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A VISUAL BASIC PROGRAM FOR DESIGNING OF GEAR PAIR FOR FRONT POWER TAKE-OFF UNIT OF HIGHER HP TRACTORS

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Abstract: With a view to fully utilize the available power, reduce soil compaction and for timeliness of tillage operations, now-a-days, front three point linkages are used in higher power tractors. So, front power take-off (PTO) also desirable with front three point linkages to operate passive implements like rotavator, mower etc. With this requirement in view, a program was developed in visual basic for design of a front PTO gear pair of higher hp tractors. In this design, the drive was taken from engine crankshaft running at rated speed of 2200 rpm and was reduced to 1000 rpm using helical gear pair. For validation of developed Visual Basic program, contact stresses and bending stresses analysis of gear and pinion was done in KISS software. The maximum bending root stress on pinion and gear was 330 N/mm^2 and 300 N/mm^2 against limiting stress value of 430 N/mm^2 where contact stresses are 1000 N/mm^2 against limiting value of 1500 N/mm^2 and therefore, designed gear pair was safe against bending and pitting. The output results of KISS software was used for validation of developed visual basic program. Developed program shows approximately the same results as in KISS software. Developed VB program is cheap, simple and can be used to design and for analysis of single gear pair.

Key words*: front power take-off, gear pair, bearings, contact stress, Visual Basic*

INTRODUCTION

The biggest challenge before the agricultural sector of India is to meet the growing demands of food for its increasing population from 1.22 billion in the year 2010 to 1.46 billion by the year 2030 [1]. This can be achieved by increasing cropping intensity and

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reducing turnaround time through increased mechanization. Tractors form an integral part of farm mechanization and have a crucial role to play in increasing agricultural productivity. As a result of different programmes implemented by the Government of India over the years, the total farm power availability is estimated to have been increased from 0.47 kW/ha in 1981-82 to 1.5 kW/ha in 2005-06 having a 46.75% contribution of tractor power [2]. India is the largest producer of tractors with 5.45 lakh of tractors in 2011 increased from 3.46 lakh in 2008 [3]. The tractor population of 41-60 hp segment has increased from 54,685 (22.8% of total number of tractors in a year) in 2000-2001 to 91,741 (31.5%) in 2005-2006. Usage of higher hp tractors (> 60 hp) has also increased from 265 tractors in 2000-2001 to 2068 in 2003-2004 [4]. To fully utilize the available power and for timeliness of tillage operations, now-a-days, front three point linkages are used in higher hp power tractors. Hence, there is need of front power take-off (PTO) drive with front three point linkages to operate passive implements like rotavator, mower etc. Therefore, a front PTO was designed which include the design of helical gear pair, selection of roller bearings and needle roller bearings, design of transmission shaft and gear as well as pinion shafts. PTO required the greatest amount of power among the major components during rotary tillage operations [3]. So, gear pair should design precisely to avoid breakage of PTO during operation.

So, by keeping the above facts in mind present study was undertaken with the objectives of designing the front PTO for higher power range tractors, analysis of bending and contact stresses of gear and pinion, development of a visual basic program and validate it with the results of KISS software.

MATERIAL AND METHODS

Front PTO design includes design of helical gear pair, selection of roller bearings, design of transmission shaft and gear as well as pinion shafts. To operate front PTO, the drive was taken from engine crankshaft in front side running at rated speed of 2200 rpm and was reduced to 1000 rpm using single gear pair. A visual basic program was developed in Visual Basic 6.0 for designing and selection of gears.

Design of gear pair: The gears are the central element in the transmission. Helical gear pair was designed as helical gears have more bending strength than spur gears. Front PTO output speed should be 1000 rpm at rated speed in anticlockwise direction as seen from front side [5]. Therefore, the gear reduction required will be the ratio of rated rpm to the 1000 rpm.

Geometry of gear: Geometry of any gear basically contains the module, pressure angle, face width, helix angle for helical gear, addendum, circular pitch diameter, root circle diameter, axial pitch, tooth height etc. Two standard pressure angles used in gear design are 14.5° and 20°.

Calculation of safety factor against bending: Failure by bending will occur when the tooth bending stress equals or exceed the yield strength [4]. The actual tooth root stress σ_F and the permissible (tooth root) bending stress σ_{FP} should be calculated separately for pinion and gear. Tooth root stress σ_F is the maximum tensile stress at the surface in the root, so it should be less than σ_{FP}. The ratio of σ_{FP} to σ_F is called safety factor against bending.

Calculation of safety factor against pitting: The calculation of surface durability is based on the contact stress, σ_H, at the pitch point or at the inner point of single pair tooth contact. The higher of the two values obtained is used to determine the load capacity. σ_H and the permissible contact stress, σ_{HP}, shall be calculated separately for wheel and pinion. σ_H shall be less than σ_{HP}. This comparison will be expressed in safety factors S_H which is the ratio of permissible contact stress, σ_{HP} and contact stress, σ_H. Fig. 1 shows the Pro-e model of designed front PTO drive.

Development of program in Visual Basic: The rapid progress in developing new software and programming languages always tend to facilitate the interaction between users and computers. As a result, many computer modeling and simulation programs have been developed. Recently, a computer program is developed in Visual Basic environment to determine the all types of machinery costs in hour and hectare basis [5]. Presently Visual Basic and Visual C++ are widely used to develop such software. In this paper, a program was developed for designing gear pair which gives the complete geometry of gear and pinion, analysis of the gear and pinion against the bending and contact stresses. This program was developed by using empirical equations of gear parameters given in ISO 6336-1, ISO 6336-2, ISO 6336-3, ISO 6336-5, ISO 6336-6 and written in Visual Basic programming language. Fig. 2 shows the flowchart used to design the program.

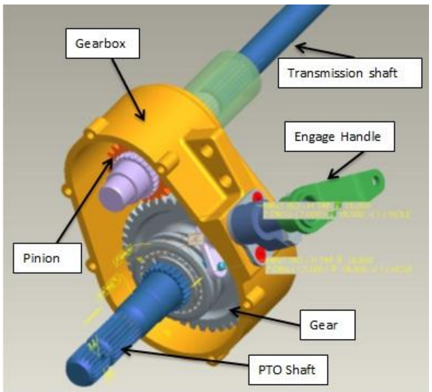


Figure 1. Pro-e model of front PTO

The program starts with an opening screen as shown in Fig. 3 which consist input parameters like power to be transmitted through gear pair, input and output speed, module, pressure angle and helix angle. Output parameters box consists center distance, number of computed teeth of gear and pinion. In choose teeth box, user should choose the integer number of teeth on the basis of computed number of teeth of gear and pinion. On the basis of chose teeth, VB program compute the center distance and computed gear ratio. User required to filled standard center distance while referring computed center distance.

Second screen as shown in Fig. 4 is opened which consists of gear and pinion geometry parameters like base circle diameter, pitch circle diameter, addendum, dedendum, base pitch etc. Third screen as shown in Fig. 5 used to select the material properties of gears. Using drop down menu, user can choose material type like normalized low carbon steel, through hardened wrought steel etc. Similarly, user can also choose reference profile of tooth for gear using drop down menu. Main output parameters on this screen will be nominal stress number, σ_{Flim} and allowable stress number σ_{Hlim} in N/mm^2. Fourth screen as shown in Fig. 6 consist of input box in which user have to fill required service life, application factor and life factor. On clicking on load cycle pushbutton, number of load cycles (in millions) will be calculated by VB program. The major output of this screen will be tooth root stress and tooth root safety factor against bending.

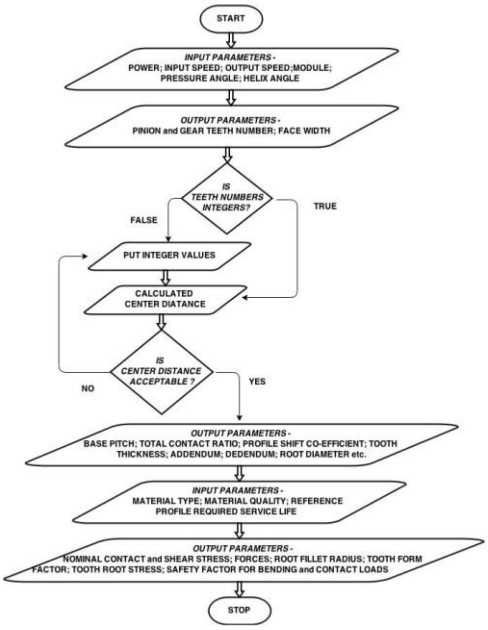


Figure 2. Flow chart for gear design in Visual Basic

Figure 3. Input screen of VB program

Figure 4. Geometry of gear and pinion in software

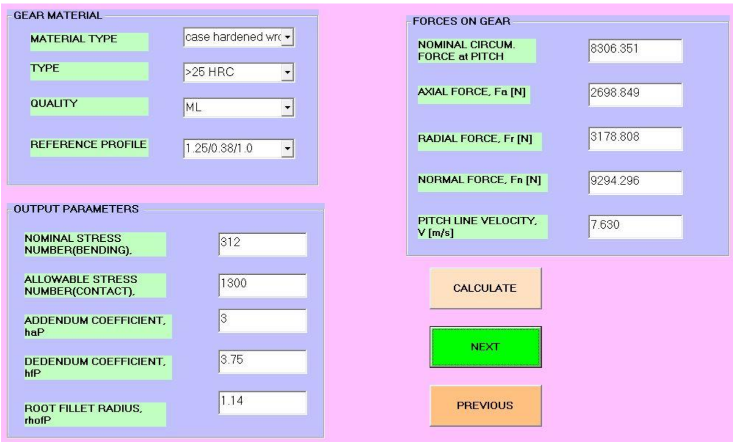


Figure 5. Mechanical properties and forces on gear pairs

Figure 6. Screen showing tooth root stress and other factors

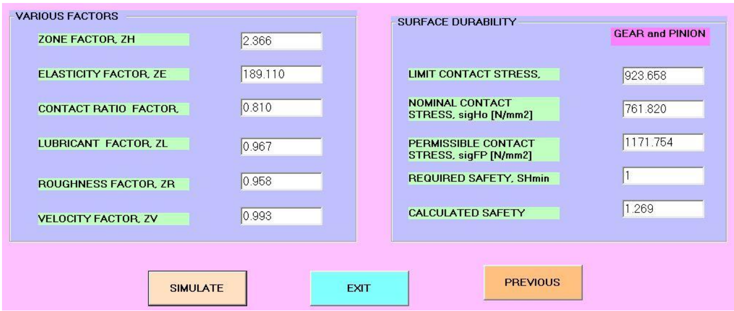


Figure 7. Screen showing contact stresses and other factors

Fifth screen as shown in Fig. 7 shows the nominal contact stress, σ_{Ho}, permissible contact stress, σ_{FP} and safety factor of gear and pinion against pitting. The minimum required safety factor against bending is 1.4 while for pitting it is 1. Value of safety factor against bending is more than safety factor for pitting because tooth breakage usually ends the service life of a transmission. Sometimes, the destruction of gear transmission can be a consequence of the breakage of one tooth. Therefore, the chosen value of safety factor against tooth breakage should be larger than the safety factor against pitting.

RESULTS AND DISCUSSION

Front PTO gear pair was designed and strength analysis was done in KISS software. Fig. 8 shows the tooth root stress on pinion which causes breakage of tooth and Fig. 9 shows the contact stress on gear which causes the pitting of tooth.

Fig. 10 and Fig. 11 show the tooth root stress on pinion and gear graphically. In this x-axis represent the middle axis of tooth and y-axis represent tooth root stress. As distance increased from middle of tooth to the side of tooth, tooth root stress increases. Initially, it increases slowly but as approaches to root of tooth it increases rapidly because at root thickness of tooth is less. For pinion, tooth root stress value reaches to 315 N/mm^2 while for gear its value is 300 N/mm^2. 3-D view of tooth root stress on pinion and gear are shown in Fig. 12 and Fig. 13. Root diameter is represented along x-axis, y-axis represent tooth root stress while z-axis represents width of tooth. In Fig. 12, up to root diameter 62 mm, there is no stress, so represented by blue color. Below 62 mm diameter, it starts increases and attains maximum value of 315 N/mm^2 at 59 mm diameter which is represented by red color. In Fig. 13, up to diameter 140 mm, there is no stress but at 138 mm diameter it reaches 300 N/mm^2. Fig. 14 shows the 3-D view of contact stresses on tooth of gear. At periphery of tooth, the contact stresses are nearly zero while at middle of face width contact stresses are higher which value is 1000 N/mm^2 and represented by red color. The results based on the graph and 3-D views presented in Figs. 11 - 15 are given in Table 1. The bending strength factor for gear is 1.955 and 1.876 for pinion. The contact stress factor for gear and pinion is 1.269. The minimum required value for bending strength factor and contact stress factor are 1.4 and 1 respectively. Hence, gear pair is safe against bending and pitting stresses.

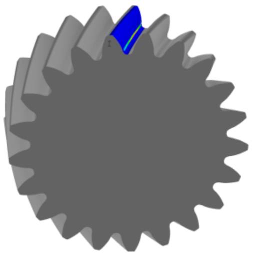


Figure 8. Tooth root stress on pinion

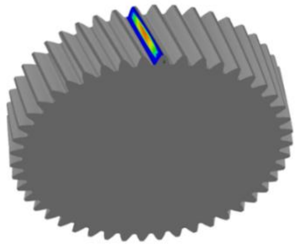


Figure 9. Contact stress on gear

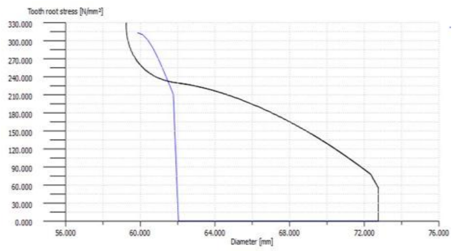


Figure 10. Graphical representation of tooth root stress on pinion

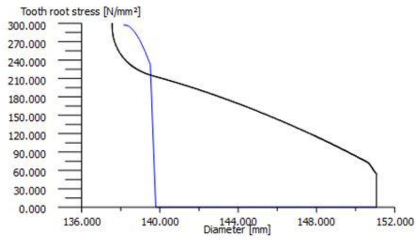


Figure 11. Graphical representation of tooth root stress on gear

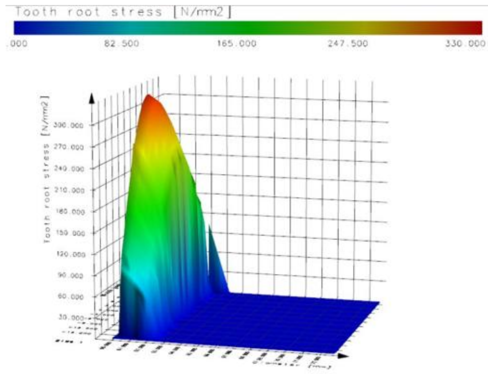
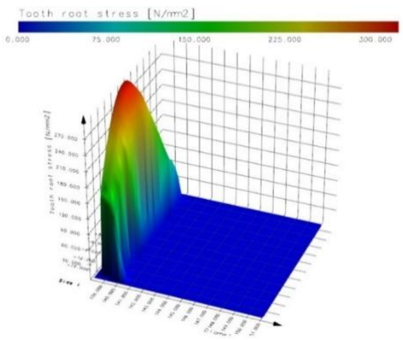


Figure 12. 3-D view of tooth root stress on pinion



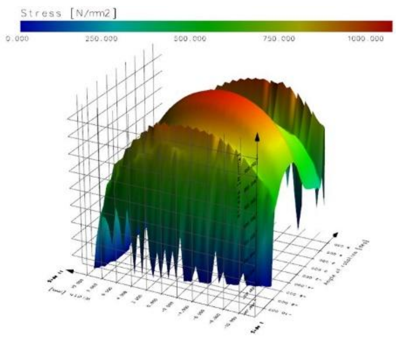


Figure 13. 3-D view of tooth root stress on gear *Figure 14. 3-D view of contact stress on gear*

Table1. Results of stress analysis of gear and pinion

Parameter	*Maximum stress (N/mm^2)*	*Permissible stress value (N/mm^2)*	*Bending strength factor*	*Contact stress factor*	*Remarks*
Root stress on pinion	*310.522*	*582.53*	*1.876*	*NA*	*Safe against bending*
Root stress on gear	*324.010*	*633.29*	*1.955*	*NA*	*Safe against bending*
Contact stress	*923.658*	*1171.75*	*NA*	*1.269*	*Safe against pitting*

Table 2. Validation of gear design Visual Basic program for front PTO drive

Parameters	*KISS Software values*		*VB program values*	
	Pinion	*Gear*	*Pinion*	*Gear*
Dedendum, mm	*3.45*	*3.71*	*3.481*	*3.688*
Addendum, mm	*3.3*	*3.041*	*3.269*	*3.062*
Root diameter, mm	*59.342*	*137.665*	*59.281*	*137.726*
Tip diameter, mm	*72.842*	*151.165*	*72.781*	*151.226*
Tooth thickness, mm	*4.92*	*4.81*	*4.908*	*4.758*
Working angle (°)	*20.07*	*20.19*	*19.048*	*19.868*
Bending lever arm, mm	*3.01*	*3.18*	*2.837*	*3.088*
Tooth root stress, $N\ mm^{-2}$	*324.43*	*332.19*	*310.522*	*324.010*
Pressure angle at pitch circle (°)	*20.942*		*20.942*	
Base helix angle (°)	*16.881*		*16.881*	
Center distance, mm	*105.672*		*105.672*	
Axial pitch, mm	*30.499*		*30.5*	
Overlap ratio	*0.971*		*0.984*	

Table 3 - Difference between VB program and KISS software values

Parameters	*Pinion*	*Gear*
Dedendum, mm	*-0.031*	*0.022*
Addendum, mm	*0.031*	*-0.021*
Root diameter, mm	*0.061*	*-0.061*
Tip diameter, mm	*0.061*	*-0.061*
Tooth thickness, mm	*0.012*	*0.052*
Working angle (°)	*1.652°*	*0.322°*
Bending lever arm, mm	*0.173*	*0.092*
Tooth root stress, $N\ mm^{-2}$	*13.908*	*8.18*

To validate the developed Visual Basic program, the output parameters of VB program were compared with the KISS software output values. As shown in Tab. 2 the parameters which computed using VB program and necessary for manufacturing a gear pair, are not much differ from the results obtained from KISS software. Some parameters like pressure angle at pitch circle, base helix angle, center distance etc. are perfectly same and in other parameters like dedendum, addendum, tooth root stress etc. an insignificant difference is present between VB program and KISS software results as shown in Tab. 3. There is a slightly difference because in designed VB program the values of trigonometric functions like tanβ etc. and other constant values like pi (π) are taken up to three decimal places only and output of one equation was the input of next equation, so the error was cumulative in nature.

CONCLUSIONS

A computer based user-friendly simulation program for designing gear pair for front PTO developed in Visual Basic software. This program was validated with output values of KISS software. The program has been found to give very close prediction of various gear parameters. Contact stress and bending stress analysis of designed gears was done and the designed pair was found safe against contact stresses and bending stresses. The calculated life of bearings using KISS software is found greater than the required service life. The advantage of developed VB program over KISS software is that KISS software is very costly and complicated one where developed VB program is cheap, simple and can be alternative of KISS software for designing and do analysis of single gear pair.

BIBLIOGRAPHY

[1] Anonymous. 2013. *Demographic overview-India, International database*, United States Census Bureau, http://www.census.gov. (Last accessed on August 17, 2016).

[2] Bector V., Singh S., Sharda A., Bansal A. 2008. Status and recent trends of tractor power in Indian agriculture, *Agri. Engg. Today* 32(1), 16-26.

[3] Lee, D., Kim, Y., Chung, S., Choi, C., Lee, K., Shin, B. 2015. Analysis of the PTO load of a 75 kW agricultural tractor during rotary tillage and baller operation in Korean upland fields. *Journal of Terramechanics*, 60, 75-83.

[4] Kamandar, M.R. 2011. The selection, designing and analysis of heavy duty tractor gearox, *Applied Mechanics and Materials*, 110, 2948-2957.

[5] Pranav, P.K., Phukan, Y., Saha, B. 2016. Computer program for cost estimation of agricultural machines. *Scientific Journal Agricultural Engineering*, XL(1), 1-10.

VISUAL BASIC PROGRAM ZA KONSTRUISANJE ZUPČASTOG PRENOSNIKA PREDNJEG PRIKLJUČNOG VRATILA KOD TRAKTORA VEĆE SNAGE

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Sažetak: Kod traktora veće snage se danas koriste prednji hidraulični uređaji radi potpunog iskorišćenja raspoložive snage, smanjenja sabijanja zemljišta i rokova za obradu zemljišta. Tako je i prednje priključno vratilo potrebno sa podiznim uređajem, za pogon gonjenih priključaka kao što su freza, kosačica i sl. Imajući u vidu ovaj zahtev, razvijen je program u visual basic za konstrukciju zupčastog prenosnika prednjeg priključnog vratila kod traktora veće snage. U ovoj konstrukciji korišćen je pogon sa kolenastog vratila motora brzine 2200 min^{-1}, redukovane na 1000 min^{-1} spiralnim zupčastim parom. Za ocenu razvijenog programa i analizu kontaktknog opterećenja i opterećenja na savijanje zupčanika i vratila korišćen je KISS softver. Maksimalno opterećenje na savijanje vratila i zupčanika bilo je 330 N/mm^2 i 300 N/mm^2 u odnosu na graničnu vrednost od 430 N/mm^2, a kontaktno opterećenje 1000 N/mm^2 u odnosu na hgraničnu vrednost od 1500 N/mm^2, tako da je konstruisani zupčasti par bio bezbedan pri udaru i savijanju. Izlazni rezultati programa KISS su korišćeni za ocenu razvijenog visual basic programa. Razvijeni program je pokazao približno iste rezultate kao i program KISS. Razvijeni program je je jevtin, jednostavan i može da se upotrebi pri konstruisanju i analizi pojedinačnih zupčastih parova.

Ključne reči: *prednje vratilo, zupčasti par, ležajevi, kontaktno opterećenje, Visual Basic*

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