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Contact Stress Analysis of Barrel Coupling

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***Abstract*— The Barrel coupling is a device used to transmit torque between two rotating parts. It consists of a sleeve provided with semicircular toothing around its internal diameter and hub that is externally toothed.A series of cylindrical barrels of hardened steel are inserted in the holes formed by toothing to act as power transmission elements. This research works aims to study effect of torque transmission and radial load on contact stress in barrel coupling. Contact stress predicted by using Hertzian analytical method. Barrel coupling is modelled using CATIAV5 software. Contact stresses are predicted using Ansys software by adopting different types of contact algorithms and contact types. Frictional contact with Augmented Langrange and pure penalty gives better results of contact stresses than langrange Multiplier contact algorithm. Coupling is used transmit torque in electric overhead crane. Results obtained from Ansys is in close agreement with Hertzian analytical method.**

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

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 internal wear of the toothing, without the need to disassemble any part of the coupling. 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 ± 1º 30’ and an axial displacement that varies between ± 3 mm and ± 8 mm.

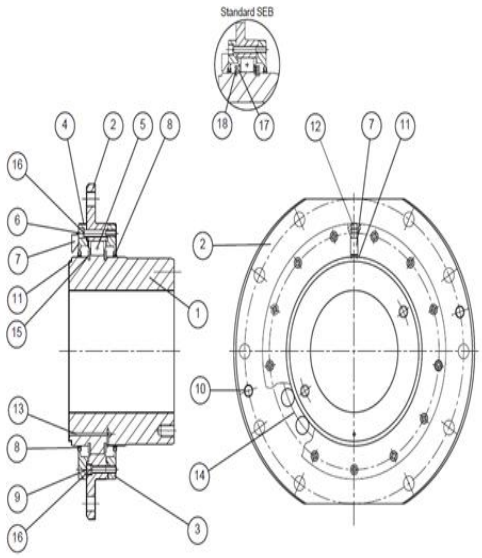

Fig.1 Barrel coupling

1.Hub	6.Allen screw	11.wear limit grooves	16.grower washer
2.sleeve	7.Indicator	12.grease connection	17.barrel guide ring
3.Inner cover	8.seal	13.grease overflow	18.seeger ring
4.outer cover	9.Allen screw	14.assembly reference	
5.Barrel	10.threaded holes for disassembly	15.barrel guide rings	

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.

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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 Nm

The static pull in the drum is given by

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

$$F_p = \frac{(60000+2000)}{4 \times 0.95}$$

F_p=13684.2 N

2)Based on consumed power

$$P_C = \frac{F_p \times v_r}{60000}$$

$$P_C \frac{13684.2 \times 4}{60000}$$

P_c= 0.91 Kw

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

$$T = \frac{0.91 \times 9550 \times 1.8}{4}$$

T= 3910 Nm

B Calculation of radial load F_r

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

$$F = \left(13684.2\left(1-\frac{400}{1200}\right)+\frac{7000}{2}\right)$$

F_r=12267.3 N

C. Stribeck'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 Fr.

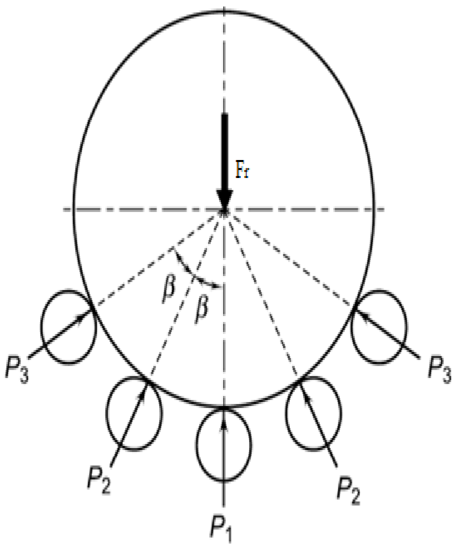

Fig.2 Distribution of forces

Stribeck's equation is given by :

$$Fr = \frac{Total\ no.of\ barrels \times P1}{Stribeck's\ factor}$$

Where Stribeck's factor = 5.

P1 ,P2 ,P3 …. 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

P1=3625 N (maximum radial load acted on barrel)

$$P2 = P1(cos18)^{(3/2)}$$
$$P2 = 3625(cos18)^{(3/2)}$$
=3362.15 N

$$P3 = 3625(cos36)^{(3/2)}$$
= 2637.18 N

$$P4 = P1(cos54)^{(3/2)}$$
$$P4 = 3625(cos54)^{(3/2)}$$
= 1633.56 N

$$P5 = P1(cos72)^{(3/2)}$$
$$P5 = 3625(cos72)^{(3/2)}$$
=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

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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

$$\sigma_x = C_\sigma(b|\Delta),$$
$$\sigma_y = 2\vartheta C_\sigma(b|\Delta)$$
$$\sigma_z = C_\sigma(b|\Delta)$$

Maximum principle stress is given by following

$$\sigma_{max} = C_\sigma(b|\Delta)$$

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

$$b = \sqrt{2q\Delta/\pi}$$
$$\Delta = \frac{1}{1/2(R_1) + 1/2(R_2)}\left(\frac{1-\vartheta^2}{\varepsilon_1} + \frac{1-\vartheta^2}{\varepsilon_2}\right)$$

By putting values

Maximum contact stress,

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

E. Contact stress due to Torque:

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

Tangential load = f_t =80357.14 N

Contact area is given by

$$A_s = \pi \times D1 \times l$$

Contact area for 20 barrels=20× A

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

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

=[(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 P1 obtained from stribeck's equation.Now we obtained further contact stress value to radial load P2,P3,P4 ,P5 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.

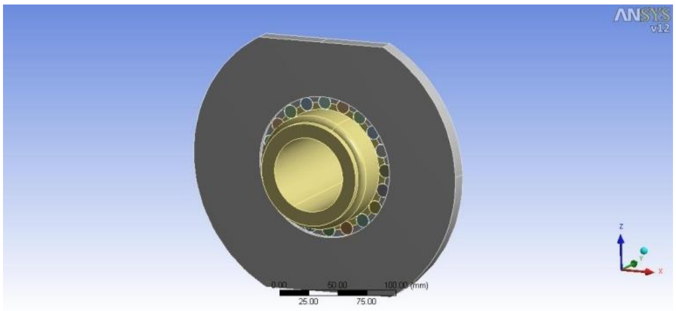

Fig.3 Creation of Geometry of barrel coupling

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 shafts, 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.

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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 Targe 170 as target element is applied.

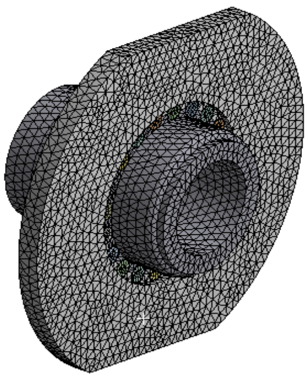

Fig.4.Meshing

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 stribeck's equation and applied to barrels in lower half portion of coupling.

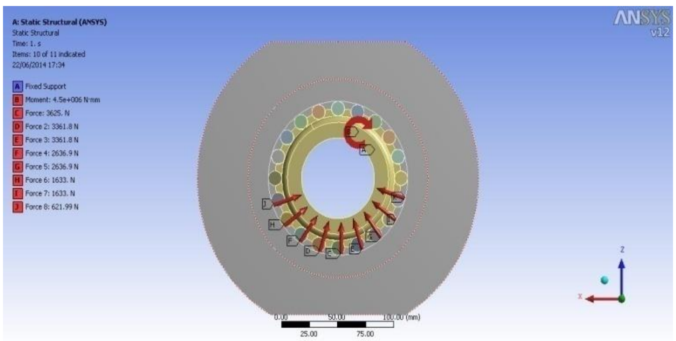

Fig.5 Application of boundary and loads

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.

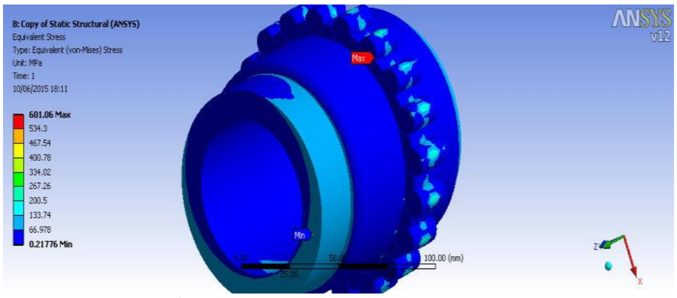

fig.6 contact stress on barrel coupling

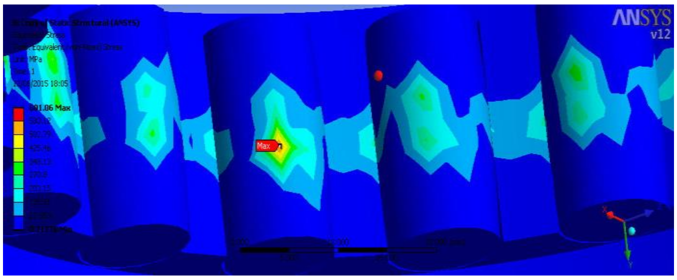

Fig.7 contact stress on barrels

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 rectangular 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 augumented langrange.after that we have changed contact algorithm pure penalty and langrane 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,augumented 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 in finite element analysis.

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Results obtained are compared to Hertz analytical method. Results are shown in Table I and figure 8

TABLE I
COMPARISON BETWEEN CONTACT ALGORITHMS WITH FRICTIONAL CONTACT IN FEM

Radial Load in N	**Contact stress(Mpa) by Augumented Langrange**	**Contact stress(Mpa) by Pure penalty**	**Contact stress(Mpa) by Langrange Multiplier**	**Hertz Analytical Method**
3625	601.06	601.06	531.04	650.8
3361.8	580.12	580.12	472.07	627
2636.9	502.79	502.79	413.1	555.3
1633	425.46	425.46	354.12	437
621.99	270.8	270.8	236.18	269.8

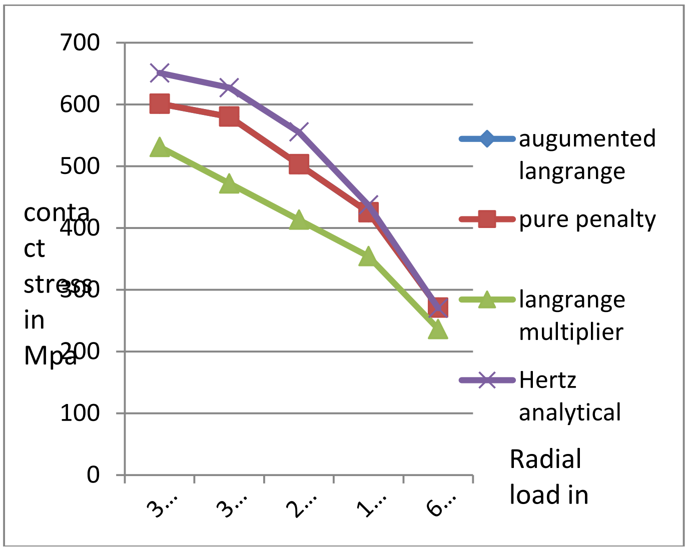

Fig.8 Comparison between contact algorithms with frictional contact in FEM

Results shown in table I revealed that contact stress obtained from contact algorithm pure penalty and augumented 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 augumented 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 augumented 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.

TABLE III
COMPARISON BETWEEN CONTACT ALGORITHMS WITH BONDED CONTACT IN FEM

Radial Load in N	**Contact stress(Mpa) by Augumented Langrange**	**Contact stress(Mpa) by Pure penalty**	**Contact stress(Mpa) by Langrange Multiplier**	**Hertz Analytical Method**
3625	109.72	109.72	109.72	650.8
3361.8	97.533	97.533	97.533	627
2636.9	85.342	85.342	85.342	555.3
1633	73.351	73.351	73.351	437
621.99	60.96	60.96	60.96	269.8

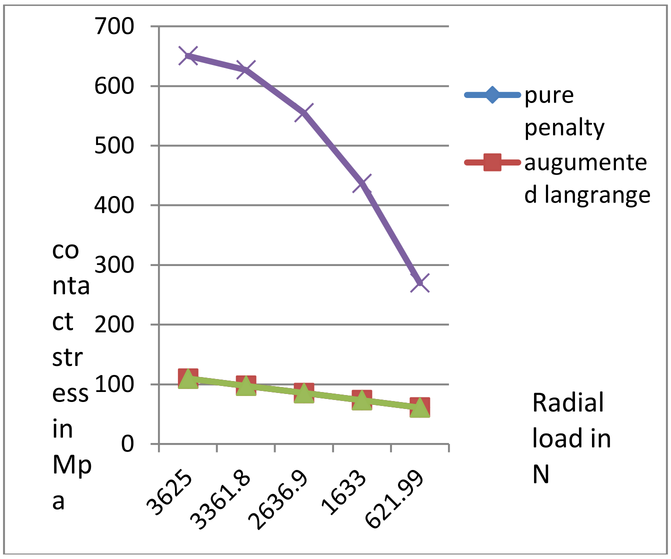

Fig.9 Comparison between contact algorithms with bonded contact in FEM

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 seperation 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.

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TABLE IIIII
COMPARISON BETWEEN CONTACT ALGORITHMS WITH NO SEPERATION CONTACT IN FEM

Radial Load in N	Contact stress(Mpa) by Augumented Langrange	Contact stress(Mpa) by Pure penalty	Contact stress(Mpa) by Langrange Multiplier	Hertz Analytical Method
3625	490.78	490.78	732.81	650.8
3361.8	436.27	436.27	651.39	627
2636.9	381.74	381.74	569.37	555.3
1633	327.2	327.2	488.54	437
621.99	272.67	272.67	407.12	269.8

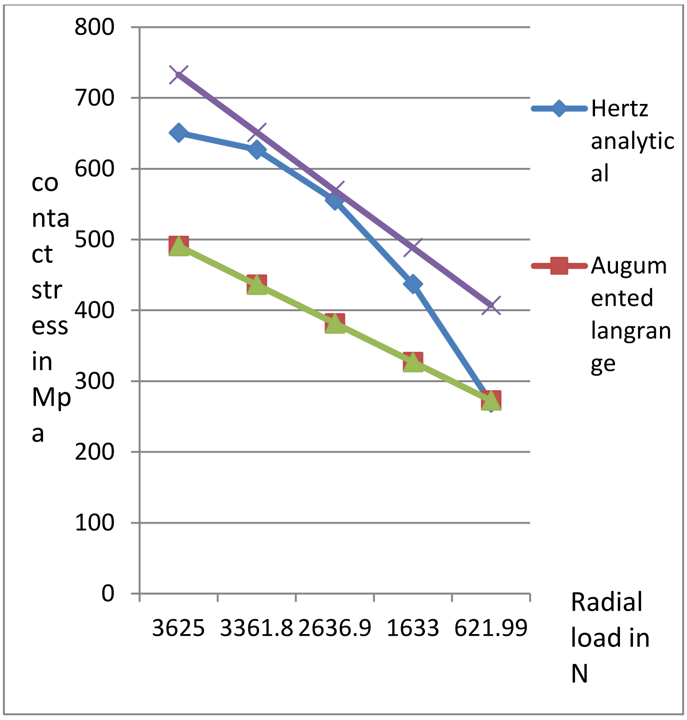

Fig.10 Comparison between contact algorithms with no seperation contact

Results in table III and Fig.10 revealed that pure penalty and augumented 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 augumented langrange gives better than langrange multiplier.

D. Comparison between contacts with augumented langrange algorithm

In this contact algorithm as augumented 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 AUGUMENTED LANGRANGE

Radial Load in N	Contact stress(Mpa) With frictional contact	Contact stress(Mpa) With bonded contact	Contact stress(Mpa) With no separation contact	Hertz analytical method
3625	601.06	109.72	490.8	650.8
3361.8	580.12	97.533	436.27	627
2636.9	502.79	85.342	381.74	555.3
1633	425.46	73.351	327.2	437
621.99	270.8	60.96	272.67	269.8

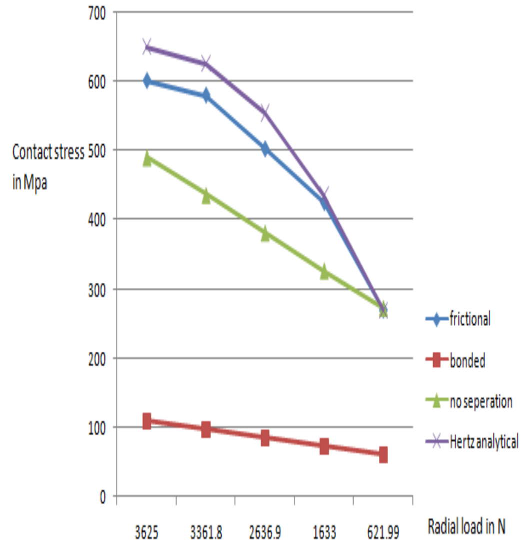

Fig.11 Comparison between contacts with augumented langrange algorithm

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 V
COMPARISON BETWEEN CONTACTS WITH PURE PENALTY

Radial Load in N	Contact stress(Mpa) With frictional contact	Contact stress(Mpa) With bonded contact	Contact stress(Mpa) With no separation contact	Hertz analytical method
3625	601.06	109.72	490.8	650.8
3361.8	580.12	97.533	436.27	627
2636.9	502.79	85.342	381.74	555.3
1633	425.46	73.351	327.2	437
621.99	270.8	60.96	272.67	269.8

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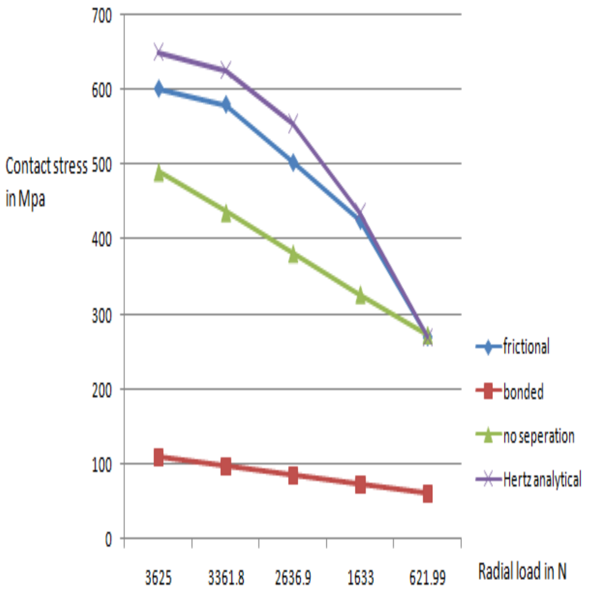

Fig.12 Comparison between contacts with pure penalty algorithm

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 VI
COMPARISON BETWEEN CONTACTS WITH LANGRANGE MULTIPLIER

Radial Load in N	Contact stress(Mpa) With frictional contact	Contact stress(Mpa) With bonded contact	Contact stress(Mpa) With no separation contact	Hertz analytical method
3625	531.04	109.72	732.81	650.8
3361.8	472.07	97.533	651.39	627
2636.9	413.1	85.342	569.37	555.3
1633	354.12	73.351	488.54	437
621.99	236.18	60.96	407.12	269.8

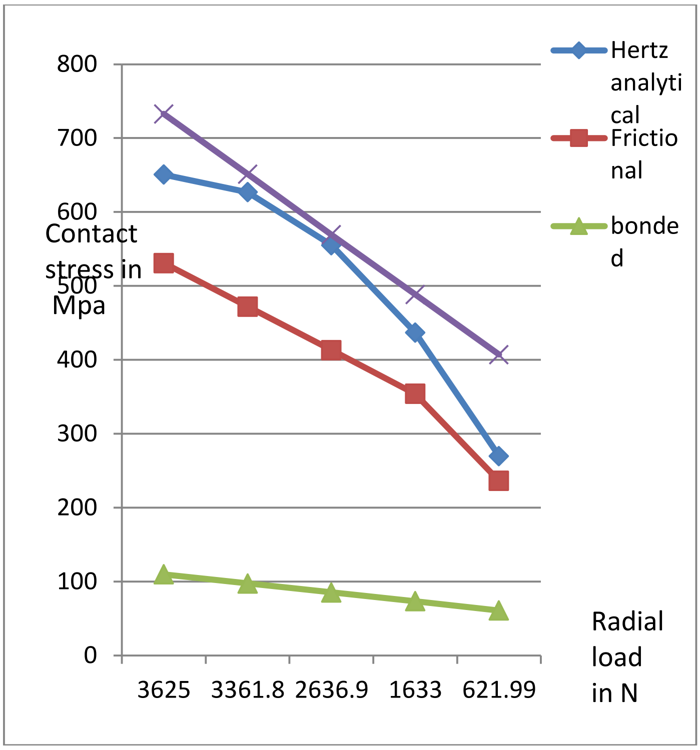

Fig.13 Comparison between contacts with Langrange Multiplier algorithm

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 depend on contacting area between barrel and sleeve, barrel and hub surfaces.
2. Contact stresses of barrel coupling are depending on pitch circle diameter of sleeve and hub, diameter of barrel, material properties of coupling.
3. The results estimated for frictional contact with pure penalty and augmented langrange algorithm is close agreement with Hertz analytical method where as langrange multiplier predicts higher than 7% error.
4. Frictional contact method is most effective contact than no separation and bonded contact. Bonded contact does not show significant change with contact algorithm.

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