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Charging process analysis of an opposed-piston two-stroke aircraft Diesel engine

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Abstract. This paper presents the research results on a 1D model of an opposed-piston two-stroke aircraft Diesel engine. The research aimed at creating a model of the engine in question to investigate how engine performance is affected by the compressor gear ratio. The power was constant at all the operating points. The research results are presented as graphs of power consumed by the compressor, compressor efficiency and brake specific fuel consumption. The optimal range of compressor gear ratio in terms of engine efficiency was defined from the research results.

1 Introduction

Civil aviation widely uses four-stroke piston spark ignition engines to power light aircrafts such as: planes, helicopters, gyroplanes, unmanned aircrafts or UAVs. This paper shows the model of a supercharged two-stroke compression-ignition aircraft engine. This type of engine is discussed in few works only so we have decided to investigate it more broadly in view of the fact that the two-stroke compression-ignition opposed-piston Diesel engine shows many advantages, such as a simple design, a good balance, high thermal efficiency, high specific power and lower wear rate due to friction forces compared to the four-stroke row engine [1].

Its drawback is increased fuel consumption and higher emission of toxic substances compared to the four-stroke engine. It was therefore resolved that the two-stroke Diesel engine carbon dioxide emission ought to be reduced [2].

A supercharged system is a key component of the model of the entire engine because a correctly defined process boosts the results in a correctly mapped working process. A correctly mapped forced induction is fundamental due to the changes in pressure in the combustion chamber and engine performance. Forced induction can be carried out with a turbocharger or a mechanical compressor. The simulation and the experimental research into a turbocharger for car engine applications using a 1D modelling of the transient and intransient flow is discussed in [3]. The research showed that time lag in the model of a turbocharger enables us to more efficiently predict the change in pressure and oscillations in mass air flow because of more precisely calculated mass and energy stored in the device, which is obtained by a better described propagation of pressure waves.

In two-stroke engines, apart from parameters and the design of the forced induced system, cylinder scavenging is important. This process was investigated using a simulation model and in experiments [4]. The paper [5] discusses the numerical optimization of basic operational parameters of the two-stroke aircraft Diesel engine using a patented valve that improves scavenging.

Cylinder filling efficiency in two-stroke engines is inversely proportional to scavenging efficiency. Keeping balance between these two processes is fundamental for engine performance and brake specific fuel consumption. The optimization method involving improved performance and reduced fuel consumption for a forced induced compression-ignition two-stroke aircraft engine is described in [6].

The paper [7] presents different methods for force inducing the two-stroke aircraft Diesel engine. A mechanical compressor increases brake specific fuel consumption more than a turbocharger. However, this method of forced induction has a lower mass and dimensions, which is particularly important while designing aircraft structures.

According to [8], power absorbed by the compressor changes with the required amount of air and EGR valve and results from engine mapping. Two different gear ratios between the compressor and the drive shaft for different speeds and load conditions can improve engine performance.

The resulting engine power is affected not only by forced induction but also the nature of combustion. The researchers [9] investigated combustion in the opposed-piston two-stroke Diesel engine for heat release rate and combustion time.

This work presents a model of an engine and a forced induction system for the designed 100 kW opposed-piston two-stroke Diesel engine aircraft system and gives an account of preliminary simulation. The research aimed at using the created engine model to investigate the effect of compressor gear ratio on the engine performance.

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The software used to develop the model was AVL BOOST, which is a tool dedicated for 1D modelling of piston internal combustion engines. This tool allowed us to map processes that occur in the combustion engine and parameters of the operating medium in the cylinder for the defined operating conditions and offered faster calculations than in 3D models. However, we could combine 1- and 3D models to obtain more information and more predictable simulation results [10]. Carlucci, Ficarella and Trullo [11] followed this approach to study the process of charge exchange in a two-stroke six-cylinder aircraft Diesel engine. The AVL BOOST also allows studying the correlation between the engine configuration and engine performance, *e.g.* power, torque, fuel consumption, *etc*.

2 Model of the engine

The model of the engine is created in the AVL BOOST. It was a zero-dimensional physical modelling based on the principle of conservation of mass and energy. The assumption is as follows: the 1D flow in the ducts means that the pressure, temperature and speed distribution calculated from the dynamic equations of gas correspond to average cross-sectional values in the duct. Flow losses due to phenomena that occur in the 3D space at individual points in the engine are considered as specific flow rates. This kind of model enables transient-state research that successfully reflects varied flight conditions, *e.g.* pressure, temperature. This model calculates the flows of air and fuel in the intake system and the cylinder including force inducing devices as well as combustion and the flow of exhaust gas to the environment.

The principles of the model of the engine are as follows:

1) a 1D model shows the length and diameter of engine elements for the flow only;

2) a model of the two-stroke engine;

3) identical geometries of all three cylinders;

4) twice as large surfaces of piston bottoms than in reality to reflect the nature of the operation of the opposed-piston engine;

5) geometric orientation of cylinders is neutral to their performance;

6) temperature of the component walls is constant;

7) the system for charge exchange is symmetric;

8) fuel is supplied directly to the combustion chamber forming a mixture of a given composition.

The AVL Boost model of the engine required some simplifications due to the nature of the 1D mathematical model. These are as follows:

1) the flow of medium through the intake system and exhaust passes mainly round-section ducts;

2) the components of a large volume and a small distance between the outlet and inlet sections to be modelled by means of the Plenum (constant-volume element);

3) the fuel flow depends on mass flow rates and is automatically regulated to maintain a desired mixture composition.

Our model shows the entire route of the charge flow between the air inlet and outlet. The model reflects the process of fuel injection into the cylinders and combustion that occurs there. The calculation uses the module for the two-stroke engine which consists of (Fig. 1): an air inlet, a compressor connector, a compressor, an air cooler, intake and outlet ducts, cylinders (changes in cylinder volume), and Plenum-created elements (of fixed volume). Preliminary research of 3D models of charge exchange enabled us to introduce similar-to-a-target-solution dimensions of the parts of the intake and outlet system including channels in the engine block. To obtain maximum filling, other elements of these systems were optimized.

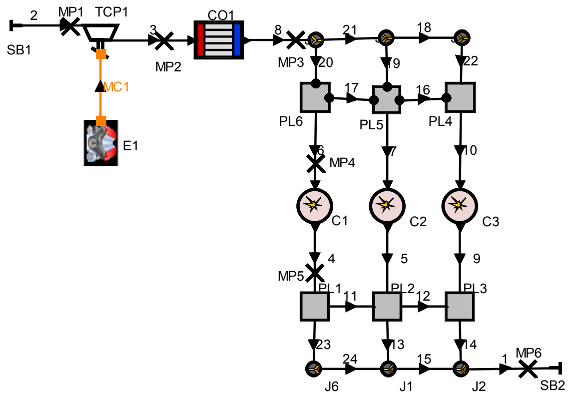

Fig. 1. Model of the engine.

The engine model is based on the mass flow in the intake and outlet ducts, and refers to the equations for the isentropic circular cross-section flow. Mapping the geometry of the intake ducts and other key elements is performed with a view to obtaining as favourable as possible conditions for engine operation. The model also is based on the assumption of mechanical radial compressors. To correctly model charge exchange, there are introduced the characteristics of window opening. The model includes the mechanical centrifugal compressor (radial), whose key function is to increase the degree of engine filling (volumetric efficiency). Supplying more air into the engine (compared to a naturally aspirated engine) contributes to obtaining higher indicated mean efficient pressure. The use of compressors in the two-stroke engine is necessary to provide for charge exchange.

The model of an air cooler is between the compressor and the Plenary element 1 placed before the cylinders. There are defined the dimensions, mass flow rate and temperatures of the air inlet, coolant and air outlet.

The compression ratio was defined by the correlation of the cylinder volume and the combustion chamber and the combustion chamber volume, which means that theoretical coefficient of compression is specified so as to ensure highly efficient scavenging.

The parameters of heat transfer into the cylinder walls were defined in line with the data from the literature and the Woschni heat transfer model, entered into the applied software (1978).

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The model of combustion is based on the Vibe model. The model of heat release is calculated from the semi-empirical formula with the given parameters. The following values of the parameters are applied in the model: combustion start -3 deg, combustion time 35 deg, shape m = 1, parameter a = 6.9. These parameters were specified with reference to the data from the literature [12].

The friction resistance was also defined in the engine by specifying the mean friction pressure.

3 Model of the supercharger system

Modelling circulation parameters of the forced induced engine requires creating and implementing mathematical models for individual elements of the system, including a centrifugal compressor. The mapping of a compressor with its real characteristics is affected much by the quality of mapping its performance under varied load conditions, which can be successful if precise calculations and relevant physical characteristics of the analyzed phenomena are included in the mathematical description.

Forced induction is also important due to pressure changes resulting from the use of aircraft engine (in line with the ISA). The compressor can increase pressure when filling the engine so the required power at the nominal altitude is maintained.

If analyzing the requirements and principles behind the model of thermodynamic process in terms of forced induction, the first aspect is that a compressor drive in this model is a mechanical connection with a crankshaft. Due to a direct connection of a compressor and a crankshaft, a mechanically forced induced engine reacts faster to considerable changes in load due to rapid changes in power equipment settings than the turbocharged engine. However, a compressor drive consumes a significant part of the engine-generated energy (up to 10%) and reduces overall circulation efficiency. Therefore, the transfer of energy to compressor drives should be included in the model.

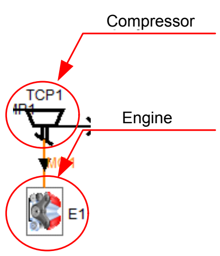

Fig. 2. Model of a compressor connected with the model of an engine.

The first stage of 1D modelling was the application of a simplified model of forced induction. Here, compression, isentropic efficiency and mechanical efficiency were defined. These values were, however, constant throughout the operating cycle, independent of the instantaneous changes in the mass flow rate. This model also disregards the impact of engine speed on the performance of a charging compressor. These values were as follows: compression π = 1.28; isentropic efficiency η_v = 0.8; mechanical efficiency η_v = 0.97. These values were as preliminary in the calculations and were specified from the literature data [13] and verified in the models of engines.

The model of a boost system uses Rotrex C30-94 compressor of a planetary transmission gear, capable of increasing engine speed several times, the required compression ratio (approx. 1.3) and the mass flow rate (0.145 kg/s) for the maximum power (100 kW) fall within the range of compressor efficiency above 65%.

To obtain the required engine power (100 kW), the gear ratio of the compressor is 1:13.9. A real compressor has a built-in reducer of a gear ratio of 1:9.5 so an additional gear belt ratio of 1:1.46 should be applied. For such a gear ratio and crankshaft speed of 4000 rpm, the rotor speed of the compressor is about 55500 rpm.

Fig. 3 shows the changes in compression and mass flow of a boosting compressor in a single operating cycle. The compression ranges from 1.26 to 1.42 and mass flow rate from 0.085 to 0.2 kg/s. These fluctuations result from wave phenomena in the intake system.

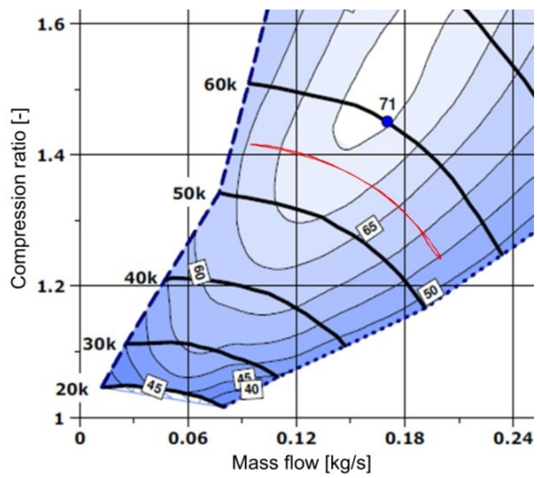

Fig. 3. Selected characteristics of the C30-94 compressor [14].

Pressure pulsations were introduced intentionally by appropriately selected dimensions of the intake system. This solution enables a dynamic forced induction and a reduced power to drive the turbocharging compressor.

When this characteristic of the compressor was applied the power to drive it increased from 4.2 kW to 6.7 kW so specific fuel consumption also rose from 203 to 207 g/kWh. This is due to the assumed in the earlier calculation high isentropic efficiency and pressure pulsation. Pressure pulsation triggers temporary changes in compressor efficiency, which reduces its mean efficiency and increases the demand for the power to drive the compressor.

4 Studying the results

This section discusses the research results on performance of our engine versus the charging compressor gear ratio.

For a fixed geometry of the intake system, calculations were conducted for different values of forced induced

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pressure changed by modifying the ratio of the gear driving the compressor. The AVL BOOST has no option to set the ratio of an additional gear between a crankshaft and a compressor. Therefore, the gear ratio of the compressor was regarded as the total gear ratio if mechanical efficiency is 98%. To obtain the required cruise power N = 73 kW, the dose of fuel was changed in each case to obtain higher AFR (Fig. 5). The calculations were made for 7 variants of the supercharger pressure (Table 1) ranging from pd_1 = 2.21 bar to pd_7 = 1.35 bar. The environmental conditions were defined by pressure p_o= 1 bar and temperature t_o = 15 C. Due to the value of ambient pressure, the forced induced pressure and compression in the subsequent cases are identical.

Table 1. Summary of the configuration of the forced induced pressure.

No.	Symbol	Total gear ratio [-]	Forced induced pressure [bar]	AFR [-]
1	pd_1	1:25	2.21	32.70
2	pd_2	1:22.2	1.92	32.28
3	pd_3	1:20	1.73	30.92
4	pd_4	1:18.2	1.59	29.54
5	pd_5	1:16.7	1.49	28.70
6	pd_6	1:15.4	1.41	26.27
7	pd_7	1:14.3	1.35	24.43

Figure 4 shows the operating points for the research variants of forced supercharger pressure. In all the cases, similar ranges of variation of supercharger pressure were recorded and we could therefore conclude that this phenomenon is correlated with the dimensions of intake ducts.

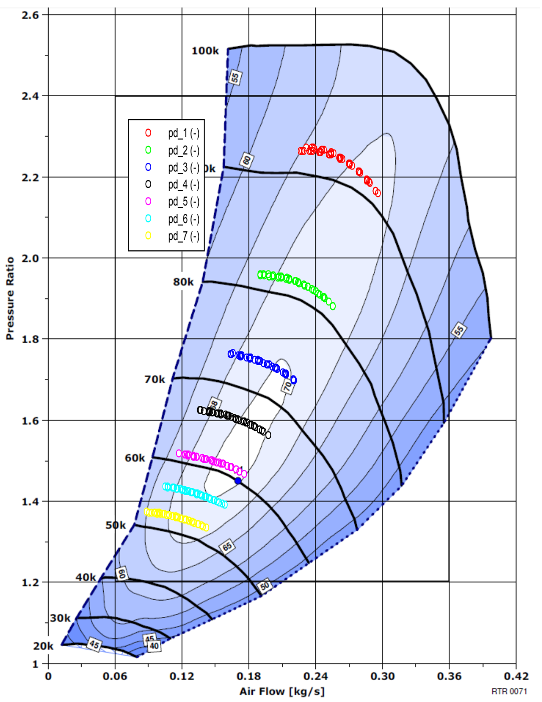

Fig. 4. Operating points for different variants of the forced induced pressure in the characteristics of the Rotrex C30-94 compressor [14].

The highest mean compressor efficiency was obtained for variants pd_3, pd_4, pd_5 (Figure 8) which were more than 69%. The lowest efficiency was obtained for the maximum forced induction (version pd_1) of about 64%. The efficiency in the last two variants, *i.e.* pd_6 and pd_7 is slightly lower and amounts to approximately 1%. Fuel mass given in mg/operating cycle (Fig. 7) increases from 71 g at 1.35 bar to 90 g at 2.21 bar, which results from the fact that the compressor needs more mechanical power.

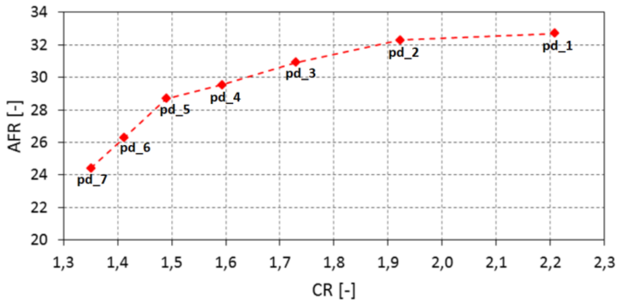

Fig. 5. AFR vs. compression ratio.

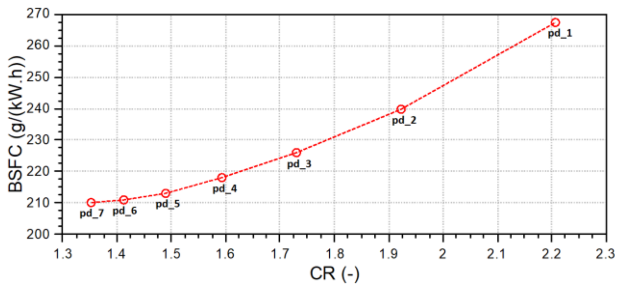

Fig. 6. Brake specific fuel consumption vs. compression ratio.

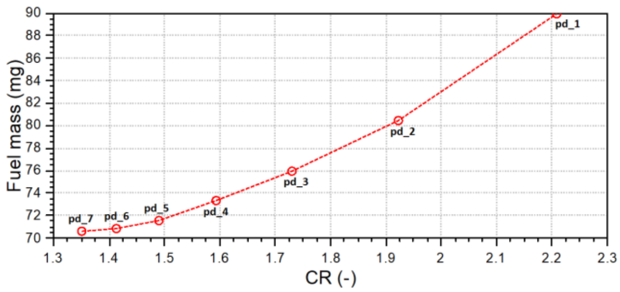

Fig.7. Supplied fuel mass per operating cycle vs. compression ratio.

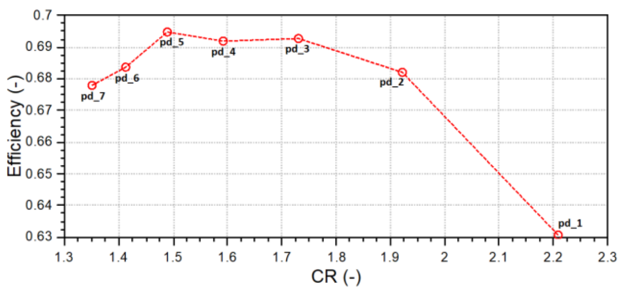

Fig. 8. Compressor efficiency vs. compression ratio.

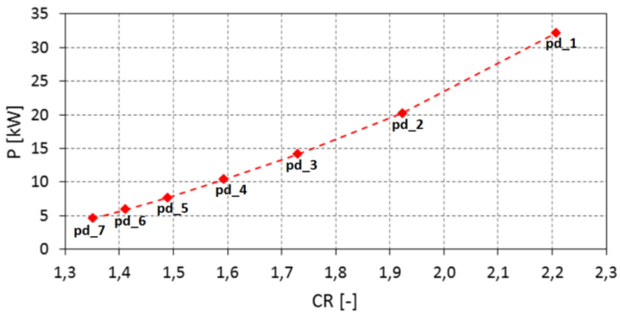

Fig. 9. Mechanical power consumed by the compressor vs. compression ratio.

5 Conclusions

Regarding the mechanical power needed by the compressor (Fig. 8), high compression deteriorates the overall efficiency of the engine (see Figure 3). Figure 6 shows the change in the specific unit fuel consumption from 210 g/kWh to nearly 270 g/kWh, which reduces the overall efficiency from 41% to 32% compared to the unchanged indicated efficiency.

Any further analysis and stand research should refer to the last four variants of forced induction, *i.e.* pd_4, pd_5, pd_6, and pd_7 because for higher pressure, the engine needs too much power so specific fuel consumption increases.

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References

1. J. P. Pirault, M. Flint, *Opposed Piston Engines: Evolution, Use and Future Applications* (SAE International, 2010)
2. P. Tribotte, F. Ravet, V. Dugue, P. Obernesser, N. Quechon, J. Benajes, R. Novella, D. De Lima, Procedia Soc. Behav. Sci. **48**, 2295–2314 (2012)
3. V. DeBellis, S. Marelli, F. Bozza, M. Capobianco, Enrgy Proc. **45**, 909–918 (2014)
4. Y. Liu, F. Zhang, Z. Zhao, Y. Dong, F. Ma, S. Zhang, Appl. Therm. Eng. **104**, 184–192 (2016)
5. G. Cantore, E. Mattarelli, C. A. Rinaldini, Enrgy Proc. **45**, 739–748 (2014)
6. P. Carlucci, A. Ficarella, D. Laforgia, G. Trullo, Enrgy Proc. **82**, 31–37 (2015)
7. P. Carlucci, A. Ficarella, D. Laforgia, A. Renna, Energy Convers. Manag. **101**, 470–480 (2015)
8. S. Naik, D. Johnson, J. Koszewnik, L. Fromm, F. Redon, G. Regner, K. Fuqua, Practical Applications of Opposed-Piston Engine Technology to Reduce Fuel Consumption and Emissions, SAE Technical Paper 2013-01-2754 (2013)
9. F. Ma, Ch. Zhao, F. Zhang, Z. Zhao, Z. Zhang, Z. Xie, H. Wang, Energies **8**, 6365-6381 (2015)
10. J. Bohbot, M. Miche, P. Pacaud, A. Benkenida, Oil Gas Sci. Technol. – Rev. IFP **64(3)**, 337-359 (2009)
11. P. Carlucci, A. Ficarella, G. Trullo, Energy Convers. Manag. **122**, 279–289 (2016)
12. G. P. Blair, *Design and Simulation of Two-Stroke Engines* (Society of Automotive Engineers, 1996)
13. J. B. Jones, R. E. Dugan, *Engineering Thermodynamics* (Prentice Hall, 1995)
14. www.rotrex.com/media/7df7470f-60fb-4413-837e-ebe527259be4/PDF/Rotrex%20Technical%20Datasheet%20C30%20Range_pdf