POWER, TORQUE AND FUEL CONSUMPTION CHARACTERISTICS FOR A DIESEL ENGINE EQUIPPED WITH AN AXIAL COMPRESSOR

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Abstract: *Article highlights the performance of a supercharged engine with axial compressor by determining power, torque and fuel consumption characteristics. It has developed a mathematical model in Mathcad for determining the dynamic performance. The study presents the influence of parameters that vary constantly. In this study is analyzed and compared by graphical, the parameters that increase engine's power. The characteristics of the power, torque and fuel consumption are determined for the same engine: a simple aspirated, supercharged by a turbocharger supercharged with a biflow axial compressor. At the end of the article it's made a comparison on the dynamic performance of a compression ignition engine.*

Keywords: *power, torque, dynamic performance, biflow, axial compressor*

1. Biflow axial compressor working

The axial compressor presented in Fig. 1 eliminates mostly disadvantages which include: high difficulty in dynamic balancing process, attrition issues and lubrication problems due to very high speeds.

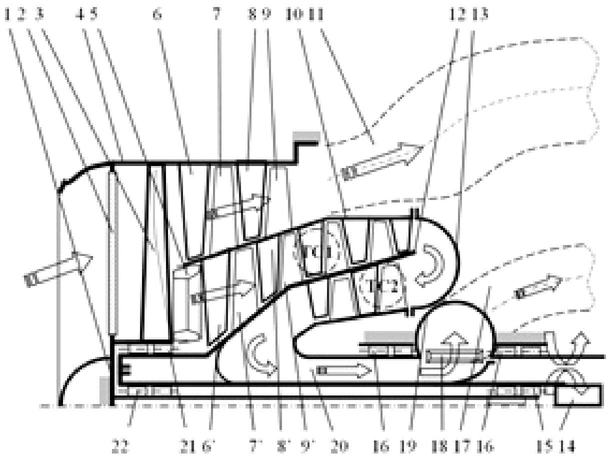

Figure 1: *Axial section through a biflow axial compressor*

The main advantage of this aggregate is that it would reduce the disadvantage of overeating at low speed, because the speed of the electric is independent of the internal combustion engine. Changing the speed at DC motors is very easy to complete.

Axial compressor compresses the air flow with special construction blades arranged on a rotor, after which the flow proceeds to be compressed axially once again another row of blades arranged on a stator. This compression produced by the rotor and stator blades is an axial compressor stage compression.

1.1 The axial compressor's role in the cooling system of the compression ignition engine

The air for cooling the engine gets in the first stage from axial compressor through guiding blades 6, Fig. 1 and then reach in the mobile blades 7. After compressing the air in the first stage this will enter in a new stage compressor made of stator blades 8 and the mobile blade 9 placed on the rotor 10. According to [Mihai, 2011], the compressed air reaches in the engine cooling system. In conventional cooling systems where fans are used, it ensures a higher rate due to the two rotors 12 and 13 of axial compression.

1.2 Ensuring the role of supercharging

The turbocharger system of the internal combustion engine presented in Fig. 1 consists of rotors 12 and 13 driven to the contrary. The air will enter in the first stage compressor supercharging through diffuser 5 and is directed in the rotor's blades 6' rotor blades 7'. In the double rotor stage, the parameters will be far superior as against classic stage axial compressor due to duplication of peripheral speed from triangle speed. The compressed air from the first stage will enter in a new stage of compression, made of the rotor's blades 8' and second rotor's blades 9'. The compression process continues in the same principle in the rest of the axial double rotor steps of the compressor.

It should be noted that there is a return of nearly 180^o of the flow path of air from the first section TC1 compression to the second TC2. Structurally, there is a pronounced reduction of the output section to the input from both TC1 and TC2 compression sections.

The rotor 12 may be driven by a motor shaft 14 of the electric D.C. which can change the speed. The outlet 19 provides the necessary air in the boost process. Compressed air flow, indicated by 20, which has passed through several stages of compression, will go towards sewerage 17 through exit 18.

2. Determination of global output parameters of an axial compressor

The axial compressor is part of a blades compressor, and where the airflow enters in an axial direction parallel to the rotation axis and the output is also the axial direction. To determine the overall calculation of the axial compressor we will use the section of the stage compression.

The triangles speed [Pimsner, 1988], [Pimsner, 1986] in sections I and II are shown in Figure 2, where: *C* – absolute speed, *U* - is the tangential speed of the average fiber and *W* – relative speed. The air fluid leaves the rotor therefore the air that will enter in the stator and on exit is the absolute speed C_3 at an angle α_3.

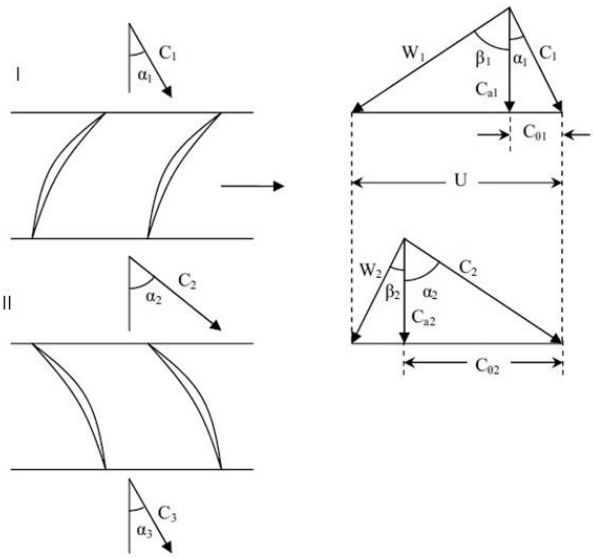

Figure 2: *Speed triangles for one stage [Pimsner, 1988]*

Usually, absolute speed out of the stator is equal to speed which entry into stator $C_1 = C_3$ and equal with the angles $\alpha_3 = \alpha_1$.

By applying speed triangles gear it can be seen more easily changing angles and speeds inside a compression stage. The compressor will have input dates presented in Table 1.

Table 1. Compressors activation parameters

Parameter	U.M.	Value
Engine power P	*[kW]*	*78*
Pressure ratio π_c		*2.5*
Compressor efficiency η_c	*[%]*	*0.88*
Fluid pressure input p_1	*[Pa]*	$0.93392 \cdot 10^5$
Enthalpy of the fluid inlet i_1	*[kJ/kg]*	*288.3*
Fluid temperature input T_1	*[K]*	*288.15*
Parameter that characterizing the geometry of stage I $d = \frac{D_b}{D_v} = 0,4 \div 0,6$	-	*0.4*
Tangential speed u_{1v}	*[m/s]*	*306*
The component of the absolute speed c_{1a}	*[m/s]*	*107.1*
Constant of fluid inlet R	*[J/kg K]*	*287.16*
Air adiabatic coefficient		*1.4*
Specific heat c_p	*[kJ/kg K]*	*1*
Speed coefficient λ	-	*0.3455*

The compressor is considered mono-rotor and the diameter at the cap D_v = const. Thus it can be calculated mechanical work to air compress:

$$l_c = i_1 \frac{\pi_c^{\frac{x-1}{x}} - 1}{\eta_c} \left[kJ/kg \right]. \qquad (1)$$

The compressor will absorb the air mass flow and will also ensure the moment coefficient condition imposed at the beginning of the calculation:

$$M_a = \frac{P_c}{l_c} \left[kg/s \right]. \qquad (2)$$

At entering the compressor will calculate the absolute velocity component:

$$u_{1m} = \frac{1}{2} u_{1v} (1 + d_1) \left[m/s \right]. \qquad (3)$$

It will determine the theoretical surface area input to the compressor taking into account the coefficient of speed:

$$A_{t1} = \frac{M_a \sqrt{T_1}}{0.04\, p_1\, 0.3455} \left[m^2 \right]. \qquad (4)$$

It will calculate the diameter at the cap in the input section:

$$D_{1v} = \sqrt{\frac{4 \cdot A_{t1}}{\pi \left(1 - d_1^2\right)}} \left[m \right]. \qquad (5)$$

Gear triangle shows that the axial direction of the absolute velocity component is preserved $c_{1a} = c_{2a}$ and will determine the cross-sectional area of the compressor exit. During compression of the working fluid both temperature and pressure will suffer some increase [Pimsner, 1986].

$$T_2 = \frac{1}{c_p} \left(i_1 + l_c \right) \left[K \right] \qquad (6)$$

$$p_2 = p_1 \pi_c \left[Pa \right]. \qquad (7)$$

It can calculate the surface area of the compressor output:

$$A_{t2} = \frac{M_a \sqrt{T_2}}{0,04\, p_2\, 0.39} \left[m^2 \right]. \qquad (8)$$

The parameters that characterizing geometry of the stage, out of the axial compressor, are presented below. According to [Pimsner, 1988] it's recommended a nominal diameter $\bar{d}_n = 0,8 \ldots 0,9$ from the input area. The mean diameter for the input compressor:

$$\bar{d}_2 = \sqrt{1 - \frac{4 A_{t2}}{\pi D_{1v}}} \; [m]. \qquad (9)$$

When the absolute velocity component of the working fluid leaving the last stage of compression it can be determinate with:

$$u_{2m} = \frac{1}{2} u_{1v} \left(1 + \bar{d}_2 \right) \left[m/s \right]. \qquad (10)$$

Also with d_2 we can calculate the diameter at the bottom at the last stage of compression:

$$D_{2b} = D_{1v} \bar{d}_2 \left[m \right]. \qquad (11)$$

3. Study of the excess air coefficient effect of the compression ignition engine parameters

In terms of thermodynamic combustion process is analyzed global, meaning that not studying the mechanism of conducting burning, called combustion kinetics, which is an extremely complex chemical phenomenon and not studying any intermediate products of combustion. Air-change coefficient is defined as the following ratio $\alpha = L_{real}/L_{min}$ and is generally determined using the indirect route flue gas composition analysis carried out by gas analyzers. The amount of oxygen required for the combustion process L_{min} is determined based on the stoichiometric oxidation reaction of the fuel elements.

It was considered $\alpha = 1.7$, value suggested by the specialty literature for compression ignition engine and an average ratio compression. The air-change coefficient has an influence on the efficiency of the combustion process.

3.1 The variation efficiency of the combustion process

Changing the air-change coefficient exerts an important influence on engine performance supercharged, influence that is highlighted by the curves from Figure 3, [Negurescu, 2009].

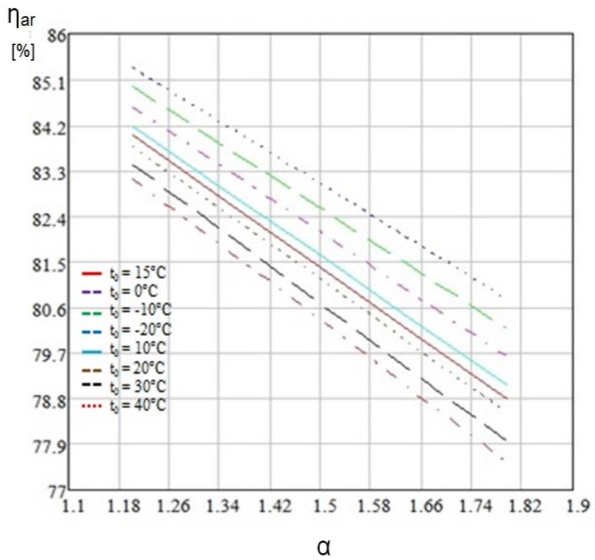

Figure 3: *The efficiency of the combustion process at different ambient temperatures*

Ambient temperature of *– 25°C* has a beneficial effect on efficiency of the combustion process because air density increases with decreasing temperature [Radcenco, 1977], [Pilusa 2012].

It can be seen that the density of air is higher at low temperature because it varies from *85.5%* for the excess air coefficient $\alpha = 1.2$, and reaches about *81.0%* when $\alpha = 1.7$. The efficiency of the combustion process suffer a significant change at $+30^oC$ to $+40^oC$ where is a decrease of approximately *6.15%*.

Increasing the α from *1.2* to *1.8*, determine increases the irreversibility losses due to burning process – Fig. 3, which translates into a decrease efficiency of the combustion process to approximately *4.07%* Fig. 3 and can be calculated with Eq. (12).

$$\eta_{ar} = \eta_1 \left[1 - \frac{T_0 \left[C_1 \left(\frac{1+\alpha N_1}{\alpha N_1} + \gamma_r \right) \right]}{\eta_{ar} \left[\left(1 - \frac{\gamma_r}{\frac{\gamma_r - 1}{\varphi_a}} \right) \right]} \right] \cdot \frac{\ln \left(\lambda^{\frac{1}{k_v - 1}} \rho^{\frac{k_p}{k_p - 1}} \delta^{\frac{k_u - n_u}{k_u - 1}} \right)}{\frac{1}{\alpha N_1} \frac{H_i}{R \cdot T_0} \frac{1}{\left[\tau_R + (1-\tau_R) \frac{T_S}{T_0} \right]}} \cdot 100$$

$$C_1 = \frac{1-\Psi_a}{T_{sf}} \frac{\varepsilon}{\varepsilon - 1} \frac{1-g_r}{\varphi_a}$$

$$C_2 = \frac{\varepsilon (1-\Psi_a)}{\varepsilon - 1}. \qquad (12)$$

3.2 The study on modification of exergy losses

The increasing of excess air coefficient determines a significant reduction of exergy losses with exhaust gas, because the efficiency of combustion process decreases,

As can be seen in the graphs of Figure 4 irreversibility losses of combustion have a linear variation coefficient of excess air – Eq. (13), [Negurescu, 2009], [Radcenco, 1977].

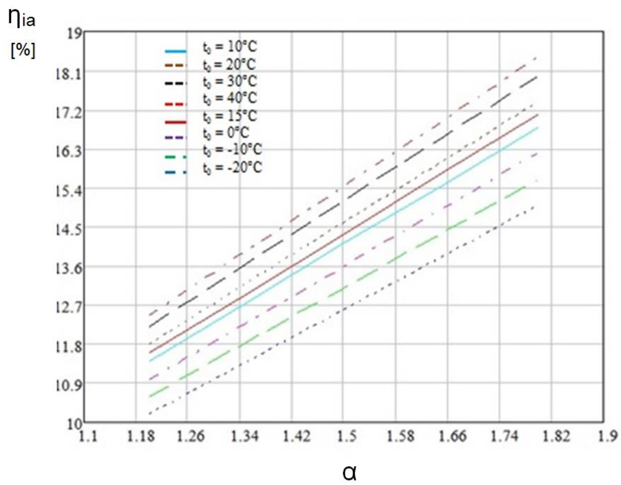

Figure 4: *The influence of the excess air coefficient of the combustion irreversibility losses due to* $t_0 = -20^0 C \div +40^0 C$

Changing the temperature of the ambient to *+ 30 °C ÷ + 40 °C* cause the variation losses due irreversible $\pi_{i_{ar}}$ of the combustion process where there is an increase of about *12%* to *18.5%*.

$$\pi_{i_{ar}}=1-\left[1-\frac{T_0\left[\frac{\eta_v}{T_{sf}}\left(\frac{1+\alpha N_1}{\alpha N_1}+\gamma_r\right)\right]}{\eta_{ar}\left[\frac{\varepsilon(1-\Psi_a)}{\varepsilon-1}\left(1-\frac{\gamma_r}{\frac{\gamma_r-1}{g_r}}\right)\right]}\right] \quad (13)$$

$$\cdot\frac{\ln\left(\lambda^{\frac{1}{k_v-1}}\rho^{\frac{k_p}{k_p-1}}\delta^{\frac{k_u-n}{k_u-1}}\right)}{\frac{1}{\alpha N_1}\frac{H_i}{R\cdot T_0}\frac{1}{\left[\tau_R+(1-\tau_R)\frac{T_S}{T_0}\right]}}\cdot 100\ [\%].$$

When the air density increases t_0 = *-20°C* for *α = 1.7* there is an improvement of *19%*.

3.3 The study on variation of thermal loading coefficient

Decrease the coefficient of thermal load Fig. 5 is faster than the increasing of indicated efficiency that fluctuate from *34.4%* to *48.59%* at *15°C* ambient temperature [Negurescu, 2009], [Radcenco, 1977].

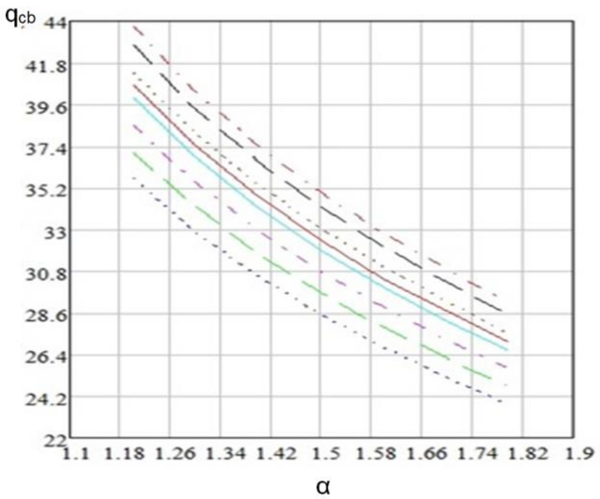

Figure 5: *The influence of the excess air coefficient on thermal load coefficient*

The increasing of *α* has influence on temperatures in the main processes operating cycle and determine reducing the temperature of the processes. Burning with high dosages exert adverse influences in the thermal load volume in the cylinder because as excess air coefficient decreases there is an increase in dosage - parameter characterizing the combustion and increasing losses due to irreversibility of the combustion process [Howell, 2013], [Meca, 2007]..

The increase of *α* leads to a reduction in the average temperature of the combustion temperature which characterizes the heat regime of the engine, T_m = *1756 → 1192 [K]*. The excess air ratio does not significantly influence the economy, but has a direct influence on engine's power. It is considered as relevant when the parameter *α* increases by a percentage, the power drops by *3.75 %*.

$$T_m=\frac{T_0\,\eta_{ar}\,q_{cb}(p_s)}{\frac{p_2}{p_0}\cdot\frac{1}{\left[T_0^2(1-\tau_R)\left[1+\frac{\left(\frac{p_s}{p_1}\right)^{\frac{k_a-1}{k_a}}-1}{\eta_s}\right]\right]}\eta_v}\cdot \quad (14)$$

$$\cdot\frac{1}{\left(\frac{\alpha N_1+1}{\alpha N_1}+\gamma_r\right)\ln\left(\lambda^{\frac{1}{k_v-1}}\rho^{\frac{k_p}{k_p-1}}\delta^{\frac{k_u-n_u}{k_u-1}}\right)}[K].$$

Average thermodynamic temperature T_m of combustion process shows a sharp decline which justifies increasing losses caused by combustion irreversibility – Eq. (14) [Negurescu, 2009],[Vianna 2005].

The influence on temperatures that define the work of the turbine is practically negligible, which can be seen in Figure 6.

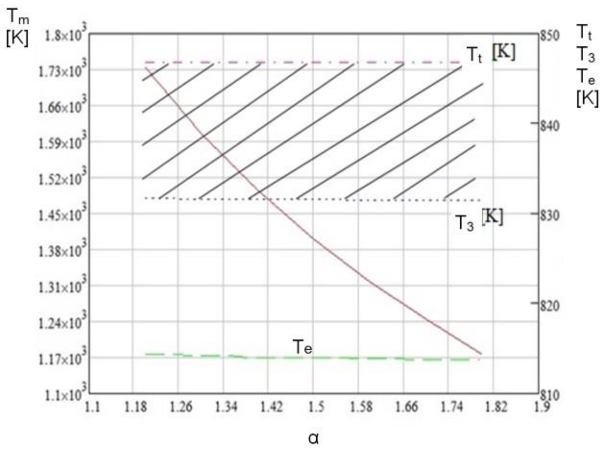

Figure 6: *The influence of excess air coefficient on temperatures characteristic of the operating cycle*

$$T_t = \left(\frac{\beta - 1}{\beta + \frac{1}{\alpha N_1}} \frac{k_a}{k_e} \frac{k_e - 1}{k_a - 1} \right) T_{am} + \frac{1 + \frac{1}{\alpha N_1}}{\beta + \frac{1}{\alpha N_1}} T_e \ [K]. \tag{15}$$

The significance of the Eq. (15) are:
T_t - temperature upstream of the turbine based on the coefficient scanning,
T_{am} - the gas temperature at the turbine end of the isentropic expansion,
T_e - exhaust gas temperature.

The temperature upstream of the turbine represents the upper limit of operation of the turbine.

4. Determination of specific power, torque and fuel consumption

This study aims to highlight the thermodynamic aspects and performance of a supercharged engine with a dual axial flow compressor, depending on certain working parameters. It is considered *"characteristic of an internal combustion engine"* as a graphical representation of the variation in size of the motor or performance indicators (such as power, torque, specific fuel consumption, etc.), depending on the mode parameters (speed, load, etc.) [Andreescu, 2010]. Next we will determine the adaptability and elasticity coefficient – equation 4.1 and 4.2.

The adaptability of the engine it's the ability to defeat higher resistance through its own possibilities, increasing torque to lower revolutions per minute due to increased external resistance [Andreescu, 2010], [Challen, 1999].

$$c_a = \frac{M_{e\max}}{M_p} > 1. \tag{16}$$

The elasticity of the engine it's the ability to achieve through its coverage of revolutions per minute a constant running, a wider range of speeds without having to change gear.

$$c_e = \frac{n_M}{n_P} < 1. \tag{17}$$

To determine the characteristics of power, torque and fuel consumption we can use polynomial equations.

$$P_e = P_{e\max} \left[\alpha' \frac{n}{n_p} + \beta' \left(\frac{n}{n_p} \right)^2 - \gamma' \left(\frac{n}{n_p} \right)^3 \right] \tag{18}$$

$$M_e = M_p \left[\alpha' \beta' \left(\frac{n}{n_p} \right) - \gamma' \left(\frac{n}{n_p} \right)^2 \right], \tag{19}$$

where α', β', γ' are dimensionless coefficients and determine the form of solving the system:

$$\begin{cases} \alpha' + \beta' - \gamma' = 1 \\ \alpha' + c_e \beta' - c_e^2 \gamma' = c_a . \\ \alpha' + 2\beta' - 3\gamma' = 0 \end{cases} \tag{20}$$

The solutions of the system are:

$$\alpha' = \frac{2c_e^2 - 3c_e + c_a}{(1 - c_e)^2}, \ \beta' = \frac{3 - 2c_a - c_e^2}{(1 - c_e)^2} \\ \gamma' = \frac{2 - (c_e + c_a)}{(1 - c_e)^2}. \tag{21}$$

Axial-flow compressor eliminates much of the disadvantages present in existing systems of overeating. Changing the speed on electric motors is currently done easily depending on the energy requirements of the internal combustion engine. The global determination shows that the pressure axial compressor reaches about $p_c = 2.3\ [10^5 Pa]$ a higher value of p_s considered in the study. The coefficient of admission has an increased to about *0.70%* and determine the variation of the temperature of intake to the cylinder. A positive influence can be found on specific mechanical work which has an increase of about *7%*, which it's reflected on the indicated average pressure. The most important variation is found in the mean effective pressure with an increase of *13%*, parameter that directly affects the effective power of the engine.

Figure 7 shows torque and power characteristics for compression ignition engine with turbocharger, with a double biflux axial compressor and a normal engine.

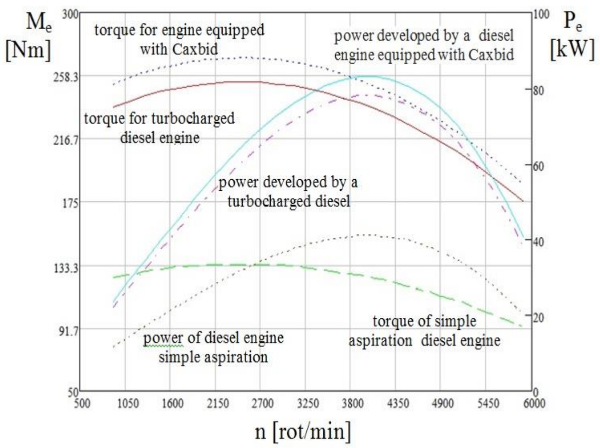

Figure 7: *The characteristic of power and torque for overeating compression ignition engine*

It can be seen an increase of power with *6%*, reaching at value P_e*=83[kW]* and torque has an improvement of *5.6[%]* at the number of rotations for developing the maximum torque.

Specific fuel consumption characteristic can be determined by [Andreescu, 2010]:

$$c_e = c_{eP}\left[1,2 - \left(\frac{n}{n_P}\right) + 0,8\left(\frac{n}{n_P}\right)^2\right] \qquad (22)$$

In Fig. 8 is shown the specific fuel consumption of the engine:

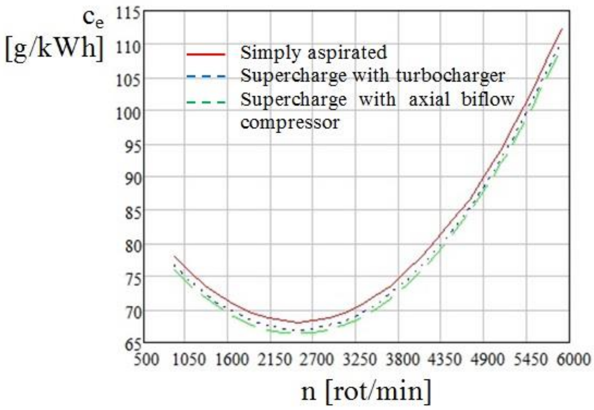

Figure 8: *Fuel consumption for compression-ignition engine supercharged with turbocharger and axial biflow compressor, and simply aspirated*

The specific fuel consumption of the engine is determined by the reduction of losses, primarily by reducing the exergy loss with heat of turbine exhaust gases - in a reduced extent of exergy losses through the cylinder wall with the heat discharged into the combustion.

5. Conclusions

The excess air coefficient has an influence on the efficiency of the combustion process η_{ar} because if it`s modified with one unit, this suffers an improvement of about *1.3%*, at t_0 = *15ºC*. If the engine works at high temperatures *30ºC÷40ºC* the efficiency of combustion process is canceled and reach at *83.99 %*, a decrease of about 1.5 %.

The increase of α leads to a reduction in the average temperature of the combustion temperature which characterizes the heat regime of the engine, $T_m = 1756 \rightarrow 1192[K]$.

The excess air coefficient does not significantly influence the economy, but has a direct influence on engine power, where the modification of α with a unit the power is increased with *3.75 %*.

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