Application of Reinforced Periwinkle Shell for Nissan Bus Piston

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Abstract- ***This study investigated the feasibility of using reinforced periwinkle shell as a novel material for Nissan bus pistons. The project involves the design, finite element analyses of two pistons: one made from reinforced periwinkle shell and the other from standard piston material. The results show that the reinforced periwinkle shell piston exhibits exceptional thermal performance, with a minimum heat flux of -1.5844e7 W/m² and a maximum heat flux of 1.5829e7 W/m². In contrast, the standard piston material exhibits a minimum heat flux of -2.4904e7 W/m² and a maximum heat flux of 2.4904e7 W/m². The reinforced periwinkle shell piston also demonstrated improved durability and lifespan, with a thermal conductivity of 15 W/mK. The findings of this study highlight the potential of reinforced periwinkle shell as a sustainable and high-performance material for automotive applications, offering improved engine efficiency, reduced emissions, and enhanced overall performance.***

Indexed Terms- ***Reinforced Periwinkle Shell, Nissan Bus Piston, Thermal Performance, Durability, Sustainability.***

I. INTRODUCTION

A piston is a moving part that is enclosed by a cylinder and has piston rings to keep it gas tight. Its function in an engine is to move energy from the expanding gas in the cylinder to the crankshaft through a connecting rod or piston rod. A unique aluminum alloy and a few additional metal components make up pistons. After undergoing various stages and fulfilling international requirements, they transform into complex items. Manufacturers must create pistons with lower densities, improved fuel and oil efficiency, and less air pollution; as a result, these pistons must be lightweight and highly solid. A car's engine is sometimes referred to as its "heart," and its piston is arguably its most crucial component. The piston, a crucial component of the engine, withstands the forces of inertia and cyclic gas pressure. The piston may sustain fatigue damage as a result of this operating situation. According to investigations, the upper portion of the piston experiences the most stress, and fatigue failure is primarily caused by stress concentrations (ABBA-AJI and Mala Ali, 2021).

Early investigations on periwinkle shells in automotive settings began in the 1990s, with an emphasis on how they might be used in automotive composites (Aimikhe 2021; Nkwoada 2021). Periwinkle shell-based composites have since been created for a range of automotive uses, such as brake pads, engine mounts, and body panels, thanks to advancements in research (Lekia 2023; Sadiq 2022; Raheem 2019).

Figure 2: Periwinkle Shells.

1.2 Aim and Objectives

The aim of this study is to apply reinforced periwinkle shells with aluminum alloy as a sustainable material for Nissan Bus Piston and Investigate the thermal integrity of the piston taking into consideration the thermal properties of the produced reinforced periwinkle shell Composite (RPSCs) and compared to standard piston using ANSYS.

II. DESIGN ANALYSES AND CALCULATION OF PISTON PARAMETERS

The Engine used for this work is a 4 cylinder four stroke NISSAN model, 12V petrol engine. The engine specifications are given in Table 2.1.

Table 2.1 Engine Specifications

Parameters	Symbols	Value
Engine type		Four stroke, petrol engine
Number of cylinder		4-cylinders
Bore diameter	D	86mm
Piston stroke	↕	86mm
Displacement volume	X	2389cc
Compression ratio	CR	9.5:1
Maximum power	P	80kW
Maximum torque	T	300Nm
Number of revolutions/cycle	N	4400rpm

Source: NISSAN Motors

2.1 Measurement of the Piston's Sealing Part Thickness

According to Prasad et al. (2016), the sealing part's thickness (s) is as follows:

$s=0.05*D$ 2.1

$s=0.05*86=4.3mm$

Consequently, $s=0.05*86=4.3mm$

2.2 Piston Crown Radius Estimation

According to Dilipkumar and Madhura (2015), the crown inner radius (ri) is calculated as

$follows: ri = {}^{D}/_{2} - (s + t + dt).$ 2.2

Thus, according to Table 2.1, D is the bore diameter, which is 86 mm.

The radial thickness (t) and radial clearance (dt) for this diameter are 3 and 0.08 mm,

respectively. R1 = 86⁄2 - (4.3 + 3 + 0.08) = 35.62mm (Venkata et al., 2013).

2.3 Calculating the Stress on the Piston Crown

In accordance with Jadhav et al. (2015), the stress (σb) acting on the piston crown is as follows:

$Mb/Wb = Pzmax\ (ri\ /\delta) = \sigma b\ 2$ 2.3

$where\ \delta =$

$thiccces\ of\ the\ pis\ cs\ cs\ ys\ yhs\ pssyor\ cven\ as: \delta = dt \times D = 0.08 \times 86 = 6.88 \approx 7mm\ pzmax = 2MPa$

$\sigma b\ is\ equal\ to\ 2(35.62/7)/\ 2 = 20.35\ MPa.$

2.4 Calculating the Piston's Angular Velocity

According to Venkata et al. (2013), the piston^' s angular velocity (ω) is as follows:

$\omega = 2\pi \times \frac{rpm}{60}$ 2.4

Table 2.1 provides the engine rpm as 4400 rpm t = /4400 min or t = 0.000227 sec.

$consequently,\ \omega = 2 \times 3.142 \times 4400\ /\ 60 = 460.806\ rad/sec \approx 460.8\ rad/sec$

2.5 Calculating the Piston's Linear Velocity

According to Khurmi and Gupta (2004), the linear velocity is as follows:

$Angular\ velocity \times radius$ 2.5

= 460.8 × 36 = linear velocity Velocity linear = 16588.8 mm/s

2.6 Determination of the piston head thickness

The piston head thickness (tH) is calculated using Grashoff's formula, according to Sandeep and Vishnu, (2005) as:

$$tH = \sqrt{\frac{((3pD^2)}{(16\sigma t))}}\ mm \qquad 2.6$$

Where p = maximum pressure in N/mm D = cylinder bore/outside diameter of the piston in mm.
σt = permissible tensile stress for the material of the piston.

Permissible tensile stress for the material of the piston: Here the material is a particular grade of AL-Si alloy whose permissible stress is 50 Mpa-90MPa. However the permissible stress of 90MPa was used for a worse case scenario. Before calculating thickness of piston head, the diameter of the piston has to be specified.

The piston size that has been considered has an L×D, specified as 73.5×86, where L is the length of the piston, Factor of safety (F.O.S) According to Khurmi and Gupta, (2004) is given as 1.2.

Therefore;

$$\sigma t\ =\ \sigma p\ /F.O.S\ \ =\ 90/\ 1.2\ /=\ 75MPa \qquad 2.7$$

The maximum pressure experienced by the piston, according to Venkata et al. (2013) is between 1.4 and 1.8MPa, however, the pressure of 1.4MPa was used. Using equation 2.6:

$$tH = \sqrt{\frac{(3*1.4*86^2)}{(16*75)}} = 5.1mm$$

$tH \approx 5mm.$

2.7. Piston Ring Analysis:

According to Budynas and Nisbett (2014), the piston' s radial breadth (b) is as follows: D √(3pw/σt) = b 2.8 Let b = (t1) be the ring^' s radial thickness.

$$Let\ b\ =\ (t1)\ be\ the\ ring's\ radial\ thickness.$$

$$\frac{where\ Pw\ is\ the\ fuel\ pressure\ on\ the\ cylinder\ wall\ in\ N}{mm^2\ and\ D\ is\ the\ cylinder\ bore\ in\ mm}$$

$$.In\ accordance\ with\ Venkata\ et\ al.\ (2013),\ its\ value$$

$$\frac{\frac{is\ restricted\ to\ 0.025N}{mm^2\ to\ 0.042N}}{mm^2}.\sigma\ t\ for\ the\ current\ material\ is\ 90Mpa\ t1\ =$$

$$72\ \sqrt{((3*0.042)/90)}\ =\ 2.69mm.$$

2.8 Calculating the Piston Rings' Minimum Axial Thi ckness

As per Venkata et al. (2013), the minimum axial thic kness is as follows:

t2 = D/ (10 × nr) 2.9

in which nr is the number of rings
86/ (10 × 3) = 2.87 mm is t2.
According to Prasad et al. (2016),
The rings' thickness is as follows:

t2 = 0.7 x 2.69 = 1.883 mm (0.7b to b) 3.0

Top Land and Ring Land Width:

The top land's breadth ranges from b1 = tH to 1.2 tH. Top land width is b = tH to 1.2tH = 1.2 × 4.2 = 5.04 mm.

Other lands' width (b2)
The width of additional ring lands ranges from 0.75 × 2.87 = 2.15 mm for b2 = 0.75t2 to t2 b2 = t2.

2.9 Piston Barrel (skirt) Analysis

The maximum piston barrel thickness (t3) is as follo ws, per Jadhav et al. (2016):
0.03 × D + b + 4.5mm = t33.1 where b is the piston's ring groove's radial depth (or piston's radial width).
The top end piston barrel thickness (t3) is determined by multiplying 0.03 × 86 by 2.69 + 4.5 = 9.77 mm.
Sandeep and Vishnu (2015) state that the following f ormula is used to determine the piston barrel thickness and skirt length: Piston barrel thickness at the open end:

$$t2\ =\ (0.25t3\ to\ 0.35t3)0.35\ \times\ 9.77\ =\ 3.42\ mm\ =\ 3.2\ t2$$

$$The\ formula\ for\ the\ skirt's\ length\ (Is)\ is\ ls\ =\ (0.6\ D\ to\ 0.8\ D).$$

$$=\ 0.8\ x\ 86\ =\ 68.8\ mm\ =\ 3.3 =\ ls$$

III. CALCULATING THE PISTON PIN'S LENG TH IN THE CONNECTING ROD BUSHING

According to Manish et al. (2016), the length of the piston pin

$$(l1)\ in\ the\ connecting\ rod\ bushing\ is\ determined\ as\ follows:$$

$$l1\ =\ 45\%\ of\ the\ piston\ diameter 0.45\ x\ 86\ =\ 38.7\ mm\ =\ 3.4\ l1$$

3.0 Calculating the diameter of the piston pin

$$Manish\ et\ al.\ (2016)\ state\ that\ the\ piston\ pin\ diameter\ is\ as\ foll$$

$do = (0.28\,D\ to\ 0.38\,D).0.38\,x\,86 = 32.68\,mm = 3.5 =$

IV. RESULTS AND DISCUSSION

4.1 Results

Table 4.1: Thermal Analysis Results of the Standard Piston

S/N	Material Properties	Values	Units
1	Thermal Conductivity	130	W/mK
2	Specific Heat Capacity	714.8	J/kgK
3	Melting Temperature	660	^{O}C
4	Sample Weight	20.392	mg

Table 4.2: Imposed Boundary Condition

S/N	Material Properties	Values	Unit
1	Convection coefficient	230	W/m^2
2	Lateral surface temperature	350	^{O}C
3	Coolant temperature	60	^{O}C

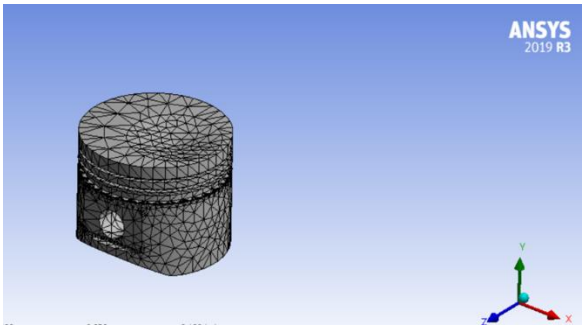

Figure 4.1: Finite Element Mesh of the Piston

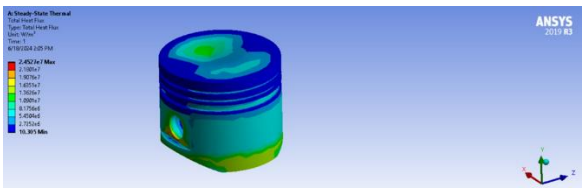

Figure 4.2: Total Heat Flux of the Thermal Analysis of the Standard Piston

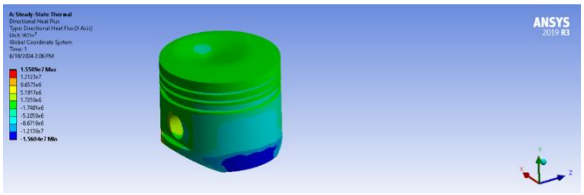

Figure 4.3: Directional Heat Flux of the Thermal Analysis of the Standard Piston

4.2 Thermal Analysis Results of the Local Piston

S/N	Material Properties	Values	Units
1	Thermal Conductivity	15	W/mK
2	Specific Heat Capacity	5	J/kgK
3	Melting Temperature	700	^{O}C
4	Sample Weight	22.50	mg

Imposed Boundary Condition

S/N	Material Properties	Values	Unit
1	Convection coefficient	230	W/m^2
2	Lateral surface temperature	350	^{O}C
3	Coolant temperature	60	^{O}C

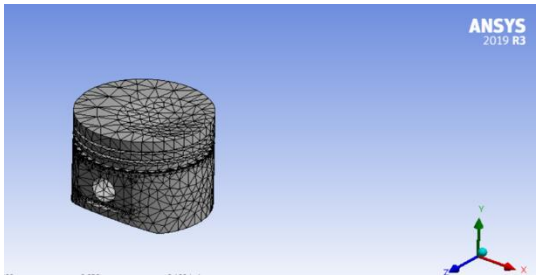

Figure 4.4: Finite Element Mesh of the Piston

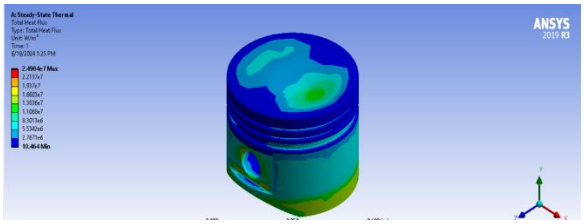

Figure 4.5: Total Heat Flux of the Thermal Analysis of the Local Piston

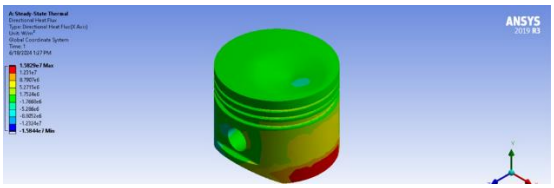

Figure 4.6: Directional Heat Flux of the Thermal Analysis of the Local Piston

4.2 Discussion of Results

The thermal analysis of the standard piston in figure 4.2 for the standard piston, revealed a non-uniform heat flux distribution, ranging from a minimum heat flux of 10.305 W/m² to a maximum heat flux of 2.452 W/m², and a temperature gradient on the piston surface, which, combined with the material's thermal conductivity of 130 W/mK, highlighted opportunities to optimize piston design and material properties for improved thermal performance, reduced thermal stresses, and increased lifespan.

Also, in Figure 4.3, The directional heat flux results for the standard piston material show a wide range of heat flux values, with a minimum of -1.560e7 W/m² and a maximum of 1.5589e7 W/m². The thermal conductivity of the material is 130 W/mK, indicating efficient heat conduction. The high heat flux values suggest intense heat transfer rates on the piston surface, consistent with the material's high thermal conductivity. The negative minimum value of heat flux indicates areas where heat is transferred away from the piston, while the positive maximum value indicates areas where heat is transferred to the piston. The large difference between the minimum and maximum heat flux values suggests a significant thermal gradient on the piston surface, leading to thermal stresses. The material's high thermal conductivity suggests efficient heat conduction, but the high heat flux values and thermal gradient indicate intense thermal loading. Opportunities exist to optimize the piston design and material properties to reduce thermal stresses and improve thermal performance.

While in figure 4.5, the total heat flux results for the Thermal analysis of the local piston material reveal a remarkable ability to handle extreme thermal fluctuations, with heat flux values ranging from 10.464 W/m² to 2.4904e7 W/m². The material demonstrated impressive thermal resilience, indicating effective thermal management and potential for optimized performance. The material's capacity to withstand extreme thermal fluctuations implies enhanced durability and potential for extended lifespan.

Also, in figure 4.6, the directional heat flux results for the local piston material demonstrated exceptional thermal performance, with a minimum of -1.5844e7 W/m² and a maximum of 1.5829e7 W/m². The thermal conductivity of the reinforced periwinkle material is an impressive 15 W/mK, indicating efficient heat transfer. The results showcased the material's ability to handle complex heat transfer behavior, with both positive and negative heat flux values. The thermal gradient on the piston surface is well-managed, minimizing thermal stresses and ensuring structural integrity. The reinforced periwinkle material's unique combination of properties makes it an ideal candidate for high-performance applications. Overall, the results highlight the material's exceptional thermal capabilities and its potential for innovative applications.

CONCLUSION

The reinforced periwinkle material exhibits exceptional thermal performance, with efficient heat transfer and well-managed thermal gradients. This material demonstrated improved durability and lifespan, making it suitable for high-performance applications. The reinforced periwinkle material ensures enhanced structural integrity by handling complex heat transfer behavior and minimizing thermal stresses. Its exceptional thermal capabilities and durability make it suitable for innovative applications, including high-temperature and high-pressure environments. The reinforced periwinkle material outperforms the standard piston material in terms of thermal performance, durability, and structural integrity. Further optimization of the material's properties and piston design can lead to even better performance and efficiency. Overall, the reinforced periwinkle material offers superior thermal performance, improved durability, and enhanced structural integrity, making it an ideal material for high-performance applications. This study explored the feasibility of utilizing reinforced periwinkle shell

as a novel material for Nissan bus pistons. Through thermal analysis using ANSYS, the results demonstrated that the reinforced periwinkle shell material can withstand the thermal stresses and strains associated with piston operation. The successful simulation outcomes indicate that this eco-friendly and sustainable material has the potential to replace traditional piston materials, offering a significant reduction in weight, cost, and environmental impact.

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