Design Evaluation of an Animal Driven Prime Mover

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Abstract— ***In this paper, design evaluation of an Animal Driven Prime Mover (ADPM) which uses a speed enhancing gear-box is presented for its applications in chaff cutting, oil expelling, grinding wheat, etc. The gear-box was used earlier with the ADPM for water pumping. In a rural set-up, bullocks are available in plenty to be used as an ADPM for above tasks instead of using expensive and environment-polluting diesel generators to run those machines. The design of different components of the ADPM is presented here, along with the computer-aided-design (CAD) to animate the system for better visualization and verifying the designs.***

Keywords— *Animal Driven Prime Mover (ADPM); Computer aided design (CAD); Gear; Shaft and Bearing Analysis;*

I. INTRODUCTION

A recent study by the Ministry of Statistics and Programme Implementation (MOSPI), showed that even in present day India, there are still a few states in which the percentage of electrified villages is at a staggering low of 30% [1]. The problems related to electricity that the farmers have to deal with increasing wide gap between supply and demand is the reason for all the woes. Bullocks are mainly owned by marginal and small farmers who have small land holding. Using animals in the harvest operations reduces the drudgery and increases efficiency of these farmers. Animal power is a renewable source of energy and hence cheap for small farmers [1] compared to other forms of environment polluting energy sources, e.g., diesel, whose price is skyrocketing. Hence, in recent times, there has been growing interest in using bullocks for running several pre- and post-harvest operations in the villages. In order to assist such group of people, Rural Technology Action Group (RuTAG) at IIT Delhi took up a project in 2010 [2] to study the feasibility of the use of an animal driven gearbox to drive a water pump for irrigation purposes. The gear box was made by Panchal Pumps and Systems, Kanpur (UP) which was tested OK for such use. The mechanical design of the components, e.g., gears, shafts, etc., were checked from the fundamental principles of the mechanical engineering design. Based on the analysis certain changes were suggested to reduce the size and weight of the gear-box without sacrificing the performance. The modified gear-box was analyzed in computer-aided-design (CAD) environment using Autodesk Inventor software as well [3], before it was re-fabricated and tested satisfactorily in the field.

Based on the successful use of the gear-box in water pumping, it was felt that the same can be extended for the use in other applications as well, e.g., chaff cutting, paddy thrashing, oil expelling, grinding wheat, etc. For that several attachments need to be fixed with the gear-box, which were fabricated by same company. In this paper, those components are checked for their failures. A CAD model was also built, as shown in Fig. 1, to visualize the motions. The designs were mainly carried out from the theories of mechanical failures, as available in Mechanical Design text books, e.g., [4, 5]. The design exercise showed some failures of the components. So, they were modified for the motion and power transmission.

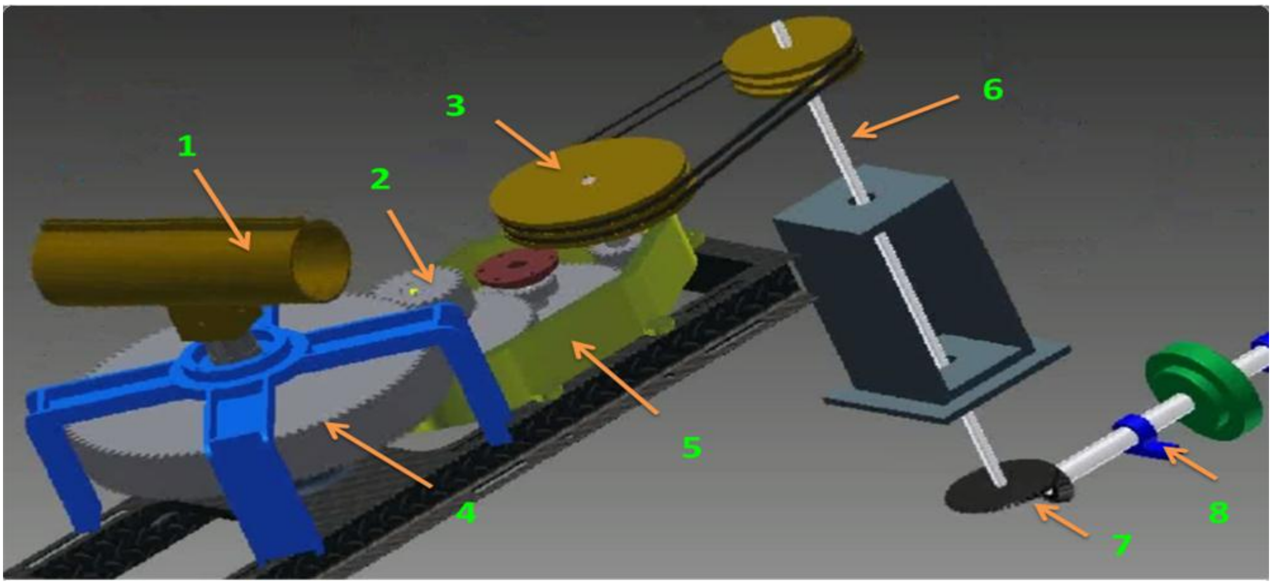

Fig. 1. CAD model for power transmission and different components

1. Lever Holder; 2. Cattle drive gear pinion; 3. V belt pulley; 4. Cattle drive gear; 5. Gearbox casing; 6. Vertical shaft; 7. Bevel gear arrangement; 8. Roller bearing support for the final output shaft

The paper comprises of the following sections: Section 2 provides the strength analysis of a spur gear based on AGMA (American Gear Manufacturers Association) standards [5], whereas Section 3 explains the selection of bearings [4]. Section 4 gives the design of shafts based on the strength [4], and finally, Section 5 concludes the paper.

II. DESIGN OF SPUR GEARS

Design of a spur gear train (velocity ratio=63), originally designed using Lewis' bending strength and contact strength, is done here using AGMA standards [5]. Under these standards, various factors such as manufacturing technique, usage conditions, geometry, life, reliability, stress concentration, etc., are to be considered, in which two popular criteria of gear design are as follows:

1) Bending strength criterion

2) Pitting resistance criterion

A. Input and Output Conditions

There are three gear pairs, as shown in Fig. 1, namely:

1) Cattle drive gear and Drive shaft pinion (Items 4 and 2 of Fig. 1)

2) Drive shaft gear and Idler shaft pinion (Not visible in Fig. 1, inside item 5.)

3) Idler shaft gear and Driven shaft pinion (Not visible in Fig. 1, inside item 5.)

Gear box was designed for transferring an input power of 650 W at input speed of 2 rpm at the cattle drive gear shaft. The output speed is 125 rpm at idler pinion shaft. The symbols for the input parameters are shown in Table I.

B. Input Torque

Input torque was provided at the cattle drive gear shaft and is given by

$$T_{in} = \frac{60P}{2\pi N} = 48.4 \text{ N-m} \tag{1}$$

C. Gear Tooth Forces

The tooth forces are useful for stress calculations and their different components are obtained below. Since the torque is transmitted by the tangential component (F_t), it is calculated as

$$F_t = \frac{2T}{D} \tag{2}$$

The radial (F_r) and total forces (F) are then given by

$$F_r = F_t \tan\alpha \tag{3}$$

$$F = F_t / \cos\alpha \tag{4}$$

D. AGMA Methodology

Two fundamental stress equations used in the AGMA standard are

1) Bending stress, and

2) Pitting resistance (contact stress).

This methodology takes into the consideration of issues like

1) Transmitted load magnitude; Overload

2) Dynamic augmentation of transmitted load

3) Size

4) Geometry: pitch and face width

TABLE I. LIST OF INPUT PARAMTERS

Sr.No.	Parameter	Symbol	Unit
1.	Power	P	W
2.	Angular speed of gear	N	rpm
3.	Number of teeth	Z	-
4.	Pressure angle	α	degree
5.	Pitch diameter	D	mm
6.	Face width	b	mm
7.	Poisson's ratio	υ	-
8.	Modulus of elasticity of gear and pinion	E_g and E_p	MPa
9.	Module	m	mm
10.	Brinell hardness number	H_S	BHN
11.	Ultimate tensile stress	σ_{ult}	MPa
12.	Yield tensile stress	σ_y	MPa
13.	Lewis form factor	Y	-

5) Distribution of load across the teeth
6) Rim support of the teeth
7) Lewis form factor and fillet resistance
8) Surface and operating conditions
9) Life and reliability
10) Fatigue and temperature conditions

The parameters used in AGMA formula are shown in Table II.

The allowable strength for bending and pitting are modified by taking into consideration factors like life, reliability and temperature conditions. The equation for the allowable bending stress in SI units is as follows:

$$\sigma_{all} = \frac{S_t}{S_F}\frac{Y_N}{Y_\theta Y_Z} \tag{5}$$

The equation for the allowable contact stress is:

$$\sigma_{C,all} = \frac{S_C}{S_H}\frac{Z_N Z_W}{Y_\theta Y_Z} \tag{6}$$

AGMA allowable stress numbers for bending and pitting are taken for unidirectional loading at 10^7 stress cycles with 99% reliability, whereas safety factors (S_F and S_H) are the ANSI/AGMA standards 2001-D04 and 2101-D04 which contain a safety factor S_F guarding

TABLE II. LIST OF PARAMETERS IN AGMA FORMULAS

Sr. No.	Parameter	Symbol	Unit
1.	Tangential force	F_t	N
2.	Overload factor	K_o	-
3.	Dynamic factor	K_v	-
4.	Size factor	K_s	-
5.	Transverse module	m	mm
6.	Face width	b	mm
7.	Load-distribution factor	K_H	-
8.	Rim-thickness factor	K_B	-
9.	Geometry factor for bending strength	Y_J	-
10.	Elastic coefficient	Z_E	$\sqrt{\text{N/mm}}$
11.	Surface condition factor	Z_R	-
12.	Pitch diameter of pinion	D_p	mm
13.	Geometry factor for pitting resistance	Z_I	-
14.	Allowable bending stress number	S_t	MPa
15.	Stress cycle factor for bending stress	Y_N	-
16.	Temperature factor	Y_θ	-
17.	Reliability factor	Y_Z	-
18.	AGMA factor of safety for bending	S_F	-
19.	Allowable contact stress number	S_C	MPa
20.	Stress cycle life factor	Z_N	-
21.	Hardness ratio factor for pitting resistance	Z_W	-
22.	Stress cycle factor for contact stress	S_H	-

against bending fatigue failure and safety factor S_H guarding against pitting failure. Note here that S_F denotes strength-over-stress where the stress is linear with the transmitted load. It is given by,

$$S_F = \frac{(S_t Y_N)/(Y_\theta Y_Z)}{\sigma} = \frac{\text{fully corrected bending strength}}{\text{bending stress}} \quad (7)$$

Similarly, S_H too is strength-over-stress where the stress is *not* linear with the transmitted load F_t. It is given by,

$$S_H = \frac{(S_C Z_N Z_W)/(Y_\theta Y_Z)}{\sigma_C} = \frac{\text{fully corrected contact strength}}{\text{contact stress}} \quad (8)$$

While the definition of S_H does not interfere with its intended function, a caution is required when comparing S_F with S_H in an analysis in order to ascertain the nature and severity of the threat to loss of function. It is suggested to compare S_F with S_H^2 when trying to identify the threat to loss of function with confidence.

E. Results and Remedies from the AGMA Methodology

From the above analyses, the results are summarized in Table III. The following are the suggested remedies for improving safety factor:

1) One of the remedies could be to increase strength and hardness of the gear materials. In short improving material property.

2) Increase the face width and pitch diameter.

3) It is found that former pairs are under-designed and latter pairs are overdesigned. So, the gear ratio of latter pairs can be increased and accordingly decreased for former pair to maintain final gear ratio.

III. SELECTION OF BEARINGS

Bearing selection for two shafts, namely, the output shaft (driven shaft) on which the driven belt pulley is mounted and the vertical shaft (Idler shaft indicated with item 6 in Fig. 1) which is smaller in length as compared to the output shaft was carried out. The two shafts are mounted with bevel gears (item 7 of Fig. 1). The ball bearings were selected for reduction in friction between the bearing parts and the shafts. The vertical shaft is also mounted with V-pulley requiring two bearings, whereas the output shaft shown in Fig. 1 is provided with flat belt pulley at the other end (not shown) requiring four bearings. The equivalent load on the bearings can be calculated as

$$P = XF_r + YF_a \quad (9)$$

where,

P : Equivalent dynamic radial load,

F_r : Constant radial load,

F_a : Constant thrust load,

Y : Thrust load factor and

X : Radial load factor

The life of the bearings can be given by [4,5]

$$L = \left(\frac{C}{P}\right)^3 \quad (10)$$

TABLE III. SAFETY FACTORS OF THE GEARS

Gear or pinion	**Present factor of safety**		**Safety factor S_F for bending fatigue failure**	**Safety factor S_H for pitting failure**	**S^2_H**	**Remarks**
	Bending	Surface fatigue				
Cattle Drive Gear	1.25	1.46	2.86	0.71	0.51	Gear is under-designed for pitting resistance
Cattle Drive Pinion	1.55	1.61	2.77	0.79	0.63	Pinion is under-designed for pitting resistance
Drive Gear	1.12	1.11	2.37	1.34	1.79	Gear is less safer for pitting resistance and over-designed
Drive Pinion	1.13	1.23	1.89	1.21	1.46	Pinion is less safer for pitting resistance and over-designed
Driven Gear	2.02	1.73	4.11	1.89	3.59	Gear is less safer for pitting resistance and over-designed
Driven Pinion	2.44	1.91	4.03	1.82	3.31	Pinion is less safer for pitting resistance and over-designed

where,

L : Fatigue life expressed in millions of revolutions

P : Equivalent radial dynamic load

C : Dynamic load Capacity

A. Calculations for Vertical Shaft

The symbols used for the input parameters for the vertical shaft are shown in Table IV.

TABLE IV. INPUT PARAMETERS FOR VERTICAL SHAFT

Sr. No.	Parameter	Symbol	Value
1.	Power	P	650 W
2.	Mass of the pulley	M	5 kg
3.	Axial force	F_a	2649 N
4.	Tangential force	F_t	1078 N
5.	Radial force	F_r	1325 N
6.	Rpm	N	64
7.	Diameter of shaft	D_1	20 mm
8.	Pitch diameter of gear	D	180 mm
9.	Semi groove angle	2β	34°
10.	Angle of contact	θ	2.72 rad
11.	Coefficient of friction	μ	0.2
12.	Center distance between pulleys	x	474.8 mm
13.	Mean diameter of smaller pulley	d_1	200 mm
14.	Mean diameter of bigger pulley	d_2	400 mm
15.	Torque on tight side	T_1	-
16.	Torque on slack side	T_2	-

Since the center distance between the two pulleys shown in Table IV, the following calculation can be carried out:

$$\sin\alpha = \frac{d_2 - d_1}{2x} \tag{11}$$

From (11), we get $\alpha = 12.16°$ (Angle between the line joining the centers of both the pulleys and to the line which is parallel to the tangent to both pulleys and passing through center of smaller pulley). Angle of contact for smaller pulley is then calculated as

$$\theta = 180° - 2\alpha = 155.68° = 2.72 \text{ rad}$$

Other expressions required to obtain the tensions are as follows:

$$\frac{T_1}{T_2} = e^{\frac{\mu\theta}{\sin\beta}} \tag{12}$$

$$\text{Power transmitted by belt } = (T_1 - T_2)v \tag{13}$$

Where, v is the speed of belt given by

$$v = \frac{\pi d_1 N}{60} \tag{14}$$

Using (12-14), we obtain,
$T_1 = 1160$ N and $T_2 = 191$ N

Reactions in vertical and horizontal planes were then obtained by applying equilibrium conditions in both the vertical and horizontal planes, thus reactions at two bearings were found as

$$R_1 = F_{r1} = 4050 \text{ N}, \ R_2 = F_{r2} = 4154 \text{ N and } F_a = 2649 \text{ N}$$

Dynamic load capacities for the ball bearings were then calculated by taking the values of $X = 0.56$ and $Y = 1.5$, because ratios of $(F_a / F_r)_1 = 0.63$ and $(F_a / F_r)_2 = 0.63$ were evaluated. Thus, in both the cases, $(F_a / F_r) \geq 0.63$. This is greater than the ratio of (F_a / C_0) because from the SKF Catalogue the value of C_0 (static load capacity) for 20 mm and 30 mm diameters for maximum capacity is in the range of 6200-16600 N and 6800-24000 N, respectively. Thus the ratio of (F_a / C_0) varies in between 0.130-0.427. Thus, taking the values of X and Y factors from the SKF catalogue to the corresponding ratios of ball bearings (single row deep groove), we get the forces as:

For 1st Bearing: $P_1 = 6241$ N;

For 2nd Bearing: $P_2 = 6300$ N;

Taking, $Loadfactor = 1.2$, the expression for the life L is as follows:

$$L = \frac{60 N L_{10h}}{10^6} \tag{15}$$

It is expected that the device will run for 5 hours every day for 250 days in a year for 10 years. Hence,

L_{10h} = 12500 hrs; and L = 48 million revolution; Note that the dynamic load capacity is related as

$$C = PL^{\left(\frac{1}{3}\right)} Loadfactor \tag{16}$$

TABLE V. INPUT PARAMETERS FOR OUTPUT SHAFT

Sr. No.	Parameter	Symbol	Value
1.	Power	P	650 W
2.	Axial force	P_a	1325 N
3.	Tangential force	P_t	1078 N
4.	Radial force	P_r	2649 N
5.	Rpm	N	128
6.	Diameter of shaft	D_1	40 mm
7.	Pitch diameter of gear	D	90 mm
8.	Angle of contact	θ	4.36 radian
9.	Coefficient of friction	μ	0.2
10.	Center distance between pulleys	x	-
11.	Mean diameter of smaller pulley	d_1	-
12.	Mean diameter of bigger pulley	d_2	500 mm
13.	Tension on tight side	T_1	-
14.	Tension on slack side	T_2	-

The factors for the two bearings are

For 1st Bearing: $C_1 = 27218$ N;

For 2nd Bearing: $C_2 = 27475$ N;

Using SKF catalogue for the above dynamic load capacities, the bearing for 20 mm diameter shaft will be

SKF6404, and for 30 mm diameter shaft it will be SKF6306.

B. Calculations for Output Shaft [Driven]

Table V shows the symbols used for the calculations of output shaft, on which bearing (item 8 of Fig. 1) is placed. Here, a total of four bearings are used for support of the output shaft, on which a wheel is mounted at other end (not visible). A flat belt on the wheel is mounted to transmit power to the appliances. For that, a set of calculations similar to (12-14) were performed to obtain $T_2 = P_2 = 25$ N and $T_1 = P_1 = 219$ N.

Now, the reactions were calculated on the vertical and horizontal planes. Since there are four bearings there will be eight unknown reactions. So, one cannot calculate them. However, whatever may be their directions their net effect will be corresponding to the applied load on a single plane. Hence, denoting the net reactions as Rh_{nett} and Rv_{nett} in horizontal and vertical directions which are the combined effect of all the reactions in horizontal as well as in vertical plane, irrespective of their actual directions, they are found as $Rh_{nett} = 835$ N and $Rv_{nett} = 2649$ N, where $F_r = 2778$ N. The dynamic load capacities for the ball bearings are then obtain again for the values of $X = 0.56$ and $Y = 1.8$ and $F_a = 1325$ N because the ratios of $\left(\frac{F_a}{F_r}\right) = 0.476$, which is greater than the ratio of $\left(\frac{F_a}{C_0}\right)$=0.0137-0.142, because from the SKF Catalogue the value of C_0 (static load capacity) for 40 mm and 50mm diameters for maximum capacity is in the range of [9300-36500]N and [10000-52000]N, respectively.

Using calculations similar to (15), one gets, $C = 21655$ N. So for the 40 mm shaft and 45 mm shaft, SKF 6308 and SKF 6310 bearings, respectively, are suitable.

IV. DESIGN EVALUATION OF THE SHAFTS

Strength analysis of the shafts was done based on the loading diagram shown in Fig. 2. The beams have been assumed as overhung for simplification of strength analysis. The shafts present in the ADPM are cattle drive gear shaft, drive gear shaft, idler gear shaft, and driven gear shafts. The strength of these four shafts needs to be checked.

A. Cattle Drive Gear Shaft (EN8)

The cattle drive gear shaft was made of EN8 which is a ductile material. So, *maximum shear stress theory* should be used to check the strength of the shaft. For that, torque transmitted to the shaft is given by

$$\text{Power, } P = T\omega \tag{17}$$

where P =650 W, N = 2 rpm which gives, torque (T) = 3104 N-m. Hence, the tangential force, F_t, is expressed as

$$F_t = \frac{2T}{D} \tag{18}$$

where the diameter of gear is 474 mm. From (18), we get F_t = 13096 N. Using (19),

$$F_c = F_t / \cos\Phi \tag{19}$$

Using (19), normal force F_c can be obtained for this

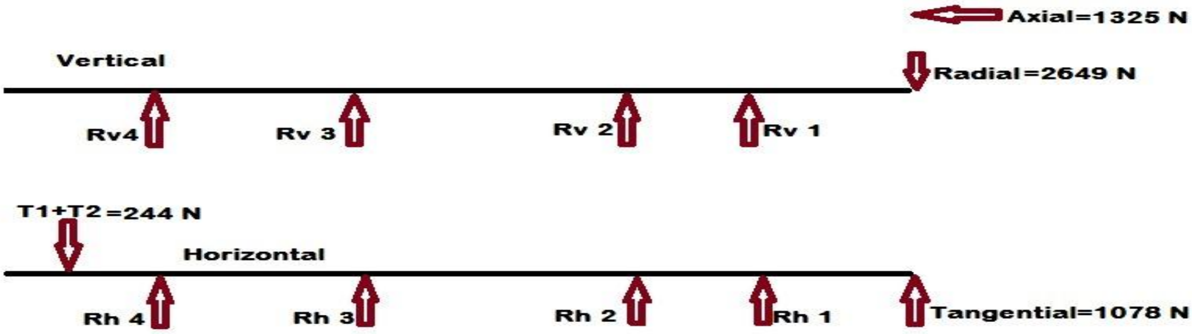

Fig. 2 Reaction in different planes for Driven output shaft

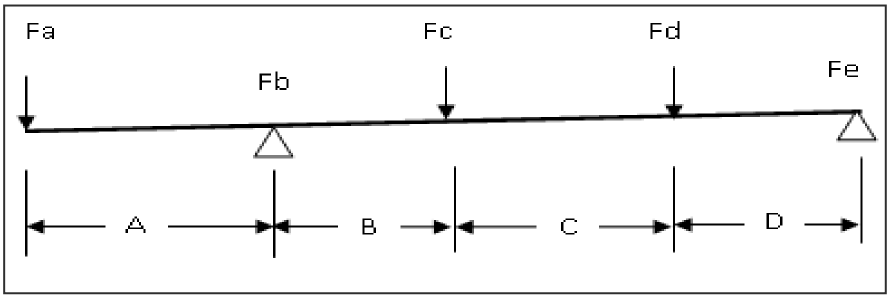

Fig. 3 Load diagram on shafts

shaft shown in Fig. 3, where A=157 mm, B=58 mm, C=0, D=112 mm. It is F_c = 13936 N. Other forces are F_a = -776 N and F_d = 0. From the above forces, maximum bending moment was calculated as M = 453 N-M. Using the formula given below, equivalent torque can be found using T = 3104 N-m, and M = 453 N-m in (20).

$$T_{eq} = 0.5\left(\sqrt{M^2 + T^2}\right) \tag{20}$$

Then stress is found using (21) as

$$\tau = \frac{16T_{eq}}{\pi d^3} = 181 \text{ MPa} \tag{21}$$

The shear strength of the shaft material is 495 MPa. Thus, the factor of safety was calculated as 2.73, which is the ratio between the ultimate strength and the stress subjected.

B. Drive Gear Shaft (EN8)

Similar calculations were done for the other three shafts. For this shaft maximum shear stress theory (MSST) was used. Note from Fig. 3, A=87.5 mm, B=33.5 mm, C=0, D=89.3 mm. Hence, F_a = 13936 N, F_c = -13003 N, F_d =0. The factor of safety was calculated as 2.6.

C. Idler Gear Shaft (EN8)

For this shaft, A=0, B=38.5 mm, C=33 mm, D=59 mm, F_a =0 , F_c = -13003 N, F_d =5433, using MSST, the factor of safety is calculated as 3.2.

D. Driven Gear Shaft (EN8)

For this shaft, A=97.5 mm, B=61 mm, C=0, D=62.5 mm, F_a =1328 N, F_c = 1858 N, F_d =0, again the MSST, the factor of safety is calculated as 1.8.

From strength calculation of shaft, it can be seen that the factors of safety are in acceptable range and further changes in dimensions of shafts are not required.

V. CONCLUSIONS

Several components of the attachment to be used for the transmission of motion and power from the ADPM geared to run several appliances like chaff cutting, paddy thresher, oil expelling, etc, were checked for their safety. The results for the spur gears are summarized in Table III. Necessary steps need to be taken for the under-designed and over-designed gears are also discussed. One can change the material or the face-width of the under-designed gears. Alternatively, one can alter the gear ratios to take care of the under- and over-designed gear, as suggested in Section II. The design of other components like bearings and shafts are found satisfactory. In addition, a CAD model in Autodesk Inventor was developed for the motion simulation, which helps one to visualize the movements of the shaft and gear components. In future the CAD model will be used for more sophisticated analysis, e.g., Finite Element Analysis (FEA) of the components.

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