освобождение ОГ от них после завершения окисления остаточного азота.

При проведении испытаний также была выполнена сравнительная оценка дымности ОГ дизеля по методу, использованному в [2]. Установлено, что дымность ОГ при работе по ЗЦ на 35–40 % ниже по сравнению с работой на атмосферном воздухе. В первую очередь это объясняется повышенным содержанием кислорода в ИГС. В какой-то мере снижению сажеобразования также способствует высокая концентрация диоксида углерода в циркулирующей смеси [3]. Малая концентрация сажи в рециркулируемых газах позволяет уменьшить отложения ее на внутренних поверхностях устройств, обеспечивающих работу дизеля по ЗЦ, и повысить надежность и эффективность всей установки.

Выводы

Проведенный сравнительный анализ показывает, что по большинству показателей рабочего цикла ЗЦ работы дизеля уступает традиционно используемому способу работы. Значительно повысить эффективность работы дизеля по ЗЦ можно, используя его в составе когенерационной установки. Перераспределение составляющих теплового баланса такой установки приводит к увеличению количества теплоты, уносимой с ОГ по сравнению с обычной работой дизеля и возможности ее утилизации для потребностей энергопотребляющего объекта. Количество теплоты, утилизируемой из систем охлаждения и смазки дизеля, при этом практически не изменяется. Повышенный расход топлива дизелем приводит к генерации больших количеств диоксида углерода, что можно считать положительной особенностью при использовании установки в качестве активатора нефтяной скважины.

В случае применения установки для энергоснабжения изолированного объекта расход топлива приобретает важное значение. Топливная экономичность дизеля в ЗЦ работы может быть улучшена использованием ИГС с теплоемкостью, близкой к теплоемкости воздуха. При этом необходимо применение химической нейтрализации избыточного диоксида углерода и включение в состав ИГС таких газов как аргон или же менее эффективного азота.

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SIMULATION OF DIESEL ENGINE AND VARIABLE GEOMETRY TURBOCHARGER (VGT) WITH VANELESS TURBINE VOLUTE

Introduction

It is known that from the thermodynamic point of view, the turbocharger system is attractive because it makes use of the exhaust gas energy [1]. Thus, turbocharging systems are widespread not only in traditional internal combustion engines (such as diesels and gasoline) but also in engines running on alternative fuels such as natural gas. It is known that natural gas is a promising alternative fuel to meet a strict engine emission standards in many countries [2]. Natural gas engines can operate on lean burn. By increasing boost pressure level, a lean burn natural gas engine can produce higher power and torque and thus its full-load thermal efficiency can even be very close to that of gasoline or diesel engine [3]. It favors the constant development of supercharger technologies.

On the other hand, the transport internal combustion engine is characterized by a large number of transient and equity modes. Engines with a non-adjustable turbocharger have matched piston and boost characteristics in a narrow range of calculated modes (such as rated power and torque) whose share in the operation does not exceed 15 percent. Operation in a different mode causes a mismatch in characteristics of internal combustion engine (ICE) and the turbocharger which in turn leads to higher emissions, worse fuel consumption and low acceleration performance. Adjustment of piston and turbo systems is an urgent task that can significantly improve the technical and economic characteristics of the engine across the full range of operation. Moreover, an opinion exists today that without fuel and air delivery systems adjustment it is impossible to meet the latest stringent emission regulations such as Euro 5 and nearest Euro 6.

Current situation

Following advanced research with full access to the International patent base WIPO [4], the problem of turbochargers adjustment was investigated. It was established that turbine adjustment of the turbocharger is more efficient than compressor adjustment. The recent trends in turbocharger adjustment technologies have been: from 1980 to 1992, adjustment by changing the partiality and throttling of the flow at the turbine inlet; from 1992 to 1998 – partial exhaust gas turbine bypass, and from 1998 up to now the major method has been the nozzle ring variable geometry. It was not taken into account new electric supercharger technologies because they have no commercialization yet.

World experience shows that nozzle ring adjustment is the most effective method of turbocharger adjustment as it allows air supply tweaking to achieve the optimal engine output across the full range of operation modes: from idle speed to the maximum value. However, the main drawback of this method is a complex actuation mechanism for nozzle rings that results in a high cost and low reliability of such turbochargers.

Hence, a new method needs to be developed to allow turbocharger adjustment of centripetal turbines with vaneless turbine volute which would minimize the losses in turbine efficiency when changing (reducing) the flow. At the same time, the new design must offer simplicity and lower costs in comparison with the nozzle ring method and have enough adjustment depth in order to provide an optimal LAMBDA (the ratio of actual air-fuel ratio to stoichiometry for a given mixture) across the full operating range of the engine.

A new patented method for centripetal vane machine adjustment (control) [5-7] satisfies the above requirements and has passed all the stages of “scientific implementation” from the idea up to a prototype. The basic design of the new method is shown in Fig.1. The new method is based on the cross-section variation of the turbine volute acceleration section end by means of a specially shaped element *1* located in the inlet part of the volute *2*. The adjustment is carried out by curvilinear progressive motion of the shaped element *1* in the direction of incoming gas flow or in the opposite direction, whereby the geometrical shape, location and the size of the flow area of the volute acceleration section are determined according to the curvilinear progressive motion of the shaped element. Position *I* of the shaped element corresponds to the minimal cross-section of the end of volute acceleration section A_{min} and, accordingly, to the maximal depth of turbine adjustment. Position *II* of the shaped element corresponds to the maximal cross-section of the end of volute acceleration section A_{max} and, hence, to the minimal depth of adjustment.

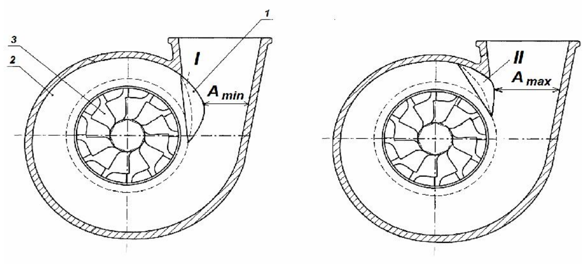

Fig.1. Method for adjustment of a centripetal turbine with vaneless volute (vaneless distributor)

1 – specially shaped element; 2 – turbine volute; 3- wheel

A joint refined model of ICE and VGT

To conduct the comparison analysis of the ICE with various variable geometry boost systems, the joint model of ICE and turbocharger was used. The model was developed by V. Petrosyancz [8] and gives a most adequate representation of an ICE with a non-adjustable, free turbocharger. The model was modified to allow for simulation of a variable geometry turbocharger engine. The model structure is shown in Fig.2

The major model function is described through the following equation:

$$Z = f(X, Y, U) \qquad (1)$$

where Z $(Z_1, Z_2 ... Z_n)$ –vector of output parameters, X $(X_1, X_2 ... X_n)$ - vector of geometrics, Y $(Y_1, Y_2 ... Y_n)$ – vector of physical constants, U $(U_1, U_2 ... U_n)$ – vector of input parameters determining the engine operating mode.

The model refinement is due to the introduction into the turbine model of a variable geometrics vector $\overline{X_T'}$ which changes during adjustment. Consequently, the major functions defining the turbine flow and efficiency characteristics become a function of three variables (a two-variable function in the original model) and take the following form:

$$G_T = f(\pi_T, \omega_T, \overline{X_T'}) \tag{2}$$

$$\eta_T = f(\pi_T, u/c, \overline{X_T'}) \tag{3}$$

For nozzle ring geometry adjustment, the variable vector $\overline{X_T'}$ is defined by the angle of vane rotation in the nozzle ring:

$$\overline{X_T'} = \{\alpha_e\} \tag{4}$$

where α_e is the effective angle of vane rotation in the nozzle ring.

In case of adjustment in a vaneless turbine volute of the turbocharger:

$$\overline{X_T'} = \{A, r_u\} \tag{5}$$

where A is the cross-section of the end of volute acceleration section, r_u - section gravity center radius.

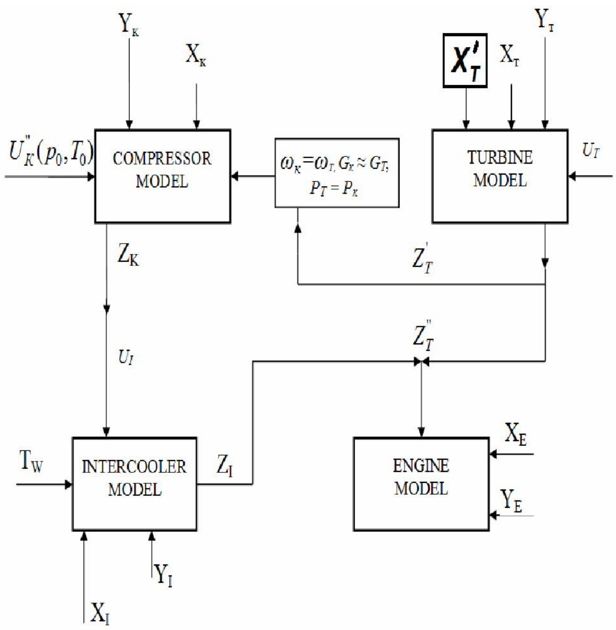

Fig.2. A view of the model structure

The major functions defining the compressor characteristics remain unchanged:

$$G_{\hat{E}} = f(\pi_{\hat{E}}, \omega_{\hat{e}}) \tag{6}$$

$$\eta_{\hat{E}} = f(G_{\hat{E}}, \omega_{\hat{E}}) \tag{7}$$

$$Đ = f(G_{\hat{E}}, \pi_{\hat{E}}) \tag{8}$$

where P – compressor surge line.

A precondition for the model convergence is the balance of power and rotating speed of the turbine and compressor rotor:

$$Đ_{\grave{O}} = Đ_{\hat{E}} \tag{9}$$

$$\omega_{\hat{E}} \approx \omega_{\grave{O}} \tag{10}$$

The joint model described above was used for the simulation of a diesel with different systems of variable geometry turbocharging.

Simulation Procedures

A simulation was conducted of the joint operation of a diesel and a turbocharging system with adjustment by means of varying the cross-section of the end of turbine volute acceleration section. A four-cylinder turbocharged diesel engine with bore 120 mm and stroke of 140 mm served as the object of simulation. The engine had no EGR system. The diesel was equipped with a commercial turbocharger TKR - 7,5 TV - 02 with the end of turbine volute acceleration section effective cross-section of $\boldsymbol{A}$ = 1055 mm^2. For simulation, the following operating modes were selected:

- Rated power operation mode with P_e = 117,7 kW, n = 2000 rpm,
- peak torque mode with P_e = 106,6 kW, n = 1500 rpm,
- torque curve mode with P_e = 106,6 kW and n = 1000 rpm.

The following values of $\boldsymbol{A}$ were chosen for the simulation : 1055 mm^2, 910 mm^2, 800 mm^2, 745 mm^2, 600 mm^2. For every cross-section above the simulation was conducted for 3 operating modes. For further analysis the values of $\boldsymbol{A}$ were selected to minimize brake specific fuel consumption (BSFC) and maintain an LAMBDA that would be acceptable from the point of view of minimizing smoke emission. The simulation was carried out under the limitations on maximum combustion pressure and exhaust gas temperature (660 Celsius degree) before turbine, as well as the turbocharger maximum rotor speed and no surging of compressor. To validate the model the simulated and the measured data were compared. The comparison was conducted on the simulation and experimental data for the same engine running in 3 modes chosen for the experiment with a commercial turbocharger TKR - 7,5 TV - 02 and $\boldsymbol{A}$ = 1055 mm^2.

Results and Discussion

To find the optimal values of $\boldsymbol{A}$ the influence of adjustment in turbocharger with vaneless turbine volute on the diesel parameters was investigated. The engine parameters for various effective cross-sections of the turbine volute acceleration section end ***A*** for the 3 investigated modes are shown in Table 1.

Rated Power Operation Mode

In the rated power operation mode the minimal BSFC 229,8 g/kWh was obtained with ***A*** = 1055 mm^2. Under these conditions the LAMBDA acquires the value of 2,03. In the rated power mode the BSFC and LAMBDA for an adjustable turbocharger match those for a non adjustable. Reducing the value of ***A*** (Table 1) to 910 mm^2 leads to a deterioration of BSFC by 3 g/kWh while LAMBDA reaches 2,132. A drop in efficiency is explained by the increase in pumping losses exceeding the gain in engine indicated efficiency due to the increase in exhaust gas resistance caused by a reduction in ***A***.

Table 1. Simulation Results

P_e= 117,7 kW, n = 2000 rpm				
A (mm^2)	*BSFC* (g/kWh)	LAMBDA	p_i (MPa)	p_p (MPa)
1055	229,8	2,03	1,429	0,0403
910	232	2,12	1,449	0,0576
P_e = 106,6 kW, n = 1500 rpm				
A (mm^2)	*BSFC* (g/kWh)	LAMBDA	p_i (MPa)	p_p (MPa)
1055	215,5	1,68	1,595	0,0012
910	213,9	1,83	1,602	0,0072
800	214,65	1,94	1,615	0,0184
P_e = 55 kW, n = 1000 rpm				
A (mm^2)	*BSFC* (g/kWh)	LAMBDA	p_i (MPa)	p_p (MPa)
1055	246,5	1,36	1,267	-0,0105
910	238,9	1,51	1,263	-0,0146
800	233,4	1,69	1,2595	-0018
745	231,2	1,79	1,2576	-0,0198
600	228,8	2,094	1,2581	-0,0193

Mode of torque curve with n = 1000 rpm

Peak Torque Mode

The joint work of a diesel and a VGT is possible with values ***A*** from 1055 to 800 mm^2. The best BSFC 213,9 g/kWh and LAMBDA 1,83 were obtained with ***A*** of 910 mm^2. In this case BSFC improves by 2 g/kWh in comparison with a non-adjustable turbocharger, LAMBDA grows by 9% (see Table 1). Reduction in cross-section of the end of turbine volute acceleration section to 800 mm^2 has practically no effect on BSFC (an increase by 1 g/kWh does not exceed the simulation error margin) but leads to a rise in LAMBDA from 1,833 to 1,935 that serves as evidence of the higher combustion efficiency and reduction in smoke emission. Thus the optimal value of ***A*** for this mode is 800 mm^2.

This mode is characterized by the greatest variation range for the value of ***A*** enabling the joint work of a diesel and a turbocharger. Effective cross-section of the end of turbine volute acceleration section ***A*** can acquire the values from 1055 to 600 mm^2. The optimal value of ***A*** is 600 mm^2 with BSFC of 228,8 g/kWh and LAMBDA of 2,094. In this case, the diesel brake specific fuel consumption improves by 17,7 g/kWh in comparison with the diesel with a non-adjustable turbo. The value of LAMBDA grows by 35%.

Discussion of main results

As we can see from the results above, the efficiency of application of variable geometry turbocharging systems is most tangible in torque curve modes located in the range of speed from peak torque to the mode of torque curve with *n* = 1000 rpm. In this range, a non-adjustable turbocharger satisfies the diesel operation requirements neither in fuel consumption efficiency nor in emission levels. Values of LAMBDA around 1,0 produce an unacceptable level of smoke.

With the best efficiency for a simulated diesel in mind, a dependence was established between the effective cross-section of the end of turbine volute acceleration section ***A*** from engine speed *n* (algorithm of adjustment) under operation at torque curve. The algorithm of turbocharger adjustment with vaneless turbine volute is shown in Fig.3. It's easy to see that this algorithm can be described by an almost straight line.

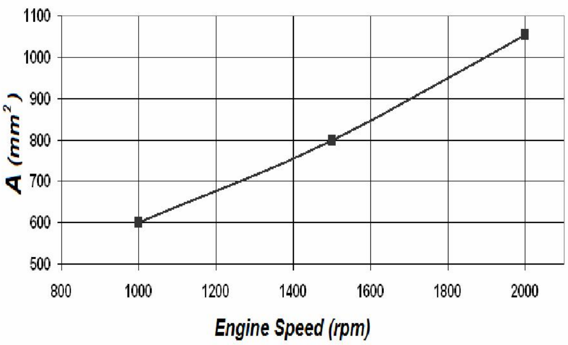

Fig.3. Effective cross-section of the end of turbine volute acceleration section vs engine torque curve modes

Based on the algorithm of adjustment shown in Fig.3 the main engine and turbocharger dependences were obtained for a diesel supplied with a commercial non-adjustable turbocharger, and a VGT turbocharger running on torque curve modes and shown in Fig.4.

As we can see from Fig.4, turbocharger adjustment with vaneless turbine volute allows to improve BSFC in the operation mode with engine speed of 1000 rpm by 17,7 g/kWh (chart b, Fig.4).

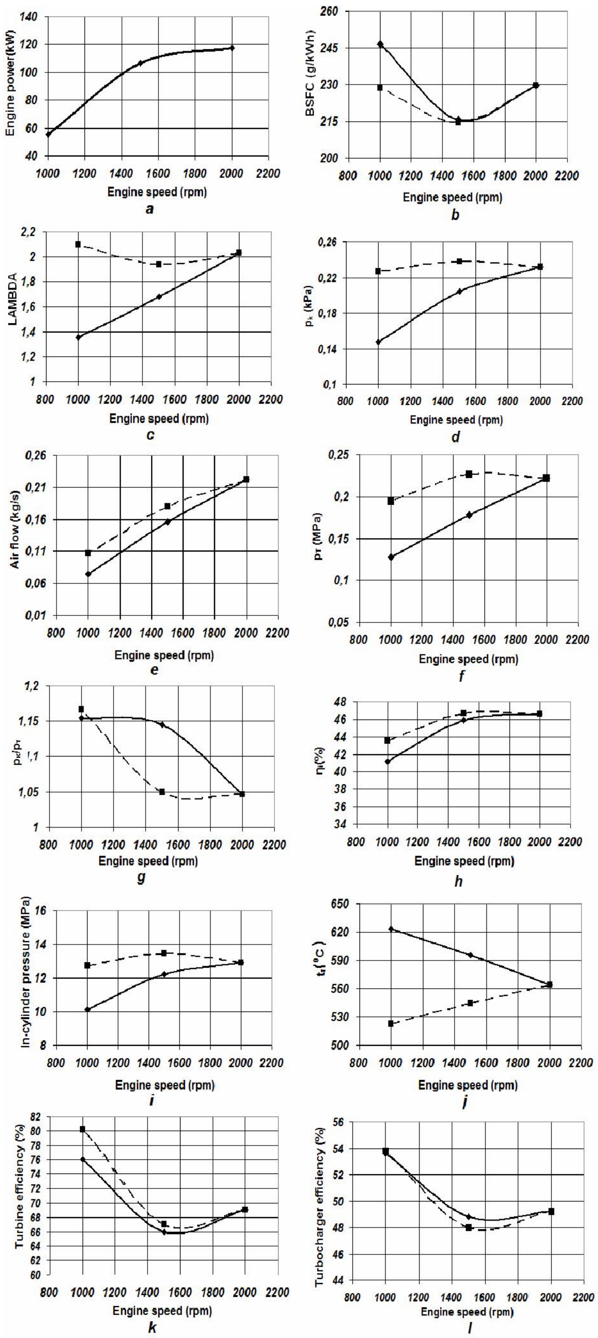

Fig.4. The main engine and turbocharger dependences a – j – engine parameters; k,l - turbocharger parameters

In this case the mechanism of the adjustment influence on diesel working process is as follows. Raising boost pressure p_k from 141,7 kPa up to 226,5 kPa (chart d, Fig.4) results in an LAMBDA increase from 1,355 to 2,094 (chart c, Fig.4). The pressure of exhaust gases before turbine p_T has also increased from 127,4 to 194,6 kPa (chart f, Fig.4). However, because the turbine efficiency η_T grows by 4,2% (chart k, Fig.4) the ratio p_k/p_T also increases (chart g, Fig.4) which indicates a reduction in engine pumping losses. Thus the improvement of combustion processes has an effect on engine indicated efficiency η_i that increases by 2,3% (chart h, Fig.4). Besides, the maximum in-cylinder pressure p_z rises as well, growing from 10,3 up to 12,71 MPa (chart i, Fig.4). An increase of fresh air delivery into the cylinder results in a decrease in exhaust gas temperature before turbine t_T by 100ºC (chart j, Fig.4).

In the peak torque mode, adjustment of turbine with vaneless turbine volute also results in an improvement of combustion efficiency. Thus, BSFC dropped by 2,8 g/kWh, LAMBDA increased from 1,609 to 1,94. The temperature of exhaust gases before turbine t_T dropped by 64ºC, indicated efficiency η_i increased by 1 percent, p_z grew from 11,98 to 13,43 MPa, turbine efficiency η_T increased by 2%. The p_k/p_T ratio dropped from 1,164 to 1,074, which signifies an increase in pumping losses.

In the rated power mode all parameters of an adjusted and a non-adjusted turbocharger remained unchanged.

When reviewing the results of the simulation in the mode of torque curve with P_e = 55 kW and n = 1000 rpm, one can spot a characteristic trend in the variation of the basic parameters of the engine and VGT with vaneless turbine volute during adjustment. Table 2 demonstrates the main parameters of diesel with VGT.

Table 2. Diesel parameters with VGT

Diesel parameters		
List of parameters	***A*** = 1055 mm²	***A*** = 600 mm²
Brake specific fuel consumption BSFC, g/kWh	246,5	228,8
LAMBDA	1,36	2,09
Boost pressure $p_к$, кPa	147	226,5
Air flow *AF, kg/s*	0,075	0,107
Indicated efficiency, η_i, %	41,2	44
Pumping losses share Δ	-0,01	-0,018
Turbine parameters		
Turbine velocity ratio *u/c*	0,68	0,7
Internal turbine efficiency η_{iT}, %	76	70,1
Turbine efficiency, η_{eT},%	47	49,6
Rotor speed, *rpm*	60000	92100
The level of reactivity, ρ	0,37	0,39
Flow outlet angle from vaneless turbine volute α_1, degrees	16	11
Speed ratio of vaneless turbine volute, φ	0,96	0,92
Angle of the flow attack *i*, degrees	9,9	7,2
Coefficient of wheel's velocity, ψ	0,77	0,62

As it can be seen from Table 2, a reduction of effective cross-section A from 1055 mm^2 to 600 mm^2 leads to a decline in turbine internal efficiency. This can be explained by a change in the flow attack angle and redistribution of losses in the flow-through parts of the turbine. However, we also witness an increase in the effective turbine efficiency from 47% to 49% due to a drop in the share of mechanical losses introduced to the engine effective efficiency when increasing the rotor speed (by 32 000rpm). This results in better BSFC and higher LAMBDA values in the engine.

Conclusions

1. A simulation of a diesel engine and variable geometry turbocharger with vaneless turbine volute was conducted. The adjustment of turbocharger by changing the cross-section of the end of turbines volute acceleration section allows to significantly improve the combustion efficiency especially in the torque curve modes. Thus in the peak torque mode with no changes in fuel consumption the adjustment allows to reduce smoke emission by increasing LAMBDA up to 1,94. In the mode with P_e = 55 kW and n = 1000 rpm when LAMBDA increased to 2,01, the fuel consumption dropped by 17,7 g/kWh.

2. A curve of effective cross-section of the end of turbine volute acceleration section – vs – engine speed under diesel torque curve modes was obtained. It allows to find the rational values of A across the full range of engine speeds from 1000 to 2000 rpm in the engine torque curve mode. Thus, for rated power mode the value of A is 1055 mm^2, for peak torque mode - 800 mm^2 and for the mode with n = 1000 rpm – 600 mm^2.

3. It was established that the adjustment algorithm for VGT with vaneless turbine volute can be described by an almost straight line. The depth of adjustment is 43,1%.

Nomenclature

P_e : Engine power
n : Engine speed
p_i : Engine indicator pressure
p_p : Engine pumping pressure
A : (Effective) cross-section of the end of volute acceleration section
p_k : Boost pressure
p_T : Exhaust gases pressure before turbine
G_T : Exhaust gases flow
π_T : Turbine's pressure ratio
ω_T : Turbine rotor's rotating speed
G_K : Air flow
π_K : Compressor's pressure ratio
η_K : Compressor efficiency
ω_K : Compressor rotor's rotating speed

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