# CASCADE EXCHANGE PRESSURE SUPERCHARGING SYSTEM OF THE TRANSPORT ENGINE WITH DEEP COOLING OF INTAKE AIR

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**Keywords:** cascade pressure exchange, thermal compressor, working cycle, temperature, power efficiency, performance, load characteristic.

**Abstract:** The new principle of the organisation of working process of the combined supercharging system of Internal Comdustion Engine (ICE) with the Cascade Pressure Exchanger(CPE) has been described. It allows considerably to raise level of forcing of the engine by supercharging at the expense of expansion of effective air supply area and coolings of supercharging air to temperature below an ambient without attraction of additional mechanical energy on refrigeration cycle realization. Substantive postulates of imitating model of work of combined engine(CICE) have been stated. Some results of calculation-experimental investigations of supercharging system of the engine 6ЧН12/14 have been adduced.

## Introduction

Possibility of improving of operational parameters of Diesel engines by application of gas-turbine supercharging is appreciably restricted to the problems connected with an increase of temperature of an air charge in cylinders and unsatisfactory quality of air supply on the off-design conditions. Noted deficiencies are especially considerably manifested at operation of the engine in the conditions of hot climate. The high ambient temperature causes falling of plant power in view of restriction of a cyclic fuel supply because of decrease in density of air (a mass charge of air) and inadmissibility of excess of the maximum temperature of the cycle of thermo-intensity limits of materials of the turbine and cylinder-piston group.

Thus decrease of the ratio of boundary temperatures of thermodynamic cycle predetermines decrease of plant efficiency on the average on 6.3 % on everyone 10K air temperature increases of on an entry in the piston part of the engine.

From this point of view, the application of the cascade pressure exchangers (CPE) - the devices based on a direct energy exchange between exhaust gases and compressed air - in the capacity of the basic air supply unit is considered to be long-term trend of development of supercharging systems. Basic difference of the CPE from known wave exchangers of «Comprex» supercharging systems is essentially large power efficiency of exchange processes, low sensitivity of parameters of work to deviations of operating conditions from design conditions, rather low rotational speed of its rotor. High efficiency of the CPE is manifested in considerable excess of the consumption of compressed air concerning compressing gas [1].

Tests of the pilot supercharging system with the CPE on the basis of the Diesel engine 6ЧН12/14 have confirmed its ability to provide pressure supercharging invariability in all range of speed regimes of the engine. And on a nominal speed regime at supercharging pressure 230 кPa and temperature of compressing gases 700K the excess of supercharging air concerning the consumption through a piston part of the engine has made 82,5 % at insignificant excess of a backpressure to exhaustion of gases from the cylinders of supercharging pressure level.

Noted feature of the CPE work allows not only to realize almost any demanded external characteristic of supercharging, but also to carry out deep cooling of supercharging air by means of gas-expansion excess of air discharged under pressure into the CPE to temperature below the ambient one with its subsequent use in the capacity of a coolant of the second step of the cooler. The circuit design of Supercharging System of Deep Cooling of supercharging air (SSDC) is shown on fig. 1.

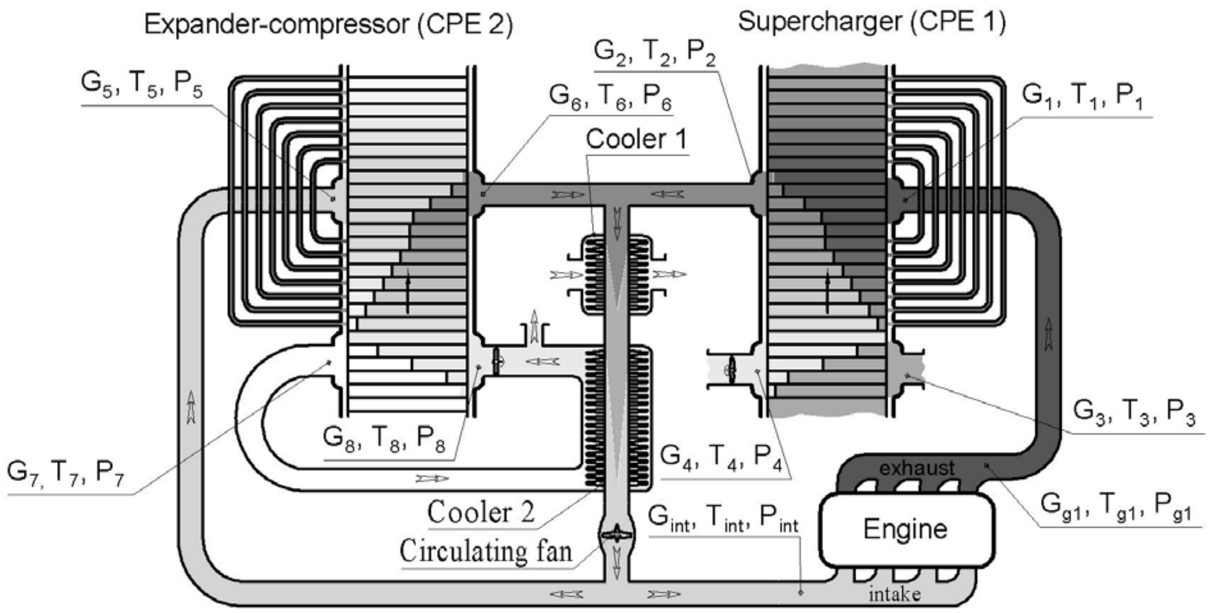


**Figure1: Circuit diagram of the SSDC – CPE**

In the given installation one of the units of the cascade pressure exchange (the CPE1) carries out function of the supercharger of supercharging air, another (the CPE2) carries out function of the gas-expansion machine (expander)-compressor.
Necessary condition of effective work of the analyzed system is the coordination of consumption characteristics of its compound units. Search of dimensional parameters of the cascade exchangers, which have been adjusted on the required operating mode of the combined engine, is carried out by the methodology described in the work [2].

Relationships of sizes of the CPE1 and the CPE2 depend on degree of raise of pressure $\pi_к$. With increase of $\pi_к$ the share of air discharged under pressure by the CPE I decrease. This air is taken away on realization of a refrigeration cycle. At the same time the results of calculation of systems of various head pressure show that decrease in the consumption of the coolant exerts insignificant influence on cooling ability of the SSDC at raise of supercharging pressure $P_s$ from 0.18 to 0.3 MPa. It is explained by the compensatory effect of decrease in temperature of the coolant at the expense of raise of degree of its expansion in the gas-expansion machine-compressor.

**Imitating model of ssdc - cpe work**

To estimate the parameters of the SSDC - CPE work with the fixed dimensional parameters in a wide range of CICE operating conditions it is expedient to use the mathematical model simulating working processes of compound units of the system simultaneously.

Substantive principal propositions of imitating model of the CICE work on the off-design conditions are resulted below. The model is based on search of conditions of joint work of the CPE units and the piston part of the engine.

The basic assumptions of calculation are: one-dimensionality of a flow of working mediums in the flowing elements of system, absence of thermal and mechanical losses in connecting pipe-lines, absence of leaks in mobile integrations of the CPE rotor. In addition the head pressure of the circulating ventilating fan 1 is supposed to be invariable in an offered range of search of design values of working mediums consumptions in the pressure line of the CPE supercharging system.

The condition of joint work of the combined engine units on the stable regime is the balance of working mediums consumptions in head pressure elements of air and gas mediums:

$$G_{int}=G_2+G_6-G_5; \quad (1)$$

$$G_{g1}=G_1 \quad (2)$$

Generally the relations of consumptions in pressure pipe-lines of supercharging system depend on state variables of flows and the pressure differential between high pressure (HP) windows of every CPE.

The task of search of the regime of joint work of units comes to definition of supercharging pressure $P_s$ and the pressure differential between HP windows of both CPE at which the conditions (1) and (2) are realized.

According to the assumptions the pressure differential $\Delta P_{5\text{-}6}$ between high pressure (HP) windows of the CPE2 depends on the head pressure of the circulating ventilating fan $\Delta P_f$ and hydraulic resistance of coolers 1 and 2, correspondingly $\Delta P_{cool1}$ $\Delta P_{cool2}$.

$$\Delta P_{5\text{-}6} = \Delta P_f - \Delta P_{cool1} - \Delta P_{cool2} \quad (3)$$

The hydraulic resistance of coolers is expressed by the simplified dependence:

$$\Delta P_{cooli} = \zeta_{\Sigma i} \cdot \frac{\rho_i}{2} \cdot \left( \frac{G_{oi}}{F_i} \right)^2, \quad (4)$$

where $\zeta_{\Sigma i}$ - total factor of hydraulic resistance, the -volumetric consumption and flow density accordingly; $F_i$ -reference area of the flow cross-section of the heat exchanger.

The pressure differential $\Delta P_{1\text{-}2}$ between high pressure (HP) windows of the CPE1 depends on the air consumption $G_s$ through the piston part of the engine. In the same time consumption $G_s$, in its turn, depends on the boost pressure $P_s$ and pressure differential $\Delta P_{s\text{-}g1}$ between the ICE inlet and outlet valves. And then:

$$\Delta P_{int\text{-}g1} = \Delta P_f - \Delta P_{1\text{-}2}\ \Delta P_{cool1} - \Delta P_{cool2}.$$

The working cycle of the ICE, the CPE 1, and the CPE 2 is simulated on the basis of supposed values of $P_{int}$, $\Delta P_{cool1}$ and $\Delta P_{cool2}$. By results of calculation of values $G_{int}$, $G_{g1}$, $G_1$, $G_2$, $G_5$, $G_6$ the hydraulic resistances of air coolers $\Delta P_{cool1}$, $\Delta P_{cool2}$ of the first and second stage are specified, and the values of pressure differentials $\Delta P_{1\text{-}2}$ and $\Delta P_{5\text{-}6}$ in high pressure (HP) windows of exchangers. are corrected.

The balance of consumptions on a condition (2) is attained by respective alteration of $\Delta P_{1\text{-}2}$ (for example, at $G_1 > G_{g1}$ it is necessary to reduce $\Delta P_{1\text{-}2}$). In case of balance default on the equation (1) the search of regimes of joint work is carried out at other values of $P_2$. Pressures in high pressure (HP) windows of the cascade exchangers and cross-sections of gas-distributing channels engine are determined by expressions:

$$P_1=P_2 + \Delta P_{1\text{-}2};$$

$$P_{int=P2} - \Delta P_{cool1} - \Delta P_{cool2};$$

$$P_{g1}=P_1$$

$$P_6=P_2$$

$$P_5 = P_6 + \Delta P_{5\text{-}6}$$

The temperature of supercharging air before an inlet collecting channel of the engine and also air temperature in the window 5 of the CPE2 is determined on the basis of calculation of thermal flows in the cooler1 simultaneously with temperature calculation of low temperature coolant in the cooler2.

In the algorithm resulted above, calculation of operational parameters of the CPE units is often repeating calculation operation. Gas-dynamic processes in the flowing elements of exchangers can be most fully described by the numerical method «Disintegration of arbitrary rupture», and are in detail stated by authors in the work [3] with reference to a working cycle of wave pressure exchanger. At a stage of preliminary determination of consumed characteristics of the cascade exchangers the sufficient accuracy is provided by method of «Body-section diffusions», offered below [4].

According to the given method, volume нhead pressure cells and mass transferring channels of the CPE is conditionally divided on the equal elementary calculated layers by a motionless grid. The magnitude of layers is chosen from a condition of full dissolution in them of the working medium which has arrived during a rated temporary step from contiguous layers (fig.2a.), and every layer has homogeneous state variables. At the first stage of computation all elementary calculated layers are considered as the closed thermodynamic systems (layers of the fixed weight), except the first and the last which are open thermodynamic systems. Change of the state variables in an elementary rated layer is generally considered as result of simultaneous affecting of three factors on a working medium originally containing in this layer. The factors are: heat brought from the outside, the work of pushing through of conditional boundary lines between contiguous layers, the work of dilution by the working medium arriving from the side of contiguous layers.

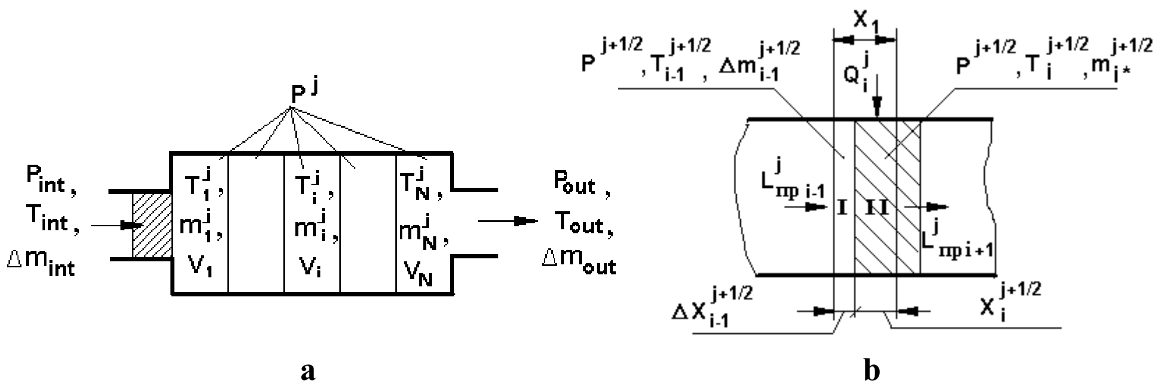


**Figure 2: Circuit designs:**
**a** – The flowing cell; **b** – The elementary layer

It should be meant that the work of deformation of the elementary layer is equivalent to the sum of works of pushing through of conditional interfaces from the side of contiguous layers:

$$\Delta L_{DEF_i} = \Delta L_{PUS_{i-1}} + \Delta L_{PUS_{i+1}} \,.$$

At an assumption about insignificant difference of compression polytrope index for various layers, the energy of deformation of a separate layer represents a part of the general work of pushing through of a working medium of all volume of the cell. This part corresponds to a relative volumetric fraction of the rated element (layer). Really, total work of pushing through of the working medium of all volume of a cell is spent for compression of the working mediums containing in rated layers.

$$\Delta L_{PUS_\Sigma} = \sum_{i=1}^{N} \frac{n_i \cdot R_{г}}{n_i - 1} \cdot T_i \cdot m_i \cdot \left( \pi_k^{\frac{n_i - 1}{n_i}} - 1 \right), \qquad (9)$$

where $T_i$ – air temperature in i-N layer;

$m_i$ – weight of air in i-N layer;

$n_i$ – compression polytropic index in i-N layer,

$R_g$ -gas constant;

N – number of rated layers.

In view of that for equal elementary layers with equal pressure $T_i \cdot m_i = const$, we will gain

$$\Delta L_{DEFi} = \frac{V_i}{V_\Sigma} \cdot dL_{PUS_\Sigma} \ . \qquad (10)$$

According to the first law of thermodynamics for open system, the change of parameters in extreme rated layer from the side of working medium arriving is presented by the equation:

$$\Delta U_1 = (C_v T_{C\,\text{int}} + \frac{W^2}{2})\Delta m_{C\,\text{int}} + \Delta L_{DEF1} + \Delta Q_1 \ , \qquad (11)$$

where $\Delta U$ - an *increment* in an internal energy of the working medium;

$C_v$ -gas specific mass tsochore heat capacity;

$W^2/2$ -kinetic energy of the flow in the minimum cross-section of an inlet (outlet) window;

$\Delta m_{\text{int}} = \mu f_{\text{int}} \cdot P_{\text{int}} \sqrt{\frac{2 \cdot k_г}{k_г - 1} \cdot \frac{1}{R_г \cdot T_{\text{int}}} \cdot \left[ \left( \frac{P}{P_{\text{int}}} \right)^{\frac{2}{k_г}} - \left( \frac{P}{P_{\text{int}}} \right)^{\frac{k_г+1}{k_г}} \right]} \cdot \Delta\tau$ - the mass of the working medium which has arrived into the cell during a time rated step;

$\Delta Q_1$ -heat flow at heat exchange between the working medium and walls of volume of the cell;

$T_{C\,\text{int}}$ - temperature of the inlet gas.

For a case when $C_V = const$:

$$\Delta T_1 = \frac{(C_v T_{C\,\text{int}} + \frac{W^2}{2})\Delta m_{C\,\text{int}} + \Delta Q_1 + \Delta L_{DEFi} - C_v T_1 \Delta m_{C\,\text{int}}}{C_v m_1} \ , \qquad (12)$$

where $m_1$ - initial mass of gas in a layer;

$T_1$ - initial temperature of gas in the elementary rated layer;

$\Delta T_1$ - change of temperature in the first elementary rated layer.

Calculation of state variables on each rated step is carried out in two stages for internal rated elements. At the first stage the parameters in internal layers are changing under the influence of the brought heat and energy of deformation. Then change of temperature in internal layers is determined from the equation:

$$\Delta T_i = \frac{\Delta Q_i + \Delta L_{DEFi}}{C_v \cdot m_i} \ . \qquad (13)$$

Change of the state variables of the working medium originally containing in rated layers, leads to deformation of these layers and displacement of their boundaries concerning nodes of a motionless grid (see fig. 2b):

$$\Delta x_1^{j+\frac{1}{2}} \cdot f_1^{j+\frac{1}{2}} = \frac{x_1^j f_1^j \left[ (N-1)a_1 - \sum\limits_{k=2}^{N} a_k \right]}{\sum\limits_{k=1}^{N} a_k} \ , \qquad (14)$$

$$\Delta x_i^{j+\frac{1}{2}} \cdot f_i^{j+\frac{1}{2}} = \frac{x_i^j f_i^j \left[ (N-1) a_i - \sum_{k=1}^{i-1} a_k - \sum_{k=i+1}^{N} a_k \right]}{\sum_{k=1}^{N} a_k} \tag{15}$$

Where $a_i = m_i^{j+\frac{1}{2}} \cdot R_г \cdot T_i^{j+\frac{1}{2}}$.

The total deformation of each internal layer caused by displacement of the right and left conditional boundaries, leads to the change of pressure in volume of the cell in the end of the first stage of a rated step:

$$p^{j+\frac{1}{2}} = \frac{m_i^{j+\frac{1}{2}} R_г T_i^{j+\frac{1}{2}}}{x_i^{j+\frac{1}{2}} \cdot f_i^{j+\frac{1}{2}}}. \tag{16}$$

Thus, in the end of the first stage of rated step in every rated element restricted by the grid nodes generally two various temperature layers (zones I and II, fig. 2b) can be contained.

To the beginning of the second rated step the distribution of state variables in cell volume is determined by averaging within every layer:

$$\begin{aligned}
&m_1^{j+1} = m_1^{j+\frac{1}{2}} - \Delta m_1^{j+\frac{1}{2}}; && T_1^{j+1} = T_1^{j+\frac{1}{2}}, \\
&m_i^{j+1} = \Delta m_{i-1}^{j+\frac{1}{2}} + m_i^{j+\frac{1}{2}}; && T_i^{j+1} = \frac{\Delta m_{i-1}^{j+\frac{1}{2}} \cdot T_{i-1}^{j+\frac{1}{2}} + m_i^{j+\frac{1}{2}} \cdot T_i^{j+\frac{1}{2}}}{m_i^{j+1}}, \\
&m_N^{j+1} = \Delta m_{N-1}^{j+\frac{1}{2}} + m_N^{j+\frac{1}{2}}; && T_N^{j+1} = \frac{\Delta m_{N-1}^{j+\frac{1}{2}} \cdot T_{N-1}^{j+\frac{1}{2}} + m_N^{j+\frac{1}{2}} \cdot T_N^{j+\frac{1}{2}}}{m_N^{j+1}},
\end{aligned} \tag{17}$$

where $\Delta m_i^{j+\frac{1}{2}}$ - mass of gas that is in the volume $\Delta x_i^{j+\frac{1}{2}} \cdot f_i$,

$m_i^{j+\frac{1}{2}}$ - mass of gas which has remained in the volume $x_i^{j+\frac{1}{2}} \cdot f_i$.

The working medium consumption in gas-distributing windows of the CPE on the stable operating mode is determined under formulas:

$$\begin{aligned}
G_{IHP} &= Z \cdot \frac{n}{60} \sum_{0}^{\varphi_{IHP}/\Delta\varphi} \Delta m_{C\,\mathrm{int}\,i}, \\
G_{OHP} &= Z \cdot \frac{n}{60} \sum_{0}^{\varphi_{ohp}/\Delta\varphi} \Delta m_{Couti},
\end{aligned} \tag{18}$$

where $\Delta m_{Cinti}$ and $\Delta m_{Couti}$ – mass which has arrived into the cell and mass, got out from it for the rated step;

$Z$-quantity of cells of the CPE rotor;

$n$ -rotational speed of the CPE rotor;

$\varphi_{IHP}$ and $\varphi_{OHP}$ - angles of rotation of the rotor corresponding to the connection of one cell with input high pressure window and outlet high pressure window.

So simulation of working cycle of the CPE is reduced to consecutive calculation of state variables simultaneously in two contiguous cells connected by the transfer channel of the stator. Thus, the calculation is conducted in a direction from a displacement line to scavenge line. The calculation is carried out up to coincidence of state variables of a current rated cycle with corresponding values of parameters on the previous working cycle.

**Rated parameters of work of the engine with SSDC - CPE**

The results of simulation of regimes of joint work of supercharging units and cooling with the piston part of the engine on speed and loading characteristics of the combined engine (CICE) are presented on fig.3.

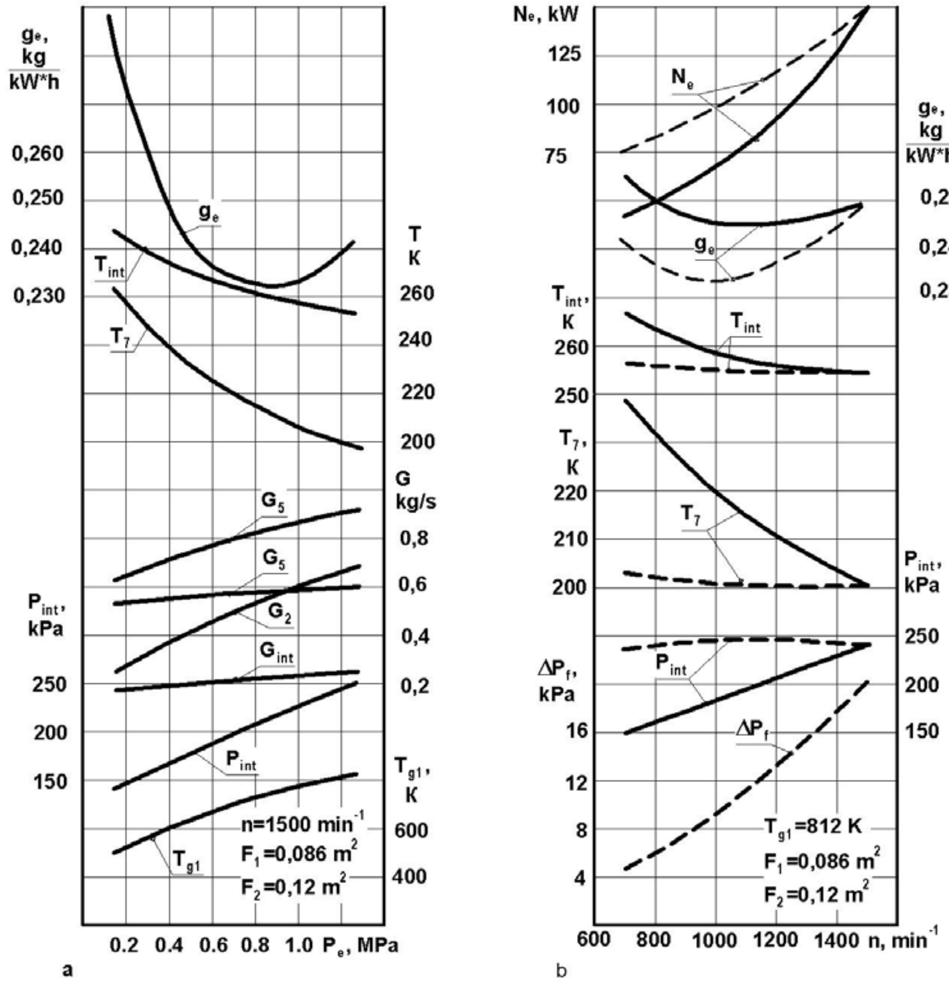


**Figure 3: Load (a) and speed (b) performance of the engine 6ЧН 12/14 with SSDC – CPE:**
-with regulating of head pressure $\Delta P_f$ of the CPE2 circulating ventilating fan;
without regulating.

As it is seen from fig.3, the consumption of air $G_5$ in the contour of high pressure of the CPE2 considerably exceeds the consumption of air $G_2$ discharged under pressure by the CPE1 in spite of the fact that only a part of air, discharged under pressure by the CPE1 is consumed on the refrigeration cycle realization. "Self-multiplication" of consumption in the refrigeration cycle contour is explained by simultaneous work of the CPE2 in a regime of the compressor refunding considerable flow $G_6$ of compressed air in the contour of the gas-expansion machine.

At decrease in a rotational speed of the crankshaft of the engine equipped by the SSDC with non-controllable adjustment, certain falling of supercharging $P_{int}$ and raise $T_{int}$ occurs. The mechanism of such change of parameters is caused by excessive cross-over of air discharged under pressure the CPE1 in the contour of the CPE 2 in view of re-regularity of the last on regimes of low rotational speeds of the crankshaft. Thus decrease in the general pressure in the SSDC, in its turn, causes falling of expansion degree of air in the gas-expansion machine and increase of the coolant temperature. Correction of the CPE2 consumption characteristic, as it has been shown above, is easily attained by differential pressure regulating between windows of high pressure by means of suitable alteration of rotational speed of the ventilating fan. The potential of such regulating is shown on fig.3b.

Adaptability of supercharging system with deep cooling (SSDC) of supercharging air at operation of the combined engine on the load characteristic is manifested in the head pressure reinforcement and cooling ability of the system on regimes of the maximum loads (fig. 3 a), where raise of density of the air charge and decrease in its temperature is most expedient. At ambient temperature 20ºC and cooling of supercharging air in the refrigerator of the first stage to 40 ºC on regime Pe = 1.5 MPa, temperature decrease of supercharging air in the refrigerator of deep cooling makes 58ºC, and on regime Pe = 0.2 MPa the supercharging air temperature decreases on 33ºC.

Ability of the CPE supercharging system to provide highly effective supercharging and cooling of supercharging air below ambient temperature without attraction of additional mechanical energy on refrigeration cycle realization stimulates possibility of substantial increase of traction and economic characteristics of engines, especially at their operation in the conditions of hot climate. In comparison with the base engine the SSDC - CPE use allows to lower the specific effective fuel consumption on work design conditions on 10% and to raise the effective power on 36%. In addition, the analyzed engine has higher power indexes on partial speed operating modes due to high level of supercharging and low temperature of supercharging air even at a low rotational speed of the crankshaft. For example, on a regime of n=800 $min^{-1}$ power of the offered engine exceeds power of base one on 65%.

Impressing characteristics of the refrigeration cycle of supercharging system with deep cooling of supercharging air admit possibility of parallel use of the engine supercharging units in the capacity of refrigerating machinery of the automobile or railway refrigerator for carriage of perishable cargoes.

## Conclusions

1. Application of the supercharging systems of the cascade exchange of pressure with deep cooling opens the prospect of considerable improving of traction and economic parameters of Diesel engines working in difficult climatic and operational conditions.

2. The presented imitating model enables to determine the parameters of compound units of the combined engine with the SSDC - CPE on the off-design conditions with sufficient accuracy for practical purposes.

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