VILNIAUS GEDIMINO TECHNIKOS UNIVERSITETAS

TRANSPORTO INŽINERIJOS FAKULTETAS

TRANSPORTO TECHNOLOGINIŲ ĮRENGINIŲ KATEDRA

Linas Adomavičius

DVITAKČIO VARIKLIO HIDRODINAMINIAI TYRIMAI
HYDRODYNAMICAL INVESTIGATION OF THE TWO-STROKE ENGINE

Baigiamasis magistro darbas

Transporto inžinerijos studijų programa, valstybinis kodas 62403T104

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Transporto inžinerijos studijų programos baigiamasis magistro darbas

Pavadinimas **Dvitakčio variklio hidrodinaminiai tyrimai**

Autorius **Linas Adomavičius** Vadovas prof. Marijonas Bogdevičius

Kalba

lietuvių

X užsienio

Anotacija

Magistriniame darbe pristatoma dvitakčio variklio modeliavimo programa, kuri apskaičiuoja dujų tekėjimą dvitakčiame variklyje. Programa sukurta naudojantis programiniu paketu „Fortran“ ir pateikta antrame priede.

Programa modeliuoja dujų tekėjimą dvitakčiame variklyje ir pateikia slėgio bei debito rezultatus. Modeliavimui taikytas koncentruotų tūrių metodas. Slėgiai variklio dalyse yra aprašomi slėgio kitimo formule.

Programa skirta lenktyniniams dvitakčiams varikliams, jų galiai didinti. Darbe labai smulkiai analizuojami mažo darbinio tūrio varikliai, kurie naudojami automobilių modelių greičio varžybose.

Naudojant programą įmanoma atlikti variklio įsiurbimo ir prapūtimo fazių optimizavimą. Optimizavimo rezultatai rodo, kad galima pagerinti cilindro užpildymą šviežiu oro-kuro mišiniu net 0,5 % - tai yra labai geras rezultatas, kadangi dvitakčiai varikliai yra labai išvystyti.

Pirmo laipsnio diferencialinėms lygtims spręsti panaudotas skaitinis Rungės-Kuto IV metodas.

Prasminiai žodžiai: dvitaktis variklis, dujų tekėjimas, masės debitas, galia, išmetimas, prapūtimas, fazės, kanalas, Rungės-Kuto metodas, Fortranas, slėgis, įsiurbimas, vožtuvas, alkūninis velenas, karteris, programa, galios vamzdis, forsavimas, optimizavimas.

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Transport Engineering study programme master thesis

Title: Hydrodynamical investigation of the two-stroke engine

Author: Linas Adomavičius Academic supervisor: prof. Marijonas Bogdevičius

Thesis language

Lithuanian

X Foreign

Annotation

Inside the master thesis there is represented computer simulation program of the gas flow through the two-stroke engine. The simulation program is developed in software package "Fortran" and is represented as Annex 2.

The program simulates gas flow through the two-stroke engine and gives results of pressure changes and mass flow. The simulation is done by using concentrated volume method. The pressure inside the engine parts is described by pressure changes equation.

The program is intended for the two-stroke engine tuning for the race. In the thesis is deeply investigated small swept volume engine used for tether model car race.

Using the program is possible to do engine inlet and transfer timing optimization. The optimization results were very successful and showed that the cylinder filling with fresh air fuel mixture can be increased by 0.5%.

For the numerical solution of the first order differential equations there were used Runge-Kutta IV method.

Keywords: two-stroke engine, gas flow, mass flow, power, exhaust, transfer, timing, port, duct, Runge-Kutta method, Fortran, pressure, inlet, valve, crankshaft, crankcase, cylinder, scavenging, program, power pipe, tuning, RPM, BDC, TDC, optimization.

CONTENT

ANNEX 1 (5 PAGES)

GAS FLOW THROUGH THE TWO-STROKE ENGINE.

PUBLICATION FROM YOUTH SCIENTIST CONFERENCE “SCIENCE – FUTURE OF LITHUANIA. TARNSPORT” (MOKSLAS – LIETUVOS ATEITIS. TRANSPORTAS) 2010-05-06

ANNEX 2 (31 PAGES)

COMPUTER PROGRAM “GAS FLOW SIMULATION IN THE TWO-STROKE ENGINE” MADE IN “FORTRAN”

LIST OF FIGURES AND TABLES

1. INTRODUCTION

The aim of this thesis is to represent the computer program which calculates the gas flow through the two-stroke internal combustion engine. During my master studies I learned a little bit of gas dynamics and here my theoretical knowledge is applied for the practice. I chose the two-stroke engine because I have very big experience in tuning them for the tether model car race. After more than ten year of experience I finally understood the main idea of the engine work, which is so simple, that mostly nobody pays attention to it. In general any internal combustion engine is the unit which converts the fuel internal energy (burning process) to the mechanical energy (power output through the crankshaft). There is two ways to increase the engine power output: to increase efficiency of fuel burning or to decrease all mechanical losses. The first way will be discussed in this thesis. There will be no discussion about mechanical design of the engines.

1.1 Review of the two-stroke engines application

The two-stroke internal combustion engine was invented by Sir Dugald Clerk in England at the end of the 19^{th} Century. At these times they were used in tricycles, motorcycles and cars. Two-stroke engines can be grouped in several groups according application area. Mainly engines are grouped by similarity of engine design and are represented in Table 1.1.1.

Table 1.1.1 Two-stroke engines groups

Engines group	*Application area*	*Fuel type*	*Swept volume*
Model engines	Engines of car, boat, plane models	methanol	0.1…50 cc
Small engines	Engines of chainsaws, trimmers, small portable and industrial devices	gasoline	20…1000 cc
	Engines of motorcycles, scooters, go-karts, boats (outboard)	gasoline	50…1000 cc
Car engines	Engines of cars	gasoline	>125 cc
Truck engines	Engines of trucks	diesel	>1000 cc
Marine engines	Engines of marines	diesel	>1 m^3

First two-stroke engines for the models were produced between First and Second World War in the USA. They use mainly methanol fuel and have combustion ignition (diesel type engines) and are of very small swept volume (from 0.1 cc till 50 cc). Today they have about 90% of the model engine market. It is because of the few reasons: they are simple, cheap, small and extremely powerful. At the moment it looks like there will be no changes in the model engines market.

Small engines group has two main application areas: for the devices used in work and for various transport means. The two-stroke engine is very applicable for hand devices such as chainsaw or trimmer, because it is lightweight, very powerful and it can work in any position (bottom up). The main reason is for using two-stroke engine in small engine group that the two-stroke engine is much pore powerful than the four-stroke engine and comparatively small. The use of two-stroke engines are decreasing year by year in motorcycles, scooters, the reason is legislative law concerning pollution. The two-stroke engine with all the pluses has one big minus today - it pollutes environment much more than the four-stroke engine. So two-stroke engines are disappearing in the markets which are regulated by environmental law.

The two-stroke engines were used in cars too. Today it is history, but the last best known example for us is “Wartburg” from VDR (East Germany). “SAAB” powered by two-stroke engine won Monte-Carlo rally many years ago. Today all cars use four stroke engines because they are more reliable, fuel-efficiently, less noisy and less pollutes environment.

The two-stroke diesel engines were used for trucks and were quite popular even in mass production 30-40 years ago. Today there is no mass production of two-stroke diesel engines for trucks.

Marine engines are the most successful application of two-stroke diesel engine. Marine engines are extremely big: about 12 meters tall and 50 meters long, stroke of such enormous engine’s cylinder is about 1.8 meter, bore 1 meter. Such cylinder produces more than 4000 hp and usually it rotates about 60-100 rpm. Thermal efficiency of marine engine is more than 50%. They are most efficient ever made engines.

1.2 What will be discussed in thesis

Computer program will be represented in this thesis which calculates parameters of gas flow through the two-stroke engine. I chosen to simulate model engine of 2.5 cc swept volume. Such small engine is used in tether model car race. I have big practice with tuning of such kind engines and that is the main reason why it was chosen such engine. Moreover you can easily notice that all theory and calculation schemes applied in this work can be used for any small two-stroke engine. The theory for each two-stroke engine is the same.

The program is intended for two-stroke engine tuning – to get more power output from it. This is because I will use calculated data in engine tuning for the race.

Inside the program there is simulated disc type inlet valve, transfer and exhaust valves controlled by piston.

The program gives output of mass-flow and pressure changes in crankcase, transfer ports, cylinder, and exhaust pipe. Calculations are based on concentrated volume method. In general it is quite simple method and not the most suitable for such type of calculations. I chosen it because during my master studies I learned it quite well, better methods which includes wave propagation are very sophisticated and they requires very deep knowledge of gas dynamics, higher mathematics, numerical solutions, programming. The program does not take into account burning process, because of lack of my knowledge. This thesis is my first step to real scientific calculations. All the more sophisticated methods usually are used in doctoral thesis.

The concentrated volume method is based on the first order differential equation. I described engine by nine differential equations. The problem is to know how to solve them. I learned simple numerical solution method Runge-Kutta IV for this reason. But it is not enough, because someone has to solve it, by hand it is impossible, I made program with Fortran 6.0. The program works very well. It is possible to do optimization of the engine by changing engine parameters in it. Also program can be applied for any two stroke engine by changing engine parameters inside the program.

2. THEORY OF THE TWO-STROKE ENGINE

In this chapter there is represented method of operation of the two-stroke engine, various designs of engines, introduction to exhaust power system. All discussion relates only small engines in this chapter. In reality there are more various designs of engines or engines which operate with some difference than it is described in this chapter. It is very important to understand operation principle of the two-stroke engine before going further to calculation.

2.1 Operation method of the two-stroke engine

There are represented main stages of two stroke engine in the figure 2.1.1. The picture is taken from the book "Design and Simulation of Two-Stroke Engine" Blair (1996).

There is shown compression and induction stage in the figure 2.1.1 (A). At this stage piston goes to top dead center (TDC). When piston goes to TDC the crankcase volume increases and creates lower pressure than atmospheric inside. Inlet port opens (inlet port is controlled by piston according to picture) and fresh air-fuel mixture flows inside the crankcase because of pressure difference (pressure in crankcase is smaller than atmospheric) – this is called induction. At the same time when exhaust port is closed and piston moves toward TDC in cylinder starts compression. When TDC is reached the ignition sparks air-fuel mixture in cylinder and burning starts. The pressure increases and pushes down the piston toward bottom dead center (BDC). In Fig.2.1.1 (B) we see that the piston goes down (after TDC) and opens exhaust port and at the same time closes inlet port. The process when piston goes down and opens exhaust port and at the same time transfer port is closed is called blowdown process. There is very important to design engine so, that during blowdown process there will be removed as much as possible already burned air-fuel mixture which is exhaust gas after burning process. After blowdown process we have few processes going at the same time. During all the time when piston was going to BDC the pressure was increasing in crankcase. When opens transfer ports all the fresh air-fuel mixture goes to cylinder through transfer ports figure 2.1.1 (C). At this stage in cylinder we have that fresh air-fuel mixture enters cylinder and at the same time exhaust port is still open. Because of this fact we have that some fresh air-fuel mixture goes straight through exhaust port to exhaust pipe – this process is called "short-circuiting". The process when fresh air-fuel mixture enters cylinder is called scavenging. Scavenging occurs only as long as air-fuel mixture flows out of crankcase to cylinder (from opening of transfer ports till BDC). As you can guess scavenging is the most important process concerning efficiency of the two-stroke engine. Depending of how efficient scavenging is depends how efficient fresh air-fuel mixture is transferred to cylinder. The “short-circuiting” is depending also on scavenging design.

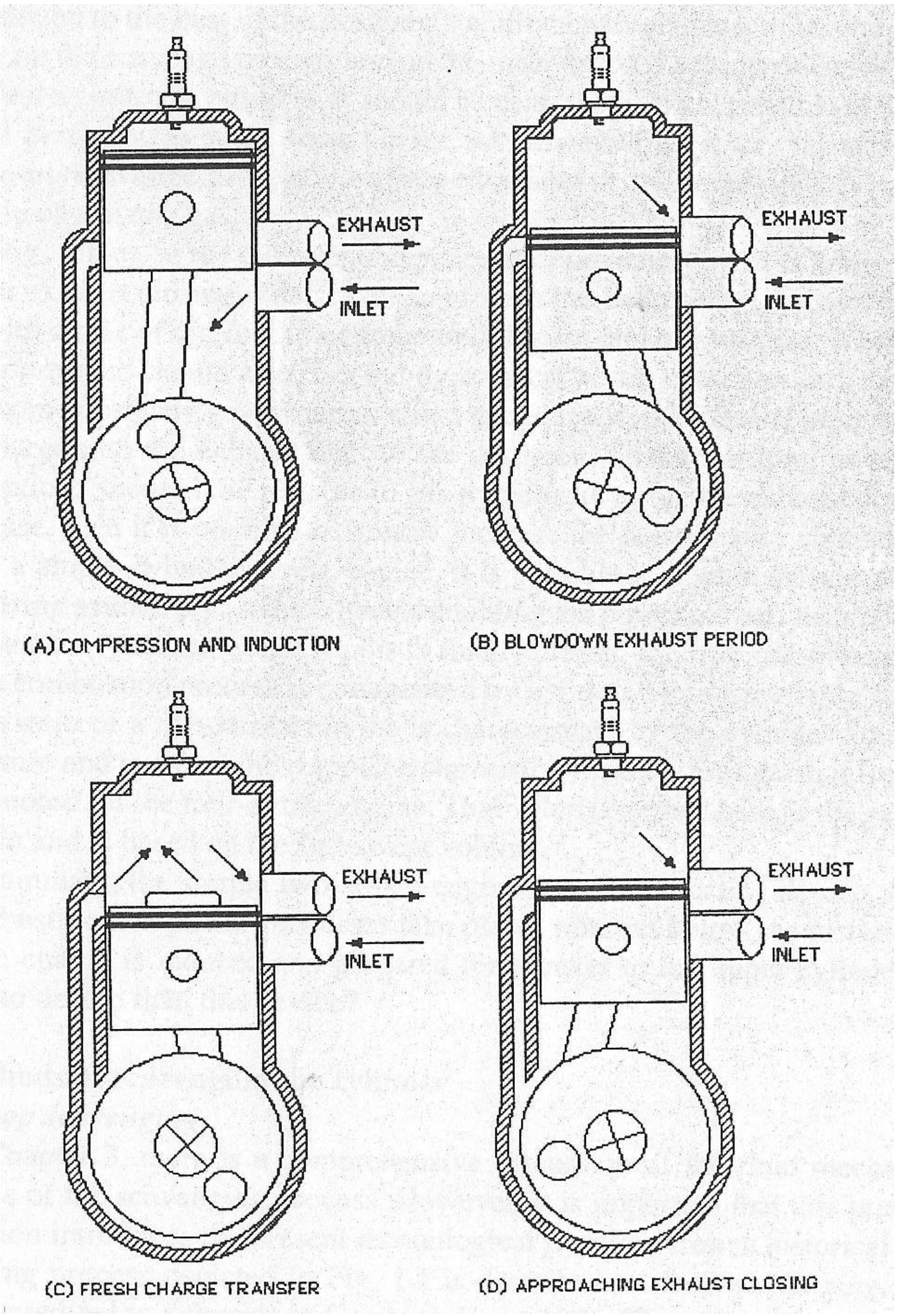

Figure 2.1.1 Operation stages of two-stroke engine

We see that piston after BDC is going upwards in figure 2.1.1 (D) and is just before exhaust port is closed. The process when piston goes from BDC till exhaust port is closed is called trapping. And the point when exhaust port is closed is trapping point. When piston goes up to TDC the pressure in cylinder do not rise till trapping point because exhaust port is open. Some part of fresh air-fuel mixture spills out through open exhaust port also during the trapping.

As you can notice, one power stroke occurs in one revolution in the two-stroke engine, whereas one power stroke occurs in two revolutions in four-stroke engine.

2.2 Various designs of the two-stroke engines

There are represented few design types of inlet valve and various scavenging schemes of engines.

Usually the inlet port timing is controlled by piston, figure 2.1.1. This type of inlet valve is used in cheap, simple engines.

The disc valve is the most sophisticated solution for the induction control. When disc valve is used there is possible to design any timing you want. There is complicated to design disc valve intake system for multi cylinder engine. The engine design with disc type valve is shown in figure. 2.2.1(A). This system was invented in 1950s by racing motorcycle company MZ.

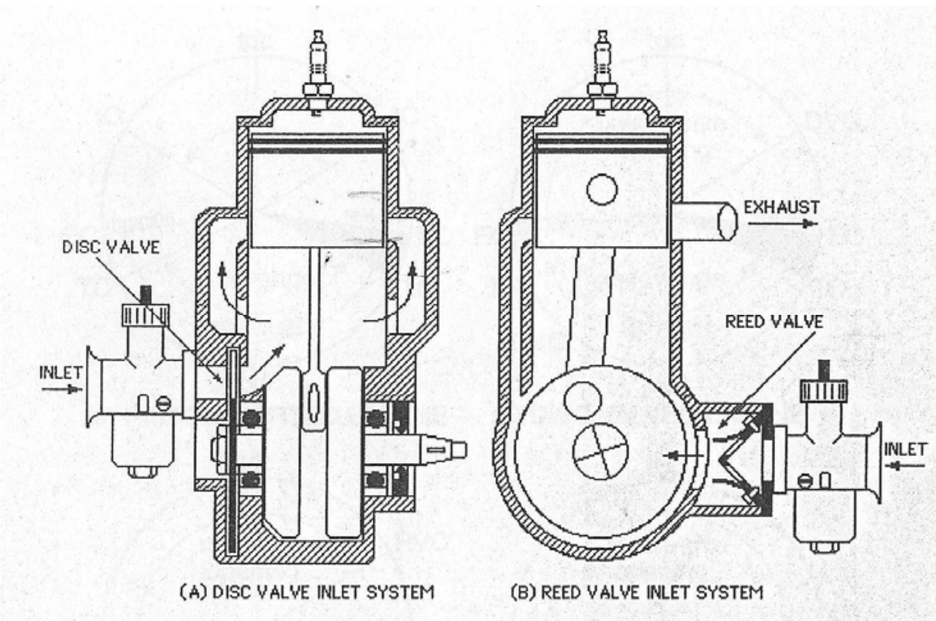

Figure 2.2.1 Engines with disc type (A) and reed valve (B) inlet systems

The reed valve of inlet system was the latest development of inlet systems. At the moment it is the most popular system in the World for the motorcycle, scooter engines and chainsaws. It is quite simple and very suitable system for any engine. The disadvantage is that there is not possible to adjust timing because it depends on the material of the reed valve plate, engines speed and design.

As it was written in previous chapter scavenging is very important process in efficiency of engine. There are two main types of scavenging types for small engines: loop scavenging and cross scavenging.

Loop scavenging was invented by Schnurle in Germany about 1926. The various scavenge port plans are shown in picture figure 2.2.2.

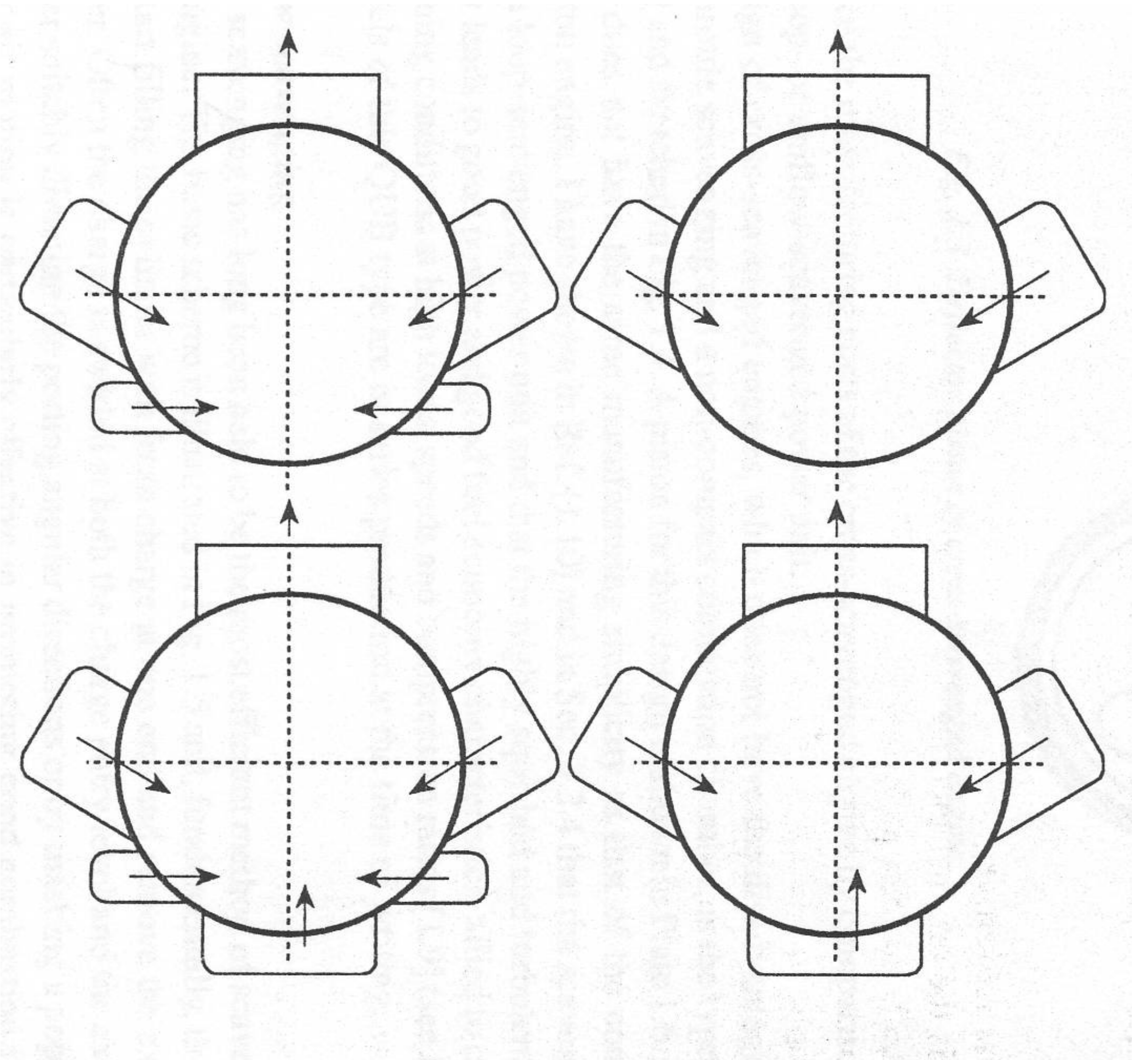

Figure 2.2.2 Various scavenge port plan layouts of loop scavenging.

The main idea of loop scavenging is to direct scavenge ports away from exhaust port, but across the piston. To make transfer ports across piston is important because of cooling the piston crown. Moreover flap top piston can be used in the loop scavenge system. There is possible to have better combustion efficiency with the flat top piston.

Cross scavenging is that one which was in the first ever made two stroke engine. It is very simple but at the same time it has a lot of negative sides. At first piston should have special deflector shape of the top, for this reason it is heavier. Combustion chamber is of complex shape because of non flat piston top. Moreover at higher rpm such engine tends to detonation and pre-ignition. The advantage is that it is easier to manufacture engines with cross scavenging, they are more compact if engine consist of many cylinders. In the picture below figure 2.2.3 there is shown cross scavenged engine with specially designed piston.

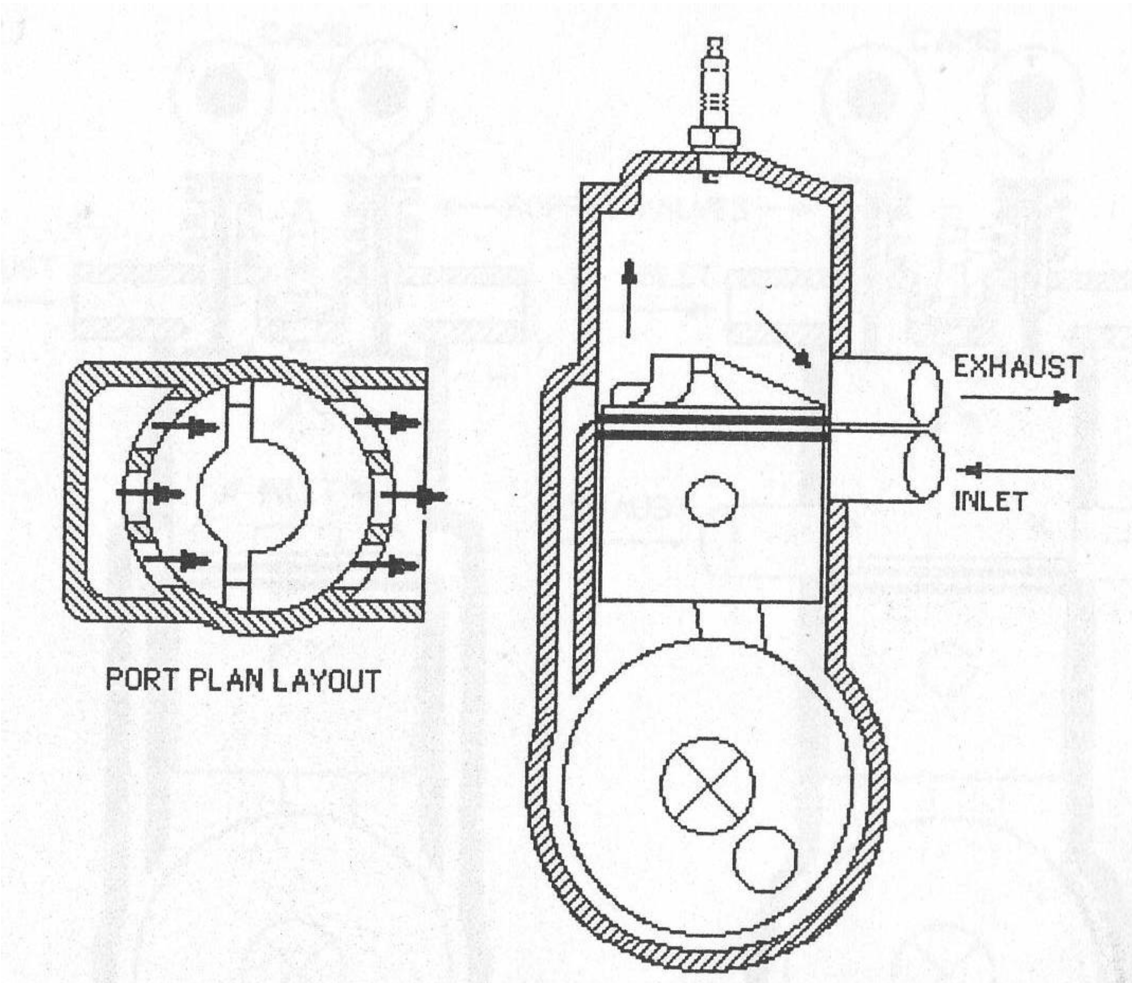

Figure 2.2.3 Cross scavenged engine.

2.3 The exhaust pipe of racing engine

If there are no restrictions by race regulations, there is used so called “power” pipe exhaust system for two stroke racing engines. The “power” exhaust system (in some books it is called expansion chamber) was invented about 1950's by East German motorcycle company "MZ". Correctly tuned exhaust system can increase engine power more than 25% and that is amazing fact. The system works according gas dynamic law. The expansion chamber is pipe with varying cross-sectional area along its length. The design of expansion chamber is very complex and depends on many engine parameters. In this subchapter there will be represented main principles, how does it works.

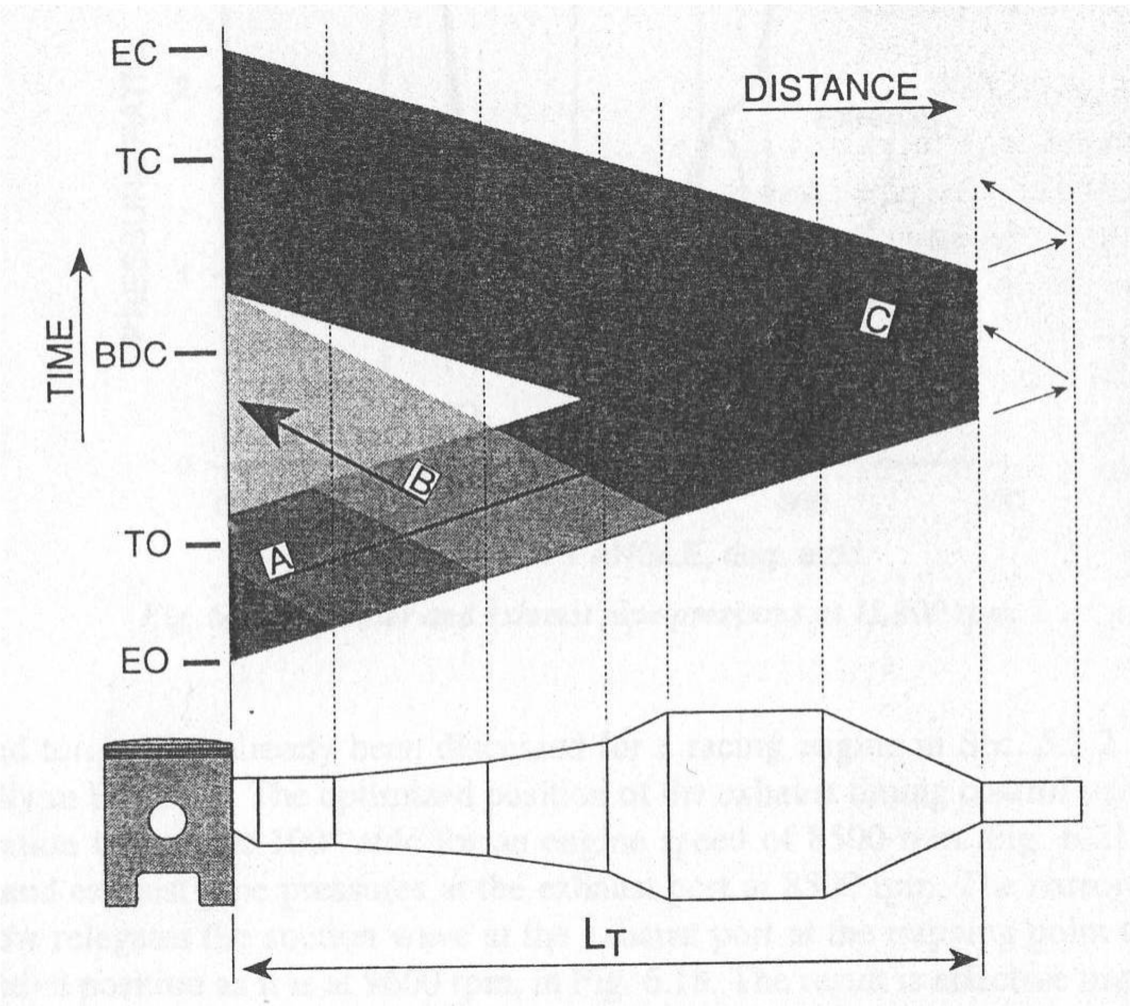

Figure 2.3.1 Pressure wave operation in "power" pipe

In figure 2.3.1 there is shown expansion chamber and pressure waves operation in it during one revolution. As it was mentioned before, the expansion chamber is of complex shape. At the left side we see the beginning of exhaust pipe starting from piston. In vertical axis there is time scale of one revolution, here EO means exhaust opens, TO – transfer opens, TC – transfer closes, EC – exhaust closes. So let's start from exhaust opening. When exhaust opens we have big pulse of burned gases during blowdown process, this pulse creates positive compression wave. When compression wave (in figure compression wave of blowdown is band of black color C) reaches cross-sectional changes in expansion chamber (divergent cones) according gas dynamic law it produces reflection waves which has low pressure and goes into opposite direction than positive compression wave. The divergent cones produce reflection waves which move to cylinder making suction effect (the band of grey color A and B). The created suction effect of reflected waves helps to take out burned gases out of cylinder and at the same time helps for better scavenging when transfer ports opens. The positive compression wave moves toward divergent cone where it reflects and goes back to cylinder. During scavenging and trapping there are spilled out to exhaust pipe some unburned fresh air-fuel mixture or unburned gas, so this returning compression wave pushes all back to cylinder. Also this high pressure wave prevents from spillage during the trapping period. In other words returning compression wave creates higher pressure at the exhaust port than at this moment is in cylinder. So we can state that it works like a valve. All the procedure described above repeats each revolution.

For a long time I was thinking that the exhaust gas is moving but not waves - this is not true. In order to understand how the expansion chamber works the reader should understand a little bit about wave theory. The exhaust gas medium is going out of cylinder to exhaust pipe and then to atmosphere and is the medium through which moves pressure waves. Pressure waves influences the speed and direction of the medium flow for very short periods. These short periods are very important in the two-stroke engines, because exhaust or scavenging takes less than 0.003 seconds in racing engines. These processes are shorter more than 3 times in the tether car racing models.

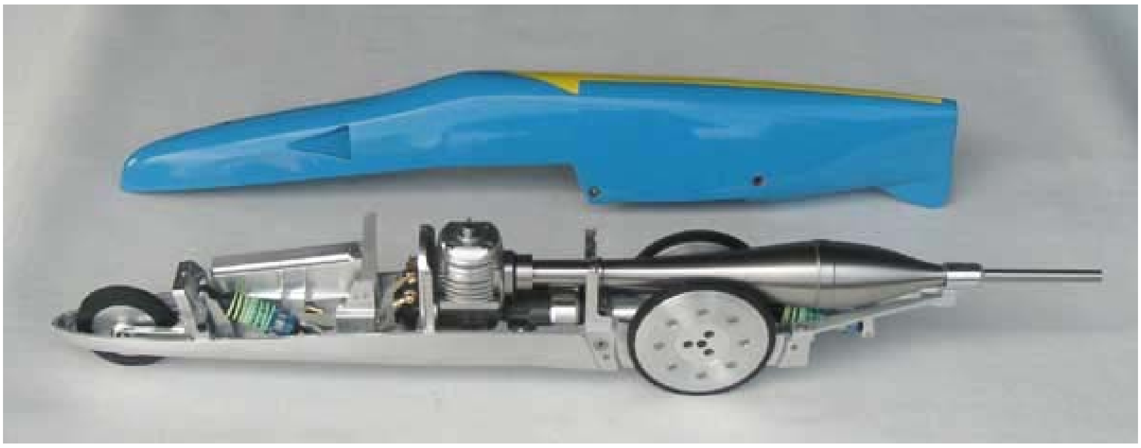

Figure 2.3.2 Tether race car model with removed cover. The engine connected with expansion chamber.

In figure 2.3.2 there is picture of tether race car model. There is shown model with removed cover and there is possible to see clear special design expansion chamber connected to engine.

Expansion chamber gives superior result but anyway it has some imperfections. At first it has to be quite long in comparison with engine size and there cannot be made any improvements because of fundamental reasons. For this reason it cannot be used in chainsaw or any other hand device. Another reason is that expansion chamber can work only in narrow band of revolutions. This band of revolution can be broadened but anyway only for some limit depending on the design. So engine with expansion chamber has to work always in narrow band of revolutions, it is ok in racing applications but for working purpose or normal life there are nothing fascinating. If the engine works not in the right revolutions the expansion chamber gives opposite affect which results in high fuel consumption and low power output.

3. REVIEW OF TWO-STROKE ENGINE SIMULATION METHODS IN THE WORLD

In this chapter there are represented simulation methods and calculation methods of the two-stroke engines or its parts concerning gas flow through two stroke engine. Mathematical modeling of engines is very complicated process, because it includes differential equations which are complex and requires very deep fundamental knowledge about processes happening in engine during its operation. Even if simple simulation is done to calculate gas flow through engine it requires kinematics description of engine, gas flow, burning process. All these processes should be related. Science of gas dynamic has made big jump when started numerical solutions for differential equations and it was about 1950's. Numerical solutions accelerated together with computer technologies. The biggest progress in simulating engines was done during the last 50 years.

3.1 Review of various scientific publications and books

When a short induction pipe is fixed to the engine, the delivery ratio can be improved by its effect in a wide range of the engine speeds (Nagao *et al.* 1961). Selecting the pipe length so as to coincide three quarters of the period of pressure fluctuation with the period of inlet port opening, the delivery ratio is most remarkably improved by the utilization of pressure fluctuation. Under the condition that the number of pressure fluctuation during the inlet port opening is below 0.5, the delivery ratio decreases through equipment with an induction pipe.

To improve the delivery ratio characteristics, it is not necessary to change the angle area. It is effective to change only the timing of the inlet port by shifting the disc valve with an optimum cut angle around the crankshaft in accordance with the change of engine speed (Komotori *et al.* 1969).

As the reduced angle area of the scavenging and exhaust ports is small, it is useless to change the angle area or timing of the inlet port to increase delivery ratio in the higher engine speed range (Komotori *et al.* 1969).

The drop in the delivery ratio caused by the increase in the crankcase volume is fairly compensated by the properly tuned exhaust and induction pipes. Therefore, it seems that scarcely any advantage is brought by making the crankcase volume excessively small (Nagao et al. 1959).

In the doctoral thesis "Development of an Efficient 3-D CFD Software to Simulate and Visualize the Scavenging of a Two-Stroke Engine" (Trescher, 2005) there is represented the whole procedure of mathematic modeling of the two-stroke engine. The thesis is written very well, but only for very well educated reader. The author of this thesis confirms once more that for special problems of fluid dynamic there should be derived models and it is not recommended to trust on the commercial CFD programs too much. There is used very sophisticated mathematical modeling and sophisticated numerical solution. In this thesis there is missing results comparison with tests or results which can be used as guidelines for design. As conclusion I can state that it this is like programming manual for two stroke engine.

There is written entire book about two-stroke engine simulation by Blair (1996). In this book there is analyzed two-stroke engine step by step and is provided guidelines for simulation. But this book can be used only for very advanced users, because it requires very good knowledge of gas dynamics and higher mathematics. The represented simulation is very complex. I like this book very much, because it represents many graphs of calculations and experiments, this book can be used for finding guidelines in engine design. In my opinion it is the best book concerning two-stroke engine design in the World at the moment.

3.2 Review of the book "Emissions from Two-Stroke Engines" by Nuti (1998)

The book "Emissions from Two-Stroke Engines" by Nuti (1998) makes overview of two-stroke engines. This book represents many graphs of experimental results concerning various data: scavenging, burning, emissions. There is short review about simulation methods in this book. And I would like to represent it in more details, because my thesis is concerned with engine simulation.

Since ancient times, one of the main goals of engineering sciences has been to be able to predict the behavior of a mechanical part or mechanism directly in advance of and without experimental work. Today, especially in the field of engines, it is important to have available a valid model to simulate the real behavior and to save money. Even more important, there must be time to make a new project/product available on the market. That is the reason why the industry has been pushing researchers to prepare increasingly more sophisticated models in parallel with computing means. It is easy to understand that the more complex the engine, the more useful and important it is to have a model available. Although it can be rather easy and inexpensive to change the cylinder of the small two-stroke SI engine, it is very expensive and infeasible to perform similar experimental setup of a large two-stroke CI engine for marine applications. In the latter case, it is critical “to make it right the first time”.

With simple mechanical problems, it is rather easy to have a valid model based on well-consolidated physical laws. However, many complex phenomena are involved in the field of engines, and thus many models are available. Among these models are some relative to “cool” internal gas dynamics (i.e. internal flow without combustion). Others are relative only to combustion phenomena. Only a few, which have developed in recent years as a result of the dramatic improvement in computer capabilities, are relative to the whole working process of engines.

Moreover, another problem that is valid in any class of models is the consideration of the “dimension” characteristic of the ambient in which the physical phenomena are simulated. Thus, the following types of models are possible:

- A one dimensional (1-D) model that, for example, works well in ducts, where the axial dimension is much more important than the traverse dimension
- A two-dimensional (2-D) model in any volume where a condition of symmetry is realized
- A three dimensional (3-D) model, which is most generic

Obviously, each model from 1-D to 3-D involves continually increasing complexity and computational time.

One-Dimensional model

The purpose of 1-D methods is to calculate the response of the different parts of the engine-namely, the inlet, crankcase, scavenge and exhaust ducts, and muffler-to determine, by using calculations on the entire flow through the engine, the predicted performance and the final value of different parameters such as charging efficiency, short-circuit ratio, and noise pressure levels. Usually the method consists of splitting the engine into several smaller parts that then can be schematized with ducts of different sections and lengths and connecting volumes as shown in figure 3.2.1.

Following the boundary conditions, the conservation equations are solved in any part; the solution is carried out with different numerical methods and pressure versus time history is represented for each point.

Generally, the phenomena related primarily to a 3-D aspect, such as scavenge model or fuel injection/air flow interaction or combustion models, cannot be directly introduced. Thus, the phenomenological methods are used and connected to the 1-D codes with simple algorithm such as triangular combustion model or a global trapping efficiency value at the beginning the compression phase.

The basic equation can include heat transfer and friction in ducts, but obviously they cannot take into account the turbulence effect in any part of the engine. The methods have the advantage of a rapid response on overall engine performance, making it easy to compare the influence on engine output of different geometrical configurations. The general drawback lies in the fact that accurate external calibrations, derived from experimental tests, are necessary, and then it is important to have solid knowledge of the physical behavior of every engine family. The valid use is only to obtain comparative results for a well-defined class of engines.

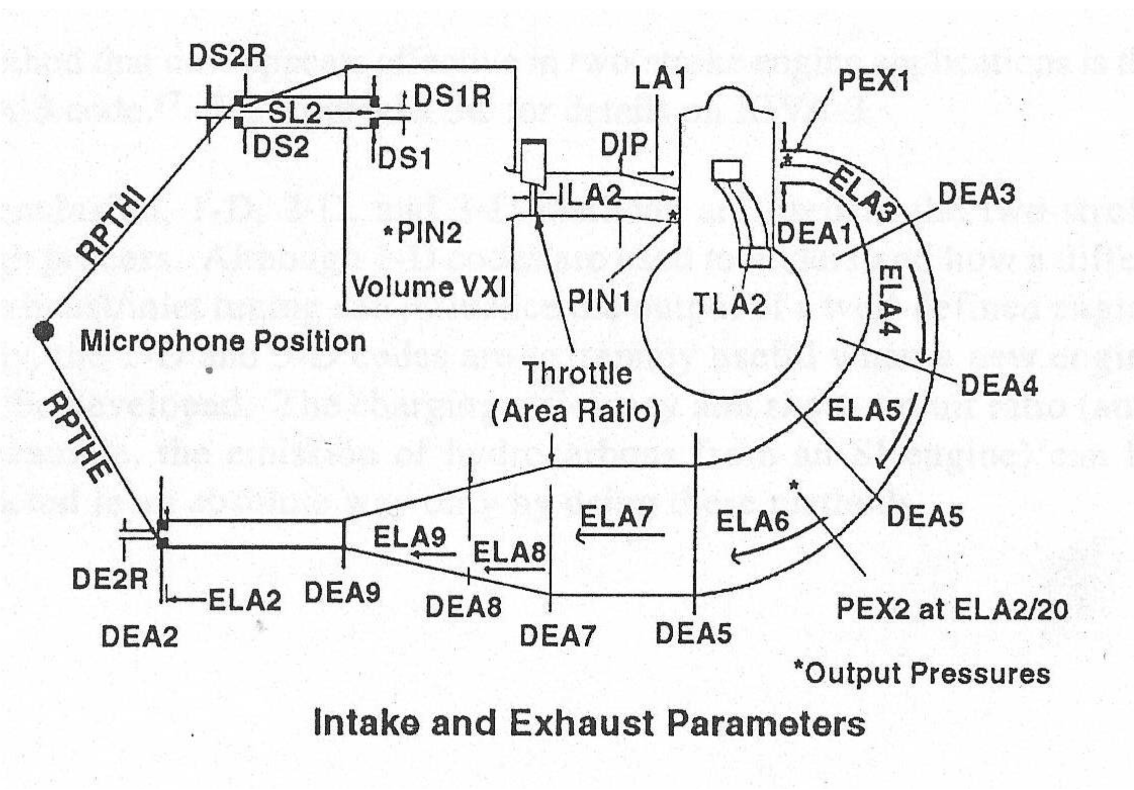

Figure 3.2.1 1-D calculation scheme of gas flow through a tuned engine

Many methods have been developed for solving differential equations. These range from the "characteristic" method, which could also be solved graphically, to modern numerical methods made possible only by advent of computers, such Lax-Wendroff and others. One of the most effective methods, particularly dedicated to the two stroke engine, is the "Two-Stroke Simulation Program" developed by Queen's University of Belfast. This method includes friction, heat transfer, and multiple ducts as well as catalyst insertion and multi-cylinder capabilities.

Two-Dimensional model

These 2-D methods simulate the scavenging phase as well as subsequent compression, combustion and expansion phases in conditions of axial symmetry. The mathematical model usually consists of solving five conservation law equations. These equations are derived from 3-D solution reduced to a 2-D problem. Physically, the variations of fluid properties (velocities, pressures and temperature) in the third dimension Z are assumed to be small compared to the variation in the X and Y directions. Therefore, only an average constant value is considered in the Z direction. The real advantage of these methods is the huge simplification, compared to 3-D calculations that usually involve a supercomputer if the result must be available in an acceptable working time. The 2-D system also can solve turbulent fluid mechanics equations (averaged Navier-Stokes with turbulence model k-ε) in acceptable way. The drawbacks are the definition of the initial and boundary conditions that must be precisely defined because of their significant influence on the simulation program outputs. Besides the initial conditions influenced by the geometrical layout of the engine, the most difficult parameters to be specified are the turbulence coefficients k and ε. Today, the validity of 2-D models is mainly in the very simplified versions. Following the development of computing facilities, the trends in practical applications are oriented toward use of complete and more exhaustive 3-D models.

Three-Dimensional model

The purpose of these 3-D methods is to provide a theoretical evaluation of the internal gas dynamics, including all features of the cylinder geometry, the different properties of the various gases, the injection of the liquid phase, and the actual heat transfer. Many codes are available with satisfactory behavior both on CPU optimization and output results; however the most critical aspect dealing with two-stroke engines is the schematization of the motion of the piston interfering fixed ports. The mesh formation method must be accurate, and special conditions are necessary.

A method that now appears effective in the two-stroke engine applications is the KIVA-3 code.

In conclusion 1-D, 2-D, 3-D methods are used in the two-stroke design process. Although 1-D coeds are used to understand how different exhaust/inlet tuning can influence the output of a well defined engine family, the 2-D and 3-D codes are extremely useful when a new engine must be developed. The charging efficiency and short-circuit ratio can be predicted in an absolute way only by using these methods.

4. SIMULATION OF THE TWO-STROKE ENGINE

In this chapter there is represented my calculation scheme of the two-stroke engine, equations of the gas flow and equations related to engine kinematics.

The engine was simulated according so called one dimensional scheme. That was done because of simplicity and lack of knowledge of better simulation methods. In the simulation there was not included burning process, wave propagation, turbulence effects. The gas medium was chosen to be air. The simulation was done using concentrated volume method. The temperatures in different parts of engine were taken as constants from experience.

4.1 The two-stroke engine data which is used for simulation

I participate in tether model car race and I chosen to simulate engine which is used for this race. The engine is of 2.5 cc swept volume. The engine use fuel of 80% methanol and 20% castor oil. It is so called combustion ignition engine (CI).

Technical data of engine:

Bore – 14.9 mm;

Stroke – 14.25 mm;

Number of transfer ports – 3;

Crankcase volume at BDC – 12.5 cc;

Combustion chamber volume at TDC – 0.19 cc;

Inlet system – disc valve;

Inlet timing – opens 35° after BDC, closes 65° after TDC;

Transfer timing - 135°;

Exhaust timing - 195°.

There are described only main engine parameters. The entire list of engine parameters can be found inside the program, there are 39 parameters describing engine.

Below there is shown transparent simulated engine picture without head. The drawing was done using “Solidworks” program.

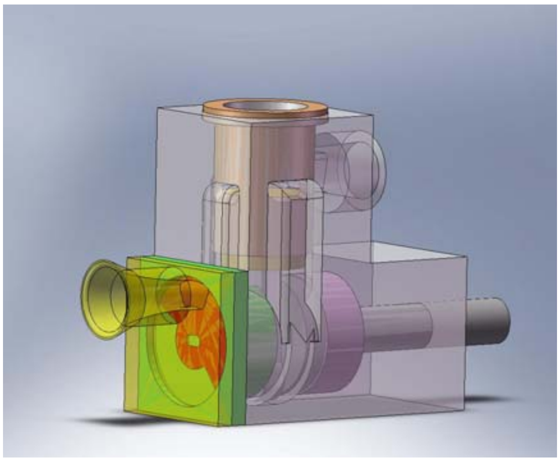

Figure 4.1.1 The 3D transparent model picture of simulated engine

The picture is 3D model, which was used for calculations. The 3D model was very useful when there were need to calculate crankcase volume, transfer port volume, transfer, inlet, and exhaust port areas.

Below in figure 4.1.2 and 4.1.3 there is shown assembled and disassembled Swedish 5 cc tether model car engine manufactured by sportsmen Nils Bjork. The design is the same as investigated 2.5 cc engine. The differences are only in swept volume and number of transfer and exhaust ports.

Figure 4.1.2 5 cc tether model car engine

Figure 4.1.3 Disassembled two-stroke engine of tether model car

The pictures are given just for the reader imagination and understanding how the tether model car engines look like. The design of very small engines has its own peculiarities, but all the elements and working principles are the same as in normal two-stroke engines.

4.2 Calculation scheme of the simulated engine

For the calculation I chose concentrated volume method. It is the simplest method to simulate gas dynamics. With this method there is possible to do only 1-D simulation. Calculation scheme is shown in figure 4.2.1.

Entire engine was divided into parts and each part was described by equations (for the equations see next subchapters). The initial data and some engine geometrical properties are represented in calculation scheme. The engine is divided into crankcase, three separate transfer ducts and cylinder by equations. Exhaust pipe is divided into four parts. The transfer ducts are separated from crankcase because they have quite big volume.

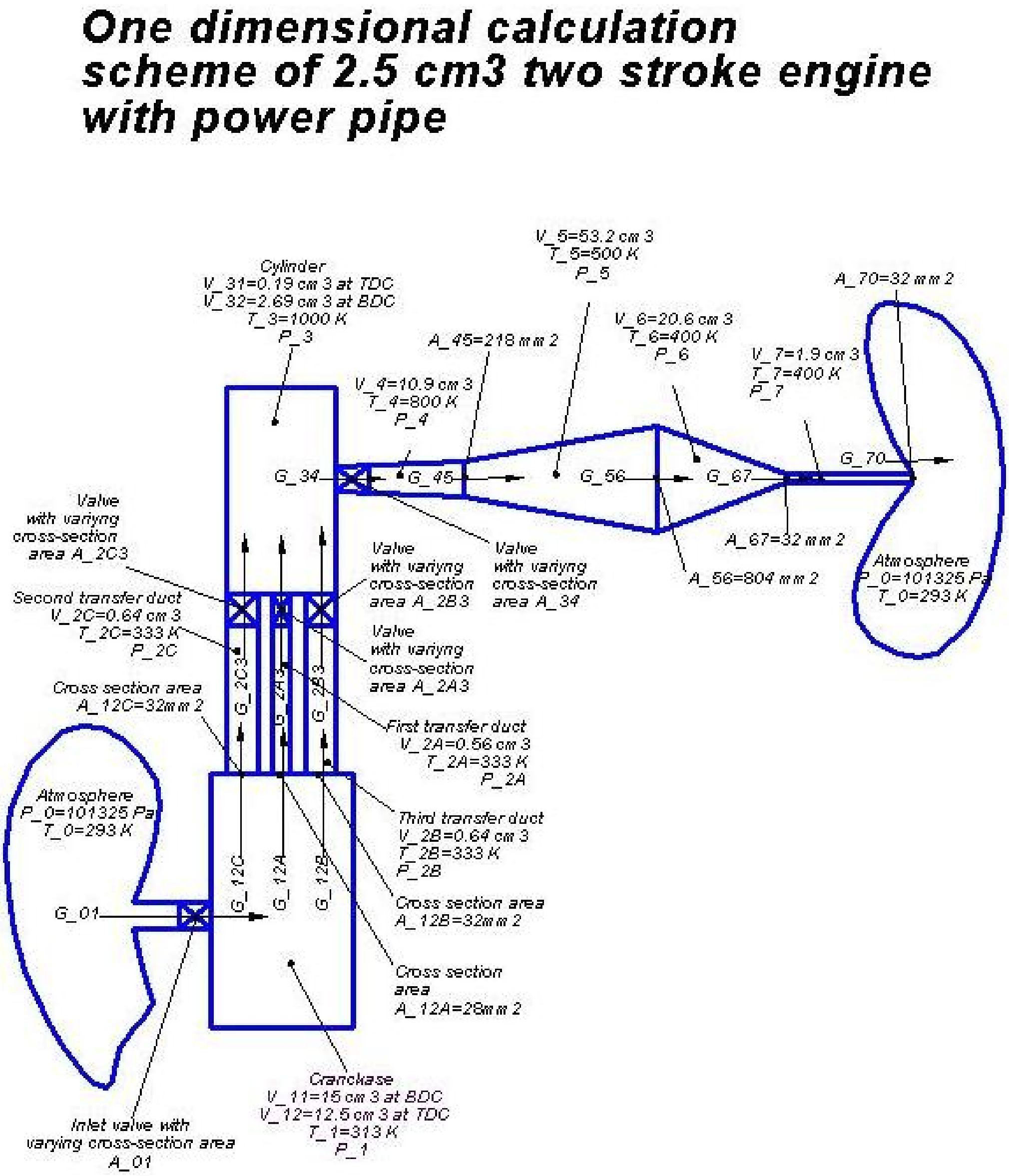

Figure 4.2.1 Calculation scheme of simulated engine

4.3 Gas flow calculation through the two-stroke engine

In the next subchapters there will be introduced many formulas. In some formulas you will find some variables denoted as engine parameter together with its number in brackets. According this information you can easily find this parameter in the simulation program and you can change according to your requirement.

As it was mentioned in introduction or in later sections, the gas flow is calculated according concentrated volume method. There is short introduce to this method.

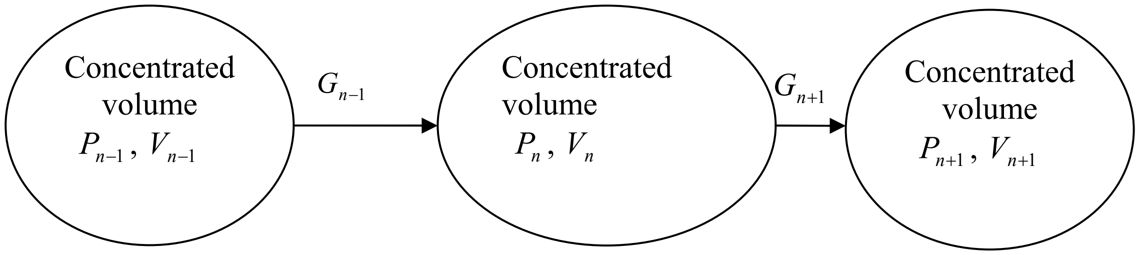

Figure 4.3.1 Calculation scheme of concentrated volume method

In the calculation scheme above (figure 4.3.1) there is represented standard situation of gas flow task. There are several concentrated volumes connected by pipes or ducts. In order to calculate pressure changes and mass flow between them, the concentrated volumes have to be described by equations.

The pressure change in any volume can be expressed by formula (about formula derivation please refer to Bogdevičius and Prenkovskis (2003)):

$$\dot{p}_n = \frac{\gamma R T_n}{V_n}\left(G_{n-1} - G_{n+1}\right) - \frac{\gamma p_n}{V_n}\dot{V}_n \tag{4.1}$$

$\dot{p}_n$ - The pressure derivative in the concentrated volume;

γ - Specific heat of gas;

R - Gas constant which equals for all gases $287 J/(kg \cdot K)$;

T_n - Temperature in the concentrated volume, K;

V_n - Volume of the concentrated volume, m^3;

G_{n-1} and G_{n+1} - Mass flow in or out to the concentrated volume, kg/s. The inflow is positive, when mass flow enters the concentrated volume and negative, when outflows;

p_n - Pressure at the concentrated volume, Pa;

$\dot{V}_n$ - The pressure derivative of concentrated volume, if there is volume changes during the time.

The mass flow can be calculated:

$$G_{n-1}=\begin{cases}\mu_{n-1}A_{n-1}p_{n-1}\sqrt{\dfrac{2\gamma}{(\gamma-1)RT}\cdot\left(\left(\dfrac{p_n}{p_{n-1}}\right)^{\frac{2}{\gamma}}-\left(\dfrac{p_n}{p_{n-1}}\right)^{\frac{\gamma+1}{\gamma}}\right)}, \text{if } p_{n-1}>p_n\\ -\mu_{n-1}A_{n-1}p_n\sqrt{\dfrac{2\gamma}{(\gamma-1)RT}\cdot\left(\left(\dfrac{p_{n-1}}{p_n}\right)^{\frac{2}{\gamma}}-\left(\dfrac{p_{n-1}}{p_n}\right)^{\frac{\gamma+1}{\gamma}}\right)}, \text{if } p_n>p_{n-1}\end{cases} \tag{4.2}$$

The inflow (positive mass flow) exist according to the condition that $p_{n-1} > p_n$, if there is $p_{n-1} < p_n$, then there is mass outflow and its value should be negative.

μ_{n-1} - Coefficient which evaluates surface roughness of the ducts;

A_{n-1} - Cross-sectional area of the duct which connects two concentrated volumes;

p_{n-1} - Pressure in the previous concentrated volume, Pa.

The calculation scheme shown in figure 4.3.1 is an example there can be many gas inflows or outflows in concentrated volume. But in order to solve the task each concentrated volume has to be described by pressure change equation and inflow/outflow must be described by mass flow equation. The pressure change equation is the first order differential equation. There is not possible to solve it by hand if there is need for thousand solutions. For this reason it is solved by computer. The numerical solution with computer will be discussed in later chapters.

The above pressure changes equation is with respect to time. Because I do engine simulation for me there is need to convert pressure change equation with respect to θ, crankshaft angle. This will give advantage in easier numerical solution and visualization of results. All the variables like mass flow and volume changes are also expressed with respect to θ.

The equation (4.1) is expressed with respect to θ:

$$\frac{dp}{dt}=\frac{dp}{d\theta}\cdot\omega \tag{4.3}$$

θ - Crankshaft angle, rad;

ω - Angular velocity of the crankshaft, rad/s.

According equation (4.3) we get that equation (4.1) transforms into such one:

$$\dot{p}_n(\theta) = \frac{\gamma R T_n}{V_n(\theta)\cdot\omega}\left(G_{n-1}(\theta) + G_{n+1}(\theta)\right) - \frac{\gamma p_n(\theta)}{V_n(\theta)\cdot\omega}\dot{V}_n(\theta) \qquad (4.4)$$

The θ in bracket near of the variables means, that the variables are expressed with respect to crankshaft angle θ.

Because we changed the derivative variable, also mass flow equation (4.2) changes to a new one, where all variables are expressed with respect to θ:

$$G_{n-1}(\theta) = \begin{cases} \mu_{n-1}A_{n-1}(\theta)p_{n-1}(\theta)\sqrt{\dfrac{2\gamma}{(\gamma-1)RT}\cdot\left(\left(\dfrac{p_n(\theta)}{p_{n-1}(\theta)}\right)^{\frac{2}{\gamma}} - \left(\dfrac{p_n(\theta)}{p_{n-1}(\theta)}\right)^{\frac{\gamma+1}{\gamma}}\right)}, \text{if } \mathrm{p_{n-1}}(\theta) > \mathrm{p_n}(\theta) \\ -\mu_{n-1}A_{n-1}(\theta)p_n(\theta)\sqrt{\dfrac{2\gamma}{(\gamma-1)RT}\cdot\left(\left(\dfrac{p_{n-1}(\theta)}{p_n(\theta)}\right)^{\frac{2}{\gamma}} - \left(\dfrac{p_{n-1}(\theta)}{p_n(\theta)}\right)^{\frac{\gamma+1}{\gamma}}\right)}, \text{if } \mathrm{p_n}(\theta) > \mathrm{p_{n-1}}(\theta) \end{cases}$$

(4.5)

All these changes let us to describe all processes in engine according crankshaft angle θ.

4.4 Equation of pressure changes in crankcase, $\dot{p}_1$

As the introduction for the concentrated volume method was done lets start describing the two-stroke engine by equations according represented theory. All the notation in formulas coincides with calculation scheme represented in subchapter 4.2 (Figure 4.2.1). There is possible to follow what is described by formulas by looking at calculation scheme of the engine.

The gas medium is air. All variables are expressed depending on θ (angle of crankshaft). The start point of calculations is TDC.

There is substituted all known values to equation (4.4) and we get pressure change equation of the crankcase:

$$\dot{p}_1(\theta) = \frac{\gamma R T_1}{V_1(\theta)\cdot\omega}\left(G_{0\_1}(\theta) - G_{1\_2A}(\theta) - G_{1\_2B}(\theta) - G_{1\_2C}(\theta)\right) - \frac{\gamma p_1(\theta)}{V_1(\theta)\cdot\omega}\dot{V}_1(\theta) \qquad (4.6)$$

$\dot{p}_1(\theta)$ - Pressure derivative of the crankcase;

$\gamma = 1.4$ - Specific heat ratio of air;

$R = 287 J/(kg\cdot K)$ - Gas constant;

$T_1 = 313K$ - Temperature in the crankcase;

$V_1(\theta), m^3$ - Crankcase volume, which is defined by formula (4.8);

$\omega, rad/s$ - Angular velocity of the crankshaft, engine parameter (1), which is expressed by formula:

$$\omega = 2\pi n / 60 \tag{4.7}$$

$n, \min^{-1}$ - Crankshaft's number of revolutions per minute;

$G_{0\_1}(\theta), kg/s$ - Mass flow from atmosphere to crankcase, which is expressed by formula (4.9);

$G_{1\_2A}(\theta), kg/s$ - Mass flow from crankcase to transfer channel (A), which is expressed by formula (4.11);

$G_{1\_2B}(\theta), kg/s$ - Mass flow from crankcase to transfer channel (B), which is expressed by formula (4.12);

$G_{1\_2C}(\theta), kg/s$ - Mass flow from crankcase to transfer channel (C), which is expressed by formula (4.13);

$p_1(\theta), Pa$ - Pressure in the crankcase.

$\dot{V}_1(\theta)$ - Crankcase volume change rate, which is defined by formula (4.14).

Crankcase volume function, $V_1(\theta)$

$$V_1(\theta) = V_{11} - A_p\left(L_{cr} + L_{ct}(1-\cos(\theta)) - \sqrt{L_{cr}^2 - (L_{ct}\sin(\theta))^2}\right) \tag{4.8}$$

$V_{11} = 15 \times 10^{-6} m^3$ - Crankcase volume at TDC, engine parameter (2);

$A_p = 174.37 \times 10^{-6} m^2$ - Piston area, engine parameter (6);

$L_{cr} = 26.5 \times 10^{-3} m$ - Connecting-rod length, engine parameter (4);

$L_{ct} = 7.125 \times 10^{-3} m$ - Half-stroke length, engine parameter (5).

The derivation of the crankcase volume function can be found in the book of Blair (1996).

Mass flow rate function, $G_{0\_1}(\theta)$

$$G_{0\_1}(\theta)=\begin{cases}\mu_{0\_1}A_{0\_1}(\theta)p_0\sqrt{\dfrac{2\gamma}{(\gamma-1)RT}\cdot\left(\left(\dfrac{p_1(\theta)}{p_0}\right)^{\frac{2}{\gamma}}-\left(\dfrac{p_1(\theta)}{p_0}\right)^{\frac{\gamma+1}{\gamma}}\right)}, \text{if } \mathrm{p}_0>\mathrm{p}_1(\theta)\\ -\mu_{0\_1}A_{0\_1}(\theta)p_1(\theta)\sqrt{\dfrac{2\gamma}{(\gamma-1)RT}\cdot\left(\left(\dfrac{p_0}{p_1(\theta)}\right)^{\frac{2}{\gamma}}-\left(\dfrac{p_0}{p_1(\theta)}\right)^{\frac{\gamma+1}{\gamma}}\right)}, \text{if } \mathrm{p}_1(\theta)>\mathrm{p}_0\end{cases}$$

(4.9)

$\mu_{0\_1}=0.8$ - Coefficient evaluating surface roughness of inlet duct, engine parameter (7);

$A_{0\_1}(\theta), m^2$ - Area of the of the inlet disc type valve, which is expressed by formula (4.10);

$p_0=101325Pa$ - Ambient (atmosphere) pressure;

$p_1(\theta), Pa$ - Crankcase pressure.

Area function of the inlet valve, $A_{0\_1}(\theta)$

There is simulated inlet valve of disc type. In the engine designers world disc type valve has its own specific name “Zimmerman”. Such type inlet valves were invented by motorcycle company “MZ” from ex East Germany in about 1950’s. You can have any timing you require with such type of inlet valve. The principle design is shown in Figure 2.2.1. It is the most popular design of inlet valves in tether model car race engines. The schematic drawing for the calculations of disc type inlet valve is represented in Figure 4.4.1.

$$A_{0\_1}(\theta)=\begin{cases}\pi\dfrac{\phi}{2\pi}\left(r_{\max}^2-r_{\min}^2\right), \text{if } 0<\theta\le\theta_{in1}=\alpha_{in}-\phi\\ \pi\dfrac{\phi-\theta}{2\pi}\left(r_{\max}^2-r_{\min}^2\right), \text{if } \theta_{in1}<\theta\le\theta_{in2}=\alpha_{in}\\ 0, \text{if } \theta_{in2}<\theta\le\theta_{in3}=\pi+\beta_{in}\\ \pi\dfrac{\theta}{2\pi}\left(r_{\max}^2-r_{\min}^2\right), \text{if } \theta_{in3}<\theta\le\theta_{in4}=\theta_{in3}+\phi\\ \pi\dfrac{\phi}{2\pi}\left(r_{\max}^2-r_{\min}^2\right), \text{if } \theta_{in4}<\theta\le 2\pi\end{cases} \tag{4.10}$$

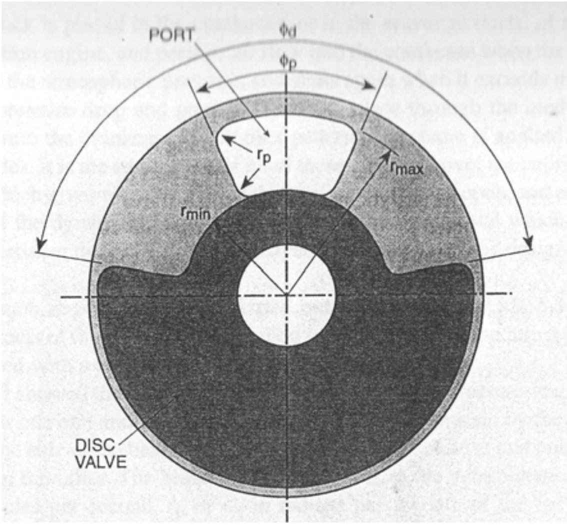

Figure 4.4.1 Inlet valve of the disc type with necessary dimensions for timing calculations

$\phi = 50^0 = 0.8727 rad$ - Inlet duct angle in cross-section area, engine parameter (8), in the figure 4.4.1 this parameter is denoted by ϕ_p;

$\alpha_{in} = 65^0 = 1.1345 rad$ - Inlet port closes after TDC, engine parameter (9);

$\beta_{in} = 35^0 = 0.6109 rad$ - Inlet port opens after BDC, engine parameter (12);

$r_{\max} = 10.55 \times 10^{-3} m$ - Inlet port outer radius, engine parameter (10);

$r_{\min} = 6 \times 10^{-3} m$ - Inlet port inner radius, engine parameter (11).

Mass flow rate function, $G_{1\_2A}(\theta)$

$$G_{1\_2A}(\theta) = \begin{cases} \mu_{1\_2A} A_{1\_2A} p_1(\theta) \sqrt{\dfrac{2\gamma}{(\gamma-1)RT} \cdot \left(\left(\dfrac{p_{2A}(\theta)}{p_1(\theta)} \right)^{\frac{2}{\gamma}} - \left(\dfrac{p_{2A}(\theta)}{p_1(\theta)} \right)^{\frac{\gamma+1}{\gamma}} \right)}, \text{if } \mathrm{p}_1(\theta) > \mathrm{p}_{2A}(\theta) \\ -\mu_{1\_2A} A_{1\_2A} p_{2A}(\theta) \sqrt{\dfrac{2\gamma}{(\gamma-1)RT} \cdot \left(\left(\dfrac{p_1(\theta)}{p_{2A}(\theta)} \right)^{\frac{2}{\gamma}} - \left(\dfrac{p_1(\theta)}{p_{2A}(\theta)} \right)^{\frac{\gamma+1}{\gamma}} \right)}, \text{if } \mathrm{p}_{2A}(\theta) > \mathrm{p}_1(\theta) \end{cases}$$

(4.11)

$G_{1\_2A}(\theta), kg/s$ - Mass flow rate from crankcase to transfer channel (A);

$\mu_{1\_2A} = 0.8$ - Coefficient evaluating surface roughness of transfer ducts, engine parameter (16);

$A_{1\_2A} = 28 \times 10^{-6} m^2$ - Cross-sectional area of transfer duct, engine parameter (14);

$p_{2A}(\theta), Pa$ - Pressure in first transfer duct (A).

Mass flow rate function, $G_{1\_2B}(\theta)$

$$G_{1\_2B}(\theta)=\begin{cases}\mu_{1\_2B}A_{1\_2B}p_1(\theta)\sqrt{\dfrac{2\gamma}{(\gamma-1)RT}\cdot\left(\left(\dfrac{p_{2B}(\theta)}{p_1(\theta)}\right)^{\frac{2}{\gamma}}-\left(\dfrac{p_{2B}(\theta)}{p_1(\theta)}\right)^{\frac{\gamma+1}{\gamma}}\right)}, \text{if } \mathrm{p}_1(\theta)>\mathrm{p}_{2\mathrm{B}}(\theta)\\ -\mu_{1\_2B}A_{1\_2B}p_{2B}(\theta)\sqrt{\dfrac{2\gamma}{(\gamma-1)RT}\cdot\left(\left(\dfrac{p_1(\theta)}{p_{2B}(\theta)}\right)^{\frac{2}{\gamma}}-\left(\dfrac{p_1(\theta)}{p_{2B}(\theta)}\right)^{\frac{\gamma+1}{\gamma}}\right)}, \text{if } \mathrm{p}_{2\mathrm{B}}(\theta)>\mathrm{p}_1(\theta)\end{cases}$$

(4.12)

$G_{1\_2B}(\theta), kg/s$ - Mass flow rate from crankcase to transfer channel (B);

$\mu_{1\_2B}=0.8$ - Coefficient evaluating surface roughness of transfer port, engine parameter (16);

$A_{1\_2B}=32\times10^{-6}m^2$ - Cross-sectional area of transfer duct, engine parameter (20);

$p_{2B}(\theta), Pa$ - Pressure in second transfer duct (B).

Mass flow rate function, $G_{1\_2C}(\theta)$

$$G_{1\_2C}(\theta)=\begin{cases}\mu_{1\_2C}A_{1\_2C}p_1(\theta)\sqrt{\dfrac{2\gamma}{(\gamma-1)RT}\cdot\left(\left(\dfrac{p_{2C}(\theta)}{p_1(\theta)}\right)^{\frac{2}{\gamma}}-\left(\dfrac{p_{2C}(\theta)}{p_1(\theta)}\right)^{\frac{\gamma+1}{\gamma}}\right)}, \text{if } \mathrm{p}_1(\theta)>\mathrm{p}_{2\mathrm{C}}(\theta)\\ -\mu_{1\_2C}A_{1\_2C}p_{2C}(\theta)\sqrt{\dfrac{2\gamma}{(\gamma-1)RT}\cdot\left(\left(\dfrac{p_1(\theta)}{p_{2C}(\theta)}\right)^{\frac{2}{\gamma}}-\left(\dfrac{p_1(\theta)}{p_{2C}(\theta)}\right)^{\frac{\gamma+1}{\gamma}}\right)}, \text{if } \mathrm{p}_{2\mathrm{C}}(\theta)>\mathrm{p}_1(\theta)\end{cases}$$

(4.13)

$G_{1\_2C}(\theta), kg/s$ - Mass flow rate from crankcase to transfer channel (C);

$\mu_{1\_2C}=0.8$ - Coefficient evaluating surface roughness of transfer port, engine parameter (16);

$A_{1\_2C}=32\times10^{-6}m^2$ - Cross-sectional area of transfer duct, engine parameter (21);

$p_{2C}(\theta), Pa$ - Pressure in second transfer duct (B).

Function of crankcase volume change, $\dot{V}_2(\theta)$

$$\dot{V}_2(\theta)=-A_p\left(L_{ct}\sin(\theta)+\frac{L_{ct}^2\sin(\theta)\cos(\theta)}{\sqrt{L_{cr}^2-(L_{ct}\sin(\theta))^2}}\right) \tag{4.14}$$

$\dot{V}_2(\theta)$ - Function of crankcase volume change.

4.5 Equation of pressure changes in the transfer duct (A), $\dot{p}_{2A}$

$$\dot{p}_{2A}(\theta)=\frac{\gamma R T_{2A}}{V_{2A}\cdot\omega}\left(G_{1\_2A}(\theta)-G_{2A\_3}(\theta)\right) \tag{4.15}$$

$\dot{p}_{2A}(\theta)$ - Change of pressure in first transfer duct (A);

$\gamma = 1.4$ - Specific heat ratio of air;

$R = 287 J/(kg\cdot K)$ - Gas constant;

$V_{31} = width \times length \times thickness = 7\times 4\times 20 = 560 mm^3 = 560\times 10^{-9} m^3$ - First transfer duct volume, engine parameter (13);

$T_{2A} = 333K$ - Temperature in first transfer duct (A).

Flow mass rate function, $G_{2A\_3}(\theta)$

$$G_{2A\_3}(\theta)=\begin{cases}\mu_{2A\_3}A_{2A\_3}(\theta)p_{2A}(\theta)\sqrt{\dfrac{2\gamma}{(\gamma-1)RT}\cdot\left(\left(\dfrac{p_3(\theta)}{p_{2A}(\theta)}\right)^{\frac{2}{\gamma}}-\left(\dfrac{p_3(\theta)}{p_{2A}(\theta)}\right)^{\frac{\gamma+1}{\gamma}}\right)}, \text{if } \mathrm{p_{2A}}(\theta)>\mathrm{p_3}(\theta)\\ -\mu_{2A\_3}A_{2A\_3}(\theta)p_3(\theta)\sqrt{\dfrac{2\gamma}{(\gamma-1)RT}\cdot\left(\left(\dfrac{p_{2A}(\theta)}{p_3(\theta)}\right)^{\frac{2}{\gamma}}-\left(\dfrac{p_{2A}(\theta)}{p_3(\theta)}\right)^{\frac{\gamma+1}{\gamma}}\right)}, \text{if } \mathrm{p_3}(\theta)>\mathrm{p_{2A}}(\theta)\end{cases}$$

(4.16)

$G_{2A\_3}(\theta), kg/s$ - Mass flow rate from first transfer duct to cylinder;

$\mu_{2A\_3} = 0.8$ - Coefficient evaluating surface roughness of transfer port, engine parameter (17).

$A_{2A\_3}(\theta), m^2$ - First transfer port area, which is expressed by equation (4.17);

$p_{2A}(\theta), Pa$ - Pressure in first transfer duct;

$p_3(\theta), Pa$ - Pressure in cylinder.

Transfer port area function corresponding to crankshaft angle

$$A_{2A\_3}(\theta)=\begin{cases}0, \text{if } 0<\theta_{tr1}=\pi-\dfrac{\alpha_{tr}}{2}\\ W_{2A\_3}\left(-L_{ct}\cos(\theta)-\sqrt{L_{cr}^2-L_{ct}^2\sin(\theta)^2}+L_{ct}\cos(\theta)+\sqrt{L_{cr}^2-L_{ct}^2\sin(\theta)^2}\right), \text{if } \theta_{tr1}<\theta\le\theta_{tr2}=\pi+\dfrac{\alpha_{tr}}{2}\\ 0, \text{if } \theta_{\mathrm{tr2}}<\theta<2\pi\end{cases}$$

(4.17)

$A_{2A\_3}(\theta), m^2$ - First transfer port area;

$W_{2A\_3} = 7\times 10^{-3} m$ - Width of the first transfer port, engine parameter (15);

$\alpha_{tr} = 136^0 = 2.3736 rad$ - Transfer timing, engine parameter (18).

4.6 Equation of pressure changes in the transfer duct (B), $\dot{p}_{2B}$

$$\dot{p}_{2B}(\theta) = \frac{\gamma R T_{2B}}{V_{2B} \cdot \omega} \left(G_{1\_2B}(\theta) - G_{2B\_3}(\theta) \right) \tag{4.18}$$

$\dot{p}_{2B}(\theta)$ - Change of pressure in transfer duct (B);

$\gamma = 1.4$ - Specific heat ratio of air;

$R = 287 J/(kg \cdot K)$ - Gas constant;

$V_{2B} = width \times length \times thickness = 8 \times 4 \times 20 = 560 mm^3 = 640 \times 10^{-9} m^3$ - Transfer duct volume (B), engine parameter (22);

$T_{2B} = 333K$ - Temperature in first transfer duct (B).

Flow mass rate function, $G_{2B\_3}(\theta)$

$$G_{2B\_3}(\theta) = \begin{cases} \mu_{2B\_3} A_{2B\_3}(\theta) p_{2B}(\theta) \sqrt{\dfrac{2\gamma}{(\gamma-1)RT} \cdot \left(\left(\dfrac{p_3(\theta)}{p_{2B}(\theta)} \right)^{\frac{2}{\gamma}} - \left(\dfrac{p_3(\theta)}{p_{2B}(\theta)} \right)^{\frac{\gamma+1}{\gamma}} \right)}, \text{if } \mathrm{p_{2B}}(\theta) > \mathrm{p_3}(\theta) \\ -\mu_{2B\_3} A_{2B\_3}(\theta) p_3(\theta) \sqrt{\dfrac{2\gamma}{(\gamma-1)RT} \cdot \left(\left(\dfrac{p_{2B}(\theta)}{p_3(\theta)} \right)^{\frac{2}{\gamma}} - \left(\dfrac{p_{2B}(\theta)}{p_3(\theta)} \right)^{\frac{\gamma+1}{\gamma}} \right)}, \text{if } \mathrm{p_3}(\theta) > \mathrm{p_{2B}}(\theta) \end{cases}$$

(4.19)

$G_{2B\_3}(\theta), kg/s$ - Mass flow rate from transfer duct (B) to cylinder;

$\mu_{2B\_3} = 0.8$ - Coefficient evaluating surface roughness of transfer port, engine parameter (17).

$A_{2B\_3}(\theta), m^2$ - Transfer port (B) area, which is expressed by equation (4.20);

$p_{2B}(\theta), Pa$ - Pressure in transfer duct (B);

$p_3(\theta), Pa$ - Pressure in cylinder.

Transfer port area function corresponding to crankshaft angle

$$A_{2B\_3}(\theta) = \begin{cases} 0, \text{if } 0 < \theta \le \theta_{tr1} = \pi - \dfrac{\alpha_{tr}}{2} \\ W_{2B\_3}\left(-L_{ct}\cos(\theta) - \sqrt{L_{cr}^2 - L_{ct}^2 \sin(\theta)^2} + L_{ct}\cos(\theta) + \sqrt{L_{cr}^2 - L_{ct}^2 \sin(\theta)^2} \right), \text{if } \theta_{tr1} < \theta \le \theta_{tr2} = \pi + \dfrac{\alpha_{tr}}{2} \\ 0, \text{if } \theta_{\mathrm{tr2}} < \theta < 2\pi \end{cases}$$

(4.20)

$A_{2B\_3}(\theta), m^2$ - Transfer port (B) area;

$W_{2A\_3} = 7 \times 10^{-3} m$ - Width of the first transfer port, engine parameter (15);

$\alpha_{tr} = 136^0 = 2.3736 rad$ - Transfer timing, engine parameter (18).

4.7 Equation of pressure changes in the transfer duct (C), $\dot{p}_{2C}$

$$\dot{p}_{2C}(\theta) = \frac{\gamma R T_{2C}}{V_{2C} \cdot \omega}\left(G_{1\_2C}(\theta) - G_{2C\_3}(\theta)\right) \tag{4.21}$$

$\dot{p}_{2C}(\theta)$ - Change of pressure in transfer duct (C);

$\gamma = 1.4$ - Specific heat ratio of air;

$R = 287 J/(kg \cdot K)$ - Gas constant;

$V_{2C} = width \times length \times thickness = 8 \times 4 \times 20 = 560 mm^3 = 640 \times 10^{-9} m^3$ - Transfer duct volume (C), engine parameter (22);

$T_{2C} = 333K$ - Temperature in first transfer duct (C).

Flow mass rate function, $G_{2C\_3}(\theta)$

$$G_{2C\_3}(\theta) = \begin{cases} \mu_{2C\_3} A_{2C\_3}(\theta) p_{2C}(\theta) \sqrt{\dfrac{2\gamma}{(\gamma-1)RT} \cdot \left(\left(\dfrac{p_3(\theta)}{p_{2C}(\theta)}\right)^{\frac{2}{\gamma}} - \left(\dfrac{p_3(\theta)}{p_{2C}(\theta)}\right)^{\frac{\gamma+1}{\gamma}} \right)}, \text{if } \mathrm{p}_{2C}(\theta) > \mathrm{p}_3(\theta) \\ -\mu_{2C\_3} A_{2C\_3}(\theta) p_3(\theta) \sqrt{\dfrac{2\gamma}{(\gamma-1)RT} \cdot \left(\left(\dfrac{p_{2C}(\theta)}{p_3(\theta)}\right)^{\frac{2}{\gamma}} - \left(\dfrac{p_{2C}(\theta)}{p_3(\theta)}\right)^{\frac{\gamma+1}{\gamma}} \right)}, \text{if } \mathrm{p}_3(\theta) > \mathrm{p}_{2C}(\theta) \end{cases}$$

(4.22)

$G_{2C\_3}(\theta), kg/s$ - Mass flow rate from transfer duct (B) to cylinder;

$\mu_{2C\_3} = 0.8$ - Coefficient evaluating surface roughness of transfer port, engine parameter (17).

$A_{2C\_3}(\theta), m^2$ - Transfer port (B) area, which is expressed by equation (4.23);

$p_{2C}(\theta), Pa$ - Pressure in transfer duct (B);

$p_3(\theta), Pa$ - Pressure in cylinder.

Transfer port area function corresponding to crankshaft angle

$$A_{2C\_3}(\theta)=\begin{cases} 0,\text{ if } 0<\theta\le\theta_{tr1}=\pi-\dfrac{\alpha_{tr}}{2} \\ W_{2C\_3}\left(-L_{ct}\cos(\theta)-\sqrt{L_{cr}^2-L_{ct}^2\sin(\theta)^2}+L_{ct}\cos(\theta)+\sqrt{L_{cr}^2-L_{ct}^2\sin(\theta)^2}\right),\text{ if } \theta_{tr1}<\theta\le\theta_{tr2}=\pi+\dfrac{\alpha_{tr}}{2} \\ 0,\text{ if } \theta_{tr2}<\theta<2\pi \end{cases}$$

(4.23)

$A_{2B\_3}(\theta), m^2$ - Transfer port (B) area;

$W_{2A\_3}=7\times10^{-3}m$ - Width of the first transfer port, engine parameter (15);

$\alpha_{tr}=136^0=2.3736rad$ - Transfer timing, engine parameter (18).

4.8 Equation of pressure changes in the cylinder, $\dot{p}_3$

$$\dot{p}_3(\theta)=\frac{\gamma RT_3}{V_3(\theta)\cdot\omega}\left(G_{2A\_3}(\theta)+G_{2B\_3}(\theta)+G_{2C\_3}(\theta)-G_{3\_4}(\theta)\right)-\frac{\gamma p_3(\theta)}{V_3(\theta)\cdot\omega}\dot{V}_3(\theta) \tag{4.24}$$

$\dot{p}_3(\theta)$ - Change of pressure in cylinder;

$T_3=1000K$ - Cylinder temperature;

$p_3(\theta), Pa$ - Pressure depending on crankshaft angle in cylinder.

Cylinder volume function, $V_3(\theta)$

$$V_3(\theta)=V_{31}+A_p\left(L_{cr}+L_{ct}(1-\cos(\theta))-\sqrt{L_{cr}^2-(L_{ct}\sin(\theta))^2}\right) \tag{4.25}$$

$V_3(\theta),\ m^3$ - Function of cylinder volume depending on crankshaft angle;

$V_{31}=0.19\times10^{-6}m^3$ - Cylinder volume (volume of combustion chamber) at TDC, engine parameter (29);

Flow mass rate function, $G_{3\_4}(\theta)$

$$G_{3\_4}(\theta)=\begin{cases} \mu_{3\_4}A_{3\_4}(\theta)p_3(\theta)\sqrt{\dfrac{2\gamma}{(\gamma-1)RT}\cdot\left(\left(\dfrac{p_4(\theta)}{p_3(\theta)}\right)^{\frac{2}{\gamma}}-\left(\dfrac{p_4(\theta)}{p_3(\theta)}\right)^{\frac{\gamma+1}{\gamma}}\right)},\text{ if } \mathrm{p}_3(\theta)>\mathrm{p}_4(\theta) \\ -\mu_{3\_4}A_{3\_4}(\theta)p_4(\theta)\sqrt{\dfrac{2\gamma}{(\gamma-1)RT}\cdot\left(\left(\dfrac{p_3(\theta)}{p_4(\theta)}\right)^{\frac{2}{\gamma}}-\left(\dfrac{p_3(\theta)}{p_4(\theta)}\right)^{\frac{\gamma+1}{\gamma}}\right)},\text{ if } \mathrm{p}_4(\theta)>\mathrm{p}_3(\theta) \end{cases}$$

(4.26)

$G_{3\_4}(\theta), kg/s$ - Mass flow rate from cylinder to exhaust pipe;

$\mu_{3\_4} = 0.8$ - Coefficient evaluating surface roughness of exhaust port, engine parameter;

$p_4(\theta), Pa$ - Pressure in exhaust pipe first section.

Exhaust port area function corresponding to crankshaft angle

$$A_{3\_4}(\theta) = \begin{cases} 0, \text{if } 0 < \theta \leq \pi - \frac{\alpha_{ex}}{2} \\ W_{3\_4}\left(-L_{ct}\cos(\theta) - \sqrt{L_{cr}^2 - L_{ct}^2\sin(\theta)^2} + L_{ct}\cos(\theta) + \sqrt{L_{cr}^2 - L_{ct}^2\sin(\theta)^2}\right), \text{if } \theta_{tr1} < \theta \leq \theta_{tr2} = \pi + \frac{\alpha_{ex}}{2} \\ 0, \text{if } \theta_{\text{tr2}} < \theta < 2\pi \end{cases}$$

(4.27)

$A_{3\_4}(\theta), m^2$ - Exhaust port area depending on crankshaft angle;

$W_{4\_5} = 11 \times 10^{-3} m$ - Width of the exhaust port, engine parameter (26);

$\alpha_{ex} = 196^0 = 3.4208 rad$ - Exhaust timing, engine parameter (27).

Function of cylinder volume change, $\dot{V}_3(\theta)$

$$\dot{V}_3(\theta) = A_p\left(L_{ct}\sin(\theta) + \frac{L_{ct}^2\sin(\theta)\cos(\theta)}{\sqrt{L_{cr}^2 - (L_{ct}\sin(\theta))^2}}\right) \tag{4.28}$$

$\dot{V}_3(\theta)$ - Function of crankcase volume change.

4.9 Equation of pressure changes in the first power pipe part, $\dot{p}_4$

$$\dot{p}_4(\theta) = \frac{\gamma R T_4}{V_4}\left(G_{3\_4}(\theta) - G_{4\_5}(\theta)\right) \tag{4.29}$$

$\dot{p}_4(\theta)$ - Change of pressure in first part of power pipe;

$V_4 = 10.9 \times 10^{-6} m^3$ - First power pipe part volume, engine parameter (31);

$T_5 = 800K$ - Temperature in first power pipe part.

Flow mass rate function, $G_{4\_5}(\theta)$

$$G_{4\_5}(\theta)=\begin{cases}\mu_{4\_5}A_{4\_5}p_4(\theta)\sqrt{\dfrac{2\gamma}{(\gamma-1)RT}\cdot\left(\left(\dfrac{p_5(\theta)}{p_4(\theta)}\right)^{\frac{2}{\gamma}}-\left(\dfrac{p_5(\theta)}{p_4(\theta)}\right)^{\frac{\gamma+1}{\gamma}}\right)}, \text{if } \mathrm{p}_4(\theta)>\mathrm{p}_5(\theta)\\ -\mu_{4\_5}A_{4\_5}p_5(\theta)\sqrt{\dfrac{2\gamma}{(\gamma-1)RT}\cdot\left(\left(\dfrac{p_4(\theta)}{p_5(\theta)}\right)^{\frac{2}{\gamma}}-\left(\dfrac{p_4(\theta)}{p_5(\theta)}\right)^{\frac{\gamma+1}{\gamma}}\right)}, \text{if } \mathrm{p}_5(\theta)>\mathrm{p}_4(\theta)\end{cases}$$

(4.30)

$G_{4\_5}(\theta), kg/s$ - Mass flow rate from first part of power pipe to second part;

$\mu_{4\_5}=0.8$ - Coefficient evaluating surface roughness of exhaust pipe, engine parameter (39).

$p_5(\theta), Pa$ - Pressure in second part of power pipe;

$A_{4\_5}=2.18\times10^{-4}m^2$ - Cross-section area in junction of firs and second part of power pipe.

4.10 Equation of pressure changes in the second power pipe part, $\dot{p}_5$

$$\dot{p}_5(\theta)=\frac{\gamma RT_5}{V_5\cdot\omega}\left(G_{4\_5}(\theta)-G_{5\_5}(\theta)\right) \qquad (4.31)$$

$\dot{p}_5(\theta)$ - Change of pressure in second part of power pipe;

$V_5=53.2\times10^{-6}m^3$ - Second power pipe part volume, engine parameter;

$T_5=500K$ - Temperature in second power pipe part.

Flow mass rate function $G_{5\_6}(\theta)$

$$G_{5\_6}(\theta)=\begin{cases}\mu_{5\_6}A_{5\_6}p_5(\theta)\sqrt{\dfrac{2\gamma}{(\gamma-1)RT}\cdot\left(\left(\dfrac{p_6(\theta)}{p_7(\theta)}\right)^{\frac{2}{\gamma}}-\left(\dfrac{p_6(\theta)}{p_7(\theta)}\right)^{\frac{\gamma+1}{\gamma}}\right)}, \text{if } \mathrm{p}_5(\theta)>\mathrm{p}_6(\theta)\\ -\mu_{5\_6}A_{5\_6}p_6(\theta)\sqrt{\dfrac{2\gamma}{(\gamma-1)RT}\cdot\left(\left(\dfrac{p_6(\theta)}{p_5(\theta)}\right)^{\frac{2}{\gamma}}-\left(\dfrac{p_5(\theta)}{p_6(\theta)}\right)^{\frac{\gamma+1}{\gamma}}\right)}, \text{if } \mathrm{p}_6(\theta)>\mathrm{p}_5(\theta)\end{cases}$$

(4.32)

$G_{5\_6}(\theta), kg/s$ - Mass flow rate from second part of power pipe to third part;

$\mu_{5\_6} = 0.8$ - Coefficient evaluating surface roughness, engine parameter (39).

$p_6(\theta), Pa$ - Pressure in third part of power pipe;

$A_{5\_6} = 8.04 \times 10^{-4} m^2$ - Cross-section area in junction of second and third part of power pipe.

4.11 Equation of pressure changes in the third power pipe part, $\dot{p}_6$

$$\dot{p}_6(\theta) = \frac{\gamma R T_6}{V_6}\left(G_{5\_6}(\theta) - G_{6\_7}(\theta)\right) \tag{4.33}$$

$\dot{p}_6(\theta)$ - Change of pressure in third part of power pipe;

$V_6 = 20.6 \times 10^{-6} m^3$ - Third power pipe part volume, engine parameter;

$T_6 = 400K$ - Temperature in third power pipe part.

Flow mass rate function, $G_{6\_7}(\theta)$

$$G_{6\_7}(\theta) = \begin{cases} \mu_{6\_7} A_{6\_7} p_6(\theta) \sqrt{\dfrac{2\gamma}{(\gamma-1)RT} \cdot \left(\left(\dfrac{p_7(\theta)}{p_6(\theta)}\right)^{\frac{2}{\gamma}} - \left(\dfrac{p_7(\theta)}{p_6(\theta)}\right)^{\frac{\gamma+1}{\gamma}} \right)}, \text{if } \mathrm{p}_6(\theta) > \mathrm{p}_7(\theta) \\ -\mu_{6\_7} A_{6\_7} p_7(\theta) \sqrt{\dfrac{2\gamma}{(\gamma-1)RT} \cdot \left(\left(\dfrac{p_6(\theta)}{p_7(\theta)}\right)^{\frac{2}{\gamma}} - \left(\dfrac{p_6(\theta)}{p_7(\theta)}\right)^{\frac{\gamma+1}{\gamma}} \right)}, \text{if } \mathrm{p}_7(\theta) > \mathrm{p}_6(\theta) \end{cases}$$

(4.35)

$G_{6\_7}(\theta), kg/s$ - Mass flow rate from third part of power pipe to fourth part;

$\mu_{6\_7} = 0.8$ - Coefficient evaluating surface roughness, engine parameter (39).

$p_7(\theta), Pa$ - Pressure in fourth part of power pipe;

$A_{6\_7} = 0.32 \times 10^{-4} m^2$ - Cross-section area in junction of third and fourth part of power pipe.

4.12 Equation of pressure changes in the fourth power pipe part, $\dot{p}_7$

$$\dot{p}_7(\theta) = \frac{\gamma R T_7}{V_7}\left(G_{6\_7}(\theta) - G_{7\_0}(\theta)\right) \tag{4.36}$$

$\dot{p}_7(\theta)$ - Change of pressure in fourth part of power pipe;

$V_7 = 1.9 \times 10^{-6} m^3$ - Fourth power pipe part volume, engine parameter;

$T_7 = 400K$ - Temperature in fourth power pipe part.

Flow mass rate function $G_{7\_0}(\theta)$

$$G_{7\_0}(\theta) = \begin{cases} \mu_{7\_0} A_{7\_0} p_7(\theta) \sqrt{\frac{2\gamma}{(\gamma - 1)RT} \cdot \left(\left(\frac{p_0}{p_7(\theta)} \right)^{\frac{2}{\gamma}} - \left(\frac{p_0}{p_7(\theta)} \right)^{\frac{\gamma+1}{\gamma}} \right)}, \text{if } \mathrm{p}_7(\theta) > \mathrm{p}_0 \\ -\mu_{7\_0} A_{7\_0} p_0 \sqrt{\frac{2\gamma}{(\gamma - 1)RT} \cdot \left(\left(\frac{p_7(\theta)}{p_0} \right)^{\frac{2}{\gamma}} - \left(\frac{p_7(\theta)}{p_0} \right)^{\frac{\gamma+1}{\gamma}} \right)}, \text{if } \mathrm{p}_0 > \mathrm{p}_7(\theta) \end{cases};$$

(4.37)

$G_{7\_1}(\theta), kg/s$ - Mass flow rate from fourth part of power pipe to atmosphere;

$\mu_{7\_0} = 0.8$ - Coefficient evaluating surface roughness, engine parameter (39);

$p_0 = 101325Pa$ - Pressure of atmosphere;

$A_{7\_0} = 0.32 \times 10^{-4} m^2$ - Cross-section area in junction of power pipe and atmosphere.

4.13 Runge-Kutta method

As it was noted before all the differential equations are solved by Runge-Kutta method. This subchapter will be a guide to it. I reviewed many books in order to learn it. It is rather simple method, but in some books it was described so cumbersome, that it was impossible to understand it. I found best description of this method by Van Iwaarden (1985).

The methods developed by Runge and Kutta are particular set of self-starting numerical methods; that is, sufficient initial information is contained in the problem to start integration. The simple Euler method is a first-order Runge-Kutta method, and improved Euler is a Runge-Kutta of order two. However, greatly increased efficiency is attained if further refinements are made to these early methods.

In the Euler method we based our replacement value in the integrand only on the value of the function at the last computed point. The improvement made in the second-order method was to look ahead in the direction field for an additional value, which was then averaged with the current value to create replacement. Our third-order method will combine two slope values in the field ahead with the current value, while the fourth order method will calculate three slope values in the direction field ahead of the point for combination with current information and a subsequent replacement.

We look first at the actual equations used to move from current point $(x_n; Y_n)$ in the numerical approximate solution to the next point $(x_{n+1}; Y_{n+1})$.

The formula for the fourth-order Runge-Kutta method is:

$$Y_{n+1} = Y_n + \frac{1}{6}(k_1 + 2k_2 + 2k_3 + k_4) \tag{4.38}$$

Where

$$k_1 = h \cdot f(x_n, Y_n)$$

$$k_2 = h \cdot f\left(x_n + \frac{1}{2}h, Y_n + \frac{1}{2}k_1\right)$$

$$k_3 = h \cdot f\left(x_n + \frac{1}{2}h, Y_n + \frac{1}{2}k_2\right)$$

$$k_4 = h \cdot f(x_n + h, Y_n + k_3)$$

The point at which computations are made are those with x-coordinates $x_n, x_n + \frac{1}{2}h$ (twice), and $x_n + h$. When using the Runge-Kutta IV method, we need to recalculate the entire set of four k values each time we move forward to a new point. Runge-Kutta IV requires slightly more machine work than do lower level methods, but its increased accuracy compensates adequately for that.

A general study of these methods indicates that a Runge-Kutta scheme of order p requires exactly p function evaluations if $p \leq 4$, and more than p evaluations if $p>4$. Thus the Runge-Kutta method usually gives optimal result.

5. CALCULATION RESULTS

In this chapter there will be represented results of the calculation program. Results will be discussed and compared with results from Blair (1996) book "Design and simulation of the two-stroke engine". In later subchapters there will be represented how can be optimized engine by using the program.

The calculations are done at 40000 rpm of the engine. According my race experience it is optimum revolutions for this kind of engine.

Runge-Kutta IV method is used for the calculation of the differential equations.

The output results of the program are pressures at each part of the engine and mass flows. From the equations described in the fourth chapter there is clear, that inside the program there is port timing functions and crankcase volume function. These things are like engine parameters, but very important for gas flow and especially for the calculations. The start of this chapter begins from the plots of theses functions.

5.1 Port timing in the simulated engine

As it was described earlier, the engine uses disc type inlet valve, the transfer and exhaust ports are controlled by piston. The formulas describing port area versus crankshaft angle is described in the fourth chapter, for the inlet valve formula (4.10), transfer ports formulas (4.17), (4.20) and (4.23), for the exhaust port timing formula (4.27).

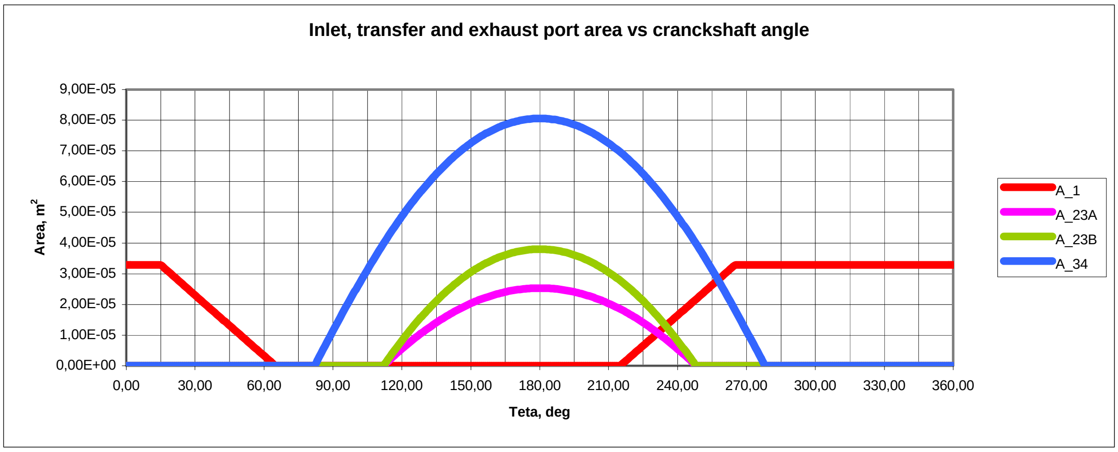

Figure 5.1.1 Inlet, transfer and exhaust port area vs. crankshaft angle

In the figure 5.1.1 we can follow port timing. As it was mentioned in earlier chapters, all the calculations begin from TDC. We start that crankshaft angle θ is zero and the piston position is in TDC, when $\theta = 180^0$ then piston is at BDC, and again when $\theta = 360^0$ the piston returns to TDC. The graph in figure 5.1.1 represents the ports areas according to crankshaft angle through one revolution. Because the engine is reciprocating system, for the next revolutions everything is the same.

Let's start from inlet port timing. In the figure it is of the red line and denoted by A_1. The notation is done in such manner that you can follow it in calculation scheme represented in subchapter 4.2. At the TDC the port is still open at the maximum area. It starts to decrease after 15^0 till final closing at 65^0. According engine data the inlet closes 65^0 after TDC. All the time till piston is going further till BDC and little bit later, the inlet is closed. The inlet starts to open at 35^0 after BDC. In the graph it means $180^0 + 35^0 = 215^0$. Fully opens somewhere at 265^0. And is open till the next revolution 65^0. The maximum area of the inlet is $3.29 \times 10^{-5} m^2 = 32.9 mm^2$.

In the graph there are represented only two transfer ports, the back and one side transfer port. The second side transfer port is the same as first one, because engine is designed symmetrically. There is no need of its representations, because it will overlay the first one. Usually transfer port and exhaust port timing is described simply by numbers: transfer port timing 136^0 and exhaust port timing 196^0. These numbers are very clear for experienced user, but what does it means practically. It means how long these ports are opened, but we need some reference point. For those who are experienced it is clear that both transfer and exhaust ports are opened symmetrically according BDC. If we analyze engine work from TDC we need to convert these values to reference point TDC. So if transfer timing is 136^0, it means that transfer port opens $180^0 - transfer\, timing / 2 = 180^0 - 136^0 / 2 = 112^0$ and closes $180^0 + transfer\, timing / 2 = 180^0 + 136^0 / 2 = 248^0$. The transfer port area of the back transfer port is denoted by pink color and side transfer port by green color. Till 112^0 the transfer port is closed and later it starts to open. Fully opens at BDC and than starts to close till it is fully closed at 248^0. The maximum areas of these ports differ because of their geometry. The side port is wider and for this reason it has bigger maximum area. The maximum area of the back transfer port is $2.53 \times 10^{-5} m^2 = 25.3 mm^2$, the maximum area of the side transfer port is

$3.8\times10^{-5}m^2 = 38.0mm^2$. The maximum area of all transfer ports is their sum: $25.3 + 2*38.0 = 101.3mm^2$.

The exhaust port timing is 196^0. Exhaust port opens $180^0 - exhaust\,ti\min g/2 = 180^0 - 196^0/2 = 82^0$ and closes $180^0 + exhaust\,ti\min g/2 = 180^0 6 + 196^0/2 = 278^0$. The maximum area at BDC of the exhaust port is $8.05\times10^{-5}m^2 = 80.5mm^2$.

When the engine tuning is done transfer and exhaust timing is changed. Usually all the micro engines on the market for the model car race competition are sold with very low timing: transfer timing is about 128^0, whereas exhaust timing is about 184^0. This is because they are prepared for wide range application. For the tether model car race timing is increased enormously as you can see from my calculations. During the tuning it is important to know how to measure timing. In practice it is done easily by mounting protractor to crankshaft and looking by eye when piston opens port and closes it. Measuring angle between these two points we will get transfer or exhaust timing.

5.2 Representation of the crankcase and cylinder volume function

Because crankcase and cylinder volume functions are used in calculation there is important to show that they are calculated correctly. The crankcase and cylinder volumes according the crankshaft angle are represented in figure 5.2.1.

Crankcase volume is denoted by V_1 and cylinder volume is denoted by V_3 volume. The crankcase volume at TDC is $15cm^3 = 1.5\times10^{-5}m^3$ from the engine data and cylinder volume is $0.19cm^3 = 0.19\times10^{-6}m^3$. From general assumption without any doubt we can state that functions of the cylinder volume and crankcase volume are opposite only differs initial volumes. The figure 5.2.1 confirms our statement. As the crankcase volume decreases because piston goes from TDC to BDC the cylinder volume at the same time increases. The cylinder volume at the TDC corresponds to combustion chamber volume.

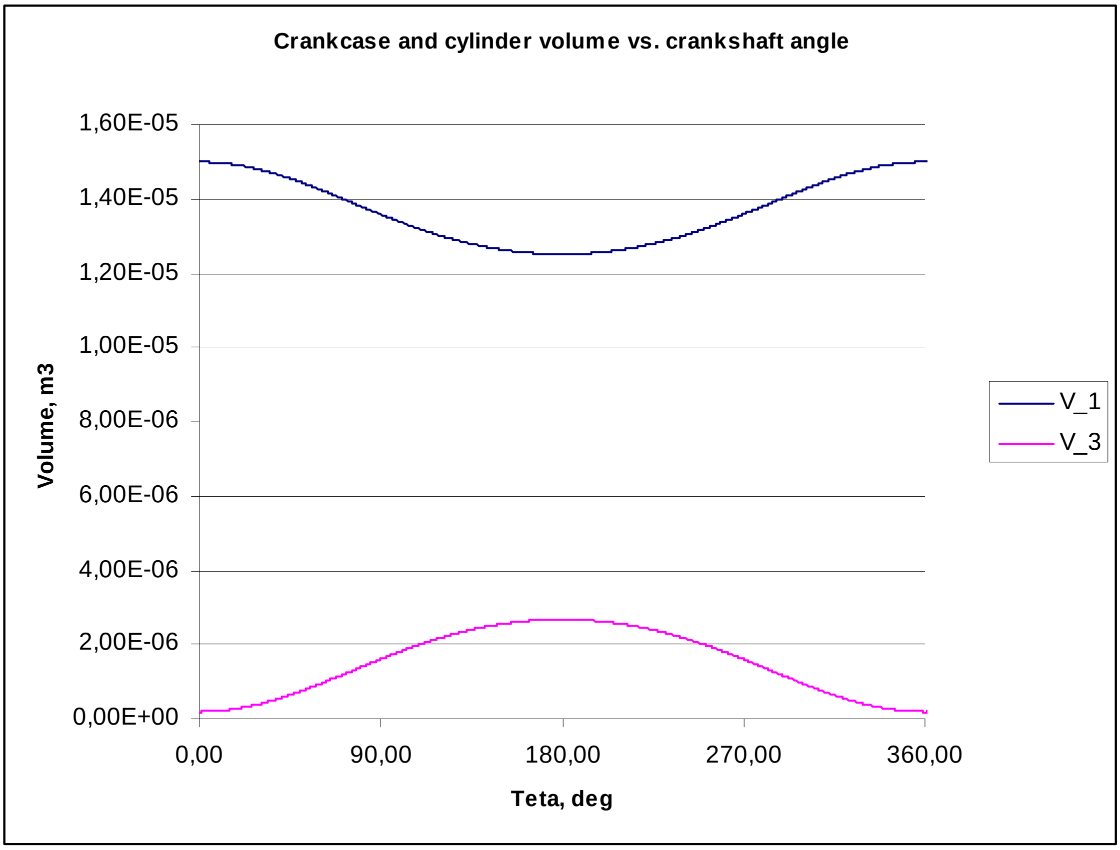

Figure 5.2.1 Crankcase and cylinder volume functions according crankshaft angle

In the practice for the engine performance it is quite important parameter crankcase compression ratio - CR_{CC}:

$$CR_{CC} = \frac{V_{CC} + V_{SV}}{V_{CC}} \tag{5.1}$$

V_{CC}, m^3 - Crankcase clearance volume (volume at BDC);

V_{SV}, m^3 - Swept volume of the engine.

According engine data we can calculate that my engine's crankcase compression volume is:

$V_{CC} = 12.5 \times 10^{-5} m^3$ - Crankcase volume of my simulated engine;

$V_{SV} = 2.5 \times 10^{-5} m^3$ - Swept volume of my simulated engine;

$$CR_{CC} = \frac{12.5 \times 10^{-5} + 2.49 \times 10^{-5}}{12.5 \times 10^{-5}} = 1.20$$

According calculation there can be stated, that compression ratio of simulated engine is fairly low in comparison with usual two stroke engines. Below there is written quotation of Blair (1996).

While it is true that the higher this value becomes, the stronger is the crankcase pumping action, the actual numerical value is greatly fixed by the engine geometry of bore, stroke, connecting rod length and the interconnect value of flywheel diameter. In practical terms, it is rather difficult to organize the CR_{CC} value for a 50 cm^3 engine cylinder above 1.4 and almost physically impossible to design a 500 cm^3 engine cylinder to have a value less than 1.55. Therefore, for any given engine design the CR_{CC} characteristic is more heavily influenced by the choice of cylinder swept volume than by designer. It then behooves the designer to tailor the engine air-flow behavior around the crankcase pumping action, defined by the inherent CR_{CC} value emanating from cylinder size in question. There is some freedom of design action, and it is necessary for it to be taken in the correct direction.

According quotation there is clear, that the smaller the swept volume the smaller CR_{CC} value can be reached. Swept volume of micro engines is very small in comparison with motorcycle engines. Because of this fact it is naturally, that micro engines will have very small value of CR_{CC}.

In calculations there is used also derivatives of cylinder and crankcase volumes. There is represented plot of derivatives of cylinder and crankcase volumes in the figure 5.2.2.

The crankcase volume derivative is denoted by black color whereas cylinder with pink color. From the graph is seen, that volume changes of crankcase and cylinder are the same but opposite. The rates of changes are equal to zero at TDC and BDC and the highest values at somewhere 90^0 and 270^0 of the crankshaft. These facts confirm that derivatives are calculated well.

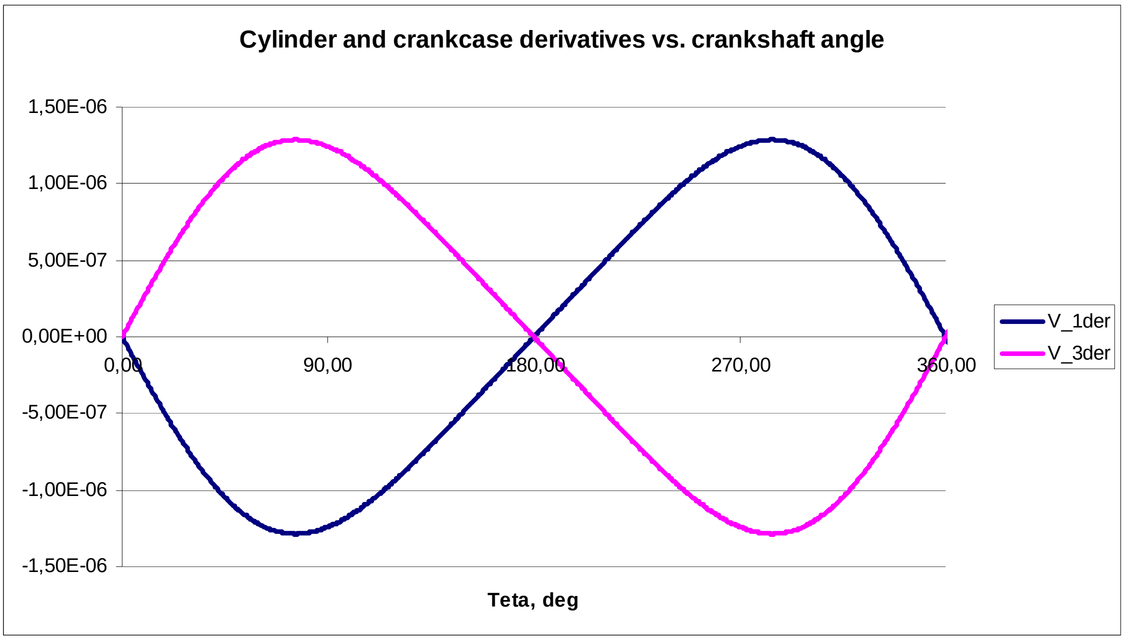

Figure 5.2.2 Cylinder and crankcase volume derivatives values through one revolution of crankshaft

5.3 The crankcase pressure

The main task of calculation program is to calculate pressures and mass flow inside all the parts of the engine. In this subchapter we will get a look to calculation results. Results are shown as graph of pressure changes during one revolution.

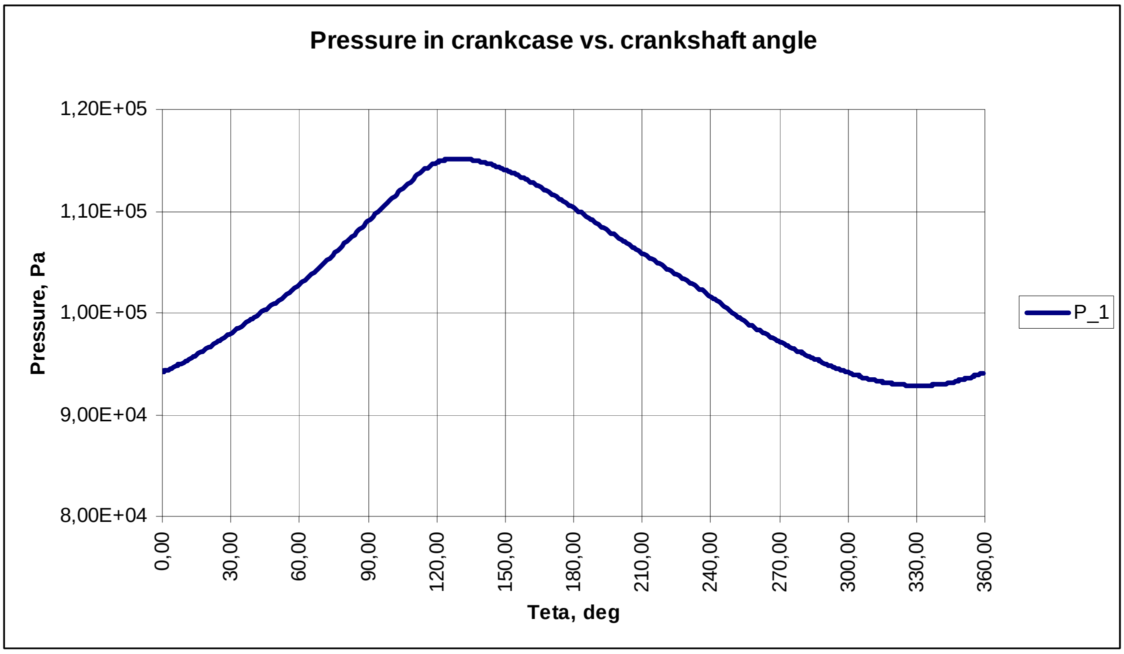

Figure 5.3.1 Pressure in crankcase during one crankshaft revolution at 40000 rpm

Why it is so important pressure diagrams for the engine design and tuning? This question can be answered only then the operation theory of engine is understood very well. The two-stroke engine is aspirated by atmospheric pressure. The flow inside the crankcase occurs only when pressure inside the crankcase is smaller than atmospheric. According this pressure diagram we can decide about inlet and transfer timing. Starting from TDC (when $\theta = 0^0$) the crankcase pressure is below atmospheric (100000 Pa) and inlet valve should be open till atmospheric pressure is reached. The atmospheric pressure is reached somewhere at 45^0 after TDC. According inlet design we have it closes only 65^0 after TDC. That results in the spillage out of crankcase to atmosphere. This is negative effect for the inlet process. Compression phase starts when inlet port is fully closed in the crankcase. From the figure 5.3.1 there is seen that compression starts a little bit earlier than inlet opens, this can be explained by gas dynamic effect, that compression is very fast and small amount of spillage does not influence it. The compression lasts till the transfer ports opens. From subchapter 5.1 we know that it occurs at 112^0 after TDC. But the pressure is still increasing in crankcase despite the transfer ports are opening. This is because piston still moves to BDC and decreases the volume of crankcase. Anyway somewhere at 130^0 after TDC the pressure in crankcase starts to decrease. The atmospheric pressure in crankcase is reached only at 249^0 after TDC. This can be explained by gas inertia and mass flow inertia. The piston moving after BDC increase the crankcase volume so vacuum starts in crankcase but as it was mentioned before, just at 249^0 or 69^0 after BDC. According design data the inlet opens 35^0 after BDC and this is too early and can be stated as serious mistake of the engine tuner. After opening inlet valve is open till 65^0 of the next revolution.

I did not done pressure measurements of the calculated pressures. For this reason I searched for information in order to compare my results with other authors. In comparison there is shown Figure 5.3.2 from Blair (1996). There is shown crankcase pressure curve of the chainsaw engine at 9600 rpm.

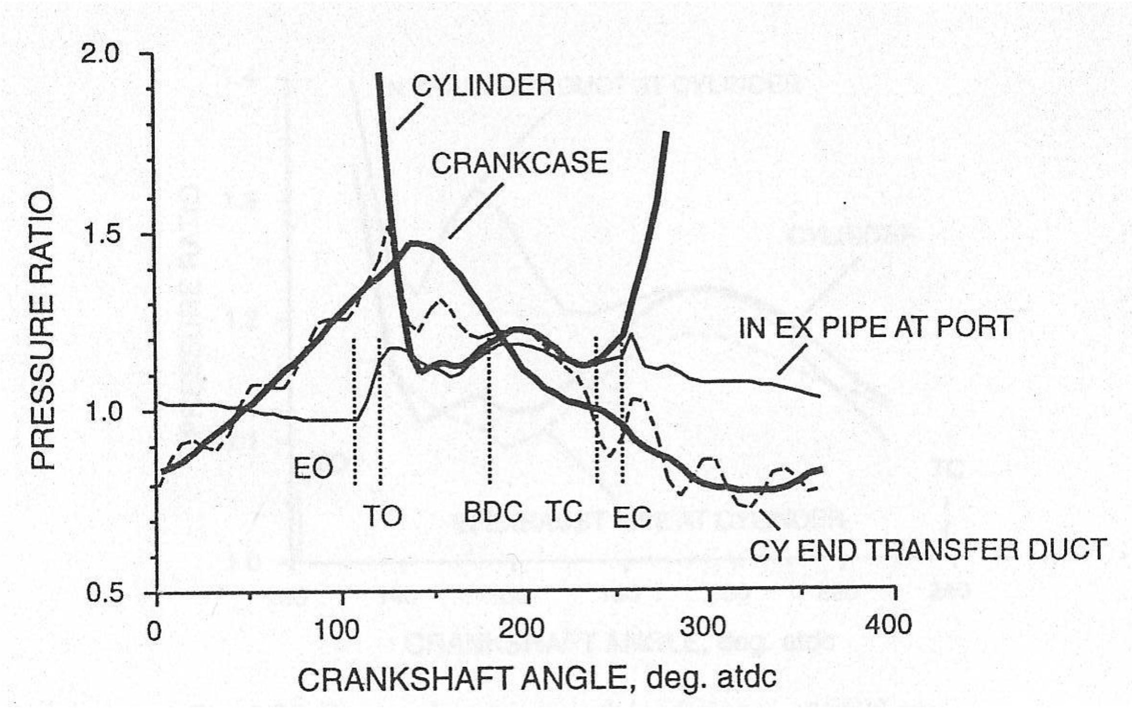

Figure 5.3.2 Crankcase and other pressures in chainsaw two-stroke engine according Blair (1996)

The shapes of pressure curves are similar in my calculations and Blair. In the figure 5.3.2 there is shown more pressure curves, but at the moment we will not discuss them. Blair gives pressures not in absolute value but in ratio with atmospheric pressure.

The maximum pressure in crankcase of my calculations is 115000 Pa. It means, that maximum pressure ratio is only 1.15. Obviously it is fairly low value in comparison with chainsaw pressure curve, but according crankcase compression ratio calculation of the model car engine $CR_{CC} = 1.20$. So maximum possible pressure ratio is 1.2 whereas according calculations we got 1.15, in my opinion it is more the true, than the lie.

5.4 Pressure of the transfer ducts

Pressures in transfer ducts are represented in figure 5.4.1. There are represented pressures of back transfer port and one side transfer duct. The pressure of the back duct is denoted by black color and of the side duct by pink color. Both curves overlay each other only small difference exists between 120^0 and 180^0 after TDC. This can be explained by different volume of the ducts. The acting behavior of both transfer ducts are the same. In my opinion processes in the transfer ducts are the same as in crankcase. So for the curve comments refer to previous subchapter.

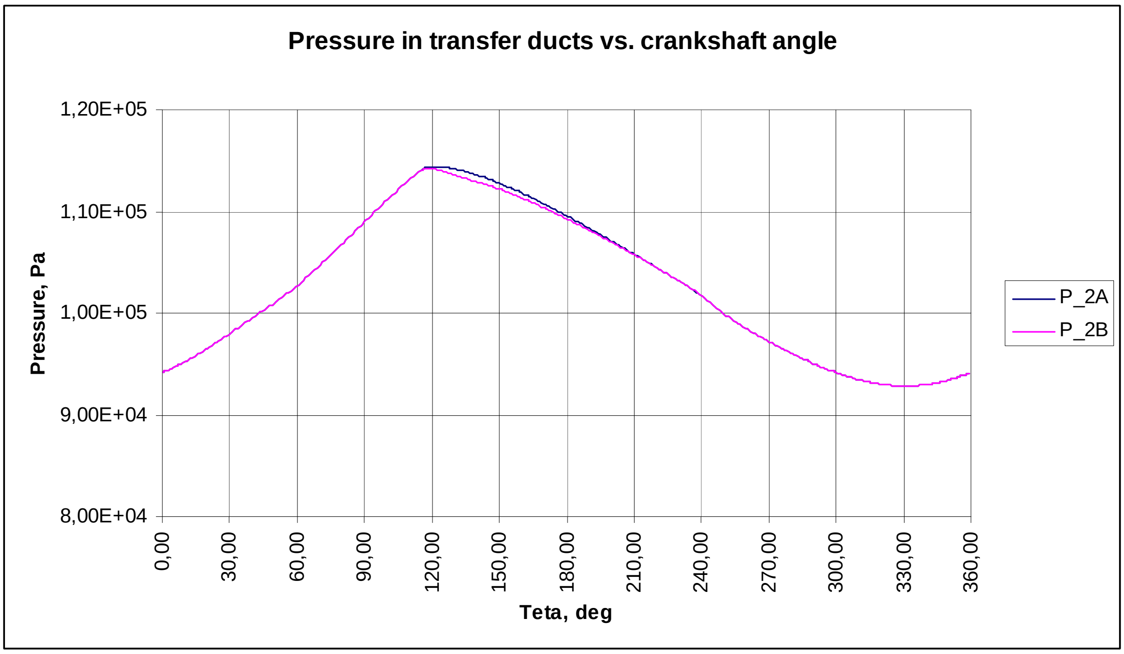

Figure 5.4.1 Pressure in the transfer ducts during one engine revolution at 40000 rpm

5.5 The cylinder pressure

The pressure in cylinder is the most interesting and important thing in all these calculations. In order to show cylinder pressure clearly there is done two graphs. The two graphs are done because cylinder pressure varies a lot and for this reason there can be difficulties to notice important things in it. In the figure 5.5.1 there is represented the full view of cylinder pressure.

From the figure 5.5.1 there is seen that pressure in cylinder reaches its maximum at TDC of 2230000 Pa=22.3 Bar, or the pressure ratio of 22.3. In my opinion it is normal compression ratio for the compression ignition engine type.

As it was mentioned in the beginning of this thesis the combustion process is not simulated. But anyway pressure diagram is corresponding to reality. For getting more confidence, that calculations are right I inserted cylinder pressure and temperature diagram of chainsaw engine from Blair (1996).

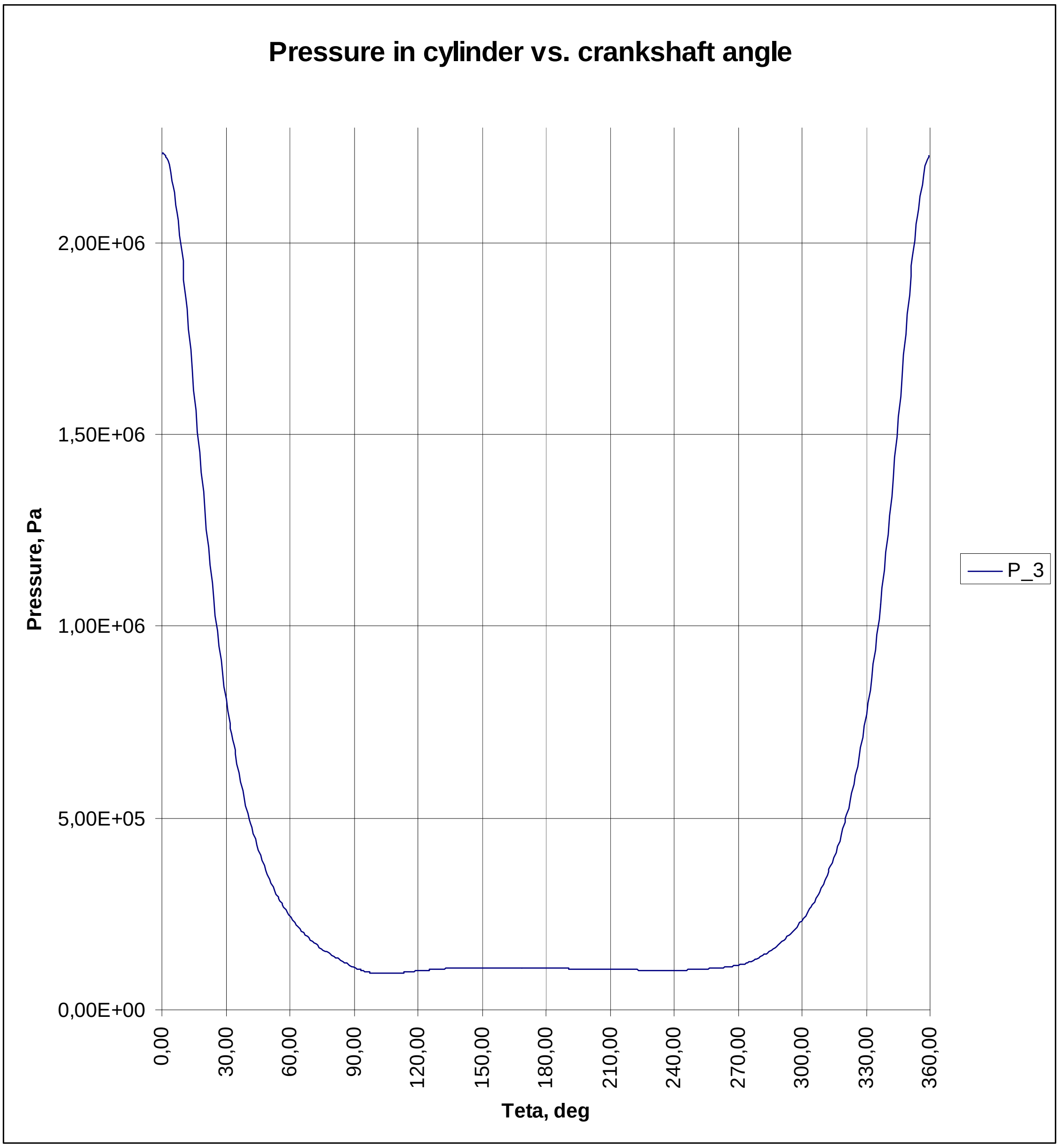

Figure 5.5.1 Pressure in cylinder of the engine of one revolution at 40000 rpm

When we speak about four-stroke engines the compression ratio is understand simply by ratio of combustion chamber and cylinder volume at BDC. In two-stroke engine theory the compression ratio can be understood in two ways and it is important to know what is going about. One of the ways that compression ratio means the same as for four-stroke engine and another one that it means so called trapped compression ratio. Trapped compression ratio means that the compression ratio is calculated between combustion chamber and cylinder volume after the exhaust port is closed.

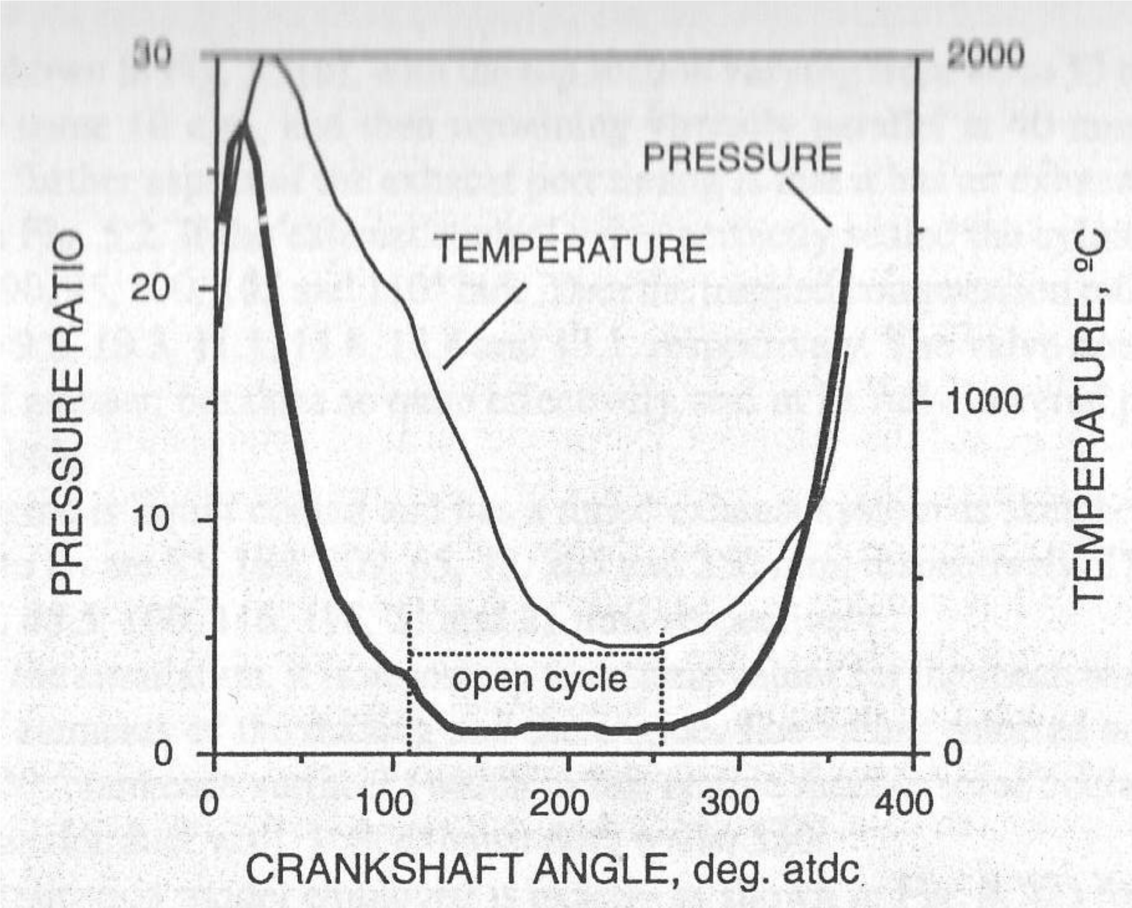

Figure 5.5.2 Pressure and temperature in the cylinder of chainsaw engine at 9600 rpm from Blair (1996)

$$CR_t = \frac{V_{ts} + V_{CV}}{V_{CV}} \tag{5.2}$$

CR_t - Trapped compression ratio;

V_{ts} - Trapped swept volume;

V_{CV} - combustion chamber volume;

According simulated engine design data:

$V_{ts} = 1.35cm^3$;

$V_{CV} = 0.19cm^3$;

$$CR_t = \frac{1.35 + 0.19}{0.19} = 8.11$$

The cylinder pressure figure 5.5.3 is the same one as first but magnified in order to see small pressure changes during the scavenging.

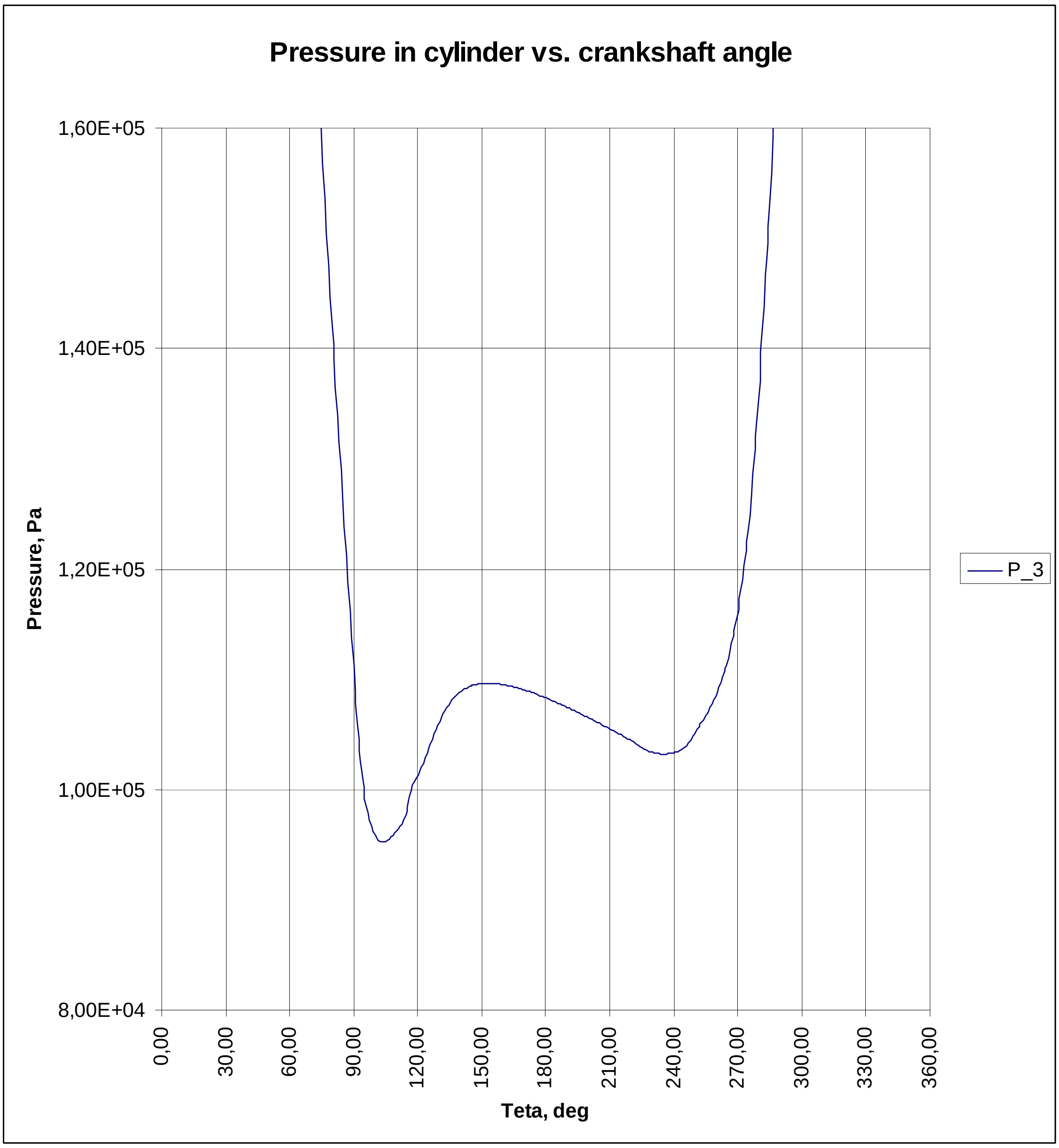

Figure 5.5.3 Pressure in cylinder of simulated engine at 40000 rpm

In the figure above there are clearly seen pressure changes during the scavenging process. According this graph there is seen some pressure drop below atmospheric pressure between 95^0 and110^0. This phenomenon cannot be explained as usual or logical. In my opinion it is shortage of the concentrated volume method and also the fact, that there is no combustion process during simulation. The simulation of the fast processes in cylinder can give vacuum effect or other unpredictable things. Without combustion simulation there is possible to happen for such things. Without taking into account the fact of this small pressure drop, the graph looks quite well. For comparison with very good result see picture 5.5.2.

As the piston goes down from TDC, the pressure decreases in cylinder. When piston crown opens exhaust port at 82^0 the cylinder becomes not closed system and pressure inside is influenced by exhaust system. Despite the illogical pressure drop discussed above the curve of pressure tends to act according normal conditions. When transfer ports open at 112^0 after TDC and until 150^0 the sudden pressure increase occur. During this period compressed air flows out of crankcase to the cylinder. This gives some increment for the short time, because at all this time there is exhaust port open. After 150^0 till the exhaust port closure there is small pressure drop till 240^0. This can be explained easily by the fact that till exhaust port is open there is impossible to compress air in the cylinder more than pressure in exhaust system is. In order to raise pressure in cylinder there is need to have higher pressure at exhaust system after BDC. If you remember the theory of racing engine exhaust system, it is designed especially for that. It gives suction effect till approximately BDC and high pressure “valve” after BDC till exhaust port closure. This looks very simple, only one thing is important, that it should happen at right time.

The calculations of pressures are intended for making decisions about port timings. From my experience in micro engine tuning timing is the most important thing in engine tuning. If the timing is wrong the engine immediately loose power. When engine is combined with power pipe, the timing is critical as pipe tuning also. They can work very well only when they are tuned for each other in other way they will disturb each other and the engine will work abnormally.

In order to tune exhaust timing the pressure diagram of the exhaust pipe is necessary. In the next chapter there will be represented exhaust pipe pressure diagrams and discussion about it. The system works well when it is connected well in any other case it will do only the disappointment.

5.6 Exhaust pipe pressure

The exhaust pipe design is very important for the successful two-stroke engine work. For the racing engine exhaust system is critical. The design of the exhaust system for the racing engine can increase power by 30% and at the same time wrong design can decrease by 50% leading in overheating or extremely low power of engine. In order to understand what happens inside the exhaust power there will be represented pressure diagrams. The most interesting thing in exhaust system how does it act and influences processes in cylinder.

In order to get more precise results I divided power exhaust pipe into four parts.

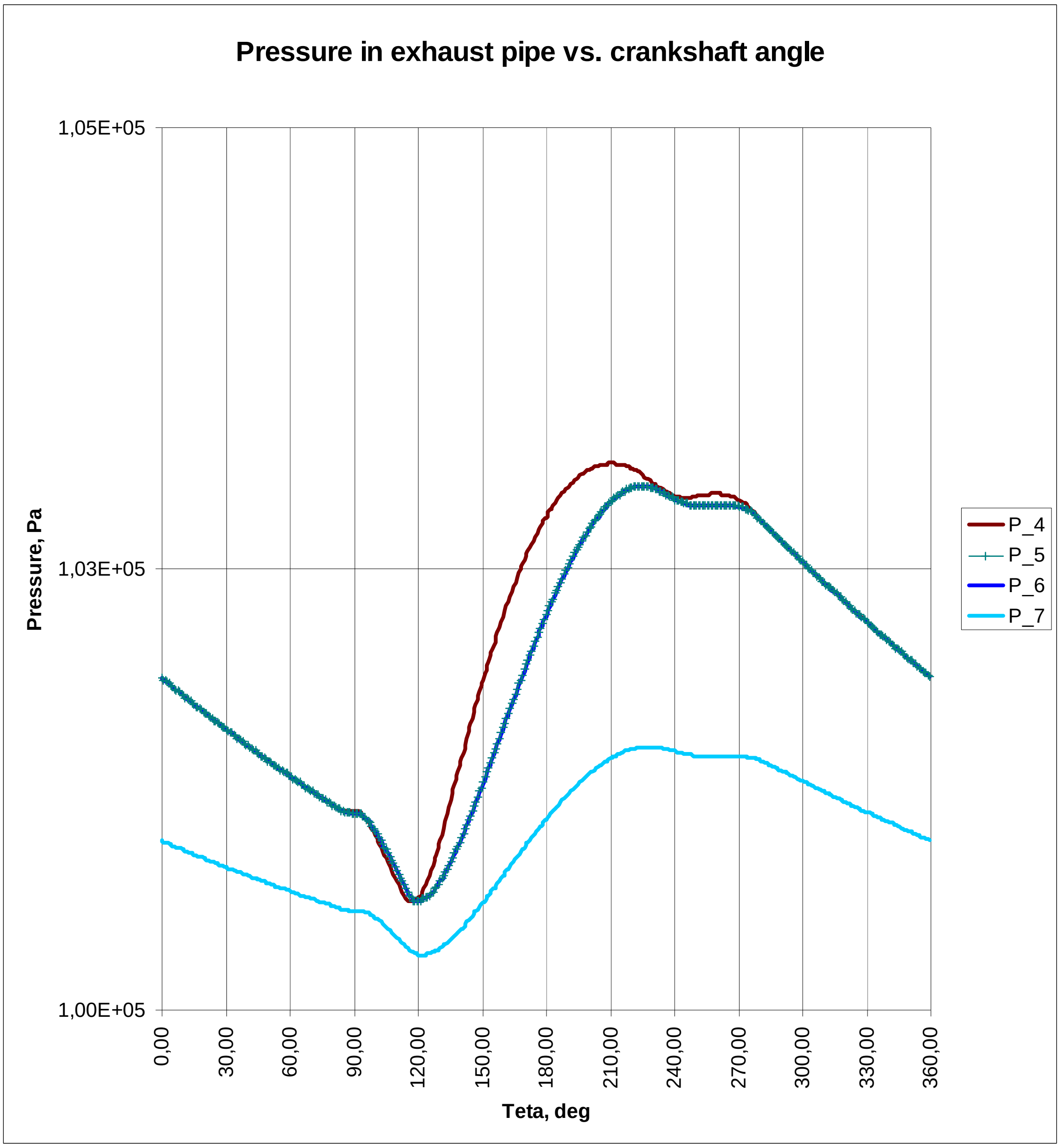

Figure 5.6.1 Pressure in exhaust pipe vs. crankshaft angle at 40000 rpm

In the figure 5.6.1 there is represented pressures in the exhaust pipe during one revolution. Notation of pressures is according calculation scheme represented in subchapter 4.2. The pressures in the second and third exhaust parts are the same and overlay each other (P_5 and P_6). The highest pressure changes are in the first part of exhaust pipe. Concentrated volume method gives static pressure values, for this reason we do not see pressure wave action. As it was mentioned earlier more sophisticated method for the pressure calculation in exhaust pipe is characteristic method, because it includes pressure wave actions.

All the pressures are decreasing till the pressure in cylinder starts to increase at 120^0 after TDC. Then there is big jump of pressure increment and it lasts till exhaust port closure 278^0 .

The pressure in the fourth part of the pipe is smaller and with smaller amplitude too. This is because the fourth part volume is quite small in comparison with other parts and the inlet and outlet areas are also very small. This part is used to adjust the static pressure in the pipe. The small output area does not allow high mass flow out of pipe and this is necessary to keep high static pressure. In the tuner's speech the fourth part of the pipe is called stinger. The stinger controls static pressure ant temperature in the pipe at the same time. The diameter of the stinger is very important and depends on the ambient conditions. If the ambient temperature is low the diameter of the stinger's hole should be smaller and vice versa. Also if the static pressure in the pipe is too high the stinger diameter should be increased.

In order to describe better pressure action in exhaust pipe I did one more figure 5.6.2 which includes both pressures of exhaust pipe first part and cylinder. The unnatural pressure drop in cylinder described in the previous subchapter influences the pressure in the pipe too. When the pressure rises in cylinder, the pressure rises in exhaust pipe too, because the exhaust port is open (120^0 - 200^0 after TDC). 150^0 after TDC the pressure in cylinder decreases but in the exhaust pipe it is still increasing till the pressure values of cylinder and exhaust pipe become similar. Somewhere at 240^0 the pressure in cylinder starts to increase because of the piston movement to TDC and here the exhaust pressure in the pipe helps for this action. The pressure in pipe is quite high and prevents from spillage out of cylinder. After exhaust port closure, the pressure in the pipe decreases, because it is connected through the fourth part of pipe with atmosphere.

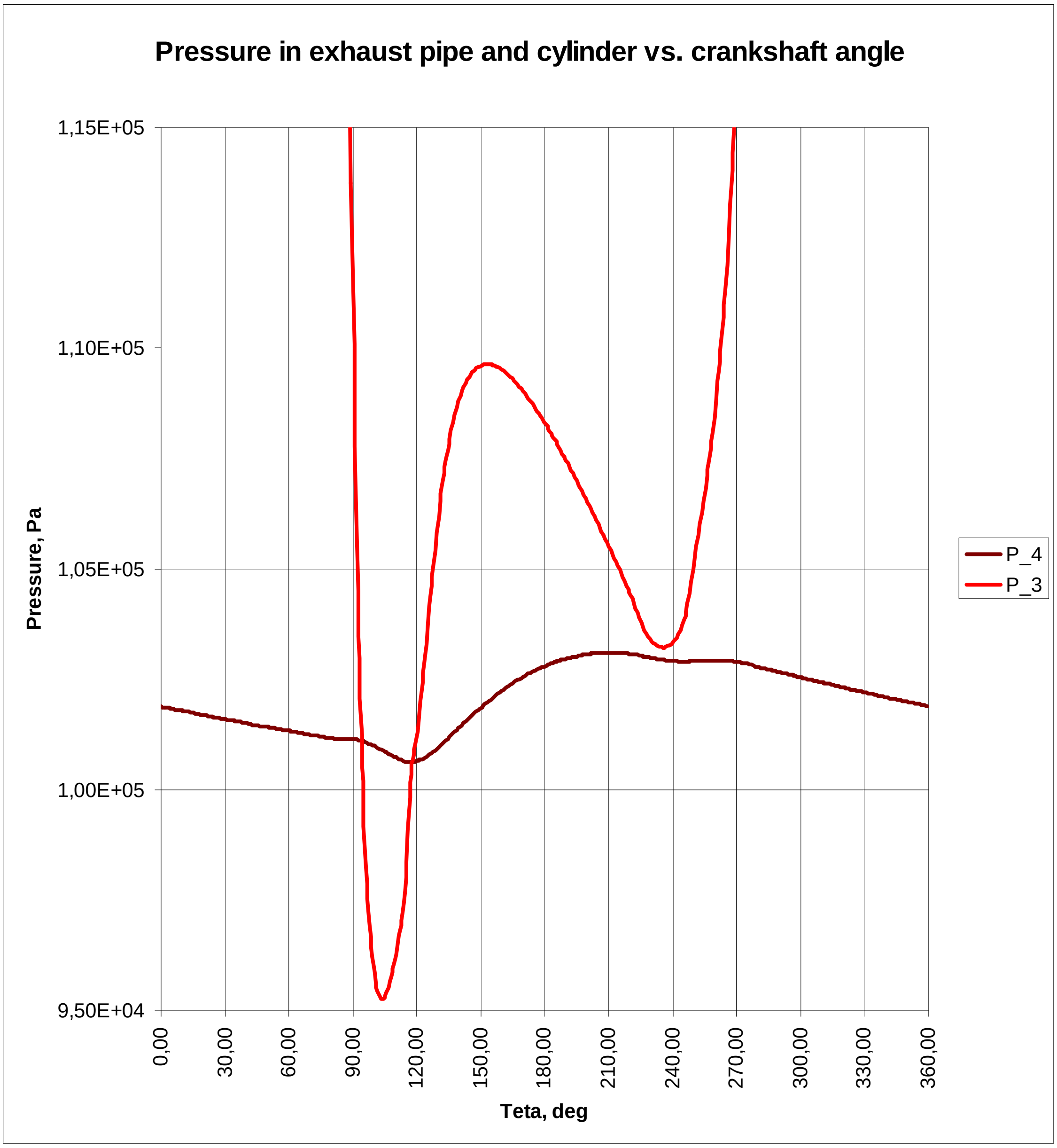

Figure 5.6.2 Pressure in exhaust pipe first part and cylinder at 40000 rpm

5.7 Mass flow into the crankcase

All in all during my calculations I was so interested in pressure results and did not pay any attention to mass flow results. Actually after the long consideration about most informative parameter of the calculations I found that mass flow is the best one. If mass flow is positive it goes according to the direction shown in calculation scheme and vice versa. Moreover inside the program I calculated how much air is pumped inside the engine during one revolution. In my opinion it is critical value according which can be done optimization. In this subchapter there will be discussed mass flow inside the crankcase all optimization procedure will be described in later subchapters.

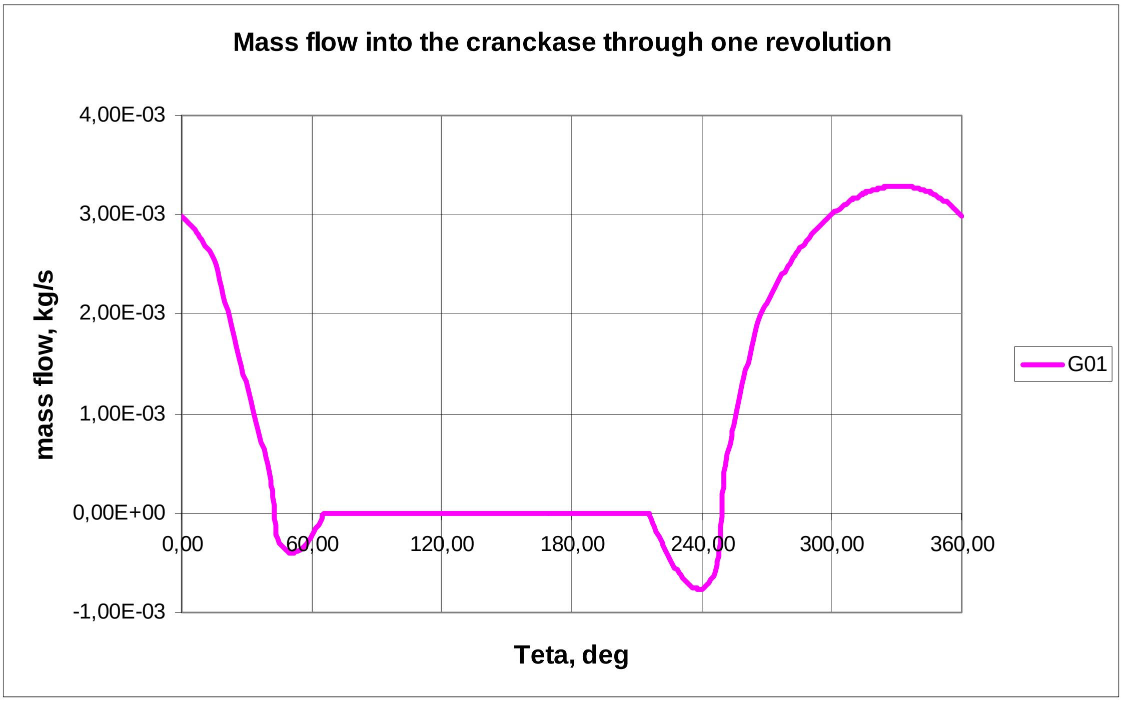

Figure 5.7.1 Mass flow inside the crankcase at 40000 rpm and inlet valve timing 35^0 after BDC and 65^0 after TDC

The figure 5.7.1 shows mass flow rate through one revolution into the crankcase. Mass inflow into the crankcase is $0.14741 \times 10^{-5} kg$ with initial inlet timing. According the figure there is spillage out of the crankcase just before inlet port closes at 45^0 till 65^0 after TDC and when inlet opens at 35^0 till 70^0 after BDC. Using this information we can do adjustments of the inlet timing to close it earlier and open later. This optimization procedure will be represented in later subchapters.

5.8 Mass flow into cylinder

There is represented mass flow out of crankcase to cylinder through transfer ducts in figure 5.8.1. The curve of the pink color represents mass flow of back transfer port, whereas green one of the side transfer port. There are two side ports but in figure there is shown only one, because they are symmetric and overlay each other. From the graph there is clear, that somewhere between 220^0 and 250^0 after TDC there is spillage out of cylinder to the transfer ducts. According the graph information we can adjust transfer timing. As it was noted in previous chapter, the mass flow is the best representative parameter of the engine timing.

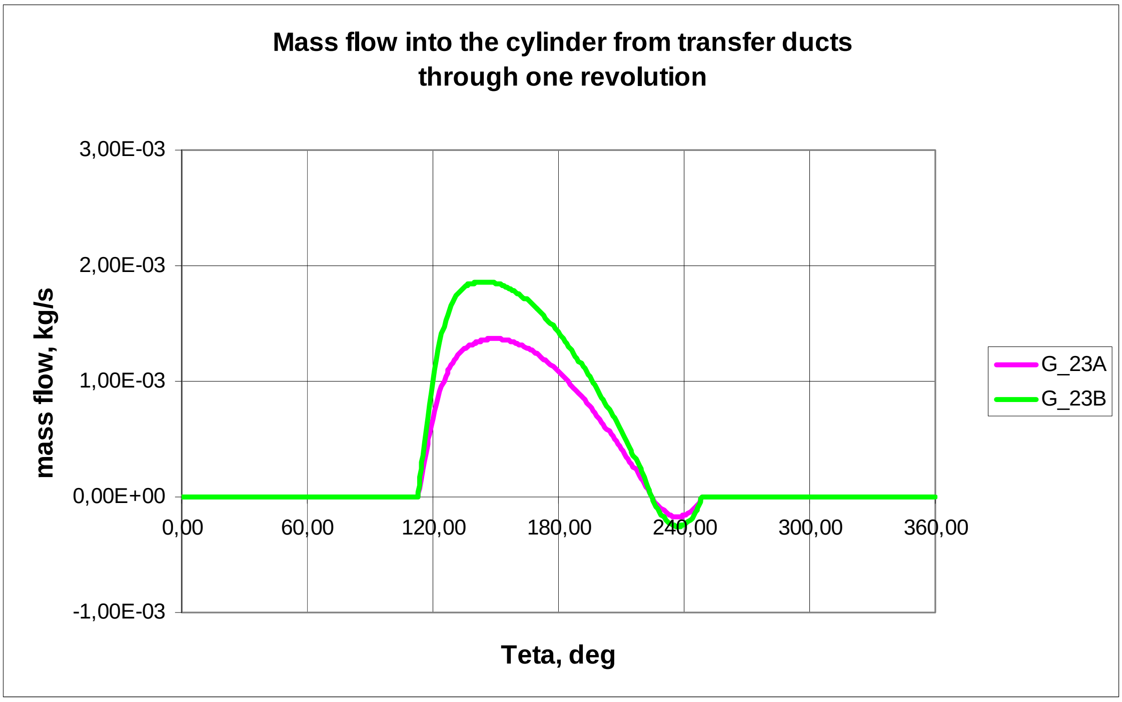

Figure 5.8.1 Mass flow to cylinder from crankcase

5.9 Mass flow out of cylinder

Mass flow out of cylinder is represented in figure 5.9.1. The unpredictable pressure drop in early exhaust phase influences mass flow too, as it was written in previous subchapter 5.5. Without any doubt the mass flow inside cylinder during early exhaust phase (somewhere from 90^0 till 120^0 after TDC) is impossible. Later stages of exhaust shows that outflow from cylinder exist till the port closure. In my opinion it is right result according my calculations. The small hollow between 220^0 and 250^0 after TDC is because of the fact that the pressure in crankcase is smaller than in cylinder and the transfer port is still open and the pressure in exhaust system is higher than in crankcase. The same hollow was in mass flow to cylinder graph (figure 5.8.1). All this information shows that we need some changes in transfer timing.

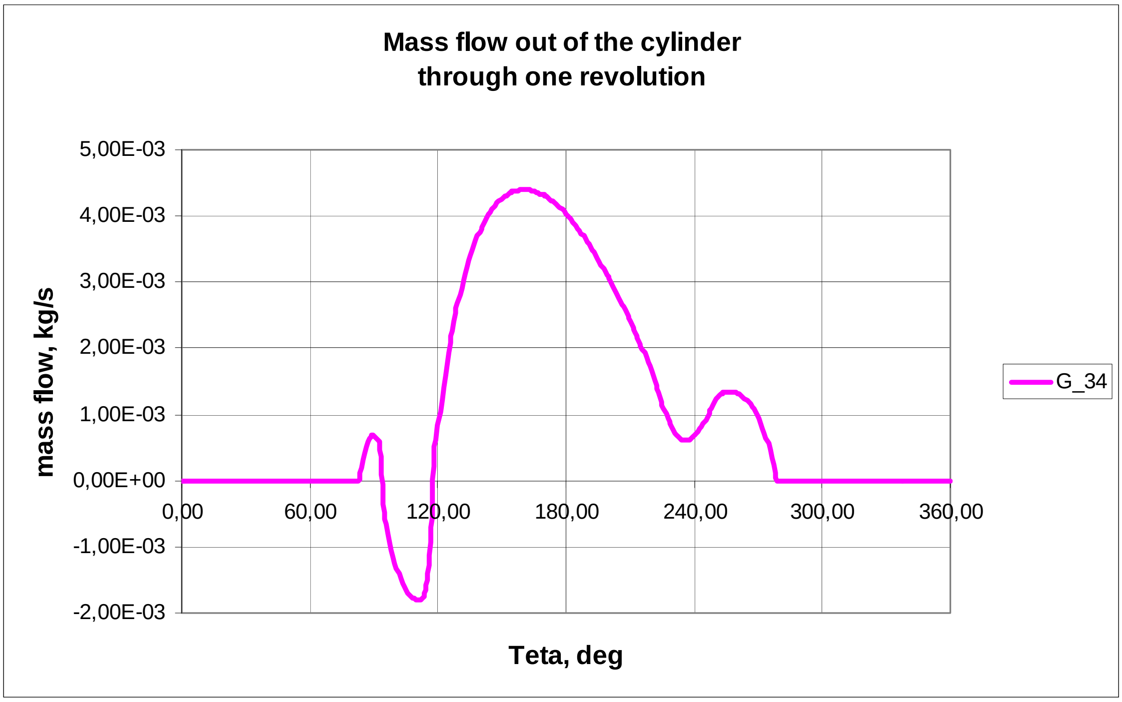

Figure 5.9.1 Mass flow out of the cylinder to the exhaust system

5.10 Mass flow through the exhaust pipe

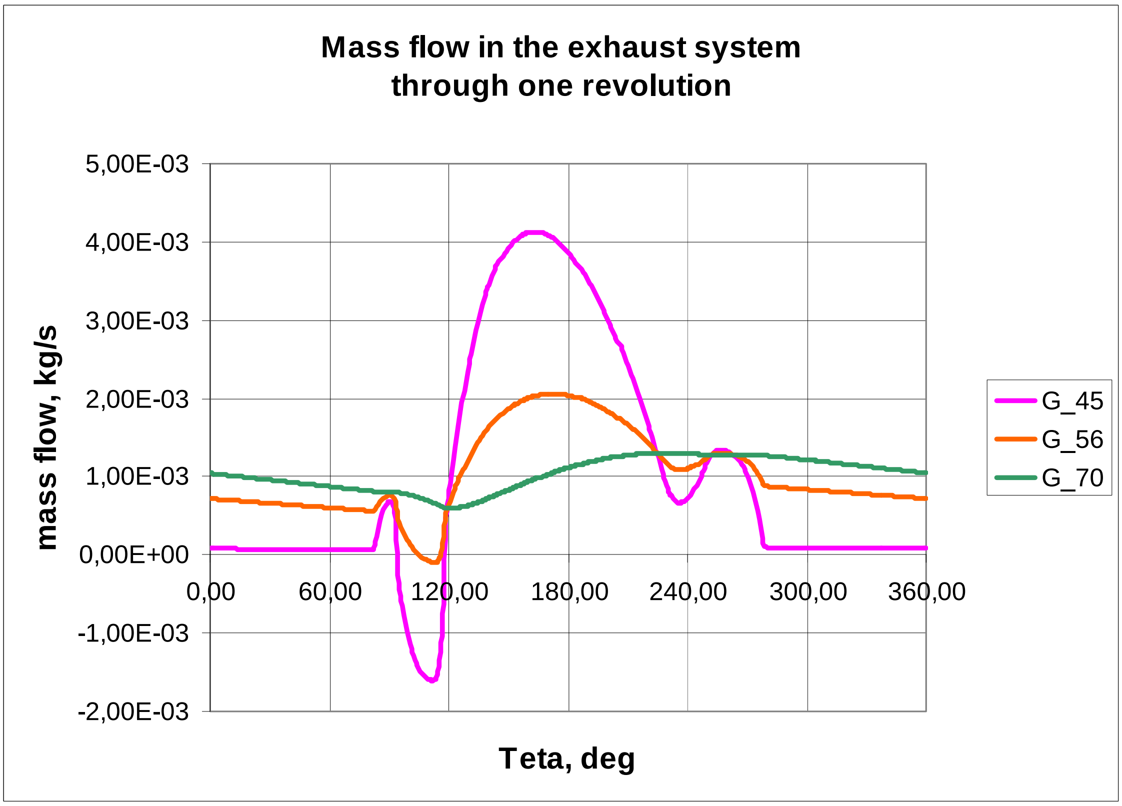

Figure 5.10.1 Mass flow through the exhaust system

In the figure 5.10.1 there is represented mass flow through the exhaust pipe. There is not shown mass flow G_67 because it overlays with G_70. As you can notice from figure 5.9.1 the mass flow out of cylinder is the same as mass flow out of the first exhaust pipe part to the second one. The second one is bigger in volume and it damps a little bit variation of mass flow intensity. The third and fourth part of exhaust pipe damps it much more. According the graph the mass flow in last to parts of exhaust pipe are always of the same direction as it is denoted in calculation scheme.

5.11 Optimization results

In this subchapter there are represented results of mass inflow into cylinder by changing engine parameters.

Table 5.11.1 Optimization table of the inlet and transfer timing

No.	Crankcase volume at BDC, cc	RPM of the engine	Inlet timing, deg		Transfer timing, deg	Exhaust timing, deg	Air mass inflow into cylinder through one revolution, 10^{-5}kg
			Opens after BDC	Closes after TDC			
1	12,50	40000	35	65	136	196	0,1554
2	13,75	40000	35	65	136	196	0,1504
3	12,50	40000	33	65	136	196	0,1552
4	12,50	40000	31	65	136	196	0,1549
5	12,50	40000	37	65	136	196	0,1555
6	12,50	40000	39	65	136	196	0,1556
7	12,50	40000	41	65	136	196	0,1557
8	12,50	40000	43	65	136	196	0,1558
9	12,50	40000	45	65	136	196	0,1558
10	12,50	40000	47	65	136	196	0,1558
11	12,50	40000	49	65	136	196	0,1558
12	12,50	40000	51	65	136	196	0,1557
13	12,50	40000	53	65	136	196	0,1556
14	12,50	40000	45	67	136	196	0,1557
15	12,50	40000	45	69	136	196	0,1556
16	12,50	40000	45	63	136	196	0,1558
17	12,50	40000	45	61	136	196	0,1556
18	12,50	40000	45	59	136	196	0,1554
19	12,50	40000	45	65	134	196	0,1555
20	12,50	40000	45	65	132	196	0,1552
21	12,50	40000	45	65	138	196	0,1560
22	12,50	40000	45	65	140	196	0,1562
23	12,50	40000	45	65	142	196	0,1562
24	12,50	40000	45	65	144	196	0,1562
25	12,50	40000	45	65	146	196	0,1561
26	12,50	40000	45	65	148	196	0,1560
27	12,50	38000	45	65	142	196	0,1614
28	12,50	36000	45	65	142	196	0,1664
29	12,50	34000	45	65	142	196	0,1714

The main goal of optimization procedure is to find the inlet and transfer timing values which gives maximum mass inflow into the cylinder. The results are shown in table 5.11.1. Optimization procedure is done step by step. At first there is changed inlet opening timing, then inlet closing timing. The cell of the optimum timing value is colored by green color. The crankcase volume of 12.5 cc is also colored by green color, because it is real value of crankcase at BDC and it cannot be changed. The column of engine revolutions is colored by green color, when it equal 40000 rpm. From my racing experience I am intended that engine revolutions should be of 40000 rpm. At these revolutions I did my best results and this is the reason why I chose it as optimum. The last three calculations 27-29 (in the table 5.11.1) are just the examples to show how program calculates. At the lower revolutions we have better cylinder filling with air, but these revolutions are too low to achieve good results.

In the first line of the table there are represented initial values of engine parameters. In the second calculation all the engine parameters are the same only increased crankcase volume by 10%. According calculations increment in crankcase volume decrease cylinder filling approximately by 3%. This fact confirms that cylinder filling with air depends on crankcase volume and for the best result the crankcase volume has to be as small as possible. My calculations confirm the same conclusion as represented in publication by Nagao et al. (1959).

By changing step by step inlet opening I found that the highest cylinder filling is when inlet opens at 45^0 after BDC. This value in the table is colored by green color. The optimum inlet closure timing is the same as initial value 65^0 after TDC. By changing inlet timing the mass inflow is improved by $0.1558\times10^5 - 0.1554\times10^{-5} = 0.0004\times10^{-5}$ kg or $\frac{0.0004\times10^{-5}}{0.1554\times10^{-5}}\cdot100\% = 0.26\%$. It is quite small number and it shows that inlet timing does not influence so much cylinder filling.

After finding the optimum inlet timing there were continued calculations to find optimum transfer timing. According calculations the best transfer timing is 142^0. At this timing together with optimum inlet timing inflow is 0.1562×10^{-5} kg. This result is quite impressive because in comparison with initial value it gives 0.5% more inflow into cylinder. The transfer timing in usual race engines do not exceed 138^0. For the next race season I will prepare the engine with transfer timing of142^0 and it will be like experiment according my calculations.

During the optimization procedure there were no optimizations for the exhaust timing. The reason is that exhaust timing is connected with the power pipe and any changes in exhaust timing influences power pipe performance. Because in my calculations power pipe is not simulated very well there are no matter to try to optimization of it.

6. CONCLUSIONS

The intention of my master thesis was to develop computer program which calculates pressure changes and mass flow through the two-stroke engine. This information is useful for the engine tuning. The program is created with software package “Fortran 6.0”. The program simulates small two-stroke engine of the tether race car of 2.5 cc swept volume. Anyway program can be easily modified for the calculations of any type and swept volume engine. With this program can be done engine timing optimization of the inlet and transfer ports.

According the program calculations there is possibility to increase cylinder filling with air by 0.5% by changing inlet and transfer timing (subchapter 5.11). This is quite big improvement, because two-stroke engines are developed at very high level.

According program output data there can be plot graphs of pressure changes and mass flow which helps to analyze engine behavior and to take decisions about its timing.

The program is more oriented for race engines in order to improve power output.

The pressure and mass flow are calculated according concentrated volume method. This method is rather simple, but gives quite reliable results. The calculation results of the program are compared with other researcher’s results by graph comparison. The differential equations of concentrated volume method are solved using Runge-Kutta IV method.

6.1 For further investigation

As final conclusions are very bright the program has some shortages which are the way for further investigation.

The pressure changes in cylinder have some unexpected and unreal behavior (see subchapter 5.5). In my opinion it is because program does not include combustion simulation. The next step for program improvement should be combustion simulation.

Racing engines uses power pipes. Power pipes cannot be simulated correctly by concentrated volume method. For power pipe simulation there should be used characteristic method. This is very advanced method and usually is done by the doctoral students.

Gas-flow inside the ducts includes much more theory than it is represented in my thesis. There is the space for the following improvements which can take into account such processes: heat losses, boundary level behavior, gas properties, temperature of gases, combustion process, air and fuel mixing, gas flow and etc. Gas flow through the engine is very complex task and my thesis is just the first step to solve it.

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INVESTIGATION OF THE GAS FLOW THROUGH THE TWO-STROKE ENGINE

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Abstract. There is represented two-stroke engine simulation program for the calculation of the gas flow through the internal combustion two-stroke engine. The program is designed for engine of 2.5 cc volume. Such engines are used in tether model car competition. Inside program there is simulated disk type inlet valve, transfer ports and exhaust port. This program gives output results of pressure changes and mass flow in crankcase, transfer ports, cylinder and exhaust pipe. The software is dedicated for high performance engine tuning. Moreover program can be easily adapted to any two-stroke engine. The program is created with programming software package "Visual Fortran 6.0".

Keywords: two-stroke engine, simulation, pressure, mass flow, exhausts port, cylinder, crankcase, inlet port, disk valve, transfer port, TDC, BDC.

Introduction

In the tether model car race there is used small two-stroke internal combustion engines. These engines are designed to give the highest power output, because probability to win in tether model car race is directly dependant on the engine power. Usually most of sportsmen tune their engines by experience or trial and error method. Today manufacturing costs of engines are very high, so trial and error method is too expensive for engine power improvements. In order to have engine design data there should be done calculations. For this reason there is created program of gas flow dynamics through two-stroke engine.

Review of similar articles of two-stroke engines

Below there is short review of various articles concerning two-stroke engine.

When a short induction pipe is fixed to the engine, the delivery ratio can be improved by its effect in a wide range of the engine speeds (Nagao *et al.* 1961). Selecting the pipe length so as to coincide three quarters of the period of pressure fluctuation with the period of inlet port opening, the delivery ratio is most remarkably improved by the utilization of pressure fluctuation. Under the condition that the number of pressure fluctuation during the inlet port opening is below 0.5, the delivery ratio decreases through equipment with an induction pipe.

To improve the delivery ratio characteristics, it is not necessary to change the angle area. It is effective to change only the timing of the inlet port by shifting the disc valve with an optimum cut angle around the crankshaft in accordance with the change of engine speed (Komotori *et al.* 1969).

As the reduced angle area of the scavenging and exhaust ports is small, it is useless to change the angle area or timing of the inlet port to increase delivery ratio in the higher engine speed range (Komotori *et al.* 1969).

The drop in the delivery ratio caused by the increase in the crankcase volume is fairly compensated by the properly tuned exhaust and induction pipes. Therefore, it seems that scarcely any advantage is brought by making the crankcase volume excessively small (Nagao et al. 1959).

There is written entire book about two-stroke engine simulation by Blair (1996). In this book there is analysed two-stroke engine step by step and is provided guidelines for simulation. But this book can be used only for very advanced users, because it requires very good knowledge of gas dynamics and higher mathematics.

After review of articles it can be stated that the two-stroke engine performance depends only on gas flow through it and burning process in it, if there is not taken into account its mechanical design.

Calculation scheme and equations

The engine simulation program works according to calculation scheme shown in Fig. 1. Entire engine and exhaust pipe is divided into concentrated volumes. The exhaust pipe is divided into four parts in order to get more precise results. It is so called one dimensional calculation scheme of the engine. All the equations inside the program are with reference to crankshaft angle θ. The calculation starts from top dead centre (TDC). The gas medium is taken to be air. In program there is not simulated combustion process. Because the program is intended for particular engine and particular speed so all engine parameters are taken from real engine (2.5 cc tether model car engine). The angular velocity is taken when engine makes 40000 rpm. Each concentrated volume is defined by differential pressure change equation (Bogdevičius *et al.* 2003):

$$\dot{p}_n(\theta)=\frac{\gamma RT_n}{\omega V_n(\theta)}\left(G_{n-1\_n}(\theta)-G_{n\_n+1}(\theta)\right)-\frac{\gamma p_n(\theta)}{V_n(\theta)}\dot{V}_n(\theta). \quad (1)$$

Where: $\dot{p}_n$ - pressure derivative in concentrated volume; γ - specific heat of air; R - gas constant, $287J/(kg\cdot K)$; T_n - temperature in concentrated volume, K; ω - angular velocity of cranckshaft, 4189 rad/s; V_n - volume of concentrated volume, m^3; G_{n-1_n} - mass flow into the concentrated volume, kg/s; G_{n_n+1} - mass flow out of the concentrated volume, kg/s; $\dot{V}_n$ - volume derivative of concentrated volume.

Mass flow is defined by formula:

$$G_{n\_n+1}(\theta)=\begin{cases}\mu_{n\_n+1}A_{n\_n+1}(\theta)p_n\sqrt{\frac{2\gamma}{(\gamma-1)RT_n}\cdot\left(\left(\frac{p_{n+1}(\theta)}{p_n(\theta)}\right)^{\frac{2}{\gamma}}-\left(\frac{p_{n+1}(\theta)}{p_n(\theta)}\right)^{\frac{\gamma+1}{\gamma}}\right)}, \text{if } \mathrm{p_n}(\theta)>\mathrm{p_{n+1}}(\theta)\\ \mu_{n\_n+1}A_{n\_n+1}(\theta)p_{n+1}(\theta)\sqrt{\frac{2\gamma}{(\gamma-1)RT_{n+1}}\cdot\left(\left(\frac{p_n(\theta)}{p_{n+1}(\theta)}\right)^{\frac{2}{\gamma}}-\left(\frac{p_n(\theta)}{p_{n+1}(\theta)}\right)^{\frac{\gamma+1}{\gamma}}\right)}, \text{if } \mathrm{p_{n+1}}(\theta)>\mathrm{p_n}(\theta)\end{cases}, \quad (2)$$

Where: μ_{n_n+1} - coefficient evaluating surface roughness of the walls, 0.8; A_{n_n+1} - cross section area between two concentrated volumes, m^2.

Cross section area A_{0_1} of the inlet valve is described by formula:

$$A_{0\_1}(\theta)=\begin{cases}\pi\frac{\phi}{2\pi}\left(r_{\max}^2-r_{\min}^2\right), \text{if } 0<\theta\le\theta_{in1}=\alpha_{in}-\phi\\ \pi\frac{\phi-\theta}{2\pi}\left(r_{\max}^2-r_{\min}^2\right), \text{if } \theta_{in1}<\theta\le\theta_{in2}=\alpha_{in}\\ 0, \text{if } \theta_{in2}<\theta\le\theta_{in3}=\pi+\beta_{in}\\ \pi\frac{\theta}{2\pi}\left(r_{\max}^2-r_{\min}^2\right), \text{if } \theta_{in3}<\theta\le\theta_{in4}=\theta_{in3}+\phi\\ \pi\frac{\phi}{2\pi}\left(r_{\max}^2-r_{\min}^2\right), \text{if } \theta_{in4}<\theta\le 2\pi\end{cases} \quad (3)$$

Where: $\phi=50^0=0.8727rad$ - Inlet duct angle in cross-section area, engine parameter; $\alpha_{in}=65^0=1.1345rad$ - Inlet port closes after TDC, engine parameter; $\beta_{in}=35^0=0.6109rad$ - Inlet port opens after BDC, engine parameter; $r_{\max}=10.55\times10^{-3}m$ - Inlet port outer radius, engine parameter; $r_{\min}=6\times10^{-3}m$ - Inlet port inner radius, engine parameter.

Cross section area A_{2A_3} between transfer port and cylinder is described by formula:

$$A_{2A\_3}(\theta)=\begin{cases}0, \text{if } 0<\theta\le\pi-\frac{\alpha_{tr}}{2}\\ W_{31\_4}\left(-L_{ct}\cos(\theta)-\sqrt{L_{cr}^2-L_{ct}^2\sin(\theta)^2}+L_{ct}\cos(\theta)+\sqrt{L_{cr}^2-L_{ct}^2\sin(\theta)^2}\right), \text{if } \theta_{tr1}<\theta\le\theta_{tr2}=\pi+\frac{\alpha_{tr}}{2}\\ 0, \text{if } \theta_{tr2}<\theta<2\pi\end{cases} \quad (4)$$

A_{2A_3}, m^2 - First transfer port area; $W_{2A\_3}=7\times10^{-3}m$ - Width of the first transfer port, engine parameter; $\alpha_{tr}=136^0=2.3736rad$ - Transfer timing, engine parameter; $L_{cr}=26.5\times10^{-3}m$ - Connecting-rod length, engine parameter; $L_{ct}=7.125\times10^{-3}m$ - Half-stroke length, engine parameter.

Simlar as cross section area A_{2A_3} there is described cross sectional areas of A_{2B_3} between second transfer port and cylinder, A_{2C_3} between third transfer port and cylinder and A_{3_4} between cylinder and exhaust pipe. All these cross section areas are described by the same formula only there is difference of engine parameters for them.

According to calculation scheme there was created equation system of nine differential equations of pressure changes with nine unknown pressures. This differential equations system was solved using numerical Runge-Kutta IV method (Van Iwaarden 1985). In order to get stable solution there was chosen step of $\Delta\theta=1.745\times10^{-5}$ rad. There was simulated seven revolutions of engine and total number of steps is 2520000!!! For each step there is solved more than 20 equations.

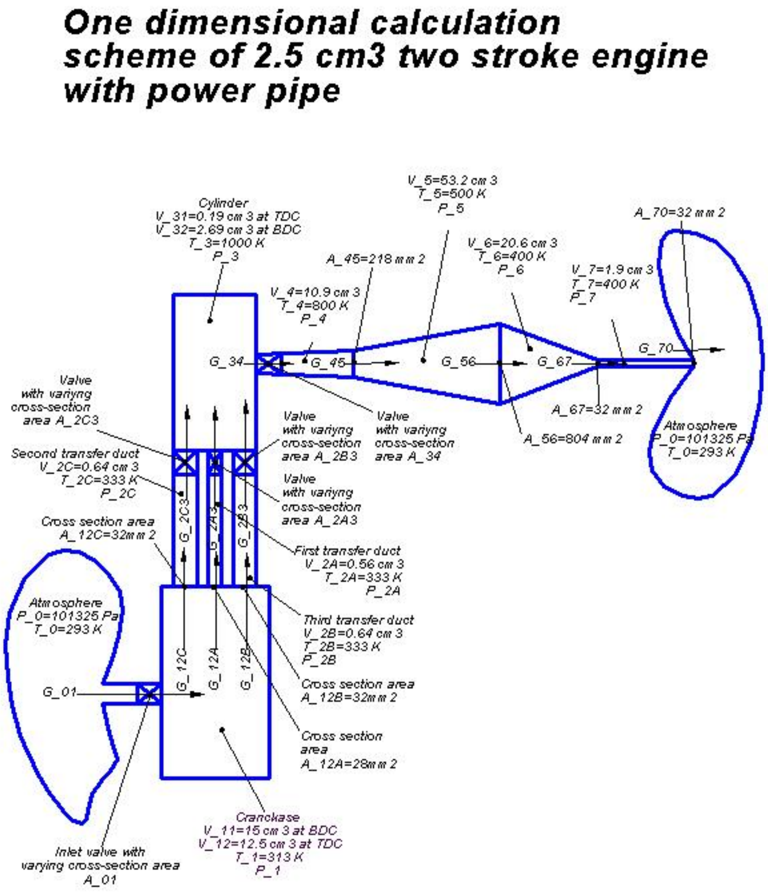

Fig. 1 Calculation scheme of two-stroke engine with exhaust pipe.

Calculation results

The results are represented by graphs. The program provides stable result only after three revolutions (Fig. 2).

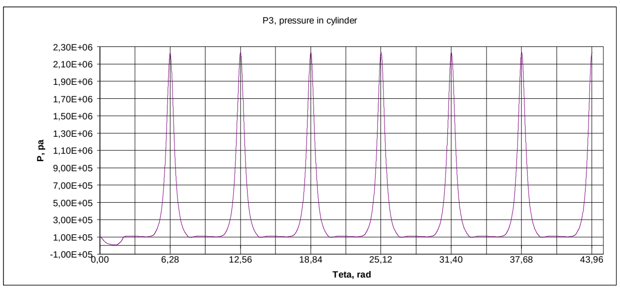

Fig. 2 Pressure changes in cylinder

On the X-axis in Fig. 2 is θ angle of the crankshaft. The graph starts from TDC the same as program. One revolution of engine occurs when θ equals multiple of 6.28 rad. There is possible to notice, that in first revolution there is strange shape of pressure line. Each minor gridline of X-axis shows bottom dead center (BDC). The maximum pressure in cylinder is 220000 Pa. It is quiet real result, because the engine compression ratio is about 14.15:1. At the BDC the pressure in cylinder equals approximately to atmospheric pressure (100000 Pa). In order to get better view of pressure in cylinder there is done another graph of the sixth revolution where θ starts from 31.42 till 37.70, Fig. 3. In this figure there is included all calculated pressures in two-stroke engine. Pressures identification corresponds with calculation scheme which is shown in Fig. 1. The minor X-axis gridline corresponds to BDC. I have only one remark by myself concerning pressure diagram. It is that pressure in cylinder goes down atmospheric pressure before BDC for a short period of time. In my opinion and what I found in other books it is impossible. This error can be because of very fast process which is happening in engine and another reason is that there is not simulated combustion process.

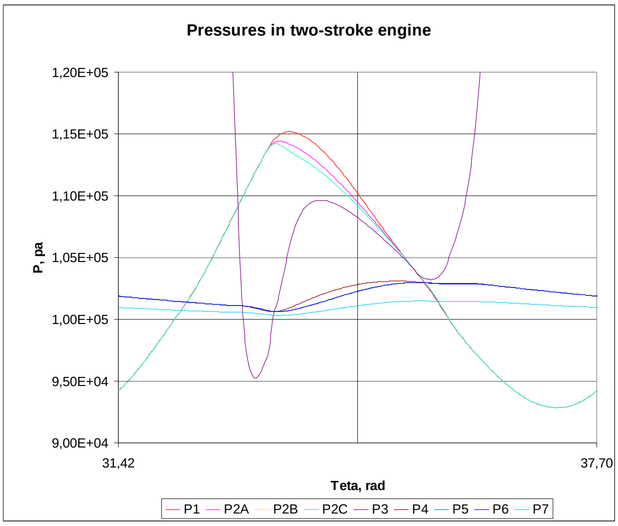

Fig. 3 Pressures in two-stroke engine.

According to the graphs the pressure in crankcase reaches maximum value of 115500 Pa. The pressures in transfer ports are more or less the same as in crankcase. The pressures in exhaust pipe are always higher than ambient pressure. It means that for burned gas to go out of cylinder there is need to overtake pressure resistance of exhaust pipe.

In the graph below Fig. 4 there is shown port areas versus crankshaft angle θ.

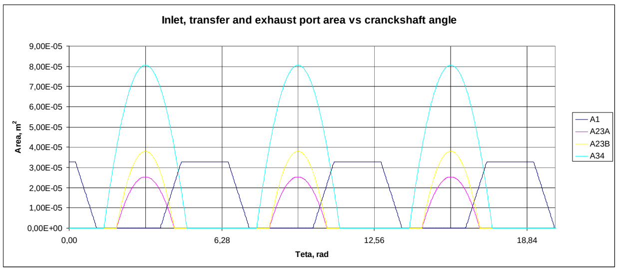

Fig. 4 Inlet, transfer and exhaust areas versus crankshaft angle.

The program also gives output of mass flow through the entire engine. This is very informative data Fig. 5. During the analysis of mass flow we can easily see if the timing of engine is too short or too long. Our interest is to achieve, that there were no any negative mass flow, which means opposite direction which is shown in calculation sheme Fig. 1.

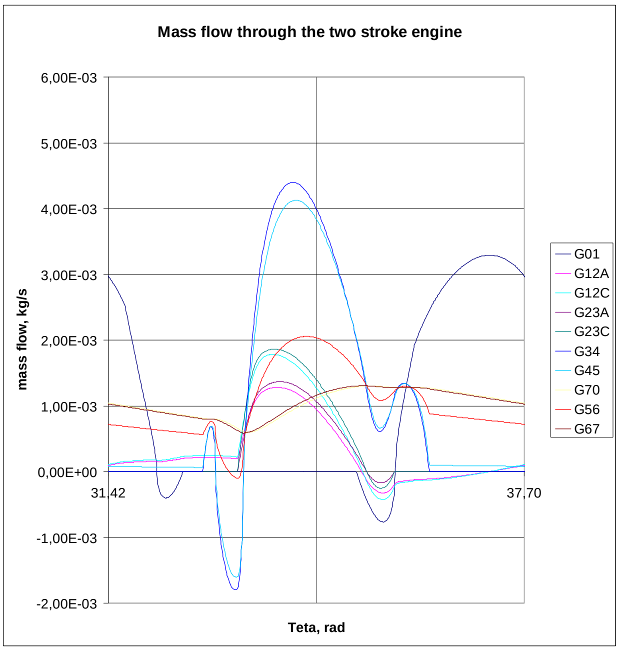

Fig. 5 Mass flow in two-stroke engine.

Conclusions

The two-stroke engine simulation program gives quite reliable results especially of the mass flow and pressures in crankcase and transfer ports.

There are very well simulated inlet, transfer and exhaust ports.

The pressure and mass flow in cylinder is not reliable result for the short period when cylinder pressure is less than ambient pressure. This is because the program does not simulate combustion process. Simulation of combustion process can be continuation work on the basis already done.

The exhaust pipe is simulated as four concentrated volumes. This can be improved by simulating exhaust pipe with characteristic method. Because inside exhaust pipe goes pressure waves and they have great importance for the engine performance. The simulation of concentrated volumes does not take into account behavior of pressure waves.

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DUJŲ TEKĖJIMO DVITAKČIAME VARIKLYJE TYRIMAS

L. Adomavičius

Santrauka

Pristatoma modeliavimo programa skirta skaičiuoti dujų tekėjimo parametrus dvitakčiame vidaus degimo variklyje. Progrma sukurta 2,5 cm^3 darbinio tūrio varikliui. Tokio tipo variliai naudojami greituminių automodelių varžybose. Programoje yra sumodeliuotas diskinio tipo įsiurbimo vožtuvas, prapūtimo ir išmetimo kanalai. Programa apskaičiuoja ir pateikia slėgio bei masės debito rezultatus karteryje, prapūtimo kanaluose, cilindre ir išmetimo vamzdyje. Programa skirta sportinių variklių galios didinimui. Programa taip pat gali būti pritaikyta bet kokiam dvitakčio variklio modeliavimui. Programa yra sukurta naudojant programavimo paketą „Visual Fortran 6.0“.

Reikšminiai žodžiai: dvitaktis variklis, modeliavimas, slėgis, masės debitas, išmetimo kanalas, cilindras, karteris, įsiurbimo vožtuvas, diskinis vožtuvas, prapūtimo kanalas, viršutinis mirties taškas (VMT), apatinis mirties taškas (AMT).