Journal of Physics: Conference Series **530** (2014) 012026 doi:10.1088/1742-6596/530/1/012026

Determination of thermodynamic gas parameters in branched pipes in internal combustion engines

W Mitianiec
Cracow University of Technology, Mechanical Faculty
Al. Jana Pawla II 37, 31-864 Krakow, Poland

E-mail: wmitanie@usk.pk.edu.pl

Abstract. The paper presents theoretical and experimental results of calculation of basic gas parameters in the branched pipes. These parameters are required in one-dimensional computer models for prediction of non-steady gas flow in complicated multi-cylinder engine ducts. The gas flow near the junction is described with assumption of compressed and unsteady flow. Mathematical equations describing the gas flow are given in the paper on basis of mass, energy balance in the junction, pressure drop between pipes and conservation of energy in the section of supplied pipe. Equation systems enable to solve values of pressure, gas velocity, sound speed, density and concentration of gas components in every pipe connected to the joint. The different cases of the flow area are considered. The obtained parameters at the junction outflow are needed as initial values for calculation of unsteady gas flow in the outflow pipes. Verification of the method was conducted experimentally and pressure loss coefficients are given in the paper. Additionally by using Fluent program with high mesh density of the T-pipe junction the thermodynamic parameters (pressure, velocity, temperature) are compared with those obtained from 0D model. The model enables calculation the thermodynamic parameters of inflow and outflow systems in multi-cylinder IC engines in computer program.

1. Introduction

Numerical calculation in 0D and 1D models of gas flow in multi-cylinder reciprocating internal combustion engines is widely used for prediction of thermodynamic parameters of gases in the branched ducts of inlet and outlet systems [1]. These ducts are connected to the branch of “Y” and “T” type, where the gas is transferred from supplier pipes to the outflow pipes in varying of changing in the time gas properties. In the engine ducts the gas parameters are a function of temperature and chemical composition. Taking into account the dynamics of the gases with incident and reflected pressure waves the filling and emptying of the cylinder depends on the flow in the branches particularly in multi-cylinder engines. The concept of calculations, based on the balance of mass, energy, and the values of the pressure loss between the pipes oriented relative to each other, has to determine the thermodynamic parameters of gas and the mass fraction of each component in the branch node. Although the model refers to the three branched ducts, and test results are considered to “T” type of branched pipes, one can apply it to the layout consisting of more pipes. This allows a definition of incoming or outgoing pressure waveforms in the ducts from the valves or the ports of individual cylinders in internal combustion engines. Unsteady gas flow in the engine ducts influences on the mass filling ratio of the gas in each cylinder. In the simulation of any engine, where its performance is related to the mass trapped in the cylinder and pressure wave motion through the branched pipes was considered by Blair [2]. The incident and reflected pressure waves define gas

Journal of Physics: Conference Series **530** (2014) 012026 doi:10.1088/1742-6596/530/1/012026

velocity, density and mass flow rate. The non-isentropic analysis is required for determination all gas parameters in the branched pipes where unsteady gas flow takes place for computation of the intake or outflow engine systems. Calculation of the gas flow in the parallel branched pipes with different resistance was considered by Yoo *at al* [3]with taking into account κ-ε turbulent model.

2. Model Flow 3-pipe branch

The mass rate of gas flowing from the individual pipes to the node in the branched ducts depends on the different pressure in pipes and the node. Schematic of a typical combination of three pipes located relative to each other by an angle θ is shown in figure 1. It was assumed that the gas flow is unsteady and non-isentropic resulting from the existence of friction (pressure loss) in the node and the pipes. A positive direction of gas flow was assumed where the gas is directed towards the node A, which corresponds to the gas pressure wave. It is known the flow direction and gas parameters in the cross-section S in the duct 3, which is located Δx away from section 3. These parameters are also known for the remaining two pipes, and they are as a result of solution of wave motion of the gas in the pipes. For this reason, a positive pressure wave amplitude P_R is known in every branch, which has moved to the node. Most of the duct connections in combustion engines have "T" shape form considered in the work and it is shown in figure 2. We should determine the parameters of gas in sections 1, 2 and 3 for different gas flow cases.

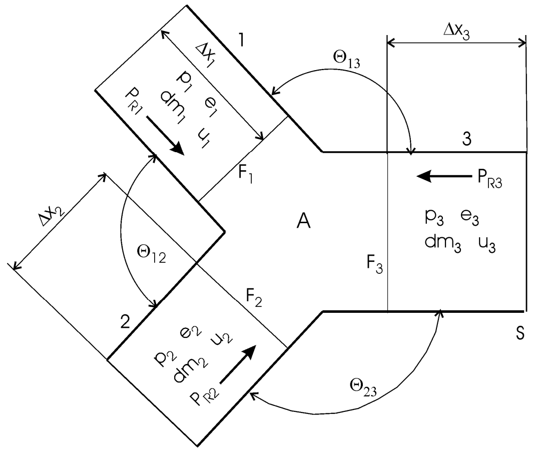

Figure 1. Overall diagram of mass flow of gas through 3 branched pipes.

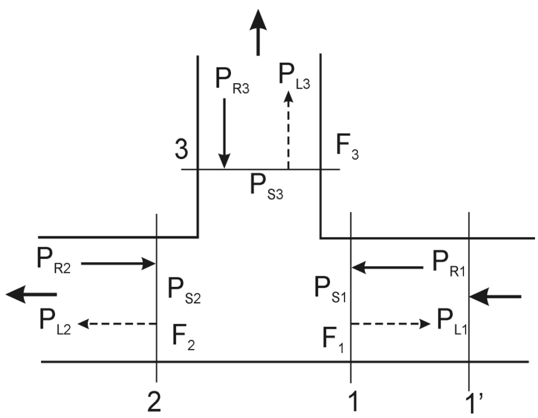

Figure 2. Diagram of pressure impulse of gas in 3 branched pipes of type "T".

The gas in the pipes of internal combustion engines consists of different chemical composition. The gas composition is determined by mass ratio of the individual species and their individual gas constant *R*. In each branch the gas flows into the node or flows out from the node with charge mass *dm*, velocity u_j, pressure p_j and specific energy e_j. The incident and reflected pressure waves and other gas parameters at unsteady flow in the pipes can be solved by using different calculation techniques for example: Rieman, QUB, Lax-Wendroff or Harten-Lax-Leer methods [1]. The relative gas pressure *P* at each point of the system is defined by formula:

$$P = \left({p}/{p_0} \right)^{\frac{\kappa-1}{2\kappa}} \tag{1}$$

where p — absolute pressure in the system, p_0 — reference pressure and κ — ratio of the specific heats at a given temperature *T*.

The pressure wave arising at the end of each duct P_S is a function of the impulse of wave in negative direction P_L and the impulse of wave in positive direction P_R derived by Benson [4]:

$$P_{Sj} = P_{Lj} + P_{Rj} - 1 \tag{2}$$

wherein *j* is the sequence number of the pipes.

In contrast, the gas particle velocity reaching the node A is:

$$u_j = c_j \cdot a_{0j} \cdot (P_{Lj} - P_{Rj}) \tag{3}$$

where $a_{0j} = \sqrt{\kappa R T_{0j}}$ is the local speed of sound at reference conditions and variable c_j is a function of the specific heats ratio and calculated from the following dependence:

$$c_j = \frac{2 \cdot \kappa_j}{\kappa_j - 1} \tag{4}$$

For gas flow in the ducts of internal combustion engines it can be assumed that for every branched pipe near the node reference temperature T_{0j} is the same ($T_0 \approx 298K$).

3. The mathematical model of the flow

In many previous methods there assumed that the node A (figure 1) the thermodynamic state and gas dynamic state is the same for all waves running across the pipes, and the pressure in the pipes near the node is the same [2,4].

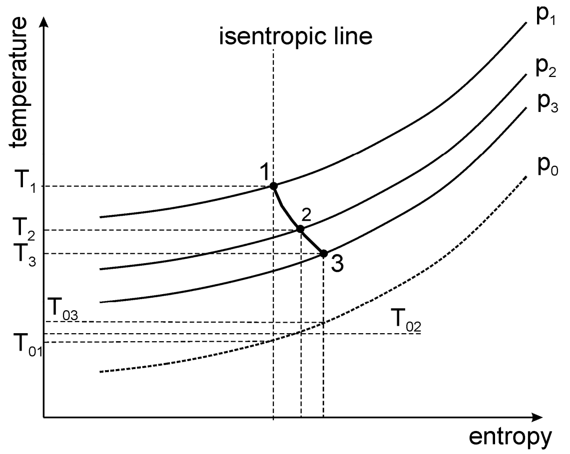

Figure 3. Non-isentropic gas flow from one supplying pipe to two receiving ducts.

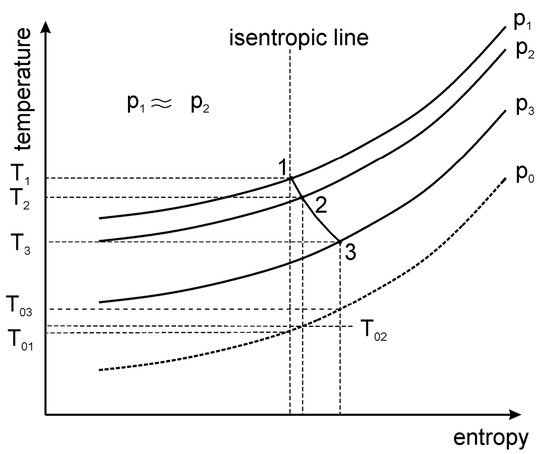

Figure 4. Non-isentropic gas flow from two supplying pipes to one receiving duct.

As a result of experimental studies conducted by Bingham [5], and the author [6,7] the mass flow from one pipe to the other depends on the pressure drop between the ducts. The actual flow in the branch is non-isentropic as shown in figure 3 at a flow from one pipe to the other two, and in figure 4, when two pipes feeding the one pipe. At the isentropic flow the quantities T_{01}, T_{02} and T_{03} are respectively the reference temperatures at pressure which is usually the ambient pressure p_0. When one pipes feeds the other two the difference between pressure p_2 and p_3 pressure is small (figure 3). Therefore, it can be considered that the reference temperature T_{02} and T_{03} are equal (figure 4). When the gas is supplying a single pipe, for example the pipe 3 with two pipes 1 and 2, the pressure in the pipes 1 and 2 differ slightly ($p_1 \approx p_2$), which also means that $T_{01} \approx T_{02}$ (figure 4). The pressure drop between the pipe 1 and the other pipes k is expressed by the given equations:

$$p_0 \cdot \left((P_{S1})^{1/c_1} - (P_{Sk})^{1/c_k} \right) = C_{L1k} \rho_k u_k^2 \tag{5}$$

where the loss factor determined by the flow [5] amounts:

$$C_L = 1.6 - \frac{1.6\Theta}{167} \tag{6}$$

The flow loss coefficient C_L in equation (6) for an angle $\Theta > 167°$ between two ducts is set to zero. For an angle of 90° the flow loss coefficient is 0.737. The assumption of pressure drop eliminates an implementation of the complex system of momentum conservation equations to the calculations.

In order to find the amplitude of the pressure wave reflected from the node in each branched pipe the law of conservation of mass, law of conservation of energy and pressure drop equation (4) between the pipes are applied. At the branching occurring in the intake system one can be omitted a consideration of change of residual exhaust gas ratio γ, gas constant R and the ratio of specific heats κ. The gas composition and thermodynamic parameters of the gas in the intake system are almost identical along the pipes. For the conservation of flow continuity the following condition must be met:

$$\sum_{j=1}^{k} \dot{m}_j = 0 \tag{7}$$

Gas pressure at the ends of the pipes near the node is determined from the formulas (4). For the three branching pipes, there are two equations of the pressure drop. In contrast, the energy flow equation for the three pipes can be written as follows:

$$\dot{m}_1 \cdot e_{01} + \dot{m}_2 \cdot e_{02} + \dot{m}_3 \cdot e_{03} = 0 \tag{8}$$

where e_{01}, e_{02} and e_{03} are the accumulation energies of fluid (gas enthalpy increased by the specific kinetic energy). This corresponds to a gas state on the line of the pressure p_0 in Figure 3 and 4. The accumulation energy of gas can be written as follows:

$$e_{0j} = \left(c_p\right)_j \cdot T_j + \frac{u_j^2}{2} = \frac{\kappa_j}{\kappa_j - 1} \cdot R_j \cdot T_j + \frac{u_j^2}{2} = \frac{1}{2} \cdot \left(c_j \cdot a_j^2 + u_j^2\right) \tag{9}$$

where a_j — gas sound speed.

Speed of sound a_j can be expressed using the pressure wave resulting P_{Sj} as:

$$a_j = a_{0j} \cdot P_{Sj} = a_{0j} \cdot \left(P_{Lj} + P_{Rj} - 1\right) \tag{10}$$

After substitution of equations (3) and (10) to (9) the accumulated energy is a function of reflected and positive (incidence) pressure waves:

$$e_{0j} = c_j a_{0j}^2 \left[\left(P_{Lj} + P_{Rj} - 1\right)^2 + c_j \left(P_{Lj} - P_{Rj}\right)^2\right] \tag{11}$$

The density of the gaseous medium obtained from gaseous state and sound speed eq. takes the form:

$$\rho_j = \frac{\kappa_j \cdot p_0}{a_{0j}^2} \left(P_{Lj} + P_{Rj} - 1\right)^{\frac{2}{\kappa_j - 1}} \tag{12}$$

The flow rate in the duct j with cross section area F_j can be expressed by the equations (3) and (9) as:

$$\dot{m}_j = \frac{2\kappa_j^2}{\kappa_j - 1} \cdot \frac{p_0}{a_{0j}} \left(P_{Lj} + P_{Rj} - 1\right)^{\frac{2}{\kappa_j - 1}} \cdot \left(P_{Lj} - P_{Rj}\right) \cdot F_j \tag{13}$$

Finally, one can write the appropriate systems of equations for the two types of gas flow:
a) gas flow from the first to the second and third duct,
b) gas flow from the first and second duct to the third duct.
Due to the similar arrangement of the equation there are given only equations for the first case:
1. Pressure drops between the pipes 1-2, 1-3 from eq. (5):
2. The equation of mass conservation in the node:

$$\sum_{j=1}^{3} \frac{2\kappa_j}{\kappa_j - 1} \cdot \frac{p_0}{a_{0j}} \left(P_{Lj} - P_{Rj}\right) \cdot \left(P_{Lj} + P_{Rj} - 1\right)^{\frac{2}{\kappa - 1}} \cdot F_j = 0 \tag{14}$$

3 The equation of energy conservation in the node as a function of known values of positive pressure waves P_{R1}, P_{R2}, P_{R3} and additionally relative speed sound a_{01} which is based on equations 11 and 13:

$$\sum_{j=1}^{3} m_j e_{0j} = f\left(P_{R1}, P_{R2}, P_{R3}, a_{01}\right) \tag{15}$$

4. The equation of conservation of energy in the supply section 1 – 1':

$$\frac{a_{1'}^2}{\kappa - 1} + \frac{\left(\rho u^2\right)_{1'}}{2} = \frac{a_1^2}{\kappa - 1} + \frac{\left(\rho u^2\right)_1}{2} \tag{16}$$

For disposal there are five equations for each case and five unknown values: three pulses of the reflected wave P_{Lj} and two reference sound speeds: a_{02} and a_{03}. This nonlinear equations system can be solved for example by using the Newton-Raphson method.
Below there is presented one example of the air flow in the branched ducts as shown in figure 2 with the basic data for three different configurations of the branched ducts:
- cross section areas of the ducts $F_1 = 4\ \text{cm}^2$, $F_2 = 3\ \text{cm}^2$, $F_3 = 1{,}5\ \text{cm}^2$
- positive pressure impulses: $P_{R1} = 1{,}06$, $P_{R2} = 0{,}98$, $P_{R3} = 0{,}98$.

The supplier pipe was the pipe No 1 and two others (No 2 and 3) were being supplied by the branch node. As a result of solution of the equation system (5, 14 – 16) the required gas parameters were obtained in the pipes near the branch node for the given data and the results are presented in table 1.

Table 1. Calculation results of gas flow in branched ducts (figure 2)

θ_{12} °	$\theta_{2:}$	θ_{23} [-]	P_{S2} [-]	P_{S3} [-]	u_1 m/s	u_2 m/s	u_3 m/s	$\dot{m}_1$ kg/s	$\dot{m}_2$ kg/s	$\dot{m}_3$ kg/s	a_{01} m/s
120	120	1,043	1,0321	1,0321	126,8	-120	-119,8	0.0775	-0,0511	-0,0264	340,4
90	150	1,045	1,0293	1,0341	122,6	-115,7	-123,6	0.0758	-0.0483	-0.0275	341,5
180	90	1,039	1,0385	1,0245	133,9	-129,8	-106,6	0,0802	-0,0576	-0,0226	338,5

The reference acoustic velocities in the pipes being supplied have almost the same value because of a small difference of gas temperature and density.

4. The spatial gas flow in the three branched ducts

The presented model of gas flow through the branching pipes refers to unsteady flow, but determination of the gas parameters is held on an assumption of stationary gas motion in a very small period of time. In this paper the analysis of the gaseous medium is limited to the branching "T“ ”ype, in which the theoretical flow loss coefficients calculated according to the equation (5) are respectively $C_{L12} = 0$, $C_{L13} = 0.738$ and $C_{L23} = 0.738$. In order to determine the nature of the flow the spatial gas flow model was created using the finite element method in the package CFD software ANSYS FLUENT Ver.13. The geometrical model was created in CAD system and next it was translated to the step model. Such universal CAD model was imported to ICEM CFD preprocessor, where the mesh was created with definition of surface and volume zones.

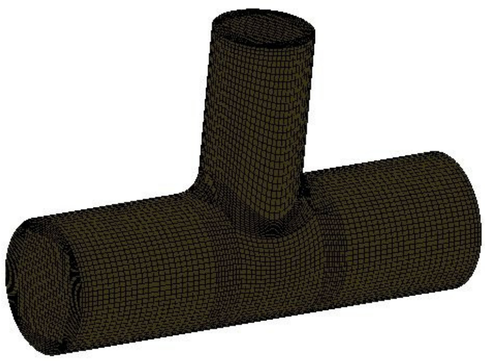

Figure 5. Mesh model of “T” pipes branching.

The grid of model shown in figure 5 contains 134425 hexahedral cells and 141000 nodes with high mesh density near the walls and in the pipes connection due to the turbulence flow. The total length of the model in both directions was 0.1 m and diameter of the main pipe was 4 cm and 3 cm of the perpendicular pipe. For simplicity, it was assumed that the flowing gas is the air treated as an ideal gas with RNG k-ε turbulence model. Elongation of the pipes did not influence on the calculation results. The air flow was analyzed at assumption of supplying of one branching pipe (2) by two pipes (1 and 3), wherein the inlet and outlet boundary parameters were as follows at reference pressure 1.013 Mpa:

inlet 1 : $p_1 = 6800$ Pa, $u_1 = 80$ m/s, T_1=500 K,
inlet 3 : $p_3 = 8000$ Pa, $u_3 = 40$ m/s, T_3=450 K
outlet 2: $p_2 = 0$ Pa.

Formulations of the boundary conditions of the gas flow in the branched pipes were considered by Chae *et al* [9]. Waveforms of pressure, velocity and density of the gas obtained from solution of the spatial flow model in the x-z plane cross section are shown in figures 6 – 9, respectively. The purpose of the analysis of this model was to determine the gas pressure difference across sections near the node, and checking the validity of the formulas (5). The analysis confirmed the assumption that in the case of gas flow from two ducts to a single duct, the static pressure (figure 5) is almost equal (the supply lines in figure 4).

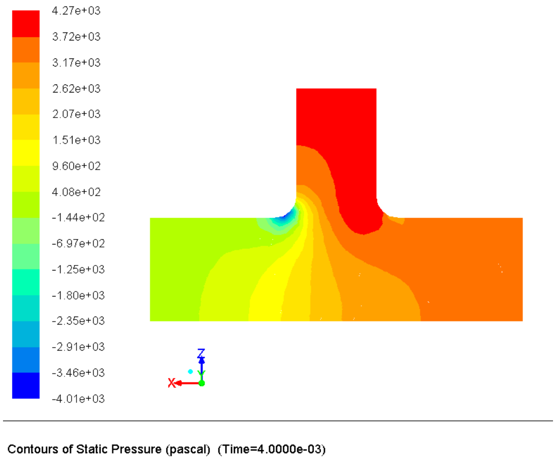

Figure 6. Static pressure in "T" branch type of 3D model.

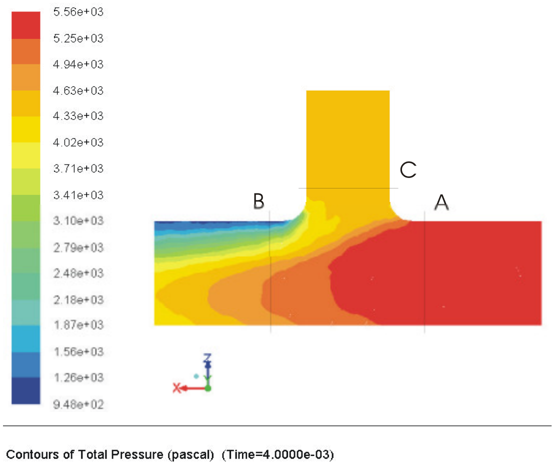

Figure 7. Total pressure in "T" branch type of 3D model.

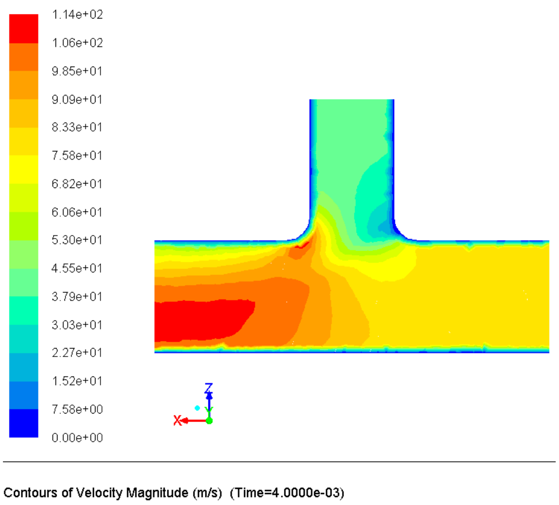

Figure 8. Contours of velocity magnitude (m/s) in x-z cross section.

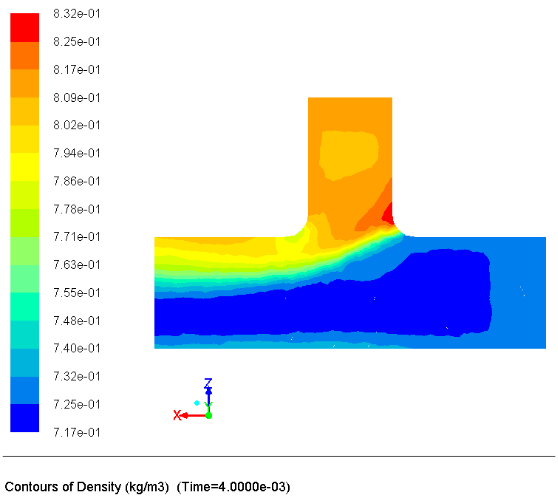

Figure 9. Contours of gas density at x-z cross section

During unsteady air flow high differences of velocity and density occur (figure 8 and 9). Differences in the density of the gas (figure 9) and specific energy (figure 11) have the rapid change of values in the node as a result of different temperature of the medium in inlet 1 and 3 (figure 10). The static temperature of the gas (figure 10) and total specific energy (figure 11) have the same distribution profile. In the numerical calculation of transient motion of the gas one assumes the average length of the cells equal to 2 cm, and therefore for benchmarking the distance 2 cm from the node was indicated in figure 7 by the letters A, B and C.

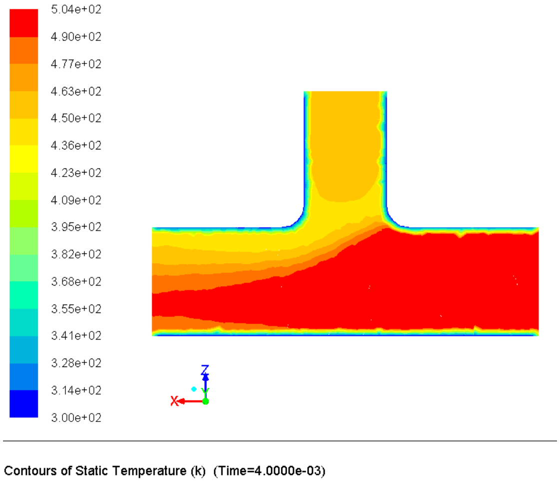

Figure 10. Static temperature (K) at x-z cross section.

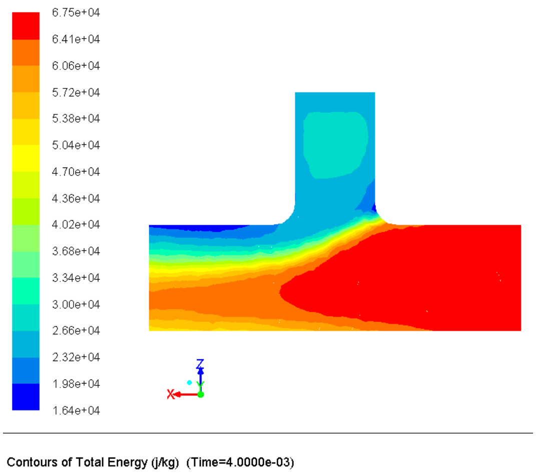

Figure 11. Contours of total specific energy (J/kg) in x-z cross section.

The table 2 shows the average pressure, velocity and density in these conductors. The loss factors determined on the spatial air motion analysis deviate significantly from the coefficients determined by the formula (6) because of non-constant thermodynamic parameters in the sections A, B and C obtained from simulation.

Table 2. Gas parameters in „T" branch of pipes

Parameter (mean values)	A	B	C
static pressure p [Pa]	3650	0	4250
density ρ [kg/m^3]	0.73	0.75	0.81
velocity u [m/s]	75,0	97.3	40
Loss coefficient in p. B according to (5)	C_{L12}=0,514		C_{L32}=0,598

5. Experimental determination of pressure losses

Pressure drops in the junction depend on the angles between the pipes and direction of gas flow. The experimental work has been carried out with the use of air in order to determine the loss coefficients C_L for the various directions of gas motion. Air velocity and pressure measurement have been conducted on the volume intake system with "T" branched pipes in steady conditions by using of thermo-anemometers and pressure gauges. Variation of pressure drop between pipes 1 and 2 ($\theta_{12} = 180°$) with the air inflow to the cylinder is presented in the figure 12. The pressure drop is the function of dynamic pressure of gas in the duct being supplied. The pressure difference between lines 1 and 2 is linearly dependent on the total value ρu^2. The existence of a pressure difference has been confirmed experimentally for the same gas flow direction where C_L≈0.6. The pressure drop between pipes 3 and 2 (flow from the duct 3 to 2) is also expressed as a linear function of the parameter ρu^2 (figure 13) and the pressure loss coefficient is 0.64 and according to the Blair formula [2] this factor should be 0.737 and is almost close to the value achieved experimentally. The experiments done by Daisuke et al [8] enabled to define the pressure loss coefficient in a polynomial function of the dynamic pressure $\rho u^2/2$. At the air flow from the cylinder connected with the duct 2 to the ducts 1 and 3 with the open ends to the atmosphere, the pressure loss coefficient between the duct 2 and 1 was determined with value C_{L21}≈ 0.598. The pressure loss coefficient C_{L23} between the pipe 2 and 3 in reverse flow direction is about 0.64 and these values differ to the value determined in [5] about 13%.

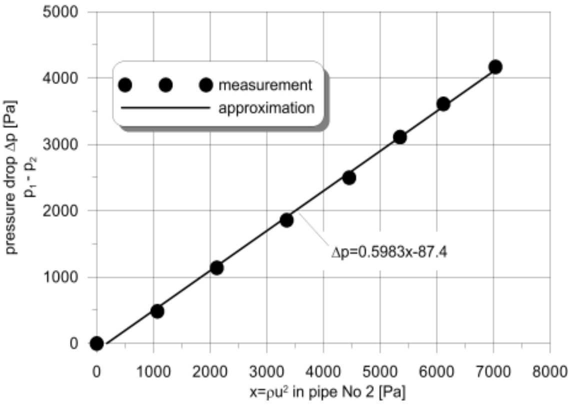

Figure 12. Air pressure drop between pipe No 1 and 2 (pipes No 1 and 3 open).

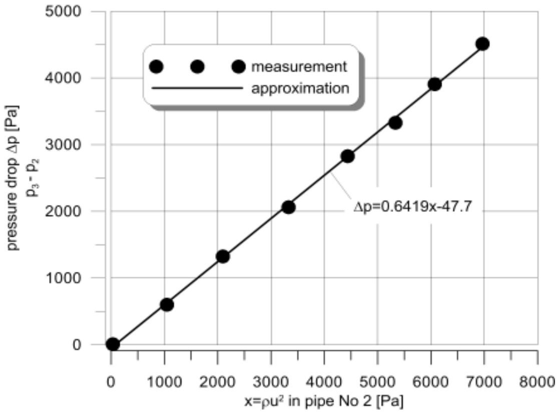

Figure 13. Air pressure drop between pipe No 3 and 2 (flow into pipe No 2).

6. Conclusions

The given mathematical model and carried out the analysis of spatial motion of gas in the manifold was developed independently by the author on the basis of the work [2].

1. During charge flow through manifolds the unequal pressures in the pipes reaching the node takes place. Two pipes supplying the gas into the branched node have the same pressure. The mathematical model enables gas flow calculation in the branched pipes in the combustion engines.
2 Solution of the equations (5, 14-16) and knowledge of the nature of the flow due to the pressure loss coefficient allows a determination the charge thermodynamic parameters in individual ducts. The use of CFD method to analyze the gas flow through the branch "T" of three ducts confirmed the assumption of the analytical method of one-dimensional gas motion but with correction of flow coefficient.
3 If the flow from two pipes into the third pipe takes place at the same time, the pressure in the supplying pipes is almost identical. In the case of gas flow from one pipe to two pipes the pressures in the pipes being supplied are different.
4. The pressure losses obtained from the experiments and simulations between pipes set at angle 180° are not equal zero as it comes from formula (6). Further experimental test are foreseen for modification of the formula on the coefficient of pressure losses (6).

7. References

[1] Chen C and Veshagh A 1992 A comparison between alternative methods for gas flow and performance prediction of internal combustion engines *SAE 921734* Warrendale
[2] Blair G 1999 Design and simulation of four stroke engines *SAE R-161* Warrendale
[3] Yoo G Choi H and Kim C 2012 Characteristics of turbulent flow distribution in branch piping system DOI: 10.1007/s11771-012-1397-3 *Springer-Verlag* Heidelberg
[4] Benson R 1982 The thermodynamics and gas dynamics of internal combustion engines **Vol. 1** and **2** Clarendon Press Oxford
[5] Bingham J and Blair G 1985 An improved branched pipe model for multi-cylinder automotive engine calculations *Proc IMechE* **Vol 199** No D1 Belfast 65-77
[6] Mitianiec W 1999 Improvements of crankcase delivery ratio in two-stroke engines with volume intake system *SAE 1999-01-1334* Warrendale
[7] Mitianiec W 1994 Zawory membranowe i układy pojemnościowe Ossolineum Kraków
[8] Daisuke S Chongho Y and Toshiharu K 2013 Pressure Change in tee branch pipe in oscillatory flow *Hindavi Pub Cor AdvMechEng* **Vol 2013** ID 257283
[9] Chae K S Lee K T Hwang C J and Lee D J 2006 Formulation and validation of boundary conditions at a branched junction for nonlinear waves *J. of Sound and Vib.* **2006|295**|1-2|13-27