Chapter 10

FINITE ELEMENT ANALYSIS OF SPUR BEVEL GEARS WITH ASYMMETRIC PROFILE

10.1 INTRODUCTION

The aim of asymmetric tooth is to improve the performance of gears such as increasing the load capacity or reducing noise and vibration. The application of asymmetric tooth side surfaces is to increase the load capacity and durability for the drive tooth side [134]. The tooth form has left-right symmetry in the involute cylindrical gear, and the same performance can be obtained at forward and backward rotation. However, both the forward and backward rotations are not always expected in the gear units for power transmission. Therefore, two sides of the gear tooth are functionally different for most gears. Even if one side (drive side) is significantly loaded for longer periods, the opposite side (coast side) is unloaded or slightly loaded for short duration only [135-136]. In several papers [136-139], the lower pressure angle profile for the drive side and higher pressure angle profile for the coast side have been considered. This kind of application makes it possible for the gear to reduce the bending stress. In this study, asymmetric spur bevel gear with lower drive side pressure angle than coast side pressure angle are taken into consideration. The purpose of this study is to determine bending load carrying capacity and its dynamic characteristics.

10.2 ASYMMETRICAL BEVEL GEAR GEOMETRY

As the asymmetric bevel gear consists of different pressure angles, thus the pinion and the gear are provided with two side profiles with base circles of different diameters. For a gear mechanism with asymmetric bevel spur gears, the following equations as in conventional gears can be outlined. The radius of the pitch circle for pinion and gear:

\[ r_{pi} = \frac{Z_i m_n}{2} \]  

where, \( m_n \) is the normal module. \( Z_i \) is the no. of teeth for \( i = 1 \), for the pinion wheel and \( i = 2 \) for the gear.

The base circle radii for pinion and gear can be obtained as:
where, \( \alpha_j \) is the pressure angle, \( i = c, d \) represents the coast side and the drive side, \( j = 1, 2 \) represents the pinion and the gear respectively.

The radius of the gear at critical section is given by:

\[
r_{cij} = r_{bij} + 0.157m
\]

(10.3)

### 10.2.1 Analytical Representation of Involute Profile

From Fig. 10.1, an involute curve has been generated from the base circle of radius \( r_b \). A and B are two points on this involute at a distance \( r_A \) and \( r_B \) respectively from base circle centre ‘O’. From the geometry, we get the following relations,

\[
r_b = r_A \cos \alpha_A
\]

(10.4)

\[
r_b = r_B \cos \alpha_B
\]

(10.5)

From equation 10.4 and 10.5

\[
\begin{align*}
    r_A \cos \alpha_A &= r_B \cos \alpha_B \\
    \cos \alpha_B &= \frac{r_A}{r_B} \cos \alpha_A
\end{align*}
\]

(10.6)

From equation 10.6, the pressure angle at any point on the involute in relation to the parameters of another known point.

Fig. 10.3 a and 10.3 b, shows a gear tooth enclosed between the tip circle and the base circle. From the properties of the involute it could be determined,

\[
\text{Arc DE} = \text{Straight length DB}
\]

\[
\text{Arc CE} = \text{Straight length CA}, \text{ Also}
\]

\[
\text{Angle EOD (in radian)} = \frac{\text{Arc DE}}{r_B} = \frac{DB}{r_B} = \tan \alpha_B
\]

(10.7)

Similarly

\[
\text{Angle EOD (in radian)} = \frac{\text{Arc CE}}{r_A} = \frac{CA}{r_A} = \tan \alpha_A
\]

(10.8)

Now,

\[
\text{Angle EOB} = \text{Angle EOD} - \alpha_B \text{ or Angle EOB} = \tan \alpha_B - \alpha_B
\]

(10.9)

\[
\text{Angle EO} = \tan \alpha_A - \alpha_A
\]

(10.10)
With reference to Fig. 10.1, the involute angle has been derived. $\phi$ is the pressure angle at 'P and $\theta$ is the roll angle at P.

$$\tan \alpha = \frac{R}{r_b}$$

$$\text{inv}\alpha = \theta - \alpha = \frac{\text{Arc}AB}{r_b} - \alpha$$

$$\text{inv}\alpha = \frac{EP}{OB} - \alpha = \frac{R}{r_b} - \alpha = \tan\alpha - \alpha$$

$$\text{inv}\alpha = \tan\alpha - \alpha \quad (10.11)$$

Now, with reference to equation 11 equation 9 and 10 can written as,

$$\text{Angle } EOB = \text{inv}(\alpha_b) \quad (10.12)$$

$$\text{Angle } EOA = \text{inv}(\alpha_A) \quad (10.13)$$

From Fig. 10.3b $\text{Angle } EOF = \text{Angle } EOB + \text{Angle } BOF \quad (10.14)$

The centreline through point F divides the tooth into equal and symmetric halves, and hence the circular tooth thickness, $s_B$ at the circle passing through B can be written as:

$$\frac{s_B}{2} = r_b \times \text{Angle } BOF \text{ and similarly } \frac{s_A}{2} = r_A \times \text{Angle } AOF$$

From equation 10.14,
\[ Angle \ EOF = Angle \ EOB + Angle \ BOF \]

\[ Angle \ EOF = \text{inv}(\alpha_B) + \frac{s_B}{2r_B} \] and also \[ Angle \ EOF = \text{inv}(\alpha_A) + \frac{s_A}{2r_A} \]

Now by equating both sides the relation could be established as,

\[ \text{inv}(\alpha_B) + \frac{s_B}{2r_B} = \text{inv}(\alpha_A) + \frac{s_A}{2r_A} \] and hence

\[ \frac{s_B}{2} = r_B \left[ \frac{s_A}{2r_A} + \text{inv}(\alpha_A) - \text{inv}(\alpha_B) \right] \] \hspace{1cm} (10.15 a)

For a standard tooth, tooth thickness at pitch circle could be written as

\[ S = \frac{\text{Circular \ pitch}}{2} = \frac{p_c}{2} \]

Also Circular pitch could be written as \[ p_c = \pi m \], hence \[ S = \frac{\pi m}{2} \]

And the module \( m \) could be written as \[ m = \frac{d}{Z} \]

Where, \( d = \text{Diameter of pitch circle} \)

\( Z = \text{Number of teeth} \)

If \( r, r_a \) are the pitch circle radius and addendum circle radius respectively and \( \alpha_a \) is the pressure angle at addendum and \( \alpha \) is the pressure angle at pitch circle then as per equation 10.6 the below relation can be written

\[ \cos \alpha_A = \frac{r}{r_a} \cos \alpha \]

\[ \frac{s_a}{2} = r_a \left[ \frac{\pi}{2Z} + \text{inv}(\alpha) - \text{inv}(\alpha_a) \right] \] \hspace{1cm} (10.15)

The above equation can be rewritten in general form as below,

\[ s_k = 2r_k \left[ \frac{\pi}{2Z} + \text{inv}(\alpha) - \text{inv}(\alpha_k) \right] \] \hspace{1cm} (10.16)

Where \( \alpha_k \) is the pressure angle at radius \( r_k \).

**10.2.2 Derivation of tooth thickness for asymmetric case**

In asymmetric teeth, it has two sides namely the coast side and drive side as shown in Fig. 10.2. The tooth thickness at addendum could be written as,

\[ s_a = \frac{s_{ac}}{2} + \frac{s_{ad}}{2} \] \hspace{1cm} (10.17)
Fig. 10.2 Asymmetric tooth with coast side and drive side

Now the pitch circle radius and pressure angle at pitch circle on drive side is designated as \( r_d, \alpha_d \) and similarly the pitch circle radius and pressure angle at pitch circle on coast side is designated as \( r_c, \alpha_c \). The addendum circle radius and pressure angle at addendum circle on drive side is designated as \( r_{ad}, \alpha_{ad} \) and the addendum circle radius and pressure angle at addendum circle on coast side is designated as \( r_{ac}, \alpha_{ac} \). But in general the pitch circle radius and addendum circle radius will be the same on both sides and hence we can designate the respective radius as \( r_d = r_c = r \) and \( r_{ad} = r_{ac} = r_a \).

Now considering the drive side, the involute thickness at addendum could be written as,

\[
\frac{S_{ad}}{2} = r_a \left[ \frac{\pi}{2Z} + \text{inv} \alpha_d - \text{inv} \alpha_{ad} \right]
\]

And for coast side the involute thickness at addendum could be written as,

\[
\frac{S_{ac}}{2} = r_a \left[ \frac{\pi}{2Z} + \text{inv} \alpha_c - \text{inv} \alpha_{ac} \right]
\]

And hence the total tooth thickness at addendum could be written as by considering equation 10.17,

\[
s_a = r_a \left[ \frac{\pi}{2Z} + (\text{inv} \alpha_c + \text{inv} \alpha_d) - (\text{inv} \alpha_{ac} + \text{inv} \alpha_{ad}) \right]
\]  \( (10.18) \)

Based on equation 10.18, the thickness of the involute profile at any point as shown in Fig. 10.2 could be written as:
\[ s_i = r_i \left( \frac{\pi}{Z_i} + \left( \text{inv}(\alpha_c) + \text{inv}(\alpha_d) \right) - \left( \text{inv}(\alpha_{c_{ji}}) + \text{inv}(\alpha_{d_{ji}}) \right) \right) \]  \hspace{1cm} (10.19)

Where \( \alpha_{c_{ji}} = \cos^{-1} \left( \frac{r_{p_c}}{r_i} \cos \alpha_c \right) \) and \( \alpha_{d_{ji}} = \cos^{-1} \left( \frac{r_{p_d}}{r_i} \cos \alpha_d \right) \)  \hspace{1cm} (10.20)

Also, the involute angle could be written as with reference to equation 10.11,

\[ \text{inv}(\alpha) = \left( \tan \alpha - \frac{\pi \alpha}{180} \right) \]  \hspace{1cm} (10.21)

The contact ratio of the gears on the drive side is:

\[ \varepsilon_d = \frac{\sqrt{r_{a1}^2 - r_{b1}^2} + \sqrt{r_{a2}^2 - r_{b2}^2} - a_d \sin \alpha_d}{p_o} \]  \hspace{1cm} (10.22)

10.3. MODIFICATION OF TOOTH GEOMETRY FOR ASYMMETRIC SPUR BEVEL GEAR

For a particular pressure angle on the coast side and drive side of an asymmetric spur bevel gear, the involute thickness of the teeth depends on radius of addendum and dedendum circle. Thus the thickness of the involute profile of asymmetric spur bevel gear along the face width portion at any point can be obtained using the concept of similar triangles as shown in Fig. 10.4. From Fig. 10.4, the radius of the addendum, pitch and dedendum circle at any point can be written as:

Fig.10.3a Coast side of asymmetric gear  \hspace{0.5cm} Fig.10.3b Coast and drive sides of asymmetric gear
Using this Eq. (10.23) and substituting in Eq. (10.19), the equation for asymmetric tooth thickness at any point along the face width of the bevel gear for addendum, dedendum and pitch circles can be obtained by Eq. (10.24)

\[
R_{ji} = r_{ji} \left(1 - \frac{b_i}{B_j}\right)
\]  

(10.23)

where \(i=1, 2\) (i=1 for pinion and i=2 for gear) \(j = A, P, D\) (A- Addendum circle, P- Pitch circle, D- Dedendum circle) and \(b =\) Face width, \(B =\)cone distance of the bevel gear. The thickness at critical section as shown in Fig. 10.3, along the direction of face width is given by:

\[
S_{ji} = r_{ji} \left(1 - \frac{b_i}{B_j}\right) \left(\frac{\pi}{Z_i} + \text{inv}(\alpha_c) + \text{inv}(\alpha_d)\right) - \left(\text{inv}(\alpha_{ji}) + \text{inv}(\alpha_{jd})\right)
\]  

(10.24)

Fig. 10.4 Modification of tooth geometry

Fig. 10.5 Thickness at critical section

\[
S_{ji} = \frac{S_{fn1} + S_{fn2}}{2}
\]  

(10.25)

Where \(S_{fn1}\) and \(S_{fn2}\) were calculated as follows:

\[
\frac{S_{fn1}}{2} = r_{ci} \left(1 - \frac{b_i}{B_i}\right) \left(\frac{\pi}{2Z_i} + \text{inv}(\alpha_c) - \text{inv}(\alpha_d)\right)
\]  

(10.26)
\[
\frac{S_{fn}}{2} = r_{ci} \left( 1 - \frac{b_i}{B_i} \right) \left( \frac{\pi}{2Z_i} \right) + \left( \text{inv}(\alpha_{ci}) - \text{inv}(\alpha_{di}) \right)
\]

(10.27)

where \( i = 1 \) for pinion and 2 for gear respectively.

### 10.4 BENDING STRENGTH OF GEAR TOOTH

The bending strength of gear tooth is given by:

\[
\sigma_b = \frac{F_t}{b.m} Y \cdot K \cdot Y Y \beta
\]

(10.28)

Where, Lewis Form Factor: \( Y = \frac{6h_{fa}jm}{(S_{fn})^2} \)

(10.29)

Load sharing factor or contact factor: \( Y_e = 0.25 + \frac{0.75}{\varepsilon_d} \)

(10.30)

and Stress Concentration Factor: \( K_c = \left[ 1.2 + 1.3 \left( \frac{S_{fn}}{h_{fa}} \right) \right] \left( \frac{S_{fn}}{2r_{cs}} \right)^{1.2 + 2.3 \left( \frac{h_{fa}}{S_{fn}} \right)} \)

(10.31)

\( Y_\beta \) is the contact ratio factor and usually its value is equal to 1. \( Y_{Fa} \) and \( Y_e \) factors depend on tooth thickness at critical section, the distance \( h_{fa} \) from critical section to intersection of the tooth centerline and the line of action for load at tip of tooth, and the curvature radius \( r_{cs} \) of the root trochoid whereas \( Y_e \) only is related with contact ratio \( \varepsilon_\alpha \). In brief, to calculate these factors, \( S_{fn}, h_{fa} \) and \( \varepsilon_\alpha \) should be determined for each tooth number of pinion and pressure angle.

### 10.5 GEOMETRY DETAILS OF PINION

Using the above said formulae, the geometry details of bevel gear pinion wheel with 25 no. of teeth and a normal module of 2.5, for various pressure angles are listed in Table. 10.1
Table 10.1 Dimensions of Pinion

<table>
<thead>
<tr>
<th>$\alpha_c - \alpha_d$</th>
<th>$r_{a1}$ (mm)</th>
<th>$r_{p1}$ (mm)</th>
<th>$r_{c1}$ (mm)</th>
<th>$r_{b1}$ (mm)</th>
<th>$S_{a1}$ (mm)</th>
<th>$S_{p1}$ (mm)</th>
<th>$S_{c1}$ (mm)</th>
<th>$h_{fa1}$ (mm)</th>
<th>$\varepsilon_d$</th>
</tr>
</thead>
<tbody>
<tr>
<td>20 – 20</td>
<td>33.75</td>
<td>31.25</td>
<td>29.757</td>
<td>29.365</td>
<td>2.5</td>
<td>3.926</td>
<td>3.739</td>
<td>1.493</td>
<td>1.649</td>
</tr>
<tr>
<td>20 – 30</td>
<td>31.25</td>
<td>33.75</td>
<td>28.236</td>
<td>27.843</td>
<td>1.463</td>
<td>3.926</td>
<td>5.081</td>
<td>3.013</td>
<td>1.416</td>
</tr>
</tbody>
</table>

10.6 BENDING STRESS ANALYSIS USING ANSYS

Using the dimensions from Table 10.1, the model is meshed using 3-D 20-Node Structural Solid and Fig. 10.6 shows the meshed model with applied boundary conditions. The load variation along the face width of the bevel gear tooth is assumed to be trapezoidal in nature with a load of 600 N at the big end and 420 N at the small end [140]. The material constants chosen are Young’s modulus 2.1E+5 N/mm$^2$ and Poisson’s Ratio of 0.33.

Fig. 10.6 Meshed model of Asymmetric Bevel Gear
The Von Misses stress plots for various symmetric spur bevel gear obtained using ANSYS software were shown in Fig. 10.7.

Similarly, Von Misses stress plots for asymmetric spur bevel gear with different pressures angles are given in Fig. 10.8.
10.7 CONTACT STRESS ANALYSES

Contact stress analysis was carried out for two cases of design; higher pressure angle on drive side and Lower pressure angle on drive side for asymmetric bevel gear drives, and stresses were compared in both cases. The analysis results presented are obtained by using the general purpose finite element software ANSYS/Standard 10.7. For initial meshing SOLID186 element was used, TARGE170 was used as target element and CONTA173 was used as Contact element. Contact stress analysis was performed with higher pressure angle on drive side in case 1 and lower pressure angle on drive side in case 2. The finite element model used for contact analysis is shown in Fig. 10.9 and the various elements used for contact analysis are shown in Fig. 10.10 respectively.

Fig. 10.8 Von Misses Stress Plots of different pressure angles for asymmetric spur bevel gear

Fig. 10.9 Meshed Model for contact analysis

Fig. 10.10 Contact and Target elements
The finite element results obtained for asymmetric bevel gear with higher pressure angle on the drive side using ANSYS are shown in Fig. 10.11 and Fig. 10.12 depicts the finite element results for asymmetric bevel gear having lower pressure angle on the drive side respectively.

**10.7.1 Contact stress analysis with higher pressure angle on drive side**

![Image of contact stress plots for asymmetric spur bevel gear (Higher pressure angle on drive side)]

*Fig. 10.11 Contact stress Plots for asymmetric spur bevel gear (Higher pressure angle on drive side)*

**20°-27°**

**20°-30°**

**10.7.2 Contact stress analysis with lower pressure angle on drive side**

![Image of contact stress plots for asymmetric spur bevel gear (Lower pressure angle on drive side)]

*Fig. 10.12 Contact stress plots for asymmetric spur bevel gear (Lower pressure angle on drive side)*

*27°-20°*  

*30°-20°*

From the finite element results it was found that asymmetric bevel gears has lower values of contact stress with higher pressure angle on drive side than that of the coast side. The
location of maximum contact stress is far above the fillet region and the location of the maximum stress value was not found changing. In the case of higher pressure angle on drive side and lower pressure angle on drive side, it was found that higher value of stresses occurred on the opposite side of the contact. With reduced contact stress in case of asymmetric profile, the vibration level may be less which has been illustrated in the dynamic analysis study.

Table 10.2 Comparison of contact stress (ANSYS results) for asymmetric profile with different pressure angle on drive side.

<table>
<thead>
<tr>
<th>(\alpha_c - \alpha_d) (Deg)</th>
<th>Higher Pressure Angle on Drive side</th>
<th>Contact stress (kN/mm²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>20-27</td>
<td></td>
<td>3061</td>
</tr>
<tr>
<td>20-30</td>
<td></td>
<td>2164</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>(\alpha_c - \alpha_d) (Deg)</th>
<th>Higher Pressure Angle on Coast side</th>
<th>Contact stress (kN/mm²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>27-20</td>
<td></td>
<td>3280</td>
</tr>
<tr>
<td>30-20</td>
<td></td>
<td>2793</td>
</tr>
</tbody>
</table>

Also another interesting point that could be observed is that the contact was localized in case of asymmetric profile. Table 10.2 gives a summary of contact stress results for various cases of study.

10.8 MODAL ANALYSES
Modal analysis was performed to determine the dynamic behavior of asymmetric bevel gear with that of symmetric bevel gear. The analysis results presented are obtained using the finite element method. These finite element results are obtained by using the general purpose finite element software ANSYS/Standard 10.7. The first five fundamental frequencies were extracted for different profiles of bevel gear and the results are shown in
various plots below. The finite element model used for modal analysis of asymmetric bevel gear for pressure angle 20 and 27 has been shown in Fig. 10.13 and the first five fundamental frequencies and mode shapes is shown in Fig. 10.14 respectively.

Fig. 10.13 Finite Element Model for asymmetric bevel gear (20 -27)

First mode plot for 20 -27

Second mode plot for 20 -27

Third mode plot for 20 -27

Fourth mode plot for 20 -27

Fifth mode plot for 20 -27

Fig. 10.14 various modal plots for asymmetric bevel gear (20 -27)
The finite element model used for modal analysis of asymmetric bevel gear for pressure angle 20 and 30 is shown in Fig. 10.15 and the first five fundamental frequencies and mode shapes is shown in Fig. 10.16 respectively.

Fig. 10.15 Finite Element Model for asymmetric bevel gear (20 -30)

First mode plot for 20 -30

Second mode plot for 20 -30

Third mode plot for 20 -30

Fourth mode plot for 20 -30

Fifth mode plot for 20 -30

Fig. 10.16 Various modal plots for asymmetric bevel gear (20 -30)
The finite element model used for modal analysis of symmetric bevel gear for pressure angle 20 and 20 is shown in Fig. 10.17 and the first five fundamental frequencies and mode shapes has been shown in Fig. 10.18 respectively.

Fig. 10.17 Finite Element Model for symmetric bevel gear (20 -20)

First mode plot for 20 -20

Second mode plot for 20 -20

Third mode plot for 20 -20

Fourth mode plot for 20 -20

Fifth mode plot for 20 -20

Fig.10.18 Various modal plots for symmetric bevel gear (20 -20)
These plots are of different modes for symmetric and asymmetric bevel gear. The first mode is predominantly bending mode, the second and other three modes exhibit a coupling between bending and torsion.

Table 10.3 Comparison of natural frequencies for symmetric and asymmetric profile with different pressure angle on drive side.

<table>
<thead>
<tr>
<th>PRESSURE ANGLE (deg)</th>
<th>20-20</th>
<th>20-27</th>
<th>20-30</th>
</tr>
</thead>
<tbody>
<tr>
<td>MODE 1</td>
<td>3607</td>
<td>4037</td>
<td>4119</td>
</tr>
<tr>
<td>MODE 2</td>
<td>3980</td>
<td>4297</td>
<td>4368</td>
</tr>
<tr>
<td>MODE 3</td>
<td>4257</td>
<td>4980</td>
<td>5085</td>
</tr>
<tr>
<td>MODE 4</td>
<td>4790</td>
<td>5593</td>
<td>5721</td>
</tr>
<tr>
<td>MODE 5</td>
<td>5556</td>
<td>6140</td>
<td>6270</td>
</tr>
</tbody>
</table>

From Table 10.3, it can be summed up that the asymmetric bevel gear exhibits better dynamic characteristics as it has high natural frequencies compared to that of symmetric bevel gear. In case of asymmetric profile itself, higher pressure angle on drive side has better results compared to lower pressure angle on drive side.

10.9 HARMONIC ANALYSIS

The load variation along the face width of the bevel gear tooth is assumed to be trapezoidal in nature with a load of 600 N at the big end and 420 N at the small end [147]. These finite element results were obtained by using the general purpose finite element software ANSYS/Standard 10.7. The frequency versus displacement plots for various case of symmetric and asymmetric bevel gear are shown in Fig. 10.19.
From Fig. 10.19, it can be concluded that asymmetric bevel gear has better dynamic characteristics as they have resonant frequencies at higher ranges compared to that of symmetric bevel gear. There are no peaks at lower frequencies in case of asymmetric
bevel gear. Whereas, symmetric bevel gears has peaks contributing to resonant conditions at lower frequency ranges. In the case of 20-20 pressure angle in symmetric bevel gear it has may significant peaks in the lower range of frequency which is not a good feature for any rotating component. In the case of 25-25 pressure angle in symmetric bevel gear it has resonant condition at much lower frequency compared to 20-20 pressure angle. In case of asymmetric bevel gear even though 20-27 pressure angle, it has only few peaks, the disturbing resonant condition appear at much higher frequency range and in the case of 20-30 pressure angle, it has a very smooth functioning at lower frequencies and has one distinct resonant condition at higher frequency range. It can be summarized as that asymmetric bevel gear with higher pressure angle on drive side has better dynamic performance than that of symmetric bevel gear. The asymmetric involute tooth can be manufactured by the same process as in generating the symmetric involute tooth. Asymmetric profile is achievable by adopting $\alpha_c$ and $\alpha_d$ values for the profiles of two sides of the rack. Depending on the special tooling, production cost of these gears increases. Therefore, the gears with asymmetric teeth should be considered for gear systems that require extreme performance like aerospace applications and for mass production transmissions where the share of the tooling cost per one gear is insignificant. The most promising application of asymmetric profiles seems to be in molded gears and powder gears.

10.10 SUMMARY

Finite element analysis of asymmetric involute bevel gear for different drive side and coast side pressure angles was carried out. The bending stress obtained using the FEM analyses is comparable with bending stress calculated using analytical equations. From the results, it can be concluded that involute bevel gears with asymmetric profiles, have better performance in load carrying capacity when compared to symmetric bevel gear. Contact analysis to determine the contact stress for asymmetric bevel gear was performed and the results showed that asymmetric bevel gear with higher pressure angle on drive side performs better. In order to study the dynamic characteristics of asymmetric bevel gear, modal analysis and forced frequency analysis were performed. From the results, it can be inferred that the asymmetric bevel gear has higher fundamental
frequency than that of symmetric bevel gear. Asymmetric bevel gear with higher pressure angle on drive side has higher range of natural frequency, which is a welcome feature. From the results of the forced frequency analysis, asymmetric bevel gear has better dynamic characteristics than that of symmetric bevel gear. Symmetric bevel gears have disturbing peaks at lower operating frequencies, whereas asymmetric bevel gear has disturbing peaks at higher ranges of operating frequencies. Asymmetric bevel gear with higher pressure angle on drive side has better dynamic characteristics. Overall, it can be concluded that asymmetric bevel gear with higher pressure angle on drive side has better load carrying capacity, reduced contact stress, better dynamic characteristics.