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Parallel-axis gear design methodology for minimization of power loss and preliminary study of its effect on vibration characteristics

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Abstract

Gear tooth strength is mainly considered in the gear design process to ensure the ability to transmit power. With the design process, various sets of gear parameters are probably selected to meet the tooth strength. However the efficiencies of various designed gears are different. Improper gear parameter selection probably makes the gear power loss increase significantly. In this paper, the design methodology to minimize gear power loss is presented. A spur gear selected from a gear catalog is used as the reference gear. Then several gears with various parameters but having the ability to transmit the same load are designed. The power losses of the designed and the reference gears are estimated by the sliding loss model, hence the minimum power loss gear is able to choose from the various designed gear set. The results show that to minimize gear power loss along with keeping loading capacity, pressure angle and face width should be increased and module should be reduced. The preliminary study of this design methodology on vibration characteristics is done by measuring the vibration attributed to two sample gear sets at outer surface of the gearbox. It is found that the helical gear having larger pressure angle and face width than the spur gear has more capability to transmit load, lower power loss and also lower vibration.

Keywords: gear design, load capacity, power loss, vibration

1. Introduction

The method to design gear has been well established and has been widely published in many of gear handbooks and machine design handbooks. The most popular and widely used method to design gear was suggested by American Gear Manufacturer Association (AGMA). In this method gear tooth strength is mainly focused to ensure the ability to transmit power at a specific operating condition and lifetime reliability. With this consideration, various sets of gear parameter are probably selected to meet the tooth strength. However the power losses of these various gear sets are different. Improper gear parameter selection probably makes the power loss of the designed gear increase significantly.

With the important of the energy problem, there are many researches studied about the gear power loss and the method to increase the efficiency of the gear box. The former researches of authors' group [1-3] show the effect of gear geometrical parameters on the sliding loss of a spur and a helical gear pair. The sliding loss can be reduced by increasing the pressure angle, face width and reducing module. Höhn et al. [4] studied the method to increase the efficiency of a gearbox. He suggested using the helical gear with smaller module, larger pressure angle and face width instead of the standard C-type spur gear to reduce the load gear losses. Although his results give very useful suggestion to increase the gear box efficiency, the load capacity of gears is not considered in his study. Therefore it is still difficult to apply the results to the other cases.

From the former studies described above, gear geometrical parameters relate closely with the gear power loss, therefore in the gear design process it is possible to choose the proper parameters to obtain the low power loss gear along with keeping its loading capacity. In this study, the design methodology to minimize gear power loss is presented. The several gear sets having nearly the same load capacities are designed based on the AGMA suggestion. The power loss of the designed gear pairs are estimated with the gear sliding loss model proposed in the ref. [3], hence the minimum power loss gear is able to choose among the various designed gear set. Because gear vibration is also probably affected with this design methodology, the preliminary study of the vibration characteristics of the gear pairs is also done in this study.

2. Gear strength calculation

The gear design methodology suggested by AGMA is used here. Two stress equations used in this method are bending stress equation and contact stress equation. These AGMA equations can be written in the form of allowable bending load (W_t) and allowable contact load ($W_{t,c}$) as shown by equations

$$W_t = \frac{\sigma \cdot b m_t}{K_O K_v K_s} \frac{Y_J}{K_H K_B} \frac{Y_\theta Y_Z}{Y_N} \tag{1}$$

and

$$W_{t,c} = \left(\frac{\sigma_c}{Z_E} \cdot \frac{Y_\theta Y_Z}{Y_N} \right)^2 \cdot \frac{b d_{\omega l}}{K_O K_v K_s} \frac{Z_I}{Z_R} . \tag{2}$$

In these equations, σ and σ_c are allowable bending stress and allowable contact stress. b is face width. m_t is transverse module. K_O, K_v, K_s, K_H and K_B are overload factor, dynamic factor, size factor, load-distribution factor, rim-thickness factor, respectively. Y_J is geometry factor for bending strength. Y_θ, Y_Z and Y_N are temperature factors, reliability factor and stress cycle factor for bending stress. Z_E is elastic coefficient. $d_{\omega 1}$ is pitch diameter of the pinion. Z_I and Z_R are geometry factor for pitting resistance and surface condition factor, respectively.

Parameters in the Eqs.(1) and (2) can be categorized into the geometrical dependent parameters and geometrical independent parameters as shown in Table 1. Since the objective of this study is to design the low power loss gear to replace the reference spur gear, operating conditions, gear configuration, gear material and surface condition of the low power loss gear and the reference gear are considered to be the same. With this reason only the geometrical dependent parameters are considered in the design process. The effects of face width b and transverse module m_t on the load capacity are directly known from Eqs.(1) and (2), whereas the effect of pressure angle and helix angle are known from parameters Y_J and Z_I. The effects of increasing gear geometrical parameters on the load capacity are summarized as shown in Table 2. This information is used for parameter selection in the design process described further.

Table 1 Geometrical dependent parameters and independent parameters

	Bending stress equation	Contact stress equation
Geometrical dependent parameters	b, m, Y_J	$b, d_{\omega l}, Z_I$
Geometrical independent parameters	$K_O K_v K_s$ $K_H K_B Y_\theta Y_Z Y_N$	$K_O K_v K_s$ $Y_\theta Y_Z Y_N Z_E Z_R$

Table 2 The effects of gear geometrical parameters on the load capacity

Geometrical parameters	Bending load	Contact load
Module (Increase)	Increase	Increase
Pressure angle (Increase)	Increase	Increase
Helix angle (Increase)	Max. at helix angle = 10°-15°	Increase
Face width (Increase)	Increase	Increase

3. Power loss in gear transmission

Power loss in gear transmission can be categorized into sliding loss, rolling loss, churning loss and windage loss. From many former studies it has known that the rolling loss is much less than the sliding loss [5], and the windage loss is very low in the case of small gear operated at low or moderate speed [6] such as the gears used in an automobile or an agricultural machine. Moreover the churning loss depends not only on the gear geometrical parameters and operating conditions, but also the configuration of the gear box and lubricating method [7] that are out of scope of this study. Therefore only the sliding loss that is the dominant loss in the gear transmission is considered here.

The method to estimate the sliding loss used here is the same as the method presented in the ref. [3]. From these researches, the effects of increasing gear geometrical parameters on the sliding loss are known and can be summarized in the table 3. The sliding loss is increased when the module is increased. On the other hand, increasing gear pressure angle and face width will reduce the sliding loss.

Table 3 The effect of gear geometrical parameters on the gear sliding loss

Geometrical parameters	Sliding loss
Module (Increase)	Increase
Pressure angle (Increase)	Decrease
Helix angle (Increase)	Indefinite
Face width (Increase)	Decrease

4. Gear design methodology

From the effect of geometrical parameters on the load capacity and the gear sliding loss in tables 2 and 3, increasing pressure angle and face width brings the positive results both in view of gear strength and gear power loss. Hence these parameters should be set to be as large as possible. For the module, since the reduction of this parameter decreases sliding loss significantly, the module should also be selected to be as small as possible. The amount of load capacity that will be reduced when the module is reduced can be compensated by increasing pressure angle, face width and also helix angle.

The procedure of gear design in this study is shown in Fig. 1. First gear specifications that are load, operating speed, center distance, gear ratio and gear material are defined. With these specifications the reference spur gear can be selected from a gear catalogue. To design the low power loss gear to replace the reference gear, the center distance and the gear ratio are fixed to be the same as the reference gear, but the other gear parameters are changed as described in the following steps.

1. <u>Decrease module:</u> Since the number of teeth is increased when the module is decreased, the size of the gear is also probably changed. The center distance of

the gear pair must be carefully check to ensure that the design gear is probable to replace the reference gear.

2. Increase pressure angle: The standard gear pressure angles are 14.5°, 20° and 25°. Here the pressure angle is set to be 20° or 25° to obtain high load capacity along with low power loss.

3. Increase face width: Although increasing the face width gives positive results both in view of gear strength and gear power loss, the increasing of the face width still has the upper limit. Many design textbooks suggest that the face width should be around 8 to 16 times of the module [8]. The excessive face width probably leads to the force distribution and tooth bending problem. In this study the face widths are set at 12-15 times of the module depended on the value of module. These face widths are equal or wider than the face width of the reference gear.

4. Adjust helix angle: Since the proper amount of helix angle cannot be known directly, the calculation here is done by varying the amount of helix angle from 0° to 30°. The optimum helix angle can be chosen further by considering the load capacity along with the estimated sliding loss.

After adjusting the geometrical parameters, gear tooth strength is calculated by the AGMA method. This result is compared to the load capacity of the reference gear. If the tooth strength of the designed gear is lower than that of the reference gear, the parameters will be changed to obtain higher load capacity. On the other hand if the load capacity of the designed gear is much more than that of the reference gear, it is possible to reduce gear module to decrease the sliding loss. The amount of the sliding loss of the designed gear is estimated by the gear sliding loss model [3].

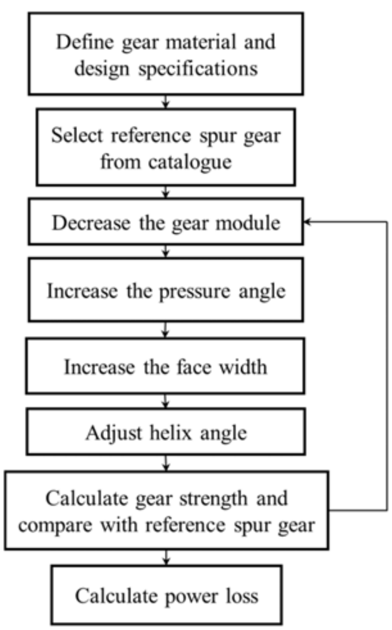

Figure 1 Procedure of gear design for minimization of power loss

5. Example of gear design

5.1 The reference gear and the designed gears

The specifications of the gear pair designed here are shown in table 4. From these specifications, the reference spur gear SSG3-30J30 having module 3 mm, pressure angle 20°, and face width 30 mm is selected from KHK's gear catalogue [9]. The transmitted load capacity of this gear from the catalogue is 228 Nm for the bending strength, and 138 Nm for the surface durability. The load capacity of this gear calculated by AGMA method equals to 285 Nm for the bending strength, and 156 Nm for the surface durability. The slight differences between the catalogue readings and the calculated loads come from the differences of various factors and the properties of materials used in calculation.

Table 4 Specifications of the designed gear.

Specifications	Values
Speed	2500 rpm
Transmitted torque	130 Nm
Center distance	90 mm
Gear ratio	1:1
Working temperature	70°C
Life-cycles	2×10^6 cycles
Material	S45C

To design the low power loss gear, the module is tried reducing from 3 mm to 2.5 and 2 mm. The pressure angle is set to be 20° and also increased to be 25°. For the gear having module 2.5, the face width is set at 30 and 35 mm that equals to 12 times and 14 times of the module. On the other hand for module 2 mm, the face width is kept at 30 mm that equals to 15 times of the module. The values of helix angle are varied from 0° (spur gear) to 30°.

5.2 Load capacity and power loss estimation

The capacities of gears designed by this method are calculated and shown in Fig.2 for the bending stresses and in Fig.3 for the contact stresses. In these figures the allowable torques of the reference gear calculated by the same method are shown by the dash lines. Most of the designed gears have higher load capacities than the reference gear except two spur gears having module 2 mm, the spur gear having module 2.5 mm, pressure angle 20°, and face width 30 mm, and also the helical gear having module 2 mm and helix angle 30°. It is obvious that the load capacities of the helical gears are much more than the spur gear. The gears having helix angle about 10°-20° have the highest load capacities. The load capacities is reduces when the helix angle is larger than 20°. Reduction of the module affects significantly to the allowable torque calculated from bending stress, but affects only a little to the allowable torque calculated from the contact stress. Increasing of pressure angle and face width increase allowable torque for both bending stress and contact stress evidently.

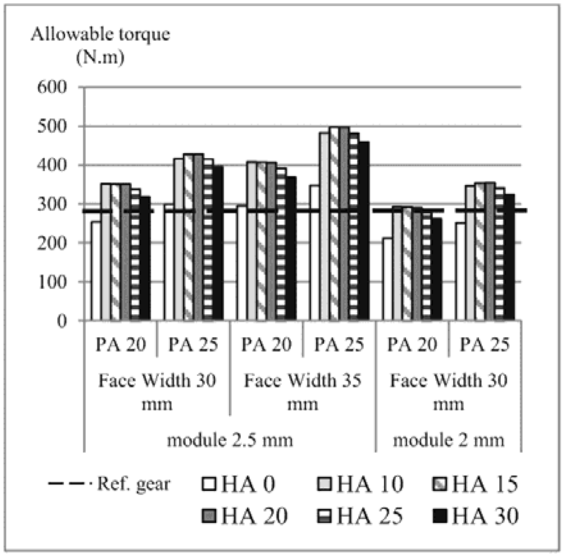

Figure 2 Allowable torques for bending stresses

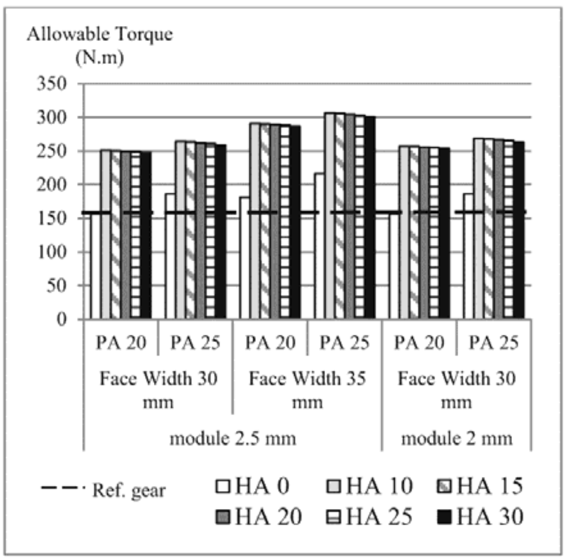

Figure 3 Allowable torques for contact stresses

The estimated sliding losses of the designed gears are shown in Fig. 4. All gears have sliding loss lower than the reference gear. Reduction of the module and increasing the pressure angle reduce the power loss considerably, whereas widening the face width does not much affect to the power loss. The helical gears having module 2 mm, pressure angle 25° and helix angle 10° is the lowest power loss gear among all gears having larger load capacity than the reference gear. This gear has the power loss less than the reference gear approximately 37.5%. The helical gears having the helix angle 15° or 20° with the same module and pressure angle with the lowest power loss gear are also recommended to use since their high load capacities and low power losses. It can be concluded from these results that the proper gear parameter selection in the design state will give the high load capacity gear and can reduce the power loss more than 35%.

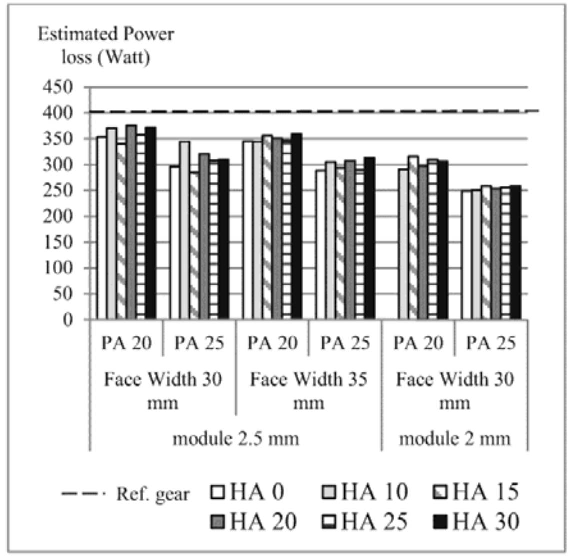

Figure 4 Sliding losses of the designed gears

6. Preliminary study of vibration characteristics

6.1 Parameters of test gears

Since the design methodology presented here is focused on the load capacities and power losses of gears, it is also important to investigate the effect of this design methodology on the other aspects. In this study the vibration characteristics of the designed gears are preliminary investigated, and some measuring results are presented here. Because the productions of designed gears in the former section are ongoing, the results shown here are obtained from the experiments done by the gear pairs already used in the laboratory. The parameters of sample gears are shown in Table 6.

Table 6 The parameters of sample gears used in vibration experiments

Parameters	Set A	Set B
Diameter (mm.)	90	90
Number of teeth	30	30
Transverse module (mm)	3	3
Helix angle (deg.)	0	33.5
Pressure angle (deg.)	20	25
Face width (mm.)	20	30
Gear ratio	1:1	1:1
Bending strength (Nm)	191.50	450.50
Surface durability (Nm)	104.88	186.13

The spur gear A in table 6 is considered as the reference gear, and the helical gear B having larger pressure angle and wider face width is considered as the designed gear. The strengths and the power losses of these gears are also shown in Table 6 and in Fig.5, respectively. It is obvious that the gear B has higher strength and lower sliding loss than the gear A as expected.

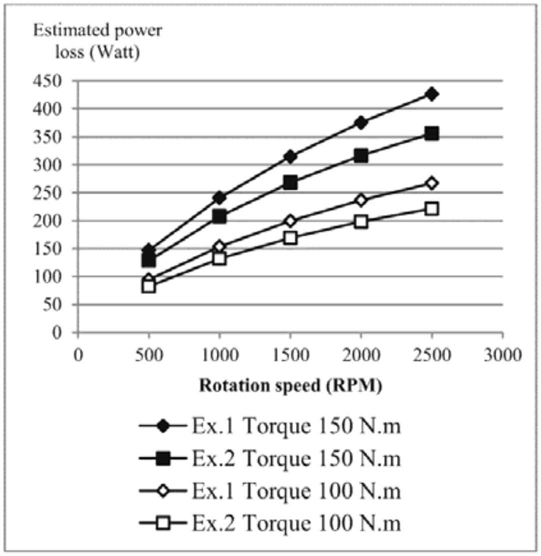

Figure 5 Estimated power losses of the example gears

6.2 Vibration investigating method

The sources of gear vibration can be categorized roughly into 3 sources [10], and the vibration attributed from the different sources occurs at different frequencies. The first source of gear vibration comes from the variation of meshing stiffness and common tooth surface form. The vibration attributed to this source is found at meshing frequency and its harmonics. The second source is the fluctuating deviation from common errors such as pitch error or misalignment of gear body on the shaft. This source brings about the vibration at low orders of shaft rotations and also the sidebands surrounding the meshing frequency and harmonics. The last source of gear vibration is the tooth surface undulation. The vibration from this source occurs at various harmonics of shaft frequency. Since the investigation here is scoped only the effect of design parameters that are common on all gear teeth, the source of vibration excitation comes from the first source, hence only meshing frequency and its harmonics are extracted and considered here.

The experiments were done by the back-to-back gear test rig as shown in Fig.6 as same as the experiments done in the ref. [1]. The details of the apparatus and the test method can be seen in the reference. An accelerometer was attached on outer surfaces of the gearbox 1 in Fig.6 at the position 1, 2 and 3 as shown in Fig.7 to measure the vertical, transverse and axial vibration, respectively. The acceleration signal was measured at one direction first, when the measurement was finished then the further directions were measured. The measurements were done at applied torque 150 Nm and the rotational speed 800-2000 rpm.

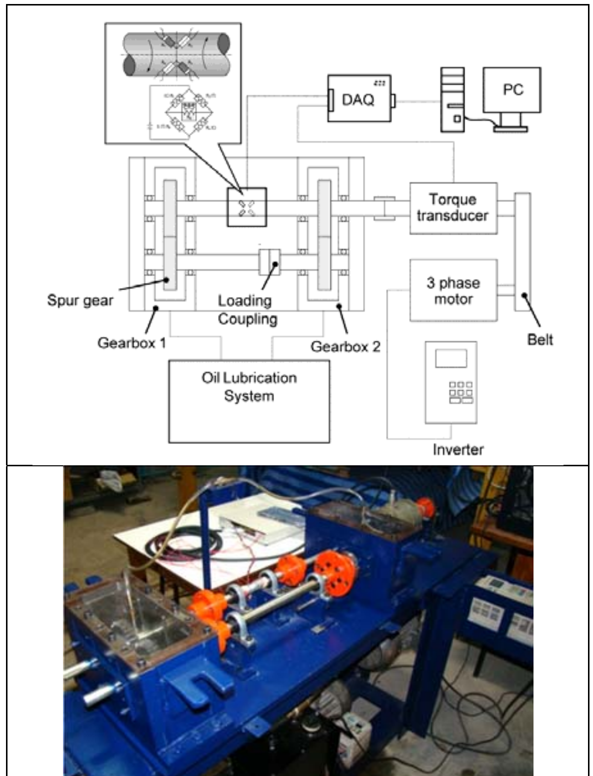

Figure 6 Back-to-back gear test rig [1]

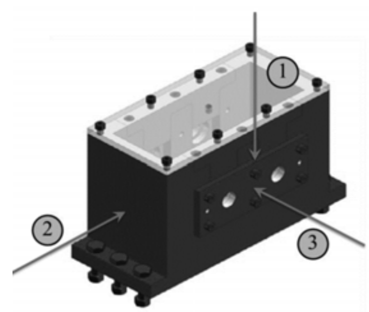

Figure 7 Positions of the accelerometer on the gearbox surface

6.3 Vibration measuring results

Figures 8 and 9 show the accelerations of meshing components of gear set A and B. The vibration amplitudes of the spur gear set A are slightly higher than the helical gear set B. The trend of vibration amplitude is also clarified by the rms value of meshing components shown in Fig.10. The vibration reduction in this example is attributed to the changing from the spur gear to the helical gear. Although the increasing of pressure angle will increase the vibration amplitude, this effect on vibration is considerably small comparing to the effect of helix angle. These results also verify the merits of the proper gear parameter selection. Not only the higher load capacity and lower power loss are probably acquired but lower vibration amplitude is also achieved from this design methodology.

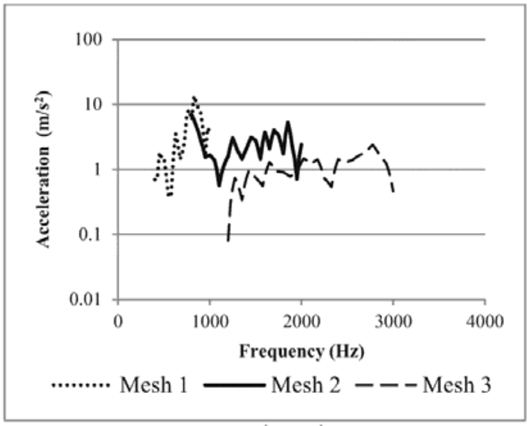

Figure 8 The meshing frequencies of the gear set A

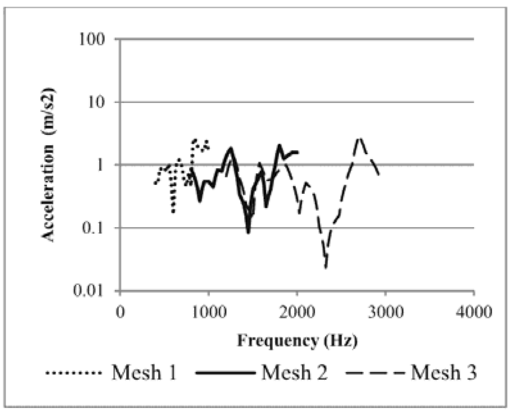

Figure 9 The meshing frequencies of the gear set B

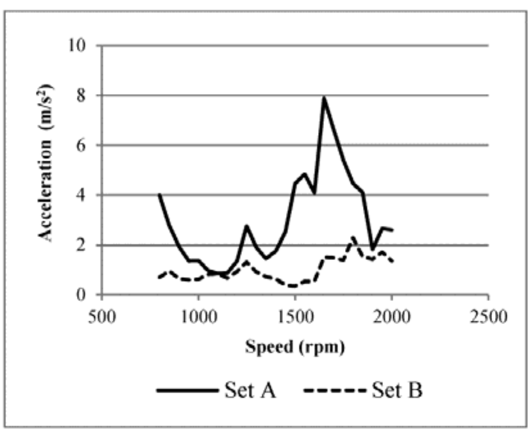

Figure 10 Comparison of vibrations (rms) of the gear A and B

7. Conclusions

The design methodology for minimization of power loss along with keeping the load capacity of gears is presented in this study. The results reveal that it is possible to achieve the high load capacity, low power loss and also low vibration gear by proper gear parameter selection in the design stage. In this design, the small module, large pressure angle, and wide face width are suggested. The helical gear is also desired due to its high load capacity and low vibration characteristics.

8. Acknowledgement

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9. References

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