The contact analysis for Barrel coupling based on Ansys

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ABSTRACT

The Barrel coupling system is the method of choice in transferring torque between rotating parts. The barrel coupling consists of a sleeve provided with semicircular toothing around its internal diameter and a hub that is externally toothed. A series of cylindrical barrels, of hardened steel, are inserted in the holes formed by this toothing to act as power transmission element. The contact mechanism of barrel and sleeve and hub behaves highly non-linear and generates contact stresses. The contact stress is the key parameter in barrel coupling design. This paper presents to determine maximum contact stress occurring in barrel coupling. The results obtained from theoretical Hertzian analytical method are compared with Finite element analysis values. In finite element analysis different contact algorithm and contacts are employed to find best possible FEA contact analysis technique. In this paper factors affecting on barrel coupling size like the Nominal transmission torque and Radial load are discussed. Barrel Coupling was modeled in Catia V5 CAD software and analysis of this was done by using ANSYS work bench.

Keywords— Barrel coupling, Radial load, contact Stresses, Transmission torque, Contact algorithm

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I. INTRODUCTION

Coupling is a mechanical device used to connect two shafts together at their ends for the purpose of transmitting power. Couplings do not allow disconnection of shafts during operation, however if torque limit is exceeded beyond designed value then coupling may get disconnected during operation. Coupling can permit some degree of misalignment among the connecting shafts. By careful selection, installation and maintenance of coupling substantial saving can be made in maintenance cost.

A Barrel coupling

The barrel coupling consists of a sleeve provided with semicircular toothing around its internal diameter and a hub that is externally toothed as shown in fig.1.7. A series of cylindrical barrels, of hardened steel, are inserted in the holes formed by this toothing to act as power transmission element. Covers with their corresponding special seals serve to assure the perfect-tightness of the inner zone, preventing the penetration of dust and guaranteeing the continuity of the necessary lubrication. Two double-lamina elastic rings mounted on the hub, one on each side of the toothing, limit the axial displacement of the barrels. Torque is transmitted to the drum’s receiving flange, generally by two diametrically opposed flat driving surfaces, located at the periphery of the coupling flange, and also by means of bolts which, at the same time, serve as connection with the drum. An indicator located on the external cover which moves relative to the marks on the hub as a function of wear, permits control of the toothing, limit the axial displacement of the barrels. Torque is transmitted to the drum’s receiving flange, generally by two diametrically opposed flat driving surfaces, located at the periphery of the coupling flange, and also by means of bolts which, at the same time, serve as connection with the drum. The same indicator also serves to control the axial position of the sleeve relative to the hub. The convex shape of the barrels and the internal spaces of the toothing allows the oscillation of the hub relative to the sleeve compensating angular misalignments of ± 1° 30’ and an axial displacement that varies between ± 3 mm and ± 8 mm.
Advantages
1. Barrel couplings have increased contact area, radial load is better distributed hence life of coupling is increased.
2. Due to barrel and gear profile, for a given radial load 40% stress reduction is obtained compared to other couplings.

II. ANALYTICAL METHODOLOGY

Analytical methodology divides into two parts; first part includes calculation of transmission torque and calculation of radial load acted on barrels. Second part includes calculation of contact stress induced due to application of torque and radial load by using Hertzian theory.

A. Calculation of nominal transmission torque T (Nm)

1) Based on installed power

$$T = \frac{9550 \times P_i \times K_1}{N}$$

$$T = \frac{9550 \times 1 \times 1.8}{4}$$

$$T=4300 \text{ Nm}$$

The static pull in the drum is given by

$$F_p = \frac{(Q + G)}{i_T \times K_2}$$

2) Based on consumed power

$$P_c = \frac{F_p \times v}{60000}$$

$$P_c = \frac{10800 \times 5}{60000}$$

$$P_c = 0.9 \text{ Kw}$$

$$T = \frac{P_c \times 9550 \times K_1}{N}$$

$$T = \frac{0.9 \times 9550 \times 1.8}{4}$$

$$T=3870 \text{ Nm}$$

B. Calculation of radial load $F_r$

$$F = \left( F_p \left( 1 - \frac{b_l}{l} \right) + \frac{w}{2} \right)$$

$$F = \left( 16315.9 \left( 1 - \frac{400}{1200} \right) + \frac{7000}{2} \right)$$

$$F=14377.4 \text{ N}$$

C. Strubeck’s Equation

It is used for distribution of radial forces among barrels on lower half part of coupling. It is based on the following assumptions:

1) The rollers are rigid and they retain their circular shape.
2) The rollers are equally spaced.
3) The rollers in the upper half portion not support any load.

Figure 3.1 shows the forces acting on the inner race through rolling elements, that supported maximum radial load $F_r$. 

Fig.1 Barrel coupling

|--------|---------------|------------------------|-----------------|

Fig.2 Distribution of forces
Stribeck's equation is given by:

$$F_r = \frac{\text{Total no. of barrels} \times P_1}{\text{Stribbeck's factor}}$$

Where Stribeck's factor = 5.

$P_1, P_2, P_3 \ldots$ are distributed forces

For this model of barrel coupling there are 20 barrels.

Total radial load 14500 N is converted into 9 forces according to stribeck's equation.

14500=1/5*20*P1

$P_1=3625 \text{ N (maximum radial load acted on barrel)}$

$P_2 = P_1(\cos 18)^{3/2}$

$P_2 = 3625 (\cos 18)^{3/2}$

=3362.15 N

$P_3 = 3625 (\cos 36)^{3/2}$

= 2637.18 N

$P_4 = P_1(\cos 54)^{3/2}$

$P_4 = 3625 (\cos 54)^{3/2}$

= 1633.56 N

$P_5 = P_1(\cos 72)^{3/2}$

$P_5 = 3625 (\cos 72)^{3/2}$

=622.7 N

D. Hertz contact stress theory
Contact between two continuous, non-conforming solids is initially a point or line. Under the action of a load the solids deform and contact area is formed. According to the contact area shape (under no external load), there are point contact and line contact. It is obvious that after load applied line contact will become rectangle contact and point contact will be an ellipse contact area. Hertz contact stress theory allows for the prediction of the resulting contact area, contact pressure, compression of the bodies, and the induced stress in the bodies. The maximum principal stresses occurring at the surface of contact are given by Hertzian equation are following:

$$\sigma_x = \frac{C_r}{b} \Delta$$

$$\sigma_y = 2\sigma_x$$

$$\sigma_z = \frac{C_r}{b} \Delta$$

Maximum principle stress is given by following:

$$\sigma_{max} = \frac{C_r}{b} \Delta$$

Where $b$ is semi width of formed rectangular contact area is given by

$$b = \sqrt{2\Delta \rho \pi}$$

$$\Delta = \frac{1}{1/2(R_1) + 1/2(R_2)} (\frac{1 - \eta^2}{\varepsilon_1} + \frac{1 - \eta^2}{\varepsilon_2})$$

By putting values

Maximum contact stress,

$\sigma_{max} = 650.8 \text{ mpa}$

E. Contact stress due to Torque:

$$T = f_t \times \frac{D_1}{2}$$

Tangential load = $f_t = 80357.14 \text{ N}$

Contact area is given by

$$A_c = \pi \times D_1 \times l$$

Contact area for 20 barrels=20*A_c

Contact stress=$\frac{\text{contact force}}{\text{Total contact area}} = f_t = 5.655\text{Mpa}$

Total contact stress=[(contact stress due to radial load)$^2$ + (contact stress due to torque)]$^{1/2}$

=[(650.8)$^2$ + (5.655)$^2$]$^{1/2}$

Total contact stress=650.8 mpa

Above contact stress value is maximum contact stress corresponding to maximum radial load $P_1$ obtained from stribeck's equation. Now we obtained further contact stress value to radial load $P_2, P_3, P_4, P_5$ obtained from stribeck’s equation.

III. FINITE ELEMENT ANALYSIS OF BARREL COUPLING

The Finite Element Method (FEM) is a numerical approximation method. It is a method of investigating the behaviour of complex structures by breaking them down into smaller, simpler pieces. These smaller pieces of structure are called elements. The elements are connected to each other at nodes. The assembly of elements and nodes are called a finite element model. Typical surface-surface contact’s analysis steps mainly include:

1. Build 3D geometry model and mesh.
2. Identify contact pairs.
3. Name target surface and contact surface.
4. Define target surface.
5. Define contact surface.
6. Set up element key options and real constants
7. Define and control rigid goal’s movement.
8. Apply the necessary boundary condition.
9. Define solution options and load steps.
10. Solve contact problems.
11. Look over and analyze results.

A. Creation of Geometry of Barrel coupling

3D geometry model is built in CATIA software and imported in ansys software.

B. Material Selection

For this application alloy steel have great advantages than others. Alloy steels have higher strength and toughness. It posses higher hardenability which has great significance in heat treatment of components and also better corrosion resistance compared to plain carbon steel. EN24 is a popular grade of through-hardening alloy steel due to its excellent machinability. EN24 is used in components such Axles, connecting rods, high tensile bolts, studs, power transmission slide gears, slide cams, differential sleeves, pinion sleeves, spindle gears and compensating washers. EN24 can be
further surface hardened to create components with enhanced wear resistance by induction or nitriding processing.

C. Define contact properties
Once material properties are given to coupling in Ansys, contact elements need to define. Contact properties are given in four stages in Ansys. In first stage contact class has to be defined. Generally there are two contact classes: rigid-flexible and flexible-flexible. In rigid-flexible contact, one or more of the contacting surfaces are treated as rigid. The other class flexible-flexible contact is the more common type. In this case, all contacting bodies are deformable. In second stage contact area has to be defined. There are two groups of contact: point-surface contact and surface-surface contact. In ANSYS, the contact is generated by pair. For the point-surface contact, the ‘point’ is contact and the ‘surface’ is target. For surface-surface contact, both contact and target are surfaces and they have to be specified which surface is contact and which is target.

In third stage behaviour of contact surface has to be specified. Contact surface has different types of behaviour according to different characteristics of contact. Normally there are frictional, no separation, bonded. In frictional contact, the contact body can slide on the target surface in the tangential direction. It can translate in the normal direction. This behaviour can simulate the contact opens and closes. Frictional contact is most reliable contact behaviour in analysis of barrel coupling as barrels fits in cavities of semicircular toothing of sleeve and hub where friction exists. In bonded contact no relative movement between each other in the rest of analysis is possible. They look like one body. In this analysis we have used first frictional contact and after that bonded and no separation contact is used for checking best possible contact. In fourth stage contact algorithm has to be specified in Ansys contact algorithms are used to solve contact problems. Pure Lagrange multiplier, pure penalty method and the Augmented Lagrangian are three contact algorithm are used to solve contact problems. In this analysis first the Augmented Lagrangian Method is used to solve contact problems with friction and after that pure penalty and langrange multiplier method is used for finding best possible combination.

D. Meshing
In first stage of meshing element type is specified for coupling. Different element type can be given to different parts of coupling just like sleeve, hub and barrels. For barrels, hub and sleeve part solid 187 tetrahedral element type is given. Barrel surface which comes in contact with sleeve inner and hub outer surface represents contact and target surface and separate contact and target element is given to that contact faces. CONTA 174 as contact element and Target 170 as target element is applied.

E. Setting boundary conditions and applying loads
Total radial load 14500 N is applied on coupling and torque of 4500 Nm acted on body. The total radial load is divided according to striebeck’s equation and applied to barrels in lower half portion of coupling.

F. Solution of contact stresses and deformation
With the help of simulation contact stress and deformation obtained. Generally von misses stress can be found out and helpful in analysis. Through simulation, result of the maximal contact stress was 601.06 Mpa while the Hertzian theory value was 650.08 MPa. The comparison revealed that there was good consistency between the Hertzian theory solution and finite element solution.
The above Figures 6 & 7 shows the analysis results of barrel coupling. It clearly indicates that maximum stress is occurred on barrel at the contacting region. From figure we can also know that the contact area had an approximate ellipse shape in contact area. The contact stress for particular this analysis is varies 601.06 Mpa to 66.98 Mpa. In this analysis we have used frictional contact with contact algorithm as augmented langrange, after that we have changed contact algorithm pure penalty and langrange multiplier with frictional contact and solution obtained, after that we have changed types of contact such as bonded, no separation with all three algorithm and solution obtained.

IV. RESULTS AND DISCUSSION

We have found contact stress in barrel coupling by Hertz analytical method. These contact stresses also obtained from finite element analysis. In finite element analysis different methodologies are for finding contact stress analysis. There are different types of contact such as frictional, bonded, no separation and different contact algorithms for contact detection such as pure penalty, augmented langrange, and langrange multiplier. We have made all possible combination of these types of contact along with contact algorithm to find best possible method of contact analysis in finite element analysis.

A Comparison between contact algorithms with frictional contact

In this frictional contact as type of contact is selected and three contact algorithm used one by one in finite element analysis. Results obtained are compared to Hertz analytical method. Results are shown in Table I and figure 8.

<table>
<thead>
<tr>
<th>Radial Load in N</th>
<th>Contact stress(Mpa) by Augmented Langrange</th>
<th>Contact stress(Mpa) by Pure penalty</th>
<th>Contact stress(Mpa) by Langrange Multiplier</th>
<th>Hertz Analytical Method</th>
</tr>
</thead>
<tbody>
<tr>
<td>3625</td>
<td>601.06</td>
<td>601.06</td>
<td>531.04</td>
<td>650.8</td>
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<td>3361.8</td>
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<td>580.12</td>
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<td>627</td>
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<td>2636.9</td>
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<td>502.79</td>
<td>413.1</td>
<td>555.3</td>
</tr>
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<td>1633</td>
<td>425.46</td>
<td>425.46</td>
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<td>270.8</td>
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<td>236.18</td>
<td>269.8</td>
</tr>
</tbody>
</table>

Results shown in table I revealed that contact stress obtained from contact algorithm pure penalty and augmented langrange is almost same within 5% to 7% to Hertz analytical method contact stress values but langrange multiplier method obtained results deviates more than analytical method. Frictional contact is with all contact algorithms. In Frictional contact the contact body can slide on the target surface in the tangential direction. The results for the augmented langrange and pure penalty algorithms are good for all problems provided they are used with surface to surface contact elements. The results for the langrange multiplier algorithms can be quite sensitive to matching of the nodes on contact region so values deviate more [10]. Frictional contact with pure penalty or augmented langrange nearly gives reliable solution.

B Comparison between contact algorithms with bonded contact

In this we have used bonded contact and contact algorithm is changed one by one. Results are obtained are shown in table II and fig.9 are compared to Hertz analytical method. In Bonded contact as soon as the contact is detected, then the nodes in contact are bonded in all directions and all the degrees of freedom are constrained. Not any relative movement between each other in the rest of analysis is possible. They look like one body, irrespective of loading, behaviour of other parts.

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</thead>
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<tr>
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<td>109.72</td>
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<td>2636.9</td>
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</tbody>
</table>

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Results in table II and Fig.9 reveals that bonded contact gives the same result of all three contact algorithm as in bonded contact there is no relative movement among parts. Results are highly deviates more than 25% from Hertz analytical method, so results are not reliable.

C. Comparison between contact algorithms with no separation contact

In this we have used no separation contact and contact algorithm is changed one by one. Results are obtained are shown in table no and graph are compared to Hertz analytical method.

TABLE III
COMPARISON BETWEEN CONTACT ALGORITHMS WITH NO SEPARATION CONTACT IN FEM

<table>
<thead>
<tr>
<th>Radial Load in N</th>
<th>Contact stress(Mpa) by Augmented Lagrange</th>
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<tbody>
<tr>
<td>3625</td>
<td>490.78</td>
<td>490.78</td>
<td>732.81</td>
<td>650.8</td>
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<tr>
<td>3361.8</td>
<td>436.27</td>
<td>436.27</td>
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<td>627</td>
</tr>
<tr>
<td>2636.9</td>
<td>381.74</td>
<td>381.74</td>
<td>569.37</td>
<td>555.3</td>
</tr>
<tr>
<td>1633</td>
<td>327.2</td>
<td>327.2</td>
<td>488.54</td>
<td>437</td>
</tr>
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<td>621.99</td>
<td>272.67</td>
<td>272.67</td>
<td>407.12</td>
<td>269.8</td>
</tr>
</tbody>
</table>

Results in table III and Fig.10 revealed that pure penalty and augmented langrange gives same result for no separation contact. Langrange multiplier gives more than 20% deviation to Hertz analytical method than other two contact algorithm. For 3D model pure penalty and augmented langrange gives better than langrange multiplier.

D. Comparison between contacts with augmented langrange algorithm

In this contact algorithm as augmented langrange is kept constant and contact changed with frictional, bonded and no separation. Results obtained are compared with Hertz analytical method.

TABLE IV
COMPARISON BETWEEN CONTACTS WITH AUGMENTED LANGRANGE

<table>
<thead>
<tr>
<th>Radial Load in N</th>
<th>Contact stress(Mpa) With frictional contact</th>
<th>Contact stress(Mpa) With bonded contact</th>
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</table>
Results obtained in table IV and fig.11 revealed that frictional contact gives nearly same within 5% to 7% to as result to Hertz analytical method. Bonded contact and no separation contact stress values deviate more than 20% . Other contact failed to give proper results.

E. Comparison between contacts with pure penalty algorithm
In this contact algorithm as pure penalty is kept constant and contact changed with frictional, bonded and no separation. Results obtained are compared with Hertz analytical method.

<table>
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<th>Radial Load in N</th>
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Results obtained in table V and fig.12 revealed that frictional contact gives nearly same as result to Hertz analytical method. Bonded contact and no separation contact stress values deviate more. Other contact failed to give proper results.

F. Comparison between contacts with Langrange Multiplier
In this contact algorithm as Langrange multiplier is kept constant and contact changed with frictional, bonded and no separation. Results obtained are compared with Hertz analytical method.

<table>
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</table>
Results obtained in table VI and Fig. 13 revealed that frictional contact gives nearly same and realistic result to Hertz analytical method. Bonded contact and no separation contact stress values deviate more. Other contact failed to give proper results.

V. CONCLUSION

1. Contact stresses are depend on contacting area between hub and barrel & contacting area between barrel and sleeve.
2. Contact stresses of barrel coupling are depending on inner diameter and outer diameter of barrel, material properties of coupling.
3. Contact stresses of barrel coupling are most efficiently calculated by Hertz contact theory.
4. The result of contact stress analysis of the barrel coupling by FEM and Hertz analytical method is almost same within 5% to 7%
5. In finite element analysis result obtained by contact algorithm pure penalty and augmented Langrange with frictional contact and Hertz analytical method is almost same within 5% to 7%.
6. In finite element analysis result obtained with frictional contact are more consistent to Hertz analytical method than obtained with bonded contact and no separation contact.
7. In finite element analysis result obtained with bonded contact is remains same for all three contact algorithm.

REFERENCES