] (solid line) compared with experimental data at the relative rotor speeds $\bar{n} = 0.9$ and 1.0 (markers). At this figure the diagram beside shows the Mach number contours on the design point mode $\bar{n} = 1.0$, $G_f = 20.74 \text{ kg/s}$.

Figure 4 shows the radial distribution of the calculated relative Mach number at the entrance and exit of rotor (solid and broken lines) in comparison with the experimental data (markers).

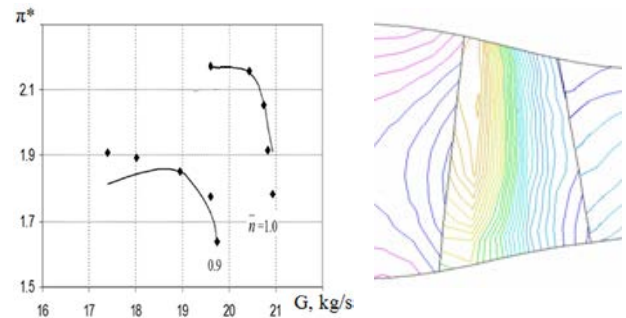


Fig. 3 ROTOR 37 total characteristic

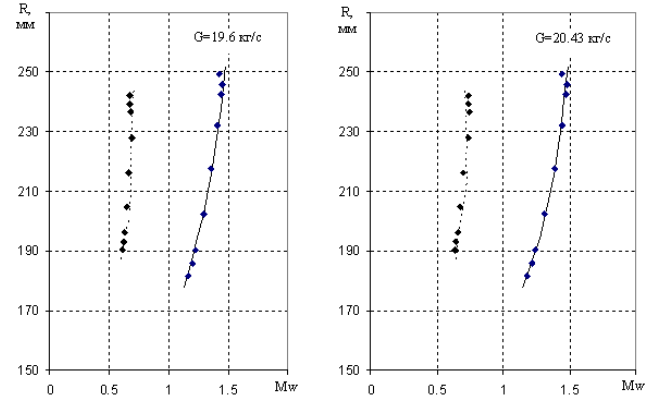


Fig. 4. Simulation of a ROTOR 37 to mode = 1.0

2.2 High-pressure centrifugal compressor stage flow modeling

Numerical simulation of a high-pressure centrifugal compressor stage is different from the calculation of flows in high-pressure axial compressor stages. Fig. 5 shows the design scheme and the corresponding computational domain.

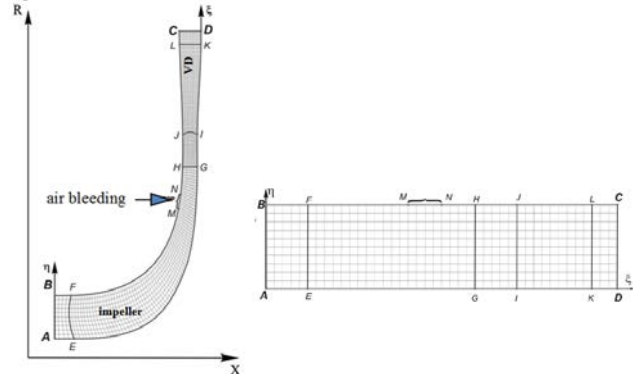


Fig. 5 Scheme of the computational domain

This approach is implemented in the SP AxCBm and applied for numerical simulation of sub- and transonic flows in the high-pressure centrifugal stage with the inlet flow rate $M_{w1} < 1.4$ [13].

As an example of the verification SP AxCBm Fig. 6 shows the total characteristics and Mach number isoline at the "design" mode for the high-pressure model stage obtained by calculation (solid line) compared with experimental data and calculations of other authors [14] (markers). Represented Mach number contour lines demonstrate the flow structure in the rotor blades channel with including the splitter blades.

3. Results and discussion

The process of designing and finishing of aircraft engine compressor can be based on the solution of the inverse prob-

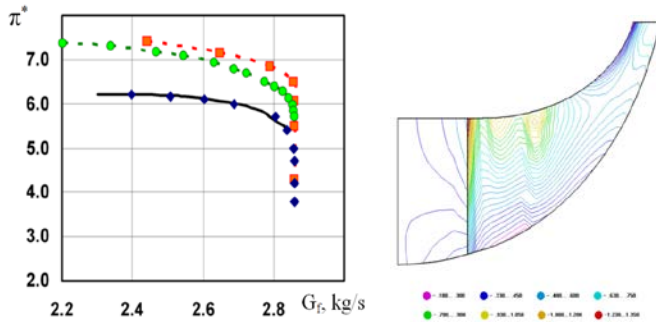


Fig. 6 Total characteristic of the model high-pressure centrifugal stage [14]

lem, which allows on the basis of given distribution of the flow parameters to receive directly the form of the blades. Another common approach in the blade design is the consistent solution of direct problems. In this case, the designer can observe the consequences of the changes and results and make changes to obtain the desired result. This approach is more capacious time, but reliably provides positive feedback during compressor stages improving.

3.1 Aircraft engine centrifugal compressor stage improving

Centrifugal compressor stage operating modes are largely determined by the quality of the vanned diffuser (VD). Calculations using 3D software package will provide a more complete picture of the flow, but it significantly reduces the range of the investigated VD variants. Total characteristics of aviation engine centrifugal stage with unchanged axial-radial impeller and a variety of vanned diffusers (VD1, VD2, VD3) are taken by means of SP AxCBm and shown at Fig. 7.

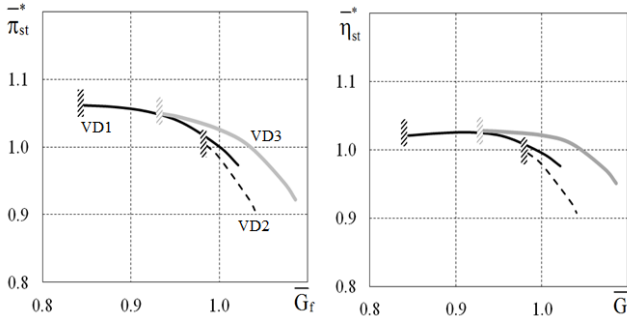


Fig. 7. Total characteristics of centrifugal stage with various vanned diffusers

Fig. 8 shows the contours of Mach numbers in the stage with different diffusers VD1, VD2 and VD3 respectively. Separately shown in greater detail the Mach numbers contours in the vanned diffusers.

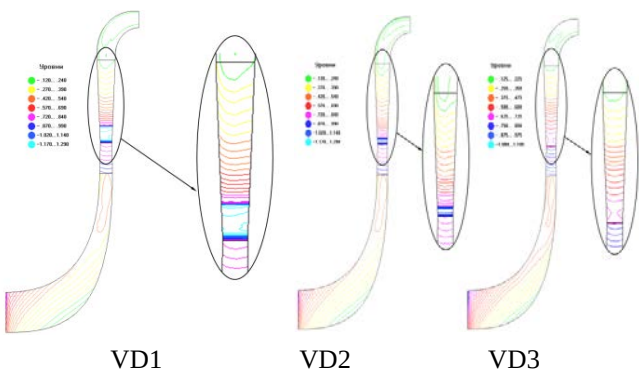


Figure 8. The contour lines of the Mach number in the centrifugal stage with various vanned diffusers on the preset mode.

Fig. 9 shows the distribution of Mach numbers along the flow median line for examined vanned diffusers.

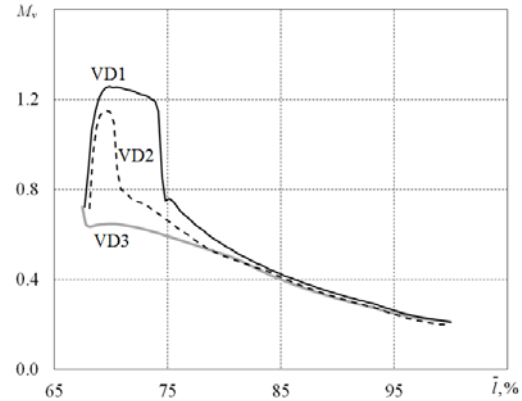


Fig. 9. Distribution of Mach numbers in various VD along the vanned diffuser median line

3.2 Modernization of high-pressure axial fan stage

The modernization of high-pressure axial fan stage was made with of SP AxSym_M to increase the compression and efficiency level at a given point. Fig. 10 shows the distribution of entrance and exit blade angles, as well as the stagger angle for the original (broken line) and modernized (solid line) options of stage (angles from row axis).

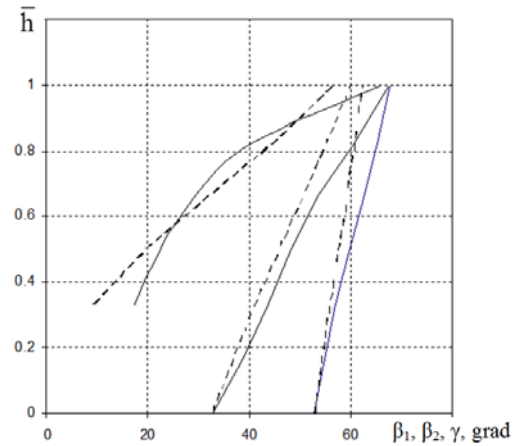


Fig. 10. Adjustment height distribution of entrance and exit blade angles, as well as the stagger angle of the profiles

Fig. 11 shows the Mach numbers contours at preset mode of flow and rotor speed. The distribution of contour lines in the blade area of the new RB is more uniform than in original stage, that suggests its higher efficiency.

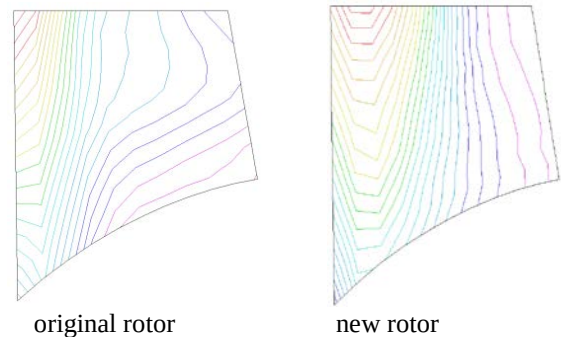


Fig. 11. Mach number contours at the specified mode
Fig. 12 demonstrates the total characteristics of the test rotors. It is obvious that modernized rotor has some advantages at a given mode (marker). The made changes improved the pressure ratio and efficiency in the modernized rotor at the specified speed and flow rate. The marked stability boundaries are received with Liblein diffusion factor. New rotor

allowed to obtain higher values of total pressure ratio and efficiency in the given point (marker).

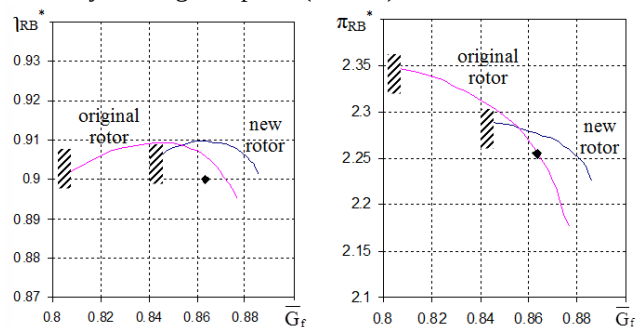


Fig. 12. The total characteristics of original and modernized rotors

3.3 Regulation of the multistage axial compressor

The influence of the multi-stage axial compressor guide vanes program control on its total characteristics was investigated with SP AxSym. Total characteristics of high pressure and low pressure multi-stage axial compressors of aviation engine without bypass are presented at Fig. 13. The broken line shows the first guide vanes program control, and solid – the second. Stability limits were determined with Liblein diffusion factor.

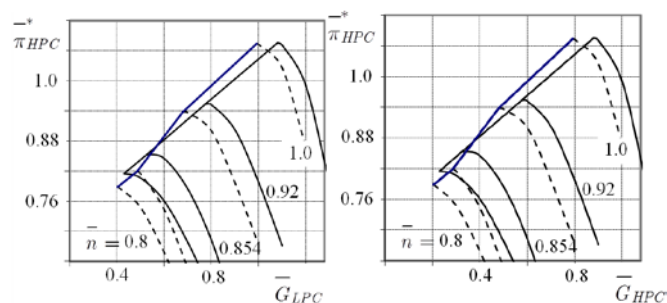


Figure 13. The influence of guide vanes program control on the total characteristic of multi-stage axial compressors

The ability to provide the flow structure in a multi-stage compressor analysis allows to examine the blade rows agreement and as a consequence to obtain more efficient guide vanes program control.

Numerical modeling of multistage axial compressor with bypass is presented in [15 et al.]

4 Conclusion

The report presents the results of numerical studies of sub- and transonic flows in the aviation engine compressors which were obtained by means of submitted original programs complex for 2D analyze of flow structure and the total characteristics of axial and centrifugal high-pressure stages and multi-stage compressors with air-bleeding and bypass from the air flow passage. This complex allows the numerical analysis in the axial and centrifugal high-pressure compressor stages, multistage axial compressors of aviation engines and their components. The complex elements may be used to improve compressor geometrical parameters for enhancing their performances.

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ENGINE COOLING SYSTEM WITH ENERGY RECUPERATION

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Abstract: Internal combustion (IC) engines are main source of power used in the automotive industry. These engines have undergone many up gradation for decades, with an initial goal of increase in power of the unit. Presently, as a result of tightening environmental regulations, research is focused on reducing harmful gases emissions along with increase of the engines efficiency.

Keywords: ENGINE, RECUPERATION, COOLING SYSTEM.

1. Introduction

Internal combustion (IC) engines are main source of power used in the automotive industry. These engines have undergone many upgrades for decades, with an initial goal of increase in power of the unit. Presently, as a result of tightening environmental regulations, research is focused on reducing harmful gases emissions along with increase of the engines efficiency.

The complexity of most engine systems certainly surpassed solutions used by the first engines designers. However, opening the car's engine covers from the 30s of the 20th century and the beginning of the 21st century, our attention is focused on rectangular radiator, rubber pipe system and expansion or overflow tank.

One gets the impression that the cooling system has been directly transferred out of the car, which is almost hundred years old. Analyzing the problem closer, it appears that in the engine cooling system almost nothing has changed. In reality, the heat transfer is improved by using different materials and bigger active surface of the radiator, but the concept has not changed.

Inherent feature of the cooling systems of IC engines is the dissipation of energy. More than 20% of the energy obtained from the fuel is transferred to the cooling system. For example, the engine which has 100[HP] and 30% of efficiency, in one hour, can produce more than 260[kJ] of energy and has radiator that dissipate almost the same amount of energy.

Presently, constant temperature of the engine is maintained as a result of taking over the appropriate amount of heat by the refrigerant circulating in the engine channels.

The control system consists of two circuits, small one – which includes engine and water pump, and a large one – with radiator (figure1).

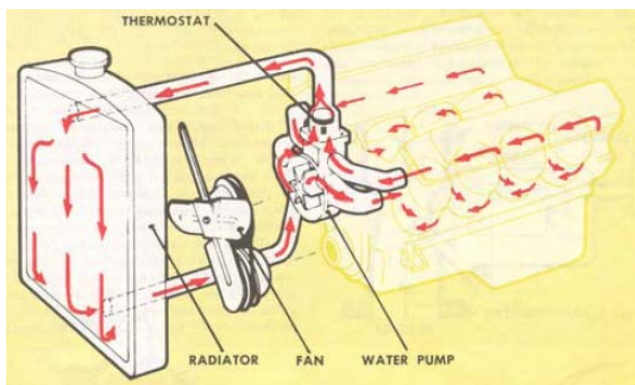


Fig. 1 Typical engine coolant system [7].

When the temperature of the coolant in a small circuit raises above a fixed value, the thermostat opens gradually, allows circulation of the coolant in a large circuit. When the coolant temperature drops, decreases the flow through the thermostat.

Considering these facts, it can be concluded that the current engines cooling system is archaic, and also its job is only to dissipate of accumulated heat. It seems feasible to modify the way the IC engines are cooled along with making use of dissipated energy.

It is worth noting, that the automakers are also looking for solutions by making use of dissipated energy. For example, BMW



proposes the use of a steam engine, which uses the heat from the engine exhaust system (figure 2).

Fig. 2 BMW adds Steam Engine to improve fuel efficiency [8].

2. Objective of the project

The objective of the project is to innovate the way the internal combustion engine is cooled and use the heat received from the engine to power a steam engine rather than being wasted in the radiator.

The proposed system would use the principle of operation of the ORC (Organic Rankine Cycle) power-station [1-6], which uses hydrocarbons as a working medium. That allows operation of the power-station at lower temperatures of the upper heat source. The system would have two duties, i.e. to keep the engine operating temperature in a certain range, and to convert the received heat for such a form of energy, that can be used in other purposes than just heating the interior of the vehicle.

Widely used in automotive, heat exchange system, where the refrigerant evaporates is an air conditioning system (figure 3).

The air supplied to the interior of the vehicle is cooled as it passes through a heat exchanger, where the refrigerant evaporates. Changing the physical state of the refrigerant requires far more energy than a change in its temperature. For example, a change of 10 degrees in temperature of 1 gram of water requires around 42[J] of energy, and the passage of 1 gram of water from the liquid state to the gaseous state requires over 2250[J] of energy, almost 54 times more. Unfortunately, medium after the evaporation is in a

gaseous state and in order to condense at ambient temperature must be compressed by the compressor. This again requires power.

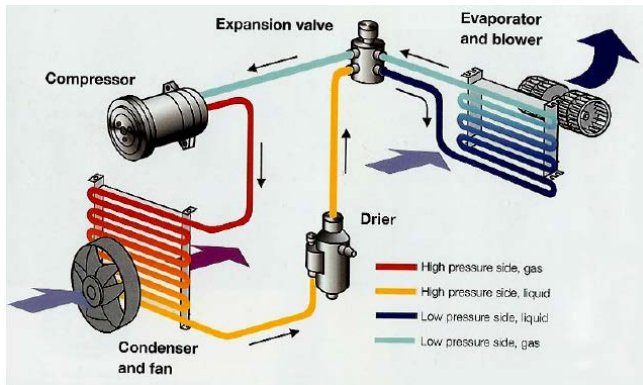


Fig. 3 The principle of vehicle air conditioning [9]

3. Proposed solution

The principle of ORC operation is shown on figure 4.

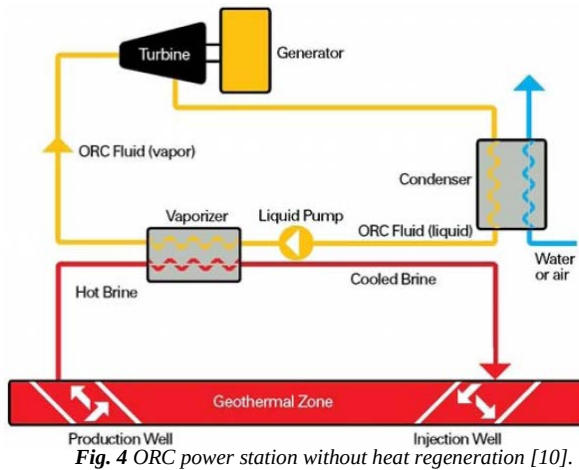


Fig. 4 ORC power station without heat regeneration [10].

Heat from the thermal source is supplied to heat exchanger (vaporizer), which causes evaporation of the compressed working fluid in the Rankine cycle. Then, the vaporized fluid drives a turbine connected to a generator. Fluid on the output of the turbine is directed to the condenser where it gets condensed. Liquid ORC fluid goes back to the pump, and then it is compressed. In the proposed solution is planned that supplied heat will come from the engine cooling system (figure 5).

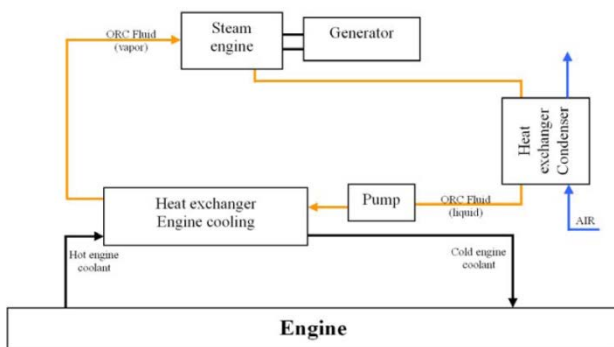


Fig. 5 Blok diagram of the proposed engine cooling system.

In addition, it is planned to use the heat from the exhaust system, while working when the engine is cold, or to increase the mass flow of refrigerant. Two solutions for such a system are shown in figure 6.

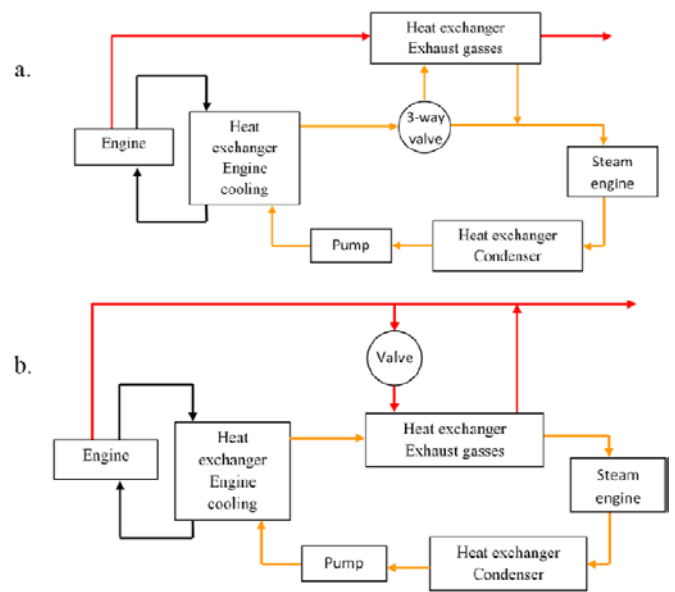


Fig. 6 Blok diagram of the proposed engine cooling system with using exhaust heat, which is controlled by: a) three-way valve in the refrigerant circuit, b) valve located in the exhaust system.

In the first case (a) three-way valve is used. It controls the amount of heat supplied to the refrigerant. The valve is mounted in the refrigerant circuit. It could lead the fluid directly to the steam engine, or in the case of a cold engine through the heat exchanger mounted on the exhaust system. In the second case (b), the valve is mounted in the exhaust system and it controls bypassing of hot gasses to heat exchanger.

4. Evaluation of system capabilities

In order to determine the amount of heat needed to be taken from the engine coolant (water) per time unit, the following simplifications were assumed:

- 30% of the heat (energy) supplied to the engine is converted into an effective power,
- 20% of the heat supplied to the engine is dissipated in the radiator.

Using these assumptions can be determined, what is the heat flow, which must be over taken in the ORC system. Assuming, that the engine power is 81,6 [HP] (60 [kW]) we obtain:

$$Q'_{ch} = 0,2 \frac{60000}{0,3} = 40000[W] \quad (1)$$

One of the fundamental problems that must be solved is to find a suitable refrigerant, which, on one hand, will evaporate at the temperature about 100[°C], by taking heat from the engine cooling system, but also allow to condensate at ambient temperature – about 50[°C] (of course at a lower pressure). Analysis of system operates using butane as a refrigerant is shown below.

The project assumes that the heat flow determined above will be fully taken over by the refrigerant (butane, figure 7). Liquid butane (50°C at 20[atm]) in the first stage is heated to boiling temperature (100°C at 20[atm]), and then the entire amount of butane evaporated ($r_{100^\circ C} = 210$ [kJ/kg]). Butane mass flow can be determined from the formula:

$$\dot{Q}_{CH} = \dot{m}c_p(100^\circ\text{C} - 50^\circ\text{C}) + \dot{m}r \quad (2)$$

$$\dot{m} = \frac{4000\text{W}}{2481 \frac{\text{J}}{\text{kgK}} 50\text{K} + 328360 \frac{\text{J}}{\text{kg}}} = 0,088 \frac{\text{kg}}{\text{s}} \quad (3)$$

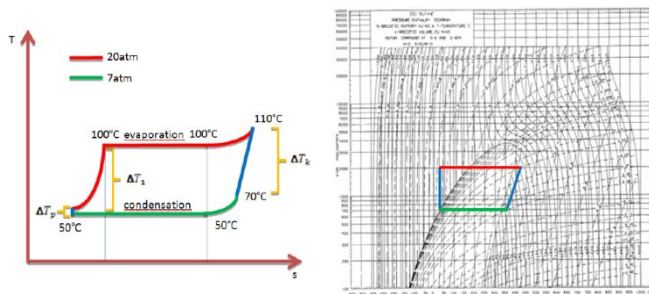


Fig. 7 Refrigerant (butane) in T-s and p-h.

The calculations of energy balance of the proposed system (for obtained above butane mass flow) were carried out using basic thermodynamic principles (figure 8). First the required power of pump ($\approx 300\text{W}$) to raise the pressure of butane from 7 to 20 atmospheres then the power which can be used from isentropic expansion of the refrigerant ($\approx 3500\text{W}$) and the required condenser cooling capacity ($\approx 36800\text{W}$) were calculated.

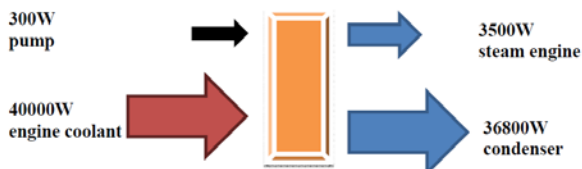


Fig. 8 Energy balance for proposed cooling system.

5. Summary

The proposed project consists of two sections- building a test rig and development of computerized heat transfer models. Building up of a test rig and simulation work will be carried out in parallel. This will help in optimization of the ORC in a certain area.

The advantage of this split would create two distinct requirements in terms of range of master's theses, but at the same time taking into account the project, complement it perfectly. Testing at the rig allow to verify the numerical model, which, in turn, the development of the real system with higher efficiency.

It is expected that the amount of saved energy could be used to drive an alternator. It is estimated that the weight of the system components should be less than 50 [kg]. ORC fluid condenser mounted at the front of the car should be smaller than the typical engine radiator. Thus the new system is not going to add huge extra mass to the vehicle.

Acknowledgement

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ADVANCED TECHNOLOGIES FOR WASTE HEAT RECOVERY IN INTERNAL COMBUSTION ENGINES

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Abstract: The escalating fuel price and carbon dioxide legislation have renewed the interest in the methods of increasing engine thermal efficiency beyond in-cylinder techniques. The aim of this study is to review the latest technologies of waste heat recovery of exhaust gases in internal combustion engines. These include turbocompounding systems, thermoelectric generators, thermoacoustic systems and closed-loop thermodynamic cycles based on Stirling, Ericsson and Rankine cycles. A number of studies revealed that Rankine cycle is the most perspective waste heat recovery system due to its higher thermal efficiency. Finally, the components of the Rankine cycle (working fluid, evaporator and expander) were studied in detail.

Keywords: WASTE HEAT RECOVERY, RANKINE CYCLE, INTERNAL COMBUSTION ENGINES, CO₂ EMISSIONS.

1. Introduction

An analysis of the European Commission shows that over 80% of the greenhouse gases in the atmosphere are composed of carbon dioxide. Over 25% of the carbon dioxide in the atmosphere is produced by transport, due to the burning of fossil fuels in internal combustion engines, see Fig. 1.

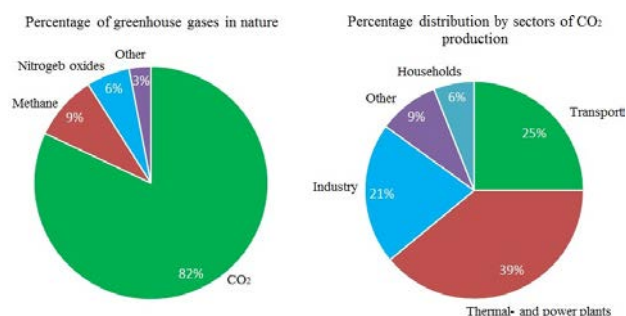


Fig. 1 Percentage of greenhouse in nature and distribution by sectors.

For this reason, the European Commission introduced legislation on carbon dioxide emissions produced by automobiles [1]. In 2020, emissions of carbon dioxide produced by light-duty vehicles must not exceed 95g CO₂/km, compared to the emissions produced in 1990 which is a reduction by 30% (Fig. 2).

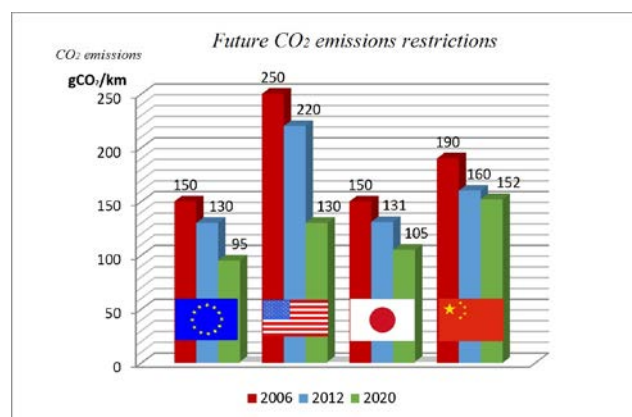


Fig. 2 Future CO₂ emission legislation.

In modern engines the overall efficiency is not more than 40%. However, most of the time engines operate with much lower efficiency than their maximum. In other words, a significant part of the energy released by the fuel in internal combustion engines is lost. That energy is lost in a form of heat in the exhaust and cooling system. Usually, the exhaust gases have higher energy than cooling system. The heat energy contained in the exhaust gases depends on

the operating point of the internal combustion engine. Milkov et. all [2] estimated this energy to be within the range of 25% to 65%. In the same engine the heat energy in the cooling system is within the range of 11% to 35%. There are two ways of increasing the overall efficiency, of reducing the fuel consumption and carbon dioxide emissions, respectively: by improving the working processes and waste heat recovery.

In spite of the existence of modern technologies for improving the working processes in internal combustion engines such as turbocharger systems, direct fuel injection, downsizing, low temperature combustion processes (CAI and HCCI), variable compression ratio, etc., the future requirements for internal combustion engines will be difficult to be met [3, 4].

For that reason waste heat recovery is a good way of increasing the overall engine efficiency, reducing the fuel consumption and CO₂ emissions.

2. State-of-the-art of waste heat recovery systems

Research has revealed that some advanced technologies can be applied to further improve waste heat recovery efficiency. Several technologies have been studied such as turbocompounding systems thermodynamic cycles (Rankine, Stirling, Ericsson and etc.), thermoelectric generators and thermoacoustic systems.

2.1 Turbocompounding systems

Over the last decade turbocompounding has been developed as an additional waste heat recovery system. Systems of this type convert part of the exhaust gas energy into mechanical or electrical energy, therefore they are of two types: mechanical and electrical turbocompounding. In mechanical turbocompounding the turbine shaft is mechanically connected to the engine crankshaft and the energy of the system is added directly to the engine output (Fig.3.)

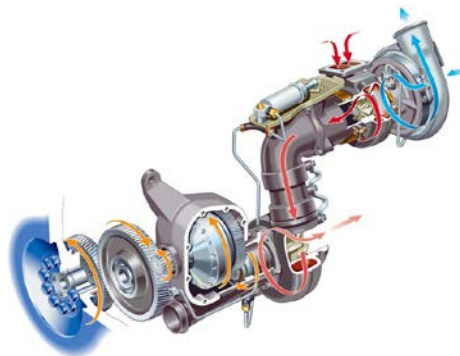


Fig. 3 Mechanical turbocompound.

In electric turbocompounding the turbine shaft is connected to a generator. In this case the energy is converted into electricity. The electrical turbocompounding can be developed without additional turbine. In this case the generator is mounted directly onto the turbocharger's shaft. (Fig.4.)

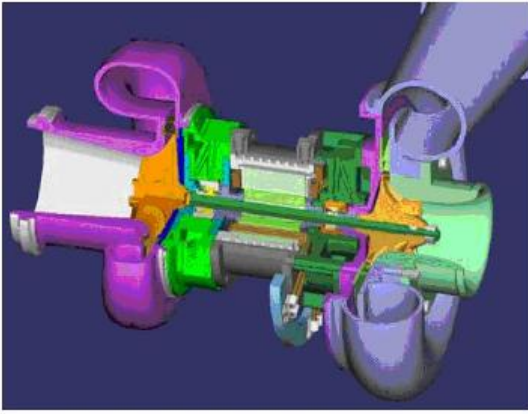


Fig. 4 Electric turbocompounding.

The disadvantage of such turbocompounding which includes an additional turbine is that the second turbine creates an additional backpressure in the exhaust system and adversely affects the gas-exchange process.

According to this study [5] the benefits of the implementation of mechanical turbocompounding are:

- Increase in attained power by 10-11%,
- Torque increase about 11%,
- Reduction in fuel consumption by 5-11%.

2.2 Thermoelectric generators

Thermoelectric generators use the properties of some materials which because of the effect of temperature difference generate voltage (effect Seebeck).

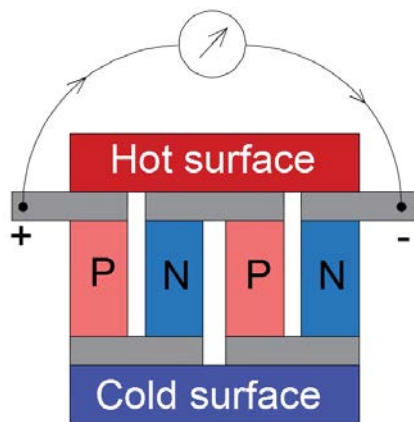


Fig. 5 Thermoelectric generators.

The principle of operation is as follows: the thermoelectric element is placed between two surfaces called hot and cold. The difference in temperature causes changes in the core of the thermoelectric material, which converts thermal energy into electricity.

The efficiency of the thermoelectric generators depends on the material and the efficiency of heat exchange between the exhaust gases and the hot part of the generator. At this point it does not exceed 5%.

Their advantage is that they do not create noise and vibration and have no moving parts or create minimal resistance in the exhaust system of internal combustion engines. However, due to their low efficiency, high cost and heavy weight at this stage they have not been used in series production engines.

2.3. Thermoacoustic systems

The thermoacoustic effect is the interaction between the temperature, density and pressure difference of sound waves. Thermoacoustic engines are thermoacoustic devices that use high amplitude sound waves to transfer heat from one place to another, or vice versa, use heat to produce high amplitude sound waves. Generally thermoacoustic engines can be divided into two types:

- constant wave thermoacoustic systems (Fig.6.);
- variable wave thermoacoustic systems (called "running" waves).

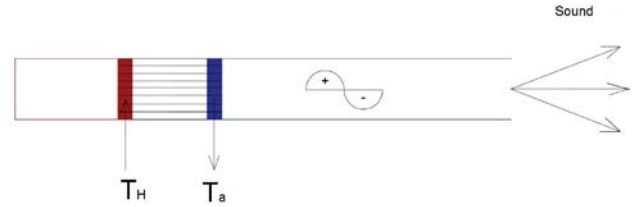


Fig. 6 Thermoacoustic system.

Thermoacoustic devices are suitable for heat recovery, as they can operate with a low heat source. Technological thermoacoustic devices have the advantage that they have no moving parts, which makes them attractive for applications where reliability is important.

Thermoacoustic engines have some disadvantages such as:

- Low power and large size;
- Use fluids with high viscosity;
- Low efficiency compared to other devices;
- Complex construction and high cost of production.

2.4. Closed-loop thermodynamic cycles

2.4.1. Stirling cycle

The Closed-loop thermodynamic cycle of Stirling was patented by Dr. Robert Stirling in 1816. This closed thermodynamic cycle operates with different types of fluids. The most commonly used ones are air, hydrogen or helium. The working process in the Stirling cycle occurs in four stages (Fig.7.):

- 1-2. $T = \text{constant expansion}$ (heat transfer from the external source)
- 2-3. $V = \text{constant regeneration}$ (internal heat transfer from the working fluid to the regenerator)
- 3-4. $T = \text{constant compression}$ (heat rejection to the external sink)
- 4-1. $V = \text{constant regeneration}$ (internal heat transfer from the regenerator back to the working fluid)

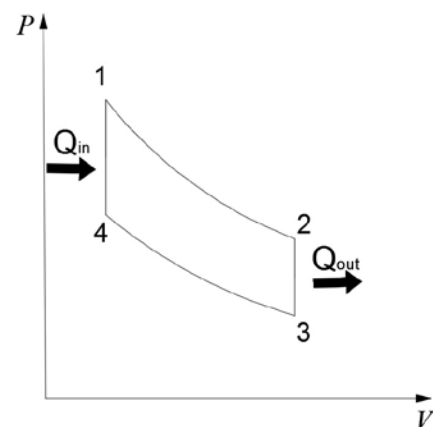


Fig. 7 Working process of Stirling cycle in P-V diagram.

A study of Thombare et al. [6] shows that the Stirling cycle has high efficiency when the cycle operates with low heat source. According to that study suitable fluids for this cycle are air and helium.

A study of Igobo at all. [7] shows that the thermodynamic cycles of Stirling are suitable for waste heat recovery in internal combustion engines, but there is still a lot of research to be done in order to improve their effectiveness as a waste heat recovery system.

2.4.2 Erickson cycle

The Ericsson cycle is similar to the Stirling cycle, except that the two constant-volume processes are replaced by two constant-pressure processes. The working process in the Erickson cycle occurs in four stages (Fig.8):

1-2. Isobaric heating of the working fluid - a process in which the temperature of the working fluid increases at constant pressure.

2-3. Isothermal expansion - a process in which the fluid expands at constant temperature.

3-4. Isobaric cooling - a process in which the liquid temperature decreases at constant pressure.

4-1. Isothermal compression - a process in which the fluid is compressed at constant temperature.

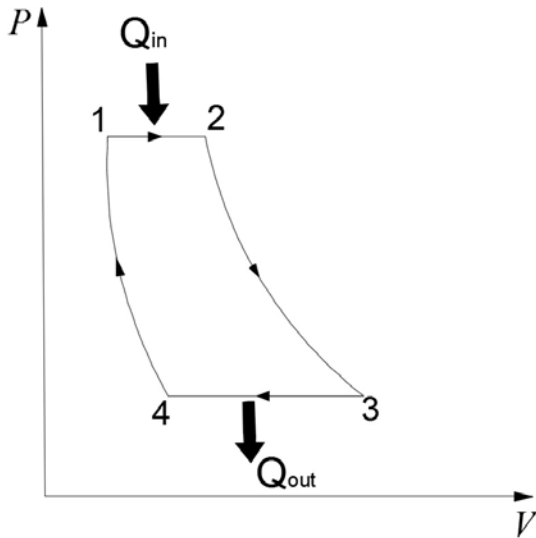


Fig. 8 Working process of Erickson cycle in PV diagram.

2.4.3 Rankine cycle

The Rankine cycle is a closed-loop thermodynamic cycle which converts heat into mechanical power. The mechanical power can be used to perform mechanical work or to produce electricity. Separately, this cycle is widely used in the production of electricity by thermal power plants.

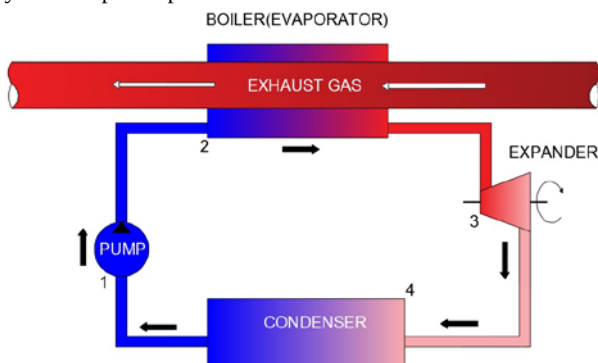


Fig. 9 Rankine cycle.

The system consists of the following elements: working fluid, pump, boiler (evaporator), expander and condenser (Fig.9). The ideal Rankine cycle presented in the T-S diagram (Fig.10) consists of the following processes:

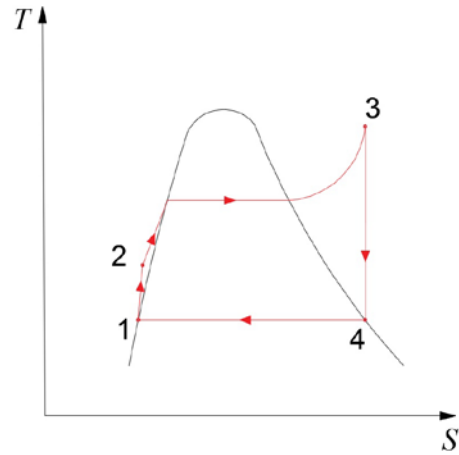


Fig. 10 Closed-loop Rankine cycle process

1-2 Isentropic compression in the pump;

2-3 Working fluid heating and evaporation in the boiler;

3-4 Isentropic expansion in the turbine;

4-1 Working fluid cooling in the condenser.

The Rankine cycle efficiency depends on: the working fluid, boiler (evaporator), expander, condenser and working fluids operating parameters (mass flow rate and pressure).

Table 1 presents the results of research on the waste heat recovery system based on the Rankine cycle.

Table 1: Review of researches on waste heat recovery system based on Rankine cycle

Author	Working fluid	Expander	Conclusions
Kang S. (2012) [8]	R245fa	Turbine	The maximum average cycle, turbine efficiencies, and electric power were found to be 5.22%, 78.7% and 32.7 kW, respectively.
Hountalas D. at al. (2011)[9]	Organic fluids and water	Piston machine	The maximum improvement in BSFC was observed in the case of the organic Rankine cycle(ORC) and is ~11.3% when both EGR and CAC heat amounts are utilized. The corresponding value for steam is 9%.
Weerasinghe W. (2010)[10]	-	-	A numerical study shown that waste heat recovery system based on steam Rankine cycle can offer substantial gains in fuel economy, potentially in the order of 20%.
Katsanos C. (2012)[11]	Water and the refrigerant R245ca	Turbine	Water - BSFC improvement by 6.1% to 7.5%; R245ca - BSFC improvement by 10.2% to 8.4%
Barrieu E. (2013)[12]	Water and ethanol	-	The order of magnitude of the saved fuel can be estimated at 4 to 10% depending on the usage and the system optimization.
Espinosa N. (2011)[13]	Ethanol and water, R245fa, fluorinol mixtures	-	The challenge for waste heat recovery techniques remains the heat rejection as Rankine cycle system efficiencies are quite low (10-15 %). The R245fa shown best results in this study.

Leduc P. (2013)[14]	Water	Piston machine	At medium engine power, corresponding to highway vehicle use, Rankine system produces about 1 kW of mechanical power.
Chen Y. (2012)[15]	R123	Turbine	The peak net power output and thermal efficiency of Rankine cycle are 15.087 kW and 13.38 %. The diesel engine power increases by 6.17 %.
Capata R. (2014)[16]	R-134a, R245fa and water	Turbine	The smallest turbine shown best results when the working fluid is R134a. The greatest amount of power is when the R245fa is used as working fluid with the largest turbine.
Yang K. (2014)[17]	R416A	-	When the degree of superheat is 40 K, engine torque is 300 N·m, and engine speed is 1900 r/min, the exergy efficiency of the ORC waste heat recovery system reaches its maximum of 54.60%.
Dolz V. (2011)[18]	Organic fluids and water	Turbine	Water Rankine cycle BSFC reduction is between 8.5 - 8.8%.
Serrano J. (2011)[19]	Water	Turbine	Water Rankine cycle achieves about 15% increment in the global mechanical power.
Boretti A. (2011)[20]	R245fa	Turbine	ORC increase the fuel efficiency by 5.1% with maximum improvement of 8.2% when the engine is operating steady state.
Roy J. (2010)[21]	R-12, R-123 and R-134a	Turbine	R-12 - RC efficiency is 12.09%; R-123- RC efficiency is 25.30%; R-134a-RC efficiency is 15.53%.
Yu G. (2012)[22]	R245fa	Turbine	The ORC system gets relatively high power (15.5 kW, 14.5 kW and 13.7 kW) and efficiencies (9.1%, 9.2% and 9.4%) under conditions 1, 2 and 3.
Zhang H. (2012)[23]	R245fa and R134a	Two turbines	In the peak effective thermal efficiency region, the augmentation proportion of the effective power for the combined system is the lowest, at 14–16%, but is highest in the small-load and high-speed region where the augmentation proportion is 38–43%.

The literature review revealed that the exhaust waste heat recovery systems based on the Rankine cycle provide high efficiency. The system efficiency can reach 15% to 20% which means that the engine efficiency improvement is by 10% to 12%. Therefore, the construction of the Rankine cycle will be presented in greater details.

3. Features of the Rankine cycle

3.1 Working fluid

In the development of the Rankine cycle waste heat recovery system, one of the most important steps is the selection of the working fluid because its operating properties have a great influence on the efficiency, size and design. A number of fluids have been studied such as: water, ethanol, benzene, organic fluids and etc. Water provides high efficiency in the case of a high temperature heat source while organic fluids are more efficient in the case of a low temperature heat source.

The working fluids used in the Rankine cycle can be divided into three groups according to the slope of the condensation line in T-S diagram: wet, dry and isentropic [24].

The temperature of the heat source is the main criterion in selecting the working fluid (Fig.11).

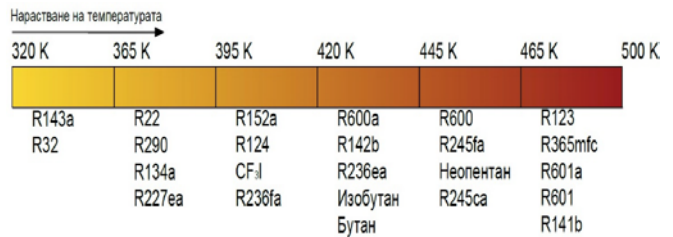


Fig. 11 Organic fluids classified by the temperature of the heat source

The results of some studies of various fluids used as the working fluids of the Rankine cycle are listed in Table 2

Table 2: Review of research of the various working fluids

Author	Studied working fluids	Criteria / Methodology	Recommended
Punov at al. (2015) [25]	Water, ethanol, R245fa and R134a	Theoretical study of the Rankine cycle	Water as the working fluid provides the highest Rankine cycle output power and efficiency
Chen at al. (2010) [26]	35 different fluids	Review and analysis of working fluid selection criteria for sub- and supercritical ORCs	R32, R125 and R143a - supercritical ORCs; R141b, R123, R21, R245ca, R245fa, R236ea and R142b - subcritical ORCs; R134a, R290, R124, R227ea, R218 - for both.
Hung at al. (2010) [27]	11 different fluids	-	The fluids with best results are the isentropic type fluids.
Lakew at al. (2010) [28]	R134a, R123, R227ea, R245fa, R290 and n-pentane	-	R227ea, R245fa and n-pentane
Mikielewicz at al. (2010) [29]	20 different fluids	Criteria for sub- and supercritical ORCs	Ethanol, R123 and R141b
Fernandez at al. (2011) [30]	Siloxanes	Criteria for sub- and supercritical ORCs	MM and MDM
Guo at al. (2011) [31]	13 different fluids	Thermodynamic and technoeconomic evaluation of the transcritical Rankine cycle	R125 best fluid to the temperature of 90 °C of the heat source, R32 and R143a to 100 °C
Guo at al. (2011)	27 different fluids	-	E170, R600 and R141b

[32]			
Datla at al. (2012) [33]	11 different fluids	-	n-pentane
Gao at al. (2012) [34]	18 different fluids	Net power exergetic efficiency, size setting of the regulator and of the required heat exchanger (evaporator and condenser) are evaluated for the selection	R152a and R143a
He и др. (2012) [35]	22 different fluids	-	R114, R245fa, R123, R601a, n-pentane, R141b and R113
Heberle at al. (2012) [36]	mixture of isobutane / isopentane and R227ea / R245fa	-	The mixture of R227ea / R245fa is recommended
Vidhi at al. (2012) [37]	7 different fluids	-	R134a

3.2 Expander

Another essential element of the Rankine cycle is the expansion machine. The expansion machine is the element of the system which converts thermal energy contained in the fluid into mechanical work. Expansion machines are of several types: turbine, reciprocating piston machine, screw, scroll and vane [24].

Two types of expansion machines are most widely used – turbines and piston machines. A higher specific power is a typical advantage of turbines but they can't provide high efficiency for low power systems. Piston machines are preferred in automotive application due to their low speed, compatible with the engine speed.

Table 3 presents the results of research of different expander machines.

Table 3: Review of research of the different expander machines.

Author	Studied expander machine	Efficiency (%)	Power (kW)
Yamamoto at al.[38]	Radial-inflow turbine	48	0.15
Nguyen at al.[39]	Radial-inflow turbine	49.8	1.44
Yagoub at al.[40]	Radial-inflow turbine	85, 40	1.50
Inoue at al. [41]	Radial-inflow turbine	70-85	5-10
Kang[8]	Radial-inflow turbine	78.7	32.7
Pei at al. [42]	Radial-inflow turbine	65	1.36
Li at al. [43]	Radial-inflow turbine	68	2.40
Zanelli at al. [44]	Scroll expander	63-65	1-3.5
Mathias at al.[45]	Scroll expander	67,81,83	1.2, 1.38, 1.75
Peterson at al. [46]	Scroll expander	45-50	0.14-0.24
Wang at al. [47]	Scroll expander	70-77	0.5-0.8

Saitoh at al.[48]	Scroll expander	65	0-0.46
Kim at al. [49]	Scroll expander	33.8	11-12
Manolakos at al.[50]	Scroll expander	10-65	0.35-2
Guangbin at al.[51]	Scroll expander	-	0.4-1.1
Wang at al. [52]	Screw expander	26-40	0.5-3
Smith at al. [53]	Screw expander	48-76	6-15.5
Baek at al.[54]	Reciprocating piston expander	70.5	24.35
Zhang at al.[55]	Reciprocating piston expander	62	-
Mohd at al.[56]	Rotary vane expander	43-48	0.025 - 0.032
Yang at al. [57]	Rotary vane expander	17.8-23	-

3.3 Boiler (Evaporator)

Heat exchangers are devices where heat transfer from one medium to another occurs. In waste heat recovery systems based on the Rankine cycle transfer of heat from the exhaust gas to the working fluid is carried out in a heat exchanger called the evaporator. The object of a number of studies is to establish the type of heat exchanger which is most suitable for application in the Rankine cycle. Heat exchangers can be classified into many groups, because of their diversity. One of the methods of their classification is by the movement of the fluids: co-current and countercurrent heat exchangers (Fig.12.)

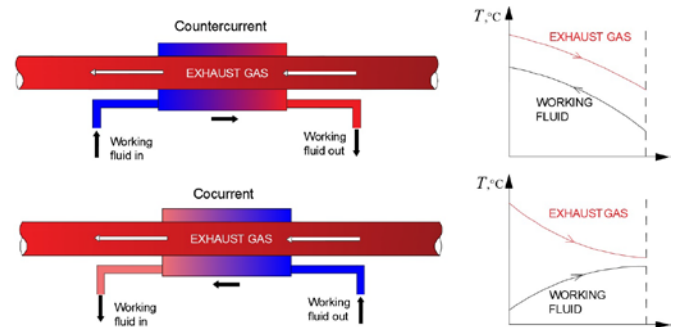


Fig. 12 Co-current and countercurrent heat exchangers.

Table 4 presents the results of research of different types of evaporators.

Table 4: Review of research of the different types of heat exchangers.

Author	Heat exchanger type	Methodology	Conclusions
Pandiyarajan at al.[58]	Shell and finned tube	Experimental study	Nearly 10–15% of total heat is recovered
Lee at al.[59]	Small fin-tube	Experimental and Taguchi method	The heat exchanger that had maximum effectiveness was not necessarily the

			optimum design.
Zhang at al.[60]	Finned tube	Mathematical ORC modeling	The overall heat transfer rate of the evaporator increases with engine power and reaches 70.4 kW at the rated power point.
Ghazikhani at al. [61]	Simple double pipe	Experimental and exergy analysis	12% reduction in BSFC
Wang at al.[62]	Multi-coil helical	Experimental and ORC numerical simulation	Total fuel saving was up to 34% under 2000 rpm and 75 Nm
Hossain at al. [63]	Shell and tube	Experimental and CFD modeling	Improving the effectiveness of heat exchanger from 0.44 to 0.76 by optimization design
Bari at al. [64]	Shell and tube	Experimental and CFD modeling	An additional 23.7% power improvement achieved by using water as the working fluid

4. Conclusions

On the basis of the review of the methods of increasing the overall engine efficiency, of reducing the fuel consumption and emissions CO₂ from internal combustion engines, the following conclusions can be drawn:

- Although there are modern systems of improving the working processes in internal combustion engines the future requirements for reducing CO₂ are difficult to be achieved;

- Since 60% of the energy in the internal combustion engine is lost as heat the waste heat recovery system is a good way of increasing the overall engine efficiency;

- The literature review revealed that the exhaust waste heat recovery system based on the Rankine cycle provides higher efficiency than others. The system efficiency can reach 15% to 20% which means that the engine efficiency improvement is between 10% and 12%.

- The Rankine cycle efficiency depends on: the working fluid, boiler (evaporator), expander, condenser and working fluids operating parameters (mass flow rate and pressure).

On the basis of these results, our future work will be focused on numerical and experimental study of the impact of waste heat recovery system Rankine cycle-based on the performance of the internal combustion engine.

Acknowledgments

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DIAGNOSTIC OF A TURBOCHARGED DIESEL ENGINE

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Abstract: The paper deals with non-dismountable diagnostic of the power of a turbocharged Diesel engine. For the purposes of diagnostic the transient operation mode of the engine was described by an extended mathematic model that takes into account the influence of suction volumes and exhaust pipes. The results obtained by this improved model were similar to the values measured by an experiment. This model produces relatively more trustworthy information regarding the influence of some typical defects on the chosen diagnostic parameters. Finally, the improvement makes possible to more accurately identify them, on the basis of changes of selected and measured diagnostic quantities the defect that has caused these changes.

Keywords: DIAGNOSTIC, TURBOCHARGED DIESEL ENGINE, TRANSIENT OPERATION, MODELING.

1. Introduction

Technical diagnostic forms an important part of comprehensive care about service reliability of engineering. It has been proved that application of diagnostic in the technological procedure of attendance and current repair of tanks and automobiles allows to keep high level of their trouble-free operation and to extend the lifetime of the vehicles parts. For these reasons it is also necessary to pay attention to the diagnostic of their engines and by its help to ensure more economic operation on the whole.

For diagnostic the power of the turbocharged Diesel engines a method of acceleration has been used. It means that the diagnostic parameters are measured during the transition mode of run of the engine and turbocharger, when the engines work under conditions that are close to its full load characteristics.

The decisive element of the correct diagnostics is the choice of suitable diagnostic quantities that have sufficient informative weight. It is suitable to carry out this choice on the basis of knowledge of the results that can be preliminarily obtained by means of a mathematic model of dynamic equilibrium of the engine and the turbocharger, compiled for this purpose. This model is completely general. Its application on to the certain engine can be carried out using data and characteristics of the respective engine and of its turbocharger as well. The characteristics of the engine can be obtained using a suitable mathematic model of the engine cycle, e.g. according to literature [1]. Then the modelling of the most frequently occurring defects can be executed, including the determination of their influence on chosen diagnostic parameters. In this very economical way, it is possible to make the choice of suitable diagnostic parameters, without demanding experimental tests.

In the primary solution we have used simplified assumptions, neglecting the volumes of suction and exhaust pipe of the engine. This simplification resulted into a much faster reaction of the turbocharger within the model than was the reaction of the real engine system (obtained from experiments) which, in its results, lowered the informative value of the diagnostic quantities being studied. For this reason the primary and simplified model of dynamic equilibrium of turbocharged Diesel engine was extended and the influence of the volumes of the suction and exhaust pipe was taken into account. The correctness of this theoretical solution was proved experimentally.

2. Methods of diagnostic

For diagnostic of the engine power the method of measuring the functional dependence of the crankshaft acceleration was used. Furthermore, this method was completed by simultaneous measuring of the functional dependence of the turbocharger angular speed, since in the [2] and [3] we have found out that the change of the turbocharger speed is one of the basic diagnostic quantities that reacts very sensitively on most of the modelled defects of the engine.

For numerical modelling of the influence of some typical defects on chosen diagnostic quantities a model based on the

dynamic equilibrium of the engine and turbocharger was compiled. The method of mathematic modeling of equilibrium and no equilibrium modes of engine and turbocharger cooperation was used for the solution of the above mentioned problems, including experimental verification.

The following procedure was used to achieve the laid out aims:

1. Creation of the mathematic model of dynamic equilibrium of the engine and turbocharger run, including the influence of the volumes of suction and exhaust piping.

2. Expression of the influence of some typical defects on the chosen diagnostic quantities of the system engine-turbocharger using the model of their dynamic equilibrium. Determination the required characteristic of the model has to be carried out by means of the model of static equilibrium.

3. Experimental verification of the theoretical solutions and results.

With the help of analysis of results of modelling and experiments, the diagnostic quantities of turbocharged Diesel engine were precised.

3. Model of dynamic equilibrium

The turbocharged engine can be assumed, from the point of view of its functioning, to work as a system consisting of two subsystems, i.e. of the engine alone and of its turbocharger, bound together aerodynamically through the mediating gas flow. The block diagram of this engine-turbocharger system is presented in fig. 1.

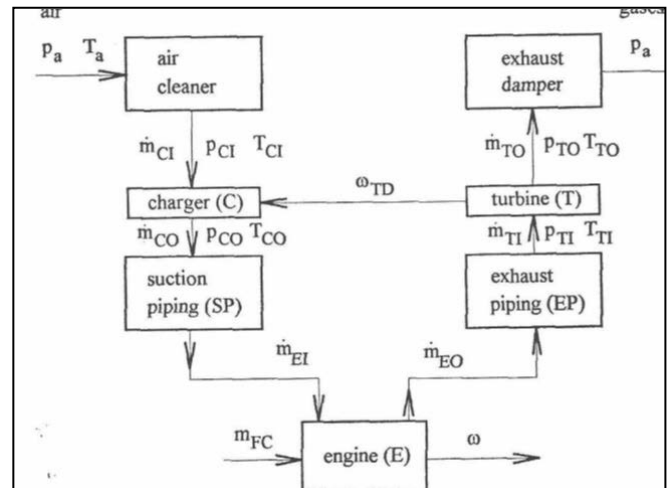


Fig. 1 Block diagram of the Diesel engine and turbocharger system

The mathematic description of individual elements of turbocharged Diesel engine is the following:

For the dynamic equilibrium of the engine alone it holds true:

$$J \cdot \frac{d\omega}{dt} = M_E - M_L \quad (1)$$

Where:

J - is the engine inertia moment, i.e. inertia moment of its moving parts reduced into the axis of the crankshaft,

ω - angular speed of the crankshaft,

M_E - engine instantaneous indicated torque,

M_L - instantaneous torque loss of the engine,

$M = M_E - M_L$ - the engine useful torque.

Similarly, for the turbocharger alone, it is valid:

$$J_{TC} \cdot \frac{d\omega_{TC}}{dt} = M_T - M_C \quad (2)$$

Where:

J_{TC} - inertia moment of the turbocharger rotor,

ω_{TC} - angular speed of the turbocharger rotor,

M_T - driving torque of the turbine,

M_C - required torque of the charger.

The torques in the above equations depends on:

- angular speed of the crankshaft (ω),
- fuel injected in the cylinder per a cycle (m_{FC}),
- total feeding pressure of the air behind the charger (p_{CO}),
- total pressure of gases in the exhaust piping (p_e), i.e. in front of the turbine.

The dynamic equilibrium of the change of the air mass in the respective volume of the suction piping can be expressed by the following equation:

$$\frac{dm_{SP}}{dt} = \dot{m}_{CO} - \dot{m}_{EI} \quad (3)$$

Where:

$\dot{m}_{CO}$ - is the air-mass-flow from the charger,

$\dot{m}_{EI}$ - is the air-mass-flow into the engine,

m_{SP} - is the air mass in the suction piping.

Similarly, for the dynamic equilibrium of the change of mass of gases, with the given volume of the exhaust piping, it is possible to write:

$$\frac{dm_{EP}}{dt} = \dot{m}_{EO} - \dot{m}_{TI} \quad (4)$$

Where

$\dot{m}_{EO}$ - mass flow of gases leaving the engine,

$\dot{m}_{TI}$ - mass flow of gases into the turbine,

m_{EP} - mass of the gases in the exhaust pipe.

The last two equations can be rewritten into the following forms. For the suction piping it holds true:

$$\frac{dp_{CO}}{dt} = \frac{n_a \cdot p_{CO}}{V_{SP} \cdot \rho_a} \cdot (\dot{m}_{CO} - \dot{m}_{EI}) \quad (5)$$

and similarly for the exhaust piping:

$$\frac{dp_{TI}}{dt} = \frac{n_g \cdot p_{TI}}{V_{EP} \cdot \rho_g} \cdot (\dot{m}_{EO} - \dot{m}_{TI}) \quad (6)$$

Where:

p_{CO} - is the total pressure of the air behind the charger,

p_{TO} - is the total pressure of the gases in front of the turbine,

ρ_a, ρ_g - are the specific masses of the air and gases,

n_a, n_g - are the exponents of polytropic of the air and gases,

V_{SP} - is the volume of the suction piping,

V_{EP} - is the volume of the exhaust piping.

4. Modelling of defects and verification of diagnostic quantities

On the basis of the analyses that were carried out in [2], some suitable quantities have been chosen that react sensitively enough on most of the typical defects that can appear during the engine service and the change of whose reflect trustworthily enough the instantaneous technical state of the engine. These quantities are the following:

- Total temperature in front of the turbine (T_{TI}),
- rpm (n) and angular acceleration (ϵ) of the crankshaft (expressed indirectly in the form of useful torque M of the engine),
- rpm of the turbocharger (n_{TC}),
- temperature of the cooling liquid (T_W).

The basis for this choice were obtained thanks to the previous modeling and experiments. The influence of typical defects on the chosen diagnostic quantities was stated by means of mathematic models and the obtained results were verified by experiments.

5. Experimental equipment

The theoretical assumptions were verified on the automobile engine SKODA LIAZ M 637 of Czech make. It is a row, six cylinder, water-cooled and turbocharged Diesel engine. The angular position of the crankshaft was measured by the Vibro-Meter KW-T pick up. The second derivation of this quantity has provided the required angular acceleration of the crankshaft. The total pressure of the working fluid behind the charger and in front of the turbine were measured by the help of the Vibro-Meter pick up PR-5K. The total temperature, in front of the turbine, was measured by the thermocouple based on NiCr-Ni alloys. The angular speed of the turbocharger was measured by the optical pick up PU 420. The above mentioned diagnostic quantities were taken by measuring device MC-32, having sampling frequency of 1 MHz.

6. Verification of the theoretical assumptions

By means of the above described mathematic model the calculation of the engine transition process from its equilibrium state at $n = 1000$ rpm was executed up to the instant of reaching its maximum speed at $n = 2000$ rpm. During this acceleration the maximum dose of fuel was adjusted.

On the basis of experience from the previous studies [2], [3] there were stated only the values of the following chosen diagnostic quantities:

- speed (rpm) of the turbocharger (n_{TC}),
- engine torque (M),
- total temperature of gases in front of the turbine (T_{TI}).

For the possible comparison of these diagnostic quantities we have expressed their dependences on the engine speed.

		Measur- ement	Calculation			
			New model		Primary model	
			value	difference	value	difference
n_{TC}	[rpm]	18735	20838	2103	31632	12897
M_{MAX}	[Nm]	864	801	63	816	48

		New model		Primary model	
n_{TC}	[%]	10.1		40.8	
M_{MAX}	[%]	7.8		5.8	

From the measured and calculated results it is clear that the use of the improved (new) mathematic model of common run of the engine and turbocharger, which respects the volumes of suction and exhaust piping, gives substantially more precise results,

especially in determining the dependence of the turbocharger speed during acceleration. It is possible to demonstrate it on the following results. In the case of the primary mathematic model (without taking into account the influence of volumes of suction and exhaust piping), at the end of the acceleration (i.e. at the engine speed of $n = 2000$ rpm), the calculated speed of the turbocharger was $n_{TC} = 31632$ rpm. This value exceeded the measured one by 40,8 %. In the case of the improved mathematic model we have obtained the respective turbocharger speed $n_{TC} = 20838$ rpm, which is only 10,1% above the measured value.

As it is evident from Table 1 and Table 2, the improvement of the mathematic model has no substantial influence on the calculated value of the engine torque. In both cases of calculations the difference between calculations and experiments is under 8 %.

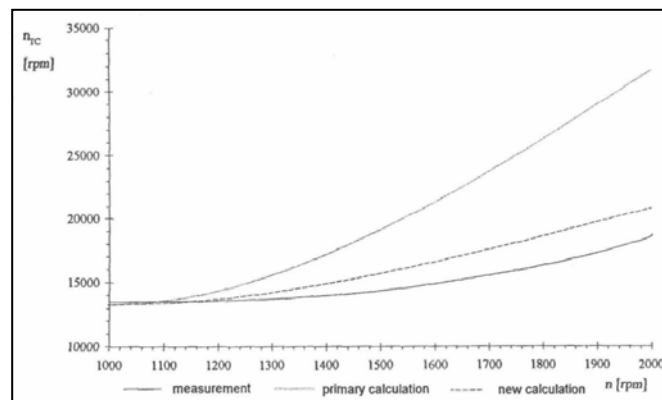


Fig. 2 The turbocharger speed (n_{TC}) dependence on the engine speed (n).

So, the modeling of the engine and turbocharger non-equilibrium run that takes into account the volumes of suction and exhaust piping gives results that differ less from those obtained by experimental measurements. As it can be seen from fig. 2, substantial improvement is reached especially in the case of the turbocharger speed which is one of the most important diagnostic quantities of the engine system.

From the above illustrated facts it follows that the improved mathematic model can give more trustworthy results.

7. Results

The relative changes of the diagnostic quantities to those that correspond to the correct technical state of the engine, which were obtained by using the improved model, are presented in Table 3.

These values were determined for the instant of reaching the engine maximum speed of

$n = 2000$ rpm. The numbers of corresponding defects taken into account in the Table 3 are the following:

№ 1 : reduction of the fuel dose through 15 %,

№ 2: decreasing the geometric beginning of the fuel injection by changing the crankshaft angle by 6 degree,

№ 3: increasing of the flow resistance in the air cleaner (expressed by the drop of the total pressure in front of the charger) by 10 % at the engine speed of $n = 1000$ rpm (which is the starting speed of the transition mode of acceleration),

№ 4 : untightness of the cylinder packing, corresponding to 9 % of the engine useful power drop,

№ 5: increased flow resistance at the engine outlet, expressed by 10 % of pressure rise behind the turbine (at the engine speed of $n = 1000$ rpm).

Abbreviations used in Table 3 :

calc - values determined by the help of calculation

meas - values obtained by the experimental measurements.

Diagnost. quantity	Deviation of diagnostic quantities in [%] from the nominal values										Meas. error		
	Defect № 1		Defect № 2		Defect № 3		Defect № 4		Defect № 5				
	calc	meas	calc	meas	calc	meas	calc	meas	calc	meas			
	n_{TC}	M_{MAX}	-1,3	-1,1	12,8	11,7	-12,6	-10,7	-5,4	-4,5		-7,1	-6,1
													±1,9

Another criterion, i.e. the informative value of the diagnostic quantities was taken in account and evaluated. Its definition is:

$$\Gamma = \frac{\partial X}{\partial P} \quad (7)$$

Where:

∂X - is the relative change of the respective quantity X and ∂P - is the relative change of the engine useful power P. This informative value Γ , being for a respective diagnostic quantity smaller than 0,8, was taken as a non-governing one. The results of the calculation of Γ for the case of turbocharger speed n_{TC} (which is one of the most important selected diagnostic quantities) are presented in Table 4.

Diagnost. quantity	Informative value Γ									
	Defect № 1		Defect № 2		Defect № 3		Defect № 4		Defect № 5	
$n_{TC} MAX$	calc	meas	calc	meas	calc	meas	calc	meas	calc	meas
	0,92	0,88	0,98	0,81	1,54	1,84	0,96	0,57	1,01	1,00

From comparison with the results in Table 3 it is obvious that good agreement between the results of calculation and measurements is in the case of both the chosen diagnostic quantities, i.e. in the case of the turbocharger speed (n_{TC}) and of the angular speed of the crankshaft (presented here by the engine torque M). Also their informative value Γ is high enough and so it is possible to take both these diagnostic quantities as governing ones. For the illustration the effect of the chosen defects on the both mentioned diagnostic quantities is presented in Fig. 3 and Fig. 4.

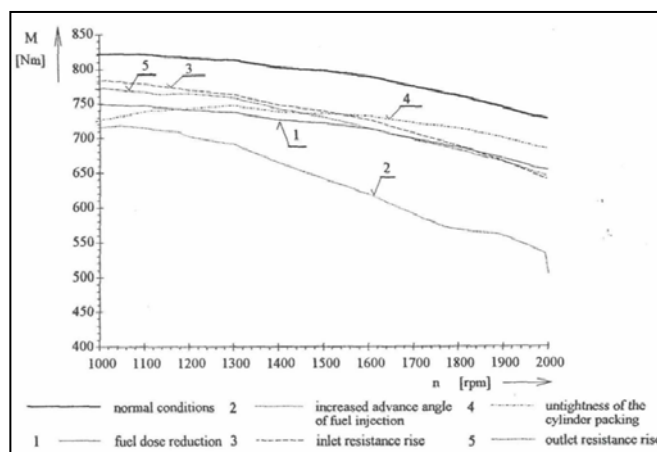


Fig.3 Dependence of the engine torque (M) on the engine speed (n) at difference chosen defects.

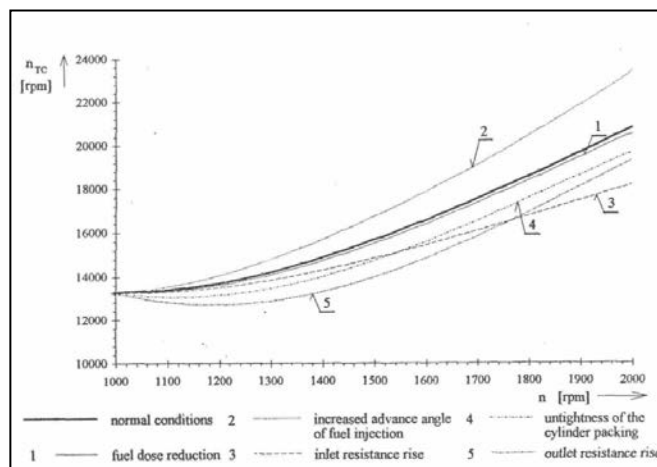


Fig.4 Dependence of the turbocharger speed (n_{TC}) on the engine speed (n) at different chosen defects

7. Conclusions

Ensuring the economic and reliable service of the turbocharged Diesel engines requires diagnostic of their actual technical state. This diagnostics should be simple enough and accurate simultaneously. It should be based on trustworthy information's that reliably correspond to the engine's real technical state. For the prediction of the effect of possible defects on these quantities, it is useful to use a reliable mathematic model which is able to give acceptable results that are close enough to the reality. Starting with the choice of suitable diagnostic quantities, it is then necessary further to precise the model. For the case of the turbocharged Diesel engine it is especially important to precise the two above mentioned diagnostic quantities, i.e. the angular acceleration of the crankshaft and the turbocharger speed.

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NUMERICAL SIMULATION OF THE METHANE INJECTION PROCESS IN THE INLET MANIFOLD OF A DIESEL ENGINE

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Abstract: The main purpose of this paper is a numerical simulation of methane injection process in the inlet manifold of a compression ignition engine. The paper gives an overview of issues related to the turbulence modeling. Computational resources such as the amount of time for the simulation and the memory of the individual models were discussed. While it is impossible to state categorically which model is the most appropriate for a specific application, general guidelines are presented in order to help you choose the turbulence model for the flow you want to model.

KEYWORDS: TURBULENCE MODELING, GOVERNING EQUATIONS, NUMERICAL SIMULATION, DIESEL ENGINE

1. Introduction

Turbulent flow is characterized by a variable-speed field. The flows carry quantities of pulse, energy and mass and simultaneously are the cause of their change.

It is a fact that there is no single universal turbulent model applicable to all types of tasks. The choice of turbulent model will depend on the physical processes occurring in the flow, of the method used, the degree of accuracy, of the computational resources and the time for the simulation. [1]

The system of equations of Navier-Stokes describe the flow fully [2]. The numerical methods which are used for direct solving of these equations are called direct numerical simulations (DNS). Turbulent flows, however, are usually characterized by eddies of various duration in time and with different linear dimensions determined by the acting forces. The results from DNS can be used for verification of the various models of the turbulence. The solving of the averaged equations of Navier-Stokes, instead of momentary, reduces the computational effort to acceptable levels. In the averaged equations unspecified members are appearing. The different models of the turbulence that solve this task can be classified according to the limit linear dimensions Δf , at which the averaging is introduced. All physical processes that take place in linear dimensions greater than Δf are covered by the averaged equations of Navier-Stokes. In this way can be resolved also those with linear dimensions smaller than Δf , as the undetermined members of the averaged equations are described by modeling.

For calculations in direct numerical simulations (DNS), the limit linear dimensions are smaller than those of the Kolmogorov, $\Delta f < \eta$ it is therefore not necessary to introduce modeling. If the limit value is equal to the integral linear dimension, $\Delta f = \ell$, the turbulence model is called a model of Navier-Stokes averaged by the method of Reynolds (RANS).

Widespread and used is the model $k - \varepsilon$, which belongs to this category. If the limit linear size is between η and ℓ , $\eta < \Delta f < \ell$ model is called model for large eddy simulation (LES). Williams [4] proposed an extension of the $k - \varepsilon$ model by reducing its limit linear size below ℓ [2]. This model of the type of very large eddy simulation (VLES), includes the properties of LES models in $k - \varepsilon$ model and thereby is covering a wider range of the dynamics of the turbulence and at the same time is maintaining the computing efforts at an acceptable level.

2. Physical models

The considered several models can be used to describe the different physical processes in simulations of diesel engine with

internal combustion (DEIC). These models are associated with the equations describing the processes meet the required mathematical requirements for simulating processes DEIC [3].

For the most common statistical 3-D flow there are four independent equations describing the main velocity field, namely the three components of the averaged equations for the velocity together with the basic equation of the continuity (or Poisson equation for pressure). There are also more than four unknowns, i.e. the three components of the velocity, pressure and the components in the member $\overline{\rho \tilde{u} \tilde{u}}$. These components are related to the stress of Reynolds, resulting from the transmission of the impulse from the changing velocity field. Since the number of unknowns is greater than the number of equations, a problem occurs at the completion of the system of equations. To complete the system is used the hypothesis for relatively distribution. This hypothesis is applied to the members $\overline{\rho \tilde{u}'' E''}$ and $\overline{\rho \tilde{u}'' Y_i''}$ in the equations of energy conservation and the equations for conservation of mass or for each one of the components K as:

$$\overline{\rho \tilde{u}'' \phi''} = -\Gamma_{K,T} \nabla \tilde{K} \quad (1)$$

The effective diffusion coefficient $\Gamma_{K,eff}$ is defined as the sum of the molecular Γ_K and turbulent $\Gamma_{K,T}$ components. Diffusion coefficients corresponding to the transfer of mass, energy and momentum are shown in Table 1.

Table 1: Coefficients of turbulent exchange

Transferred amount	Mass	Energy	Impulse
Coefficient of transfer	$\Gamma_{Y_i,T}$	$\Gamma_{E,T}$	Γ_T
Symbol	D_T	K_T / c_p	ν_T
Formula	$\mu_T / (\bar{\rho} Sc_T)$	μ_T / Pr_T	$\mu_T / \bar{\rho}$

In the above diffusion coefficients, the values of the turbulent numbers of Prandtl $Pr_T = 0,56$ and Schmidt $Sc_T = 0,9$. Similarly to the equations for the mass and energy, the hypothesis of the turbulent viscosity introduced by Boussinesq is mathematically analogous to the relationship between the pressure and the rate of deformation for a Newtonian fluid. In conclusion, the stresses of Reynolds can be written as:

$$\overline{\rho \tilde{u}'' \tilde{u}''} = -\bar{\rho} \nu_T \nabla \tilde{u} \quad (2)$$

To obtain the complete shape of the equations it is need to be determined the turbulent viscosity. There are several models for its determination, namely:

- algebraic models (universal model of turbulent viscosity, model for mixed lengths);
- models with one equation;
- models with two equations (standard $k - \varepsilon$ model and RNG $k - \varepsilon$ model)
- Reynolds Stress Model (RSM).

2.1. $k - \varepsilon$ model of the turbulence

In this model are solved two additional equations for the transfer of both turbulent components - the turbulent kinetic energy $\tilde{k}$ and the rate of energy diffusion $\tilde{\varepsilon}$. These two components refer to the original variables and they can determine $k - \varepsilon$ linear size and length of time for the formation of quantity with dimension of V_T , and in this way they turn the model into a complete one. This is widely used model for calculation of fluid dynamics (CFD).

$$\tilde{k} = \frac{1}{2} \tilde{u}''^2 \quad (3)$$

$$\tilde{\varepsilon} = \overline{\nu \nabla \tilde{u}''^T : \nabla \tilde{u}''} \quad (4)$$

$$\nu_T = C_\mu \frac{\tilde{k}^2}{\tilde{\varepsilon}} \quad (5)$$

The equation for the balance of $\tilde{k}$ can be obtained from equations (3), (4), (5) and the equation for conservation of the momentum:

$$\frac{\partial(\rho \tilde{u})}{\partial t} + \nabla \cdot (\rho \tilde{u} \otimes \tilde{u}) - \nabla \cdot \overline{\overline{\sigma}} = \nabla p + \rho \tilde{g} + \dot{F}^S = 0$$

The result is:

$$\begin{aligned} \frac{\partial \tilde{\rho k}}{\partial t} + \nabla \cdot (\tilde{\rho k} \tilde{u}) &= -\frac{2}{3} \tilde{\rho k} \nabla \cdot \tilde{u} + \overline{\overline{\sigma}} : \nabla \tilde{u} \\ &+ \nabla \cdot \left[\left(\frac{\mu_{eff}}{Pr_k} \right) \nabla \tilde{k} \right] - \tilde{\rho \varepsilon} + \overline{\overline{W}}^S \end{aligned} \quad (6)$$

$$\begin{aligned} \frac{\partial \tilde{\rho \varepsilon}}{\partial t} + \nabla \cdot (\tilde{\rho \varepsilon} \tilde{u}) &= -\left(\frac{2}{3} c_{\varepsilon 1} - c_{\varepsilon 3} \right) \tilde{\rho \varepsilon} \nabla \cdot \tilde{u} \\ &+ \nabla \cdot \left[\left(\frac{\mu_{eff}}{Pr_\varepsilon} \right) \nabla \tilde{\varepsilon} \right] + \frac{\tilde{\varepsilon}}{k} \left[c_{\varepsilon 1} \overline{\overline{\sigma}} : \nabla \tilde{u} - c_{\varepsilon 2} \tilde{\rho \varepsilon} + c_s \overline{\overline{W}}^S \right] \end{aligned} \quad (7)$$

The flow in the internal combustion engine (ICE) has a variable density, which leads to a change in the volume of the infinitesimal element. This process is called velocity expansion. The member $\left(\frac{2}{3} c_{\varepsilon 1} - c_{\varepsilon 3} \right)$ in ε the equation is for tacking into account the change in the linear dimensions due to the velocity expansion. The member $\overline{\overline{W}}^S$ is due to the interruption of the flow. The new quantities introduced by these equations are constants whose values are determined by some experimental and theoretical considerations. The standard values which are used for calculation of the internal combustion engine are shown in Table 2.

Table 2: Values of the constants in the models of the turbulence.

Константа	$c_{\varepsilon 1}$	$c_{\varepsilon 2}$	$c_{\varepsilon 3}$	c_s	Pr_k	Pr_ε	η_0	β	c_μ
Стандартен $k - \varepsilon$	1,44	1,92	-1,0	1,5	1,0	1,3	-	-	0,09
RNG $k - \varepsilon$	1,42	1,68	(3.1.3)	1,5	0,719	0,719	4,38	0,012	0,0845

2.2. Way of overview of the processes in the wall boundary layer

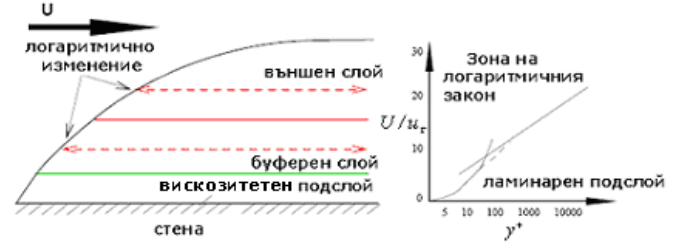


Fig. 1 The area of logarithmic law and laminar sublayer of the wall boundary layer are displayed.

The described in this way methods of examining of the turbulence in the wall boundary layer are valid mainly for homogeneous flow. The type of the flow near the wall requires more attention in the mathematical modeling due to the smaller line dimensions. Near the wall, the flow is divided into inner layer with prevailing viscosity phenomena and an outer layer, where these phenomena are not so significant (separated by the red line, see Figure 2). The layer very close to the wall in the inner layer (below the green line, see Figure 2) is called viscosity or laminar sublayer. The transition from one to the other layer is realized on logarithmic curve. While the change in the speed in the normal direction (perpendicular to the wall) vary linearly with a distance y from the wall in the laminar sublayer it is amended logarithmically. The numerical solution of the equations of the turbulence model in the wall boundary layer is not possible. It is needed more simple function that describes the flow between the last point of the network close to the wall and the wall. Such a function can be obtained through several simplifying assumptions for the flow in the wall boundary layer:

1. The flow is quasi-stationary.
2. The fluid velocity is parallel to the flat walls and vary only in the normal direction.
3. There is no change of the pressure in the flow.
4. In the area near to the wall no chemical reaction occurs and there is no source of injection.
5. The heat losses of the wall are smaller in comparison with the universal.
6. Reynolds numbers are higher (the dynamic viscosity is greater than the laminar viscosity) and the Mach numbers are smaller.

Thus, $k - \varepsilon$ the equations can be written again in accordance with the already said assumptions and in non-dimensional form using the friction velocity (velocity in the wall) u_w determined by

$$u_w = \sqrt{\frac{\tau_w}{\rho_w}} \quad (8), \text{ where } \tau_w \text{ is the tension in the wall boundary}$$

layer and ρ_w is the density of the wall. k is non-dimensional due to the application of u_w^2 , and ε due to the application of u_w^3 / y , where y is the distance along the normal to the wall. Near the wall of the equations for maintaining the components in the other directions (X and Z) disappear leaving only the equations for the direction y . These equations are:

$$c_\mu \rho \frac{k^2}{\varepsilon} y \frac{\partial u}{\partial y} = 1 \quad (u) \quad (9)$$

$$\frac{\partial}{\partial y} \left[\frac{c_\mu}{Pr_k} \rho \frac{k^2}{\varepsilon} y \frac{\partial k}{\partial y} \right] + c_\mu \rho \frac{k^2}{\varepsilon} y \left(\frac{\partial u}{\partial y} \right)^2 - \rho \frac{\varepsilon}{y} = 0 \quad (k) \quad (10)$$

$$\frac{\partial}{\partial y} \left[\frac{c_\mu}{Pr_k} \rho \frac{k^2}{\varepsilon} y \frac{\partial}{\partial y} \left(\frac{\varepsilon}{y} \right) \right] + c_{\varepsilon 1} c_\mu \rho k \left(\frac{\partial u}{\partial y} \right)^2 - c_{\varepsilon 2} \rho \frac{\varepsilon^2}{y^2 k} = 0 \quad (\varepsilon) \quad (11)$$

The solution of these equations for u , k and ε is performed by using Taylor series and in the result is received the the speed of distribution in the wall boundary layer:

$$u_0 = \frac{1}{\kappa} \ln y + B \quad (12)$$

$$\frac{u}{u_w} = \frac{1}{\kappa} \ln \left[c_{1w} \left(\frac{yu}{v_L} \right)^{7/8} \right] + B \quad (13)$$

Where κ is called von Karman constant and has a value of 0,4372 [3] for values of the constants of the model shown in Table 2, $B=5,5$ for smooth walls [3]. The above function is called the law of the wall and because of its logarithmic character, the border area is called the zone of the logarithmic law. In the laminar sublayer, the velocity is calculated by using

$$\frac{u}{u_w} = \left(\frac{yu}{v_L} \right)^{1/2} \quad (14)$$

A correction is made of the places where equation (13) and equation (14) contain the same u_w . For the given constants $u_w = 144$. The calculated velocities for the point of the network closest to the wall is connected with the values calculated by using the functions of the wall. In the final numerical performance is calculated the tangential velocity in the last cell in the vicinity of the wall. This is used to determine the value of u_w , by using the equations (8) и (13). The heat exchange in the zone adjacent to the wall is done in a similar manner.

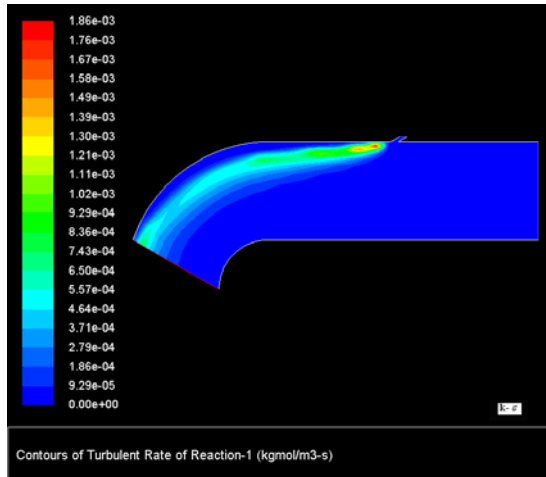


Fig. 2 Turbulent flow in the filler collector of a diesel engine with use of the model $k - \varepsilon$

2.3. RNG $k - \varepsilon$ model of the turbulence

The idea of the renormalization group theory (RNG) is using a random universal force that causes fluctuations of small dimensions. This force represents the impact of larger sizes (including internal and boundary conditions) on the eddies in the inertial range. It is selected in such a way that the global characteristics of the resultant field of the flow are the same as those in the flow driven by the main tension. The equations of the motion for fields with large dimensions are obtained by averaging the infinitely small areas (strips) of the small dimensions in order to be removed. The process of elimination is repeated until infinitely small corrections of the equations are accumulated. The influence of the small dimensions, which have been removed is taken into consideration by dynamic equations for large dimensions of the velocity field due to the presence of the eddies. At the border where successively are eliminated the larger dimensions is obtained $k - \varepsilon$ model.

This procedure covers only the conversion of the viscosity. After complete the elimination of the small dimensions, the random force is excluded from the resulting equations and the internal and boundary conditions are resumed. RNG procedure increases the number of equations, which do not contain undefined constants and the presence of correction. These equations allow the use of the model for the two ranges of the flow - with large and small numbers of Reynolds. The turbulence in ICE is characterized by large changes in density, even at small Mach numbers. In order to adjust the equation for ε of the RNG model for the cases of compressible flow the Rapid Distortion Analysis is used. The final equation for ε contains additional member and extra constant [3]:

$k - \varepsilon$ model applies mainly to the significant values of the relation between the duration of action of the tension in turbulence and the main tension caused by the injection.

In the most general case, $k - \varepsilon$ models contain constants that depend on the geometry and type of the flow, but may determine the main speeds with sufficiently good accuracy. It is however important to note that the hypotheses for the transmission of the components are not the same for all types of turbulent flows [3]. The alternative Reynolds stresses models (RSM) are more accurate in predicting the energy levels in anisotropic turbulent flows. RSM requires a solution of seven additional equations (six for shear stresses for each surface of the element and one member for the diffusion), characterized by a greater calculation intensity. RNG $k - \varepsilon$ model is proved to be particularly suitable for ICE in which the fuel injections lead to high values of the stresses [3].

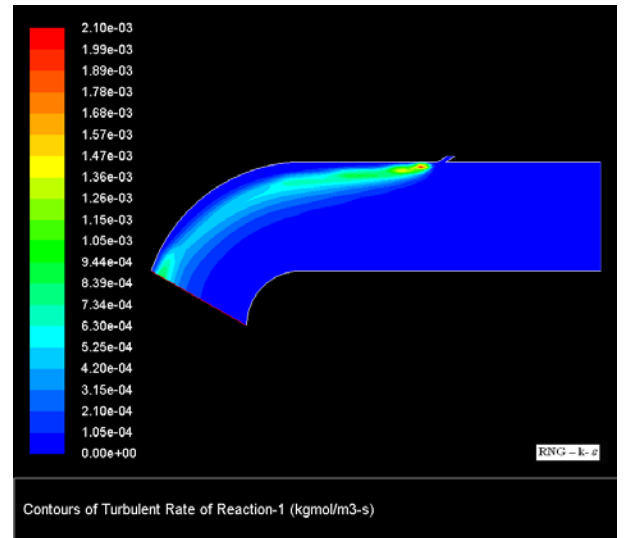


Fig. 3 Turbulent flow in the filler collector of a diesel engine with use of the model $RNG\ k - \varepsilon$

2.4. Real /actual/ (Realizable) $k - \varepsilon$ model of the turbulence

The real $k - \varepsilon$ model is a relatively new development and differs from the standard $k - \varepsilon$ model by two main features:

- The real $k - \varepsilon$ model contains a new formulation for the turbulent viscosity;
- A new equation for the diffusion level is derived ε .

The term "real" means that the model satisfies the precise mathematical limitations on Reynolds stresses related to the phenomena of turbulent flows. Neither the standard $k - \varepsilon$, nor the RNG $k - \varepsilon$ models are completely real.

The immediate benefits of the real $k - \varepsilon$ model is that it more accurately predicts the extent of spreading of both - planar and cylindrical jets. It is also possible to provide the better quality of implementation for flows involving rotation, boundary layers under strong counter pressure gradients, separation and recirculation.

Both the real and RNG $k - \varepsilon$ model show significant improvements over the standard $k - \varepsilon$ model, where the flow properties include highly distorted currents, eddies and rotations. Because the model is relatively new it is not clear in which cases it outperforms the RNG $k - \varepsilon$ model. Initial studies indicate that the real model provides the best performance of all $k - \varepsilon$ versions for some individual flows and flows with complex secondary properties.

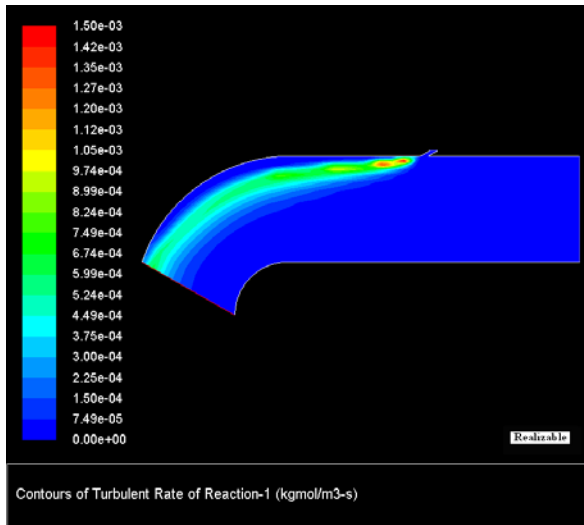


Fig. 4 Turbulent flow in the filler collector of a diesel engine with use of the model **Realizable** $k - \varepsilon$

2.5. Reynolds stresses model (RSM)

Reynolds stresses model is the most detailed model of the turbulence. By excluding the hypothesis for the viscosity of the eddies, RSM provides a solution to the equations of Navier-Stokes, averaged by the method of Reynolds, by solving the equations for transferring of Reynolds stresses, together with one equation for the diffusion rate.

This means that we need 4 additional equations for 2-D flows and 7 additional equations for the transmission for 3-D flows.

Since RSM is taking into account the impact of the turbulent currents, eddies, rotations and rapid changes in the level of tension more accurately than the models with 1 and 2 equations, it has a greater potential to make more accurate predictions for complex flows. The accuracy of the predictions of RSM, however, is still

limited because of the assumptions made in the modeling for the convergence of various members in precisely defined equations for transfer of the Reynolds stresses. Modeling of stresses and the level of diffusion is particularly challenging and often requires achieving of compromise with the accuracy of the RSM prediction.

RSM can not always lead to results that are superior to the simpler models.

However, the use of RSM is mandatory when the flow properties are the result of anisotropy in the stresses of Reynolds. Such are hurricane flows, highly turbulent flows in the combustion chamber, rotating currents and streams causing tensions in the channels.

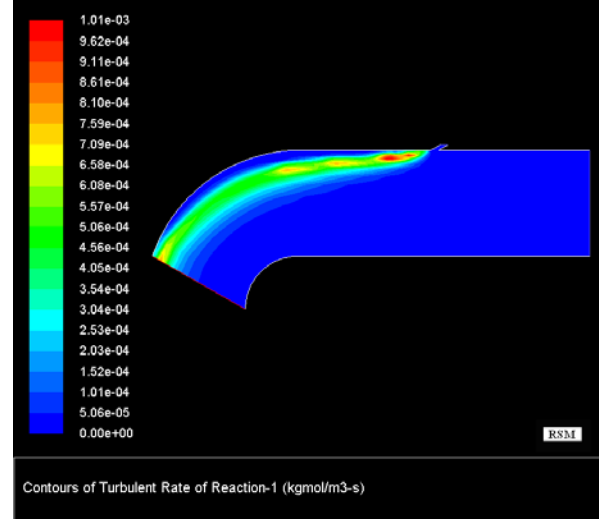


Fig. 5 Turbulent flow in the filler collector of a diesel engine with use of the **Reynolds (RSM)** model

2.6. Model of the type of very large eddies simulation (VLES)

The displayed model of the VLES type is developed by Williams [4]. It is based on two-dimensional assessment which combines the components in at integral size ℓ with those at average filtrating size Δ_f in the inertial sub-range. The model is based on $k - \varepsilon$ model shown above, but in addition it directly solves the turbulent structures with dimensions smaller than ℓ , but still larger than Δ_f , which can not be solved numerically by $k - \varepsilon$ model. In order to achieve this the numerical dimension of the net must be compared with the filter dimension Δ_f . Only the resulting equations of the model are shown. Instead of solving the entire system of equations for transfer only for a one particular level of the linear dimensions, the second level of linear dimensions is used. Yet $\tilde{k}$ and $\tilde{\varepsilon}$ equations are solved with the integral linear dimension ℓ , while all other differential equations are solved with the filter dimension Δ_f , $\eta < \Delta_f < \ell$. In order to use these two levels of the network should be used procedures called scaling up, for passage from Δ_f to ℓ , and scaling down for passage from ℓ to Δ_f respectively.

For the model of the type VLES the following important features can be noted:

All characteristics of $k - \varepsilon$ model are preserved except that VLES model allows solution of greater ranges of linear dimensions, even in the internal sub-range. In this way the VLES model is

certainly avoiding the effect of suppression of the important turbulent structures inherent to the standard $k - \varepsilon$ model. Its major disadvantage, in comparison with the standard $k - \varepsilon$ model is the fact that in order to obtain the average components statistical analysis of the results should be carried out. By combining the characteristics of the standard $k - \varepsilon$ model with those of the LES model, the calculation efforts are maintained at an acceptable level.

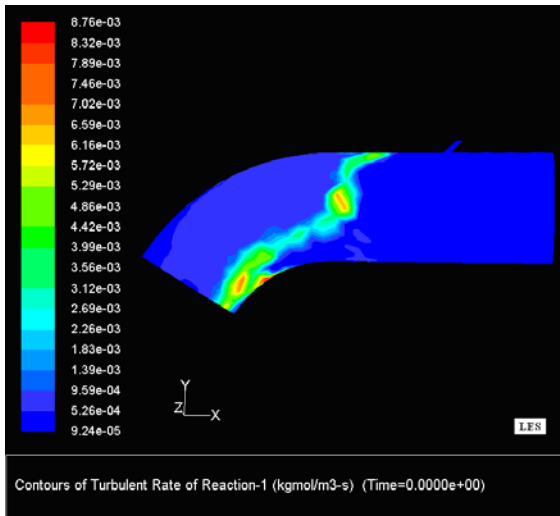


Fig. 6 Turbulent flow in the filler collector of a diesel engine with the use of VLES model

3. Results

For carrying out the Numerical analysis of the process of methane injection (CH_4) in the filler collector of a diesel engine the program FLUENT6.2.1 is used. Calculations are performed under the following boundary conditions:

$$T_0 = 300, K; p_0 = 101325, Pa; T_{wall} = 300, K;$$

$$v_{air} = 50, m/s; v_{inj, CH_4} = 250, m/s;$$

The results of the study of the applied different mathematical models are shown in the corresponding Figures.

4. Conclusions

- For the study of the processes occurring in the filler collector of DEIC while injecting methane in it the most appropriate turbulent model is the LES - model which avoids the effect of suppression of the important turbulent structures inherent to the standard $k - \varepsilon$ model;
- To obtain a numerical solution of the equations of the model of the turbulence in the wall boundary layer it is necessary to apply simplifying assumptions for the flow in the wall boundary layer;
- 2-D ($k - \varepsilon$) models are preferred over 3-D (LES) models in case of insufficient resources of calculation time and in solving a certain type of tasks;
- in studying the influence of the angle of the methane injection in the filler collector of DEIC over the mixture preparation process, the best mixture formation is observed at an angle equal to 60 degrees;
- the optimal combustion process is expected at maximum intensity of the turbulent flow that occurs in the LES model.

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