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NUMERICAL ANALYSIS OF THE USE OF DIFFERENT CAMSHAFTS IN A SPARK IGNITION INTERNAL COMBUSTION ENGINE

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Abstract. *This work presents a numerical analysis where the performance of an internal combustion engine of spark ignition is evaluated, using different camshafts to control the opening and closing of the intake valves of this thermal machine. The analysis was conducted only with the scope in the camshaft that control the intake valves. The authors believe that this has a greater influence on the performance of the engine, since, during the event of engine cycle exhaustion, the combustion products are under high pressure, so that they already have a natural tendency to outside the combustion chamber. At the end of the study, the results did not show any kind of benefit that justifies the change of the original camshaft of the engine for the others analyzed. However, the results of the simulations help to understand a worldwide trend, which is the insertion of capable systems to control the opening and closing of the valves, simultaneously.*

Keywords: *Internal Combustion Engine, Computational Simulation, Camshaft*

1. INTRODUCTION

Internal combustion engines are thermal machines that uses the energy present in a fuel, whether fossil or not, to produce work. This fact makes many people mistakenly think that in order to extract more work from them, it would be enough to insert more fuel into the system per cycle. However, the combustion reaction is only possible with the presence of more oxidizing fuel. So that, there is another component necessary for this reaction to occur. Based on this assumption, is possible to start to understand that to be able to extract more work of these machines, a greater amount of air (which is our oxidant) will have to be inserted into the system.

By this reasoning it is understood that different camshafts will provide different time intervals in which the valves are opened. Therefore, for the intake valve, there will be more or less time to introduce air into the system. In this way, the present work proposes the analysis of different camshafts for the control of the opening and closing of the intake valves of a spark ignition engine.

For the development of this analysis was used the software Diesel-RK®, of Russian origin, which has a license for academic uses.

The engine used as a reference for the analysis is a four stroke engine, spark ignition of 1497 cm³ volumetric displacement, 3 cylinders, water cooled and using ethanol as fuel. The computational model of this engine, developed in the aforementioned software, was previously described in previous works by these same authors (Tonon, 2018).

2. DIESEL-RK® SOFTWARE AND INPUT PARAMETERS

The Diesel-RK® software was developed to work with internal combustion engines of alternative type inside the Bauman Moscow State Technical University (BMSTU) in Russia, to be used as a tool in the research field (Kuleshov, 2004). Currently the tool has different versions, and one of them is available, free of charge, for academic use.

The creation of a model in the tool is based on the insertion of some geometric and operation parameters of the engine to be simulated. A more detailed description of the creation of the model can be seen in Tonon (2018). An important point to be mentioned is that the heat exchanges losses are analyzed by the Woschini model (1967) and the combustion model is characterized by the Wiebe curve, that can be seen in (Ferguson and Kirkpatrick, 2001) and (Heywood, 1998). Figures 1 and 2 show the windows, in the software, where these two parameters are inserts.

It is also necessary to insert the data of the fuel used, which in this case is ethanol, and which has its properties highlighted in Tab. 1. Part of these properties were determined experimentally and others were calculated using the method described by Lacava (2014).

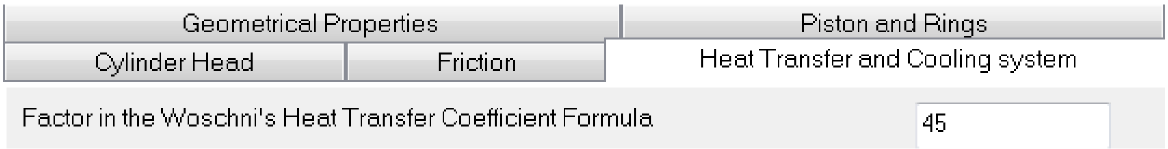

Figure 1. Woschni coefficient change window.

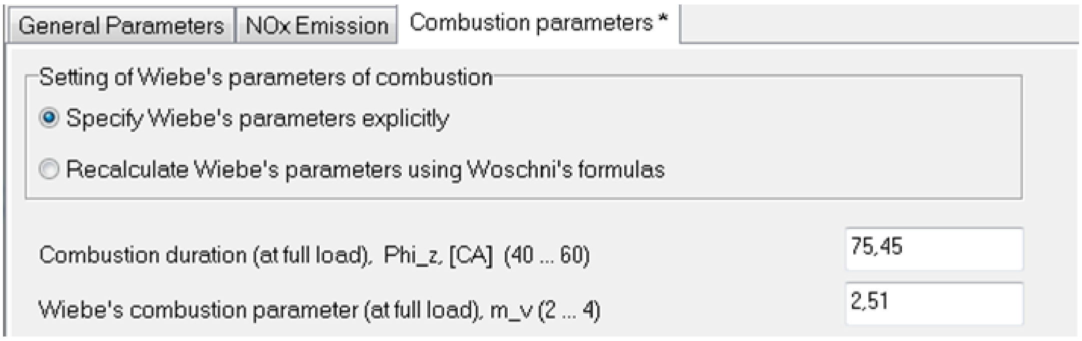

Figure 2. Input screen for Wiebe parameters.

Table 1. Parameters obtained for the equivalent fuel.

C *[Mass Fraction]*	0.4921
H *[Mass Fraction]*	0.1302
O *[Mass Fraction]*	0.3777
Lower Calorific Power *[MJ/kg]*	24.4
Fuel Density a 323 *K* in *[kg/m3]*	783
Specific Heat of Vaporization *[kJ/kg]*	1019.22
Thermal Capacity of Fuel at Injection Temperature *[J/kgK]*	2405.6
Molecular mass *[kg/kmol]*	42.365

3. GAS EXCHANGES

Gas exchanges in internal combustion engines are performed through the inlet and exhaust valves. This analysis however can become very complicated, due to the involved mechanisms and characteristics of the fluid displacement. Some hypotheses can be seen in Caton (2016) and Stone (1992), being:

- Quasi-steady flow;
- One-dimensional flow;
- Adiabatic and reversible flow;
- Compressible flow.

It can then be correlated the actual mass flow ($\dot{m}$), with the isentropic mass flow ($\dot{m}_{is}$), through the coefficient of flow (C_f), as shown in Eq. (1) (Ferguson and Kirkpatrick, 2001).

$$\dot{m} = C_f \cdot \dot{m}_{is} \tag{1}$$

And for the calculation of the isentropic mass flow, was used the Eq. (2) (Ferguson and Kirkpatrick, 2001):

$$\dot{m}_{is} = \rho_v \cdot A_v \cdot u_{is} \tag{2}$$

Since (u_{is}) is the isentropic reference velocity, (A_v) is the cross-sectional area of the valve and e (ρ_v) is the density of the fluid in the valve.

The calculation of (u_{is}) is done using Eq. (3), where (ρ_0) and (p_0) are, respectively the density and the stagnation pressure, and (p_v) is the static pressure at the valve (Ferguson and Kirkpatrick, 2001).

$$u_{is} = \left(2 \cdot \frac{\gamma}{\gamma - 1} \cdot \frac{p_0}{\rho_0} \cdot \left(1 - \left(\frac{p_v}{p_0} \right)^{\frac{\gamma}{\gamma - 1}} \right) \right)^{1/2} \tag{3}$$

The density and pressure in the valve can be correlated with the density and pressure of stagnation upstream of the flow through Eq. (4) (Ferguson and Kirkpatrick, 2001).

$$\rho_v = \rho_0 \cdot \left(\frac{p_v}{p_0} \right)^{1/\gamma} \tag{4}$$

For stagnation conditions, considering perfect gas behavior, Eq. (5) represents this.

$$p_0 = \rho_0 \cdot R \cdot T_0 \tag{5}$$

For the calculation of the sound velocity in the stagnation condition (c_0), is used Eq. (6) (Ferguson and Kirkpatrick, 2001).

$$c_0 = (\gamma \cdot R \cdot T_0)^{1/2} \tag{6}$$

Finally, the actual mass flow ($\dot{m}$) is shown in Eq. (7) (Ferguson and Kirkpatrick, 2001).

$$\dot{m} = \rho_0 \cdot C_f \cdot A_v \cdot c_0 \cdot \left[\frac{2}{\gamma - 1} \cdot \left(\left(\frac{p_v}{p_0} \right)^{2/\gamma} - \left(\frac{p_v}{p_0} \right)^{\frac{\gamma + 1}{\gamma}} \right) \right]^{1/2} \tag{7}$$

Under supersonic conditions, in some cases, it can be assumed that the pressure in the valve will be equal to the pressure inside the combustion chamber.

Considering Eq. (2), it is assumed that the flow is isentropic to an upstream condition. The flow will pass through a minimal area in the groove (A_v). In the same way as in a divergent convergent nozzle. However, it is necessary to define the minimum area. Figure 3 shows two possible areas that may be minimal in the flow (Ferguson and Kirkpatrick, 2001).

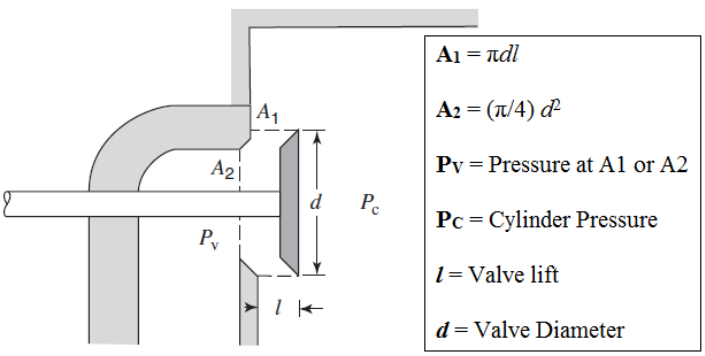

Figure 3. Idealized model of flow areas in valves (Ferguson and Kirkpatrick, 2001).

The area (A_1) is called the curtain area, and the area (A_2) is called the door area. For low lift valve conditions, the minimum area will be the curtain area, and as the lift increases, that area will grow until the door area is shorter. It is important to say that this model does not take into account the effects of valve seat.

In relation to the flow coefficient, it is defined as a ratio between the effective flow area (A_f), and the minimum area in the throat (A_v). Since (A_f) is always related to the area (A_v), it will also depend on the minimum area at the time being analyzed. In this way, the curtain area will sometimes be associated with the door area. When associated to (A_1) the coefficient flow is called discharge coefficient (C_d), while for (A_2) t will be denoted as (C_f), as shown by Eq. (8) e (9) (Ferguson and Kirkpatrick, 2001).

$$A_f = C_f \cdot A_v = C_f \cdot \frac{\pi}{4} \cdot d^2 \tag{8}$$

$$A_f = C_d \cdot A_v = C_d \cdot \pi \cdot d \cdot L \tag{9}$$

The coefficients (C_d) and (C_f), can be found in several ways. They can be measured experimentally or empirically. In the *Diesel-RK*®, an empirical correlation is used, where it is enough to provide the coefficient value for the maximum opening position. With this value, the software will determine the coefficient for each valve lift.

Some literatures work only with the coefficient of discharge (C_d). In some cases this is used as a constant value, around 0.7, that can be seen in Caton (2016) and Medina et al. (2014).

The equation shows that there are several influence variables in the gas exchange process, and that one of them is time (note that the amount of mass passing through the valves is given through a rate). In this way, the period that the valves remain open exerts a strong influence in the parameters of performance and consumption of an internal combustion engine, since changing this period, is also changed the time for the gas exchanges. In a camshaft, this time can be represented by the amount in degrees of the crank angle, that the valve stays open.

Note also that not necessarily a camshaft with a longer crank length will always be better. The amount of mass of air and fuel that will participate in the combustion is also a function of the moment the intake valve is closed. Therefore, if after the piston reaches the bottom dead center after the intake event, the valve still remains to be closed, part of the fresh mass will be expelled from inside the combustion chamber. This effect must be considered when choosing the duration of the camshaft to be used.

4. RESULTS AND DISCUSSION

The results presented in this paper are presented as percentages, for reasons of confidentiality of the company that submitted the data of the analyzed engine, which wishes to remain anonymous. In this way, torque results, which are usually presented in the *Nm*, will be presented as a percentage of a reference value, which can also not be reported.

In this work were opted to analyze the behavior of the engine when the opening time of the intake valve was changed. It was chose to analyze this valve because it is understood that it has a greater influence on performance parameters, more than the exhaust valve. This information is based on the fact that the difficulty of inducing fresh mass into the combustion chamber is greater than that of removing the combustion products from it.

This can be displayed by two callsigns. The first is the fact that a combustion chamber is at a much greater pressure than a central exhaust of a chamber, in a way that there is a natural combustion apparatus for the chamber forums. The second is the working flow code for the exhaust valves, which in Diesel-RK® itself is larger than that for the inlet valves (Kuleshov, 2004).

Four camshafts with different lengths of the original were analyzed. These have a duration of 215, 225, 255 and 265 degrees of the crank angle.

Figure 4 shows the torque results for the wide open throttle (WOT) situation for all analyzed camshafts. In this figure it is noticed the existence of three regions, divided between rows of rotation. The first, between 1250 and 3500 rpm, the second between 3500 and 5500 rpm, and the third between 5500 and 6900 rpm. In the first range, camshafts with shorter durations have the best performance results. Then, there is a transition phase in the second rotation range. In the third range there is a region where camshafts with higher have better results. This result was expected, due to the fact that at low rotations the camshaft may have a shorter duration because there is more time available for the admission of fresh mass. However, at high speeds the available time is shorter, so camshafts with longer durations are desirable.

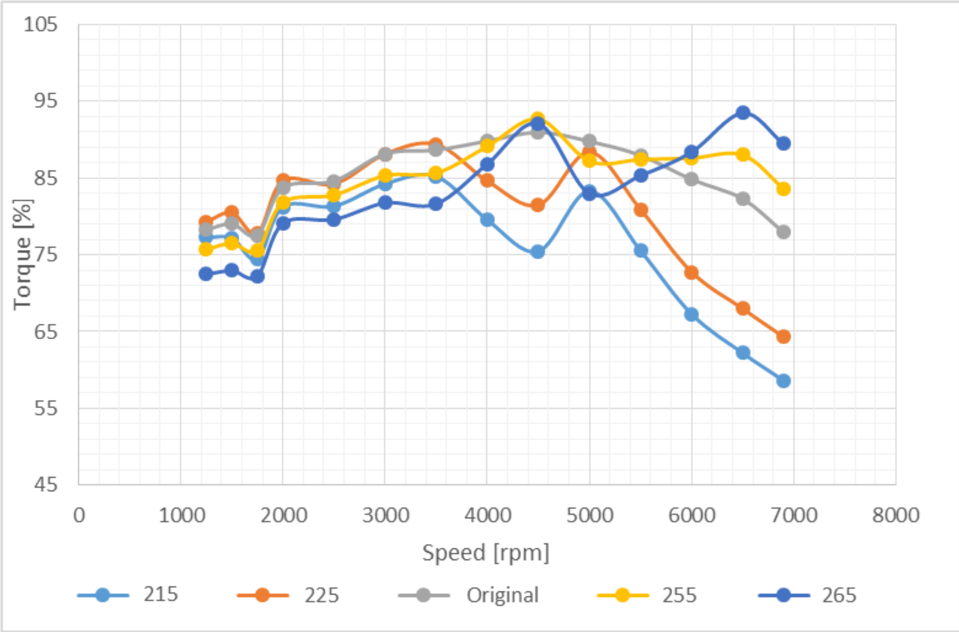

Figure 4. Results of the use of different camshafts for WOT condition.

In Figure 5 the same results are shown however for a loading condition of approximately 75%.

Figure 5. Results of the use of different camshafts for 75% load.

In this case, there are the same three regions as for the WOT case, having the same behavior characteristics. However, these are more evident. In these conditions it can be noted that the difference between the camshafts for the low and high rotation regions become more prominent. At low speeds, the camshafts with shorter durations are presenting performance results that are much more favorable than those of longer durations. The reverse happens at the highest rotations.

In Figure 6 these same results are shown for a loading condition of approximately 65%.

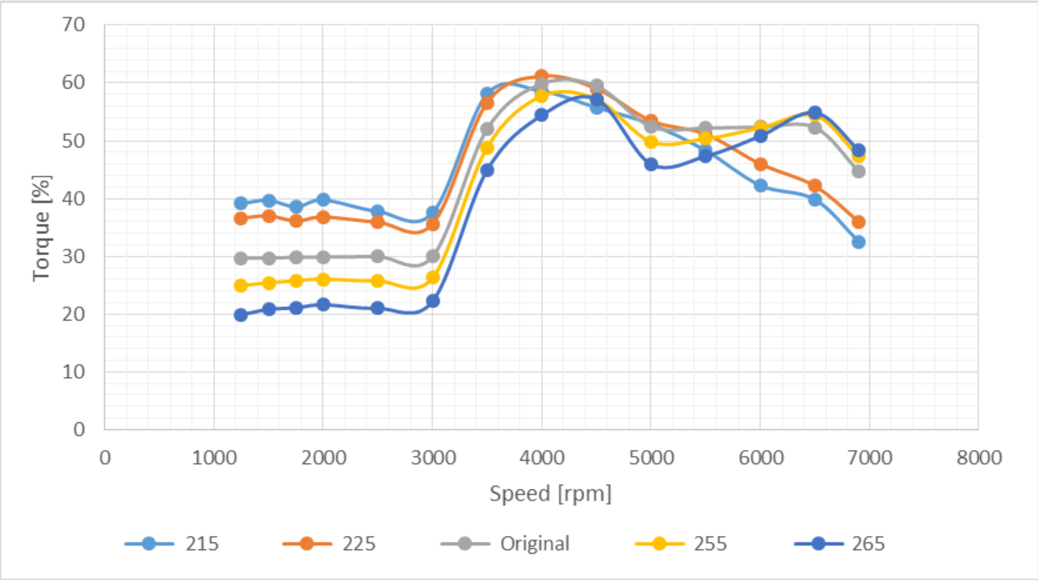

Figure 6. Results of the use of different camshafts at 65% load.

The same behavior of the previous cases is still present in this load condition. However, it is noticed that in the region of high speeds, the difference in the results between the camshafts of greater durations is smaller. The explanation for this difference in influence may be associated with control parameters, such as the intake valve opening and closing points. However, in this part of the results the objective is only to analyze what would happen with the use of other camshafts.

Figure 7 shows the last results of this part of the work, referring to a load condition of approximately 50%.

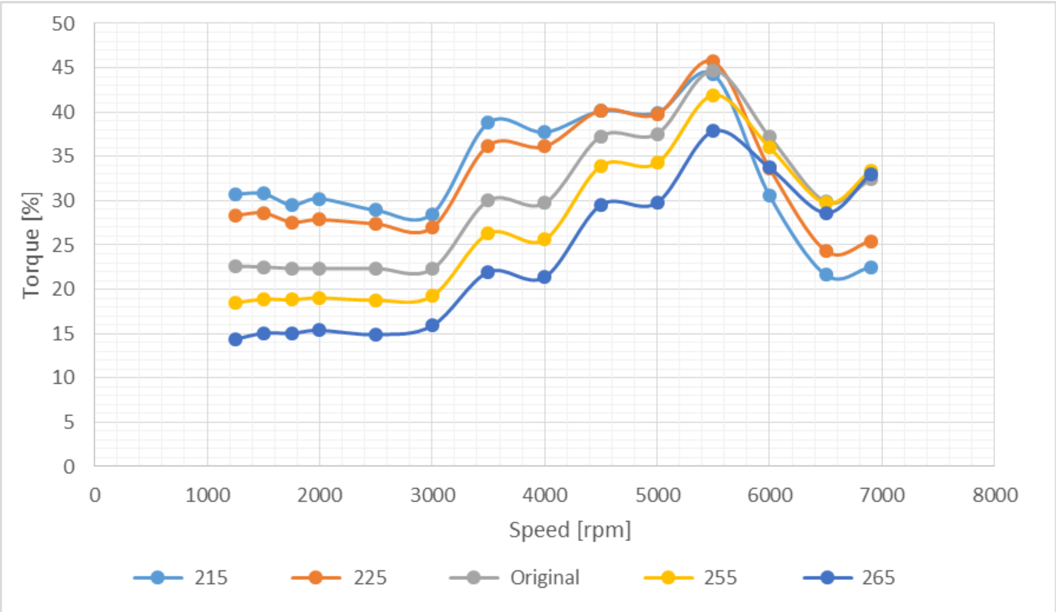

Figure 7. Results of the use of different camshafts for 50% load.

The results for this load do not present three distinct regions like the others. During most of the engine speed range, camshafts of shorter duration had higher performance results than those of longer duration, and only at the highest speeds the behavior changed. The prevalence of better results with shorter camshafts may again be associated with engine control parameters, which were not modified for the simulations.

It should be noted from the results that there is no camshaft with superior performance over the entire speed range. So that, some criteria must be adopted to determine whether it would be advantageous to replace the original camshaft with any of the others analyzed. The criterion adopted was to use the camshaft with the largest area, below the torque curve, for wide open throttle conditions. Based on the calculations, the original axis has an area only 0.5% greater than the axis of 255°, in relation to the others, the area is 2 to 10% larger. In spite of being a very small area, the original camshaft has another advantage, which is that of a smoother torque curve behavior under WOT conditions. Based on this analysis, it is concluded that it would not be interesting to use a different camshaft for this engine under the conditions analyzed.

In other words, this adopted criterion (area below the graph) is similar to what is done in a process of integrating a function. It is considered that a larger area represents a greater amount of energy being harnessed by the thermal machine. The authors acknowledge that this area does not directly represent this physical interpretation, however, it is an extrapolation that can lead to this conclusion indirectly.

5. CONCLUSION

In this work a numerical study was done, proposing to change the duration of the opening of the intake valve, through the use of different camshafts.

It was found that it would not be interesting to replace the camshaft currently used by any of the following analyzed: 215, 225, 255 and 265° of the crank angle.

However, something interesting in the presented results, is that they allow to show a tendency of the market, that is the insertion of mechanisms that can control the opening and closing of the valves. As the results show, if the engine analyzed had control of both the opening and closing of the intake valve, it would be possible to always work at the simulated maximum points, which would guarantee a gain of approximately 2.6% of the area below the torque curve, in full throttle conditions. In partial loads this gain could be up to 15%.

6. ACKNOWLEDGEMENTS

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8. RESPONSIBILITY NOTICE