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COMPUTATIONAL FLUID DYNAMIC MODEL FOR CRANKCASE FLOW ANALYSIS OF NEW TWO STROKE DIESEL ENGINE

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ABSTRACT

A multipurpose computational engine crankcase model of KUKTEM two-stroke modular diesel engine has been developed to analyze the crankcase interior flow. The complexity of crankcase flow is due to the occurrence of reed valve before the crankcase and upstream the crankcase at the auxiliary port of the engine. Furthermore, current computational fluid dynamic solver is unable to handle the multi-physics interaction of reed valve petal with air. The crankcase flow is investigated by profiling the pressure built-up inside the crankcase geometry and examination of flow pattern during transient operation. The methodology is based on solution of computational fluid dynamic (CFD) formulation. The governing equation of CFD and robust k-e model are coupled with moving dynamic mesh model and sliding mesh model to represent the isotropic motion of piston and rotational motion of crankshaft and balancing shaft. The inputs to this model are based on motoring results of the same base engine. The pressure fluctuation inside the inlet tract is used as the pressure inlet boundary while the cylinder pressure upstream of the crankcase is set as the pressure outlet boundary. The modeling results shows that the maximum pressure inside the crankcase can achieve its ideal compression pressure. The compression ratio is about 1.14 bar based on nominal crankcase compression ratio but the trend of plot having large deviation due to ineffective reed valve model. Better solution of the flow can be achieved if proper reed valve model is available where the pressure at inlet and the outlet boundary can be calculated simultaneously.

INTRODUCTION

Research on two-stroke engine performance is widely studied by researchers all over the world by early 70's. Reports on experimental and computational effort of in cylinder flow and combustion process were found in many publications but yet very little focus is given on the crankcase compression process. The main reason for these occurrence maybe resulted from understanding that the engine performance mainly influence only by in-cylinder process especially in-cylinder flow and combustion. But in the case of conventional two stroke engine, crankcase compression largely affects the engine performance since the process provides useful work to compress the fresh air before

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entering cylinder. The pressurized air is needed to expel the burned gases inside cylinder volume. The higher the fresh air being compressed, then the higher momentum they have when entering the cylinder and the better scavenging will be. Thus, the latest trends in crankcase engine design for two stroke engine are emphasizing on better compactness and higher crankcase compression ratio. The crankcase compression ratio is defined as:

$$\boldsymbol{CR_{CC}} = \frac{\boldsymbol{V_{CC}} + \boldsymbol{V_{SV}}}{\boldsymbol{V_{SV}}} \tag{1}$$

where $\boldsymbol{V_{CC}}$: crankcase clearance volume or crankcase volume at BTDC and
$\boldsymbol{V_{SV}}$: total swept volume which is define as

$$\boldsymbol{V_{SV}} = \boldsymbol{n}\frac{\boldsymbol{\pi}}{\boldsymbol{4}}\boldsymbol{d_{bo}}^2\boldsymbol{L_{st}} \tag{2}$$

where d_{bo} : diameter of bore
$\boldsymbol{L_{st}}$: length of stroke

In conventional two-stroke, intake pipe is connected to the crankcase volume by a reed valve block which holds the reed petals and this is the one and only reed valve in the nowadays design. While in this prototype engine, there is another auxiliary reed valve at the pipe connecting the crankcase and cylinder. The fact is, crankcase pressure profile is highly dependent on the pressure wave upstream and downstream of it which both is highly related to the motion of reed valve petal. Thus, to analyze the crankcase using computational fluid dynamics solver required an accurate reed valve model. The reed valve model and proper setting of initial condition will permit the solver to calculate the conditions at the inlet and the outlet simultaneously. The previous research of reed valve model is presented in details elsewhere [1,2,3,4].

The KUKTEM two-stroke modular diesel engine is a conventional type of design which employs the crankcase compression strategy. The over square engine advantageous with lower friction due to shorter stroke length. But consequently, the crankcase compression ratio is only about 1.14, quite low compared with other nearly same displacement two-stroke engine. This study will try to evaluate the characteristics of the flow inside the engine's crankcase and investigate the effect of the crankcase compression ratio on the compression process. The novelty of the engine is the design of diagonal auxiliary ports. This innovation intends to improve the scavenging process by inducing additional fresh air inwards into combustion chamber instead of main intake port. The ports are controlled by secondary reed valve, positioned vertically at cylinder sleeve. The reed valve is operated mainly by pressure differences of upstream and downstream flow area.

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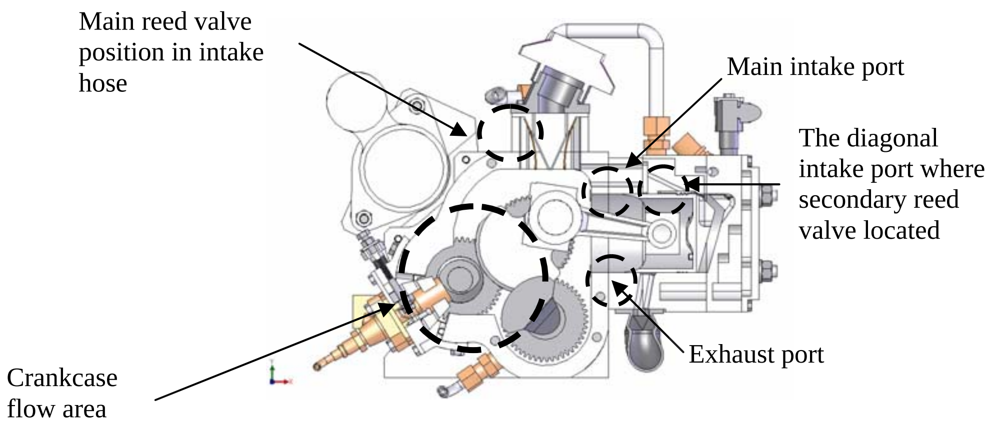

Figure 1: Cut view of the two stroke diesel.

NUMERICAL METHODOLOGY

Based on work by Cunningham and Kee [5], computational fluid dynamic model of the crankcase parts are developed. This model trying to include additional features of inner components compare to previous model of Cunningham and Kee such as the motion of reciprocating piston, the rotating crankshaft big end, and crankshaft balancer. The model excluded small components such as fuel pump and connecting rod which adding complexity and currently quiet impossible to solve simultaneously. To reduce computing requirement, the model are taken to be halve at the cylinder centerline. The model is meshed using unstructured element at the stationary region and structured element or hexagonal at the isotropy motion region. Both type of mesh are selected to fulfill the requirement of dynamic mesh and sliding mesh motion. The segregated computational fluid dynamic (CFD) solver with implicit formulation is used to solve the unsteady, compressible, 3-dimensional Navier-Stokes equation and energy conservation equation. Commercial CFD software is employed for its easy approach and user-friendliness.

These are the basic partial differential equation of Navier-Stokes written in longhand notation in terms of flowfield variables using Stoke's postulation in conservation form. The mass conservation equation

$$\frac{\partial \rho}{\partial t} + \nabla,(\rho V) = 0 \tag{1}$$

the x-momentum conservation equation

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Solid State Science and Technology, Vol. 15, No 2 (2007) 50-59
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$$\frac{\partial(\rho u)}{\partial t}+\frac{\partial(\rho u^2)}{\partial x}+\frac{\partial(\rho uv)}{\partial y}+\frac{\partial(\rho uw)}{\partial z}=\frac{\partial p}{\partial x}+\frac{\partial}{\partial x}\left(\lambda\nabla.V+2\mu\frac{\partial u}{\partial x}\right)$$
$$+\frac{\partial}{\partial y}\left[\mu\left(\frac{\partial v}{\partial x}+\frac{\partial u}{\partial y}\right)\right]+\frac{\partial}{\partial z}\left[\mu\left(\frac{\partial u}{\partial z}+\frac{\partial w}{\partial x}\right)\right]+\rho f_x \tag{2}$$

the y-momentum conservation equation

$$\frac{\partial(\rho v)}{\partial t}+\frac{\partial(\rho uv)}{\partial x}+\frac{\partial(\rho v^2)}{\partial y}+\frac{\partial(\rho vw)}{\partial z}=\frac{\partial p}{\partial y}+\frac{\partial}{\partial x}\left[\mu\left(\frac{\partial v}{\partial x}+\frac{\partial u}{\partial y}\right)\right]$$
$$+\frac{\partial}{\partial y}\left(\lambda\nabla.V+2\mu\frac{\partial v}{\partial y}\right)+\frac{\partial}{\partial z}\left[\mu\left(\frac{\partial u}{\partial y}+\frac{\partial v}{\partial z}\right)\right]+\rho f_y \tag{3}$$

the z-momentum conservation equation

$$\frac{\partial(\rho w)}{\partial t}+\frac{\partial(\rho uw)}{\partial x}+\frac{\partial(\rho vw)}{\partial y}+\frac{\partial(\rho w^2)}{\partial z}=\frac{\partial p}{\partial z}+\frac{\partial}{\partial x}\left[\mu\left(\frac{\partial u}{\partial z}+\frac{\partial w}{\partial x}\right)\right]$$
$$+\frac{\partial}{\partial y}\left[\mu\left(\frac{\partial w}{\partial y}+\frac{\partial v}{\partial z}\right)\right]+\frac{\partial}{\partial z}\left(\lambda\nabla.V+2\mu\frac{\partial w}{\partial z}\right)+\rho f_x \tag{4}$$

the energy conservation in terms of internal energy, e

$$\frac{\partial(\rho e)}{\partial t}+\nabla.(\rho eV)=\rho\dot{q}+\frac{\partial}{\partial x}\left(k\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right)+\frac{\partial}{\partial z}\left(k\frac{\partial T}{\partial z}\right)$$
$$-p\left(\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}\right)+\lambda\left(\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}\right)^2+ \tag{5}$$
$$\mu\left[2\left(\frac{\partial u}{\partial x}\right)^2+2\left(\frac{\partial v}{\partial y}\right)^2+2\left(\frac{\partial w}{\partial z}\right)^2+\left(\frac{\partial u}{\partial y}+\frac{\partial v}{\partial x}\right)^2+\left(\frac{\partial u}{\partial z}+\frac{\partial w}{\partial x}\right)^2+\left(\frac{\partial v}{\partial z}+\frac{\partial w}{\partial y}\right)^2\right]$$

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Table 1: Two Stroke Diesel engine specifications.

Parameter	Specification
Type	2-stroke, single cylinder, diesel
Bore x stroke	70 mm x 55 mm
Volume of Displacement	211 cc
Nominal Compression ratio	16.5
Rated engine speed	3000-3500 RPM
Maximum Output	2.8 kW
Mean piston speed	5.5 m/s
Mean Effective Pressure	443.2 kPa
Fuel consumption rate	280.3 g/kW.hr
Crankshaft Rotary direction	Clockwise from flywheel
Cooling type	water-cooled
Starting type	Electric starter motor
Fuel Injection Pressure	196 bar
Fuel Injection Timing	17.5 $\pm$ 0.5 btdc
Crankcase Compression ratio	1.14
Exhaust Port Opening/Closing	111^{o} ATDC, 249^{o} ATDC
Intake Port Opening/Closing	146^{o} ATDC, 214^{o} ATDC

Table 2: Numerical Model Attributes.

Num	Parameter	Attribute
1.	Material definition	Air, ideal gas density
2.	Operating condition	101325 bar, 300 K
3.	Boundary condition	Pressure inlet and pressure outlet
4.	Turbulence Model	Two equation K-ε model
5.	Turbulent specification	Intensity and hydraulic diameter

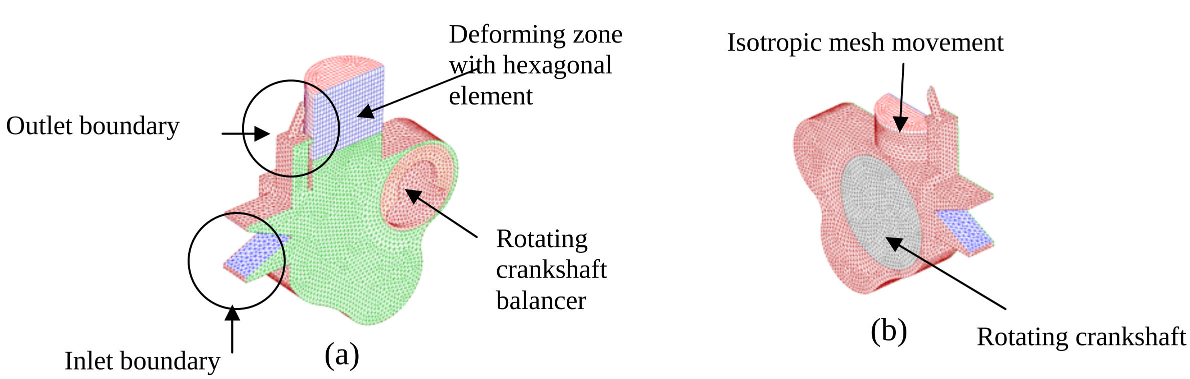

Figure 2(a) and 2(b): Meshed model using hexahedral and tetrahedral element and description of moving and stationary parts and boundary.

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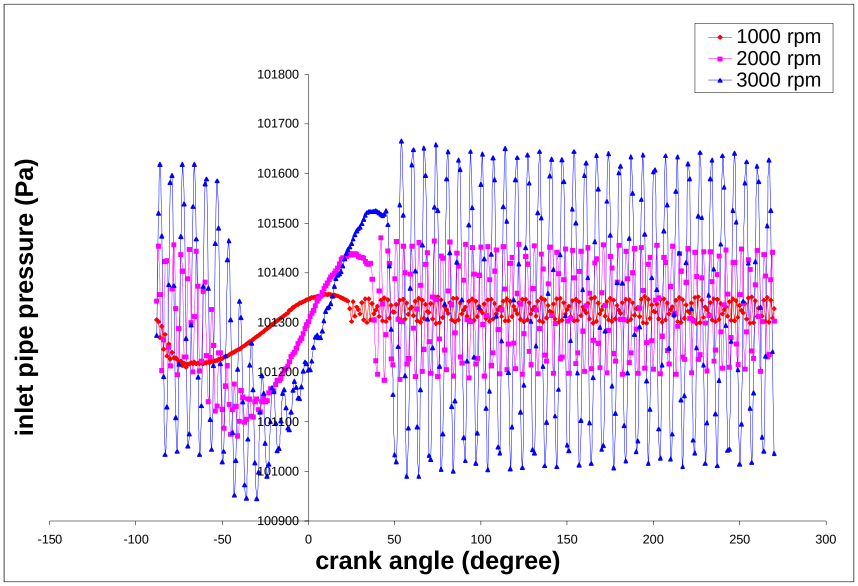

Figure 3: Pressure fluctuation inside the inlet tract prior to the reed valve is plotted against the crank angle for 1000, 2000 and 3000 rpm engine speed.

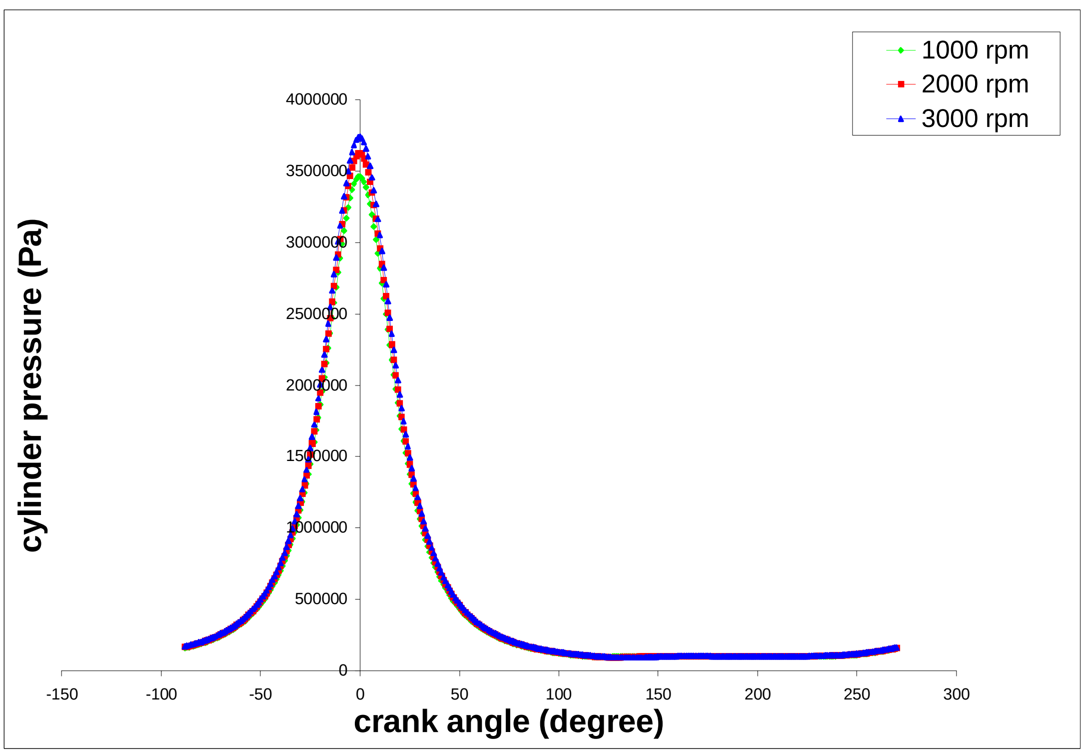

Figure 4: Cylinder pressure profile in motoring test is plotted against the crank angle for 1000, 2000, and 3000 rpm engine speed.

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The paper presented by Dave *et al*. [6] give a clear view on how an accurate reed valve model is developed. Since such model is beyond the capability of this commercial software, another approach is proposed. The reed valve motion is disregarded in this model but the effect of the reed valve existence is included by taking the instantaneous pressure fluctuation in the inlet tract from motoring test engine prior to the main reed valve as the input for the pressure inlet boundary. The pressure outlet boundary is based on the instantaneous measured cylinder pressure of the engine. It is expected that the pressure inlet and outlet boundary can give the effect as same as the reed petals to force or prevent the flow.

RESULTS AND DISCUSSION

In order to investigate the compression pressure profile, the computational model was run at engine speeds of 1000, 2000 and 3000 rpm at all valves closed condition. From the measured data, the maximum value lies in the range of 1.17491 bars at 150 CA at 1000 rpm - 1.16812 bar at 160 CA at 3000 rpm. The trend shows that the higher rpm gives a lower peak pressure due to a faster compression period. Based on the calculated pressure profile, the plot does show the same trend. The smooth line of pressure profile is produced compared to the measured value which encounters a gradual slight drop after 150 CA as the reed valves at the diagonal port are opened. The entrance of fresh charge into the cylinder through the ports will cause pressure drop inside the crankcase. At this stage, the CFD model unable to model the event of reed valve opening and closing and thus causing the pressure to keep rising until 180 CA. The velocity vector magnitude shows that the flow vector is totally prohibited by the rotation of crankshaft and crankshaft balancer and the moving downward of piston surface. Maximum velocity inside the crankcase achieved the value of 24 m/s due to the momentum transfer from the rotating crankshaft surface at 3000 rpm to the air. Based on the contour of static temperature, the temperature distribution is higher at the region with small surface area and lower in the region with large surface area. This is agreed with previous knowledge of thermodynamic and heat transfer where a large surface area will allow a higher rate of heat transfer and thus cause the region with higher surface area such as the bottom of the crankcase have a lower temperature.

The pressure distribution which is plotted based on the volume average static pressure did give a significant indication on the volumetric region with quasi-static air charge since this model did not include the rotation of fuel pump camshaft at the bottom of the crankcase as in actual operation. But for motoring purpose, no fuel pump camshaft is needed, so the results thus have predicted what was happening during the motoring operation.

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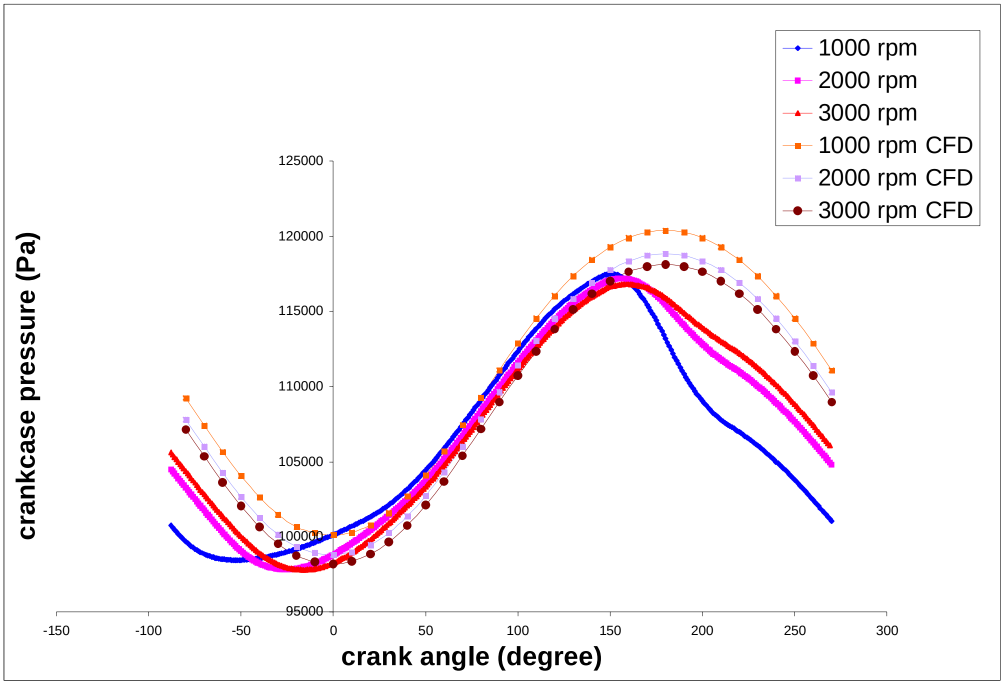

Figure 5: Crankcase pressure measured at 1000, 2000 and 3000 rpm and calculated crankcase pressure at same engine speed without pre-defined exhaust and intake pressure boundary (all valves closed).

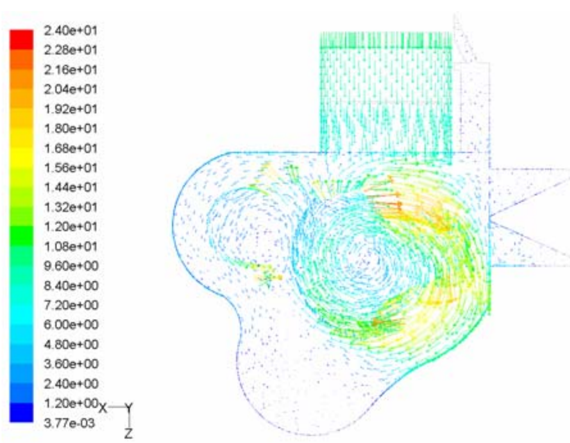

Figure 6: Vectors of velocity magnitude at 70 CA after to dead centre (ATDC) at 3000 rpm.

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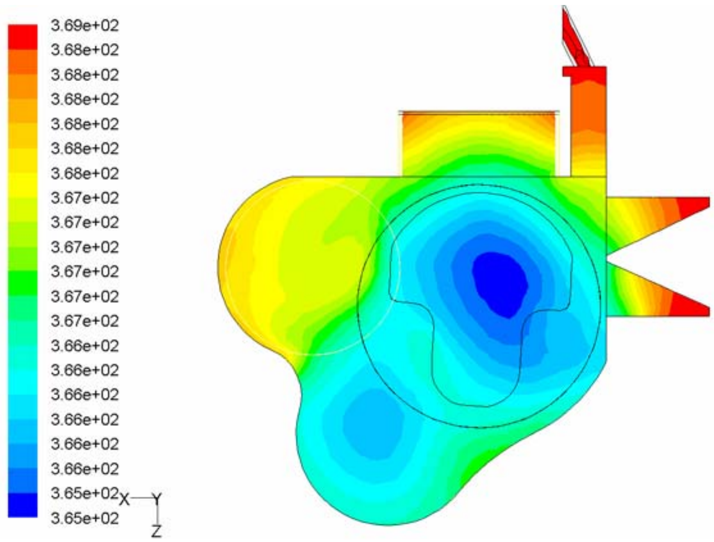

Figure 7: contours of static temperature at 180 CA ATDC at 3000 rpm.

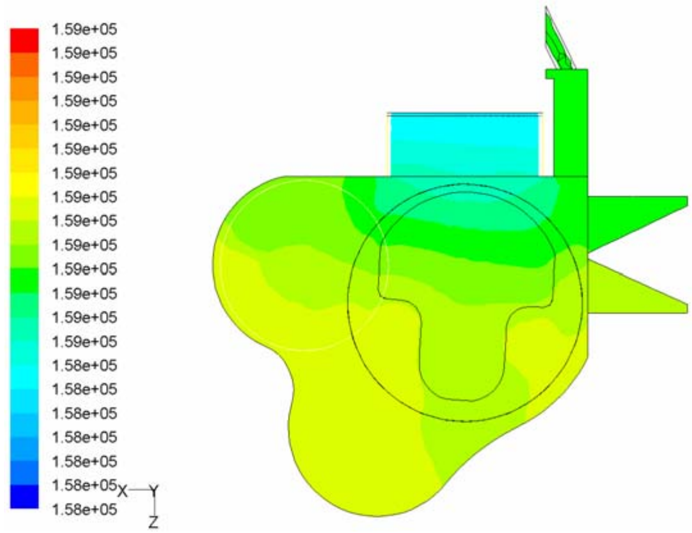

Figure 8: Contours of static pressure at 180 CA ATDC at 3000 rpm.

CONCLUSION

The most probable solution to the crankcase flow is to develop similar reed valve model such as presented by Anupam *et al*., Fleck *et al*. or Mitianec *et al*. Such models should be integrated with the current crankcase model. This paper presents the attempt taken to develop the crankcase model without the existence of that type of model. But the current result shows no significance improvements can be achieved. Major findings of this study told that the pressure profile of the crankcase compression process can be calculated and produce a reasonable value of maximum pressure inside crankcase. But the measured plot trend shows that the action of reed petal will cause the drop in crankcase pressure and that condition can only be approximate in numerical analysis by using proper boundary condition or reed valve model. But the study on the velocity vector and contour of pressure and temperature thus give a good prediction on how the flow inside the crankcase is developed.

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