THE BOOST PRESSURE'S INFLUENCE OF AN AXIAL BIFLOW COMPRESSOR ON THE ENGINE'S PERFORMANCE

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Abstract: *This study sought to highlight the thermodynamic aspects on the processes of a compression ignition engine equipped with an axial compressor under certain conditions and with certain parameters that varies. It has focused on determining the performance and losses caused by the irreversibility of the processes. This paper presents the contributions for supercharging system, design operating principle and also continue with the descriptions of the two roles that has axial compressor. The study continues with the presentation of variation of the parameters characteristic of operating cycle and dimensional measurements of axial compressor.*

Keywords: *supercharging, axial biflow compressor, boost pressure*

1. The axial compressor's role in the cooling system of the internal combustion engine

In Fig. 1 is represented the supercharging system with an axial compressor [Mihai, 2011].

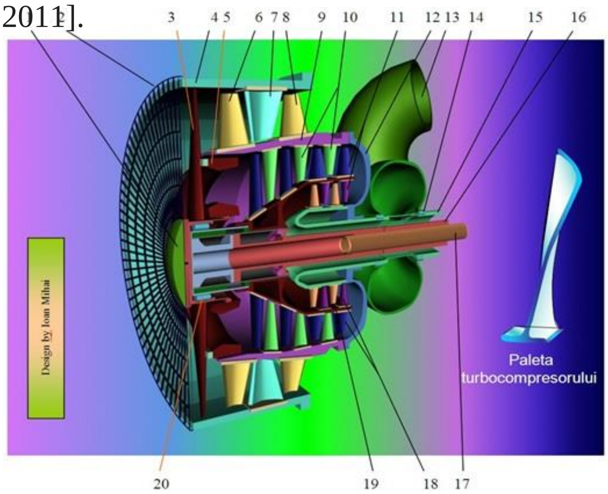

Figure 1: *Biflow axial compressor* [Mihai, 2011]

The main disadvantages of actually system are on the one hand the high price, the high operating speeds and malfunctions at low speeds. It is necessary to devise a system that overcomes these drawbacks. Thus the new concept uses a dual drive system consists of an electric motor and a power outlet from the crankshaft. Such an aggregate would reduce overeating at low revs disadvantage because the electric motor speed is independent of the internal combustion engine. Changing the speed DC motors is very easy to complete. The solution presented has two rotors that spin in opposite directions. One of the rotors is driven electric and the other one is driven by the engine. At low speeds the electric motor running at maximum speed. With increasing the speed of compression ignition engine, it can adjust speed depending on the operating conditions. The bearing system 1 of the biflow axial compressor is supported by blades 3 and the air is filtered by sieve 2. The sieve and the filtering system are attached to the stator 4 which is integral with divergent-convergent nozzle 5. The air for cooling the engine, get in the first stage from axial compressor through mobile blades 6, 8 and then reach in the fixed blades 7. After compressing the air in the first stage, will enter in a new stage compressor 10-11, 18-19 placed on the rotors 9, 15 and stator 4. The compression process continues in the same principle in the rest of the axial double rotor steps of the compressor. Elbow 12 ensures the return airflow to 180 degrees. The exhaust manifold 13 provides compressed air to the engine. Needle bearings 14 and 20 ensure the bearing of the compressor crankshaft. The compressed air reaches in the engine cooling system. In conventional cooling systems where fans are used, in this case it ensures a higher rate due to the two stages of axial compression.

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 engine and compressor parameters are presented in the Table 1.

Table 1: Engine and compressor parameters.

Parameter	Notation	U.M.	Value
Engine's power	P	[kW]	78
Pressure ratio	π_c	-	2.5
Compressor efficiency	η_c	%	0.88
Fluid pressure input	p_1	[Pa]	$0,93 \cdot 10^5$
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	[m]	0,4
Tangential speed u_{1v}	u_{1v}	[m/s]	306
The component of the absolute speed c_{1a}	c_{1a}	[m/s]	107.1
Constant of fluid inlet	R	[J/kg K]	287.16
Air adiabatic coefficient	χ	-	1.4
Specific heat	c_p	[kJ/kg K]	1
Speed Coefficient	λ		0.3455

The air fluid that leaves the rotor is therefore the air that will enter in the stator and on exit is the absolute speed C_3 at an angle α_3. Usually absolute speed out of the stator is equal to speed entry into stator $C_1 = C_3$ and equal 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. The compressor is considered mono-rotor and the diameter at the cap D_v = const. Thus it can be calculated.

The compressor will absorb the air mass flow and will also ensure power condition imposed at the beginning of the calculation [Pimsner, 1986]:

It will determine the theoretical surface area input to the compressor taking into account the coefficient of speed [Pimsner, 1988]:

$$A_{t1} = \frac{P_c \sqrt{T_1}}{0,04 \; p_1 \; 0,3455 \; i_1 \dfrac{\pi_c^{\frac{x-1}{x}} - 1}{\eta_c}} \quad \left[m^2\right]. \quad (1)$$

where all terms are found in Table 1.

Speed 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,1988].

$$T_2 = \frac{1}{c_p}\left(i_1 + l_c\right) \; \left[K\right]. \quad (2)$$

$$p_2 = p_1 \; \pi_c \; \left[bar\right]. \quad (3)$$

It can calculate the surface area of the output compressor:

$$A_{t2} = \frac{M_a \sqrt{T_2}}{0,04 \, p_2 \; 0.39} \; \left[m^2\right], \quad (4)$$

where M_a is the airflow from axial compressor and can be calculated as ratio of power and mechanical work.

2.1 Contributions to the influence of the boost pressure on the engine's parameters

Compression ignition engines can be supercharged to increase performance. The engine power is proportional to the amount of fresh fluid introduced into the cylinder.

Reducing the stroke capacity leads to a decrease of engine's power and the solution to eliminate this shortcoming is apparent from analysis of the Eq. (7) [Negurescu, 2009], [Basshuysen2, 2004].

$$P_l = \frac{P_e}{i\,V_s} = \frac{p_e\,n}{30\,\tau}\ [kW], \tag{5}$$

where p_e is the mean effective pressure and can be calculated very easy. This will increase the pressure on the intake and the fresh charge is compressed $\pi_s = p_s/p_0$ - process called supercharging. After this process it will increase the mechanical work per cycle, which leads to increased average effective pressure. Compressing the air into the supercharger unit creates pressure and heat, pressure helps to increase power [Negurescu, 2009]. Another equation showing that the power developed by the engine is directly proportional to the mean effective pressure [Basshuyen1, 2004], [Basshuysen2, 2004]:

$$P_e = p_e\,n\,S\,\frac{1}{Z}\ [kW]. \tag{6}$$

Increasing the boost pressure increase also the temperature at the end of the isentropic compression.

2.1.1 Study on flue's temperature from supercharging system

Eq. (7) shows that this temperature is directly proportional to the boost pressure and influence directly the temperature from the cooling system – intercooler [Radcenco, 1977].

The air temperature at the outlet of the cooling system has direct effect on the volume coefficient of thermal overload, over temperature and mixture of air and flue gas.

$$T_S = T_0 \cdot \left[1 + \frac{\left(\dfrac{p_s}{p_{in}}\right)^{\frac{k_a - 1}{k_a}} - 1}{\eta_S} \right] [K], \tag{7}$$

where: T_0 - ambient temperature, P_s – boost pressure, p_{in} – calculated pressure according to losses, η_S – blower efficiency. The boost temperature variation depending on the boost pressure is presented in Figure 3.

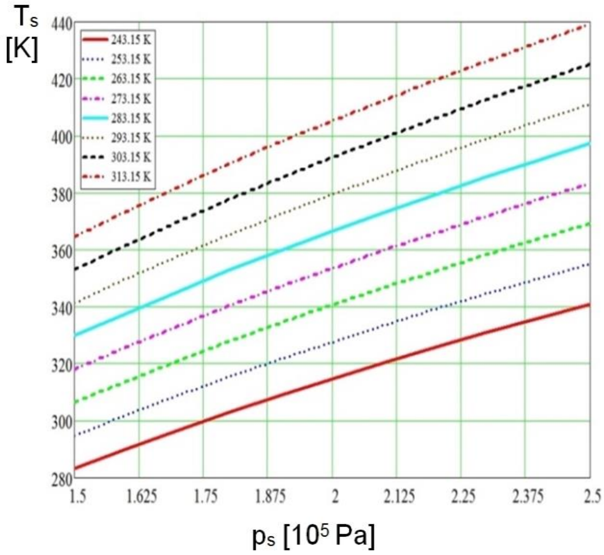

Figure 3: *Boost temperature variation depending on the boost pressure*

2.1.2 The coefficient of thermal loading variation

Increasing the boost pressure is achieved with the increased thermal load cylinder. The economy of the engine is virtually immune to changes in boost pressure considered in this study $p_s = 1,3\ldots2,5\cdot10^5 Pa$. Instead boost pressure has significant influence on the thermal load of the cylinder, [Podevin, 2000], [Radcenco,1977].

$$q_{cb}(p_s) = \left[(1-\Psi_a)\frac{\varepsilon}{\varepsilon - 1} \cdot \left(1 - \frac{\dfrac{\gamma_r}{\gamma_r - 1}}{\varphi_a} \right) \right] \cdot \left(\frac{1}{\alpha N_1} \cdot \frac{H_1}{R \cdot T_0} \cdot \frac{(1-\Psi_2)\,p_s}{p_0} \cdot \frac{T_0}{T_S} \right), \tag{8}$$

where $\Psi_a = 0.02$ - coefficient of relative losses in the intake process, ε – compression ratio, $\gamma_r = 0.035$ – coefficient gases, $\varphi_a = 1.04$ – the coefficient of deviation from the adiabatic process, N_1 – minimum combustion air, R – specific constant of the mixture, H_1 – lower calorific value of fuel. In figure 4 we can observe the coefficient of thermal loading variation depending by ambient temperature, [Radcenco, 1977], [Hiereth, 2007].

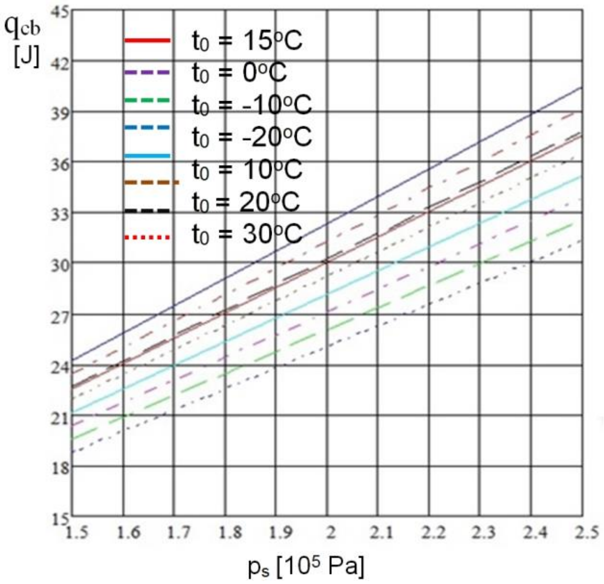

Figure 4: *The coefficient of thermal loading variation depending by ambient temperature*

2.1.3 Indicated and effective efficiency depending on the boost pressure

The thermal load of the cylinder q_{cb} growth directly influences indicated mean pressure and mean effective pressure. Also increases the indicated and effective efficiency is calculated by eq. 9 and the results are presented in Figure 5, [Radcenco, 1977].

$$\eta_i = \frac{p_i}{\left[(1-\Psi_a)\frac{\varepsilon}{\varepsilon-1}\left(1-\frac{\frac{\gamma_r}{\gamma_r-1}}{\varphi_a}\right)\right]} \cdot \frac{1}{\frac{1}{\alpha\cdot N_1}\cdot\frac{H_i}{R\cdot T_0}\cdot\frac{p_2}{p_0}\cdot\frac{T_0}{T_2}} 100[\%], \quad (9)$$

$$\eta_e = \eta_{mM}\,\eta_i\,100[\%].$$

As can be seen from the efficiency graph it have an approximate increase of about *11.3%*, *11.01%*, in terms of excess air considered at the beginning of the study $\alpha=1,7$.

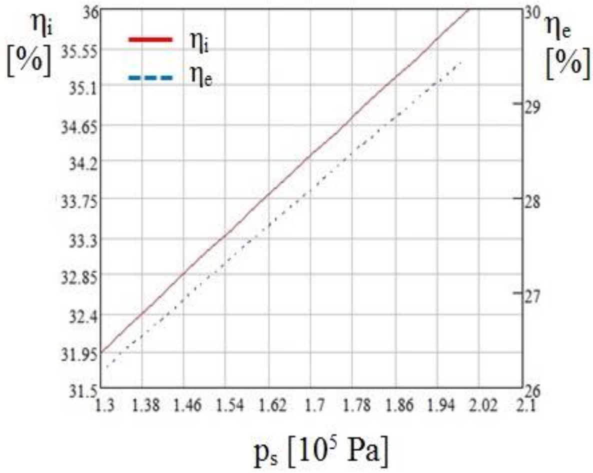

Figure 5: *Indicated and effective efficiency for diesel engine*

2.1.4 The variation of average temperature combustion

Fig. 6 represents the average combustion temperature depending by the variation of boost pressure and the ambient temperature. Basically this increase leads to increased losses due to irreversibility of the combustion process.

In conclusion it can be said that the variation of T_0 in a bid to reduce losses is more important than the supercharged pressure variation [Radcenco, 1977].

$$T_m = \frac{T_0\,\eta_{ar}\,q_{cb}(p_s)}{\frac{p_2}{p_0}\,\frac{1}{\left[T_0^2(1-\tau_R)\left[1+\frac{\left(\frac{p_s}{p_1}\right)^{\frac{k_a-1}{k_a}}-1}{\eta_s}\right]\right]}\eta_v} \cdot \frac{1}{\left(\frac{\alpha N_1+1}{\alpha N_1}+\gamma_r\right)\ln\left(\lambda^{\frac{1}{k_v-1}}\,\rho^{\frac{k_p}{k_p-1}}\,\delta^{\frac{k_u-n_u}{k_u-1}}\right)}\,[K].$$

(10)

The result of eq. (10) is represented in Fig. 6 according to ambient temperature.

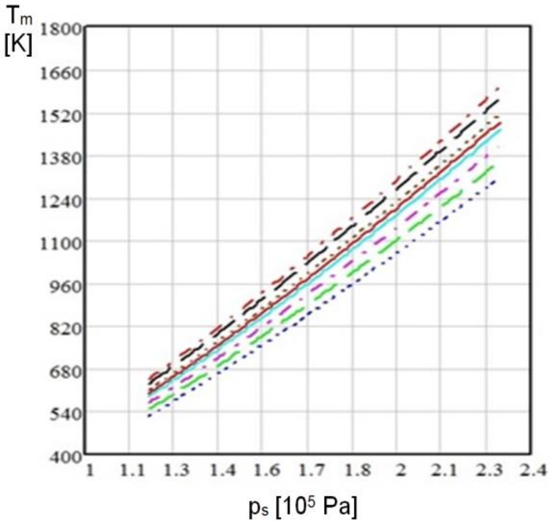

Figure 6: *Average temperature combustion variation depending p_s at ambient temperature oscillation*

3. Determination of the performance of a compression ignition engine equipped with an axial compressor

It may be observed that the increase in boost pressure by *0.1* units lead to increased engine power by approximately *7%*.

$$P_e = P_{e\max}\cdot\left[\alpha'\frac{n}{n_p}+\beta'\left(\frac{n}{n_p}\right)^2-\gamma'\left(\frac{n}{n_p}\right)^3\right] \quad (11)$$

The variation of power at full load it's represented in Figure 7, [Andreescu, 2010].

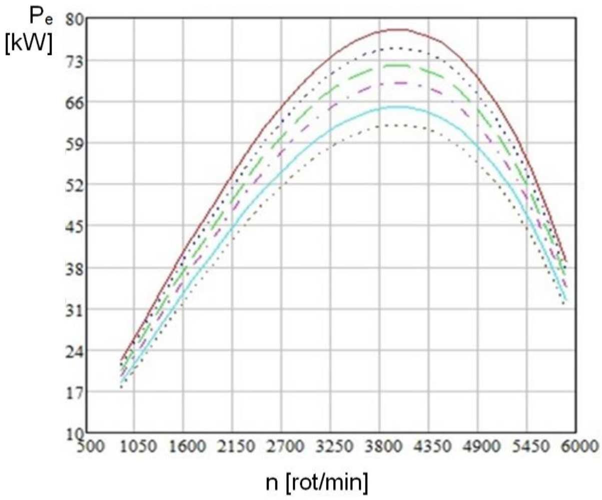

Figure 7: *The power characteristic of diesel engine*

The boost pressure directly affects the effective mean pressure, pressure that influence engine's power.

If it is considered that the boost pressure has values *p_s=1.3÷2.5[10^5Pa]* the effective power of the engine is increased of about *20%* at full load. The same influence is highlighted in torque characteristic Fig. 8. This influence is noticed in the average effective pressure which in turn is influenced by specific mechanical work, [Andreescu, 2010], [Podevin, 2000]

$$M_e = M_p\left[\alpha'\cdot\beta'\left(\frac{n}{n_p}\right)-\gamma'\left(\frac{n}{n_p}\right)^2\right] \quad (12)$$

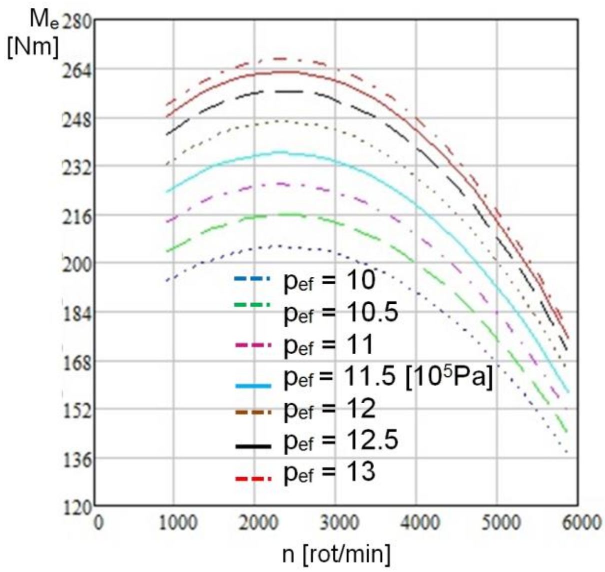

Figure 8: *The torque characteristic depending by effective mean pressure*

Effective mean pressure is calculated based on the indicated mean pressure indicated plus thermal and mechanical losses. The boost pressure directly influences the specific mechanical work through which determine the indicated mean pressure [Podevin, 2000], [Basshuysen2, 2004].

If the engine is turbocharged, the torque is obtained at a relatively low speed and maximum torque is also maintained in a wider range of speeds, [Hiereth, 2007]. If we want to evaluate or compare the efficiency of an internal combustion engine it can be use specific graphical representation of actual consumption. By analyzing the chart above that minimum is around *2.200rev/min*. In general, internal combustion engines are designed to operate with maximum efficiency around the average speeds of 2000÷3000 [rev/min].

4. Conclusions

Boost pressure is the pressure that directly affects the effective power of the engine and in generally in the cooling systems occur losses that reduce this benefits. Engine's power grows easily because the temperature of the fluid is increased in the compression process from supercharger system. This temperature directly influences the average temperature of combustion because if the boost pressure it's modified with one unit, the T_m shows an increase of about *7.1%* influence on the thermal engine.

According to [Negurescu, 2009] the high thermal conditions support normal combustion process because the fresh fluid heats more in contact with the cylinder's walls. Also the thermal loading has an increase by *5%* when the boost pressure is modified with one unit. This improvement is reflected on the effective power which has an increase of about *4.7 %*.

Increasing the boost pressure with two units determine the increasing of effective pressure with *10.6%* which leads to a decrease fuel consumption. If the engine works in economy mode, the fuel consumption decrease with approximately *11.7%*. When $p_s=1.8 - 2.5[10^5Pa]$ the fuel consumption decrease $c_e = 145.88 - 128.88[g/kWh]$.

Torque is directly proportional to the mean effective pressure, which leads to the following statement: when boost pressure changes by two units, the maximum torque increases by *7.8%*.

References

1. [Andreescu, 2010] Andreescu C., *Dinamica autovehiculelor pe roți*, Vol.1, Bucuresti, Editura Politehnica Press, 2010;

2. [Basshuyen1, 2004] Richard Van Basshuysen, R., Schäfer, F. (editors), *Internal Combustion Engine Handbook – Basics, Components, Systems and Perspective*", SAE International, eISBN: 978-0-7680-7196-2, , 2004.

3. [Basshuysen2, 2004] Richard van Basshuysen, Schafer Fred, *Internal Combustion Engine*, 2004;

4. [Mihai, 2011] Mihai, I., Olariu, E., *Cerere de brevet de invenţie nr. A/01292 din 02.12.2011*, Contract cu misiune inventivă nr. 18090/25.11.2011: [Biflow axial compressor, has air flows for external cooling and supercharging of internal combustion engines], Patent Number: RO128769-A2;

5. [Neguresсu, 2009] N. Negurescu, *Motoare cu aprindere prin scânteie, Procese*, București, Ed. Matrix Rom, 2009;

6. [Pimsner, 1986] Victor Pimsner, Corneliu Berbente, Virgil Stanciu, Valentin Luca, Dimitrie Negulescu, Victor Dobrin, Cristian Tataranu – *Procese în masini termice cu palate*, București, Ed. Tehnică, 1986;

7. [Pimsner, 1988] Victor Pimsner, *Mașini cu palate,* București, Ed. Tehnică 1988;

8. [Podevin, 2000] Podevin, P., Descombes G., Charpentier, C., *Effect of supercharging pressure on internal combustion engine performances and pollutants emissions, From Thermo-Economics to Sustainability* (G.G. Hirs editor), Part 2: ECOS 2000 Proceedings, Universiteit Twente, Nederland, pp. 703-713, 2000.

9. [Radcenco, 1977] Vs. Radcenco, *Criterii de optimizare a proceselor termice*, București, Ed. Tehnică 1977;

10. [Hiereth, 2007] Hiereth, H., Prenninger, P., *Charging the internal combustion engine*, Springer, 2007