Design investigations on a 2-speed gear reducer for agricultural application

J L Luces¹ and M C E Manuel²

¹Department of Science and Technology - Metals Industry Research and Development Center, Gen. Santos Ave., Bicutan, Taguig City, 1631, Philippines
²School of Mechanical and Manufacturing Engineering, Mapua University, Muralla St., Intramuros, Manila 1002, Philippines

joein.mirdc@yahoo.com, mcemanuel.mapua@gmail.com

Abstract. A dual-speed gear reducer was designed to offer a broader application to the common single-speed reducers. However, in order to provide additional value to the design, an investigation was made to determine the effects of varying the normal module, center distance, and facewidth on the gear components of the dual-speed gear reducer. KISSsoft gear simulation software was used to perform the computation for each variation in the mentioned parameters. The optimized values were based on the calculated parameters of gears with the root and flank safeties of not less than one or greater than 3000 hours service life as minimum standard for designing gears used for agricultural application. Based on the investigations made the increase in normal module, center distance, and facewidth also increases the flank and root safeties of both drive and driven gears. However, increasing further the normal module decreases the flank safeties of both gears. Nevertheless, increase in normal module with corresponding increase in center distance considerably increases both the root and flank safeties of gears. Finally, a dual-speed gear reducer was fabricated that proved the data generated using KISSsoft was useful for efficient manufacturing and assembly of the gears and gearbox, respectively.

1. Introduction

Torque and speed requirements in the industrial facilities, fishery, and agricultural machineries paved the way for the development of speed reducers. In agricultural processes, ranging from land preparation until post-harvest, machines performing the operations require a gear reducer to decrease the speed of normally high rpm prime movers. Zimmerman [1] established that for tractor plowing, the lower speeds are more desirable than the higher speeds. Primary tillage should be done at a slower speed of 4-6 km/hr and the secondary working such as offset disking is best done at higher speed of 7-8 km/hr (IRRI). This is supported by the Philippine Agricultural Engineering Standards (PAES) 118: 2001 which requires tractors to pull field implements at a maximum speed of 8 km/h. In post-harvest operation, such as mechanical drying using rotary type paddy dryer, the machine requires angular speeds of approximately 15-20 rpm which is far from the usual speed of electric motors which ranges from 900 - 3600 rpm and that of single cylinder engines which have more or less 3600 rpm [2]. Another agricultural machine that requires slower speed is the juice extractor for sugar cane. In a patent filed by Velumani Aravind [3], the machine used a gearbox to reduce the speed of the two different motors of about 900 and 1440 rpm to an output of 10 to 50 rpm of a juice extractor.
From these, it is apparent that reduction of speed for high-speed prime movers is required which will consequently increase torque of low power engines or motors. Nevertheless, in a more practical sense, it would be of paramount importance to optimize the use of a speed reducer by providing a dual speed output to the apparatus. This will reduce the cost of acquiring two gear reducers to serve two different purposes. This would be beneficial particularly among low income earner farmers.

One feature in gear reducer which deserved to be taken into consideration is the importance of helix angle among helical gears. Helical gears have several teeth engaged constantly that transmits considerably high torque and reduce stress among gear teeth due to the load distribution through the gear teeth. It also produces less noise to about 10-12 dB more quiet than spur gearing [4]. It is therefore essential to conduct investigation on the optimum helix angle by which a helical gear will be on its optimum performance. Facewidth on the other hand is another factor that influences the life cycle of a gear. In the study conducted by Lateno [5], he observed that the maximum bending stress and contact stress decreases with increasing face width, number of teeth and root fillet radius relative to spur gear set weight. Module also deserves to be considered in designing gears. It is known that the larger the module, the stronger the gear hence, the higher the load it could carry. Lastly, center distance between gear pairs should also be given equally utmost attention. It is important to note that center distance plays a vital role in gear designs since larger center distance also means larger gears.

In order to have an efficient and speedy calculation using a combination of certain gear parameters of specified range, simulation or computational softwares are employed in the process. Kiekbusch et al. [6] presented a paper on the development of detailed two- and three-dimensional finite element models used to determine the torsional mesh stiffness of spur gears. He made use of the finite element software ANSYS to generate different pairs of spur gears which included the adaptive meshing algorithm for the contact zones. He used 2D models to simulate varied gear pairs in a short period of time. He concluded that using a 3D model it is possible to simulate helical gears or apply modifications in tooth face such as face angle corrections or crowning. He added that the effect of meshing interferences can be studied. Likewise, he concluded that the resulting values from the finite element models and the mesh stiffness formula can be used in the dynamic simulations such as multi body simulation of gearboxes. The advantage of the developed formula is that only the basic gearing conditions are needed to obtain the torsional mesh stiffness. The finite element models presented the option to evaluate stresses in crucial areas of gears. The results showed that both the finite element models and the mesh stiffness formula are credible.

Jia, Howard and Wang [7] presented a paper regarding the dynamic modeling of multiple pairs of spur gears in mesh, including friction and geometrical errors. A dynamic model of two pair of gears in mesh with three shafts, 26 degrees of freedom, friction, effects of variable tooth stiffness, pitch and profile errors, and a localized tooth crack on one of the gears were presented in the article. It also explained how to include in the model the geometrical errors in teeth which also incorporates the effects of variations in torsional mesh stiffness in gear teeth by using a common formula to elucidate stiffness that occurs during the meshing of gears. The determination of the occurrence of geometrical errors in teeth was made using MatLab and Simulink models, developed out of the equations of motion. It also discussed the effects of pitch and profile errors on the resultant input pinion angular velocity. The simulation results revealed that pitch and profile errors of 10 microns have a substantial impact on the vibration of gearbox. The geometrical errors are noticed to provide extra energy into the harmonics of the tooth mesh and associated sidebands, shadowing the changes in modulation caused by localised damage in tooth like tooth crack. It was observed that tooth crack of 5mm in length was difficult to detect using 10 microns geometrical pitch and profile errors.

Lahtivirta, J. and A. Lehtovaara [8] emphasized the important role of finite element method (FEM) in the modelling and simulating of the gear pair in order to approximate deformations, stresses, and damage risks in different working conditions as part of the design process. To analyze stresses in the gear root and the gear contact, they developed a parameterized calculation model. The local adaptive FE mesh they used sped up the computation. They created in Matlab environment the rotation of the gear pair, the adaptive FE mesh, and the accurate surface profile based on gear hobbing process to obtain a
good control of the flank profile and the meshing parameters. They then integrated the method they used to a commercial FEM software to calculate stresses and deformations. They successfully validated the FE mesh approach they developed against analytical Hertzian theory. They also compared the developed spur gear model with ISO 6336 gear standard and a commercial gear simulation software (KISSsoft) and turned out a relatively good conformity results with the maximum contact pressure and the maximum tooth root stresses.

Yu-Ren Wu and Van-The Tran [9] mentioned that the tooth surfaces of gears are usually crowned with twist-free tooth flanks to reduce the vibration and noise of the gear systems in the high-speed rotational gearbox. With this they proposed an innovative method for longitudinal crowning tooth flank of helical gear. They constructed a linear crossed-angle function for honing external gear. They used a variable pressure angle honing cutter for free-twist of the crowned tooth flank. They investigated transmission and load characteristics of meshing gear pair with twist-free tooth flank. They developed 3-D models of twisted and twist-free meshing gear pairs and analyzed them using KISSsoft. Their study revealed that the working conditions of the meshing gear pair have substantial improvements using the proposed gear honing method.

The study of Jani and Shah [10] focused on the theoretical and finite element method to calculate bending and contact stresses on the tooth geometry of major effected parameters on helical gear pair. They used various approaches to calculate the wear and bending effects of gear failure cause in static condition using finite element analysis, AGMA standard and ANSYS. They checked the root and flank safety of a helical gear by calculating their strength manually using Lewis’s equation and KISSsoft software for the gear system.

Cosmin Petra [11] used KISSsoft to study noise reduction in spur gears transmission. His motivation focused on several characteristics of transmissions such as high efficiency, high load carrying capacity, and low sound excitation behavior. His aim was to optimize the helical gear pair to achieve the best possible noise and contact ratio. He stressed that the well-known source of vibration was the disparity of the tooth pair stiffness during meshing and appropriate measures should be applied to reduce the excitation.

In this research, a dual-speed gear reducer for a 12 HP diesel engine developed by MIRDC [12] for agricultural application was designed. Investigations on the effects of varying the normal module, center distance, and facewidth for each of the gear components was done numerically using KISSsoft gear software [13]. This was done to achieve the optimal sizes of gear components that would result to the minimal size of gear box which consequently reduce material and manufacturing costs.

2. Design Constraints and Limitations

The gear reducer was developed to increase the torque while reducing the speed of a generally higher rpm prime movers. A dual speed function was introduced to the gear reducer for broader scope of applications. The gear reducer was made of helical gears for smooth and quiet operation, higher efficiency, lesser manufacturing cost, and improved gear life. The design parameters were basically centered on the number of gear pairs used along with its corresponding number of shafts and bearings. The parameters tested for the gear components was normal module, center distance, and facewidth. Said gear components were designed using KISSsoft gear software while the modelling and drafting made use of NX 3D modelling software. The optimized values were based on the calculated parameters of gears with the root and flank safeties of not less than 1 or greater than 3000 hours service life. The power input, pressure angle, and helix angle were limited to 12 HP, 20 degrees, and 30 degrees, respectively. The maximum diameter of gears was limited only to 250 mm. The dual speed ratios were 1:6 and 1:12. The material used in this study was 1045 medium carbon steel. The gear reducer was developed for 12 HP-rated agricultural machines using helical gears.
3. Design Development and Gear Pairs Optimization

The number of gears, shafts, and bearings, as conceptualized, were added to the module in KISSsys. These are among the basic requirements of the software in order for it to perform the necessary calculation. Preliminary analysis on three (3), five (5), and seven (7) gear pairs was conducted to determine which design was appropriate, that also served as the basis of the selection and further investigation. The computation revealed that the five gear pairs was the most applicable given the limitations of the study and considering its weight. One gear of the three gear pairs have exceeded the tip diameter of 250 mm and the set also have heavy gears. On the other hand, the tip diameters of the seven gear pairs are all within the range but the weight is greater than that of the five gear pairs. Table 1 depicts data on the preliminary calculation made using KISSsoft. Additional investigations were performed to choose the optimal set of parameters. The lightest gear pairs with root and flank service life greater than 3,000 hours was desired for this study. The normal module was set from 1 to 5 with 0.25 mm increment after completion of each set of calculation. The center distance, on the other hand, was set from 100 to 200mm with 1mm increment after each set of calculation. Furthermore, facewidth was set from 30 to 80mm with 1mm increment after calculating each set. The summary of the optimal set of parameters is presented in Table 2 which include the center distance, face width, normal module, root safeties of drive and driven gears, flank safeties of drive and driven gears, root service life and flank service life. The mock-up of an experimental apparatus for further testing is shown in Figure 1.

Table 1. Preliminary calculation using KISSsoft.

| Design | Gear Pair Ratio | Center Distance (mm) | Facewidth (mm) | Normal Module (mm) | Tip Diameter1 (mm) | Tip Diameter2 (mm) | Root Safety Gear1 | Root Safety Gear2 | Flank Safety Gear1 | Flank Safety Gear2 | Root Service Life (hr) | Flank Service Life (hr) |
|--------|----------------|----------------------|----------------|--------------------|-------------------|-------------------|------------------|------------------|------------------|------------------|---------------------|---------------------|
| 1      | 6              | 263                  | 100            | 1.75               | 79.68             | 453.31            | 2.84             | 2.66             | 1.00             | 1.16             | 127.72              | 1000000             |
| 2a     | 1              | 209                  | 25             | 1.25               | 200.97            | 203.89            | 1.41             | 1.41             | 1.09             | 1.09             | 12.26               | 1000000             |
| 2b     | 2              | 209                  | 40             | 1.25               | 134.93            | 270.51            | 1.50             | 1.48             | 1.04             | 1.04             | 21.99               | 1000000             |
| Final  | 6/12           | 101.87               |                |                    |                   |                   |                  |                  |                  |                  |                     |                     |
| 2      | 1.5            | 209                  | 38             | 2                  | 146.88            | 216.01            | 3.07             | 3.13             | 1.01             | 1.02             | 28.18               | 1000000             |
| 2      | 2              | 186                  | 69             | 1                  | 128.53            | 248.36            | 2.70             | 2.65             | 1.01             | 1.05             | 31.52               | 1000000             |
| 2      | 186            | 65                   | 1              | 128.53             | 248.36            | 1.67             | 1.68             | 1.00             | 1.05             | 31.05               | 1000000             |
| 2      | 2              | 186                  | 60             | 1                  | 106.59            | 210.34            | 1.67             | 1.68             | 1.00             | 1.05             | 31.05               | 1000000             |
| Final  | 6/12           | 120.64               |                |                    |                   |                   |                  |                  |                  |                  |                     |                     |
| 3      | 1.5            | 178                  | 30             | 1.75               | 146.88            | 216.01            | 2.31             | 2.34             | 1.00             | 1.00             | 12.07               | 1000000             |
| 2      | 1.5            | 197                  | 35             | 1.5                | 160.75            | 239.20            | 1.75             | 1.75             | 1.00             | 1.04             | 17.31               | 1000000             |
| 3      | 1.5            | 207                  | 55             | 1.5                | 170.10            | 249.88            | 1.50             | 1.49             | 1.00             | 1.06             | 29.96               | 1000000             |
| 4      | 2              | 185                  | 100            | 2                  | 128.35            | 249.65            | 1.44             | 1.44             | 0.84             | 0.90             | 46.39               | 1000000             |
| 5      | 2              | 106                  | 100            | 5                  | 81.73             | 150.05            | 1.76             | 1.76             | 1.00             | 1.06             | 14.93               | 1000000             |
| 6a     | 1              | 219                  | 100            | 2.5                | 223.50            | 224.50            | 1.72             | 1.72             | 0.82             | 0.82             | 58.65               | 1000000             |
| 6b     | 2              | 106                  | 100            | 5                  | 81.73             | 150.05            | 1.76             | 1.76             | 1.00             | 1.06             | 14.93               | 1000000             |
| Final  | 6/12           | 194.24               |                |                    |                   |                   |                  |                  |                  |                  |                     |                     |

Table 2. The optimized gear parameters based on simulation results.
4. Design Investigations

Investigations on how the parameters interact with the responses were further performed by varying the center distance, normal module, and facewidth. The effect to the root and flank safeties of gears where then observed. Sample plots from the investigations are shown in Figures 2, 3, and 4. Results show the forward-leaning curve which means that the increase in center distance improves the root safety of gears. In addition, it could be noticed that as the module increases, it further enhances the root safeties of both gears. Center distance had similar effects to flank safety of gears at increasing module. However, varying the module has no significant effect to the flank safeties of both gears.

Moreover, as the normal module increased further, given a certain limit in center distance, the flank safety of gears decreases. This means that an optimum normal module should be selected in order to have a better design of gears. Furthermore, increase in facewidth improves both the flank and root safeties of both gears. However, an optimum facewidth should also be investigated to optimize the size of gears for reduced weight and consequently, cost. Preliminary investigation also revealed that increase in helix angle also improved the root and flank safeties of gears however, due to machine restriction, the maximum helix angle tested was limited until 30°.

Figure 2. Effect of center distance to (a) root safety and (b) flank safety of drive gear at increasing module (facewidth=30).
Figure 3. Effect of center distance to (a) root safety and (b) flank safety of drive gear at increasing module (facewidth=55).

Figure 4. Effect of center distance to (a) root safety and (b) flank safety of drive gear at increasing module (facewidth=80).

Figure 5. The fabricated 2-speed gear reducer.
5. Conclusions
A dual-speed gear reducer for 12 HP single cylinder diesel engine was successfully designed using KISSsoft/KISSsys gear software and NX 3D modelling software. Based on the investigations made the following conclusions are drawn:
1. Increase in facewidth, center distance and normal module also increases the root safeties of both drive and driven gears.
2. Increase in facewidth and center distance increases the flank safeties of both drive and driven gears however, increase in normal module has no significant effect on the flank safety of gears.
3. Further increase in normal module decreases the flank safeties of both gears therefore, an optimal normal module should be determined.
4. Increase in normal module with corresponding increase in center distance considerably increase both the root and flank safeties of gears.
5. Increase in helix angle until 30° also increases root and flank safeties.
6. The computation process in KISSsoft was useful in determining the optimized normal module, center distance and facewidth which further helped in determining gear pairs with lesser weight.
7. A dual-speed gear reducer was developed that proved the data generated was useful for efficient manufacturing and assembly of the gears and gearbox, respectively.

References
[1] Zimmerman O B 1920 Tractor Plowing Speeds (SAE Trans.) 471–85
[2] Johnson K G E, Mollenhauer K, Tschöke H 2010 Handbook of Diesel Engines (Springer Science & Business Media)
[3] Aravind V 2008 Sugar cane crushing and juice vending machine (WIPO PCT WO2009128086A1)
[4] Mallenkamp A 2017 Machine design: Getting the most out of gearboxes penton Last updated February 9, 2012 Accessed April 5, 2017 http://www.machinedesign.com/mechanical-drives/getting-most-out-gearboxes
[5] Lateno E 2014 Effect of change of spur gear tooth parameters on bending and contact stresses MS thesis (Ethiopia: Addis Ababa University, Addis Ababa Institute of Technology School of Mechanical and Industrial Engineering) http://etd.aau.edu.et/bitstream/123456789/5397/1/Esayas%20Lateno.pdf
[6] Kiekbusch et al. 2011 Calculation of the combined torsional mesh stiffness of spur gears with two- and three-dimensional parametrical FE models Strojniški Vestnik/Journal of Mechanical Engineering 57 11 810–818 doi:10.5545/sv-jme.2010.248
[7] Jia S, Howard I, and Wang J 2003 The dynamic modeling of multiple pairs of spur gears in mesh, including friction and geometrical errors The International Journal of Rotating Machinery 9 6 437–442 doi:10.1155/S1023621X03000423
[8] Lahtivirta J and Lehtovaara I A 2016 Modelling of spur gear contact using a local adaptive finite element mesh TRIBOLOGIA Finnish Journal of Tribology 34 no 1-2
[9] Wu Y and Tran V 2016 Transmission and load analysis for a crowned helical gear pair with twist-free tooth flanks generated by an external gear honing machine Mechanism and Machine Theory 98 36-47 doi:10.1016/j.mechmachtheory.2015.11.014
[10] Jani S and Shah J 2017 Design, modelling and analysis of helical gear pair using ANSYS and AGMA standards for calculating a bending and contact stress on gear profiles International Journal of Advance Research and Innovative Ideas in Education 3 3813-3821
[11] Cosmin P 2011 Numerical research in kisssoft for noise reduction in spur gears transmissions The 5th Edition of the Interdisciplinarity in Engineering International Conference
[12] Puerto J Q, Luces J L, Limson A J S, Dime F C, Liza F P An approach for the development of a locally made single cylinder diesel Engine Philippine Metals 2016;3:46–50
[13] KISSsoft AG 2016 KiSSsoft Release 03/2016 - User Manual no. 03/2016: 1209