Effect of Pressure Angle on Transmission Efficiency of Helical Gears

Pande A. M., Kharde Y. R.

Abstract:- In this study, a test methodology for measuring load-dependent (mechanical) power losses of helical gear pairs is developed. A high-speed four-square type test machine is adapted for this purpose. Several sets of helical gears having 3 different pressure angles are manufactured, and their power losses under dip lubricated conditions are measured at various speed and torque levels. A general trend found in the experimental testing was that the higher the pressure angle, the lower the temperature-increase of the lubricant across the gearbox while being tested at identical conditions. This is an indication of the improved efficiency. Finally it was concluded that high-pressure angle helical gears (25°) pressure angle running at high speed provide improved performance over more traditional gear pressure angles (20°).

Key Words- load-dependent power losses, helical gear, pressure angle, efficiency.

I. INTRODUCTION

The looming energy crisis and environmental concerns in regards to global warming and air quality have recently placed great emphasis on fuel economy performance and gas and particulate emissions of passenger and commercial vehicles. Both emissions and fuel consumption of a vehicle are influenced largely by the efficiency of the power train of the vehicle. Power losses of a power train can be traced back to the inherent losses of the engine in generating the power and the losses that occur during transmission of power to the wheels through the drive train. The transmission can be identified as the major component of the drive train not only in terms of its contributions to the power losses, but also the potential it presents for improving overall power train efficiency.

The power losses consist of speed- and load-dependent losses. Speed-dependent losses can be divided into windage losses, churning losses, bearing churning losses and seal losses. The load-dependent losses are made up of sliding friction loss, rolling friction loss and bearing loss. In automotive applications, for instance, geared components of systems such as manual transmissions, transfer cases, and front or rear axles, might rotate at reasonably high speeds (say, gear pitch-line velocities in excess of 20–30 m/s) to cause significant amounts of spin losses. While load-dependent and spin power losses can be comparable in magnitude under high-load and lower speed conditions, the spin losses typically dominate the load-dependent power losses at these higher operating speeds.

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II. TYPES OF GEAR LOSSES

A. Speed Dependant Losses

Windage losses: As gears rotate, lubricant is flung off the gear teeth in small oil droplets due to the centrifugal force acting on the lubricant. These lubricant droplets create a fine mist of oil that is suspended inside the gear housing/case. The effect of this oil mist is an increase in "windage frictional resistance" on the gears and hence an increase in the power consumption. In addition, the expulsion of the oily atmosphere from the tooth spaces as the gear teeth come into engagement creates turbulence within the gearbox and increases the power consumption. The combination of these factors, as well as the losses at the side faces of the gears, contribute to the total windage losses. Factors that influence the magnitude of the windage loss include the rotational speed of the gear because power losses rise with an increase in peripheral velocity. Other factors arc the tooth module. the amount of oil mist present inside the casing and the diameter of the gears.

Oil Churning Loss:- Churning losses are defined as the action of the gears moving the lubricant inside the gear case and referred in particular to the losses due to entrapment of the lubricant in the gear mesh, which is more applicable to spur gears than to helical gears. Factors influencing the oil churning loss are the viscosity of the oil, as this resists the motion of the gears: peripheral velocity: operating temperature: the tooth module: the helix angle: and the submerged depth of the gears. All rotating components that are in direct contact with the lubricant, i.e. dipped into the oil, contribute to the churning losses, and the deeper the components are submerged, the higher the losses. With larger helix angles, the power losses are lower as the gear teeth slice through the lubricant rather than displacing the lubricant along the whole gear face width.

B. Load Dependant Losses

Sliding Friction Loss:- Principally, the instantaneous sliding friction loss is a function of the instantaneous sliding velocity and the friction force, which itself is a function of the instantaneous normal tooth load and the instantaneous coefficient of friction, The magnitude of sliding velocity depends on the position of contact along the contact path with a peak velocity at the start of the approach. The velocity reduces to 0 at the pitch point of the two mating gears and rises again to a peak value at the end of the recess. The effect of the sliding friction loss is an increase in power consumption, where the magnitude depends on the point of contact. It is influenced by the angular velocity of the gears, the ratio of the rolling velocities, the point of contact, the contact ratio and the lubricant properties. The sliding friction loss is dependent on the position of contact during the engagement cycle.

Rolling friction loss:- The rolling friction loss is dependent on the instantaneous rolling velocity and the instantaneous lubricant film thickness. As the gear teeth come into mesh, an elastohydrodynamic lubricant film is developed between the teeth in contact. The action of the gear teeth during the engagement draws the lubricant into the contact zone. The parameters that influence the rolling friction loss are the lubricant film thickness, the angular velocity of the gears, the working pressure angle and the point of contact along its contact path. The lubricant properties influence the buildup of the lubricant film, its shear values and its thermal behavior. In addition, the gear material and the normal tooth load also influence the film thickness.

Spin power losses of a gearbox are either due to churning of the lubricant if the rotating components of the gearbox are immersed in an oil bath (dip-lubricated) or due to windage if the lubrication method is jet-type and the surrounding medium is air or a fine mist of air and oil. Focusing on the most fundamental component of the gearbox, i.e. a gear pair in mesh, this chapter aims at developing a novel physics-based fluid mechanics model of oil churning losses due to interactions of the gears, both as a pair and as individual entities, with the surrounding lubricant medium.

III. LITERATURE REVIEW

Robert F. Handschuh et al completed a preliminary study to determine the feasibility of using high-pressure angle gears in aeronautic and space applications[1]. ***Y. Michlin et al*** described a methodology for analysis of gear transmission with allowances for the power losses due to both the rolling and sliding friction. The latter friction, incorporated in the traditional methods, does not by itself account for the wear of the teeth at contact sites adjoining the pitch point, where the rolling losses are paramount[2]. ***Xu Hai et al*** proposed a computational model for the prediction of friction-related mechanical efficiency losses of parallel-axis gear pairs. The model incorporates a gear load distribution model, a friction model, and a mechanical efficiency formulation to predict the instantaneous mechanical efficiency of a gear pair under typical operating, surface, and lubrication conditions[3]. ***Sheng Li et al*** studied the influence of basic design parameters and tooth surface modifications on the mechanical (friction induced) power losses of a helical gear pair[4]. ***S. Seetharaman et al*** proposed a physics-based fluid mechanics model to predict spin power losses of gear pairs due to oil churning and windage[5]. ***Shanming Luo et al*** developed a geometric relationship and mathematical model of pressure angle of elliptical gears based on the theory of gearing, and some constraints of pressure angle are presented in order to prevent gear teeth from coming out of mesh[6]. ***C. Changenet et al*** presented a series of formulas which enable accurate predictions of churning losses for one pinion characteristic of automotive transmission geometry. The results are based on dimensional analysis and have been experimentally validated over a wide range of speeds, gear geometries, lubricants, and immersion depths[7]. ***Petra Heingartner et al*** reviewed some of the mathematical models proposed for the individual components associated with speed and load dependant losses, such as windage, churning, sliding and rolling friction losses[8]. ***T. T. Petry-Johnson et al*** developed a test methodology for the measurement of spur gear efficiency under high-speed and variable torque conditions. A power-circulating test machine was designed to operate at speeds to 10,000 rpm and transmitted power levels to 700 kW[9].

IV. EXPERIMENTAL SET UP

Eighteen different gear sets will be used in this study to implement the test matrix shown in Table 1. This test matrix is formed with the goal of investigating the influence of normal pressure angle on gear mesh power losses. According to the test matrix of Table 5, tests will be performed in three groups. First group consists of three gear sets A,B and C. Here, gear set A is formed by 16 and 32-tooth helical gears with a pressure angle Φ = 25° and helix angle Ψ = 25. Gear set B with normal pressure angle of Φ = 22.5° and helix angle of Ψ = 25°. Meanwhile, gear set C with Φ = 20° and Ψ = 25°

The second group in the test matrix is formed by 15 and 30-tooth helical gears. Gear set D have a pressure angle Φ = 25° and helix angle Ψ = 20. Gear set E with normal pressure angle of Φ = 22.5° and helix angle of Ψ = 20°. Meanwhile, gear set F with Φ = 20° and Ψ = 20°

The third group in the test matrix is formed by 15 and 30-tooth helical gears. Gear set G have a pressure angle Φ = 25° and helix angle Ψ = 15. Gear set H with normal pressure angle of Φ = 22.5° and helix angle of Ψ = 15°. Meanwhile, gear set I with Φ = 20° and Ψ = 15°

Each test listed in Table 5 will be repeated at discrete speed values of Ω = 2,500 , 3,400 and 4,400 rpm and torque levels of T_c = 0.0, 2.5, 5.0, 7.5 and 10.0 Nm. Here, the tests at T_c = 0.0 serve a special purpose as they represented the load-independent spin losses.

Table no 1(a) Test matrix showing dimensions of helical gear pair for pressure angle 20°

Abbr.	RH	LH	RH	LH	RH	LH
z	16.000	32.000	15.000	30.000	15.000	30.000
b	22.000	22.000	22.000	22.000	22.000	22.000
mn	2.000		2.000		2.000	
p	6.283		6.283		6.283	
aw	50.000		50.000		50.000	
Φ	20.000		20.000		20.000	
Ψ	15.000		20.000		25.000	
da	37.533	70.453	38.563	68.888	37.560	70.425
df	28.547	61.467	30.112	60.437	28.575	61.440
dw	33.333	66.667	33.333	66.667	33.333	66.667

Table no 1(b) Test matrix showing dimensions of helical gear pair for pressure angle 22.5°

Abbr.	RH	LH	RH	LH	RH	LH
z	16.000	32.000	15.000	30.000	15.000	30.000
b	22.000	22.000	22.000	22.000	22.000	22.000
mn	2.000		2.000		2.000	
p	6.283		6.283		6.283	
aw	50.000		50.000		50.000	
Φ	22.500		22.500		22.500	

Ψ	20.000		25.000		15.000	
da	38.598	68.972	37.562	70.427	37.534	70.455
df	30.028	60.402	28.573	61.438	28.545	61.466
dw	33.333	66.667	33.333	66.667	33.333	66.667

Table no 1(c) Test matrix showing dimensions of helical gear pair for pressure angle 25°

Abbr.	RH	LH	RH	LH	RH	LH
z	16.000	32.000	15.000	30.000	15.000	30.000
b	22.000	22.000	22.000	22.000	22.000	22.000
mn	2.000		2.000		2.000	
p	6.283		6.283		6.283	
aw	50.000		50.000		50.000	
Φ	25		25		25	
Ψ	15.000		20.000		25.000	
da	37.535	70.457	37.309	70.604	37.562	70.429
df	28.543	61.465	28.396	61.691	28.571	61.438
dw	33.333	66.667	32.609	67.391	33.333	66.667

V. EXPERIMENTAL FACILITIES AVAILABLE

The test rig is manufactured by me. The machine consists of a pair of opposing, identical gear boxes, each containing a pair of helical gears with gear ratio 2, supported by rigid shafts mounted on two pairs of bearings. They are labeled as 'Test Gearbox' and 'Reaction Gearbox' for the purpose of distinguishing them while they are identical in every aspect. Each gear is held in place between two bearings, a deep-groove ball bearing on the outside (SKF Model 6206) and a four-point angular contact ball bearing (SKF Model QJ 206 MA) on the inside. The deep-groove ball bearing is not axially loaded (allowed to float axially), allowing all of the axial load to be carried by the four-point angular contact ball bearing. In this arrangement, the bearings are not required to be preloaded. This avoids repeatability problems associated with setting the correct amount of preload for each test as well as changes in preload due to temperature effects that would be detrimental to the fidelity of the measurements. The gearboxes are powered by a variable speed AC motor connected to a high-speed spindle. The connection is through a 3:1 ratio belt drive. The high speed spindle is connected to the splined input shaft of the reaction gearbox through a flat belt pulley. The torque loss in the system T_L is measured using a high-precision, non-contact type torque-meter (Lebow model TMS 9000) that is mounted on the output shaft of reaction gearbox.. The lubrication systems and parameters of the test and reaction gearboxes are kept identical to ensure the same thermal conditions for both gearboxes. The maximum operating speed that the test machine can reach is 400 rpm, with a fixed center distance of 50 mm and fixed face width of 22 mm.

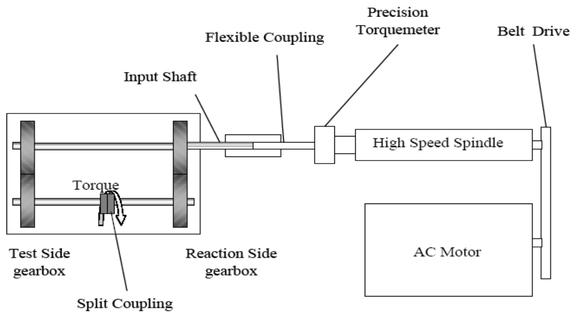

Fig.1 Experimental set up

VI. RESULT AND DISCUSSION

The influence of the basic design parameters such as pressure angle as well as the rotational speed and input torque on load independent (spin) power losses of helical gear pairs are studied experimentally in this chapter. Applying the test methodology proposed in the previous chapter, the experiments defined in the test matrix of Table 5(a), 5(b) and 5(c) are executed under both loaded and unloaded conditions in order to calculate total gear pair power loss.

Fig. 2 to 10 shows the variation of efficiency against the pressure angle. Fig 2, 3 and 4 shows the variation for helix angle 25° at different speeds. Fig 2 and 3 shows nearly linear loss in efficiency for decreasing pressure angles for all torque values.

Fig 5, 6 and 7 shows the variation of efficiency against pressure angle for helix angle 20° and for different speeds. As the pressure angle decreases the efficiency decreases for all torques and speeds values.

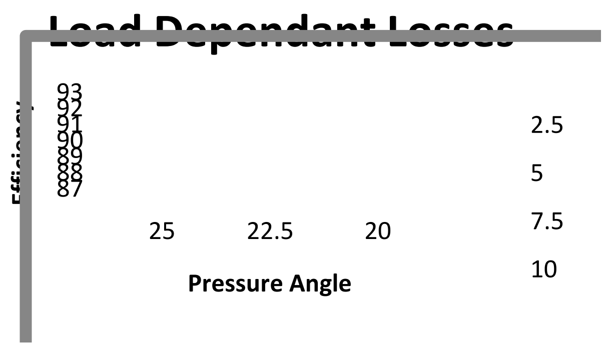

Fig. 2 Pressure angle vs. Efficiency Helix angle – 25, Speed 2500 rpm

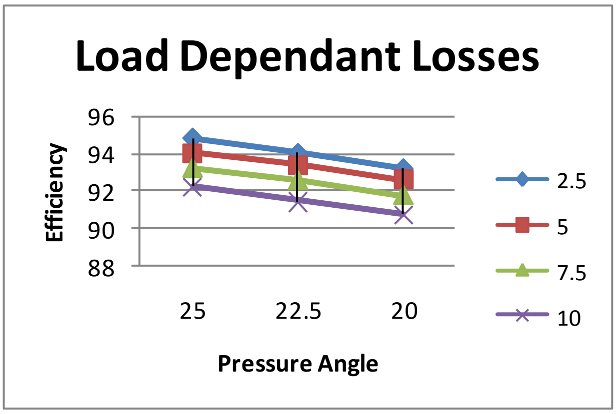

Fig. 3 Pressure angle vs. Efficiency Helix angle – 25, Speed 3400 rpm

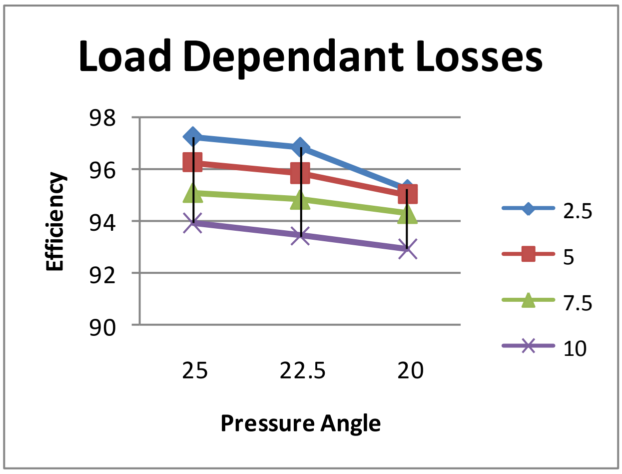

Fig. 4 Pressure angle vs. Efficiency Helix angle – 25, Speed 4400 rpm

Fig. 8, 9 and 10 shows the variation of efficiency against the pressure angle for helix angle 15°. In fig 8 torque 2.5 Nm and for 20° pressure angle efficiency is much as compared to other torque values.

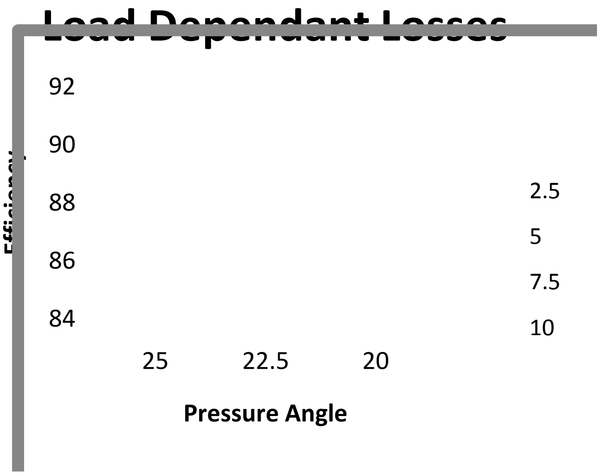

Fig. 5 Pressure angle vs. Efficiency Helix angle – 20, Speed 2500 rpm

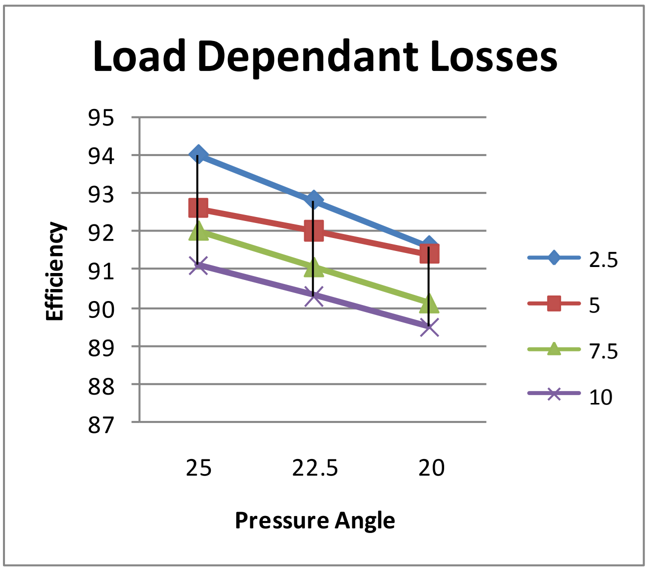

Fig. 6 Pressure angle vs. Efficiency Helix angle – 20, Speed 3400 rpm

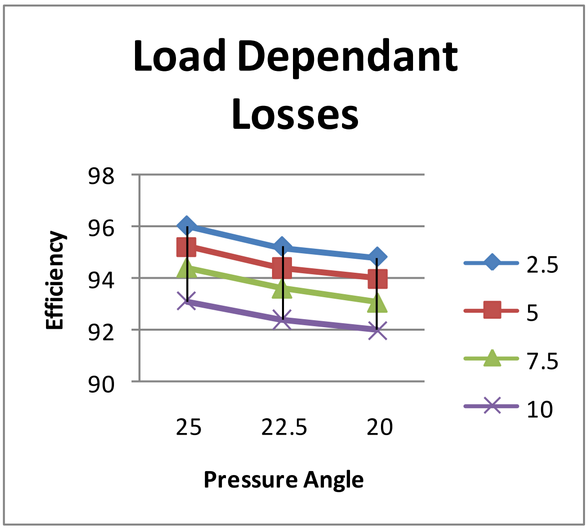

Fig. 7 Pressure angle vs. Efficiency Helix angle – 20, Speed 4400 rpm

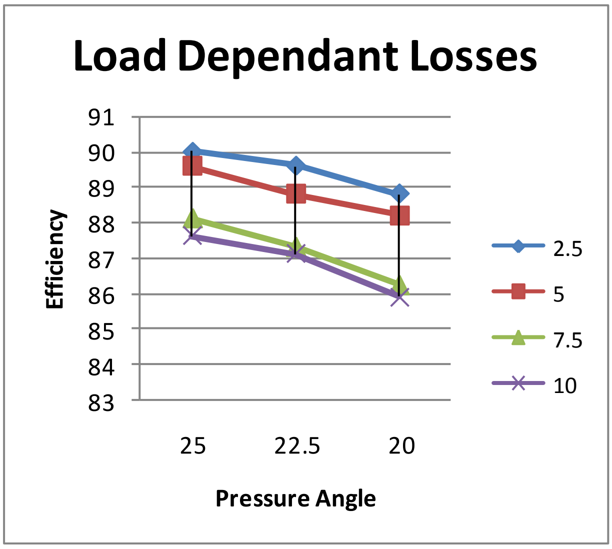

Fig. 8 Pressure angle vs. Efficiency Helix angle – 15, Speed 2500 rpm

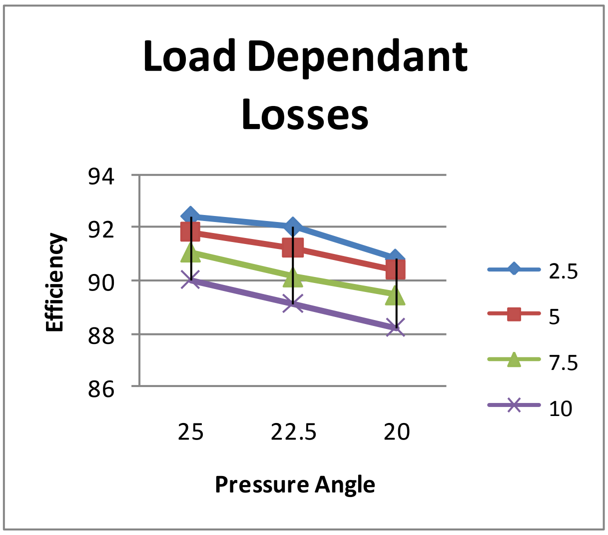

Fig. 9 Pressure angle vs. Efficiency Helix angle – 15, Speed 3400 rpm

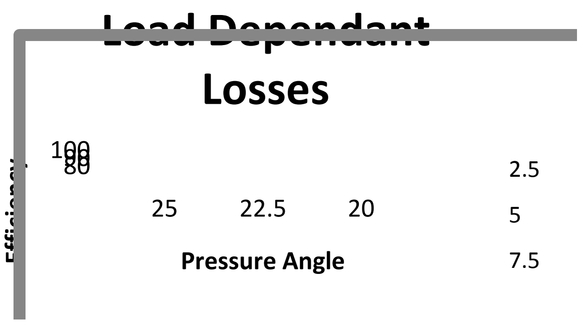

Fig. 10 Pressure angle vs. Efficiency Helix angle – 15, Speed 4400 rpm

All the above graphs show a significant loss in efficiency as pressure angle decreases. This means that higher the pressure angle, higher the efficiency and lower the pressure angle, lower the efficiency. But as the pressure angle increases the tooth thickness decreases. Therefore high pressure angles can be used for moderately low torques and for heavy torques low pressure angle can be used.

VII. CONCLUSION

An experimental test matrix, which included different pressure angles and helix angles was defined and implemented for dip lubricated conditions. These results comprised the database of helical gearbox and gear mesh power losses and efficiency. The effects of pressure angle as well as the effects of speed and torque on the gearbox mechanical power losses and gearbox mechanical efficiency were presented graphically and quantified. The study revealed nearly the same results as that of the previous study for heavy vehicles. The pressure angle and torque have the significant effect on helical gear power losses. As the pressure angle increases, the power losses get reduced because of the fact that as the pressure angle increases the tooth thickness decreases. As the torque increases the power losses also increases. An expected increase in metal-to-metal contact activity with increased torque can be pointed to one of the reasons for this. It is seen from the study that efficiency increases almost linearly with speed. This improvement in efficiency can be attributed directly to the fact that the asperity interactions are reduced with increased speed since the lubricant film thickness is increased.

NOMENCLATURE

Abbrevia	
ΔL(τ)	
FF(τ)	
TT(τ)	
ΔL(τ)	
z	Number of teeth Pinion / Gear
b	Face width (Pinion / Gear)
mn	Normal module
p	Circular pitch
aw	Center distance (working)
Φ	Pressure angle
Ψ	Helix angle
da	Tip diameter
df	Root diameter
dw	Operating pitch diameter

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