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Thermal Effects on Mullite Coated Diesel Engine Piston for Various Coating Thickness

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Abstract: Diesel engines are prime role in medium and heavy duty applications due to grander characteristics such as lower fuel consumption, high engine power output and lower emissions as compared with gasoline functioned engines. In this research is to investigate the temperature distribution and the effects of thermal barrier coated in diesel engine piston as a function of various coating thickness. Multi coatings are used to increase the performance of high temperature components in diesel engines. The piston is modelled using PRO-E wildfire software. The thermal analysis are performed on the piston for various coating thickness by means of using commercial code namely ANSYS 13.0 version. The piston temperature distribution is calculated for both conventional and coated pistons in order to control the thermal stresses and deformations within acceptable levels. Based on the above thermal analysis the optimum coating thickness is suggested for diesel engines. The performance results of optimum thickness coated piston are compared with conventional and coated pistons.
Key words: thermal barrier, coating thickness, thermal stresses.

I. INTRODUCTION

The function of the piston is to absorb the energy released after the air fuel mixture is ignited by the high temperature. The piston accelerates producing useful mechanical energy. To accomplish this, the piston must be sealed so that it can compress the mixture of air and fuel does not allow gases out of combustion chamber. This can be accomplished by piston rings which also help to preventoil from entering the combustion chamber from underneath the piston. Another function of rings is to keep the piston from contacting the cylinder wall. Less contact area between the cylinder and piston reduces friction, thereby increasing efficiency. The concept of insulating cooled heat engine components such as diesel engine piston and valves from hot working fluids with a ceramic thermal barrier coating is very efficient one. A practical system, however, has only been identified in the past few years. Thermal barrier coating TBC) is applied to metallic components to reduce metal temperature, reduce life cycle cost, increase the environmental resistance and in some cases reduce noxious exhaust emissions.

Mullite based coatings are ceramic combustion chamber coatings originally developed for adiabatic or low heat rejection engines have been shown to reduce diesel emissions. Reported results indicate that in-cylinder mullite coatings are capable of reducing the carbonaceous fraction of diesel particulates without increasing Nox or other regulated emissions. Reductions in total PM emissions may be achieved by combining mullite coatings with diesel oxygen catalysts. In-cylinder coatings are most effective in reducing emissions from older technology engines of relatively low thermal low thermal efficiency.

II. THERMAL ANALYSIS OF DIESEL ENGINE PISTON

It is important to calculate the piston temperature distribution in order to control the thermal stresses and deformations within acceptable levels. The temperature distribution enables us to optimize the thermal aspects of the piston design at lower cost, before the first prototype is constructed. As much as 60% of the total engine mechanical power lost is generated by piston ring assembly.

The piston skirt surface slides on the cylinder bore. A lubricant film fills the clearance between the surfaces. The small values of the clearance increase the frictional losses and the high values increase the secondary motion of the piston. Most of the Internal Combustion (IC) engine pistons are made of an aluminium alloy which has a thermal expansion coefficient, 80% higher than the cylinder bore material made of cast iron. This leads to some differences between running and the design clearances. Therefore, analysis of the piston thermal behaviour is extremely crucial in designing more efficient engine. The thermal analysis of piston is important from different perspectives. First, the highest temperature of any point in piston must not exceed more than 66% of the melting point temperature of the alloy. This limit temperature for the current engine piston alloy is about 640 K. Temperature distribution leads to thermal deformations and thermal stresses. The piston thermal deformation has an important role in piston skirt design which has a potential to reduce friction and piston slap. In this design, both of the thermal and mechanical stresses must be

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considered indicating the importance of piston thermal analysis.

In the recent work, Li used finite element method to analyse the piston thermal behaviour. Because of symmetry, he only used a quarter of the piston. He applied the thermal boundary conditions of piston symmetrically. He used simple combustion model for combustion side boundary condition. His numerical results matched with experiment well. The piston was subjected to the coupled action of thermal and mechanical loads. The results would be used as source data for the development of a global elastic hydrodynamic model and was provided a good tool for piston design analysis. The piston is modelled with different coating thickness using PRO-E wild fire software. The thermal analysis are performed using commercial code namely ANSYS 13.0 version. The conventional and the coated piston are compared each other to outcome the optimum thickness for diesel engine applications.

III. ENGINE SPECIFICATIONS

Thermal analysis is carried out on Texvel engine piston.

Table 1: Texvel Engine specifications

Type	Single cylinder, vertical, four stroke cycles, water cooled diesel engine
Bore in mm	85
Stroke in mm	110
Speed in rpm	1500
Connecting rod length in mm	235
Compression ratio	18:1
Brake Horse power	6.5
Loading	Rope brake
Lubrication system	Forced type
Injection timing	19.58^0 BTDC

A. Calculations Of Heat Transfer Coefficients

The piston receives the heat from the hot gases formed by burning mixture of a particular air/fuel ratio , the boundary conditions around the piston body are different from region to region .In this work the calculations of the thermal analysis depends on the theories of the convection heat transfer analysis that could be applied to piston and piston rings.

B. Combustion Chamber Side Thermal Boundary Condition

The mathematical description of the forced fluid flow on a cylinder surface is so complicated whereas in the parts of an internal combustion engine especially the piston, the effect of the hot gases on it is very complicated, and in order to calculate the heat transfer coefficient at the piston crown surface, the heat transfer is described as a forced convection heat transfer inside a cylinder . The heat transfer from the combustion gases is assumed to be similar to the turbulent heat transfer of gases in a cylinder as follows:

$$Nu = C\ Re\ m\ Pr\ n \ldots.$$

Where

Nu is the Nusselt number,

Re is Reynolds number and

Pr is Prandtl number.

The m exponent is typically assumed to be 0.8 for fully developed turbulent flow and n = 0.3or 0.4 for the cooling or heating respectively .The constant C is to be found from the experimental studies. Benson mentioned that Gunter F.Hohenberg, presented a developed relationship for the equation by using the cylinder volume as a function of the piston diameter

$$h_g = 226.8\ P^{0.8}\ T^{-0.4}(Vp+1.4)0.8$$

Where,

h_g – convective heat transfer coefficient

P= indicated mean effective pressure acting on the piston in bar = 7.67bar

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T= bulk temperature = 656^0C (reference temperature)

Vp= mean velocity of the piston = 2xLxN=2x.11x1500=5.5m/s h_g=356.36 W/m^2K

C. Heat Transfer Coefficient Between Piston Crown And Liner

The ring land heat transfer model is based on the flow between the two, parallel plates. According to Reynolds number which is less than 2000, it could be assumed that the flow is laminar .To get the value of the heat transfer coefficient Nusselt number should be found for the laminar flow between two parallel plates. where this number is ,

$$Nu=hxD_h/k \qquad =8.235$$

So the heat transfer coefficient will be equal to ,hr =8.235k/Dh , Dh=4A/P

Where,

D_h = is the hydrulic diameter

A = is the cross-section area in (m^2) and is equal to; A = 2b*1 unit depth

P = the perimeter in (m) and is equal to; P = 2,

therefore the hydraulic diameter will be equal to; D_h= 4b.

b= clearance between piston and liner=0.09mm

K= thermal conductivity of gas = 0.184 W/mK

h= 1515.24W/m^2K

T= bulk temperature = 315^0C (reference temperature)

D. Heat Transfer Coefficint In The Rings

Thermal circuit method is used to model the heat transfer in the ring land and skirt region. with the following assumptions:

1) The effect of piston motion on the heat transfer is neglected;
2) The rings and skirt are fully engulfed in oil and there are no cavitations;
3) The rings do not twist;
4) The only heat transfer mode in the oil film is assumed to be conduction.

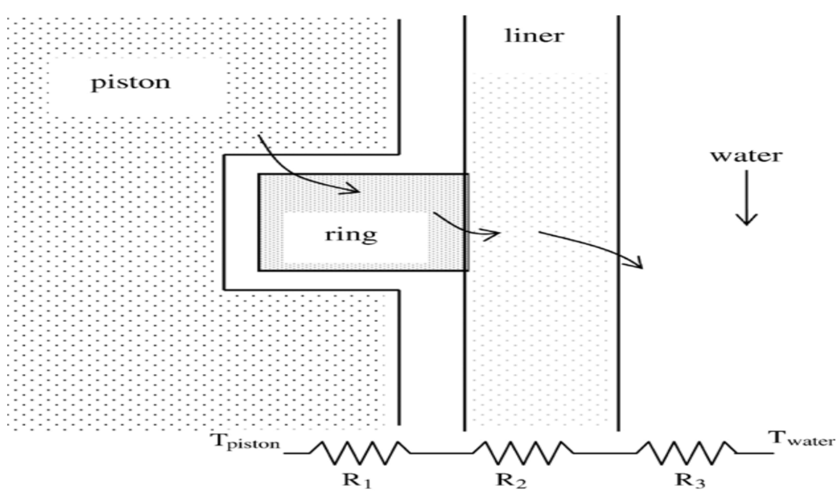
The resistances are:

$R_1=\ln(r_2/r_1)/(2x3.14xL_1xk_{ring})$ ring resistance

$R_2=\ln(r_3/r_2)/(2x3.14xL_2xk_{oil})$ oil film resistance

$R_3=\ln(r_4/r_3)/(2x3.14xL_3xk_{ring})$ block resistance

$R_4= 1/h_{water}.A_s$ water-jacket resistance



Thermal circuit resistance model for heat transfer from the rings

(R_1: ring resistance, R_2: block resistance, R_3: water-jacket resistance)

r_1=inner radius of the ring =41.08mm

r_2=Outer radius of the ring=44.45mm

r_3=Bore radius= 42.5mm

r_4=Inner radius of water jacket=45.5mm

L_1 and L_3 are the widths of the heat transfer paths 2.36 and 6mm respectively.

A_s= the effective area in contact with the coolant.

= 2x3.14xliner radiusxliner height

= $2x3x45.5x110x10^{-6}=0.30314314m^2$

When the above values is substituted in the equations we get $R_{1,}$ $R_{3,}$ and R_4 are 0.10230, 0.034811 and 0.02285 W/mK respectively.

Rtot=R1+R2+R3=0.159 w/mk

The effective heat transfer coefficient is obtained from

h_{eff}= 1/(R_{tot}xA_{eff}) A_{eff}=Piston surface in contact with ring.

= 2x3.14xtx (r_2-r_1) n

=$2x3.14x8x10^{-3}$x (41.08-44.45) $x10^{-3}$

=$6.77x10^{-4}m^3$

h_{eff}= 1/(R_{tot}xA_{eff})=9046.56 w/m2k

T= bulk temperature = 160^0C(reference temperature)

E. Heat Transfer Coefficient Of Piston Crown Underside

The crown underside is cooled by splash cooling type. The value of convective heat transfer coefficient is calculated from the following equation.

$$h= 900x (N/4600)^{0.35}$$

h=convection heat transfer coefficient

N=speed of the engine = 1500rpm.

h= 608 w/m^2k

T=bulk temperature =100^0C

F. Heat Transfer Coefficient Of Piston Skirt Underside

The skirt underside, the heat transfer coefficient value is calculated from the equation

$$h= 240x(N/4600)^{0.35}$$

h=convection heat transfer coefficient

N=speed of the engine = 1500rpm.

h= 162.135 w/m^2k

T=bulk temperature =100^0C

Table: 2 the boundary conditions of the piston

Region	Temperature (^{0}C)	Heat transfer coefficient
Combustion chamber	656	356.36
Between piston crown and liner	315	1515.24
Rings	160	9046.56
Crown underside	100	608
Skirt underside	100	162.35
Skirt outside	85	60

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IV. GEOMETRIC MODELLING

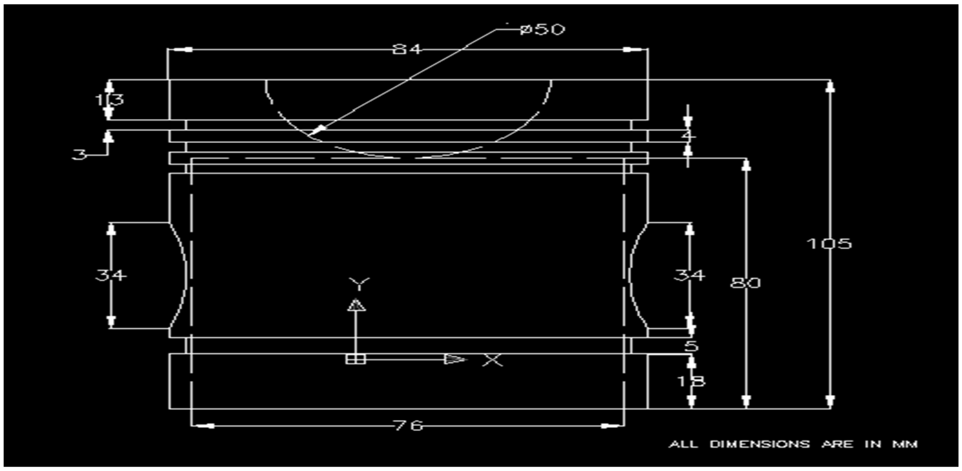

Figure 2: Dimensions of the Texvel engine piston

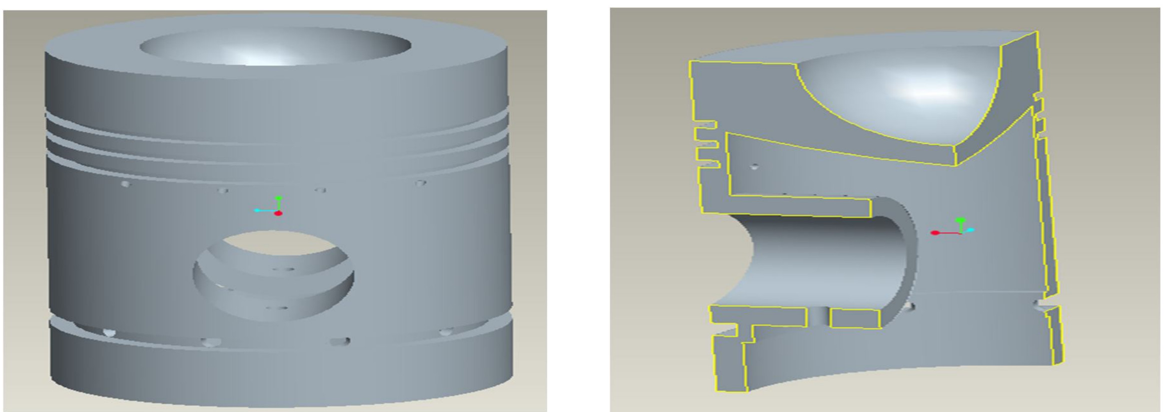

Figure 4: Texvel Engine Piston

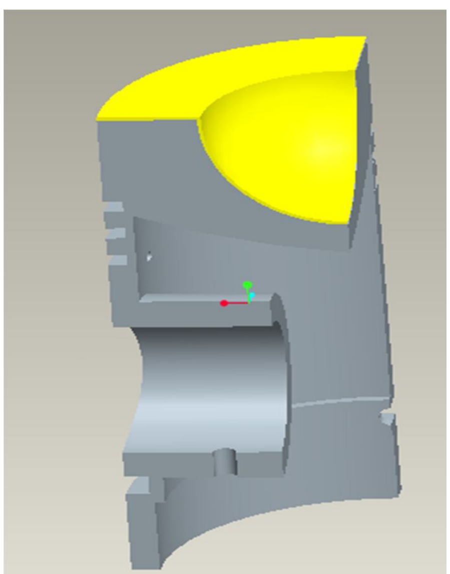

Figure 5: Piston with a coating Thickness of 0.65mm NiCrAl

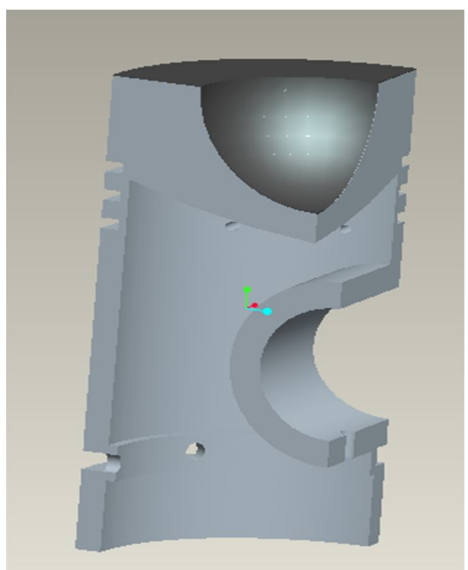

Figure 6: Piston with a coating thickness of 0.05mm Mullite + 0.60mm NiCrAl

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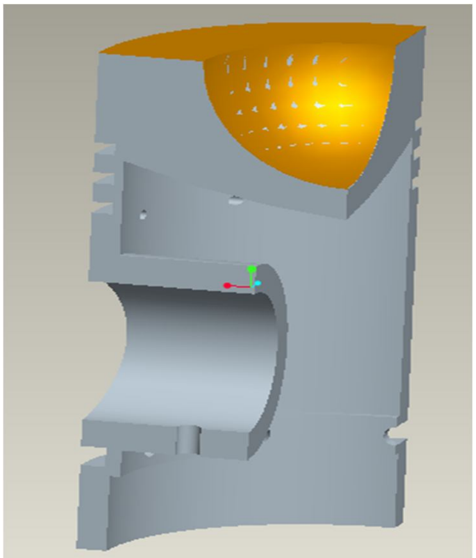

Figure 7: Piston with a coating thickness of 0.10mm Mullite +0.55mm NiCrAl

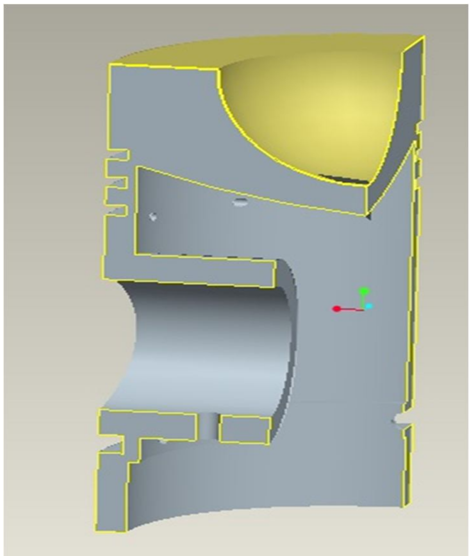

Figure 8: Piston with a coating thickness of 0.15mm Mullite +0.50mm NiCrAl

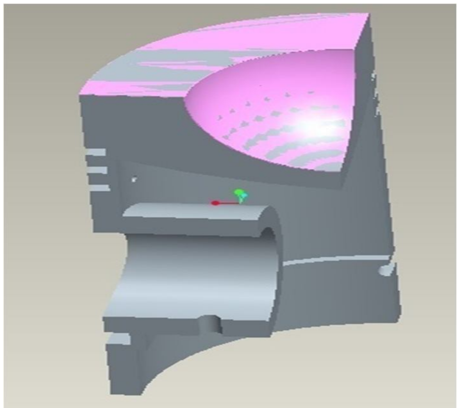

Figure 9: Piston with a coating thickness of 0.20mm Mullite +0.45mm NiCrAl

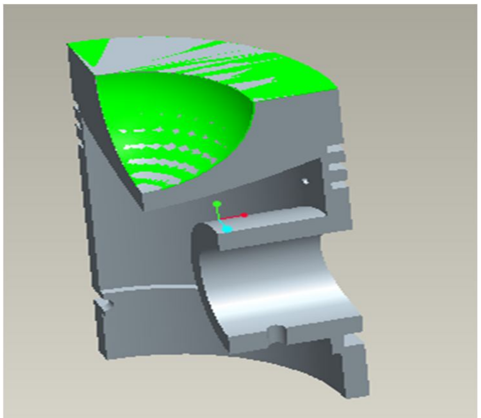

Figure 10: Piston with a coating thickness of 0.25mm Mullite +0.40mm NiCrAl

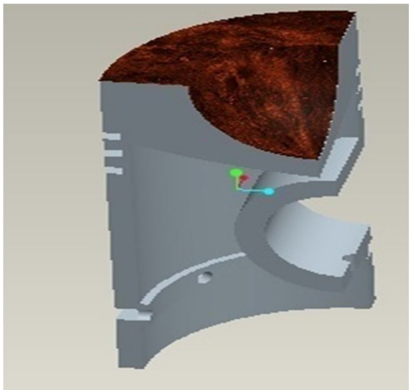

Figure 11: Piston with a coating thickness of 0.30mm Mullite +0.35mm NiCrAl

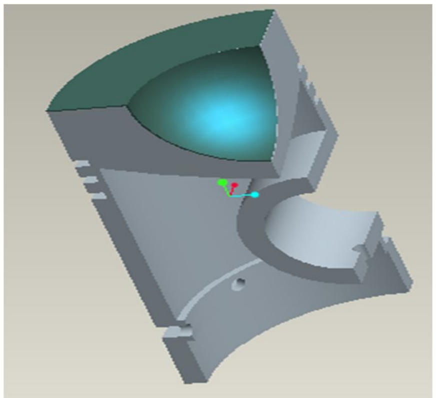

Figure 12: Piston with a coating thickness of 0.35mm Mullite +0.30mm NiCrAl

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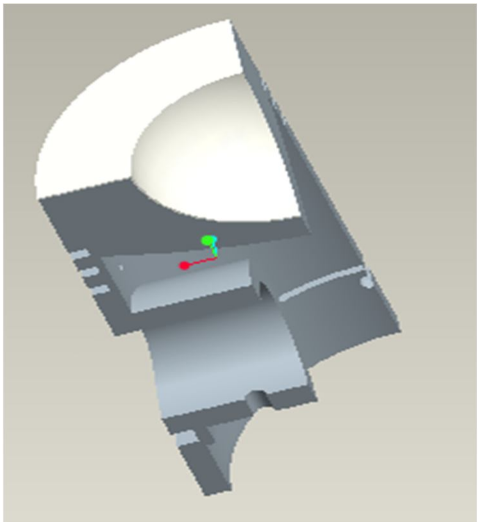

Figure 13: Piston with a coating thickness of 0.40mm Mullite +0.25mm NiCrAl

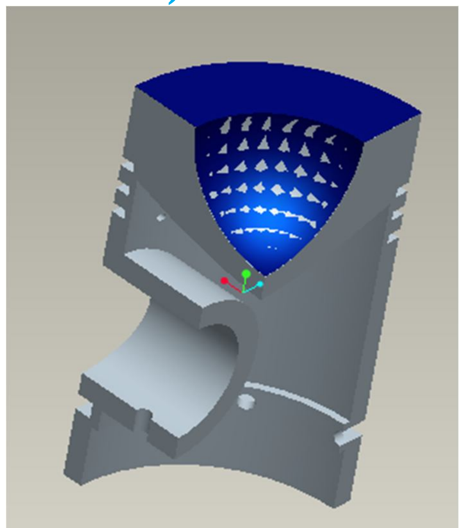

Figure 14: Piston with a coating thickness of 0.45mm Mullite +0.20mm NiCrAl

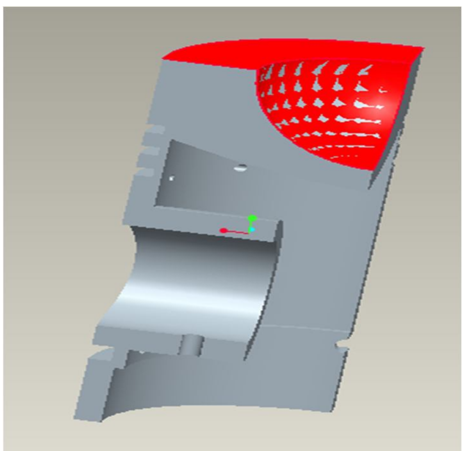

Figure 15: Piston with a coating thickness of 0.50mm Mullite +0.15mm NiCrAl

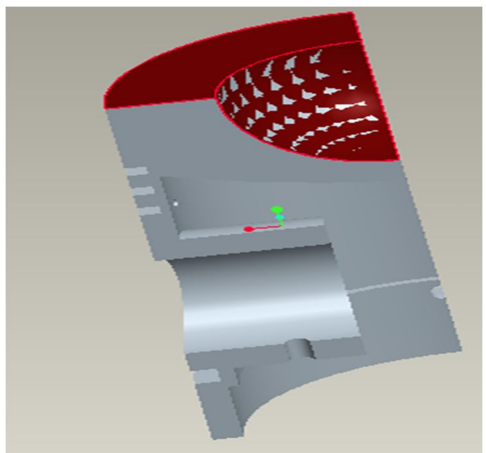

Figure 16: Piston with a coating thickness of 0.55mm Mullite +0.10mm NiCrAl

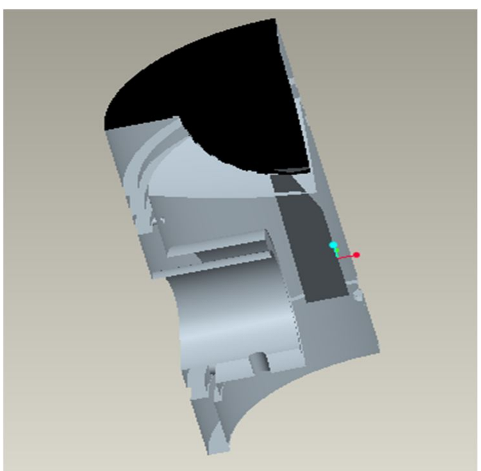

Figure 17: Piston with a coating thickness of 0.60mm Mullite +0.05mm NiCrAl

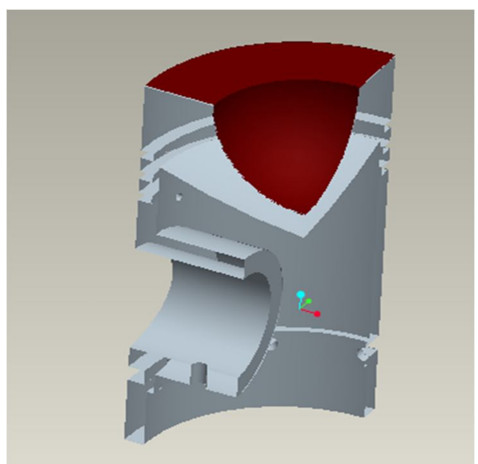

Figure 18: Piston with a coating thickness of 0.65mm Mullite

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V. RESULTS AND DISCUSSIONS

The temperature distribution along the conventional and coated piston is seen for various coating thickness involved. The temperature results are obtained and compared individually for all the coating thickness. The temperature distributions for conventional, coated and 0.55mm mullite coated piston are shown in figures.

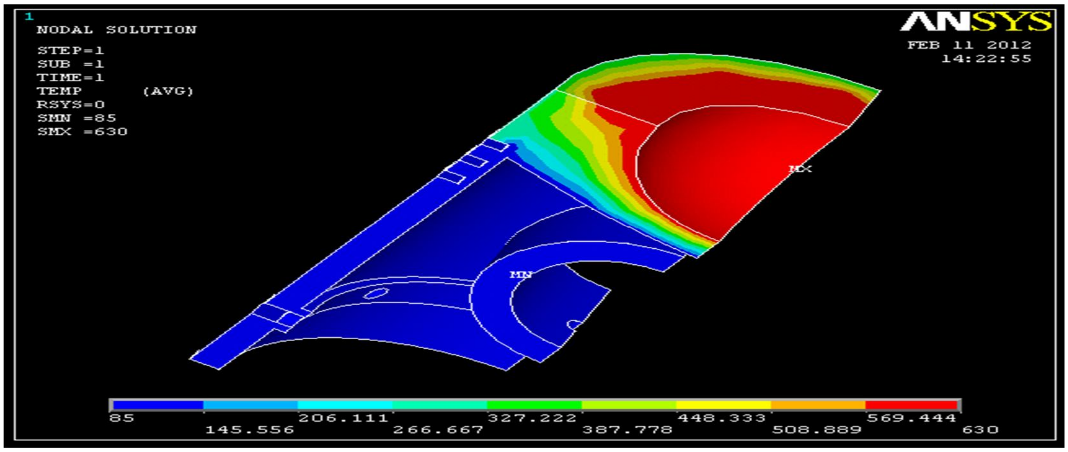

Figure 19: Temperature distribution for conventional piston

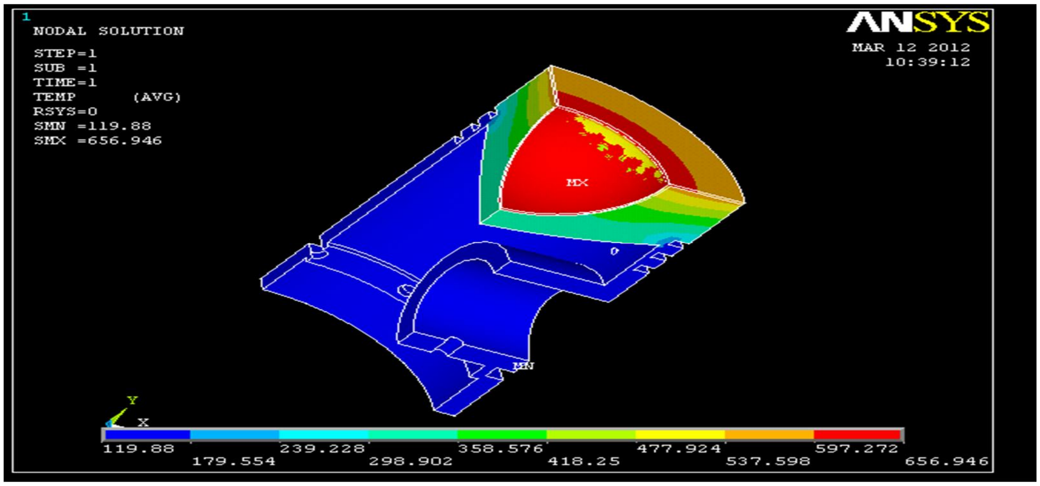

Figure 20: Temperature distribution for coated piston

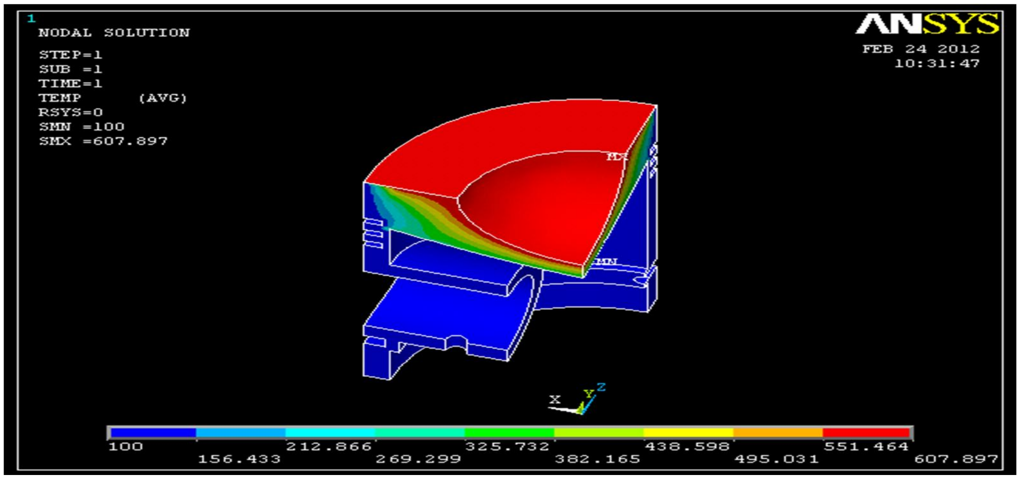

Figure 21: Temperature distribution for 0.55mm mullite coated piston

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Table 3: Performance tabulation for standard 19.58 degree injection timing

LOAD	TIME	B.P	TFC	SFC	BTE	FP	IP	BMEP	IMEP	MEE	ITE
Kg	s	KW	Kg/hr	Kg/KW hr	%	KW	KW	bar	bar	%	%
1	138	0.2	0.55	2.706866	3.068	1.8	2	0.265	2.63	10.09	30.31
5	128	1.01	0.59	0.583668	14.23	1.8	2.81	1.326	3.69	35.93	39.46
10	127	2.02	0.59	0.294132	28.23	1.8	3.82	2.651	5.01	52.87	53.22
15	102	3.03	0.74	0.244149	34.01	1.8	4.83	3.977	6.34	62.72	54.04
20	86	4.04	0.88	0.217179	38.24	1.8	5.84	5.302	7.67	69.17	55.09

Table 4: Performance tabulation for coated engine 19.58 degree injection timing

LOAD	TIME	B.P	TFC	SFC	BTE	FP	IP	BMEP	IMEP	MEE	ITE
Kg	S	KW	Kg/hr	Kg/KW hr	%	KW	KW	bar	bar	%	%
1	143	0.2	0.53	2.612221	3.179	2.5	2.7	0.265	3.55	7.473	42.4
5	133	1.01	0.57	0.561726	14.78	2.5	3.51	1.326	4.61	28.76	51.22
10	130	2.02	0.58	0.287344	28.9	2.5	4.52	2.651	5.93	44.68	64.46
15	125	3.03	0.60	0.19925	49.68	2.5	5.52	3.977	7.25	54.78	75.83
20	91	4.04	0.83	0.205246	40.46	2.5	6.54	5.302	8.58	61.76	65.29

Table 5: Performance tabulation for 0.55mm mullite coated engine 19.58 degree injection timing

LOAD	TIME	B.P	TFC	SFC	BTE	FP	IP	BMEP	IMEP	MEE	ITE
Kg	s	KW	Kg/hr	Kg/KW hr	%	KW	KW	bar	bar	%	%
1	158	0.2	0.477	2.386	3.48	2.3	2.5	0.265	3.21	8.60	43.50
5	149	1.01	0.506	0.501	16.36	2.3	3.32	1.326	4.24	30.51	54.29
10	142	2.02	0.531	0.262	31.57	2.3	4.32	2.651	5.54	46.75	67.52
15	139	3.03	0.542	0.179	46.40	2.3	5.33	3.977	6.83	56.84	81.62
20	119	4.04	0.634	0.156	52.88	2.3	6.34	5.302	8.13	63.72	83.26

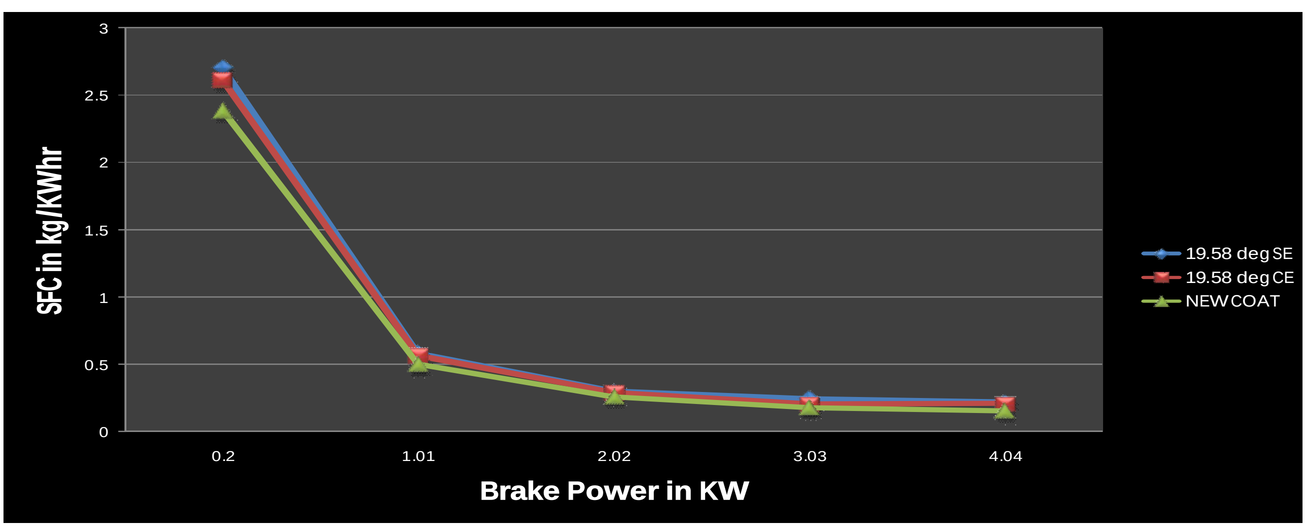

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Figure 22: Brake power Vs Specific fuel consumption.

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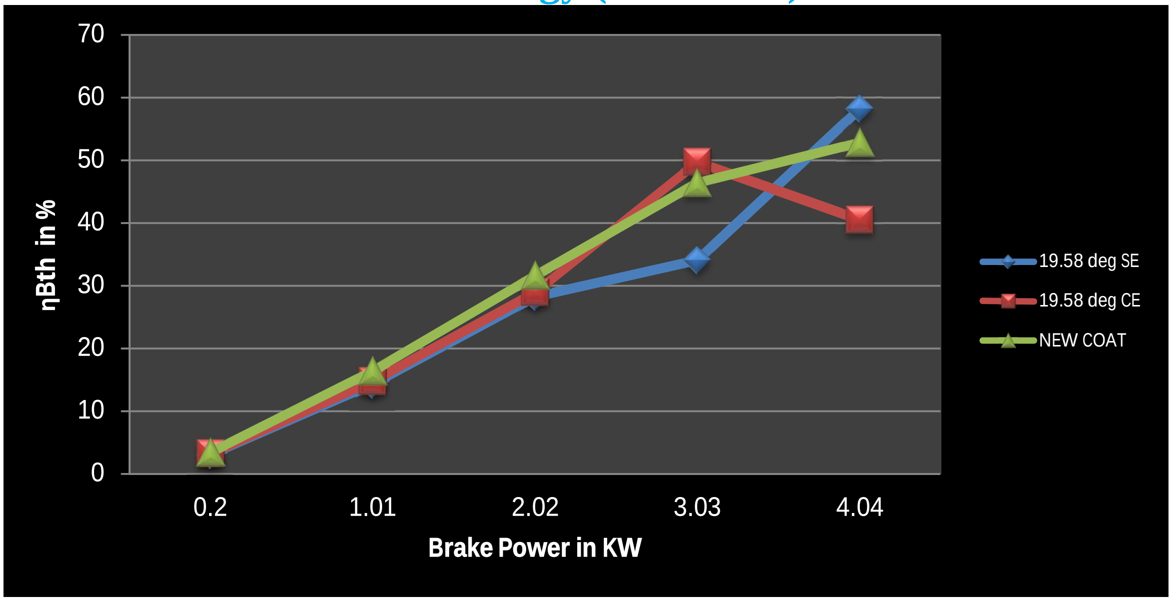

Figure 23: Brake power Vs Brake thermal efficiency.

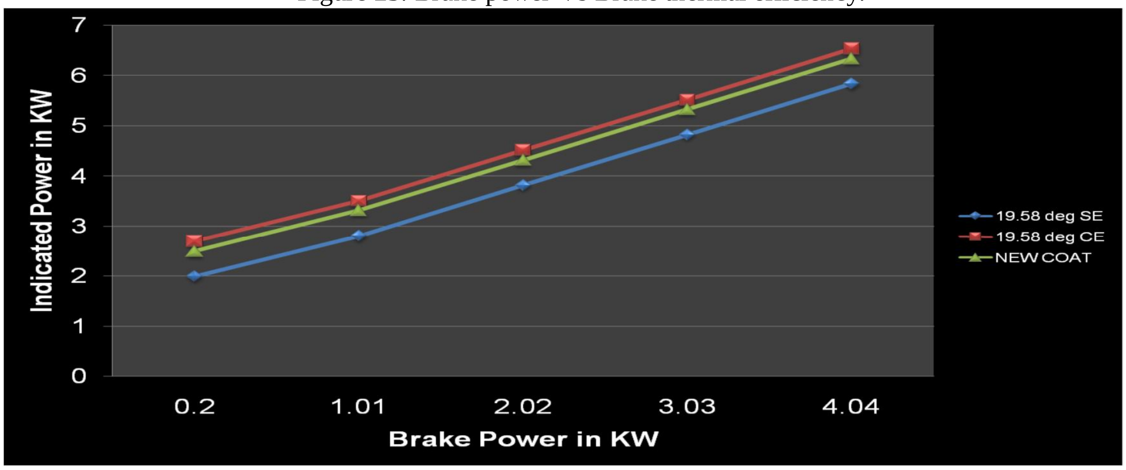

Figure 24: Brake power Vs Indicated power.

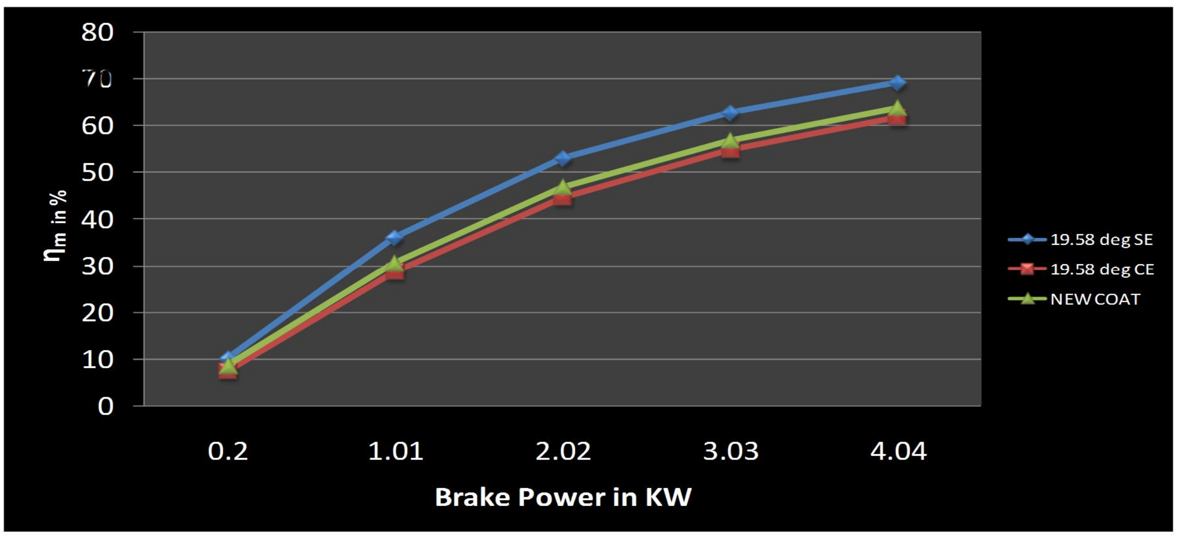

Figure 25: Brake power Vs Indicated thermal efficiency.

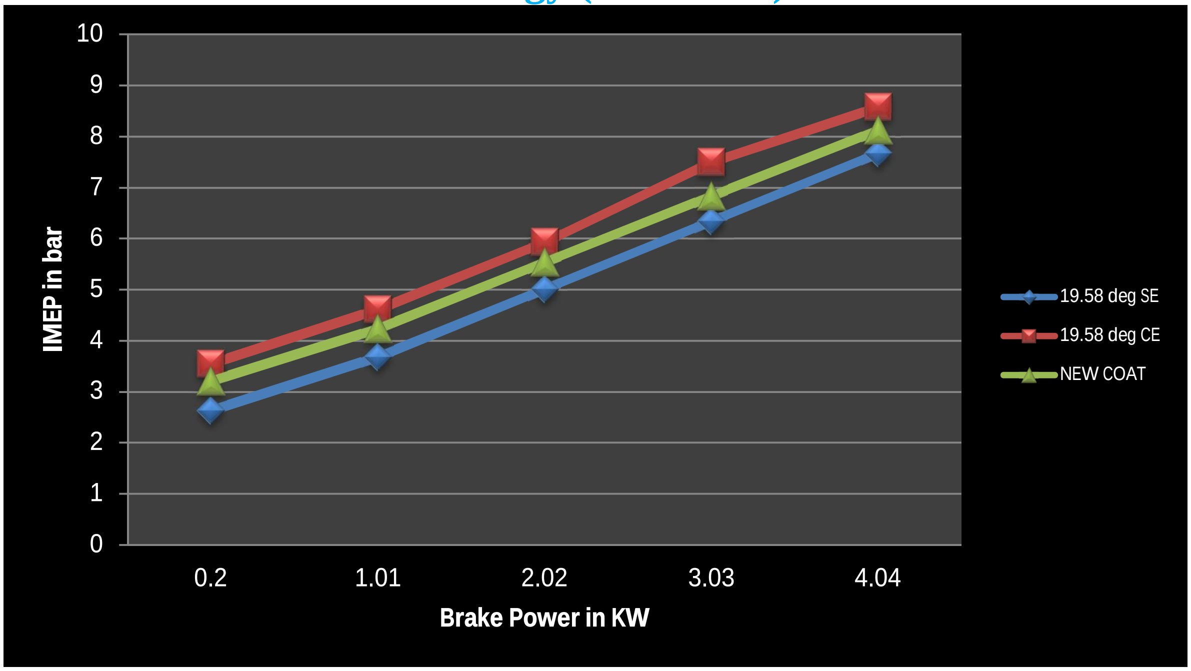

Figure 26: Brake power Vs Indicated mean effective pressure.

VI. CONCLUSION

The thermal analysis has been performed on the conventional and coated with optimum thickness of the pistons. The temperature distributions are predicted by using ANSYS software package. The results of the analysis reveal that, the combustion chamber temperature of the conventional piston is 630degree Celsius. The combustion chamber temperature values are increased from 650.579 degree Celsius to 668.786 at some intervals for 0.05mm to 0.50mm coating thickness. This is due to the very small layer of the coating thickness. The temperature of the combustion chamber in 0.55mm mullite coated piston is decreased to 607 degree Celsius. The temperature contours are also evenly distributed in 0.55mm coated piston. Then the combustion chamber temperature values are gradually increased for other coating thickness greater than 0.55mm. From the above analysis, it is concluded that 0.55mm coating thickness is the optimum coating thickness for diesel engine applications. The temperature and the stress values are reduced then the cooling load is also diminished for 0.55mm coating thickness. Thermal efficiency of the optimum thickness coated piston is 8% increase compared to the conventional and coated pistons and Specific fuel consumption is 4.5% decreased compared to the coated engine.

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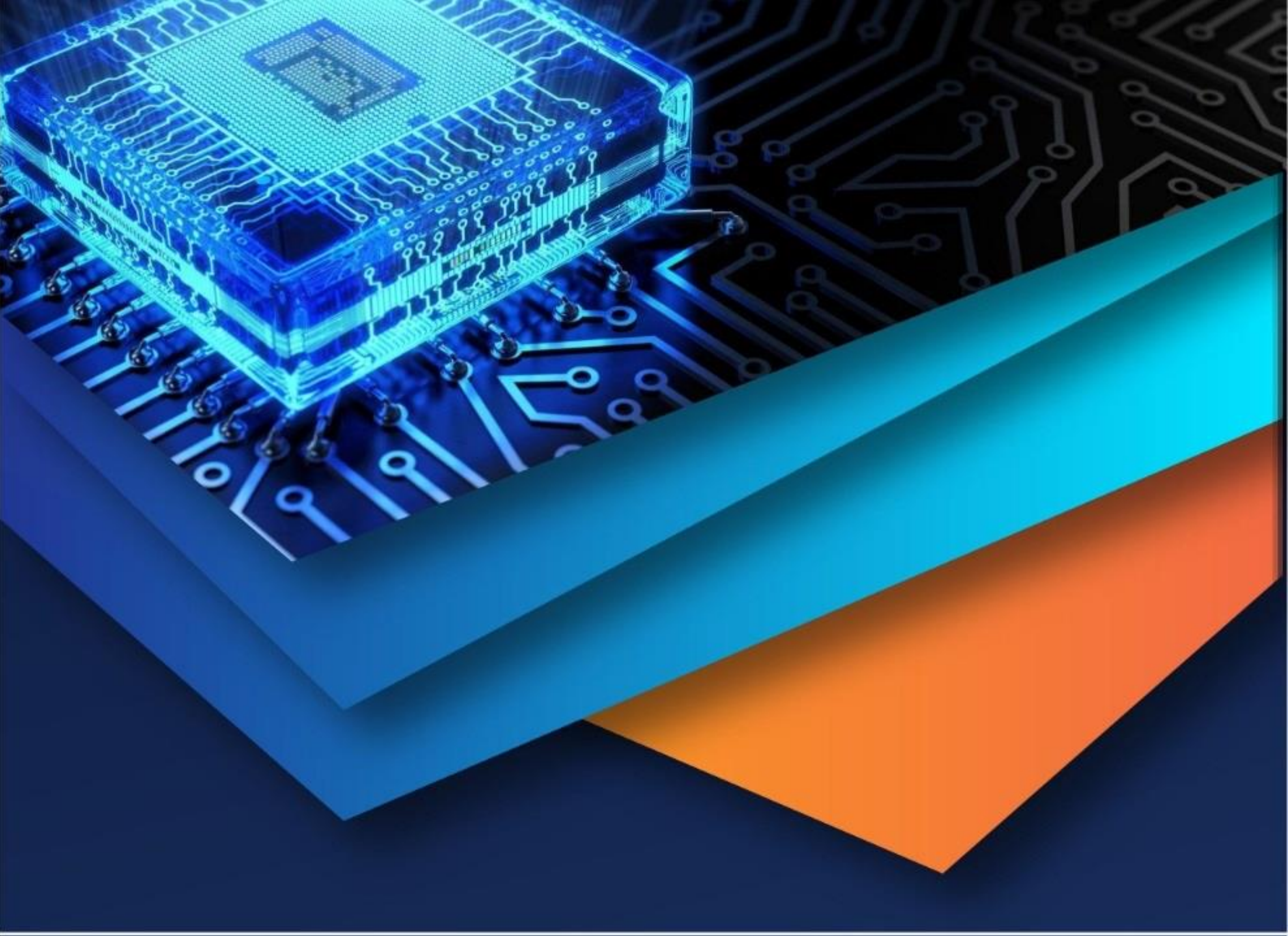

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