Research Paper Engineering

Comparison Between Experimental and Theoretical Investigation of Power Losses of Single Stage Spur Gear System

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ABSTRACT

This paper deals with the comparative analysis of experimental and theoretical power losses of single stage spur gear system. Firstly theoretical analysis is done by using various empirical formulas and then after total power loss is determined with the help of an experimental test setup. Analysis of the power loss data shows that these operating parameters influence the components of the transmission power loss significantly.

Keywords : gear, sliding power loss, rolling power loss, churning power loss

INTRODUCTION

Gears are an important transmission system and are used in automotive industry, turbines and compressors, gear pumps and most commonly in vehicle drive trains. Since gears transmit power through rotational motion at different speeds and torques, power is lost through dissipation taking place due to friction between geared elements and spin losses accounted by the surrounding of the system.

Power losses in the drive trains of passenger vehicles have been one of the major concerns in automotive power train over the past few decades. Such loss have a big impact on fuel consumption of the vehicle and helps to define how good a vehicle is in terms of its fuel economy and gas/particulate emission levels. In the present time, the study of power losses and the efficiency of gear transmission has become an important area of interest, as depleting fossil fuel resources has stressed the urgent need to improve the efficiency of transmissions.

OBJECTIVE

- Comparison of theoretical total power loss for SAE30 and SAE40 with the variation of speed.
- Comparison of experimental total power loss for SAE30 and SAE40 with the variation of speed.
- Comparison of theoretical and experimental total power loss for SAE30 lubricant with the variation of speed.
- Comparison of theoretical and experimental total power loss for SAE40 lubricant with the variation of speed.

DESCRIPTION OF EXPERIMENTAL TEST SETUP:

An experimental test setup is developed in this study to measure total power loss. That set up consists of many parts. The setup is shown in figure 1:

Figure 1: experimental test rig

THE SPECIFICATIONS OF THE GEARS ARE:

TABLE – 1

Properties	Gear 1	Gear 2
No. of teeth	32	35
Face width	13mm	13mm
Outer diameter	58.8mm	66.49mm
Addendum + dedendum	4.22mm	4.22mm
Module	1.96mm	1.96mm
Pitch circle diameter	56.84mm	64.5mm
Pressure angle	20˚	20˚

SPECIFICATIONS OF USED LUBRICANT ARE:

TABLE - 2

Type of lubricant	Density (kg/m^3)	Kinematic viscosity (cSt) at 40°C	Kinematic viscosity (cSt) at 100°C
SAE 30	885	100	11.3
SAE 40	897.1	160.8	15.2

At the input side a DC motor is used to drive the transmission. The out shaft of the transmission is connected to the rope brake dynamometer through the pulley. The different load applied at the rope brake dynamometer is adjusted by spring balance. The input power is measured with the help of multi meter. The tachometer measures input and output speed of transmission.

POWER LOSS ANALYSIS PROCEDURE:

THEORETICAL POWER LOSS ANALYSIS:

Model of bones[2] for churning power loss:

$$P_{churning} = \frac{1}{2} \cdot C_m \cdot \rho \cdot \omega^3 \cdot S_m \cdot r^3 \quad (1)$$

With S_m : Surface area in contact with the gear

For laminar flow Re < 2000

$$C_m = \frac{20}{Re} \quad (2)$$

For intermediate flow regime 2000 < Re < 100000

$$C_m = 8.6 \cdot 10^4 \cdot Re^{1/3} \quad (3)$$

For turbulent flow Re > 100000

$C_m = (5 \cdot 10^8) / Re^2$ (4)

Here
C_m: Constant based on flow regime

ρ: lubricant density

ω: angular velocity of spur gear

ω=2πN/60

r: pitch circle radius

Re: Reynolds number

Re= v.l/υ

v: rotational speed

v= r.ω

l: immersion depth

υ: kinematic viscosity

Sliding power loss:

$$P_{sliding} = \sum_{k=1}^{N} \mu_k \cdot F_{N,k} \cdot v_{s,k} \quad (5)$$

Where k: no. of teeth comes into contact
Sliding velocity and normal force:
Ohlendorf model for sliding velocity and normal force:

Most popular formula derived by Ohlendorf to calculate the product of sliding velocity and normal force:

$$\sum_{k=1}^{N} F_{N,k} \cdot v_{s,k} = P_i \cdot \frac{\pi \cdot (i+1)}{z \cdot i} \cdot (1 - \varepsilon_a + \varepsilon_1^2 + \varepsilon_2^2) \quad (6)$$

Where

P_i: input power

ε_a: profile contact ratio taken as 1

ε_1, ε_1: tip contact ratio taken as 1

i: gear ratio (ratio of the teeth on the gear and to the pinion)

Gear ratio or velocity ratio is the ratio of angular velocity of input gear to the angular velocity of output gear.

Z: no. of teeth on the pinion

Höhn model for friction coefficient[3]: Höhn calculates the average friction coefficient in function of the load, roughness, viscosity, velocity and geometrical properties. The lubricant factor can be estimated, but is often set to 1.

$$\mu = 0.048 \cdot \left(\frac{F_{bt}/b}{V_{\sum c} \cdot \rho_{redc}}\right) . \eta_{oil} . R_a . X_l \quad (7)$$

With

F_{bt} : tangential force at the base circle

R_a : arithmetic mean roughness (1.6193 microns)

X_l: lubricant factor

$v_{\sum c}$: sum speed at operating pitch circle

ρ_{redc}: reduced radius of curvature at pitch point

η_{oil}: coefficient of dynamic viscosity

The sum speed at operating pitch circle and the reduced curvature radius can be calculated like:

$$v_{\sum c} = 2 \cdot v \cdot \sin(\alpha_p)$$

v: rotational speed

$$\rho_{redc} = \frac{1}{2} \cdot d \cdot \sin(\alpha_p) \cdot \frac{i}{i+1} \cdot \frac{1}{\cos\beta}$$

d: pitch circle diameter

α_p: pressure angle of spur gear (20°)

β: helix angle (for spur gear its value is 0)

Rolling losses:

$$P_{rolling} = \sum_{k=1}^{N} F_{r,k} \cdot v_{r,k} \quad (8)$$

The rolling velocity can be derived from the angular speed. The rolling force however is dependent on the film thickness. [4]Anderson derived an equation for the rolling force which is used by most researchers is
$F_R = C_R \cdot h \cdot \varphi \cdot b$ (9)

Where
C_R : constant 0.001 chosen by Anderson

φ: thermal reduction factor (taken as 1)

Anderson proposed following equation for the film thickness:

$$h = 2.05 \cdot 10^{-7} \cdot (V_R \cdot \mu_d)^{0.67} \cdot \left(\frac{T}{2 \cdot r \cdot \cos(\alpha_p)}\right)^{-0.67} \cdot r_{eq}^{0.464} \quad (10)$$

Where

μ_d : dynamic viscosity

R_{eq} : equivalent contact radius

$R_{eq} = (R_g \cdot R_p) / (R_g + R_p)$

T= F· r

F: tangential force at the pitch circle

EXPERIMENTAL POWER LOSS ANALYSIS:
Some steps to calculate total power loss:
With the help of multi meter reading calculate input power (P_{in}) of the gear system.

$P_{in} = V \times I$ (watt) (11)

Where, P_{in} = Input power

V = Voltage

I = Current

Measure input speed with the help of tachometer and calculate input torque by following equation

$T_{in} = (60 \times P_{in}) / 2 \times \pi \times N$ N-m (12)

Where, P_{in} = Input power

N = rpm

To measure output power loss, load will be varied from rope brake dynamometer by adjusting spring balance.

$T_{out} = 9.81 \times W \times R$ N-m (13)

Where $W = (S_1 – S_2)$ Kg
T_{out} = Output torque

W = Load on pulley

R = Radius of pulley

$$P_{out} = (2 \times \pi \times T_{out} \times N_{out}) / 60 \quad \text{watt} \qquad (14)$$

Where, P_{out} = output power

N_{out} = rpm at output shaft

The total transmission power loss efficiency under loaded condition is determined simply as

$P_t = P_{in} - P_{out}$

RESULTS AND DISCUSSION:
Variation of different types of theoretical power loss w.r.t. speed for SAE30:

TABLE – 3

Speed	Churning power loss	Sliding power loss	Rolling power loss	Total power loss
30	0.0430	0.0696×10^{-3}	0.0552×10^{-12}	0.0431
60	0.1721	0.0855×10^{-3}	0.0873×10^{-12}	0.1722
90	0.3873	0.1090×10^{-3}	0.1137×10^{-12}	0.3874
120	0.6885	0.1327×10^{-3}	0.1369×10^{-12}	0.6886
150	1.0758	0.1789×10^{-3}	0.1574×10^{-12}	1.0760

Variation of different types of theoretical power loss w.r.t. speed for SAE40:

TABLE – 4

Speed	Churning power loss	Sliding power loss	Rolling power loss	Total power loss
30	0.0701	0.1135×10^{-3}	0.0765×10^{-12}	0.0702
60	0.2806	0.1393×10^{-3}	0.1211×10^{-12}	0.2807
90	0.6313	0.1777×10^{-3}	0.1577×10^{-12}	0.6315
120	1.1223	0.2163×10^{-3}	0.1899×10^{-12}	1.1225
150	1.7536	0.2917×10^{-3}	0.2183×10^{-12}	1.7539

Comparison of theoretical total power loss for SAE30 and SAE40 lubricants with the variation of speed:

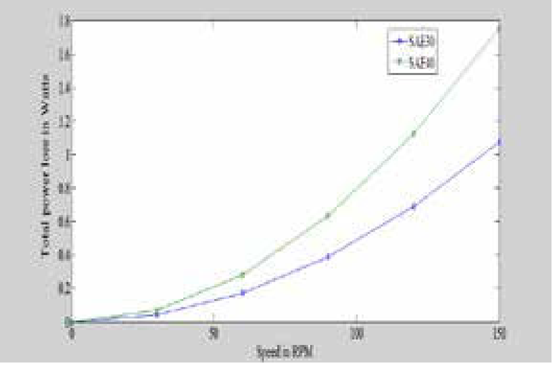

Figure 2: graph of comparison of total power loss

Variation of experimental total power loss w.r.t. speed for SAE30 lubricant:
TABLE – 5

Input Speed (RPM)	Input power (watt)	$W=(T_1 – T_2)$ (Newton)	Output power (watts)	Power loss (watts)
30	2.79	21	2.36	0.43
60	6.3	23	5.30	1
90	10.76	26	9.00	1.76
120	15.78	29	13.38	2.4
150	22.9	31	17.88	5.02

Variation of experimental total power loss w.r.t. speed for SAE40 lubricant:

TABLE – 6

Input Speed (RPM)	Input power (watts)	$W=(T_1 – T_2)$ (Newton)	Output power (watts)	Power loss (watts)
30	2.79	20	2.30	0.49
60	6.3	22	5.07	1.23
90	10.76	25	8.65	2.11
120	15.78	28	12.92	2.86
150	22.9	30	17.30	5.6

Comparison of experimental total power loss for SAE30 and SAE40 lubricants with the variation of oil:

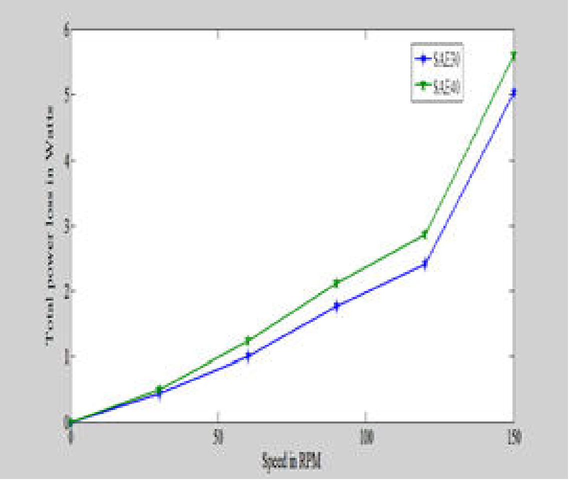

Figure 3: graph of comparison of total power loss

Comparison between experimental and theoretical total power loss For SAE30 with the variation of speed:

Here theoretical power loss values are less as compared to experimental total power loss at any particular speed.

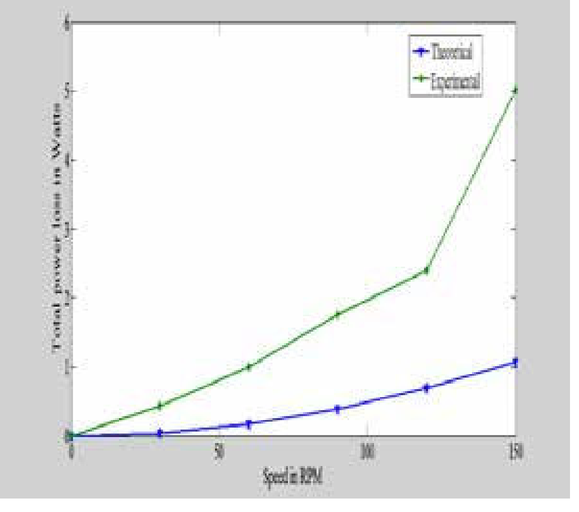

Figure 4: graph of comparison of total power loss

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Comparison between experimental and theoretical total power loss For SAE40 with the variation of speed:

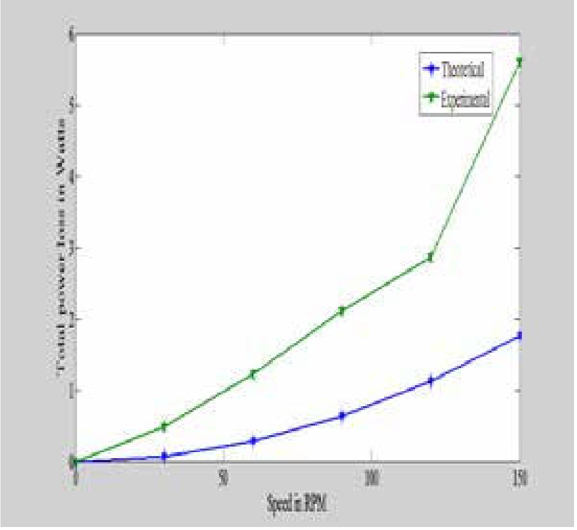

Figure 5: graph of comparison of total power loss

CONCLUSION:

The analysis of power loss data revealed that all three operating parameters density, viscosity and speed influence the total power losses from the transmission significantly.

Specifically:

- Total power loss increases with the increase of speed. Likewise increased oil viscosity causes increased total power loss.
- Theoretical power loss is less than the experimental power loss. With the rise of speed difference between theoretical and experimental power losses increases.

RECOMMENDATION FOR FUTURE WORK:

The theoretical database generated in this study exhibited clear trends in term of influence of operating parameters on transmission power loss. In order to bring a complete understanding to the measured power loss behavior, this experimental analysis must be done with the higher speeds.

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