AIJREAS VOLUME 2, ISSUE 8 (2017, August) (ISSN-2455-6300) ONLINE
ANVESHANA'S INTERNATIONAL JOURNAL OF RESEARCH IN ENGINEERING AND APPLIED SCIENCES

ONE ZONE THERMODYNAMIC MODEL SIMULATION OF AN IGNITION COMPRESSION ENGINE BY USING ANSYS

GUNNAM SEKHAR
M.Tech (THERMAL)
Department Of Mechanical Engineering,
kakinada Institute of technological sciences.
A. Agraharam-Ramachandrapuram East Godavari District.
Email id: sekhargunnam5270@gmail.com

Mr. M.CH. RAMA RAO
Assistant Professor, Mechanical Engineering Department.
kakinada Institute of technological sciences.
A.Agraharam-Ramachandrapuram East Godavari District.

Mr.D.VIJAYA PRASAD
Associate Professor,
Mechanical Engineering Department,
Kakinada Inistitute of technological sciences.
A. Agraharam-Ramachandrapuram East Godavari District.
Email id:vijayaprasad35@gmail.com

ABSTRACT

My end course project was performed in the exchange program between Faculae de Engenharia da Universidad do Porto (FEUP) and the University of Maryland, Baltimore County (UMBC). The project was proposed by Dr. Christian von Karce (CVK) who developed a thermodynamic engine model for which a computer program had been written for its implementation. The computer simulation was developed and implemented numerically by way of a "Skylab". However, heat transfer, Q (Q=0), was left out which results somewhat artificially in an "adiabatic engine". Also the combustion model used was based on a somewhat simple formulation. It was assumed that burned and unburned gases were homogeneously mixed and burning rate was a constant. Based on CVK work, my work has added heat transfer and a two zone combustion model that separates the action of the burned and unburned gases during the combustionThe result of my project is a computer simulation which may be used to obtain some fairly good estimates of engine performance. These estimates are most useful for understanding basic engine performance as well as assessing modifications as regards valve sizing, spark advance and various fuels. A particularly useful application is to do a compressor/turbine engine matching for turbo charging. The report is based on and extends a prior, informal, report by CVK.

Objectives :

The main propose of my work:

- Complete the program with the heat transfer model and insert the correct modifications to perform a simulation of a non- adiabatic engine;
- Add a new combustion model, replacing the existing one used in the initial program.

The new combustion model would take into account the turbulence in the cylinder and would then allow the variation of burn duration (which is fixed in the simple model used) to vary with engine speed. Despite this, I also had to understand the existing computer simulation implemented by CVK and the theoretical concepts behind it.

INTRODUCTION

This report presents the Thermodynamics theory describing the main physical phenomena occurring inside a spark ignition four stroke (4S) internal combustion engine (ICE) while it is running at steady speed (constant revolutions per minute, rpm). The mathematical form of the Thermodynamic theory is developed and implemented numerically by way of a

"Skylab" computer program. The result is an ICE computer simulation. This computer simulation may be used to obtain some fairly good estimates of engine performance in which the main effects of compression ratio, sparks timing, some aspects of valve timing, valve sizing, and fuel types, over a range of engine speeds. Of course not every detail of ICE performance can be accounted for, but depending on the physical details incorporated and their relative importance, many of the most important performance characteristics can be determined to a reasonable degree of accuracy. This report does not deal with any structural or mechanical aspects of an ICE beyond those of the basic geometric features relevant to the containment and external manifestations of the Thermodynamics processes occurring in the engine. These thermodynamic processes are idealized to a certain degree in order to reduce the complexity at this stage of development of the engine simulation. The simulation is based on the standard configuration of a reciprocating piston in a cylinder closed at one end, the cylinder 'head'.

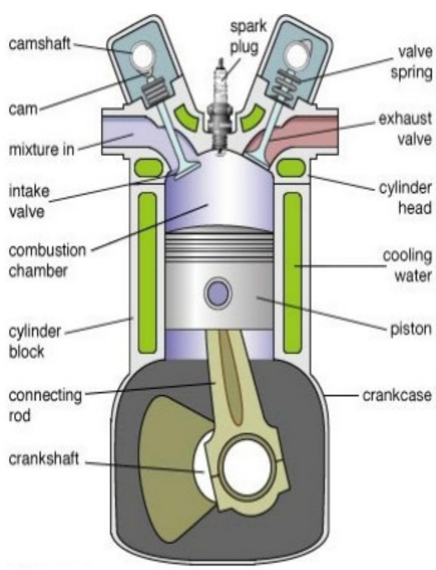


Figure 1.1 – Four stroke internal combustion engine.

The ICE Thermodynamics analysis is based on the following primary assumptions. All thermodynamics processes are assumed to be internally reversible. The working medium (fuel and air mixtures) is assumed to be an ideal gas with constant specific heats. The equations of state for the burned and unburned media are derived on the basis of equilibrium chemistry. The gas exchange process is based on quasi-steady compressible flow through an orifice. Further secondary assumptions and idealizations are discussed in the formulation of the thermodynamics model in the next chapters.

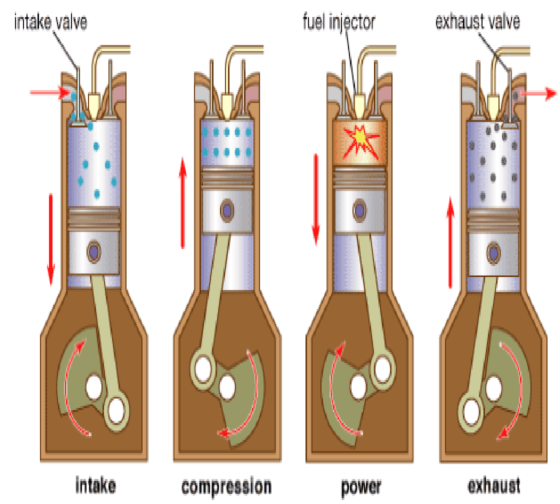


Figure 1.2 - A four-stroke spark ignition cycle.

The basic engine performance cycles are controlled by the crank rotation. The crank rotation in turn moves the piston up and down, thus varying the volume V of the space enclosed by the piston and cylinder. This varying volume is the primary controlling factor of the sequence of thermodynamic events occurring in the piston-cylinder space. Henceforth this space will be referred to simply as the cylinder. The crank rotation is measured in terms of the rotation angle θ shown in Figure 3.

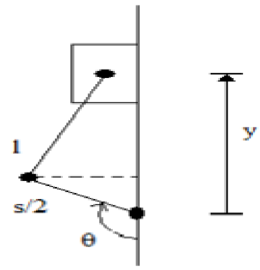


Figure 3 - Sketch of the slider crank model of piston-cylinder geometry

At θ=0 (2∗n∗π) the piston is at the bottom-most point in its travel. This point is called bottom center, BC. The cylinder volume V(θ) can be shown, by an analysis of the slider-crank mechanism to be,

$$v(\theta)=V_m*(\frac{1}{r_c}+\frac{1}{2}*(1-\frac{1}{r_c})*(1+\cos(\theta)+R_c-\sqrt{}$$

In formula (1.1), Vm is the (maximum) volume in the cylinder at BC, Rc is the ratio of connecting rod length to s , where s=stroke , and r c is the compression ratio V_M/V_C

, where Vc is the (minimum) volume of the cylinder at top center.θ=π 2*(n-1)* π Vc is called the clearance volume and Vd=Vm−Vc is the "displacement" volume, the usual measure of engine capacity or, more commonly, engine size. The calculation of the instantaneous volume equation (1.1) is discussed in "Appendix A.1". Using Vm as an input variable, some other variables such as the Vc, Vd , bore (bo) and stroke (s) have to be calculated in order to proceed. It was assumed that the bore was equal to the stroke in order to simplify some equations and to use the minimum ones possible,

$$V_d=\frac{\pi}{4}*b^2*s$$

$$\frac{Vd}{Vc}=rc-1 \quad (1.3$$

$$Vc=\frac{Vm}{rc} \quad (1.4)$$

$$h_c=\frac{4*V_c}{\pi*bo} \quad (1.5)$$

assuming that the shape of the clearance volume is a cylinder with diameter bo and height, hc

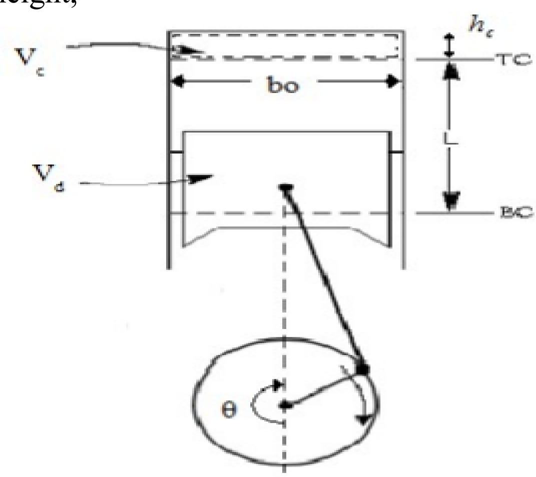


Figure 4 – Basic geometry of the internal combustion engine

The exposed total cylinder area Aw(θ) is the sum of the cylinder clearance area and the displacement area,

Aw=Ad+Ac

$$A_w=\frac{v-V_c}{bo/8}+A_c$$

$$A_c=\pi*bo*h_c+\frac{\pi}{2}bo^2$$

The operation of the valves is synchronized to the motion of the piston by way of gear or chain drives from the crankshaft. This is not shown in Figure 2. There will be no need for a description of this mechanism here since it will not be made use of. For the present it is only necessary to describe the valve configuration and actual motion of the valves as a function of the crank angle θ. This will be reserved for the chapter on the gas exchange process in order to keep the exposition simple at this stage. The function, of θ, describing the motion of the valves is given as part of the basic engine specifications utilized in this study. The camshaft and valve actuation mechanism must then be designed to realize this valve motion function. For this the reader is referred to books on engine and mechanism design given in the bibliography.

1.2.The Equations of State of the Working Gases

$$PV_u = m_u * R_u * T$$

$$u_u = Cv_u * T + hf_u$$

$$PV_b = m_b * R_b * T$$

$$u_b = Cv_b * T + hf_b$$

then, if the gases are homogeneously mixed,

$$PV = m*(xR_b + (1-x)R_u)*T = m$$

$$U = m*u = m*(x*Cv_b + (1-x)*Cv_u)*T + x*$$

CVK utilized the homogeneously mixed charge described by equations (1.13) and (1.14), where dx dx(t)/ dt is an empirically determined burning rate. One of the main aims of this study is to implement a more realistic combustion model in which unburned and burned gases remain separated, ie., a "two zone" model.

1.3.Thermodynamics and Mathematical Model of the Engine

The engine operates in a two-cycle (4 stroke) mode. Each cycle consists of a complete rotation, through an angle of 2π, of the crank. The first cycle is called the power cycle in which both valves are closed and the power of the engine is produced. This cycle consists of the compressions stroke roughly from θ=0 to θ=π, in which the fuel-air mixture in the cylinder is compressed, followed by the expansion stroke, roughly from θ=π to θ=2π, in which the main positive engine work is done. The second cycle is called the gas exchange cycle in which the burned gases from the expansion stroke are expelled (exhaust stroke, θ=2π to θ=3π) and fresh fuel-air is ingested (intake stroke, θ=3π to θ=4π). These two cycles are completely described by the main two laws of thermodynamics governing the unburned and burned fuel-air mixture in the cylinder. The two laws are the Conservation of Mass,

$$dm/dt = m_i - m_e$$

and the Conservation of Energy (1st Law of Thermodynamics),

$$\frac{dU}{dt} = \tilde{Q} - \tilde{W} + \tilde{H}_i - \tilde{H}_e$$

Here m is the mass of burned plus unburned fuel-air mixture in the cylinder at any instant of time t, m_i and m咏e are the mass flow rates into and out the cylinder, respectively Q咏,W咏, H $_i$ and H e is the heat rate into or out of the cylinder, the work rate into or out of the cylinder, the enthalpy flow rate into the cylinder, and the enthalpy flow rate out the cylinder respectively. In the application of these equations to the analysis of the engine cycles at constant speed (constant crank angular velocity, ω), it is the most convenient to transform time t to angle θ by the equation ω∗dt=dx . Then the time derivatives in equations (1.15) and (1.16) are replaced by angle derivatives and the ~ over symbols is replaced by ^ over the symbols to signify that rates are with respect to angle instead of time. Then,

$$\frac{dm}{d\theta}=\hat{m}_i-\hat{m}_e$$

$$\frac{dU}{d\theta}=\hat{Q}-\hat{W}+\hat{H}_i-\hat{H}_e$$

LITERATURE REVIEW

In this chapter a review of detailed literature survey conducted is presented. Topics covered are simulation of spark ignition engine processes, lean burn engine, and extended expansion engine. Since the present work involved variation of valve timing, literature pertaining to variable valve timing is also reviewed.

2.1 SIMULATION OF SPARK IGNITION ENGINE PROCESSES

Jerald A. Caton (2001) developed a thermodynamic cycle simulation for a spark-ignition engine which included the use of multiple zones for the combustion process. This simulation was used for the complete analysis of a commercial, spark-ignition V-8 engine operating at part load condition. This simulation is limited to only in-cylinder processes. For simplicity, the combustion chamber was assumed to be cylindrical shape. In this work, all cylinders of a multiple-cylinder engine are assumed to be identical, and assumed to follow the same thermodynamic process, and to operate with identical conditions. Overall results for a multiple cylinder engine are obtained by multiplying the results from the single-cylinder analysis by the number of cylinders. The primary feature of this cycle simulation is the first law of thermodynamics which is utilized to derive expressions for the time (crank-angle) derivative of the pressure, gas temperature, and volume in terms of engine design variables, operating conditions, and sub-model parameters.

Abd Alla (2002) has presented the preliminary simulation of a fourstroke spark ignition engine. The Wiebe's heat release formula was used to 19 predict the cylinder pressure, which was used to find out the indicated work done. The heat transfer from the cylinder, friction and pumping losses were also taken into account to predict the brake mean effective pressure, break thermal efficiency and break specific fuel consumption. Most of the parameters that can affect the performance of the four stroke spark ignition engines such as equivalence ratio, spark timing, heat release rate, compression ratio, compression and expansion index were studied. The use of a real combustion curve has a profound influence on the similarity of the pressure volume profile to that seen in a real engine. The modeling process is obviously getting closer to reality and is now worth pursuing as a design aid.

Jehad A.A. Yaminl et al (2003) studied effect of combustion duration on the performance and emission characteristics of propane-fueled 4-stroke S.I. engines. The analytical model developed includes the compression, combustion and expansion process. Two zone combustion model was adopted for combustion process and then calculations then proceed in three phases. Firstly, the initiation of the combustion, then the subdivision of the combustion chamber into two zones separated by spherical flame front and, finally a single zone encompassing the whole of the combustion chamber. To initiate combustion a unit mass of the cylinder content is considered to burn at constant volume. The internal energy of the initial reactants were set equal to the internal energy of the products. It was assumed that 12 species; H2O, H2, OH, H, N2, NO, CO2, CO, O2, O and Ar were present in the combustion products both inside the cylinder as well as in the exhaust. Model for NOX emission and CO emission were also described in detail. The developed model was validated with other researchers experimental results.

Yanlin Ge et al (2005) studied the performance of an air-standard Otto cycle with heat transfer loss and variable specific heats of working fluid 20 by using finite-time thermodynamics. Relationships between the power output and the compression ratio, between the thermal efficiency and the compression ratio, as well as the optimal relationship between power output and the efficiency of the cycle are derived. Moreover, the effect of heat transfer loss and variable specific heats of working fluid on the cycle performance were also analyzed. Results show that the effect of heat transfer loss and variable specific heats of working fluid on the cycle performance are obvious, and they should be considered in cycle analysis.

Abu-Nada et al (2006) did a thermodynamic analysis of spark-ignition engine. A theoretical model of Otto cycle, with a working fluid consisting of various gas mixtures, was studied. It is compared to those which use air as working fluid with variable temperature specific heats. A wide range of engine parameters were studied, such as equivalence ratio, engine speed, maximum and outlet temperatures, brake mean effective pressure, gas pressure, and cycle thermal efficiency. For example, for the air model, the maximum temperature, brake mean effective pressure (BMEP), and efficiency were about 3000 K, 15 bar, and 32%, respectively, at 5000 rpm and 1.2 equivalence ratio. On the other hand, by using the gas mixture model under 21 the same conditions, the maximum temperature, BMEP, and efficiency were about 2500 K, 13.7 bar, and 29%. However, for the air model, at lower engine speeds of 2000 rpm and equivalence ratio of 0.8, the maximum temperature, BMEP, and efficiency were about 2000 K, 8.7 bar, and 28%, respectively. Futher by using the gas mixture model under these conditions, the maximum temperature, BMEP, and efficiency were about 1900 K, 8.4 bar, and 27%, i.e. with insignificant differences. Therefore, it is more realistic to use gas mixture in cycle analysis instead of merely assuming air to be the working fluid, especially at high engine speeds.

2.1.1 Summary

To model engine processes thermodynamically for the present theoretical analysis, the above literature gives in depth knowledge. Modeling of engine processes includes compression, combustion, expansion and gas exchange. Ganesan (1999) literature helps to model all the above processes 22 and Jerald A. Caton (2000), Siew Hwa Chan and Zhu (2001) and Abd Alla (2002) literature gives an idea for combustion model using Wiebe function. Some researchers attempted multi-zone combustion modeling, where as in the present theoretical analysis the author attempted combustion model using Wiebe function. Kodah et al (2000) has presented the importance of the selection of the constant value of 'a' and 'm' in Wiebe function to fit with experimental data. Ganesan (1999) and Abd Alla (2002) literature is useful to model the heat transfer from the cylinder, friction and pumping losses for present theoretical analysis. Jehad A.A. Yaminl et al (2003) and Yusaf et al (2005) work helps to model equilibrium calculations. Maher Jehad A.A. Yaminl et al (2003) and Maher A.R. Sadiq Al-Baghdadi (2006) literature gives in depth knowledge NOX emission modeling using Zedlovich mechanism and CO emission by equilibrium calculations.

Author and Year	Modifications done to achieve lean burn	Remarks
Increasing swirl, squish and turbulence		
Souich Matsushita et al (1985)	Helical port with swirl control valve and programmed sequential fuel injection and feed back control of air fuel ratio.	20% fuel economy, also gives high performance and reduces emissions.
Chau et al (1988)	Four – valve head with a centralized spark plug, a two-valve head with a helical swirl intake port and two-valve head with straight intake port.	Best mixing was found in the four – valve head, which requires the least spark advance and was able to burn the leanest mixture.
Yuhiko Kiyota et al (1992)	Barrel-stratification.	Extremely lean conditions (maximum of A/F ratio 32), indicated thermal efficiency improved by 15% and reduction in NO_x emission achieved.
Chulho Yu et al (1995)	Swirl Control Valve, injection, intensifying ignition system and Cu-ZSM-5 catalyst.	Extended Lean misfire limit to A/F ratio 23. Therefore, NO_X emission was reduced 60.6%, fuel economy was improved 10.6%.
Heinz-Jakob NeuBer and Jose Geiger (1996)	Continuous Variable Tumble System.	Reductions in fuel consumption by 5 - 10 % compared to existing lean burn engine and at the same time, NOx-emissions reduced by approximately 85 %.

METHODOLAGY

3.1 Power Cycle

This chapter presents the thermodynamics theory describing the main physical phenomena occurring inside an ICE. The thermodynamic models of the four movements, or strokes, of the piston before the entire engine firing sequence is repeated

In this cycle, the valves are closed so there is no mass exchange and it is where the main power of the engine is produced. This cycle consists in a complete rotation which is characterized by two stages: (a) Compression stage - roughly from θ=0 to just before the spark plug goes off, 炶s=0.88*π , in which the fuel-air mixture in the cylinder is compressed. The value of 炶s given is typical. All engines start combustion before TC θ=∏ . This called spark advance. (b) Combustion stage - roughly from θs to just before exhaust valve opens, 炶evo=2*∏- δ , where depending on some factors, such as the flame speed and piston speed. Combustion begins during compression and most expansion.

3.2. Compression stage

3.2.1 Thermodynamic Model of the compression stage

During this stage, the energy balance on the in-cylinder gas is,

$$\frac{dU}{d\theta}=\hat{Q}-\hat{W}$$

As both valves are closed there is no mass exchange so

$$\frac{dm}{d\theta}=\hat{m}_i=\hat{m}_e=0$$

After the algebraic manipulation shown in "Appendix Above equation becomes,

$$\frac{dT}{d\theta}=\frac{\hat{Q}}{m*A}-(xR_b+(1-x)R_u)\frac{T}{AV}\frac{dV}{d\theta}-\frac{\Delta CvT+\Delta hf}{A}\frac{dx}{d\theta}$$

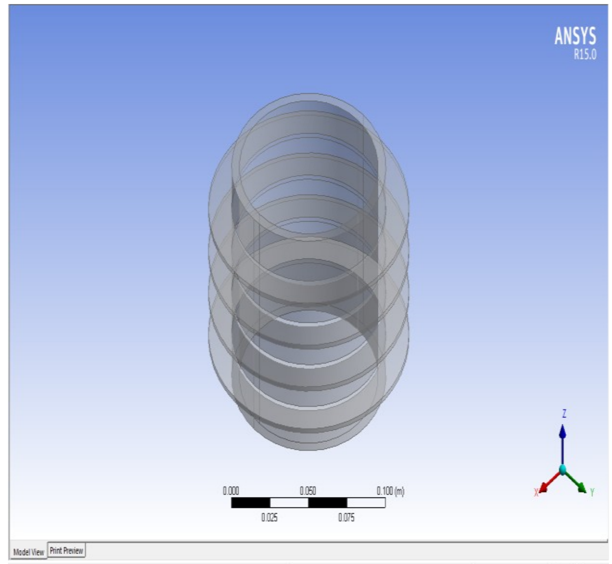
where the quantity A=x*Δ Cv+Cvu

At this stage, the mass of gases in the cylinder is primarily unburned. However, there is a small amount of burned gas, called residual gases mr which remains after the exhaust stroke. These

gases are homogeneously mixed, where x= mr/ m . Thus the above equations do apply for this part of the cycle.

RESULTS

The single zone with homogeneously mixed burned and unburned gases developed by CVK is called the single zone adiabatic model. I have added heat transfer to this model and it will be called the single zone heat transfer model so first the results obtained from the single zone heat transfer model simulation program (see appendix C.3.) are discussed. After that, it is discussed the main simulation program and how the power, efficiency and heat transfer vary with the engine speed and how this have influence on the engine performance. Another point explained is the turbulence model and how it affects the combustion stage. As illustration an engine with a compression ratio of 11, total volume of 55 cm3 , using methane (CH4) which properties are very similar to gasoline and setting the engine to run at 6000rpm was obtained the following results for the heat transfer model,



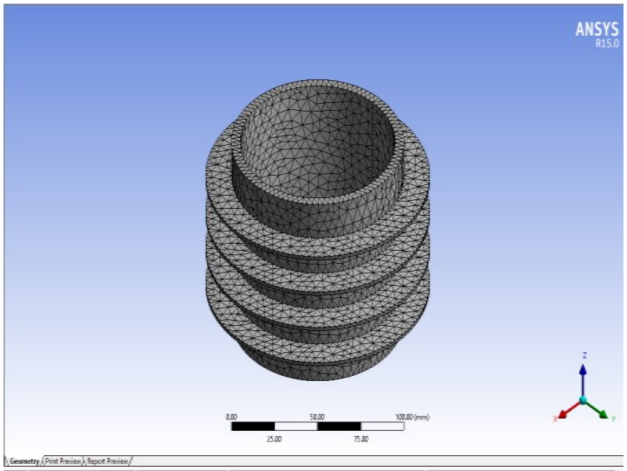
The fig 4.1 shows that study state thermal plane of the model design

Sketching Modeling

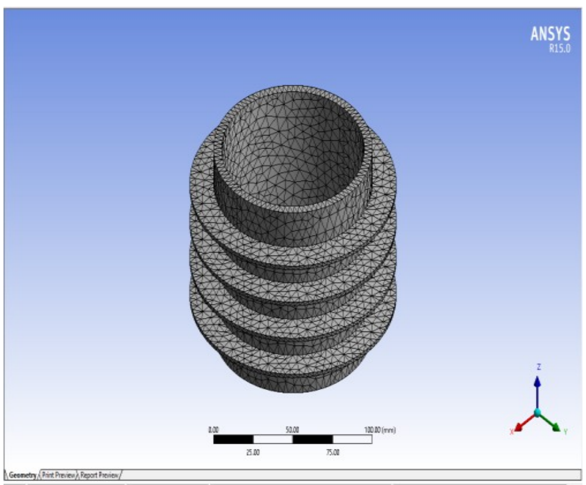
Details View

Details of Import1	
Import	Import1
Source	D:\EDUCATION\New Project\Kits P...\Part1.igs
Base Plane	XYPlane
Operation	Add Frozen
Solid Bodies	Yes
Surface Bodies	Yes
Line Bodies	No
Simplify Geometry	No
Simplify Topology	No
Heal Bodies	Yes
Clean Bodies	Normal
Stitch Surfaces	Yes
Tolerance	Normal
Replace Missing Geometry	No
Refresh	No

Table 4.1 shows that study state thermal properties



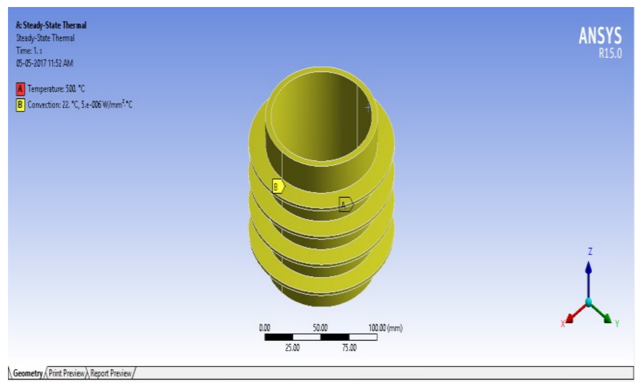
The fig 4.2 shows that study state meshing model



The fig 4.2 shows that triangle surface meshing model

The table 4.2 shows that triangle surface meshing properties

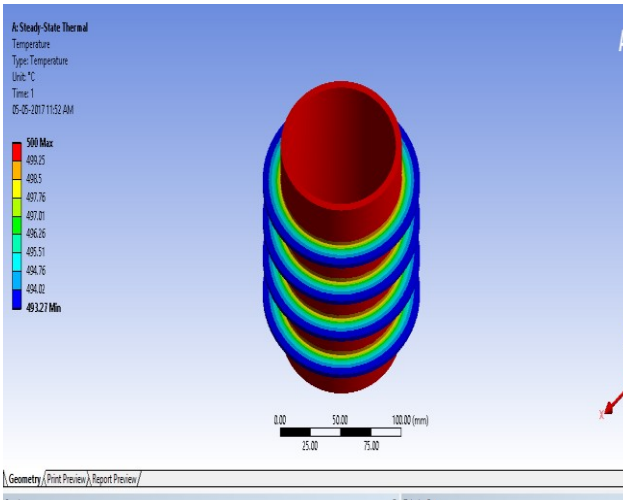
Details of "Mesh"	
Defaults	
Physics Preference	Mechanical
Relevance	0
Sizing	
Inflation	
Patch Conforming Options	
Triangle Surface Mesher	Program Controlled
Patch Independent Options	
Topology Checking	Yes
Advanced	
Defeaturing	
Statistics	
Nodes	34320
Elements	17388
Mesh Metric	None



The fig 4.3 shows that study state thermal applied the temprature

The table 4.3 shows that details of thermal state

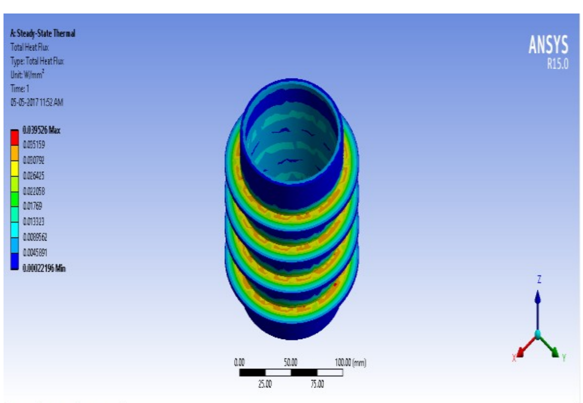
Details of "Steady-State Thermal (A5)"	
Definition	
Physics Type	Thermal
Analysis Type	Steady-State
Solver Target	Mechanical APDL
Options	
Generate Input Only	No



The fig4.4 shows that the study state model maximum value 500and minimum 493.27

The table 4.4 shows that properties of body temprature

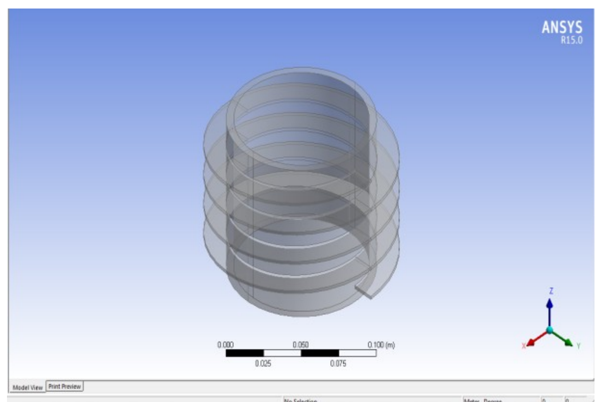
Details of "Temperature"	
Scope	
Scoping Method	Geometry Selection
Geometry	All Bodies
Definition	
Type	Temperature
By	Time
Display Time	Last
Calculate Time History	Yes
Identifier	
Suppressed	No
Results	
Minimum	493.27 °C
Maximum	500. °C
Minimum Value Over Time	
Minimum	493.27 °C
Maximum	493.27 °C



The fig 4.5 shows that the view of plane maximum value 0.039526 minimum value 0.00221596

The table 4.5 shows that total heat flux properties

Details of "Total Heat Flux"	
Scope	
Scoping Method	Geometry Selection
Geometry	All Bodies
Definition	
Type	Total Heat Flux
By	Time
Display Time	Last
Calculate Time History	Yes
Identifier	
Suppressed	No
Integration Point Results	
Display Option	Averaged
Average Across Bodies	No
Results	
Minimum	2.2196e-004 W/mm²
Maximum	3.9526e-002 W/mm²



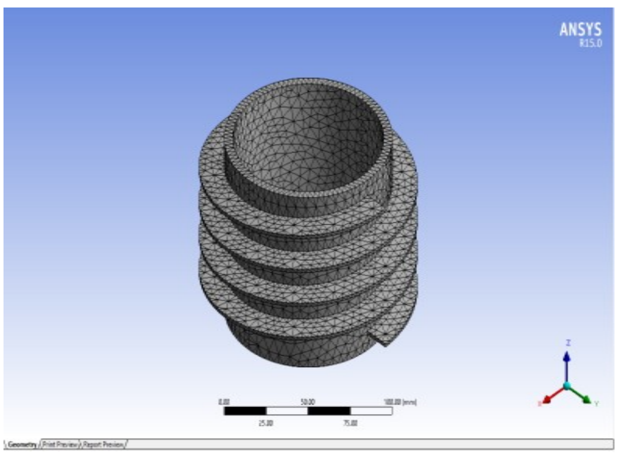
The fig 4.6 shows that stedy state thermal designing model

The table 4.6 shows that geometric modelling properties

Sketching | Modeling

Details View

Details of Import1	
Import	Import1
Source	D:\EDUCATION\New Project\Kits P...\Part2.igs
Base Plane	XYPlane
Operation	Add Frozen
Solid Bodies	Yes
Surface Bodies	Yes
Line Bodies	No
Simplify Geometry	No
Simplify Topology	No
Heal Bodies	Yes
Clean Bodies	Normal
Stitch Surfaces	Yes
Tolerance	Normal
Replace Missing Geometry	No
Refresh	No

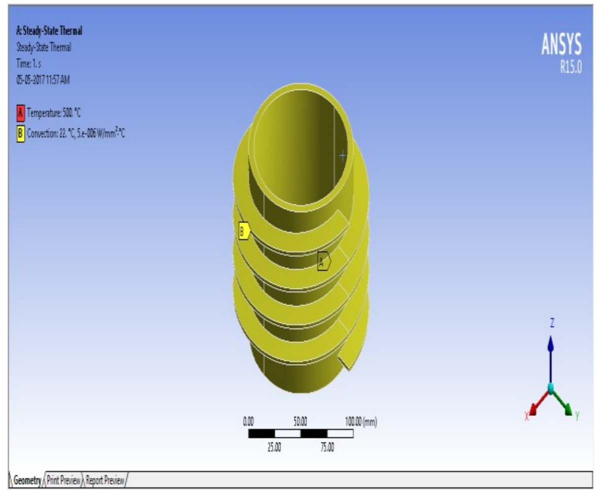


The fig 4.7 shows that 3D Meshing model

The table 4.7 shows that 3D meshing reports

Details of "Mesh"

Defaults	
Physics Preference	Mechanical
Relevance	0
Sizing	
Inflation	
Patch Conforming Options	
Triangle Surface Mesher	Program Controlled
Patch Independent Options	
Topology Checking	Yes
Advanced	
Defeaturing	
Statistics	
Nodes	34605
Elements	17480
Mesh Metric	None

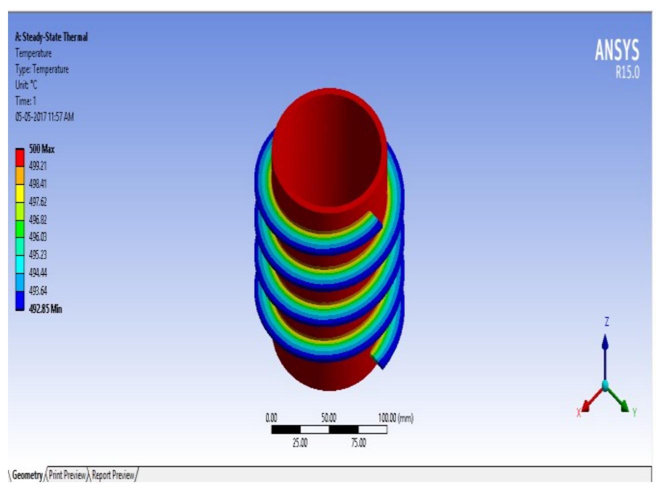


The fig4.8 shows that maximum heat flux of the 3D model

The table 4.8 shows that the total heat flux report

Details of "Steady-State Thermal (A5)"

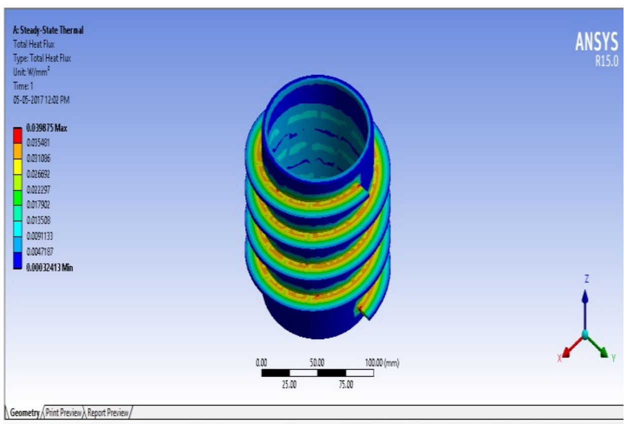
Definition	
Physics Type	Thermal
Analysis Type	Steady-State
Solver Target	Mechanical APDL
Options	
Generate Input Only	No



The fig 4.9 shows that maximum &minimum bending moments 500&402.85

The table 4.9 shows that details of body tempratures

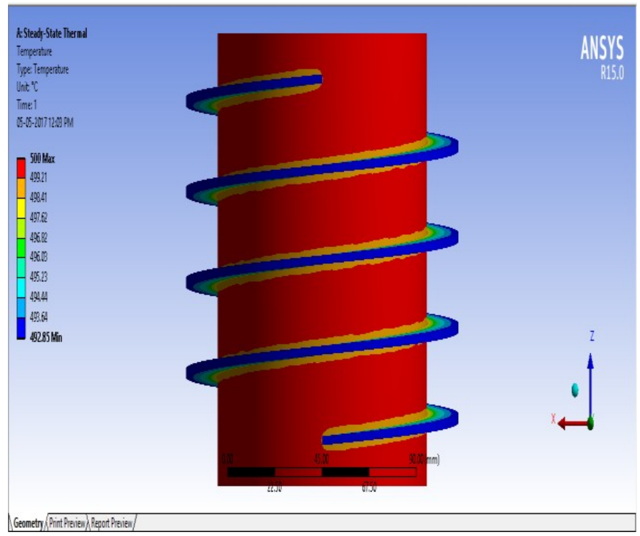
Details of "Temperature"	
Scope	
Scoping Method	Geometry Selection
Geometry	All Bodies
Definition	
Type	Temperature
By	Time
Display Time	Last
Calculate Time History	Yes
Identifier	
Suppressed	No
Results	
Minimum	492.85 °C
Maximum	500. °C
Minimum Value Over Time	
Minimum	492.85 °C
Maximum	492.85 °C



The fig4.10 shows that the maximum value 0.039875&minimum moment 0.00032413

The table 4.10 shows that total hetflux in all bodies reports

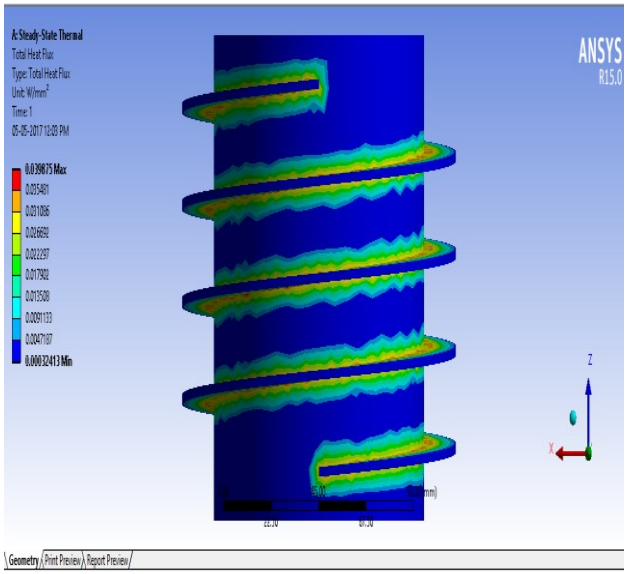
Details of "Total Heat Flux"	
Scope	
Scoping Method	Geometry Selection
Geometry	All Bodies
Definition	
Type	Total Heat Flux
By	Time
Display Time	Last
Calculate Time History	Yes
Identifier	
Suppressed	No
Integration Point Results	
Display Option	Averaged
Average Across Bodies	No
Results	
Minimum	3.2413e-004 W/mm²
Maximum	3.9875e-002 W/mm²



The fig 4.11 shows that over bending moment is 500max and 492.85min

The table 4.11 shows total geometric reports

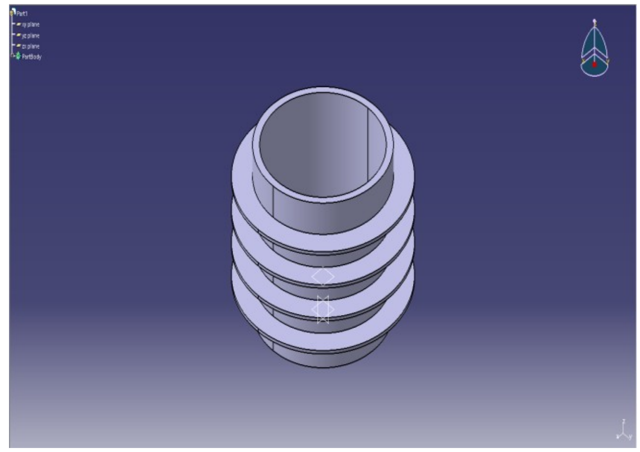
Details of "Temperature"	
Scope	
Scoping Method	Geometry Selection
Geometry	All Bodies
Definition	
Type	Temperature
By	Time
Display Time	Last
Calculate Time History	Yes
Identifier	
Suppressed	No
Results	
Minimum	492.85 °C
Maximum	500. °C
Minimum Value Over Time	
Minimum	492.85 °C
Maximum	492.85 °C



The fig 4.12 shows that total heat flux in 3D view

The table 4.12 total heat flux reports

Details of "Total Heat Flux"	
Scope	
Scoping Method	Geometry Selection
Geometry	All Bodies
Definition	
Type	Total Heat Flux
By	Time
Display Time	Last
Calculate Time History	Yes
Identifier	
Suppressed	No
Integration Point Results	
Display Option	Averaged
Average Across Bodies	No
Results	
Minimum	3.2413e-004 W/mm^2
Maximum	3.9875e-002 W/mm^2



The fig 4.13 shows that asending of 3D view in designing model

CONCLUSIONS

It was found that the model could be used in simulating any diesel engine. This could save an enormous amount of time in tuning an engine, especially when little is known about the engine. With an air-fuel ratio and volumetric efficiency map, Injection-timing could be optimized, thus minimizing wear-and-tear on the engine and dynamometer equipment. Much research could be directed towards refining the model and using it for the improvement of engine performance and reducing the NOx emissions by testing different fuels.

The fundamentals principles which govern internal combustion engine design and operations were well developed and implemented using the "Scilab" computer program. All the objectives proposed were achieved. For the heat transfer model the results obtained were quite good. However, the results obtained from the last simulation where it was added the heat transfer model plus the combustion model were not the ones expected. As mentioned previously, the model used was oversimplified, thus leading to quantitatively erroneous results although the results were qualitatively correct. We can conclude that treating the combustion model CVK developed is simpler than the one developed during this project and leads to the good results. However, the CycleComQC is more realistic because it presents a non-adiabatic engine, and the two zone model inside the combustion chamber which takes into account the turbulence inside it. However, some of the details need to be improved in order to give quantitatively more accurate results. We have concluded that the main difficulty is the pressure approximation. It is felt that with a little more time the pressure calculation can be improved by a new model that we have developed.

REFERENCES

[1] C.D. Rakopoulos, E.G. Giakoumis and D.C. Kyritsis - "Validation and sensitivity analysis of a two zone Diesel engine model for combustion and emissions prediction", Energy Conversion and Management 45 (2004).

[2] Jeremy L. Cuddihy - University of Idaho, "A User-Friendly, Two-Zone Heat Release Model for Predicting Spark-Ignition Engine Performance and Emissions", May 2014.

[3] "Computer Simulation of Compression Engine" by V. Ganesan –1 st edition, 2000.

[4] J. Heywood, Internal Combustion Engine Fundamentals. Tata Mcgraw Hill Education, 2011. [5] V. Ganesan, Internal Combustion Engines, 6 th edition, Tata Mcgraw Hill Education, 2002. [6] Zehra Sahin and Orhan Durgun - "Multi-zone combustion modeling for the prediction of diesel engine cycles and engine performance parameters", Applied Thermal Engineering 28 (2008).

[7] G. P. Blair, Design and Simulation of Four Stroke Engines [R-186]. Society of Automotive Engineers Inc, 1999. [8] C.D. Rakopoulos, K.A. Antonopoulos and D.T. Hountalas -"Multi-zone modeling of combustion and emissions formation in DI diesel engine operating on ethanol–diesel fuel blends", Energy Conversion and Management 49 (2008) 625–643.

[9] Hsing-Pang Liu, Shannon Strank, MikeWerst, Robert Hebner and Jude Osara - "COMBUSTION EMISSIONSMODELING AND TESTING OF CONVENTIONAL DIESEL FUEL", Proceedings of the ASME 2010, 4th International Conference on Energy Sustainability (May 17-22, 2010).

[10] A. Sakhrieh, E. Abu-Nada, I. Al-Hinti, A. Al-Ghandoor and B. Akash - "Computational thermodynamic analysis of compression ignition engine", International Communications in Heat and Mass Transfer 37 (2010) 299–303 .

[11] D. Descieux, M. Feidt - "One zone thermodynamic model simulation of an ignition compression engine", Applied Thermal Engineering 27 (2007) 1457– 1466. [12] Mike Saris, Nicholas Phillips - Computer Simulated Engine Performance, 2003