Effect of Helix Angle on the Gear Design Parameters in Helical Gears

Shubham Dhokale, Sanjay Bahulikar, Vishal Hemke

Abstract**:** ***Screw compressor demands quite operation. For getting lower noise it is important to have higher contact ratio. Contact ratio can be increased by increasing the Helix angle i .e. indirectly increasing overlap ratio. The paper represents the effect of change in design parameters with respect to helix angle with the keeping same module and same centre distance. Higher helix angle leads lower bending and contact stresses. The study was conducted for screw electrical compressor. Gear was design for fixed parameters except helix angle. Also the contact stresses are analyzed (FEA) on ANSYS. The result from the calculation and FEA are compared for contact stress as well as bending stress.***

Index Terms**:** ***contact ratio, contact stress, FEA, overlap ratio***

I. INTRODUCTION

The frequently used type of gear profile is the involute. It is used for cylindrical spur and helical gears as well as for conical gears like beveloid, hypoid and spiral bevel gears. Some characteristics of involute (cylindrical) gears that have made them so common are:

- Uniform transmission of rotational motion, independent of small error in centre distance.
- The sum of the contact forces is constant and the direction of the total contact force always acts in the same direction.
- An involute gear can work together with mating gears with different number of teeth.
- Manufacturing is relatively easy and the same tools can be used to machine gears with different numbers of teeth. (Applies to hobs, shaper cutters, grinding worms, shaving cutters but not to profile tools like milling cutters and profile grinding wheels).

If the gears were perfectly rigid and no geometrical errors or modifications were present, the gears would transmit the rotational motion perfectly.

The equation for transverse contact ratio and given below also the equation for overlap ratio.

$$Transverse\_CR = \frac{\sqrt{(r_1+a)^2 - r_{b1}^2} + \sqrt{(r_2+a)^2 - r_{b2}^2} - (r_1+r_2)\sin\phi}{\pi m\cos\phi}$$

$$Overlap\_Ratio = \frac{b\sin\beta}{\pi m_n}$$

The total contact ratio given by

$$Total\_CR = Transverse\_CR + Overlap\_Ratio$$

Thus to go higher contact ratio we have to increase helix angle.

Revised Manuscript Received on August 02, 2019.

Shubham Dhokale, Mechanical Engineering, Vishwakarma Institute of technology, Pune, India.

Dr. Sanjay Bahulikar, Mechanical Engineering, Vishwakarma Institute of technology, Pune, India.

Vishal Hemke, ACD D&D, Kirloskar Pneumatic company limited, Pune, India.

II. DESIGN PARAMETERS

Following are the common parameters considered while the design of the helical gears.

Sr. No.	Parameter	Value
1	Centre distance (mm)	125
2	Maximum available power (kW)	24.67
3	Normal Pressure Angle	20º
4	Input motor rpm	2940
5	Output required rpm for Application	5200
6	Application factor	1.3
7	DIN accuracy class	6

III. FINITE ELEMENT ANALYSIS

For finite element analysis the total gear pair analysis takes lot of time. For this purpose in this paper I have used only the part of gear pair which is in the contact at that instant.

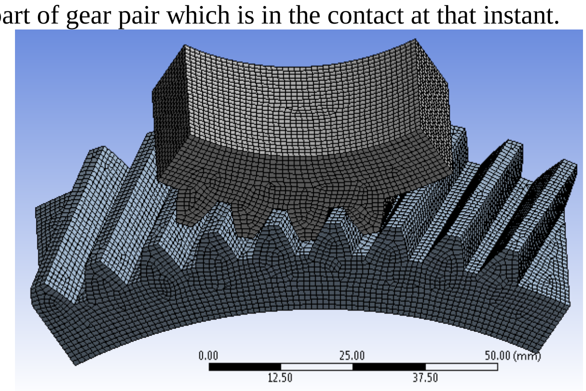

Figure 1 –finite element meshed model of gear pair

- Minimum number of teeth required for simulation-4
- 60 degree of total geometry is taken for simulation

Von mises stresses by calculations- The contact stresses are compressive in nature and as shown in fig. also there would be shear stress.

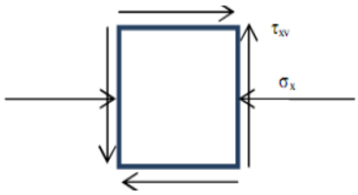

Figure 2-Stress distribution

$$\tau_{xy} = \frac{T*R}{J}$$

Where-

T – Torque

R – Pitch circle radius

J – Polar moment inertia= $\frac{\pi}{32}d^4$

Published By:
Blue Eyes Intelligence Engineering
& Sciences Publication

$$\sigma_H = c_p \sqrt{\frac{F_t}{bdl}\left(\frac{\cos\beta}{0.95CR}\right)K_v K_o (0.93K_m)}$$

$$C_p = 0.564\sqrt{\frac{1}{\frac{1-\mu_1^2}{E_1}+\frac{1-\mu_2^2}{E_2}}}$$

$$l = \frac{\sin\phi\cos\phi}{2}\,\frac{i}{i+1}$$

Results from the simulation are as below

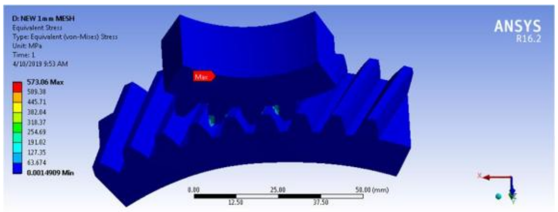

Figure 3- Von misses stresses(Helix angle 22)

Maximum Von-mises stresses due to contact- 573.06 MPa (helix angle-22)

Location-At the top teeth where actual engagement in this case

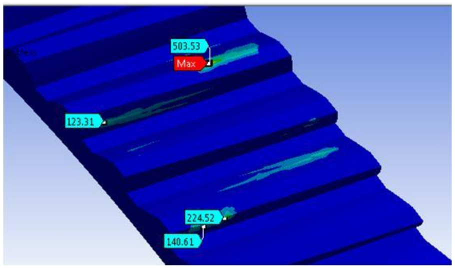

Figure 4- Stress Location (Helix angle 22)

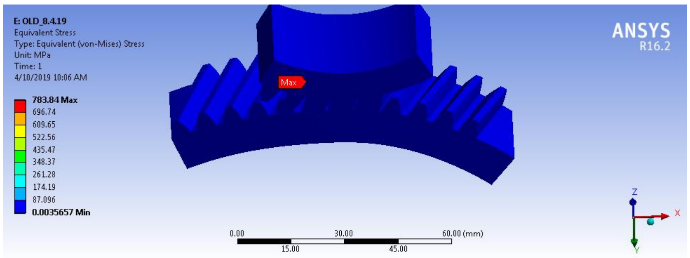

Figure 5- Von misses stresses (Helix angle 7)

Maximum Von-mises stresses due to contact- 783.06 MPa (Helix angle-7)

Location-At the top teeth where actual engagement in this case

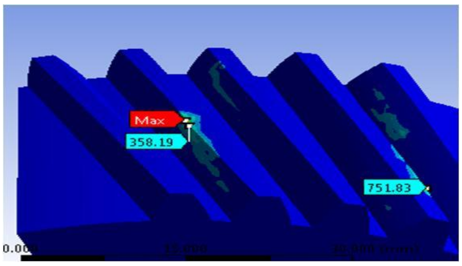

Figure 6- Stress Location (Helix angle 7)

Other results for the different helix angle given below

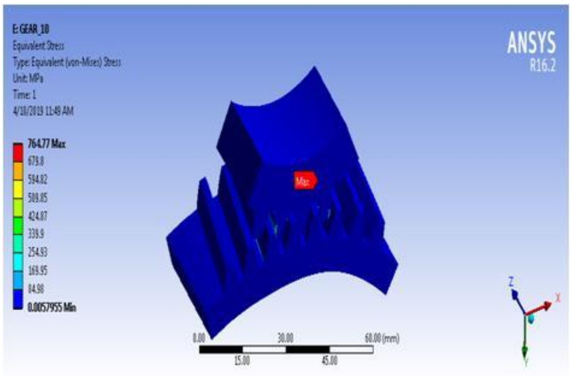

Figure 7- Von misses stresses(Helix angle 10)

Helix angle -10

Max Von mises stress-764.77 MPa

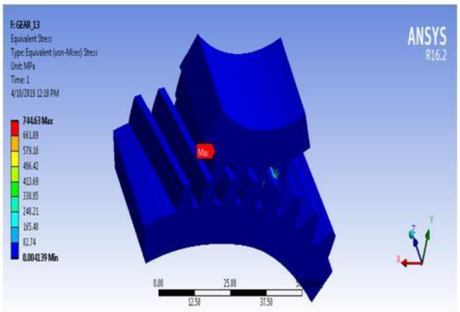

Figure 8- Von misses stresses(Helix angle 13)

Helix angle -13

Max Von mises stress-744.63 MPa

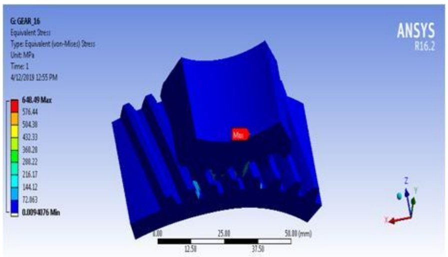

Figure 9- Von misses stresses(Helix angle 16)

Helix angle -16

Max Von mises stress-648.49 MPa

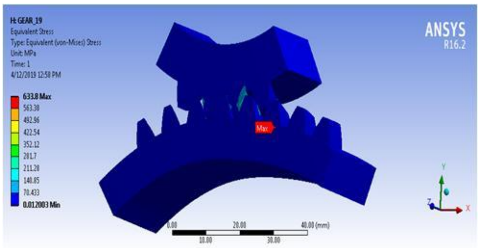

Figure 10- Von misses stresses(Helix angle 19)

Helix angle -19

Max Von mises stress- 633.8 MPa

IV. EFFECT ON GEAR DESIGN PARAMETERS

Figure 11 -Transverse Pressure Angle Vs Helix Angle

The formula for the transverse pressure angle is given by following relationship-

$$\alpha_t = \tan^{-1}(\tan\phi / \cos\beta)$$

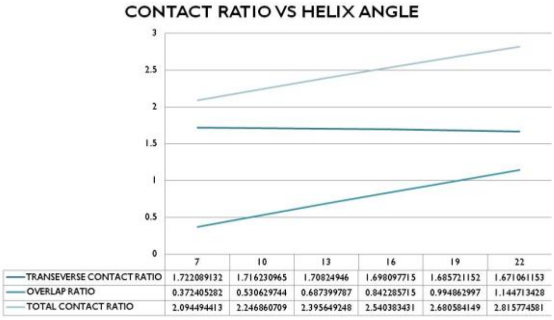

Figure 12-Contact ratio Vs Helix Angle

The formulae for TCR, Overlap ratio and Transverse contact ratio are mentioned earlier in this paper.

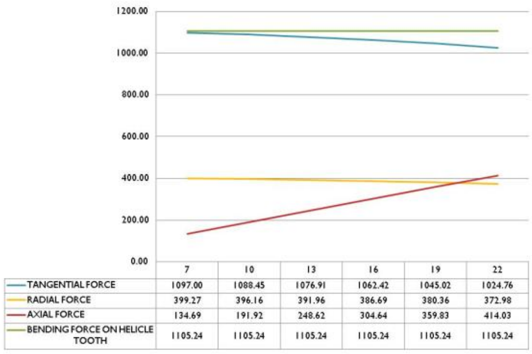

Figure 13-Force Vs Helix Angle

The formula for the transverse contact raio is given by following relationship-

$$F_t = \frac{1000W}{V}$$

$$F_b = \frac{F_t}{\cos\beta}$$

$$F_r = F_t \tan\phi$$

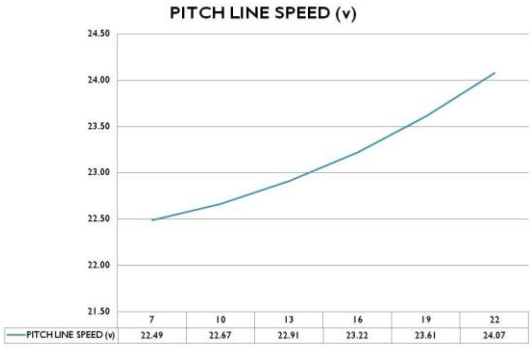

Figure 14-Pitch line velocity Vs Helix Angle

The formula for the Pitch line velocity is given by following relationship-

$$V = \frac{\pi dgNg}{60}$$

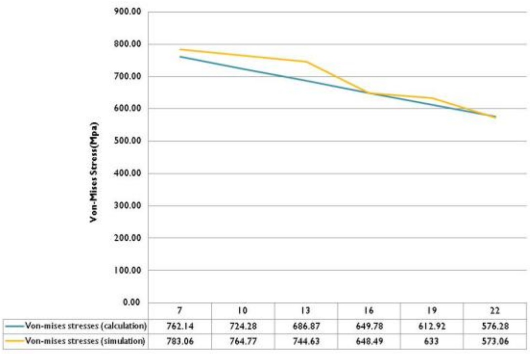

Figure 15-Von misses stresses (contact stresses) Vs Helix Angle

In above figure the results from simulation and calculation are compared also the result from FEA is compared below.

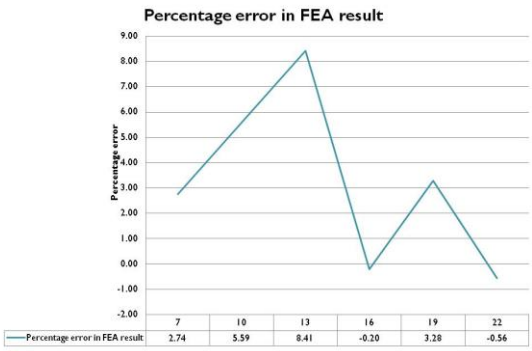

Figure 16-Percentage error vs pressure angle in FEA and calculation

V. CONCLUSION

From designing gear with the higher contact ratio leads to reduction in contact stress. The increasing overlap ratio to greater than 1 which will increase the axial force. The axial force is to taken by bearing which is situated inside the motor. This limiting value has to be considered for design. Pitch line velocity increases with increase with overlap ratio. Total contact ratio and overlap ratio increases with increases in helix angle.

ACKNOWLEDGMENT

I am thankful to my project guide Prof. (Dr.) Sanjay Bahulikar for his constant encouragement and valuable guidance. I am thankful to my industrial project guide Mr. Vishal Hemke, for his constant encouragement and valuable guidance.

REFERENCES

1. Mats Åkerblom "Gear Noise And Vibration– A Literature Survey"
2. Yoon K. Purdue. University,Doctoral Thesis, 1993."Analysis of Gear Noise and Design for Gear Noise Reduction"
3. A. Kahraman, G.W. Blankenship, Transactions of ASME, Journal of Mechanical Design, vol. 121, pp. 112-118, March 1999. "Effect of Involute Contact Ratio on Spur Gear Dynamics".
4. H.H. Lin, J. Wang, F. Oswald, J.J. Coy, Gear Technology, pp. 18-25, July/August 1994. "Effect of Extended Tooth Contact on the Modeling of Spur Gear Transmissions".
5. V.I. Nikolayev, E.I. Podzharov, SU Patent 1320568, February 12, 1986. "Involute Spur Gear".
6. E.B. Vulgakov, Moscow: Mashinostroenie,1974. "Gears with Better Properties".
7. W.S. Rouverol, Y. Watanabe, in International Symposium on Gearing and Power Transmissions, Tokyo, 1981, pp. 103-108. "Maximum-Conjugacy Gearing. –Part 1. Theory".
8. Y. Watanabe, W.S. Rouverol, in International Symposium on Gearing and Power Transmissions, Tokyo, 1981, pp. 109-114. "Maximum-Conjugacy Gearing. –Part 2. Test Results
9. Smith J. D. C08293 IMechE 1994 "Helical Gear Vibration Excitation with Misalignment".
10. Iwase Y., Miyasaka K. JSAE Review 17 (1996) pp 191-193. "Proposal of Modified Tooth Surface with Minimized Transmission Error of Helical Gears".
11. Drago R. J. Machine Design, Dec. 1980."How to Design Quiet Transmissions".
12. Masuda T., Abe T., Hattori K. Journal of Vibration, Acoustics, Stress and Reliability in Design, Vol. 108, Jan. 1986 pp 95-100. "Prediction Method of Gear Noise Considering the Influence of the Tooth Flank Finishing Method".

AUTHORS PROFILE

Shubham Dhokale
Master of technology, Design Engeeniring, Mechanical Engineering, Vishwakarma Institute of technology, Pune, India.

Dr. Sanjay Bahulikar
Dean academics, Mechanical Engineering, Vishwakarma Institute of technology, Pune, India.

.

Vishal Hemke,
Senior Manager, ACD D&D, Kirloskar Pneumatic Company Limited, Hadapsar - Pune, India.