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INFLUENCE OF INTAKE VALVE CLOSURE ANGLE ON IC ENGINE INDICATED PARAMETERS

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Abstract

The paper presents results of modelling study of influence of an intake valve closure angle on IC engine indicated parameters. The modelled engine was Andoria S231, which was working on methane. At first, optimizations of the model were done by comparison of the indicated mean effective pressure for real engine and modelled engine. Next, modelling was done for early intake valve closure angle in comparison to original closure angle. The engine was simulated as a naturally aspirated one and for the cases such indicated; parameters as indicated efficiency, mean indicated pressure, fuel consumption were calculated. During the modelling ignition, timing and air-fuel ratio were fixed. For better comparison for two cases of early intake valve closure angle the engine was modelled as super-charged one where mean indicated pressure was fixed at the same level as for the naturally aspirated engine working with original valve timing and indicated parameters were calculated and compared with in parameters determined from this naturally aspirated engine. Because of the calculations, characteristics of indicated parameters vs. intake valve closure angle were computed. As a result of this research, both the decrease in indicated efficiency, indicated mean effective pressure were shown, temperature of fresh charge, end of compression stroke and maximum in-cylinder temperature were observed for naturally aspirated engine with early intake valve closure angle.

Keywords: *over-expanded cycle, turbocharging, valve timing, CNG*

1. Introduction

One of the major engine component, which affects its performance as well as indicated parameters and exhaust emission, is the valve mechanism. The valve mechanism is responsible for proper and efficient delivery of fresh charge (air and fuel or air) to the engine and for effective exhaust gases release from the cylinder [14]. To do that, proper valve timing should be computed and implemented. Because of a wide range of the engine rotational speed under it work, the valve timing has to be a compromise between economical work with low fuel consumption by the engine and plausible performance for the user. Because of work with constant rotational speed of the industrial engine, valve timing is fixed for the engine maximum volumetric efficiency. To increase the indicated efficiency and decrease in exhaust gases emission, the over-expanded cycle can be applied to the stationary gas engines. In this case, the over-expanded cycle is achieved by modification of valve train and valve timing for intake valve [7]. The main issue of such modification is to decrease in maximum in-cylinder temperature, which influences on NO_x emission [9, 12, 13]. Lower value of the in-cylinder maximum temperature can also reduce the probability of occurrence and intensity of knock combustion [10, 11]. There are several methods to modify the valve train and change the valve timing. The most popular from it, is modification of intake valve cam slope in that way that it provides early intake closure before bottom dead centre (BBDC) or late intake valve closure after bottom dead centre (ABDC). However, with very early or very late intake valve closure relative to BDC, decrease in indicated mean effective pressure and indicated efficiency might occur. One of the methods to reduce this drawback is to apply supercharging the engine what in effect contributes to increase in both IMEP and indicated efficiency [3-6, 8].

In this paper, results of computer analysis of influence of intake valve closing angle on indicated parameters of internal combustion engine are presented. During this research, the analysis was conducted for two configurations of the engine: the naturally aspirated and the turbocharged with proper selection of turbocharger specifications.

2. Test configuration

The research was performed with aid of computation code named “SILNIK 32”. The SILNIK 32 is the program, which calculates indicated parameters of the engine. The combustion model applied to the program code is the zero dimensional dual zone model. Working fluid can be simulated as a homogenous mixture made of 10 different gases, mixed each other at various ratios. The code also gives possibility to model combustion of liquid hydrocarbon fuels with assumed the H/C mass ratio. The NCHR (normalized cumulative heat release) in this model is described by two separately defined Wiebe functions, which were merged into a single function. The first function models kinetic combustion, the second one – diffusion combustion, respectively. The parameter defined as the start of combustion for both kinetic and diffusive combustion is at the same value while both the ratio of kinetic to diffusion combustion, the angle of occurrence of maximum dp/dφ and the end of combustion can be varied. In this model, the exchange of heat between unburned and burned charge zone is not considered. For calculation of heat exchange between environment and combustion chamber walls, the average temperature by mass was taken. The coefficient of heat convection is calculated by Woschni correlation. Charge flow thru intake and exhaust valves are calculated on a basis of Sain-Venant correlation and instantaneous values of effective flow are calculated on a basis of the cam profile and dependence of flow number and valve lift. The program allows calculating the turbocharged version of the engine where temperature and pressure for fresh charge are determined by user. The turbine of the supercharger is modelled as a nozzle with special diameter where the decrease of pressure is equal to that in a real turbine engine. The ratio between isentropic-work of a supercharger to isentropic work of a turbine is the parameter, which describes proper selection for a turbocharger to the engine intake and the exhaust system [1].

The research was done in three stages. In the first, the optimization of the model was done by comparing the IMEP from a real engine with the modelled IMEP according to the equation 1:

$$\mathrm{DIW} = \frac{\mathrm{IMEP_{real}}\text{-}\mathrm{IMEP_{calc}}}{\mathrm{IMEP_{real}}} \cdot 100, \tag{1}$$

where:
DIW – difference of IMEP,
$\mathrm{IMEP_{real}}$ – indicated mean effective pressure for measured cycle,
$\mathrm{IMEP_{calc}}$ – indicated mean effective pressure for calculated cycle.

The geometrical parameters of the modelled engine were the same as for the real one. To achieve satisfactory correlation between combustion in the real engine and the modelled one the following parameters: start of combustion, maximum dp/dφ and end of combustion were taken into analysis. The comparison dealt with the gross IMEP, in which charge exchange work is neglected; hence, combustion plays the most crucial role. The parameters of the basic model are shown in Tab. 1.

Tab. 1. Parameters of basic model

Parameter	Value
ε (-)	11.5
IVC (° CA)	221
IVO (° CA)	716
EVC (° CA)	1
EVO (° CA)	507
IT (° CA) BTDC	27

The results of optimization are shown in Fig. 1.

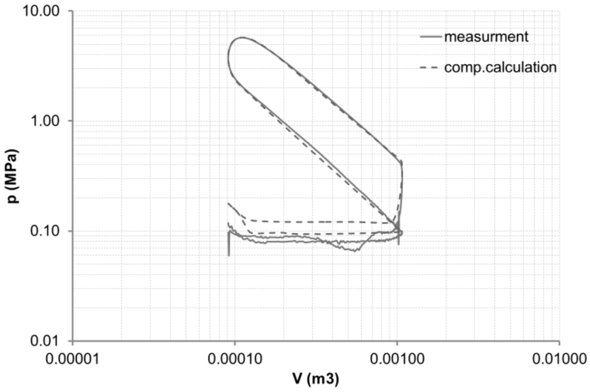

Fig. 1. Comparison between measured pressure trace and calculated pressure trace

The DIW calculated for this comparison is -0.6% between the IMEP.

In the second stage for the optimized model, the valve timing was changed by advancing the intake valve closure angle from the initial angle characteristic for the basic model to value of CA 90° with step of 10°. The model configuration during this stage of research is shown in Tab. 2.

Tab. 2. Model configuration during the second stage of research

Parameter	Value
IVC (° CA)	90-221
IVO (° CA)	716
EVC (° CA)	1
EVO (° CA)	507
IT (° CA) BTDC	27
λ (-)	1

For the each calculated point the following parameters: IMEP, indicated efficiency, volumetric efficiency, temperature of fresh charge, end of compression temperature, maximum in-cylinder temperature, heat release (HR) and normalized heat release (NHR) were stored. In this stage of the research, the engine was modelled as the naturally aspirated one.

In third stage, the engine was modelled as the turbocharged one. The boost pressure was increased to obtain IMEP at level equal for the basic model with originally valve timing for two cases of intake valve closure angle, 150° of CA and 90° of CA. For this intake valve closure angle indicated parameters was also stored. During this investigation, the parameters of the turbocharger were optimized to achieve the ratio for isentropic work of compressor to isentropic work of turbine (N_s/N_T) in the range from 0.986 and 1. In this case, to achieve such correlation, change in the diameter for the nozzle in exhaust manifold has to be modified for each case.

3. Results and discussion

In this part, results of investigation for second and third stage are shown. Fig. 2 shows in-cylinder pressure vs. volume for various angles of intake valve closure (IVC). As can be seen with earlier intake valve closure angle, there is decrease in pressure at the beginning of compression stroke. Hence, the maximum in-cylinder pressure also decreases.

With increasing early intake valve, closure angle, the volumetric efficiency also decreases, because less charge fills the engine cylinder. The dependence of intake valve closure angle against volumetric efficiency is presented in Fig. 3.

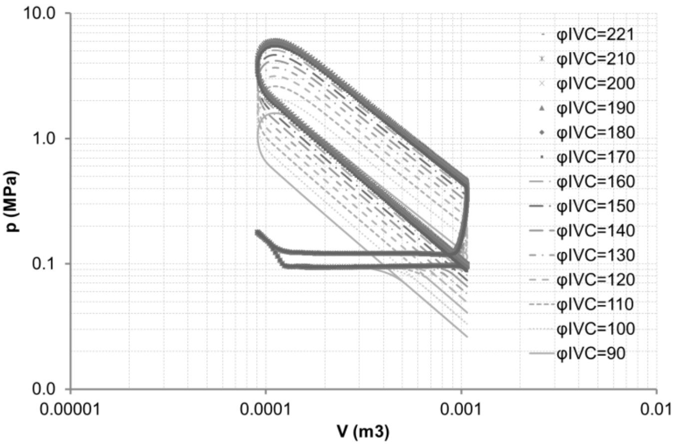

Fig. 2. Pressure vs. volume for different IVC

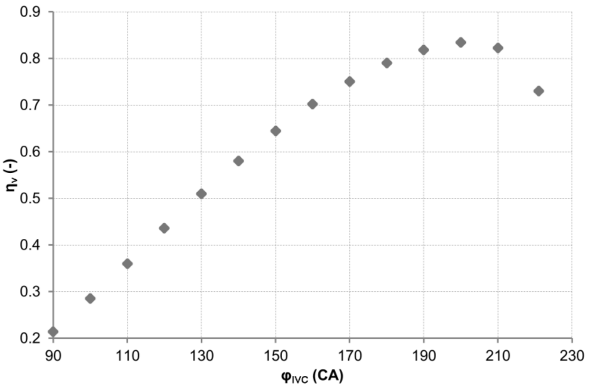

Fig. 3. η_v vs. IVC

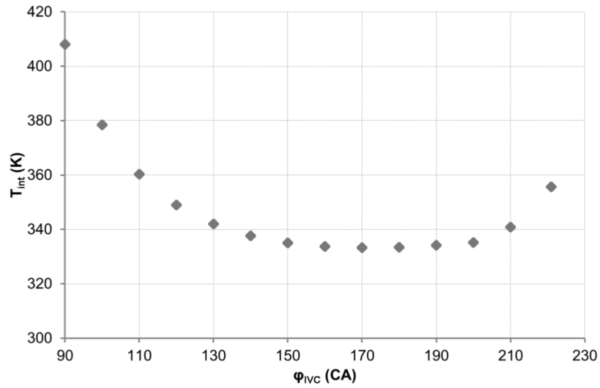

Fig. 4. T_{int} vs. IVC

As can be seen in Fig. 4 with earlier intake valve closure angle the temperature of fresh charge decreases at first to value approximately 335K for φ_{IVC}=200°CA. Than where the intake valve is closed earlier the temperature is approximately at constant level between φ_{IVC}=150...200° CA and next it starts to increase. The increase of fresh charge temperature might be caused by heating it by the in-cylinder walls because of longer stay inside hot cylinder than it occurs in the basic engine.

Figure 5 shows the dependence of intake valve closure angle for the temperature at the end of compression stroke. This temperature decreases with early intake valve closure angle almost in the whole range. However, for the earliest intake valve closure angle the temperature slightly rises what can be cause by the increase in fresh charge temperature. The maximum temperature decrease at the end of compression stroke is about 80K.

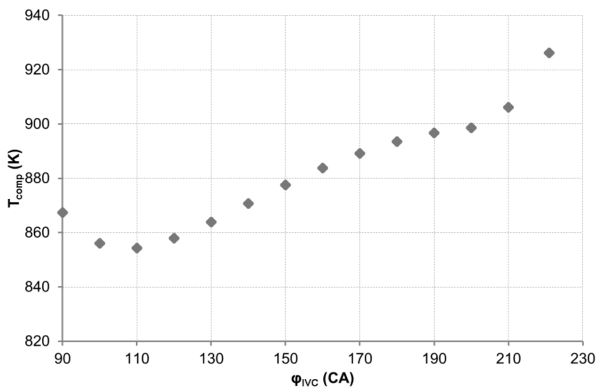

Fig. 5. T_{comp} vs. IVC

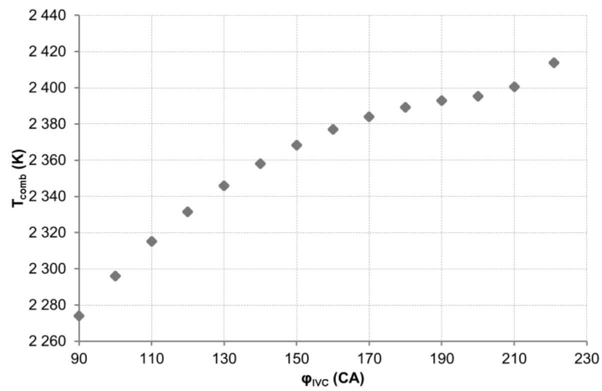

Fig. 6. T_{comb} vs. IVC

In Fig. 6, the dependence of intake valve closure angle on the maximum in-cylinder temperature is shown. As a consequence of early intake valve closure, angle the combustion temperature decreases in the whole range. As can be seen the maximum difference in temperature is about 140K. In the maximum in-cylinder, temperature has no tendency to increase in its value at the earliest intake valve closure what can be caused by combined influence of compression temperature, combustion pressure and the portion of energy included in the fresh charge.

Figure 7 presents the influence of intake valve closure angle on the heat release. As can be seen with a decreasing intake valve closure angle the heat release value is slightly increasing to intake valve closure angle about φ_{IVC}=180° of CA and next is decreasing. That phenomenon occurs because for the range of intake valve closure angle from 221° to 180° of CA a part of fresh charge is push out to the intake manifold. For the intake valve closure angle equal to 180° of CA a HR value reaches maximum and for the intake valve closure angle from the range of 180° to 90° of CA a HR decreases what is caused by decrease in volumetric efficiency and a smaller share of a fuel consumed by the engine at constant excess air ratio.

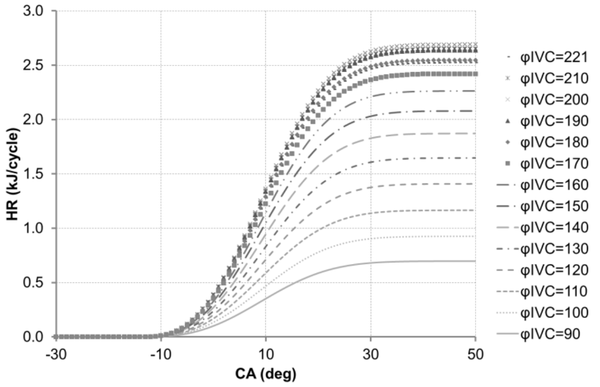

Fig. 7. HR vs. CA for different IVC

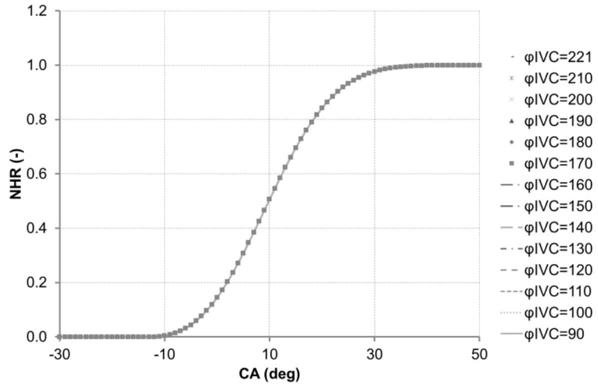

Fig. 8. HRR vs. CA for different IVC

In Fig. 8 the normalize heat release rate is shown. As can be seen there is no difference in the NHR for various intake valve closure angles.

In Fig. 9 mean indicated pressure (IMEP) against intake, valve closure is depicted. As can be seen the mean effective pressure slightly increases with early intake valve as it reaches φ_{IVC}=180° of CA, then the decrease in mean indicated pressure occurs. This is conducted with increase and decrease in volumetric efficiency as well as with the energy of charge trapped in working area of the engine after intake valve closure.

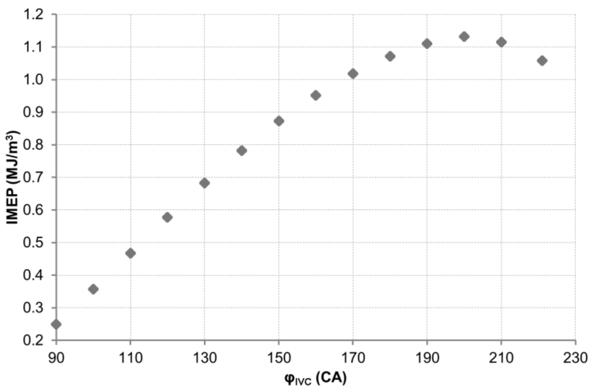

Fig. 9. IMEP vs. IVC

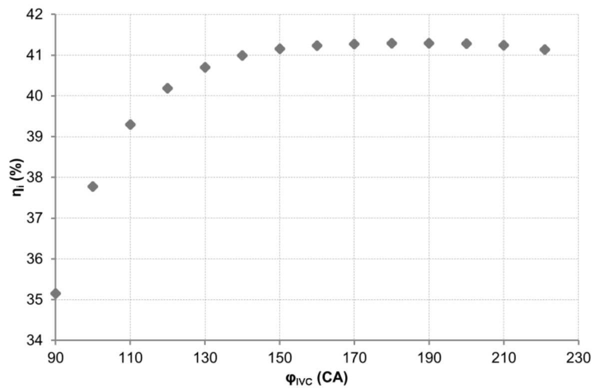

Fig. 10. η_i vs. IVC

The indicated efficiency is at constant level in wide range of intake valve closure angle. The drop in indicated efficiency occurs when the intake valve closure angle exceed φ_{IVC}=140. Between the values from 221° to 140° of CA, indicated efficiency is almost constant, what shows Fig. 10.

To reduce this decrease in indicated mean effective pressure and indicated efficiency with an engine with early intake valve closure angle, the turbocharging can be used. During this stage of research the increase in boost pressure for the engine with the early intake valve closure angle was necessary to achieve indicated mean effective pressure equal IMEP=1.06 MJ/m^3 which corresponds to IMEP for the basic engine. Fig. 11 presents the comparison of pressure trace as a function of volume for three intake valve closure angles 221°, 150° and 90° of CA.

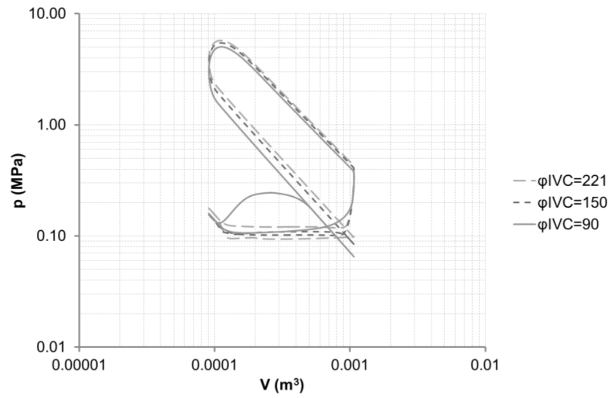

Fig. 11. Pressure-trace vs volume for three values of intake valve closure angle

As can be seen for the engine with the earliest intake valve closure angle the highest boost pressure is required to compensate drop in the IMEP. The results of the third stage of investigation are presented in Tab. 3.

Tab. 3. The results of third stage calculation

φ_{IVC} (deg)	η_v	T_{int} (K)	T_{comp} (K)	T_{comb} (K)	IMEP (MJ/m^3)	η_i (%)	p_{boost} (kPa)
221	0.73	355.7	926.2	2413.8	1.06	41.1	0
150	0.77	324.0	861.9	2379.8	1.06	42.5	14
90	0.74	324.6	755.3	2303.5	1.06	44.2	154

As can be concluded from Tab. 3, despite the turbocharging the compression temperature and maximum in-cylinder temperature decreases with early intake valve closure angle. Turbo charging also provides increase in volumetric efficiency, indicated efficiency and allow to keep indicated mean effective pressure at constant level. As mentioned, for the model option with the earliest intake valve closure angle to achieve IMEP typical for the basic model the turbocharging with high boost pressure is needed.

4. Conclusions

Because of this investigation, the followed conclusions are presented:

1. Early intake valve closure angle allows decreasing in maximum in-cylinder temperature and pressure, what can in consequence lead to decrease in NO_x emission and reduce probability of knock combustion as well. The decrease in maximum in-cylinder temperature was approximately 140K.
2. Early intake valve closure causes decrease in volumetric efficiency, what is caused by smaller amount of fresh charge trapped in working area of the engine. In this case the volumetric efficiency decreases from η_v=0.85 to η_v=0.2.

3. The change in intake valve closure angle does not affect normalized heat release rate. The heat release also decreases with the increasingly early intake valve closure angle what is caused by decrease in fresh charge trapped in working area of the engine.
4. Early intake valve closure causes decrease in indicated mean effective pressure. The denoted lost in IMEP was nearly 0.9MJ/m^3 from nearly 1.15 MJ/m^3 (φ_{IVC}=180° of CA) to 0.2 MJ/m^3 (φ_{IVC}=90° of CA).
5. Early intake valve closure cause decrease in indicated efficiency from η_i=41.3% (φ_{IVC}=180° of CA) to η_i=35.1 % (φ_{IVC}=90° of CA).
6. Turbocharging the engine with early intake valve closure angle is a good method to compense the loss in IMEP. With very early intake valve closure angle to obtain IMEP at level for basic model relatively high boost pressure is required (φ_{IVC}=90° of CA, p_{boost}=154 kPa).
7. Turbocharging causes increase in indicated efficiency especially for high boost pressure. The increase in the indicated efficiency for the turbocharged engine was almost 4% compared to the basic engine.
8. The in-cylinder peak temperature for the turbocharged engine was also lower by approximately 100K if compared to the basic engine.

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