INTERNAL COMBUSTION ENGINE

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Abstract**—** This research paper strategy was previously applied to simplified geometries as well as real engine geometries. The purpose of this work is to show its validity for real and complex engine geometry, covering a longer simulation interval then previously. After an initial description of the mesh creation process, studies of the in-cylinder charge motion, created during the intake stroke, were made to investigate the presence of a tumble motion. The results how strong tumble motion is not generated. This could also be because of numerical diffusion caused by the mesh structure. Furthermore, investigations were made to see why the present engine has a tendency to knock, possible increasing the knock risk close to the exhaust valves. Problem calculated turbulence intensity that was too low for proper flame propagation were apparent, leading to inconsistency between the calculated cylinder pressure of the CFD calculation and the calibrated1D calculation.

Keywords**—** simulation interval, in-cylinder charge motion, CFD calculation, numerical diffusion, engine geometry.

1. INTRODUCTION

The Internal Combustion Engine Group collaborates with many industrial partners and the work of this thesis was carried out in collaboration with MV Agusta S.P.A which is a company based in Varese outside Milanthatmakes exclusive high performance motor cycles. ThethesisconcernstheCFDmodelingofafourstroke, fourcylinder, spark-ignited engine which is mounted on the motorcycle models based on the F4 series made by MV Agusta. The CFD simulations were done on one of the cylinders during the intake, compression and expansion phases ,in cluding combustion simulation, taking in to account the valveandpiston motion.

CFD simulations can be a powerful engineering tool that can help to understand the thermo-physical and chemical processes that takes place in an IC engine without being forced to use possibly complicated and expensive measurement techniques. Some things can be difficult measure in at here dimensional space an done dimensional models does not account for all geometric topologies, making CFD simulation san important compliment. It can also be a valuable tool to run numerous tests and in vestigation so fan engine without having to perform.

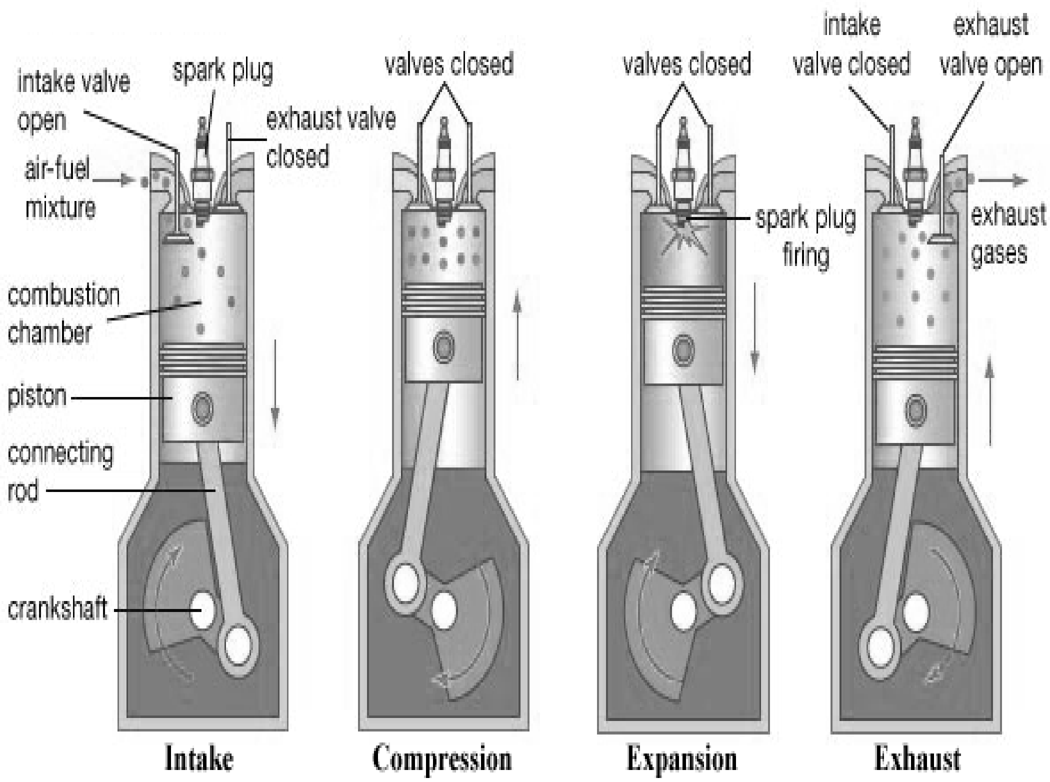
CFD research has unfortunately not yet reached a state where it completely describes all the processes that takes place in an IC engine. This is mainly due to the fact that it is a highly complex mechanical device which incorporates many, simultaneously interacting, thermo fluid and chemical processes. Because of the IC engines reciprocating nature it features extremed deformation of the solution domain due to the moving piston and valves, and the shape of the piston and cylinder head, which are key features for the engine design, are usually very complex. Aside from the geometric complexity there are also the simultaneously interacting thermo fluid processes such as; non-stationary turbulent flow, heat and mass transfer, injection, atomization ,dispersion and vaporization of the liquid fuel, ignition, combustion and the consequent formation of harmful pollutants, to mention a few. Adding all this together makes a complete modeling of the IC engine one of the most challenging tasks in the are of CFD research.

In this section a brief introduction to the Internal Combustion Engine (ICE) will be presented as well as the purpose of this thesis. Lastly a brief summary of all the chapters of the thesis is presented.

2. *The Internal Combustion Engine*

Theinternalcombustion(IC)engineistodaythemostwidelyusedenergysourceintheautomotive and naval industry. It has therefore become increasingly important to improve theefficiency of the engines to reduce fuel consumption, emission levels and noise pollution todecrease the negative effect it has on the health of the human being as well as its environment.Since the introduction of the IC engine in the late 19th century its development has resulted in aconstant reduction of fuel consumption and emission levels, while the power output per cylindervolume has continued to increase. The IC engine comes in many different types and sizes but canbe divided in the category of the two stroke cycle or the four stroke cycle which can either be runbytheOttoortheDieselprinciple.ForOttoenginesthepre-mixedair-fuelmixtureisignitedbya spark from a sparkplug while for the Diesel principle the air is compressed beforehand in thecylinder and the incoming fuel spray is ignited by the high pressure and temperature. In thischapter a brief explanation of the four stroke, spark-ignited (SI/Otto) engine will be presented,sincethisisthetypeofenginethat the workofthe thesis is performed on.

The four stroke cycle starts with the piston positioned in TDC (Top Dead Center) and as thepiston travels down towards BDC (Bottom Dead Center) the intake valve opens, letting the freshcharge of air-fuel mixture to enter the cylinder during the *intake stroke*, see Figure 1. Withthe piston at BDC the intake valve closes and as the piston travels towards TDC again the freshcharge is compressed during the *compression stroke*. As the piston reachesTDC the compressed air-fuel mixture is ignited by the spark plug and the chemical energy of thefuel is converted to heat during combustion. This increases the cylinder gas temperature and pressure there for adding work to the crank shaft during the power strok eor*expansion stroke.* When the piston reaches BDC again the exhaust valve opens and as the pistontravels towards TDC the exhaust gas is pushed out from the cylinder during the *exhaust stroke*,see). When the piston has reached TDC the cycle is restarted again. In one powercycle the crankshaft has done two full revolutions (720 crank angle degrees or CAD) and thepistonhastraveledupanddownthe cylinder fourtimes, thereforethe name fourstroke cycle.



i. b) c) d)

Figure 1. *The four phases of the four stroke IC engine: intake stroke (a), compression stroke (b), expansion orpowerstroke(c)and exhauststroke (d).*

The combustion process in the SI engine can be divided into four stages:

1. **Spark ignition.** The air-fuel mixture is ignited by the spark discharge of the sparkplug.The sparkplug releases a high energy plasma with a temperature around 60 000 K for avery short period (typically in the order of 1 ns). The spark ignition usually occurs around20-40crankangledegreesbeforeTDC.
2. **Earlyflamedevelopment.**The first phase of the combustion process is the laminarflame propagation. The flame propagates in a sphere around the sparkplug and the flamefrontisconsideredsmoothwithatypicalthicknessof0.1mmandavelocityofaround 0.5 m/s. The flame speed depends on the pressure, temperature, the air-fuel ratio and theamount of residual gases from the previous cycle. High temperature and a low air-fuelratio results in a higher flame velocity while high pressure and residual gases results in alowervelocity.
3. **Turbulent flame propagation.** After a few crank angle degrees the laminar phase istransformedintoaturbulentphase,causedbytheturbulencecreatedduringtheintakeand compression stroke. The smooth laminar flame front is broken down by vortexes intoa turbulent flame front with a very irregular and wrinkled shape. The flame speed nowreaches velocities of 10-50 m/s. The velocity still depends on pressure, temperature, theair-fuelratioandamountofresidualgases, buttheturbulenceintensityisthemostimportantfactorinthisphase.Increasing theenginespeedleadstomoregeneratedturbulence, this in turn results in higher flame velocity. This leads to a combustion timethat is unchanged (expressed in crank angle degrees) allowing for high engine speeds fortheSI engine.
4. **Flame extinction.** Lastly the flame reaches the cylinder liner causing flame extinctionbecause of the liner walls relatively low temperature. The combustion phase is usuallycompletedaround 30 crank angle degrees after TDC.

This is a simplified explanation of the typical combustion process where the spark ignited flamemoves steadily across the combustion chamber until the charge is consumed. However two typesof abnormal combustion can be identified; *knock* and *surface ignition* which will be explained in the next section.

3. Abnormal Combustion Phenomena

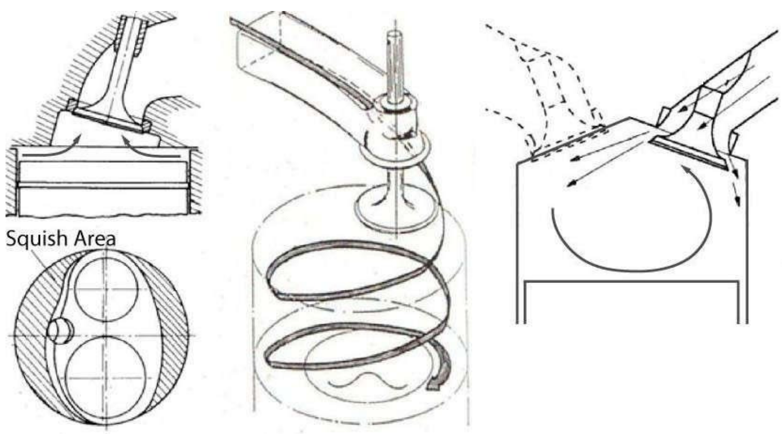
As the flame propagates across the combustion chamber the unburned mixture ahead of the flame(called the *end-gas*)is compressed, causing its pressure, temperature and density to increase. Some of the end-gas air-fuel mixture may then undergo chemical reactions prior to the normal combustion when the flame front reaches the end-gas. These chemical reactions may cause the end-gas to auto ignite, releasing a large part or all of their chemical energy. This causes the end-gas to burn very rapidly with flame velocities of up to 1000 m/s and it creates high-frequency pressure oscillations inside the cylinder that produce the sharp metallic noise called*knock* [1]. Knock is mainly caused by high pressure and temperature in the end-gas but depend salsoon the octane number of the fuel.Knockiscommoninengineswithahighcompressionratioε,whichleadstoahighcylinderpressureandanincreasedriskof knockwhentheend-gasis compressed further by the propagating flame. The pressure waves that knock creates, lead to high mechanical load on the engine, causing damage to the material. Knock can be detected byknocksensors,forexamplewithanaccelerometerinthecylinderwallsthatdetectshigh-frequency pressure oscillations. Knock can the n be n avoided by adjusting the spark timing automatically when knock is detected in the engine. By retarding the time of spark discharge themaximumcylinderpressureisloweredwithadecreasedriskofknockasresult.

The other abnormal combustion phenomenon is *surface ignition*. This phenomenon is causedwhen the air-fuel mixture is ignited by an overheated surface in the engine, like the valves or thesparkplug. It is defined as ignition by any other source than the spark of the sparkplug. It canoccur before the spark ignition (preignition) or after (postignition). Either way it means that thecombustion process is no longer controlled by the ignition of the spark. When surface ignitionresultsinknockintheend-gasitiscalled*knockingsurfaceignition*[1].

Knock can be considered a bigger problem than surface ignition, since surface ignition can beavoided with proper engine design. Knock on the other hand is an inherent constraint on engineperformance and efficiency since it limits the maximum compression ratio that can be used withany given fuel [1].

4. Charge Motion Classifications

As explained above, the turbulence level in the cylinder is very important for the turbulent flamepropagation and the efficiency of the combustion. There are many ways in which the turbulencelevel in the cylinder can be increased by different constructions of the intake ducts. The flowmotion of the incoming fresh charge can vary dramatically depending on the design of the intakeducts, valves and combustion chamber. The structure of the flow can be divided in three mainmotion categories; *squish*, *swirl* and *tumble*, see Figure 2. Squish is defined as the flow createdwhenthepistonreachesTDCandtheouterpartofthepistoncrownapproachesthecylinderhead closely, causing a flow motion directed from the cylinder liner towards the center of thecombustion chamber, see Figure 2(a). Swirl is the rotational flow motion around the cylinderaxis and is created by the shape of the intake duct, see Figure 2(b). By bringing the intake flowinto the cylinder with an initial momentum a swirl motion is created, usually persisting throughthe whole intake, compression and expansion stroke. Tumble is the rotational flow around theaxis perpendicular to the cylinder axis, see Figure 2(c). The tumble motion is also created bytheshapeof the intakeduct and the angle ofthe incoming jets offresh charge.



• b) c)

Figure 2.*Chargemotionclassificationforthreetypicalflowstructures;squish(a),swirl(b)andtumble(c).*

While all three of these charge motions can help to improve the turbulence level in the SI engine, the tumble motion is instead the most important. Tumble is generated during the intake stroke and intensified during the compression stroke where the large scale tumble motion keeps it sangular momentum but increases its intensity because of the shrinking cylinder volume. As the piston reaches TDC and the combustion chamber becomes "flat", the large scale tumble motion breaks up in small scale turbulence, creating a high level of turbulence intensity in the cylinder. To create astrong tumble motionit is notjust importantto accelerate theincoming flow, but also to direct it mainly towards the area of the exhaust valve. In a traditional intake duct the incomingcharge is distributed quite uniformly around the valve causing a flow that counteract itself andhinder the creation of a main tumble motion. By directing a bigger part of the flow towards oneside of the valve, preferably towards the exhaust valve, a strong tumble motion can be created bythis main flow jet that is not counteracted by the jets created on other sides of the valve [2]. Astrong tumble motion creates high turbulence which in turn leads to a high flame velocity. Thisincreases the efficiency of the combustion process, but the fast combustion also leads to hardengine noise. A strong tumble motion is positive for high performance and racing engines but fornormalenginesa compromisebetween enginenoiselevelsandperformancehasto bemade.

5. *Purpose of the research paper*

The purpose of this research paper can be divided into two parts. The first part is related to the strategiesof mesh management and CFD simulation, and the second part is related to the analysis of theresultsproducedforthe specificengine used for thesimulations.

Since the moving piston and valves deforms the solution domain extremely during a wholeenginecycle,thecomputationalgridwillatsomepointbecometoodistorted,leadingtonumericalerrorsandabortionof thecalculations.Astrategytosolvethiswaspreviouslydeveloped by the Internal Combustion Engine Group at Politecnico di Milano which uses a number of computational grid sorme shestocover the whole engine cycle. Eachmeshisdeformed for a certain interval and when the quality of the mesh becomes too low, the solution isinterpolated onto a new mesh. This strategy is called the MUMMI approach (Multiple MeshMotion and Mesh to Mesh Interpolation) and it had previously been employed on a simplifiedengine geometry [10] and later on a real engine geometry [12]. In the last case the calculations ofthe whole engine cycle intended to be simulated was not actually performed due to the fact thatno real boundary conditions were used. Thus, the purpose of this thesis was to use the samestrategy of how each mesh is constructed and generated as in the previous cases, but this timeemployed on the real and highly complex engine geometry of the F4 engine, including the use ofrealistic boundary and initial conditions for temperature, pressure and mass flow. The simulationwas intended to cover the intake, compression and expansion stroke starting from the opening oftheintakevalveandendingatthecompletionofthecombustionprocess.Thepurposewasalsotostudyacombustionmodeld evelopedbyWeller[3],whichispre-implementedintheOpenFOAM code, to investigate if a realistic combustion process could be simulated for the F4engine.

The second part of the purpose of this thesis is strictly engine related. First it was of interest tomakeaninvestigationofthestructureofthechargemotioncreatedduringtheintakeandcompression stroke to see if a strong tumble motion is created. As mention in Chapter 1.1.2tumble is important for the performance of the SI engine and is mostly created by the design ofthe intake port and the intake valves. Furthermore, the F4 engine has a tendency to knock at fullload for the engine speed that produces the maximum amount of torque, and it was of importancetotrytomakeaninvestigationtowhyandwheretheknockoccurs.Asmentionedin section 2

the knock risk depends on the temperature and pressure of the end gasanda study had to be made to see if, for example, the charge motion could create a temperature distribution in thecylinderthat is not uniform,causing a higher knock risk in some parts of the cylinder.

As a sub purpose it was also of interest to see if the whole process of mesh generation andsimulation could be done within a reasonable amount of time to make it valid as an engineering tool.

6. RESULT

6.1 SimulationfromEVC

1. SimulationstartsatEVC(388CAD)insteadofIVO(333CAD)

2. Changeof Air-Fuel (A/F)ratio to 12.4

3. CorrectionofcompressionratiofromIVC(600CAD)

The overestimation of volumetric efficiency is most likely because of the fact that the exhaustvalvewaskeptclosedduringthevalveoverlapperiod,andthereforeanamountofextramasswas trapped in the cylinder. In reality there will be a certain mass that exits the cylinder throughthe exhaust valve during valve overlap period, and this was not taken into account for. To solvethis problem the simulation was set to start at exhaust valve closure times instead. New initialconditions at EVC (388 CAD) for temperature and pressure was assigned from the Gasdyncalculations. The original boundary condition for mass flow at the intake duct was also used, seeFigure3.14.

The initial case had an Air-Fuel ratio corresponding to a stoichiometric mixture (14.7) which isnot the case of the F4 engine at full load and 8000 rpm. Instead the real A/F ratio should be 12.4which corresponds to a λ value of 0.84. A lower A/F mixture will lead to less air in the cylinderand therefore lower cylinder pressure because of a smaller volume of the charge (gasoline hashigher density then air). To implement the correct A/F ratio is obviously important also for thefuturecombustion simulation.

The intake stroke was then re-calculated with the first two changes (point 1 and 2). First a casewas calculated only with the new starting time at EVC. Then another case was simulated withboth the new starting time at EVC and the new A/F ratio to compare them both against the initialcase described in Chapter 4.1.1. Note that the compression ratio correction was not made to thegeometry until intake valve closure time (600 CAD) since the difference in cylinder pressure willbe negligible before the intake valve closes and the compression stroke starts. The result of the calculations for the first two changes can be seeming Figure3.

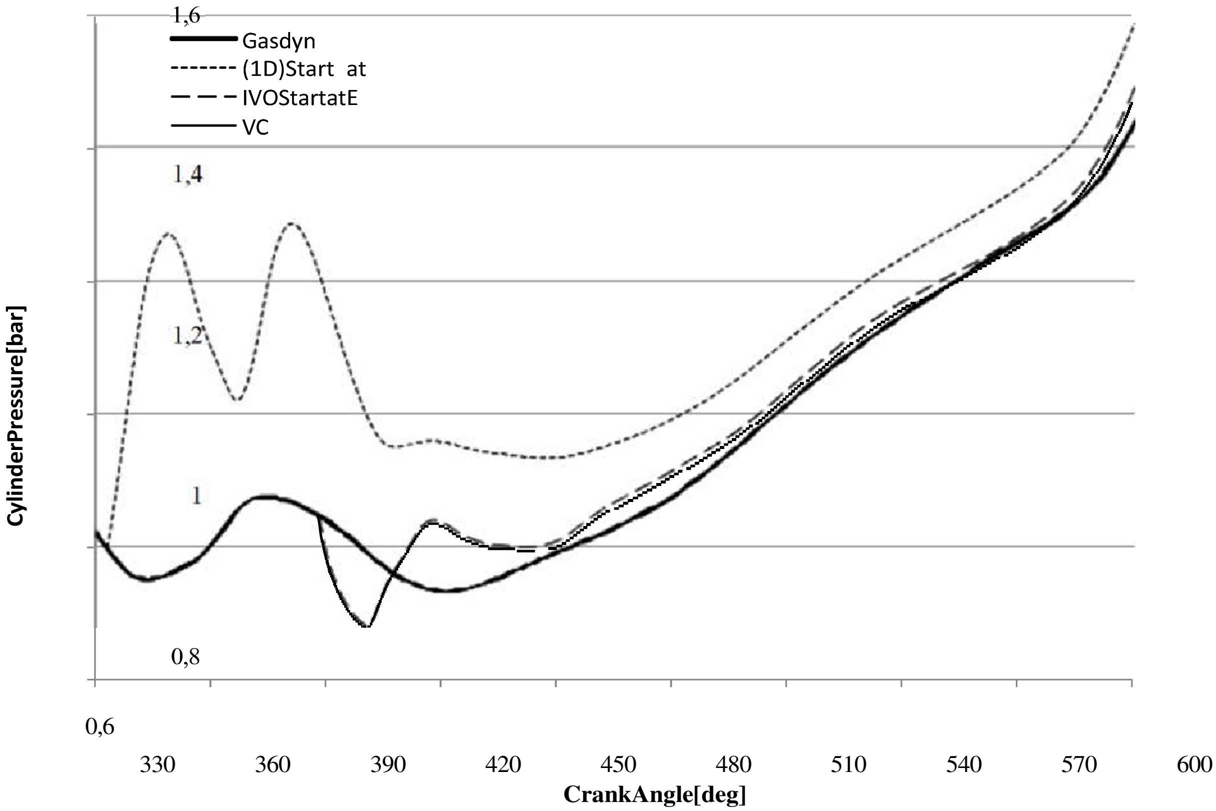


Figure3.*CylinderpressureduringintakestrokeforGasdyn(1D)andOpenFOAM(CFD)calculations,comparingthedifferenceofstartingthesimulationfromIVOandEVC,andchangingλfrom1tothecorrectvalueof0.84.*

Figure 3 shows that the biggest improvement for the consistency between the 1D and CFDcalculations was from the change of simulation starting time. As suspected the overestimation ofvolumetricefficiencywasmainlybecauseoftheneglectedvalveoverlapperiod.Asmallimprovement was also achieved by decreasing the air-fuel ratio, and therefore a smaller gasvolumeofthetrappedchargeinthecylinder,resultinginalowercylinderpressure.

It was discovered that the engine model had a compression ratio ε that was wrong compared tothe real engine. The correct compression ratio is 13 while the engine model had a ratio of 14.1.The reasons for this is probably because the engine model is slightly simplified, for example arethe crevices above the first piston ring not present. There are most likely also some additionalcrevices in the cylinder head, close to the valves and spark plug that are not present either in theengine model. Since the error in compression ratio will have a negligible effect before the valveclosesandcompressionstrokestartsat600CADitwasdecidedtoonlyre-makethemeshesfrom 600 – 780 CAD. An investigation was made to confirm this by also re-making the meshesforthetwointervals before600CAD whichdidnot showanydifferencein cylinderpressure.

6.2 Results from Final Intake and Compression Stroke Simulation

The results of the calculations can be post-processed with a software called ParaView. Thissoftware offers a graphical user interface and many options to display the calculated field valuesin a number of different ways. ParaView is started by importing an OpenFOAM case to it withthe utility paraFoam. This chapter will deal with some results of the field variables calculatedfrom the final simulation of the intake and compression stroke, which was started at EVC, has acorrectedcompression ratio andthecorrectair-fuelmixture.

6.3 FlowField

Starting from a flow field that is zero over the whole domain, an initial flow field is establishedbecause of the pressure difference between the intake duct and cylinder as well as the imposedboundaryconditionforthemassflowattheintakeduct.Afterthefirstfewcrankangledegreesofsim ulationwhereaflowfieldisestablished,theflowstructureoftheintakestrokeisstartingtobecomevisibleata round424CAD,seeFigure4.Theresultsarepresentedonacuttingplanemade through the middle ofthe intakevalve and paralleltothe symmetry plane.

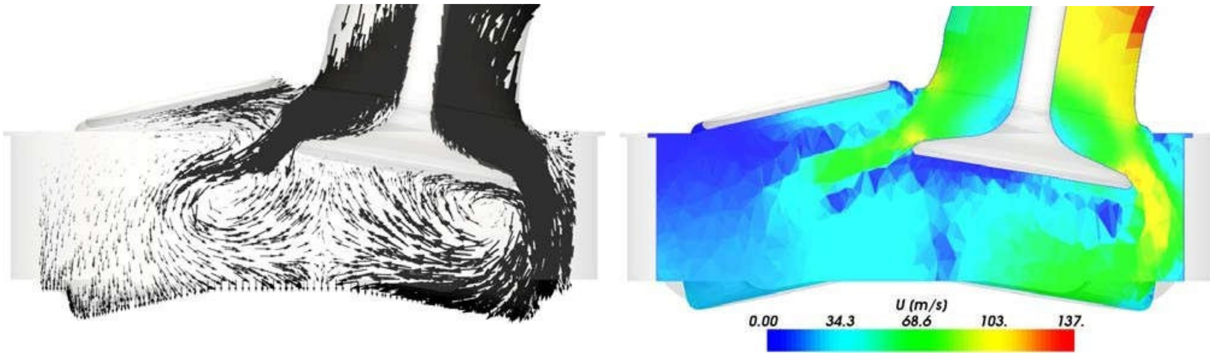


Figure 4..*Flow field (a) and velocity magnitude (b) at 424 CAD. Results are shown for a plane cut through themiddleof the intakevalveandparalleltothe symmetryplane.*

ItisvisibleinFigure4. That the two incoming jetsoneachside of the valve creates two strong vortexes, one a biton the left side of the valve and one under the valve plate since the flow is restricted by the liner wall.

7. CONCLUSION

This research paper simulations were carried out from the start of the intake stroke (IVO) until around the end ofthe combustion process, thus covering the intake, compression and part of the expansion phases.A significant amount of the total time spent on the work was dedicated to the mesh creation.Before the initial template mesh could be created, the geometry had to be cleaned and dividedinto sub-volumes which, together with the mesh creation itself, required a lot of time. Thisprocess could probably be made a lot faster with more experience and/or mesh software thatallowed for easier manipulation of the geometry and more options for mesh generation thenGambit possesses. The power of Gambit is in the Journal function, with which the rest of themeshes were created very fast and easily,taking only around 10 min each compared to theseveral hours needed to create the meshes by hand. In total, 18 different meshes were used tocoverthewholesimulationintervalandthemeshstrategyoftheMUMMIapproachwasshowntobesuccessfullyemplo yedalso forthis complex enginegeometry.

The initial calculations were started from IVO but without simulating the closure of the exhaustvalve during the valve overlap period. This lead to an overestimation of the volumetric efficiency since in reality a small part of the cylinder mass will be lost through the exhaust port during valve overlap. This resulted in a cylinder pressure that was higher than expected compared to the1D calculation, which was not acceptable. The conclusion is that either the closure of the exhaustvalve has to be simulated as well or the simulation has to be started from EVC. All the following calculations were there fore started from EVC.

An investigation was done using the *total pressure* boundary condition on an “infinite” volumeattached to the intake port, thus simulating the atmosphere. This allows the incoming mass flowto be calculated freely without the restriction of a fixed boundary condition. The calculatedcylinder pressure did not show satisfactory results at all during the intake stroke using thisstrategy, and it is clear how important it is with correct boundary conditions for the incomingmass flow. This is especially true for engines that use the pressure pulses in the intake system tocreate a ram effect before valve closure. This approach can be valid when there are no boundaryconditions available or for simple intake systems, otherwise it is important with time-varyingboundaryconditions for the massflow if the whole intakesystemis not simulated.

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