THE COMBUSTION ENGINE INTAKE MANIFOLD CFD MODELING DEPENDING ON THE AIRBOX CONFIGURATION

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The aim of the article is to present the possibilities of application of computational fluid dynamics (CFD) to modelling of air flow in combustion engine intake manifold depending on airbox configuration. The non-stationary flow occurs in internal combustion engines. This is a specific type of flow characterized by the fact that the variables depend not only on the position but also on the time. The intake manifold dimension and geometry strongly effects intake air amount. The basic target goal is to investigate how the intake trumpet position in the airbox impacts the filling of the combustion chamber. Furthermore, the effect of different distances between the trumpet neck and the airbox wall in this paper will be compared.

KEYWORDS

Computational Fluid Dynamics (CFD), combustion engine, air flow, non-stationary flow, intake trumpet

1 INTRODUCTION

CFD flow modeling is a way to optimize engine design, for example, to optimize or increase engine performance. Increasing of engine power by various methods can be realized. It is possible, for example, by application of advanced injection systems [Vondracek 2018], or by efficient supercharging [Singh 2011], or for example by more precise description of the combustion start process and utilization for modeling and diagnostics [Famfulik 2021].

For example, the fuel additives application also has an effect on the performance characteristics [Sejkorova 2017]. However, the engine tuning results may affect the emission characteristics in different engine modes [Kuranc 2017]. Another possibility is the flow optimization in the engine and peripherals. This approach is therefore the subject of this paper.

The flow modeling of vehicles external aerodynamics is a well-known issue, as described in [Jakirlic 2017, Jakirlic 2020]. Similarly, for flow heat transfer modelling also can be used [Bojko 2016]. Thus, internal flow modeling is of course also used in the field of internal combustion engines [Dresler 2018, Saravanan 2014, Ceviz 2010] and a similar approach in this paper has been used.

One of the most important conditions for achieving high performance is high volumetric efficiency {Muckova 2021]. Wave theory-based pipeline tuning is a very effective tool for engine power increasing. The reflection wave timing to maximize the effect of the wave by the pipe geometry (volume, length, diameter, etc.) can be utilized. The basic principle of intake line tuning is therefore to reach the maximum pressure in front of the open intake valve, which ensures the input of additional air into the cylinder.

2 NON - STATIONARY FLOW AND HYDRAULIC LOSSES

The non-stationary flow occurs in internal combustion engines. This is a specific type of flow, which is characterized by the fact that the variables depend not only on position but also on time. The cause of non-stationary flow in the intake tract of an internal combustion engine is the movement of the piston, which causes a pressure wave into the pipeline. [Xu 2017]

By moving the piston towards the bottom dead center (BDC), the pressure in the cylinder is reduced and a pressure disturbance is formed created. The pressure commotion creates a vacuum wave that spread through open valves into the intake tract. The pressure wave causes a non-stationary flow towards the valve. As soon as the pressure wave arrives at the mouth of the intake pipeline, it bounces back as an overpressure wave. This wave entrains air at atmospheric pressure, which is located in the intake tract, and if the reflected wave reaches the intake valve before the valve is closed, it helps to get additional air into the engine cylinder.

The following generally known equations to the model have been applied:

The Continuity Equation:

$$\frac{\partial(\rho F)}{\partial t}+\frac{\partial(\rho u F)}{\partial x}=0 \qquad (1)$$

The Momentum Equation:

$$\frac{\partial(\rho u F)}{\partial t}+\frac{\partial(\rho u^2+p)F}{\partial x}-p\frac{dF}{dx}+\frac{1}{2}\rho u^2 f\pi D=0 \qquad (2)$$

The air flow, mixture flow, exhaust gases flow and combustion in an internal combustion engine are relatively complex and complicated processes. Despite this complexity, it is possible to solve the problem as a pipeline system. Such a simplified approach has the advantage of allowing various designs and optimizations to be assessed in a relatively short time.

The main disadvantage of this approach is its limited accuracy in terms of observing complex phenomena and the results are rather qualitative and to prediction and optimization of the performance and operating parameters of the engine cannot be used.

Another related problem of the topic are the pressure losses. In order to the most efficient cylinder filling, the hydraulic losses in the intake manifold must be minimized. There are two types of hydraulic losses, the friction losses and the local losses. Friction losses along the entire length of the flow can be located. They are caused by friction against the pipeline wall (by tangential stress at the wall) and by friction between the individual layers of liquid (by internal forces in a viscous flowing liquid). These losses depend on the fluid properties, the roughness and the inside diameter of the pipeline.

Local losses occur when the flow direction or the flow cross section have been changed. These losses occur in the pipeline mouth, in the bends, in the throttle valve, or close the injector of internal combustion engine.

During the pressure flow, due to an obstacle in the pipeline, the flow is detached off the wall and the subsequent transmission of turbulent vortices and their disintegration (see Figure 1).

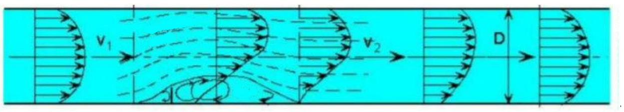

Figure 1. The influence of obstacles to the flow in a straight pipeline [Havlik 2018]

The intake manifold has a great influence to the local losses. Through the manifold the intake air from the calming chamber (airbox) flows into the pipeline. For all internal combustion engines, it is necessary to solve the engine installation problems, because of space limits. All dimensional limitations during the design works must be considered. The problems of the intake pipeline position with required length very often must be solved. In the final design, the intake manifold is then positioned very close to the walls of the airbox. This study focuses on the effect of the position of the manifold in the airbox on the fulfillment of the engine cylinder.

By suitable shaping of manifold, not only the local losses at the intake can be minimized, but even an acceleration of the flow at the intake to the manifold can be achieved. In terms of the flow and spatial conditions, the manifold with a widened neck and rounded edge (see Figure 2, shapes 6 to 9) seems to be optimal [Vizard 1999].

In practice, straight manifolds with a rounded edge also very often occur (see Figure 2, shape 3). They have a simpler construction and thus also lower production costs. The manifold with an inner diameter of 50 mm and a length of 50 mm for the study has been modeled. It is a straight manifold with a radius of rounding of 5 mm. Ahead of the intake neck of the manifold, the wall at three different distances of 30 mm, 20 mm and 10 mm has been modeled. The aim of the study is to find how the wall affects the flow of intake air into the engine.

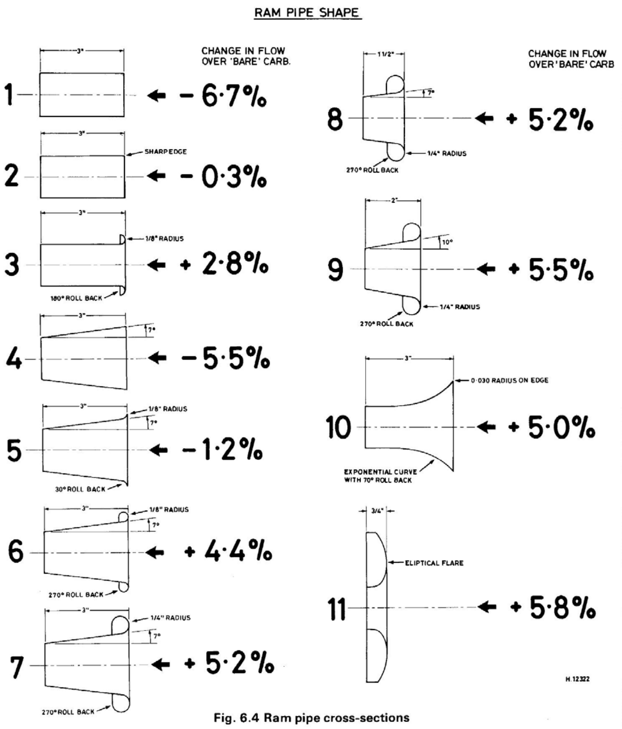

Figure 2. The influence of the shape of the intake manifold to the amount of mixture [Vizard 1999]

The course of the pressures in the intake pipeline as the inlet boundary condition for modeling has been applicated. The process cannot be modeled as a stationary flow, because in a real engine only non-stationary flow occurs.

3 ANSYS FLUENT MODEL

The simulation in the Ansys Fluent software suite has been performed. In the first phase, by utilization of the Ansys Meshing software a simple unstructured computing mesh has been created. The mesh consists of a combination of triangular (Tri) and quadrangular (Quad) elements for 2D networks, in the area of the manifold mouth and manifold walls the mash has been locally refined (because of the flow calculation in the boundary layer). The 19360 nodes and 18943 elements have been applied.

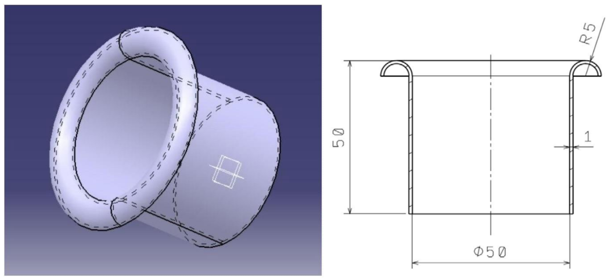

Figure 3. The model of the manifold

The computational mesh of the manifold in distance of 30 mm from the airbox wall on the Figure 4 has been shown.

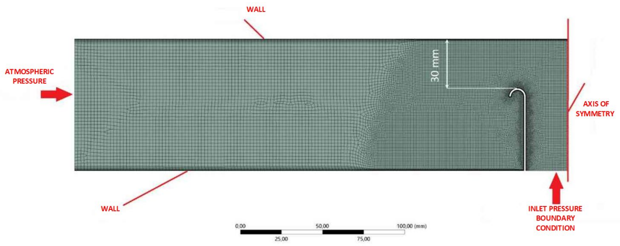

Figure 4. The computational mesh of the manifold in distance of 30 mm from the airbox wall with indication of boundary conditions

The solved problem is an axially symmetric task, and the model has been simplified to the 2D configuration and has been adapted as an axisymmetric. To solve a 2D-axisymmetric problem, the Element Behavior as axisymmetric has been set. The advantage of such modeling is speed, simplicity and computational economy. The Reynolds number characterizing the flow behavior in the pipeline is already from the flow velocity of 2,0 ms^{-1} significantly above the value defining the turbulent flow Re = d·u·v^{-1}> 3500 (where d is the characteristic dimension, u is the flow velocity and v is the kinematic viscosity). This leads to the Reynolds number around 70000. To solve this turbulent flow with high local velocity gradients by reasonable computational costs was used high-Re k-epsilon turbulence model in combination with standard wall function. Mimic of the viscous sublayer by wall function allows coarser cells in the near-wall region.

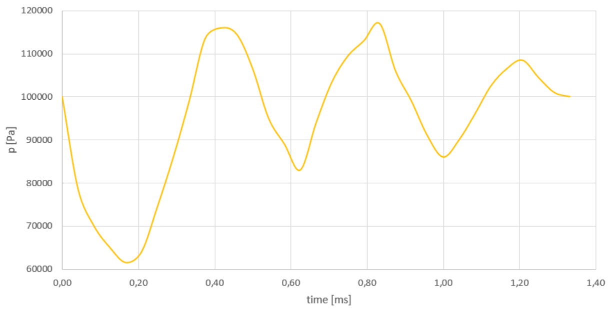

Figure 5. The course of the pressure wave coming to the manifold, caused by the movement of the piston.

The boundary conditions have been set as follows. At the end of the manifold (in the direction of flow, i.e. at the edge of the manifold adjacent to the intake tract) an inlet pressure condition has been entered. The dynamic flow, i.e. time dependent (Transient), has been considered, because of non-stationary flow in a real engine. The course of the inlet pressure wave caused by the movement of the piston from the top dead center (TDC) to the bottom dead center (BDC), from the measurement of the pressures in the intake tract has been obtained (see Figure 5).

Described pressure profile for 720° of crankshafts has been shown. This corresponds to one operating cycle of a four-stroke internal combustion engine. The course of pressures at an engine speed of 4000 $1 \cdot min^{-1}$ has been measured, so one revolution of the crankshaft lasts 15 ms, one degree of rotation of the crankshaft lasts 0.0417 ms. This corresponds to the value of setting of the calculation step (Time step) 140 steps and 300 iterations with a step size of 10^{-5} s.

An atmospheric pressure input condition of 100,000 Pa at the outlet with the surroundings has been set. Also, other boundary conditions as the Walls and Axis type have been defined.

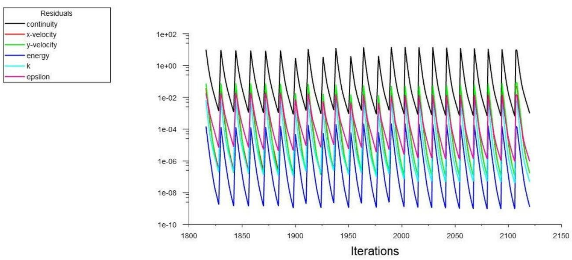

Figure 6. The course of residuals of calculated variables

Figure 6 shows the course of the residuals of part of the calculation. Residues are the differences of individual variables during the calculation between n and n-1 iteration. In this case, the values of pressure, x-velocity components and y-velocity components are monitored (this is a solution of a 2D problem).

4 RESULTS EVALUATION

The air flow through the manifold from the performed calculations has been evaluated, i.e. for each time interval the value of the amount of mass flow through the manifold in $g \cdot s^{-1}$ has been obtained. By multiplying the length of the time interval by the mean value of the given flow, the mass of the flowed air for the given time interval in grams has been obtained.

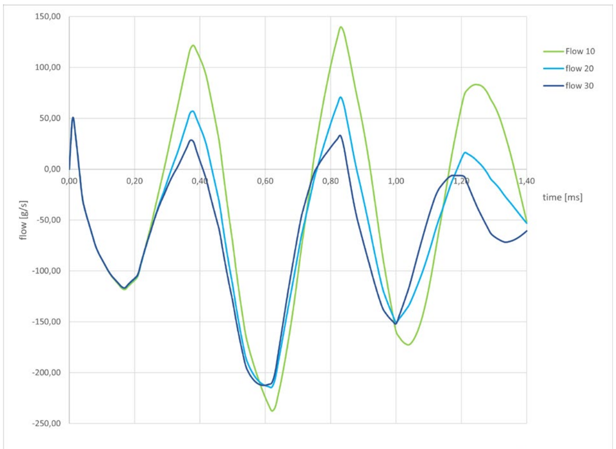

Figure 7. Comparison of the amount of air flowed through the manifold for a different position of the manifold in the airbox

The sum of these masses indicates the total amount of air flowing from the manifold to the engine intake tract over the entire cycle (720 ° of crankshaft rotation = 0.0417 ms). For the model of the manifold with 30 mm distance from the obstacle, a total of 94.9 mg of air flowed through the manifold for one working cycle, at a distance of 20 mm it was 80.5 mg of air and at a distance of 10 mm only 48.3 mg of air.

Figure 7 shows the course of the amount of air flowed through the nozzle. Negative values represent the flow of air from the manifold to the intake tract and to the combustion chamber of the engine. Positive values represent backflow. As expected, most of the air has been distributed through the manifold into the intake tract of the engine for the configuration, when the distance of manifold was 30 mm from the obstacle (wall of the airbox). The closer manifold position to the obstacle, results to the smaller total amount of flowed air. To check the calculation the pressure courses at the outlet of the manifold also have been evaluated. This is a pressure wave which in boundary conditions has been determined. This pressure wave is the same for all models. The evaluation of these data therefore only verified the accuracy of the calculations. The course of the pressures is shown in Figure 8.

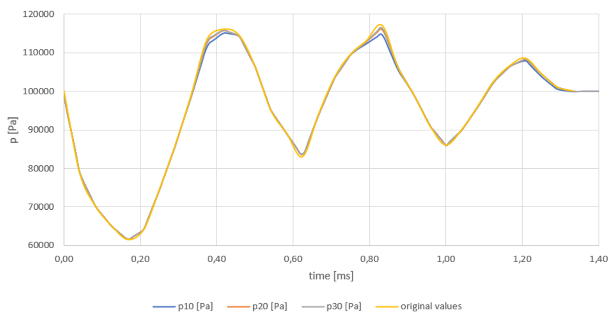

Figure 8. The course of the pressures at the outlet of the manifold

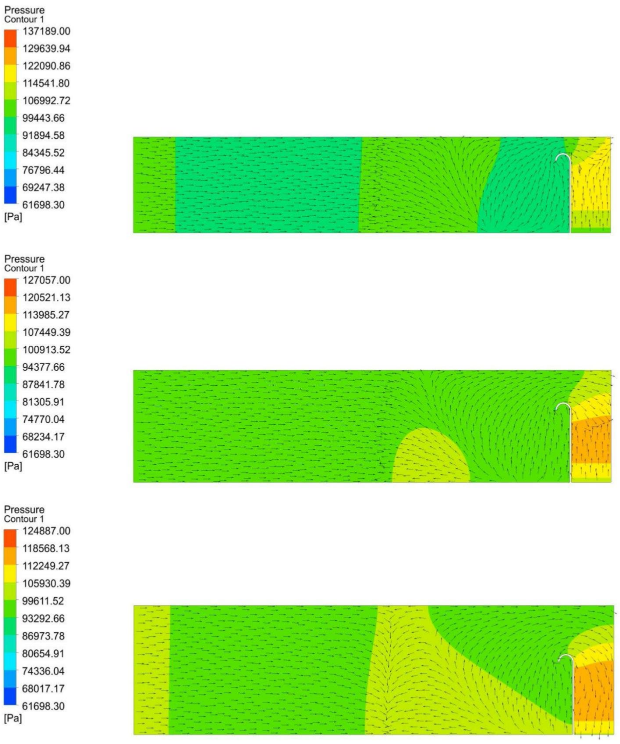

Figure 9. Visualization of the pressure fields at a time of 0.88 ms for the distance of the manifold of 10 mm (above), 20 mm (center) and 30 mm (bottom)

Figures 9 and 10 show the pressure fields at two different times (0.88 ms and 1.18 ms).

At time 0.88 ms there is still backflow for the manifold with 10 mm distance from the wall. For the manifold with the distance of 20 mm distance a vacuum wave is already coming (backflow ends). For the manifold with distance of 30 mm there is already a vacuum and the cylinders are being filled. A similar situation is for the the time of 1.18 ms.

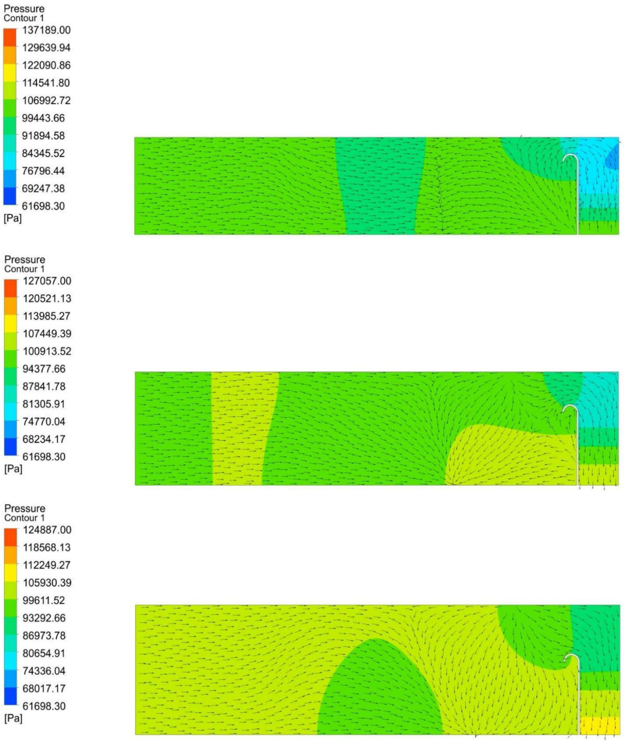

Figure 10. Visualization of the pressure fields at a time of 1.18 ms for the distance of the nozzle of 10 mm (above, 20 mm (center) and 30 mm (bottom)

In Figure 11 with step of 0.1 ms the velocity fields at the arrival of the primary and secondary negative pressure wave from the intake tract for the manifold with distance of 30 mm have been shown (vacuum induced by the movement of the piston from TDC to BDC. It is clear that the highest flow velocity is reached at the rounding of the manifold.

The radius of rounding of the manifold is one of the most important parameters in the design phase of the manifold and its size and shape have an important influence on the amount of intake air.

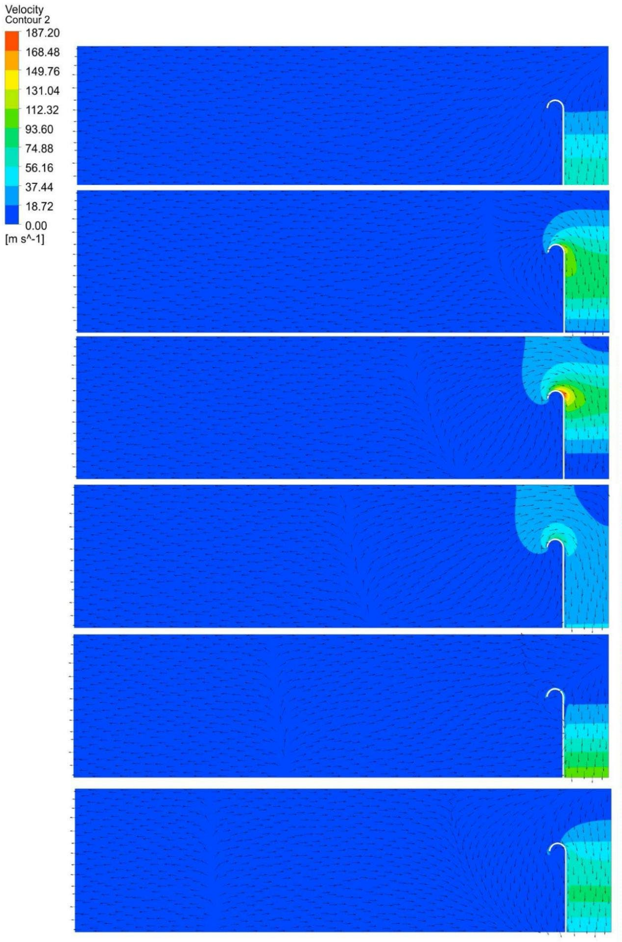

Figure 11. Velocity fields around the manifold in distance of 30 mm from the wall in time (from above), 18 ms, 0.28 ms, 0.38 ms, 0.48 ms, 0.58 ms and 0.68 ms.

5 CONCLUSIONS

In the article have been presented the possibilities of application of computational fluid dynamics (CFD) to modelling of air flow in combustion engine intake manifold depending on airbox configuration. The non-stationary flow occurs in internal combustion engines. This is a specific type of flow characterized by the fact that the variables depend not only on the position but also on the time. The intake manifold dimension and geometry strongly effects intake air amount. The influence of intake manifold position in the airbox to the filling of combustion chamber has been investigated. Furthermore, the effects of different distances between the trumpet neck and the airbox wall in this paper have been compared.

The obtained results can be applied in the design of the intake manifold of an internal combustion engine. The study showed that placing the nozzles too close to the wall of the airbox has a negative effect on the filling of the cylinder, and therefore also on the performance parameters of the engine.

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