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Journal of Vibration Engineering

ISSN:1004-4523

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INTERNALCOMBUSTIONENGINE

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Abstract— This research paper strategy was previously applied to simplified geometries as well as real engine geometries.Thepurposeofthisworkistoshowitsvalidityforrealandcomplexenginegeometry,covering alonger 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 tumblemotion is notgenerated. Thiscould 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 lowforproperflamepropagationwereapparent,leadingtoinconsistencybetweenthecalculatedcylinderpressureof the CFD calculation and the calibrated1D calculation.

Keywords—simulationinterval,in-cylinderchargemotion,CFDcalculation,numericaldiffusion,enginegeometry.

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. ThethesisisconcernstheCFDmodelingofafourstroke, 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 modelsdoesnotaccountforallgeometrictopologies,makingCFDsimulationsanimportantcompliment.Itcanalso be a valuable tool to run numerous tests and in vestigation so fan engine without having to perform.

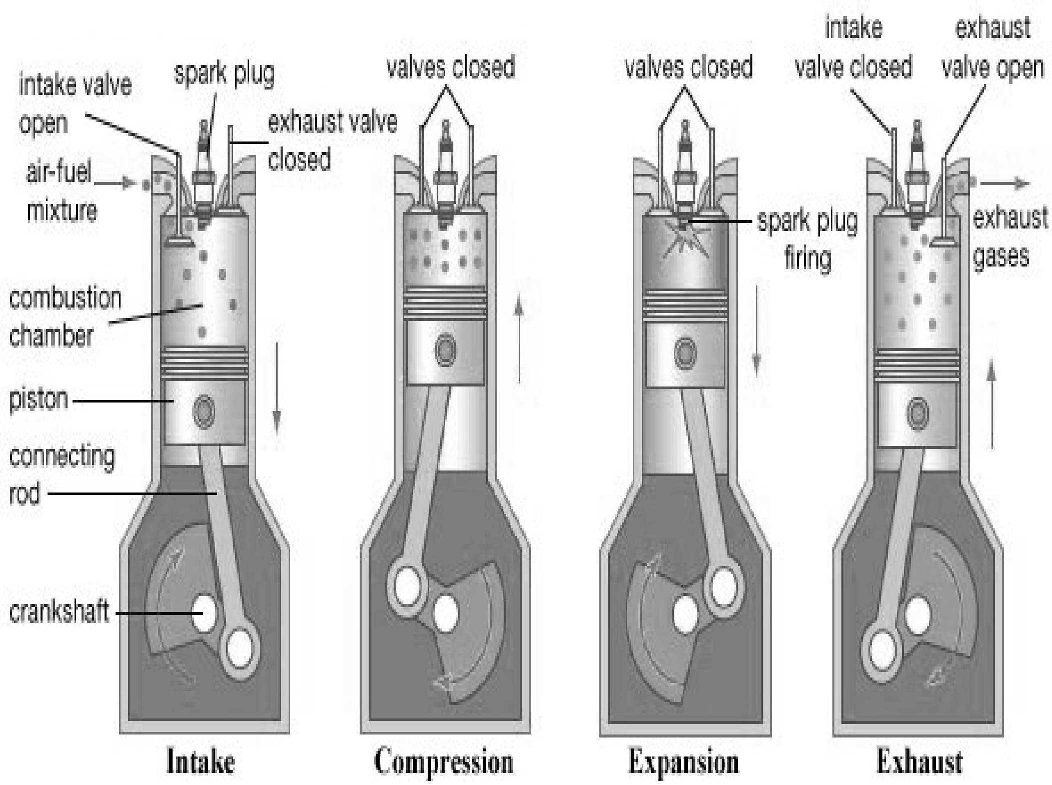
CFD research has unfortunately not yet reached a state where it completely describes all the processes that takes place in an IC engine. This is mainly due to the fact that it is a highly complex mechanical device which incorporates many, simultaneously interacting, thermo fluid and chemical processes. 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 turbulentflow,heatand masstransfer, 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 thissection a brief introduction to the Internal Combustion Engine (ICE) will be presented aswell as the purpose of this thesis. Lastly a brief summary of all the chapters of the thesis is presented.

2. *TheInternalCombustionEngine*

Theinternalcombustion(IC)engineistodaythemostwidelyusedenergysourceintheautomotiveandnavalindustry.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 outputpercylindervolumehascontinued toincrease.TheICenginecomesinmanydifferenttypesandsizesbut canbe divided in the category of the two stroke cycle or the four stroke cycle which can either be runbytheOttoortheDieselprinciple.ForOttoenginesthepre-mixedair-fuelmixtureisignitedbyasparkfroma sparkplug while for the Diesel principle the air is compressed beforehand in thecylinder and the incoming fuel sprayisignitedbythehighpressureandtemperature.Inthischapterabriefexplanationofthefourstroke,spark- ignited (SI/Otto) engine will be presented,sincethisisthetypeofenginethat the workofthe thesis is performed on.

The four stroke cycle starts with thepiston positioned in TDC (Top DeadCenter) and as thepistontravels 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 duringthe*exhauststroke*,see).WhenthepistonhasreachedTDCthecycleisrestartedagain.Inonepowercycle 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)

Figure1.*ThefourphasesofthefourstrokeICengine:intakestroke(a),compressionstroke(b), expansion orpowerstroke(c)and exhauststroke (d).*

ThecombustionprocessintheSIenginecanbedividedintofourstages:

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 flamespeed dependson thepressure, temperature, theair-fuelratio 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,causedbytheturbulencecreatedduringtheintakeandcompressionstroke. Thesmoothlaminarflamefrontisbrokendownbyvortexesintoaturbulentflamefrontwithavery irregularandwrinkledshape.Theflamespeednowreachesvelocitiesof10-50m/s.Thevelocitystill depends on pressure, temperature, theair-fuelratioandamountofresidualgases,buttheturbulenceintensityisthemostimportantfactorinthisphase.Increasing theenginespeedleadstomoregeneratedturbulence,thisinturnresultsinhigherflamevelocity.Thisleadstoa combustiontimethatisunchanged(expressedincrankangledegrees)allowingforhighenginespeeds fortheSI engine.
4. **Flame extinction.** Lastly the flame reaches the cylinder liner causing flame extinctionbecause of the liner walls relativelylow temperature. Thecombustionphaseisusuallycompletedaround30 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. AbnormalCombustionPhenomena

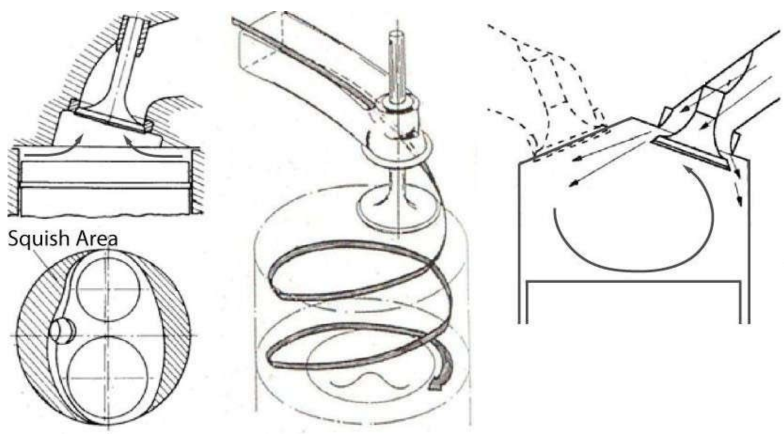
As the flame propagates across the combustion chamber the unburned mixture ahead of the flame(called the *end-gas*)is compressed, causing its pressure, temperature and density to increase. Some of the end-gas air-fuel mixture may then undergo chemical reactions prior to the normal combustion when the flame front reaches the end-gas. Thesechemical reactionsmay cause the end-gasto 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) orafter (postignition).Eitherway it meansthat thecombustion processis 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. ChargeMotionClassifications

As explained above, the turbulence level in the cylinder is very important for the turbulent flamepropagationand 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 flowcan bedivided in threemainmotion categories; *squish*,*swirl*and *tumble*,seeFigure 2. Squish isdefined as theflowcreatedwhenthepistonreachesTDCandtheouterpartofthepistoncrownapproachesthecylinderheadclosely, causingaflowmotiondirectedfromthecylinder linertowardsthe centerofthecombustion chamber,see Figure 2(a). Swirl is the rotational flow motion around the cylinderaxis and is created by the shape of the intake duct, seeFigure2(b).Bybringingtheintakeflowintothecylinderwithaninitialmomentumaswirlmotioniscreated, 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)

Figure2.*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 tumblemotionkeeps it sangularmomentumbut 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, butalsotodirectitmainly towardstheareaoftheexhaustvalve.Inatraditional intakeductthe incomingcharge is distributed quite uniformly around the valve causing a flow that counteract itself andhinder the creation of a main tumblemotion. By directing a biggerpartof theflow towards onesideofthevalve, preferablytowardsthe 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. *Purposeoftheresearchpaper*

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.AstrategytosolvethiswaspreviouslydevelopedbytheInternalCombustionEngineGroupat PolitecnicodiMilanowhichusesanumberofcomputationalgridsormeshestocoverthewholeenginecycle. Eachmeshisdeformedforacertainintervalandwhenthequalityofthemeshbecomestoolow,thesolution 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 actuallyperformedduetothefactthatnorealboundaryconditionswereused.Thus,thepurposeofthisthesiswasto usethesamestrategyofhoweachmeshisconstructedandgeneratedasinthepreviouscases,butthistimeemployed on thereal and highly complexenginegeometry of the F4engine, including theuseofrealisticboundary 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-implementedintheOpenFOAMcode,toinvestigateifarealisticcombustion 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 engineandismostlycreatedbythe designofthe intakeportandtheintakevalves.Furthermore,theF4enginehasa 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.

Asa subpurposeitwas also of interest to see ifthewholeprocess ofmeshgeneration andsimulation couldbedone 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. ChangeofAir-Fuel(A/F)ratioto12.4

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 therewillbe a certain mass that exits the cylinder throughtheexhaustvalve during valveoverlap period, and this was not takeninto accountfor.To solvethisproblem the simulation was setto start at exhaustvalve closure timesinstead. Newinitialconditions atEVC(388 CAD) for temperatureandpressurewasassigned fromthe Gasdyncalculations. The original boundary condition for massflow at the intake duct wasalso used, seeFigure3.14.

TheinitialcasehadanAir-Fuelratiocorrespondingtoastoichiometricmixture(14.7)whichisnotthecaseoftheF4 engine atfull load and 8000rpm. Instead the real A/F ratioshould be12.4whichcorresponds to a λvalueof0.84. 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 cylinderpressurewillbenegligiblebeforetheintakevalveclosesandthecompressionstrokestarts.Theresultofthe calculations for the first two changes can be seeming Figure3.

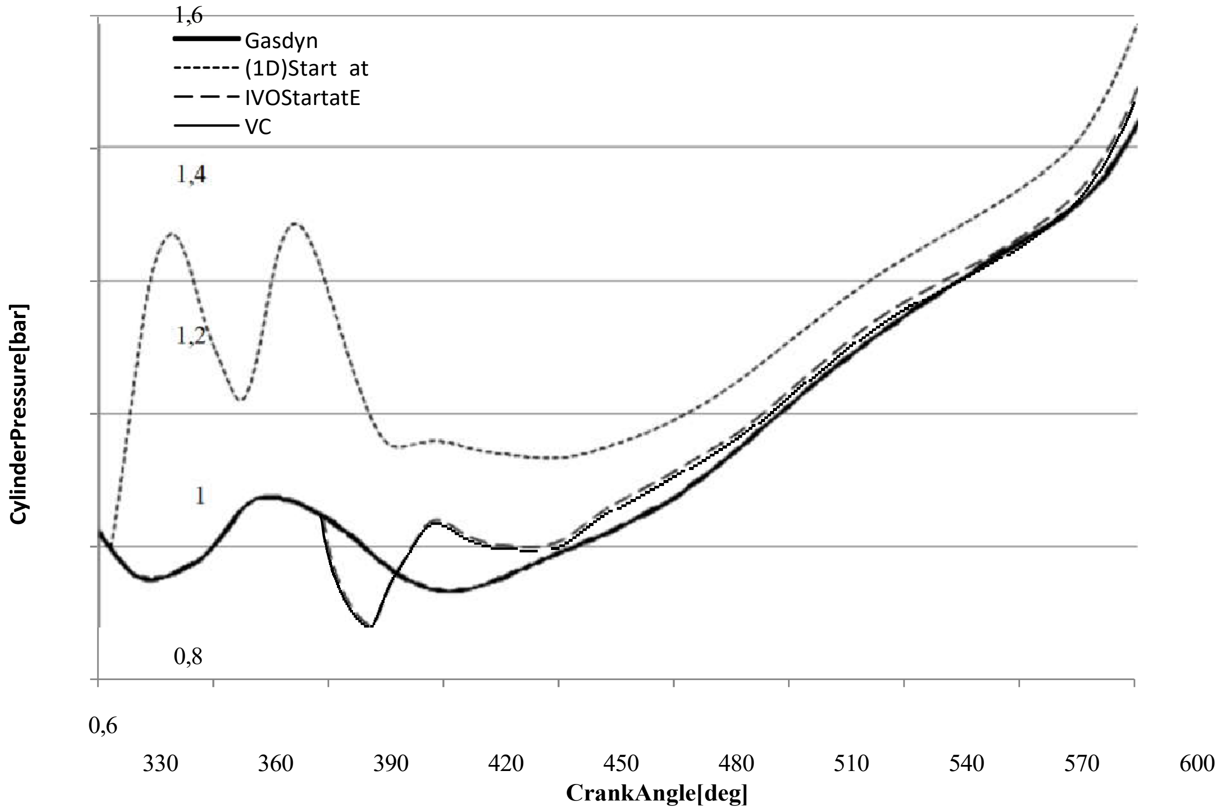


Figure3.*CylinderpressureduringintakestrokeforGasdyn(1D)andOpenFOAM(CFD)calculations,comparingthedifferenceofstartingthesimulationfromIVOandEVC,andchangingλfrom1tothecorrectvalueof0.84.*

Figure3showsthatthebiggestimprovementfortheconsistencybetweenthe1Dand CFDcalculationswasfromthechangeofsimulationstartingtime.Assuspected theoverestimation ofvolumetricefficiencywasmainlybecauseoftheneglectedvalveoverlapperiod.Asmallimprovementwas 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-makethemeshesfrom600–780 CAD. An investigation was made to confirm this by also re-making the meshesforthetwointervals before600CAD whichdidnot showanydifferencein cylinderpressure.

6.2 ResultsfromFinalIntakeandCompressionStrokeSimulation

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.

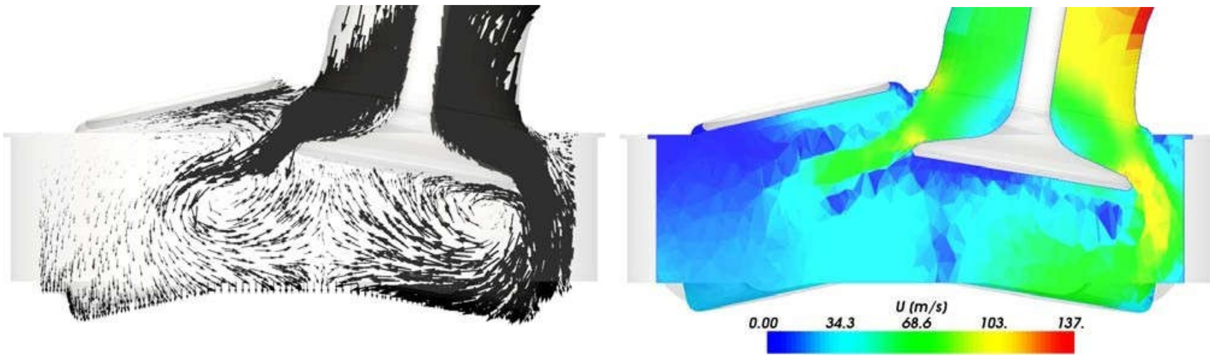


Figure 4..*Flow field (a) and velocity magnitude (b) at 424 CAD. Results are shown for a plane cut through themiddleof the intakevalveandparalleltothe symmetryplane.*

ItisvisibleinFigure4. Thatthetwoincomingjetsoneachsideofthevalvecreatestwostrongvortexes,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 ofthecombustionprocess,thuscovering theintake,compression andpartoftheexpansionphases.Asignificant amount of thetotal time spent on the work was dedicated to the mesh creation.Beforethe initial template mesh could be created, the geometry had to be cleaned and dividedinto sub-volumes which, together with the mesh creationitself,requireda lot oftime.Thisprocesscouldprobablybe madea lotfasterwithmore experience and/ormeshsoftwarethatallowedforeasiermanipulationofthegeometryandmoreoptionsformesh generationthenGambitpossesses.ThepowerofGambitisintheJournalfunction,withwhichtherestof themesheswerecreatedveryfastandeasily,takingonlyaround10mineachcomparedtotheseveralhours needed to create the meshes by hand. In total, 18 different meshes were used tocoverthewholesimulationintervalandthemeshstrategyoftheMUMMIapproachwasshowntobesuccessfullyemplo yedalso forthis complex enginegeometry.

TheinitialcalculationswerestartedfromIVObutwithoutsimulatingtheclosureoftheexhaustvalveduringthe valve overlap period. This lead to an overestimation of thevolumetric efficiency since in reality a small part of thecylindermasswillbelostthroughtheexhaustportduringvalveoverlap.Thisresultedinacylinderpressure thatwashigher than expectedcompared to the1Dcalculation,which wasnot acceptable.The conclusion isthat either the closure of the exhaustvalvehasto 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 allowsthe incoming mass flowto be calculated freelywithout 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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