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# Reasarch on Load Distribution of Different Involute Spline Clearance

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***Abstract***

A spline coupling consists of a gear with external teeth inserted into another gear with internal teeth. The gap that exists in between the spline teeth is called gap clearance. There were a lot studies conducted on the spline design in various aspects, but no study focuses on the different spline gap clearances. This research aims to analyse the load distribution of different involute spline clearance. The project involved modelling an two actual involute splines coupling models that were identical except for the tolerance in the gap clearance. Finite element analysis was conducted to investigate load distribution along the tooth profile. The results show greater stress value on the spline coupling with maximum gap clearance. The spline coupling material's stress versus number of cycles-to-failure (S-N) curve was used to determine the structural durability of each gap clearances under different loads. The spline coupling with zero gap clearance was found experiencing an infinite life, whereby an infinite number of cycles applied onto the spline coupling would not cause any failure to occur. On the other hand, the results of the simulation showed that the spline coupling with maximum gap clearance would fail after a few cycles applied to it.

***Keywords:*** *finite element analysis, involute spline, gap clearance, S-N curve.*

## I. INTRODUCTION

In mechanical transmission system, splines are often used to transfer rotary motion and torsion produced from the internal combustion engine to the transmission box, or normally term as, the gear box. The internal combustion engine and the transmission box are connected through a flywheel with a spline coupling that exist in the middle of it. The spline coupling has an involute tooth profile instead of a cycloidal tooth profile due to a few advantages that the former has over the latter. An involute tooth profile offers advantages such as smoother gear and less wear over time, simplicity in manufacturing, and more flexibility in assembly due to having a larger tolerance in manufacturing.

The involute spline coupling consists of an output shaft with external teeth inserted into a driven shaft with internalteeth both along the same rotational axis. Rotational movement and torsion are transmitted by the means of the internal and external teeth engaging with each other. The distance between the flanks of the internal and external teeth is called gap clearance and different manufacturers have different preferences on the tolerance of the gap clearance. Fig. 1 shows involute splines with (a) maximum gap clearance and (b) zero-gap clearance along with their terminology.

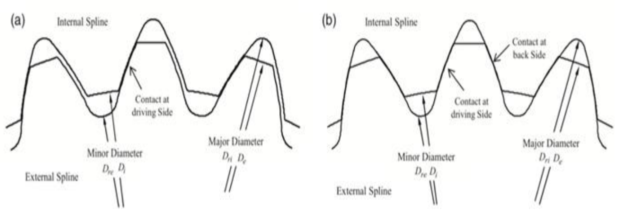


Fig.1. Schematic side views of segments of (a) a side-fit spline, and (b) a side-fit spline having back side contact.

The best tolerance for the gap clearance in between the spline teeth is still unknown to all, with different manufacturers making different claims for their own preferences. Some manufacturer claims that the best tolerance is to have no gap, while some

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prefers a certain gap. Picking the right tolerance for the gap clearance is crucial as it will able to mitigate or even prevent many issues from arising on the spline couplings. A few cases have been reported in which splines have failed. Examination of the failed parts shows that the gear tooth started wearing out at a very fast rate and begins to fail in the early stage of the gear life. An understanding of why these splines are failing and what can be done to eliminate the problem would be highly useful. Failure can be avoided much earlier, and a better tolerance of gap clearance can be implemented, which results in longer operational life of spline couplings, if the load distribution of different involute spline gap clearance is known.

Every material has mechanical properties that explains their characteristic. For instance, when a material is experiencing stress that exceeds its tensile yield strength, yielding of the material will naturally occur. Moreover, if the stress increases further towards exceeding its tensile ultimate strength, the material will break. Hence, the safety factor of the spline coupling will be taken into consideration when comparing the load distribution between the spline coupling with zero-gap and certain gap clearance.

Most journal articles focus on zero-gap clearance, or normally term as clearance-fit spline joints, as the base of their research, and the issues and failures that they discussed and analysed are not related to the different in gap clearance. Hence, this research tackles the subject of, how does the load distribution of the spline teeth contribute to the possible failures of a certain gap clearance and compare it with the load distribution of the zero-gap clearance. Issues and failures caused by both the zero-gap and certain gap clearance case are discussed and analysed in this research.

Issues and failures experienced by the spline coupling cannot be thoroughly understood unless its load distribution along the spline teeth are known. Different tolerance for the gap clearance may bring about different load distribution on the spline teeth. Hence, a comprehensive study on comparing the load distribution and structural durability of different gap clearance of spline teeth were carried out in this research by using the finite element analysis method. The comparison study of load distribution was between two spline couplings with different gap clearance conditions. With that said, two spline couplings with the same dimensions but different gap clearance, such as maximum and zero gap clearance condition, was modelled in this project.

The objectives of this project are to develop surface integral contact analysis model and conduct finite element analysis on the load distribution in different spline clearance conditions and to analyse the spline material's Stress versus Number of cycles-to-failure (S-N) curve for each gap clearance condition.

## II. RESEARCH METHOD

The research methodology of this project consists of the design of the volute and the simulation process. Design, Involute Spline Design

Two 3D models of the involute spline were drawn using the 3D modelling software, SolidWorks. Both involute spline model was identical in dimensions except for the tolerance in the gap clearance. Both involute spline models were put into simulations using the finite analysis method to analyse the load distribution of different tolerance in gap clearance. The software, ANSYS, was used to carry out the simulation of both involute spline models. The two models of the involute spline were designed accordingly to the design parameters from the standard DIN 5480-1 shown in the Table 1 below.

The involute spline was drawn by using the parametric involute curve equation, shown below, was used in generating the involute curve on the gear teeth when modelling in SolidWorks.

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Table 1: Design parameters from DIN 5480-1 [5]

| Parameters | External Teeth | Internal Teeth |
|---|---|---|
| Number of Teeth, z | 25 | |
| Pitch Circle Diameter, $D_{ref}$ (mm) | 26.458 | |
| Pressure Angle, ϕ | 30° | |
| Base Diameter, $D_b$, (mm) | 22.9136 | |
| Module, m | 1.0538 | |
| Major Diameter, $D_o$ / $D_{r_i}$ (mm) | 27.400 | 28.590 |
| Minor Diameter, $D_{r_e}$ / $D_i$ (mm) | 23.876 | 25.400 |
| Form Diameter, $D_{f_e}$ / $D_{f_i}$ (mm) | 25.298 | - |
| Tooth thickness effective, $t_v$ (mm) | 1.641 | |
| Space Width effective, $s_v$ (mm) | 1.661 | |

$$0.5(D_B)[(t\cos(t\pi)+t\sin(t\pi)] \quad (1)$$

$$0.5(D_B)[(t\cos(t\pi)-t\sin(t\pi)] \quad (2)$$

Where $D_B$ is the diameter of the base circle and t is the sweep parameter of the involute curve.

The equations (1) and (2) are the parametric involute curve equations used to create the involute curve for a gear tooth in SolidWorks, respectively. The equations were inserted when drawing an equation driven line. Under the parametric option, equations (1) and (2) were keyed in as x(t) and x(t), respectively. The sweep parameter of the involute curve ranged from 0 to 1 with $t_1$ being 0 and $t_2$ being a value from 0 to 1. The greater the value of $t_2$, the further the involute curve line was generated beyond the major diameter of gear. Hence, in this case, $t_2$ was set to 1 as it did not affect the drawing as long as the involute line intersects the major diameter of the gear.

The two models of the involute spline were design in which, one of them had no gap clearance and the other had a certain gap clearance. Effective clearance was another name for the gap clearance and it was calculated by deducting the effective tooth thickness from the effective space width of the spline teeth. From Table 1, the effective space width and tooth thickness are 1.661 mm and 1.641 mm, respectively. Thus, the effective clearance for the maximum gap case is 0.02 mm. As for the no gap clearance case, the gear teeth have a clearance-fit design whereby, the effective tooth thickness was the same as the effective space width, hence making the effective clearance between the internal and external spline teeth a 0 mm. The gap clearance of the spline coupling is the space in between the flanks of the spline teeth as marked as seen in Fig. 2 and 3.

## Simulation Process

Simulations were carried out on the two involute spline couplings using the finite element analysis method with the ANSYS software to analyse the load distributions along the spline teeth of each cases of spline gap clearance. Since the involute spline couplings was designed to have the exact same dimension as the actual spline coupling currently being used in cars transmissions, the torque input into the simulation was the same as the actual torque. This was being done so to obtain and analyse the load distribution that was as close as possible to the actual scenario.

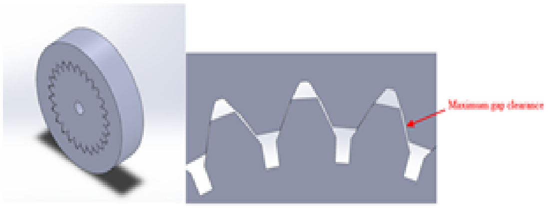


Fig.2. Involute spline coupling with maximum gap clearance.

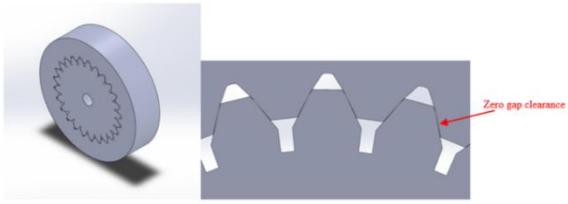


Fig.3. Involute spline coupling with zero-gap clearance.

## ANSYS Set up

The transient structural solver was used in this case because wear damage occurs over-time, so time has to be one of the variables in the simulation.

The material of the spline coupling is SCM 435 Alloy steel and it has a density of 7700 kg/m$^3$. The Young's modulus and Poisson's ratio of the alloy steel is 200 GPa and 0.27. SCM435 has an ultimate tensile strength of 991 MPa and an ultimate yield strength of 902 MPa.

Two types of connections were added into the transient structural analysis. The first connection was bonded contacts where both internal and external spline teeth contacted with each other and the second connection was a revolute joint that tells the simulation that the gears were rotating about an axis.

A few mesh settings were applied onto the internal and external spline to increase the accuracy of the simulation results. The refinement of the mesh can be validated by the average element quality and skewness of the mesh shown in the simulation. The closer the value of the average element quality to 1, the better it is in term of the quality of the mesh. On the other hand, the quality of the mesh improves as the average skewness value gets closer to 0. The spline coupling mesh is shown in Fig. 4.

For refinement, the mesh method was changed from automatic to tetrahedral method as it showed an improvement of mesh quality as in Table 2. Other than that, a contact sizing was added to refine the mesh on the spline teeth of both external and internal spline gear where they are in contact with each other. The element size for the contact sizing was set to 0.0005 mm.

The simulation of each spline gap condition produced three sets of results due to the internal spline gear being applied with moment force of 110 Nm, 155 Nm and 205 Nm. This is because when a car is in idle state, the torque input is at 110 Nm and the maximum torque input from the internal

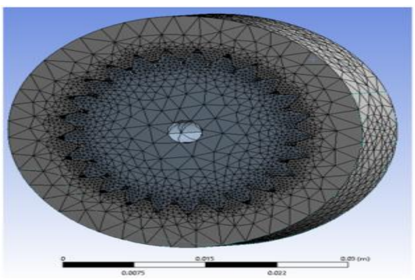

Fig. 4. Meshing of spline coupling.

Table 2: Mesh Metric

| | Spline Coupling with Max Gap | | Spline Coupling with Zero gap | |
|---|---|---|---|---|
| | Auto generated | Refined | Auto generated | Refined |
| Average Element Quality | 0.50590 | 0.70797 | 0.46511 | 0.70969 |
| Skewness | 0.38280 | 0.41124 | 0.39268 | 0.41959 |
| Number of Elements | 4116 | 125208 | 4721 | 130153 |
| Number of Nodes | 22846 | 210289 | 25867 | 218035 |

combustion engine to the flywheel is at 205 Nm. A moment force of 155 Nm was added as a control variable in this simulation. On the other hand, a fixed support was added to the external spline to study the load distribution on both spline gears once the driver gear meets the driven gear with the specific moment force. Besides, a standard earth gravity of -9.8066 m/s$^2$ was also applied to both spline gears

## III. RESULTS AND ANALYSIS

The results of the simulation are compared between the maximum gap clearance and zero gap clearance with different loads applied.

Maximum Gap Clearance

The spline coupling with maximum gap clearance was put into a static structural simulation with an applied moment to the internal spline gear. The moment that was applied onto the internal gear are 205 Nm, 155 Nm, and 110 Nm.

A.1 Maximum gap clearance with 205 Nm

The stress distribution of the spline coupling with maximum gap clearance under the applied moment force of 220 Nm, is displayed in Fig. 5. The torque is the maximum torque that comes from the internal combustion engine. The maximum and minimum stress that the spline coupling with maximum gap clearance was experiencing, under 205 Nm torque, was 649 MPa and 6.2 KPa, respectively. Due to the maximum gap clearance in between teeth, the stress distribution was highly concentrated at the region of

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the side of the teeth where the internal gear meets the external gear. The other side of the spline teeth that was not in contact with each other during operation has a lower stress concentration when compared to the other side.

The stress distribution at the internal and external spline gear was shown in Fig. 6 and 7, respectively. High stress intensity could be seen at both the edges of the spline gear teeth. This means that that region underwent shortening of material life faster than the other region.

The S-N curve in Fig. 8 shows that the spline coupling with maximum gap clearance under 205 Nm will have failure after 8 $\times 10^4$ number of cycles. This means that once 8 $\times 10^4$ number of cycles of the 649.3 MPa was applied to the material, failure would to occur.

A.2 Maximum gap clearance with 155 Nm

Fig. 9 shows the stress distribution of the spline coupling with maximum gap clearance under the applied moment force of 155 Nm. The maximum and minimum stress that the spline coupling in Fig 9 as experiencing were 490 MPa and 4.68KPa, respectively.

It can be seen in Fig. 10 and 11 that the stress experienced by the spline coupling was higher at the external spline gear than the internal spline gear. The stress distribution could also be seen to be more concentrated at both the side edges of the gear teeth.

The S-N curve in Fig. 12 shows the spline coupling with maximum gap clearance under 155 Nm torque falling into the infinite life region. This means that an infinite number of cycles can be applied to the material without failing.

A.3 Maximum gap clearance with 110 Nm

Fig. 13 shows the stress distribution of spline coupling with maximum gap clearance under the applied torque of 110 Nm. It could be seen that the maximum and minimum stress that it was experiencing were 347 MPa, and 3.3 KPa. The external spline gear seems to always be experiencing a higher stress value than the internal spline gear, as seen in Fig. 14 and 15 below.

The S-N curve in Fig. 16 also shows the spline coupling with maximum gap clearance under 110 Nm torque falling into the infinite life region. This means that an infinite number of cycles can be applied to the material without failing.

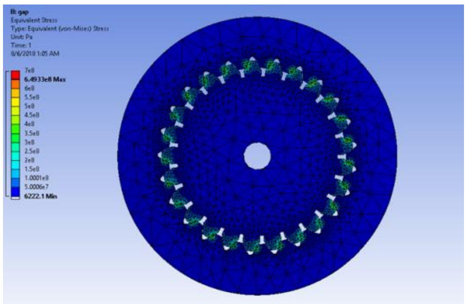

Fig 5: Front view of stress distribution of spline coupling with maximum gap clearance under 205 Nm torque.

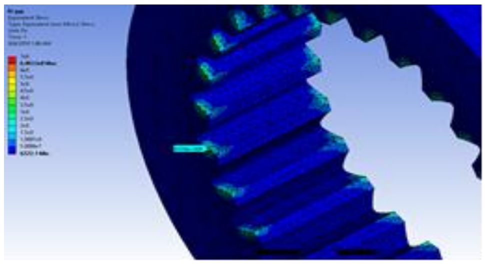

Fig 6: Stress distribution of maximum gap clearance internal spline gear under 205 Nm torque.

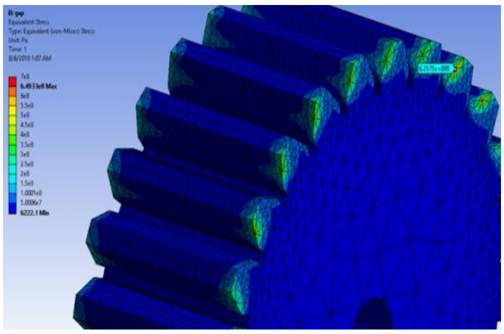

Fig 7: Stress distribution of maximum gap clearance external spline gear under 205 Nm torque

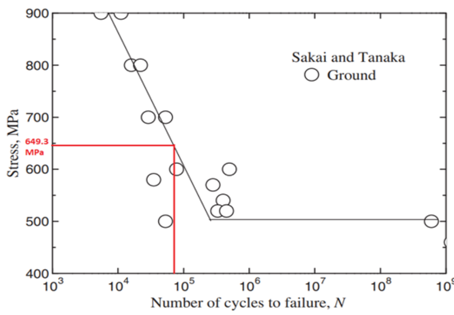


Fig 8: S-N curve for spline coupling with maximum gap clearance under 205 Nm torque

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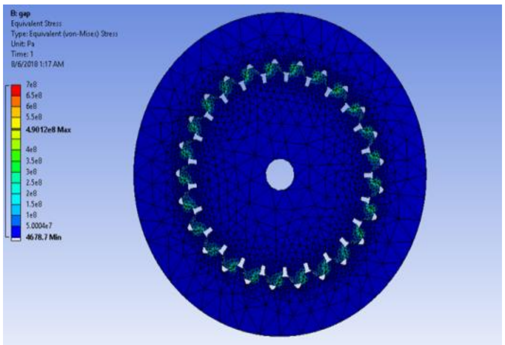

Fig 9: Front view of stress distribution of spline coupling with maximum gap clearance under 155 Nm torque

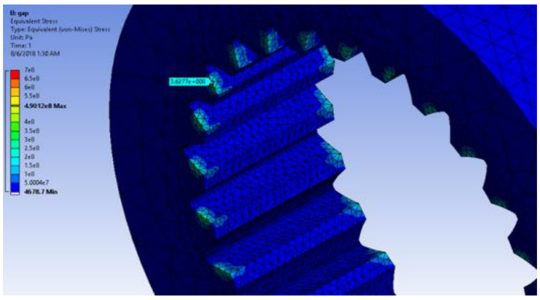

Fig 10: Stress distribution of maximum gap clearance internal spline gear under 155 Nm torque

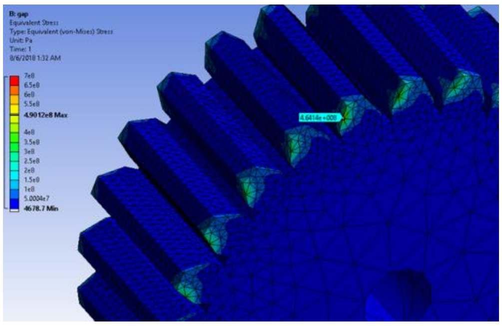

Fig 11: Stress distribution of maximum gap clearance external spline gear under 155 Nm torque

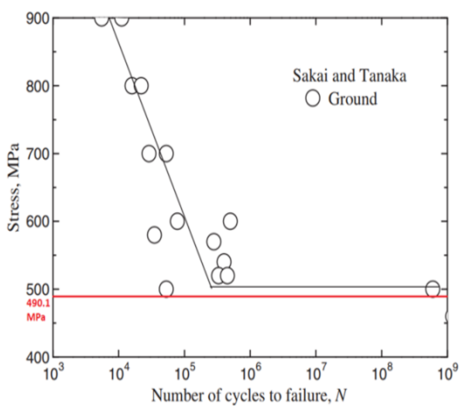


Fig 12: S-N curve for spline coupling with maximum gap clearance under 155 Nm torque

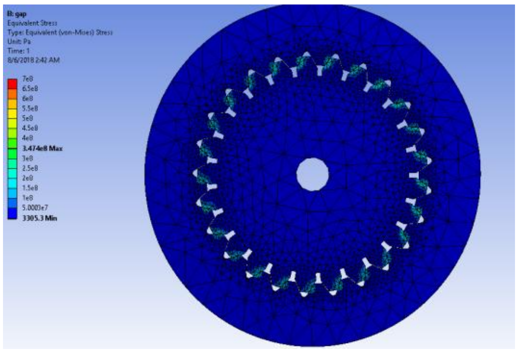

Fig 13: Front view of stress distribution of spline coupling with maximum gap clearance under 110 Nm torque

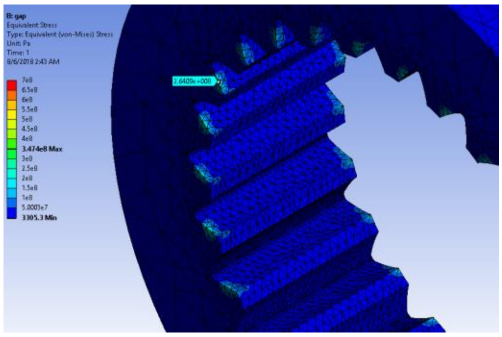

Fig 14: Stress distribution of maximum gap clearance internal spline gear under 110 Nm torque

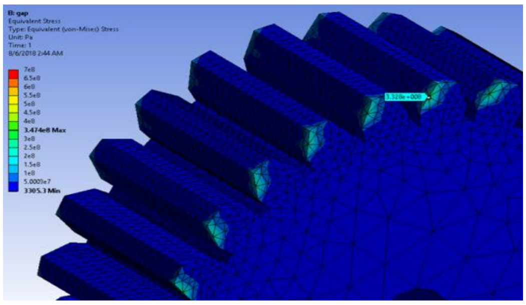

Fig 15: Stress distribution of maximum gap clearance external spline gear under 110 Nm torque

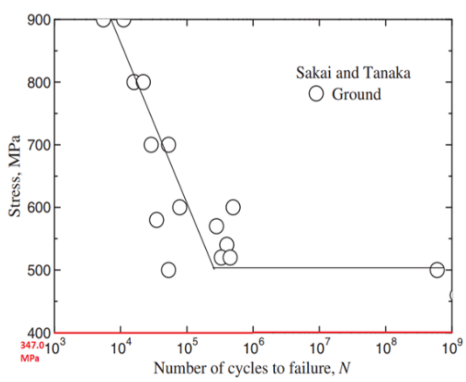


Fig 16: S-N curve for spline coupling with maximum gap clearance under 110 Nm torque

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Zero Gap Clearance

The spline coupling with zero gap clearance was also put into a static structural simulation with an applied moment to the internal spline gear. The moment that was applied onto the internal gear are 205 Nm, 155 Nm, and 110 Nm.

B.1 Zero gap clearance with 205 Nm

The stress distribution of the spline coupling with zero gap clearance in Fig. 17 shows a uniform distribution across both side of the spline teeth. This is mainly because both side of the spline teeth are in contact with each other. The maximum and minimum stress experienced by the spline coupling with zero gap clearance when being applied with 205 Nm of torque were 391.4 MPa and 3.26 kPA, respectively.

Fig. 18 and 19 also shows the stress distribution of the spline coupling with zero-gap clearance to have high concentration of stress at both edges of the spline teeth. This points to faster material life shortening that that region. The edges of the spline tooth flank do not have as high intensity of stress concentration as the edges. Hence, the edges of the spline tooth flank will not experience shortening of material life as quickly as the other edges of the spline tooth.

The S-N curve in Fig. 20 also shows the spline coupling with zero gap clearance under 205 Nm torque falling into the infinite life region. This means that an infinite number of cycles can be applied to the material without failing.

B.2 Zero gap clearance with 155 Nm

The maximum and minimum stress experienced by the spline coupling with zero gap clearance under the torque input of 155 Nm are 295.7 MPa and 3.11 kPa as shown in Fig. 21 – 23.

The S-N curve in Fig. 24 also shows the spline coupling with zero gap clearance under 155 Nm torque falling into the infinite life region. This means that an infinite number of cycles can be applied to the material without failing.

B.3 Zero gap clearance with 110 Nm

The maximum and minimum stress experienced by the spline coupling with zero gap clearance under the torque input of 110 Nm are 209.9 MPa and 2.63 kPa as shown in Fig, 25 – 27.

The S-N curve in Fig. 28 also shows the spline coupling with zero gap clearance under 110 Nm torque falling into the infinite life region. This means that an infinite number of cycles can be applied to the material without failing.

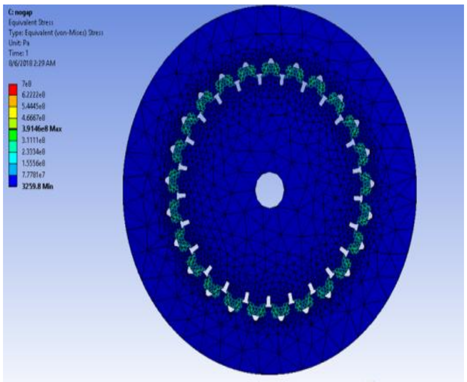

Fig 17: Front view of stress distribution of spline coupling with zero gap clearance under 205 Nm torque

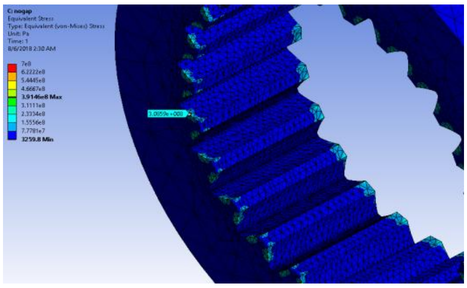

Fig 18: Stress distribution of zero gap clearance internal spline gear under 205 Nm torque

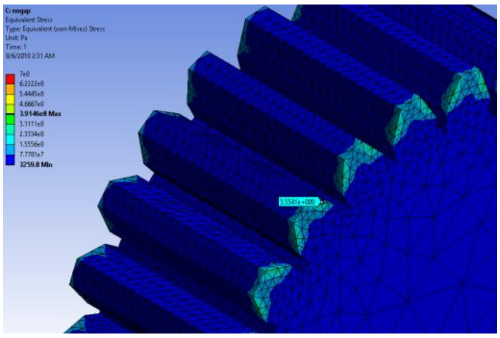

Fig 19: Stress distribution of zero gap clearance external spline gear under 205 Nm torque

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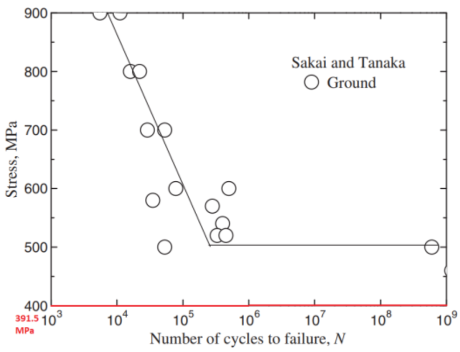


Fig 20: S-N curve for spline coupling with zero gap clearance under 205 Nm torque

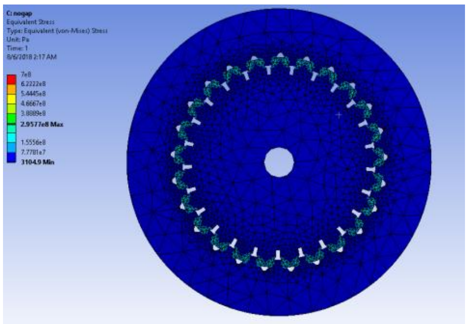

Fig 21: Front view of stress distribution of spline coupling with zero gap clearance under 155 Nm torque

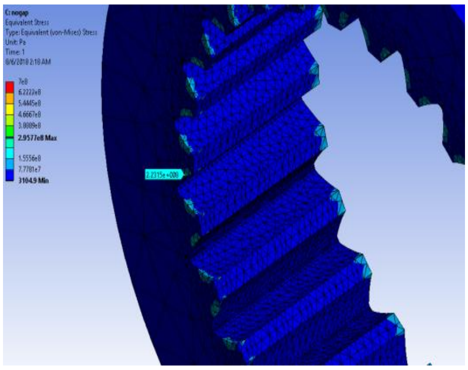

Fig 22: Stress distribution of zero gap clearance internal spline gear under 155 Nm torque

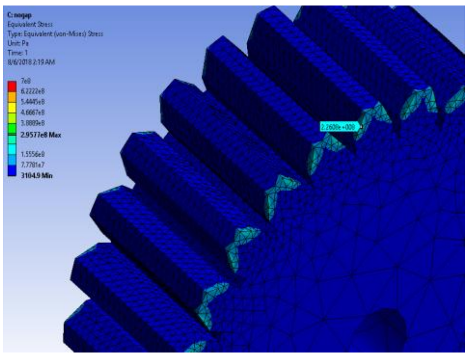

Fig 23: Stress distribution of zero gap clearance external spline gear under 155 Nm torque

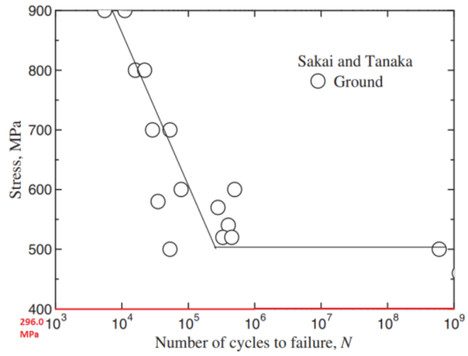


Fig 24: S-N curve for spline coupling with zero gap clearance under 155 Nm torque

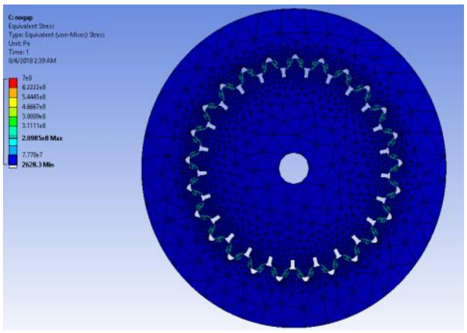

Fig 25: Front view of stress distribution of spline coupling with zero gap clearance under 110 Nm torque

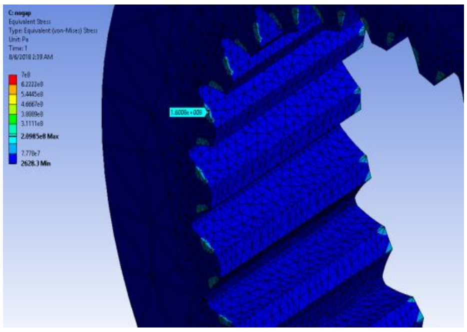

Fig 26: Stress distribution of zero gap clearance internal spline gear under 110 Nm torque

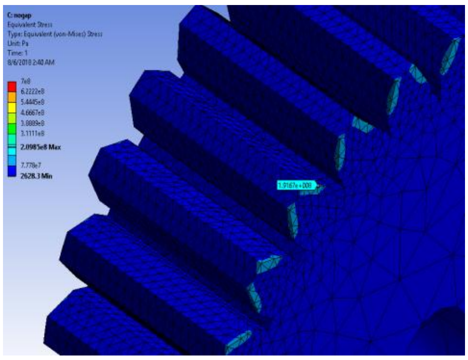

Fig 27: Stress distribution of zero gap clearance external spline gear under 110 Nm torque

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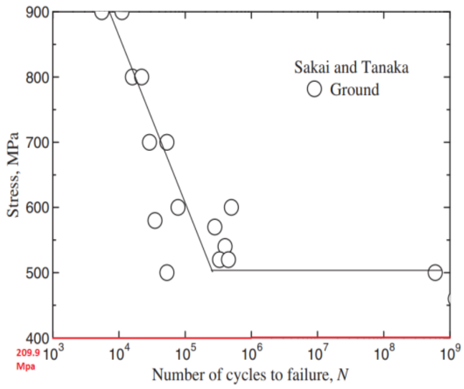


Fig 28: S-N curve for spline coupling with zero gap clearance under 110 Nm torque

## IV. DISCUSSION

As summarized in the Table 3.1, for the spline coupling with maximum gap clearance, the maximum stress it experienced increases along with the input torque. When the input torque increases from 110 Nm to 155 Nm, the maximum stress experienced by the spline coupling with maximum gap clearance increased 41%. When the input torque increases from 155 Nm to 205 Nm, an increase of 33% in maximum stress experienced by the spline coupling occurs. Whereas, for the spline coupling with zero gap clearance, the maximum stress it experienced also increased 41% when the torque input increases from 110 Nm to 155 Nm. When the torque input increases again to 205 Nm, the maximum stress experienced by the spline coupling increased for 32%, which is still very close to the increase of maximum stress, in percentage, for the spline coupling with maximum gap clearance.

Table 3.1: Results of Simulation

| | | Maximum gap clearance | | Zero gap clearance | | Increase between zero and maximum gap clearance, % |
|---|---|---|---|---|---|---|
| | | Max stress, MPa | Min stress, kPa | Max stress, MPa | Min stress, kPa | |
| Torque input, Nm | 110 | 347.0 | 3.31 | 209.9 | 2.63 | 65.3 |
| | 155 | 490.1 | 4.68 | 296.0 | 3.11 | 65.6 |
| | 205 | 649.3 | 6.22 | 391.5 | 3.26 | 65.8 |

It can also be seen that the maximum stress experienced by the spline coupling with maximum gap clearance are higher in all three torque input cases when compared with the maximum stress experienced by the spline coupling with zero gap clearance. Table 3.1 also shows that the maximum stress experienced by the spline coupling with zero gap clearance has an increase of around 65%, for all three torque input cases. Thus, the hypothesis that mentioned the spline coupling with zero gap clearance having a higher stress than the maximum gap clearance has been rejected.

However, the hypothesis that mentioned that the load distribution on the spline coupling with zero gap clearance will have a more uniform distribution when compared to the spline coupling with maximum gap clearance can be accepted. Fig. 6 and 7 clearly shows that the stress distribution on the spline with maximum gap clearance has high stress concentration at the contact side of the spline gear teeth, whereas the spline with zero gap clearance has a more uniform stress concentration at both contact side of the spline gear teeth.

Besides, high intensity of stress is seen to be concentrated at both edges of the spline gear at both the internal and external spline teeth. This means that the edge region will undergo a faster shortening of material life than the other region, where there is not that high intensity of stress distribution.

The simulation patterns where the wear was at the edge region were consistent with actual spline that went through durability test. Fig. 29 shows the spline after the test and Fig. 30 shows the contour of that spline where the edge was worn.

Fig 29: Actual spline condition after test

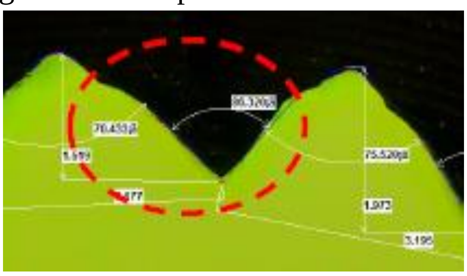

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Fig 30: Contour test that shows wear at edge

Table 3.2 shows the number of cycles to failure for each torque input applied onto the spline coupling with maximum and zero gap clearance. Only the spline coupling with maximum gap clearance that is under 205 Nm of torque will have a limited number of cycles to failure and that is $8 \times 10^4$. This shows that the stress experienced by the spline coupling with zero gap clearance will have infinite life throughout all three torque input cases. On the other hand, the spline coupling with maximum gap clearance also has two torque input cases that shows the result of infinite number of cycles to failure. However, when the torque input increases up to 205 Nm, the region of highly concentrated of stress will experience failure after number of cycles.

Table 3.2: Structural Durability of Spline Coupling with different Gap Clearance

| | Torque input, Nm | Number of cycles to failure |
|---|---|---|
| Spline coupling with maximum gap clearance | 110 | Infinite life |
| | 155 | Infinite life |
| | 205 | $8 \times 10^4$ |
| Spline coupling with zero gap clearance | 110 | Infinite life |
| | 155 | |
| | 205 | |

## V. CONCLUSIONS

To conclude, the load distribution of spline coupling with maximum gap clearances are much greater than the load distribution of spline coupling with zero gap clearances. Since the maximum gap clearance experiences greater stress distribution, the rate of material life shortening is much faster when compared to the zero-gap clearance condition. Also, the spline coupling with maximum gap clearances is more prone to experience failure at a certain torque input after a number of cycles whereas, the spline coupling with zero gap clearances experiences infinite life with the current torque input. Thus, it can be concluded that the spline coupling with zero gap clearance has the longer operational life compared to the maximum gap clearance

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