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Effect of Coefficient of Asymmetry on Strength and Contact Ratio of Asymmetric Helical Gear

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ABSTRACT

A gear is a rotating machine part having cut teeth, which mesh with another toothed part in order to transmit torque. Gears may be spur, helical, bevel or worm in which Helical Gear is most common type of gear used in engineering applications. The increased in performance requirement such as high load carrying capacity, high speed, high reliability and long life leads to new design of gear. So as to fulfill above demand here we used a concept of asymmetric gear profile in which two different pressure angles are assign to two faces of gear tooth. In literature it shows that increased in drive side pressure angle shows improve in strength of spur gear. In this paper we find the effect of coefficient of asymmetry on strength and contact ratio of different helical gear pairs having gear ratio equal to one and transmit same power. All the gear pair are design in Cre-O parametric 2.0 and the analysis is carried out in ANSYS workbench 16.0. Results from numerical analysis shows that as coefficient of asymmetry and helix angle increases Von-Mises stress and total deformation decreases. To validate the results from numerical analysis experiment stress measurement is done using digital strain indicator in which a strain gauge (UFLA-1-350-23) is mounted at the root of the gear and strain is calculate for different loads. In further study we find the effect of coefficient of asymmetry on contact ratio of helical gear pairs, here it shows that decreased in coefficient of asymmetry leads to increased contact ratio.

Keywords: Asymmetry, Contact Ratio, FEA, Helix Angle, Pressure Angle

I. INTRODUCTION

Gearing is one of the most critical components in a mechanical power transmission system. Gears are used in all applications where power transfer is required such as automobiles, wind mills, industrial equipments, air planes and marine vessels. Most of the gears used in power transmission system are spur and helical gears. Sufficiency in load carrying capacity is a serious problem. There are different ways to improve strength of gears such as replacing trochoidal fillet by circular fillet at the root of a gear [1], heat treatments, improving surface quality, using composite materials and using a cutter with large tip radius.

In some applications like wind mills, the gears experiences unidirectional loading. In such cases one side of gear subjected to high stresses and deformation and this may cause early failure. This leads to designing of asymmetric teeth. Asymmetric teeth are one in which drive and coast side have different pressure angle. Recently, the involute spur gears with asymmetric teeth have been found in applications requiring high performance. These gears, due to their asymmetric tooth profile, allow for optimal design in various applications. Due to their geometry, these gears allow for the selection of different pressure angles on the drive side and the coast side, which is absolutely necessary in obtaining key properties, such as high load-carrying capacity and minimum weight. In literature, two configurations of the involute spur gears with asymmetric teeth can be found one has the pressure angle on the drive side is higher than the coast side, and for another the pressure angle on the drive side is lower than the coast side. Gears with a larger pressure angle on the drive side compared to coast side have significant advantages. [2][3][4]

Several studies in literature have been conducted on the design and stress analysis of asymmetric gears. Kapelevich [5] proposed a method for the design of gears with asymmetric teeth. Several equations required for design of asymmetric gears were developed and presented for the synthesis of asymmetric gears. Moreover, this study included the results of the experimental study conducted on a planet gearbox of an airplane engine. In 2010 Neils Pedersen [1] shows in his work that how bending stresses can be reduced significantly by using asymmetric gear teeth and by shape optimizing the gear through changes made to the tool geometry, he also suggested the use of two new standard cutting tools. Santosh Patil et al [6] uses Lagrange multiplier algorithm between the contacting helical gear pairs of different helix angles to determine the stresses. In this study they also considered effect of friction at the point of contact which made the problem nonlinear. P. Marimuthu and G. Muthuveerappan [7] takes drive and coast side pressure angles, top land thickness coefficient, contact ratio, coefficient of asymmetry, gear ratio and teeth number as various parameters and study their influence on load sharing ratio, maximum fillet stress and maximum contact stress using Ansys Parametric Design Language code. Yamei Hua et al [8] studied transient meshing performance of gears using explicit dynamic analysis in ANSYS for different modification coefficient and helix angles. Fatih Karpat et al [9][10] developed a computer program for dynamic load simulation using MATLAB and used for the prediction of instantaneous dynamic loads of spur gears with symmetric and asymmetric teeth.

Motivation and Objectives

Involute spur gears with asymmetric teeth provide flexibility to designers for different application areas because of their non-standard design. If they are accurately designed, they can make important contributions to the improvement of designs in automobile, aerospace and wind turbine industry. This often relates to improving the performance, increasing the load capacity and reduction of vibration [1]. In the past, most of the analysis of gears with asymmetric teeth has been limited to spur gear. But since helical gears are mostly used in industrial application because of their quietness in operation, there is need to improve strength of helical gear. Since this study focuses on improved performance of helical gears by using asymmetric teeth.

II. METHODS AND MATERIAL

1. Geometric Design of Gears

As helical gear design is a time consuming process here we use computer aided design of symmetric as well as asymmetric gears. For the same we use creo parametric 2.0 CAD software. To design a gear with asymmetric teeth there are two methods available one is by direct gear design approach and another is by conventional design approach, here we use convention design method as it is simple and there is no need to develop any computer code [7]. To do geometric design of asymmetric gear following are some formulas need to be used:

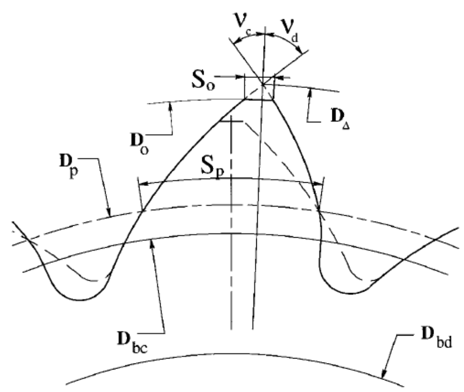

Figure 1. Formation of asymmetric teeth [5]

The top land thickness coefficient

$$m_o = \frac{S_o}{D_{bd}} = \frac{\text{inv}\nu_d + \text{inv}\nu_c - \text{inv}\varphi_{od} - \text{inv}\varphi_{oc}}{2\cos\varphi_{od}} \tag{1}$$

Where,
S_o = Top land thickness
D_{bd} = Drive side base circle diameter
D_{bc} = Coast side base circle diameter
φ_{od} = Drive side profile angle on outside circle
φ_{oc} = Coast side profile angle on outside circle
inv(x) = tan(x) - x
The coefficient m_o is selected within (0.25-0.4)/N range
The coefficient of asymmetry

$$k = \frac{D_{bc}}{D_{bd}} = \frac{\cos\gamma_c}{\cos\gamma_d} = \frac{\cos\varphi_{bc}}{\cos\varphi_{bd}} \tag{2}$$

Several gear pairs are used to know the effect of helix angle and coefficient of asymmetry on strength and fatigue life are shown in TABLE I. These all gear pairs are model in creo parametric 2.0, one of those pair is shown in Fig. 2.

TABLE I
PARAMTERS OF HELICAL GEAR PAIRS

Parameters	Gear pairs			
Material	Structural steel			
Module (m)	4.5			
Number of teeth (N)	25			
Gear ratio (G)	1			
Top Land thickness coefficient (m_o)	0.4 / N			
Helix angle (β)	10	15	20	25
Face width (b)	6 m			
Fillet radius (Rf)	0.4 m			
Clearance (c)	0.25 m			
Drive side pressure angle (φd)	20	25	30	35
Coast side pressure angle (φc)	20			
Addendum (ha)	1m			
Dedendum (hd)	1.25m			

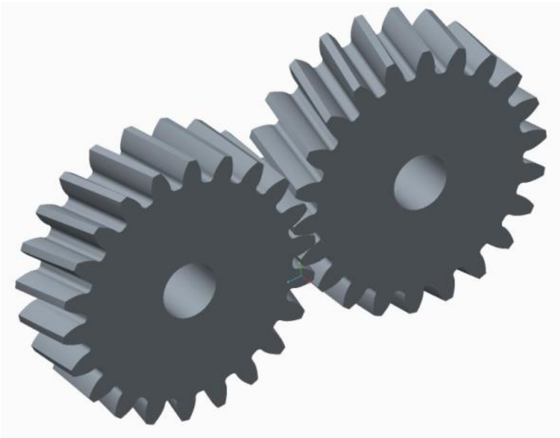

Figure 2. Helical gear model for β = 10, φd = 20, φc = 20

2. Finite Element Analysis Using Ansys 16.0

Numerical method made important contribution for solving complex computational mechanics problems, quickly as well as accurately. There are dozens of methods are available, generally used to solve variety of problems. Each method has their own advantages and disadvantages. Finite element method is one which uses differential equation and system boundary conditions to solve most of the problems related to mechanical, civil and aerospace.

All helical gear models drawn in creo parametric 2.0 are assembled and each final model in IGES format is imported in ANSYS 16.0 workbench. After importing a CAD model, the next step is to discretization of the gear pair into number of element and node. Here we used hexahedral mesh with 1mm elemental size and relevance centre is set to 40.

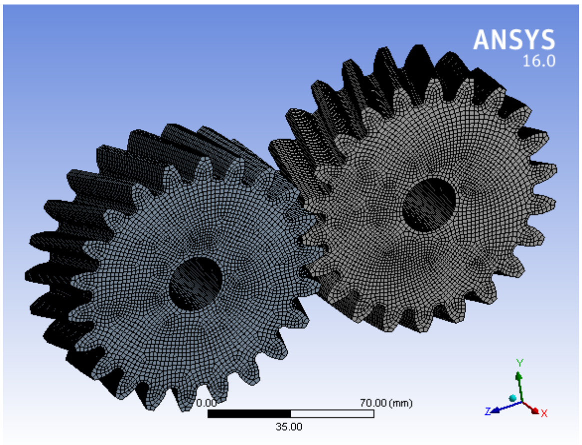

Figure 3. Meshing of helical gear pair (β = 15, φd = 20, φc = 20)

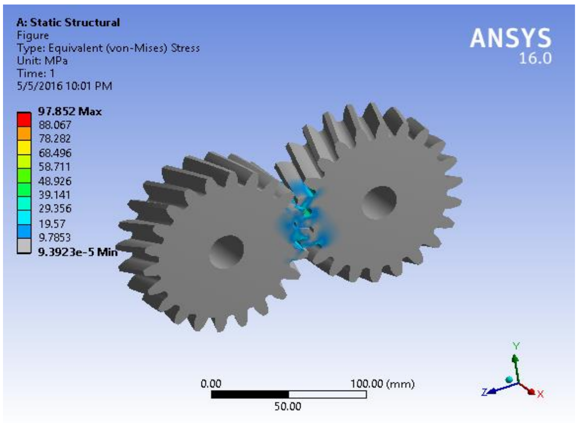

Figure 4. Equivalent von-mises stresses in helical gear (β = 15, φd = 20, φc = 20)

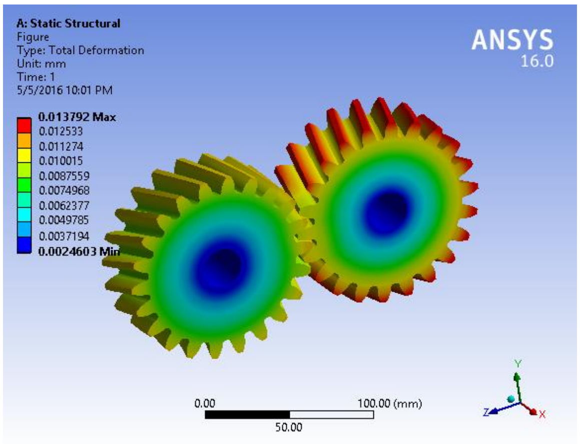

Figure 5. Total deformation in helical gear (β = 15, φd = 20, φc = 20)

Next we defined boundary conditions for a given gear pair. Gear is fixed by assigning a fixed support at inner periphery and a pinion has given cylindrical support with tangential free constraint. The contact between the gear pairs has been fixed to flexible-flexible contact using Conta 173 and Target 170 as elemental types. The Augmented Lagrangian algorithm with lower contact stiffness is used to get contact solution. All the gear pairs are used to transmit a power of 12kw and pinion is rotating at 1440 rpm. Pinion has given a moment of 79.5774Nm. Finally in post processing, solution is calculated for Equivalent (von-mises) stress and total deformation. All the gear pairs are analyzed for given loading and boundary conditions.

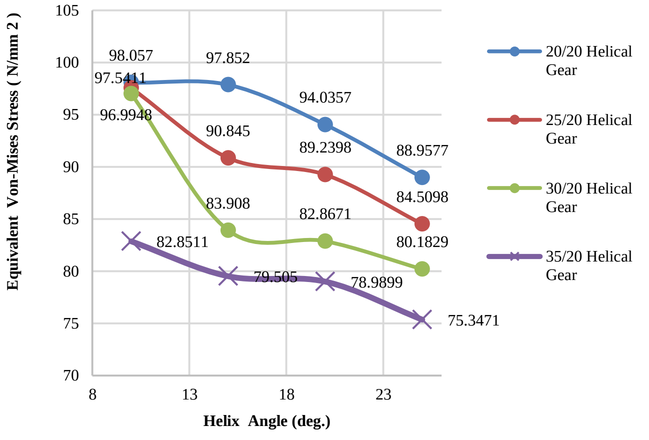

Figure 6. Effect of helix angle and drive side pressure angle on Equivalent Von-mises stress.

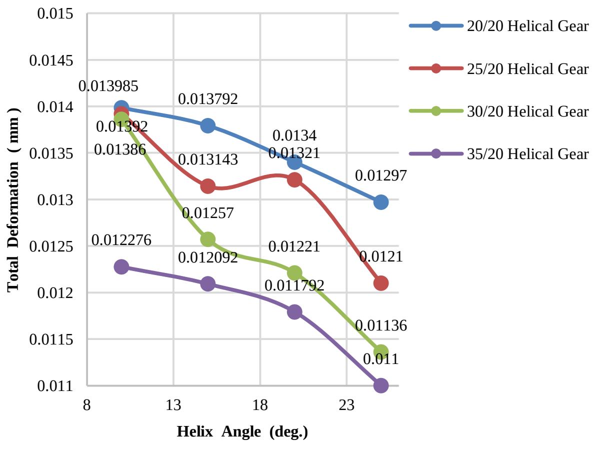

Figure 7. Effect of helix angle and drive side pressure angle on Total deformation

Table IV
Strain gauge specification

Type	UFLA-1-350-23
Gauge Factor	2.15 ± 1%
Gauge Length	2mm
Gauge Resistance	350 ± 1.0 Ω
Transverse Sensitivity	0.4 %

3. Experimental Stress Measurement

Strain gauge experimental set-up has been fabricated with the help of specimen supporting frame, spur gear loading arrangement, gear fixing arrangement, strain gauges and data acquisition system as shown in Fig. 8. In set-up, pinion is free to rotate and gear is fixed. During the experiment pinion has to be fixed at particular meshing position then torque is applied on pinion with the help of lever arrangement on the pinion in clockwise direction. In this experiment the limitations of the space for pasting the strain gauge on the gear tooth small strain gauges have been used. During pasting the UFLA-1-350-23 strain gauges all the precautions and cares were taken. Strain gauge installation and experiment were performed at room temperature. First surface was degreased and cleaned with cleaning solution. The strain gauge was aligned with right position on the surface and was pasted with the help of loctite 496 bonding adhesive. After pasting the strain gauge the silicone paste was used to protect them from environment. Data acquisition system was used to display the output strain of the strain gauge in µstrain and after that stress was calculated with the help of modulus of elasticity of gear material. Asymmetric Helical gear and pinion specimen of steel AISI 1045 material were manufactured on CNC wire EDM. The parameter of gear and pinion are shown in Table II.

TABLE II
The parameters of gear pair.

Sr. no	Parameters	Unit	Pinion and gear
1.	No. of teeth	-	20
2.	Module	mm	4.5
3.	Helix angle	Deg	10
4.	Drive side pressure angle	Deg	35
5.	Coast side pressure angle	Deg	20

6.	Face width	mm	20
7.	Contact ratio	-	1.3677
8.	Young's modulus	GPa	210
9.	Poisons ratio	-	0.28

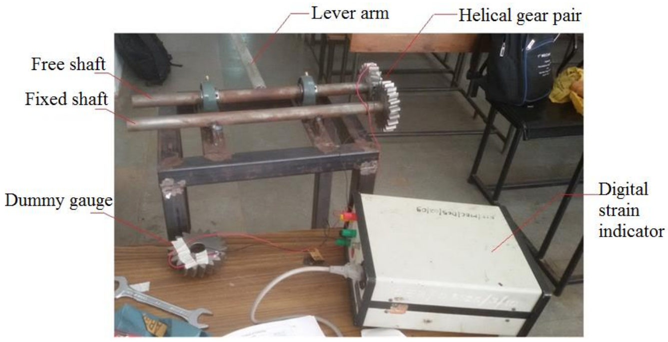

Figure 8. Experimental test setup

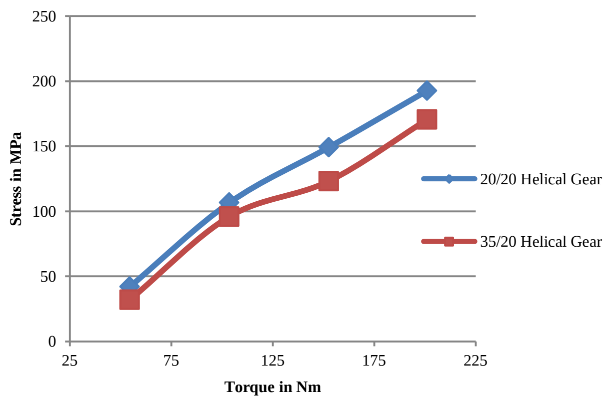

Figure 9. Variation Stress vs loading for symmetric and asymmetric gear

TABLE III
Experimental test result

Load		µ Strain		Stress in MPa	
Kg	*Nm*	*Symmetric Gear*	*Asymmetric Gear*	*Symmetric Gear*	*Asymmetric Gear*
5	54.5	203	152.25	42	31.9725
10	103	507.5	456.75	106.575	95.9175
15	152	710.5	586.67	149.205	123.2
20	201	917.56	812	192.6876	170.52

4. Effect of Asymmetry on Contact Ratio

Low-noise behavior in standard industrial gear units is becoming an important selection criterion and a factor indicating gear quality to the customer. In several gear tests and practical researches the gear contact ratio has been reported with a large effect on noise level, especially in spur gear applications. Lower noise levels are generally associated with gear design leading to higher contact ratios. For this reason a gear design with a high contact ratio is an important key for reducing noise levels. In this section we find the contact ratios for gear pairs with different coefficient of asymmetry. Alexander Kapelevich define coefficient of asymmetry (k) as the ratio of coast side pressure angle to the drive side pressure angle.

$$\text{Coefficient of asymmetry (k)} = \frac{D_{bc}}{D_{bd}} = \frac{\cos \nu_c}{\cos \nu_d} = \frac{\cos \text{Ø}_{bc}}{\cos \text{Ø}_{bd}} \quad (3)$$

For symmetric teeth k = 1

We know that the number of pairs of teeth that are simultaneously engaged is called contact ratio (ε). In case of spur gears contact ratio is the ratio of arc of action to the circular pitch.

For spur gear,

$$\varepsilon_{radial} = \frac{\sqrt{\left(\frac{D_{a1}}{2}\right)^2 - \left(\frac{D_{b1}}{2}\right)^2} + \sqrt{\left(\frac{D_{a2}}{2}\right)^2 - \left(\frac{D_{b2}}{2}\right)^2} - a_x \sin\varphi}{\pi\, m \cos\varphi} \quad (4)$$

Where,

D_a = Addendum circle diameter

D_b = Dedendum circle diameter

a_x = Centre distance between gears

φ = Pressure angle

m = Module

1 = For pinion

2 = For gear

For Symmetric Helical Gear,

In case of helical gear the contact ratio in radial direction is same as it was defined for spur gears. But the fact that the teeth in helical gears are slanted relative to the direction of rotation, gives rise to an additional type of contact ratio known as the axial contact ratio (ε_{axial}). Hence the total contact ratio (ε) for helical gears is the sum of both radial and axial contact ratios.

$$\varepsilon_{axial} = \frac{B}{P_X} \quad (5)$$

$$P_X = \frac{\pi D}{z \sin\beta} \quad (6)$$

$$\varepsilon = \varepsilon_{radial} + \varepsilon_{axial} \quad (7)$$

Where,
B = Face width
P_X = Axial pitch
D = pitch circle diameter
z = No of teeth
β = Helix angle
ε = Total contact ratio

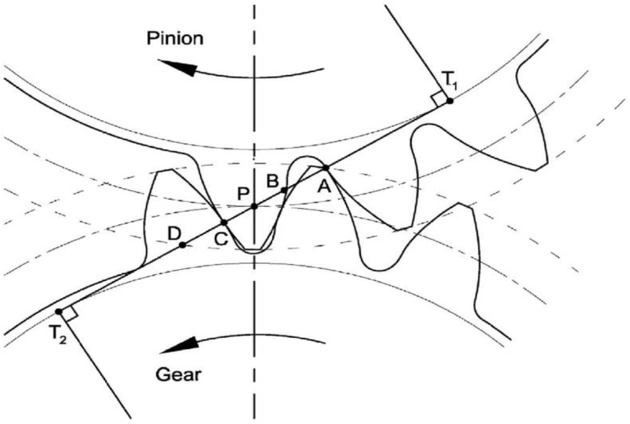

Figure 10. Critical contact position on pinion asymmetric NCR spur gear pair [5]

The radial contact ratios in case of asymmetric helical gear is given by,

$$\varepsilon_{radial} = \frac{z\,(\tan\varphi_{o1d} + m_G \tan\varphi_{o2d} - (1+m_G)\tan\varphi_d}{2\pi} \quad (7)$$

$$\varphi_{od} = \cos^{-1}\left(\frac{D_{bd}}{D_o}\right) \quad (8)$$

Where, m_G = Gear ratio
φ_d = Drive side pressure angle
D_o = Outside circle diameter

TABLE V
Variation of contact ratio with β and k

	Contact ratio (ε)						
β	*Coefficient of asymmetry (k)*						
	1	*1.036*	*1.085*	*1.1471*	*0.9644*	*0.9216*	*0.8717*
10	1.7732	1.6785	1.4105	1.3677	1.8124	1.89	1.92
15	1.8623	1.88	1.8104	1.7811	2	2.43	2.53
20	1.9413	2.0748	2.0293	2.0272	2.1721	2.51	2.622
25	2.0126	2.2958	2.2841	2.3189	2.3618	2.81	2.941

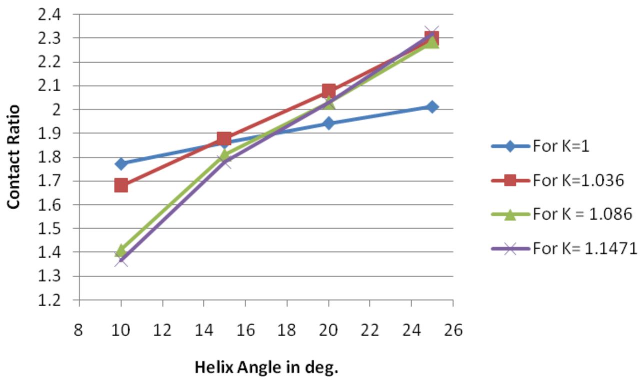

Figure 11. Contact ratio vs. helix angle for k ≥ 1

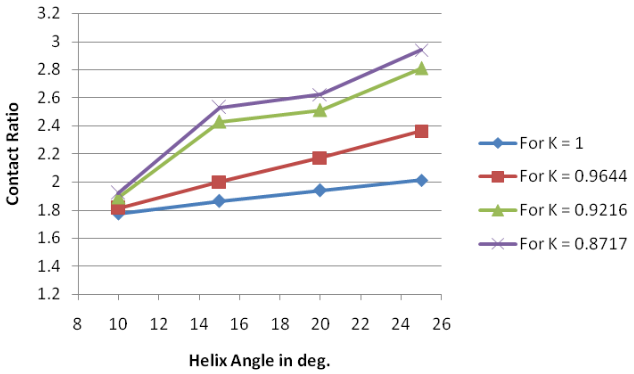

Figure 12. Contact ratio vs. helix angle for k ≤ 1

III. RESULTS AND DISCUSSION

Fig. 6 shows effect of variation in helix angle and drive side pressure angle on equivalent von-mises stress. From the results one can conclude that as the pressure angle on drive side increases von-mises stress decreases similarly it also shows that as helix angle increases von-mises stress decreases .The decreased in von-mises stress for 20 degrees drive side pressure angle is observed to be 9.05 % as helix angle increases from 10 to 25 degree. The decreased in von-mises stress for 10 degree helix angle is observed to be 15.50 % as drive side pressure angle is increased from 20 to 35 degree. Similarly the effect of variation in helix angle and drive side pressure angle on total deformation is given in Fig. 7. Both the figure shows positive result of increased helix and drive side pressure angle.

In numerical analysis it has been seen that increased drive side pressure angle increases the strength of helical gear. Similar result was found in case of increased helix angle. But the increased helix angle also increases the axial trust on shaft. The validation of given numerical solution is done in section IV , here experimental stress measurement is done using digital stain gauge apparatus. The result coming both from numerical and experimental stress analysis shows an average deviation of 9.65%.

From previous studies it is come to know that increased in contact ratio leads to decreased noise emission from power drives. To achieve the same we find the effect of coefficient of asymmetry on contact ratios of helical gear pair in section V. Fig. 11 and Fig. 12 shows that as coefficient of asymmetry is greater than one the contact ratio are first decreases but with increased helix angle contact ratio also increased. But in case of helical gears with decreasing coefficient of asymmetry contact ratio increases with increased in helix angle

IV. CONCLUSION

From this work it is concluded that as the helix angle increases from 10 to 25 degree, all gear pair gives decreased in root stress and total deformation. For a helical gear pair with 20 degree drive side pressure angle this decrement is 9.05 %. The increased in helix angle also lead to decreased total deformation for 20 degree drive side pressure angle gear pair it is observed to be 7.25 %. Similarly as the pressure angle on drive side increases from 20 to35 degree, all gear pair gives decreased in total deformation and root stress. The helical gear with 10 degree helix angle have 15.50 % decreased in von-mises stress, 12.22 % decreased in total deformation .

Finaly from this disscussion it can be concluded that incresed in drive side pressure angle with optimum helix angle gives incresed in strength of helical gears and the incresed in contact ratio is achieved by decreasing coeficint of asymmetry (ie for $k \leq 1$).

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