1. Introduction (need, background and current status of the Animal Driven Prime Mover)

Animal have been, from times immemorial very helpful resources of nature for sustinance of human beings and has been harnessed for various animal products as well as renuable energy in the form of draught power, load carrying etc. The use of animals has been very ancient, however, the technologies for harnessing of animal energy has remained at a primitive level for a very long time. As a consequence, with the adventure of fossil fuels, the use of animals has greatly dwindled. Nevertheless, from all considerations, animal have proved to be a renewable, sustanable and a historic source of energy and with increasing thrust on renewable energy at a present times, it is imperitive to retask at harnessing animal power a fresh and develop static of art technologies to efficiently harness animal power particularly for all the rural agri-industrial and transport activites.

Draught animals powered Indian agriculture to a greater extent. At present they provide energy to Indian agriculture equivalent to 27 million mega watts of electrical energy. The present total value of draught animals is Rs.408 billions. Therefore, it is necessary to promote the continued use of draught animals in agriculture which is a time tested renewable energy source for sustained agriculture in the face of dwindling reserves of the non renewable sources of energy. Use of animate sources of energy (human and draught animal power) is very predominant in the country and will continue to be so for many years due to the prevailing small sized land holdings and socio-economic conditions. The annual utilization of draught animals in different regions varies from about 300 hrs to 1500 hrs against the ideal utilization period of about 2400 hrs [1]. Note that the use of rotary motion operate agro-processing machines can increase the present utilization of the animal power. In that direction, an Animal Driven Prime Mover (ADPM) consisting of a step-up (typically 60 to 500 times) gearbox connected to a level driven by a pair of bullocks or horses is a multi-purpose for various mechanical power applications and the agro-processing and rural industrilazation applications like water pumping, chaff cutting, castor decorticator, oil expelling, winnowing, etc. An application in water pumping using an Animal Driven Prime Mover (ADPM) shown in figs. 1 and 2. The draught power in rotary mode which is usually available at 2-3 rpm can be efficiently connected to deserved highlits spreeds by design of a suitable gearbox and this naturally becomes a verable ADPM for various applications. In order to encourage the useage of ADPM, several individuals as well as organizations have been to develop such devices during last 2-3 decades, amongst them, Central Institute of Agricultural Engineering (CIAE) IN Bhopal, College of Agricultural Engineering (CAE) in Raichur, Kanpur Goshala Society (KGS), Mr. R.S. Singh in Varanasi, M/s. Panchal Pumps and Systems in Kanpur, and several others may be cited. Each of them has targeted specific application based on their local needs. For example, KGS developed ADPM primarily for the electricity generation, where as the ADPM by M/s. Panchal Pumps and Systems has been mainly used for water pumping CIAE, Bhopal has been working on ADPM for multiple applications for a long time. A combative study of salient ADPMs was reported in [2].

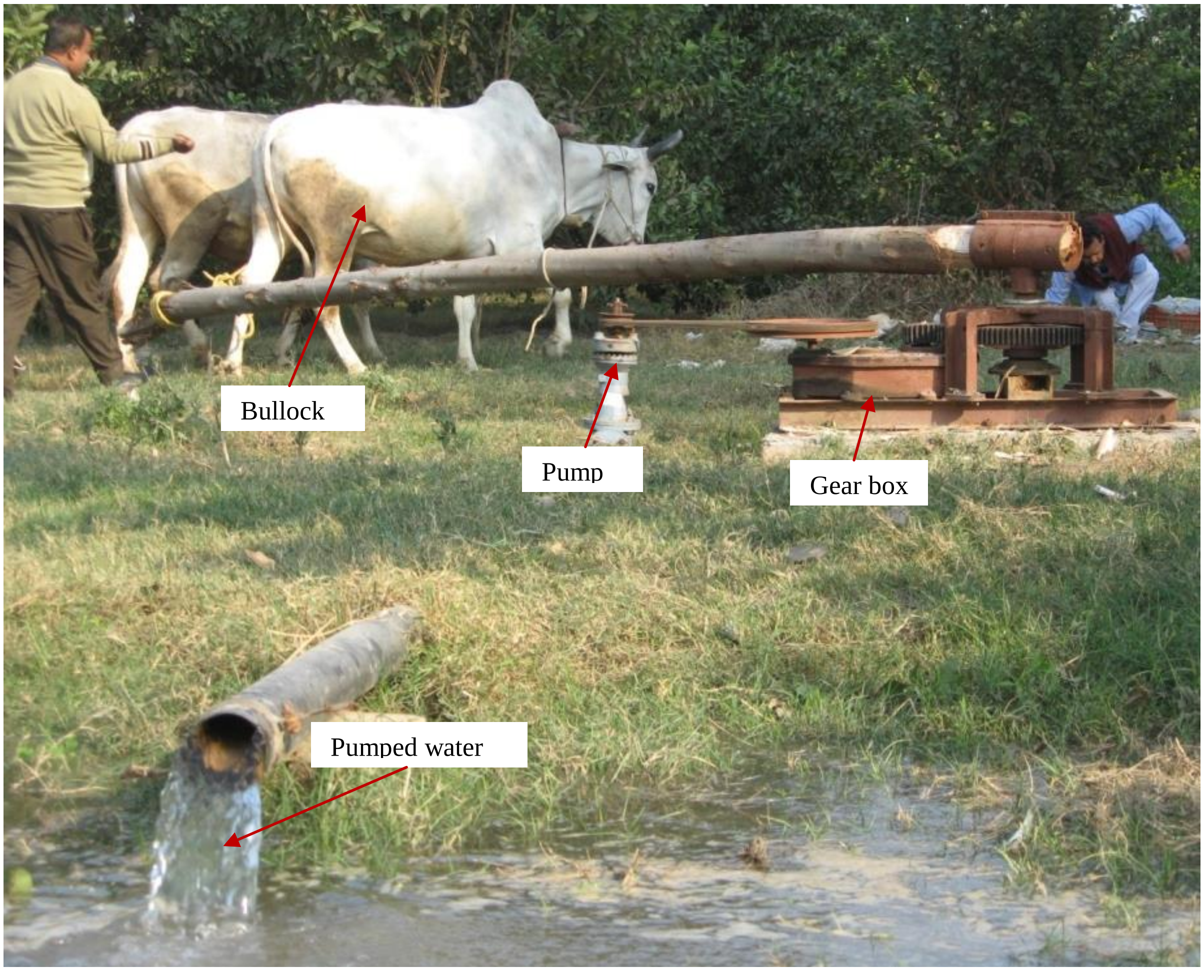


Fig. 1 Animal driven water pump with gear box and bullocks

2. Motivation and Objectives

As mentioned above, various efforts have been made to develop different prototypes of the ADPMs, but all of those remained in the pilot level. They have not been regularly evaluated, field used or

3. Work Plan and Gantt chart

The project officially begun on April 01, 2010, i.e., the day when IIT Delhi received the first installment of the payment. Immediately, actions were taken to hire the sanctioned staffs,

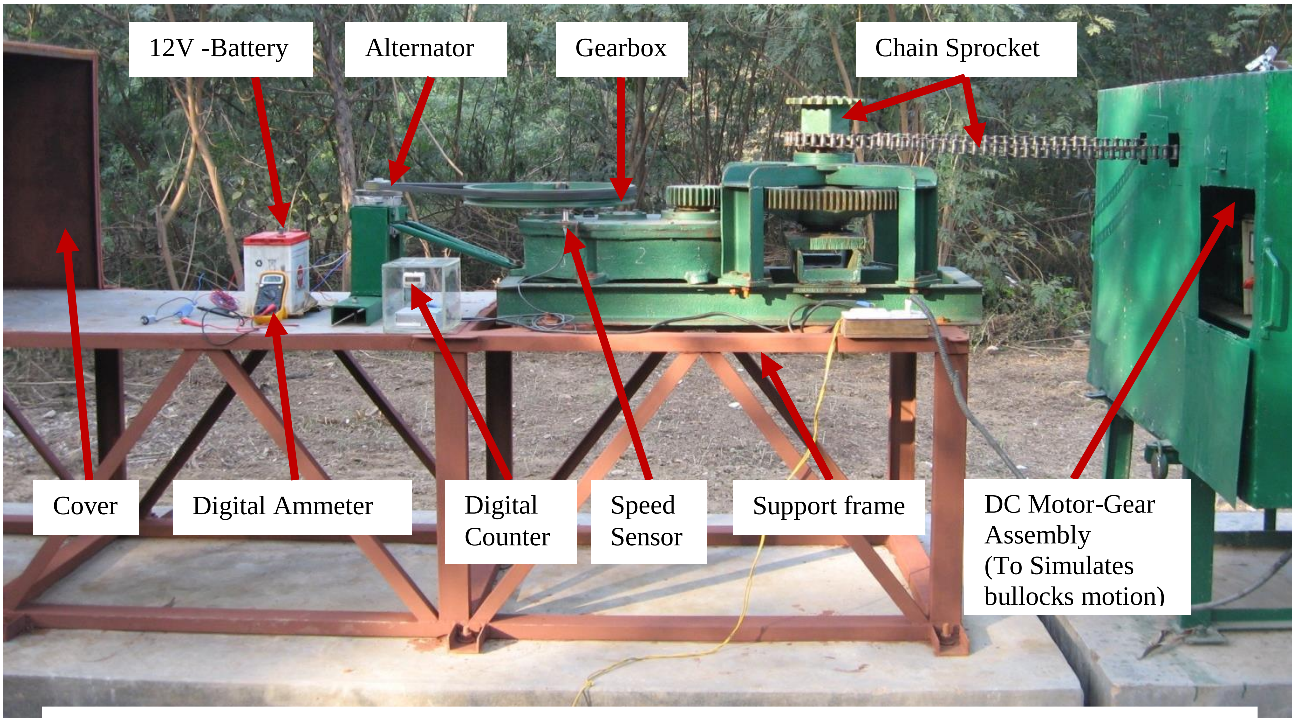


Fig. 2 Gear box by M/s. Panchal Pumps and Systems, Kanpur for testing at IIT Delhi

namely, one Sr. Project Assistant and one Project Attendant. Following the IIT Delhi interview procedures, the following people were recruited:

1. Mr. Raj Kumar Gupta, Sr. Project Assistant (Joined on 6th May, 2010)
2. Mr. Mangal Sharma, Project Attendant (Joined on 6th May, 2010)

Two M. Tech. students were also contacted during May - June, 2010 to help in the design and analysis of the gear box and obtaining the necessary data related to the project. In order to fulfill the objectives of the project, weekly meetings were held to discuss the progress of the work, and set the targets for the next week. During the weekly meetings, technical aspects, fundamental studies and literature search on the gear box, as shown in Fig. 1, were discussed. A sample page from the project discussion diary is attached in Appendix A, whereas the original work plan of the project, along with the actual one, are given in Table 1.

3. Work elements or Steps

- Field Visit and problem identification
- Detail Study and Design according to Field Visits Feedback
- Prototype Development
- Field Testing
- Re-design according to Feedback and Field Testing
- Final Prototype

- Report Writing

4. Field Visit (December 23-24, 2009)

After the project was submitted, in 2009 a field visit to Kanpur was made before the project actually started. An animal driven water pump with gear box has been installed at the agriculture farm of Mr. Pradeep Mishra, Periyar, Unnao district of UP, as shown in Fig. 1. The water pump runs well. There was a good discharge of water. The pump was running for a short duration of 10 – 15 minutes. The animal driven water pump with gear box has been used regularly for two hours in the morning and two hours in the evening, as needed by a farmer. The owner explained that one of the bullocks was not keeping well. In fact, it appeared that the pump was installed more as a demonstration to promote the technology rather than to use regularly for irrigation of the fields.

Table 1 Work plan of the project

Serial No.	Activities	2010 - 2011															
		April 2010	May	June	July	Aug.	Sept.	Oct.	Nov.	Dec.	Jan. 2011	Feb.	Mar.	April	May	June	July
1.	Staff Recruitment																
2.	Field Visits and Problem Identification																
3.	Detail Study and Design according to Field Visits and Feedback																
4.	Prototype Development																
5.	Field Testing																
6.	Re-design according to Feedback and Field Testing																
7.	Final Prototype																
8.	Report Writing																

Original: **Actual:**

The people from IIT Delhi and the local manufacturer of animal driven water pump and gear box in Kanpur visited Periyar regarding demonstrated the working of Animal Driven Water Pump with Gear box. The feedbacks based on the field visits are as follows:

1. The overall performance of the animal driven water pump and Gear box is good.
2. The gear box and the pump are shown in Figs. 3 and 4, respectively.
3. The gear box should be checked for all its components from mechanical engineering design point of view. If possible, weight of the gear box and its cost should be reduced.
4. The cost of pump (screw pump) was Rs. 28,500/-, and that of the gear box was Rs. 32,500/-.
5. An evaluation procedure for testing a given gear box should be devised.
6. A standard procedure for the gear components should also be proposed.
7. Manufacturers and users want IIT Delhi to evolve a process to judge the standard of their gear box for the use as animal driven water pump.

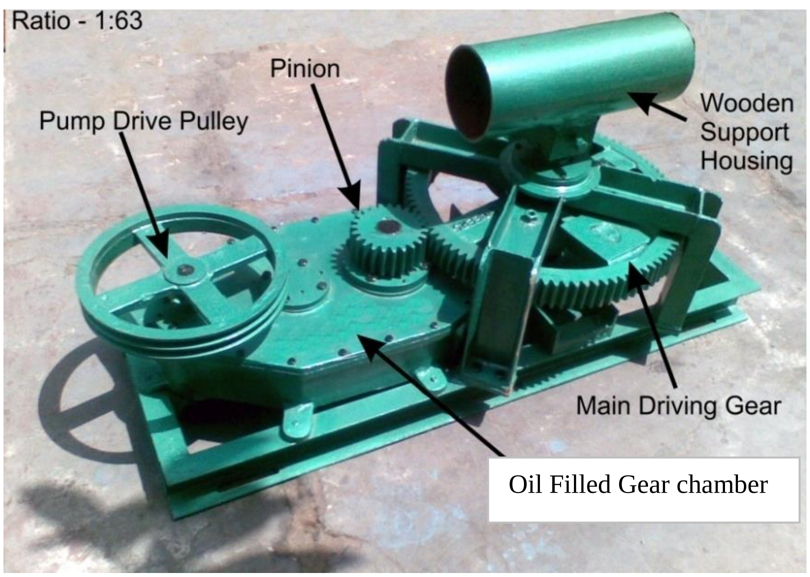


Fig. 3 Gear box

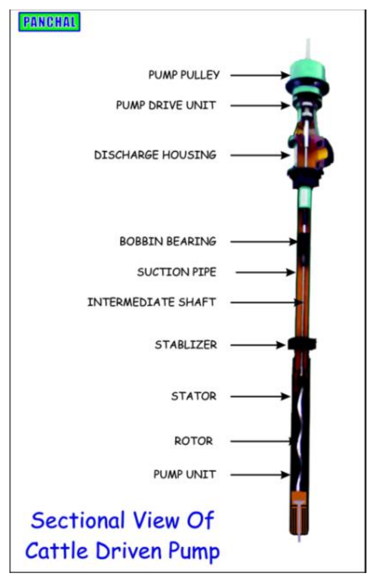


Fig. 4 Water pump

5. Evaluation of Existing Gear Box

Evaluation of the gearbox fabricated by M/s. Panchal Pumps and Systems, Kanpur is done in the following subsections.

5.1 Specifications and CAD Assembly

Specification of the gear box is given in Table 2, whereas its CAD assembly drawn in Autodesk Inventor Software is shown in Fig. 5.

Table 2 Specification of the gearbox

Input power	**2 hp (~1.5 kw)**
Overall gear ratio	1:63
Input velocity	2 rpm
Module of gear pair 1 (spur gear)	6 mm
Face width of gear pair 1	35 mm
Module of gear pair 2 (spur gear)	7 mm
Face width of gear pair 2	70 mm
Module of gear pair 3 (helical gear)	2.5 mm
Face width of gear pair 3	35 mm

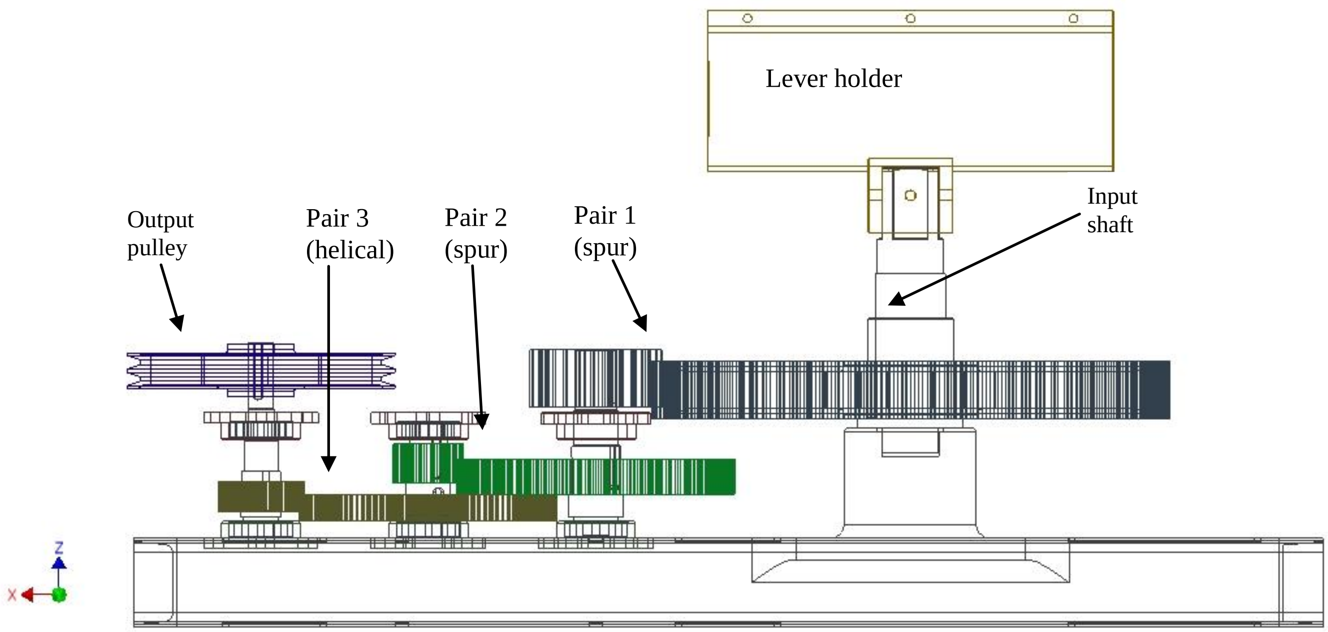


Fig. 5 CAD Assembly of the gear box

5.2 Design Checks

Gear and shafts designs and the selection of bearings are explained next.

5.2.1 Gear Design

Based on eqns. (B.1) and (B.2) given in Appendix B, the factor of safety (FOS) was calculated for the gear pairs of existing gear box. The results are shown in Table 3.

Since the FOS for the helical gear pair 3 was coming high in both bending and surface fatigue, it was felt that its value can be reduced to lower the weight and cost of this pair. Hence, effects of material and geometry parameters were carried out. They are shown in Figs. 6 to 8. Based on Figs. 6 to 8, the proposed specification for gear pair 3 is suggested as follows: Material: EN 24; Face width: 15mm (reduced from existing 35mm); Module: 2.5.

This reduces FOS for bending and surface fatigue of gear 3 to 6.87 and 5.16, respectively.

Table 3 FOS of the gears

Pair and type	Gear or pinion	Material	FOS in bending	FOS in surface fatigue
Pair 1: Spur	Gear	CI FG-210-Gr	1.29	1.22
	Pinion	EN-8= AISI 1040	1.25	1.36
Pair 2: Spur	Gear	EN-24= AISI 4340	1.99	1.87
	Pinion	EN-24= AISI 4340	1.5	1.87
Pair 3: Helical	Gear	EN-24= AISI 4340	6.87	5.16
	Pinion	EN-24= AISI 4340	5.37	5.16

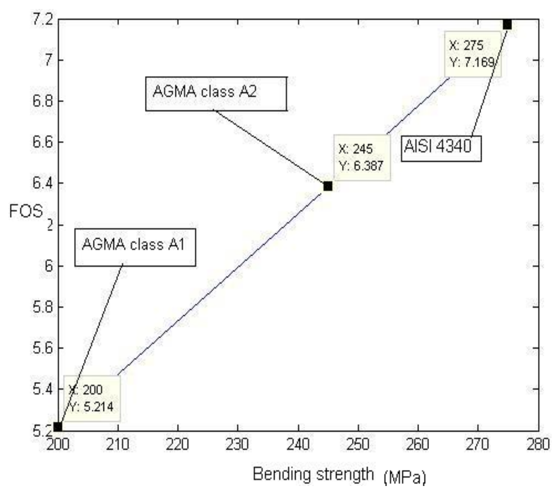


Fig. 6 FOS for different materials

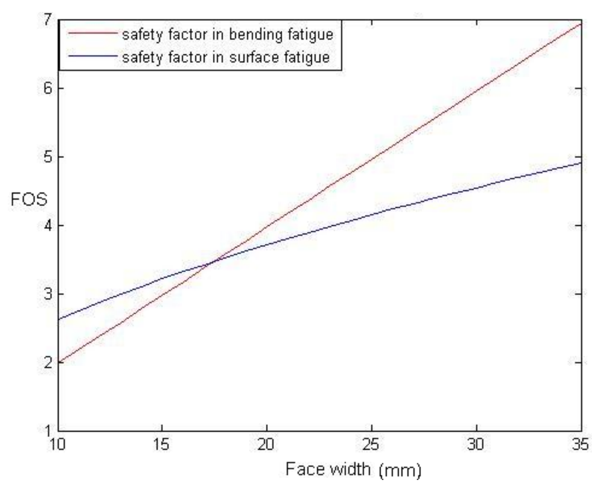


Fig. 7 FOS for different face widths

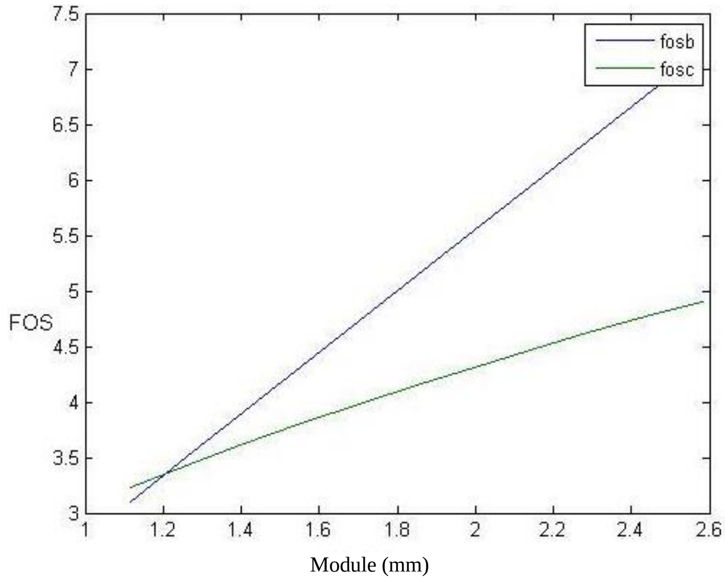


Fig. 8 FOS for different modules

5.2.2 Shaft Design

Using equations given in Appendix B, i.e., eqns. (B.6) – (B.9), the FOS for the shafts are given Table 4.

Table 4 FOS of shafts

Shaft	**Material**	**FOS**
Cattle drive gear shaft	AISI 4340	1.28
Drive gear shaft	AISI 4340	1.08
Idler gear shaft	AISI 4340	1.52
Driven gear shaft	AISI 4340	1.23

Since the values of FOS are reasonable for all shafts, no change is suggested.

5.2.3 Bearing Selection

Using eqns. (B.10) and (B.11), the estimated life is calculated, as shown in Table 5, which is compared with the desired life.

Table 5 Desired and estimated life of bearing

Bearing type	Desired life ($\times 10^6$ revolutions)	Estimated ($\times 10^6$ revolutions)
Taper roller bearing SKF-300356	1.7	42000
Ball bearing SKF 6215	1.7	456

Ball bearing large SKF 6310	5.9	2000
Ball bearing small SKF 6306	105	2300

Since estimated life of the chosen bearings is more than the desired one, and the weight or price of different bearings in these categories are not much different, no change is proposed.

5.3 Noise and Vibration

A gear box was bought from M/s. Panchal Pumps and Systems, Kanpur and installed at the Micro Model complex of IIT Delhi, as shown in Fig. 1. The set-up was used to measure the noise and vibration level.

The following instruments were used to measure the noise and vibration of gear box:

Sound level meter; Oscilloscope; Accelerometer; and Charge Amplifier

Fig. 9(a) Brochure of Animal driven water pump (front page)

Fig. 9(b) Brochure of Animal driven water pump (back page)

5.5.1 Noise

The measured noise at the points labeled 1, 2…..10 of the gear box in Fig. 9 are shown in Table 6.

Table 6 Measured noises

Point Indicates (Noise level measured in that area of the gear box)	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**
Noise level in db (decibel)	73.5	75.45	74.25	75.35	74.85	74.5	75.15	73.65	74.25	72.5

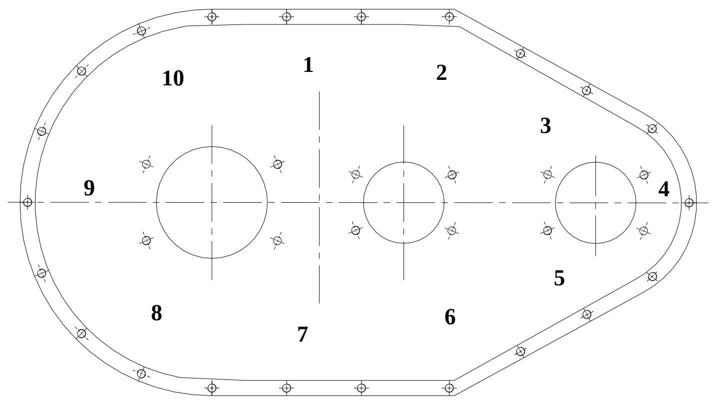
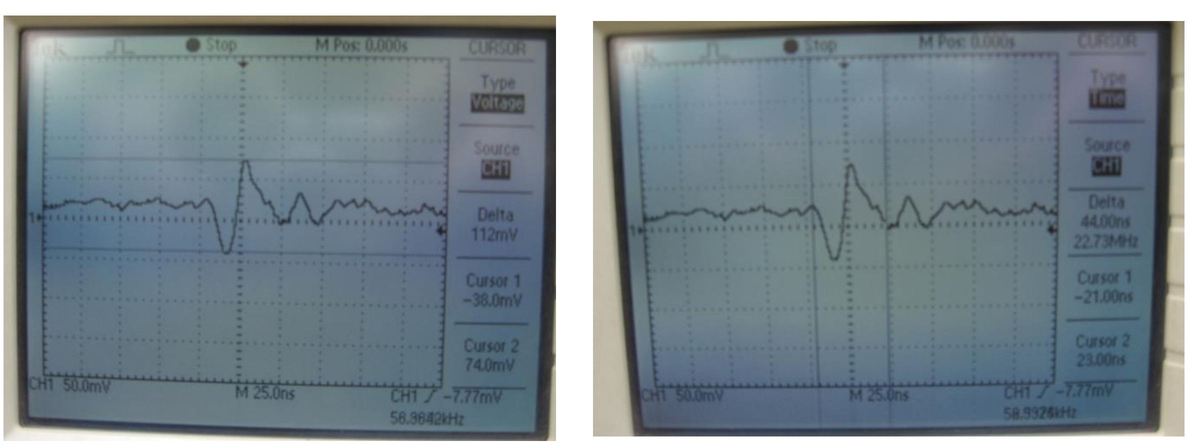


Fig. 9 Gear box

5.5.2 Vibration

The vibration level was also measured at different places (as shown in Fig. 9) of the Gear-box that was recorded on oscilloscope. The records for point 1 are shown in Fig. 10, whereas the vibration level readings for points 1, 3, 5, 7, and 9 are shown in Table 7.



(a) Voltage at location 1 (b) Frequency at location 1

Fig. 10 Oscilloscope readings

Table 7 Measured vibrations

Point Indicates (Vibration level measured in that area of the gear box)	**1**	**3**	**5**	**7**	**9**
Vibration level in mV (milivolts)	112	102	372	128	232
Vibration frequency in MHz (Mega hertz)	22.73	5.00	20.83	92.59	4.46

5.4 Efficiency

Efficiency of the gear box was calculated based on the following losses:

1. **Power loss in tooth engagement:** The power loss during the tooth engagement of the whole gear-box was estimated as 138.12 Watt.
2. **Churning power losses:** The churning power loss occurs due to viscosity of the lubricant used in the gear box, and the total churning loss was coming out to be 6.02 Watt (for SAE30 oil).
3. **Bearing losses and seal frictional losses:** Total bearing and seal loss was estimated as 7.5 Watts.

After calculating all the losses the efficiency of the gear box was expected to be 89.23%. However, experimentally obtained value was about 50%. The experiment was calculated in the following manner: as shown in Fig. 11. An alternator was attached to light car headlight bulbs. The readings to calculate the overall efficiency of the DC motor connect to the gear are shown in Table 8. If the efficiency DC motor with gear box are taken into account the gearbox under study is estimated as 70%, which is less than theoretical estimate of 89.23%. the discrepancy could be nonrealistic estimates of the losses in the gearbox.

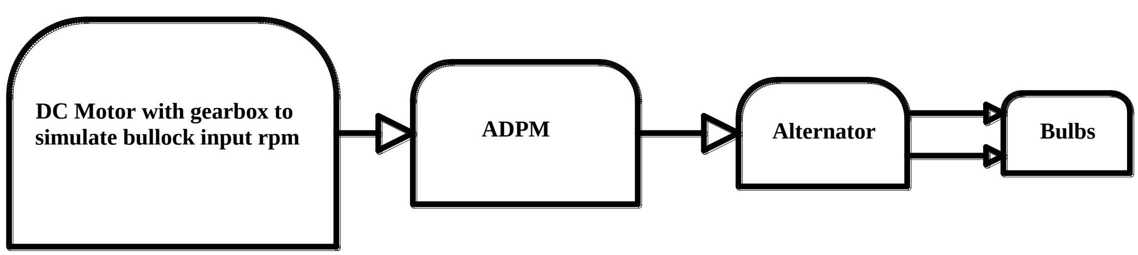


Fig. 11 Line diagram of experiment set-up

Table 8 Readings at different speeds

Motor run with load and note down the reading at different speed							
		Input Power		Output Power			
Regulator Setting	RPM (Rev.)	Voltage (Volts)	Current (Amps)	Voltage (Volts)	Current (Amps)	Efficiency (%)	Remarks
50 (90)	--	--	--	--	--	--	Bulb was not glowing
100	--	--	--	--	--	--	Bulb was not glowing
150	850	125	2.95	13.8	9	34	
200	1315	160	3.56	13.9	15*	37	
200	1250	160	3.5	13.8	16.5**	41	
270	1975	225	3	13.85	16*	33	
270	1887	220	2.86	13.8	24**	53	
* Three bulbs of 60 Watt each (one filament of each bulb was glowing)							
** Three bulbs of 60 Watt each (two filaments of each bulb were glowing)							

5. Standardization

In order to have a standardized method to evaluate an animal driven gearbox the following steps were taken.

5.1 MATLAB Program

The process of the gear-box design, i.e., spur gear and helical gear design, and other calculations of the gear-box which are of iterative type, was automated using MATLAB software, as shown in Appendix C. Later, the same calculations were performed using formulas in MS Excel which would be more convenient for any users, including the gear manufacturers, as it is readily available in all computers. A sample page for MS Excel calculations is shown in Appendix D.

5.2 Assembly and Simulation

In order to understand the gear-box and for the ease of manufacturing the final standardized gearbox, it was drawn in Autodesk Inventor. This is shown in Fig. 12. The CAD model helps one to see the animation and understand the motion of the gears.

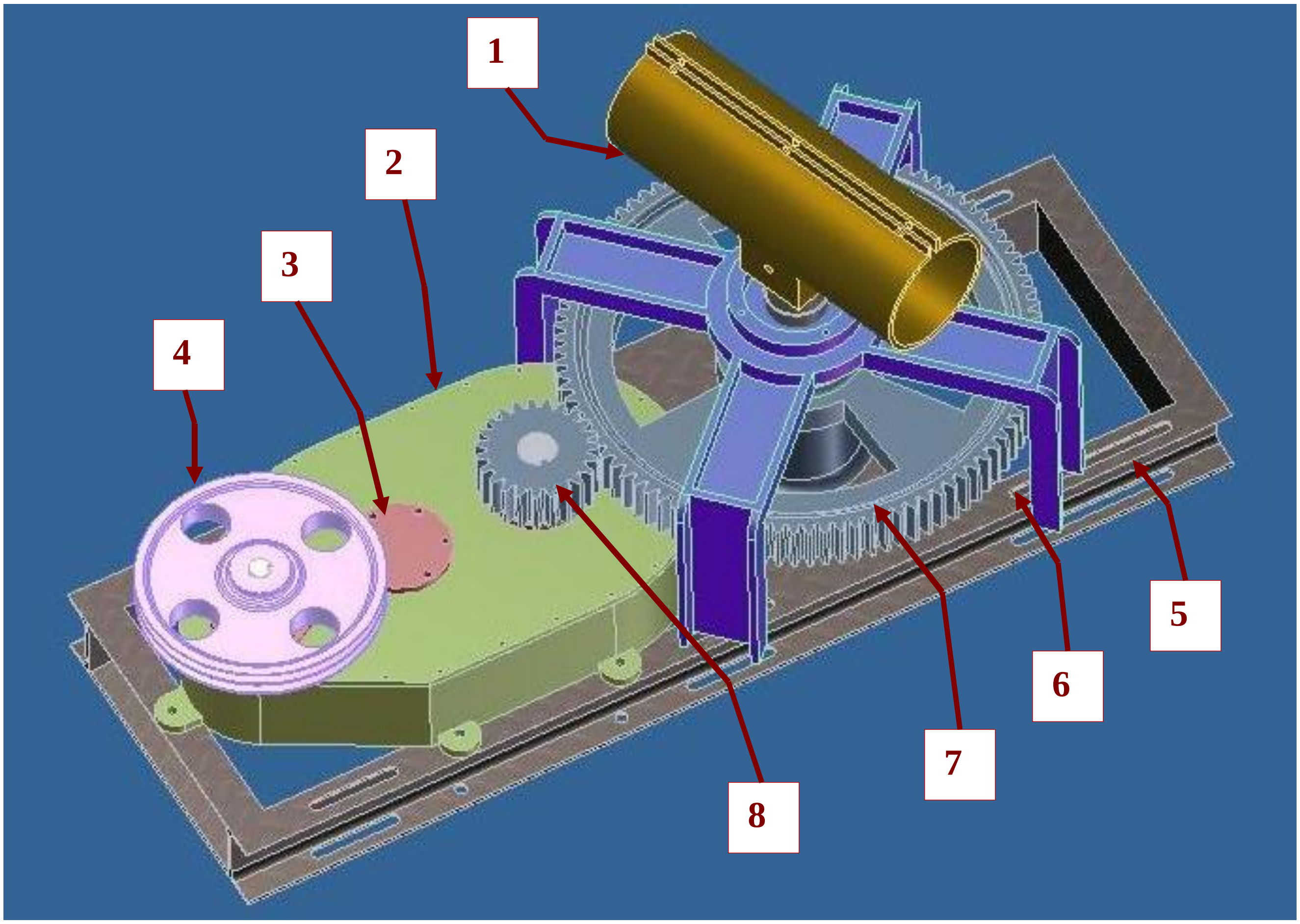


Fig. 12 Standardized gear-box in Autodesk Inventor

1. Lever Holder	2. Gearbox Casing	3. Bearing Housing
4. V-Pulley	5. Base Frame	6. Centre Locking Frame
7. Cattle Drive Gear	8. Cattle Drive Gear Pinion	

Referring to Fig. 12, the specifications of the standardized gear box are as follows.

a. Gear Pair – 1 (Spur)

Particulars	Value
Power	650 Watt
Rpm	2
Face width (gear)	45 mm
Face width (pinion)	50 mm
Module	4
Material	EN-24
FOS for bending (gear)	1.24
FOS for bending (pinion)	1.55
FOS for Surface fatigue (gear)	1.45
FOS for Surface fatigue (pinion)	1.61

b. Rib Design

Particulars	Value
Power	650 Watt
Rpm	2
Face width (gear)	45 mm
Face width (pinion)	50 mm
Module	4
Material	EN-24
Width of rib	40 mm
Depth of rib	45 mm
FOS for Rib	1.20

c. Gear Pair – 2 (Spur)

Particulars	Value
Power	637 Watt
Rpm	2
Face width gear	45 mm
Face width pinion	45 mm
Module	2.5
Material	EN-24
FOS for bending (gear)	1.25
FOS for bending (pinion)	1.27
FOS for Surface fatigue (gear)	1.17
FOS for Surface fatigue (pinion)	1.30

d. Rib Design

Particulars	Value
Power	637 Watt
Rpm	8
Face width (gear)	45 mm
Face width (pinion)	45 mm
Module	2.5
Material	EN-24
Width of rib	16 mm
Depth of rib	18 mm
FOS for Rib	1.42

e. Gear Pair – 3 (Spur)

Particulars	Value
Power	625 Watt
Rpm	8
Face width (gear)	22 mm
Face width (pinion)	24 mm
Module	2.5
Material	EN-24
FOS for bending (gear)	11.36
FOS for bending (pinion)	10.59
FOS for Surface fatigue (gear)	1.72
FOS for Surface fatigue (pinion)	1.91

f. Rib Design

Particulars	Value
Power	625
Rpm	34
Face width (gear)	22
Face width (pinion)	45
Module	2.5
Material	EN-24
Width of rib	16 mm
Depth of rib	16 mm
FOS for Rib	4.91

g. Shafts

Parameters	Cattle Drive Gear Shaft	Drive Gear Shaft	Idler Gear Shaft	Driven Gear shaft
	Normal Stress Theory	Max. shear Stress Theory		
Static Loading				
Power (P) in Watt	650	650	637	624
RPM (N)	2	4	8	34
Minimum Dia. of Shaft (d) in mm	5.80E-02	3.80E-02	3.00E-02	2.90E-02
Dia. of Gear (D) in mm	5.70E-01	2.35E-01	2.20E-01	4.60E-01
Material Strength				
Material	Cast Iron	EN8	EN8	EN8

h. Bearing

		Bearing type*
Cattle drive	Upper bearing	Ball bearing large SKF 6213
	Lower bearing	Taper bearing 32213
Drive shaft	Upper bearing	Ball bearing small SKF 6308
	Lower bearing	Ball bearing small SKF 6308
Idler shaft	Upper bearing	Ball bearing small SKF 6306
	Lower bearing	Ball bearing small SKF 6306
Driven shaft	Upper bearing	Ball bearing small SKF 6306
	Lower bearing	Ball bearing small SKF 6306

***** It is assumed that the gearbox will run for 5 hours every day for 250 working days in a year for 10 years.

5.3 Test

The standardized gear-box was tested at Lav-Kush Ashram, Karoli, Kanpur during April 26 to May 02, 2011. For the purpose of comparison with existing gearbox, both the gearboxes were tested under same test conditions, i.e., by a same pair of bullocks, etc. The test results obtained during the testing are shown in Appendix E.

5.3.1 Specifications

a. Gear-box Size

Length: 1200 mm; Width: 450 mm; Height: 450 mm; Stage: Three; Ratio: 63:1;
Belt pulley: V-belt with 4:1 ratio

b. Water pump (Screw Pump/Low rpm Pump)

Type: Screw pump; Head: 10 m; Suction pipe dia.: 85 mm; Discharge pipe dia.: 65 mm; Length/height of screw: 615 mm; Lever length: 5 m; Discharge: average 10,000 ltres per hours.

5.3.2 Efficiency

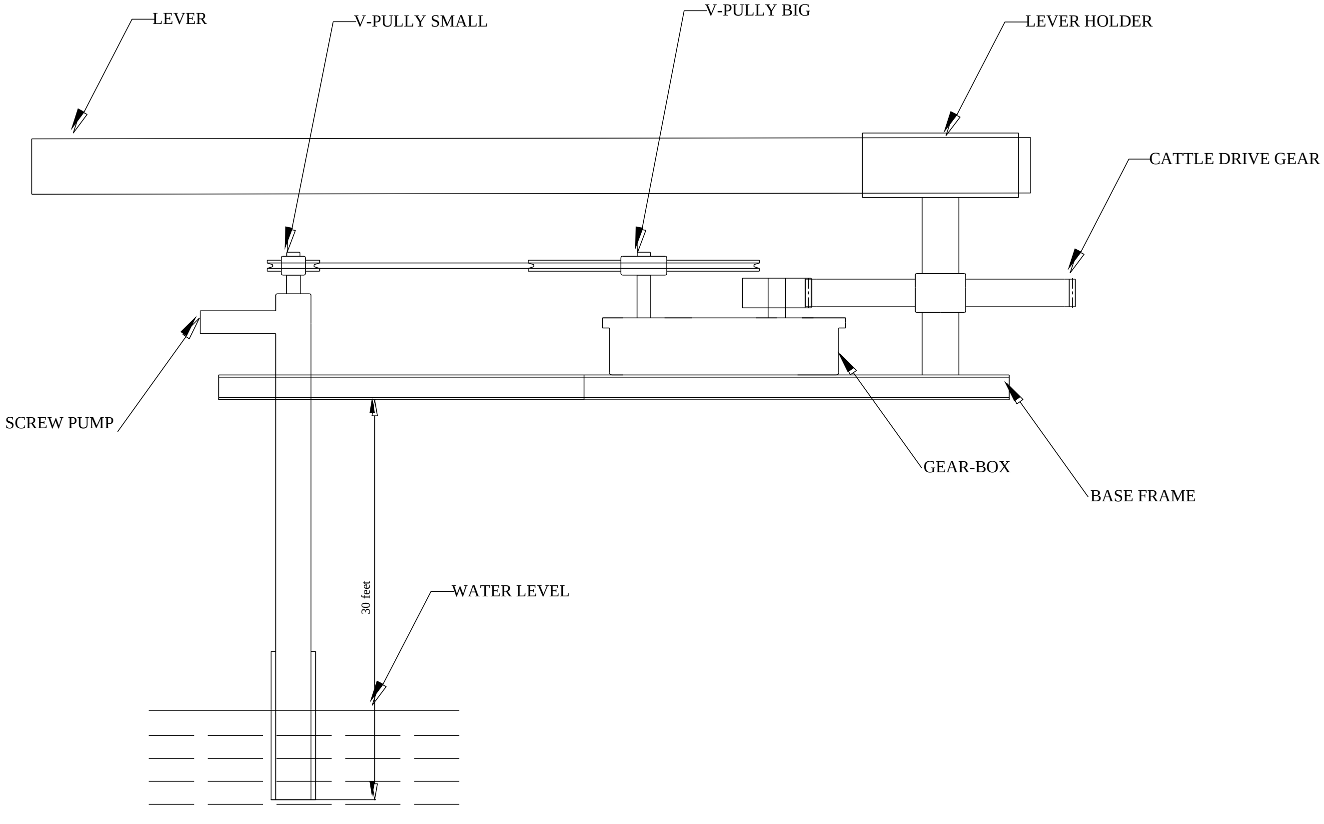
Gear box: ~70% [measured with the help of an alternator to charge a car battery and switching on head lights of cars]; Belt efficiency: ~98%; Screw pump efficiency: ~75% Overall ADPM-driven water pump: ~50%.

Table 9 Comparison of existing and standardized gear-box

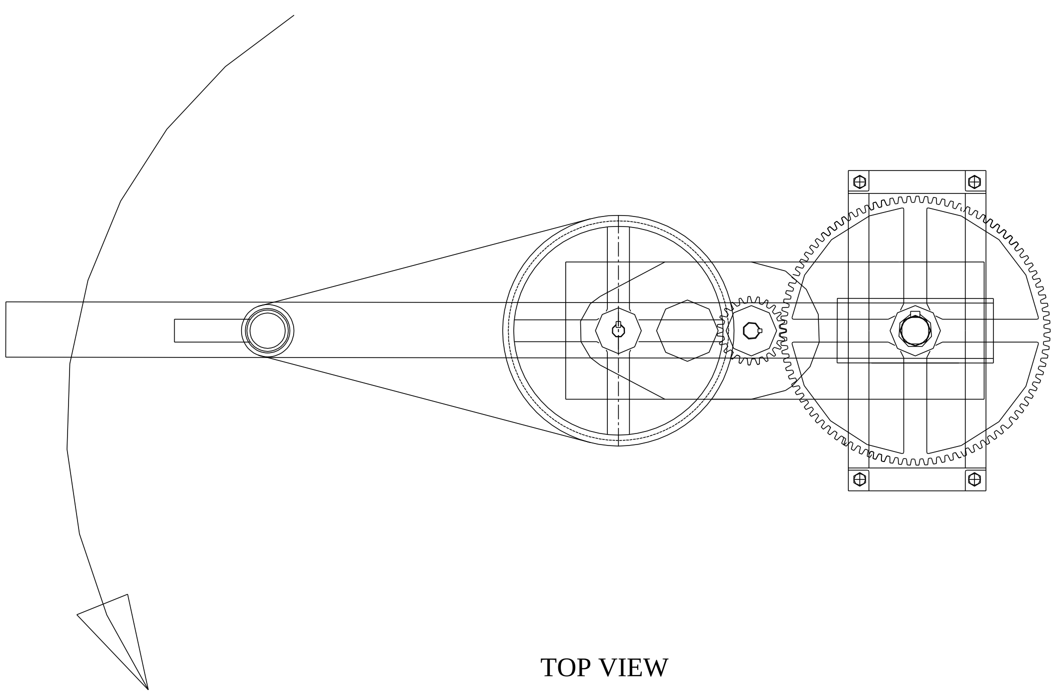
Serial No.	Particular	Existing gear-box	Standardized gear-box
1.	Efficiency		~50%
2.	Flow rate	Avg. 15,000 litres per hours	Avg. 10,000 litres per hours
3.	Water lift (Height)	(30, 40, 50, 60, 70…) feet	(30, 40, 50, 60, 70…) feet
4.	Ratio	63:1	63:1
5.	Overall Size	Length: 1350 mm; Width: 450 mm; Height: 500 mm;	Length: 1200 mm; Width: 450 mm; Height: 450 mm;
6.	Total weight	175 Kg.	125 Kg.

6. Workshop Brochure

A workshop was organized at Lok Bharti, Lucknow, Uttar Pradesh on September 14 and 15, 2011. The brochure used to share the information on the project shown in appendix F.



FRONT VIEW



TOP VIEW

Fig. 13 Schematic diagram of animal driven gear-box

7. Conclusions and Future plan

Design calculations of the initial gearbox given in Section 4 were carriedout first, which showed that some of the gears were overdesigned. Moreover, it was felt from the first field trial in 2009 and the interactions with the farmers that 1.3KW power input is relatively high. It was found that the water flow at the rate of 10,000 liters per hour in comparison to15,000 liters per hour with the existing gearbox is good enough for farming purposes, particularly, with the help of sprinklers. Hence, redesigning was done by the manufacturer (M/s. Panchal Pumps and Systems, Kanpur) and the gearbox was made much lighter, as explained in Section 5. The present gearbox is suitable for 650Watt input, which produced high discharge rate with new standardized gear box (10,000 liters per hour) compared to the existing type (8,000 liters per hour), as shown in Appendix E. Also the price of the gearbox was brought down from Rs. 32,500 to Rs. 28,500. However, the total cost of the standardized gearbox with screw pump, installation, etc. is Rs. 95,431/- for 30 feet water level in comparison to the existing one which costs Rs. 1,25,000/-. Hence, there is an overall saving of Rs. 30,000/- which can be attributed to the study of this project and the design improvements. Further, a set of design procedure is available in MS-Excel which even a non-engineer can use. In addition, the test set-up created at IIT Delhi will allow one to compare any two gear-boxes objectively without any bias.

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3. Robert L. Norton, “Machine Design: An Integrated Approach, “Second Edition, Pearson Education Asia”, pp. 683-768, 2001.

Appendix A

Sample Discussion Page

8/5/2010 Saturday 9:30 AM.

DATE / /

Prof. S.K. Saha, Dr. Modak, Mr. Amit Jain, Mr. Raj Kumar, Mr. Mangal Sharma

1) ~~Call to~~ Mr. Shrivastav for project interaction – 11/5/10 on 9:30 AM

Gear ratio (

Estimate the forces on the gear based on the power provided by the bullock

Calculate stress in the gear teeth and find F.O.S. Redo

Check shaft & bearing

14/05/2010

1) Gear ratio —: (Cattle drive gear)

No. of teeth – 96

No. of teeth of pinion – 23

$$\text{Ratio} = \frac{96}{23} = 4.17 \Rightarrow 4.17:1$$

2) Drive gear (Spur)

No. of teeth — 77

No. of teeth of idler gear — 16

$$\text{Ratio} = \frac{77}{16} = 4.81 \Rightarrow 4.81:1$$

3) Driven gear (Helical)

No. of teeth – 35

No. of teeth of Idler gear – 110

$$\text{Ratio} = \frac{110}{35} = 3.14 \Rightarrow 3.14:1$$

4) V-pulley to Pump –

V-pulley OD — 300

Pump pulley OD — 125

$$\text{Ratio} = \frac{300}{125} = 2.4 \Rightarrow 2.4:1$$

Appendix B

Design Calculations

B.1 Definitions

The terms most commonly used to describe gears are:

- *Pinion:* the smaller of a pair of meshing gears.
- *Gear:* the larger of a pair of meshing gears.
- *Ratio:* number of teeth in gear / number of teeth in pinion.

Pitch circle: a theoretical circle upon which all calculations are based. The *operating pitch circles* of a pair of gears in mesh are tangent to each other.

Circular pitch *(p):* the distance, measured on the theoretical pitch circle from a point on one tooth to a corresponding point on an adjacent tooth. The circular pitch is measured in inches or in millimeters. The circular pitch is the sum of the *tooth thickness t* and the *width of space.*

Pitch diameter *(d* for the pinion) and (*D* for the gear): the diameter of the pitch circle; it is measured in inches or in millimeters.

Module *(m):* the ratio of the theoretical pitch diameter to the number of teeth *N*. The module is the metric index of tooth sizes and is always given in millimeters.

Diametral pitch *(p_d):* the ratio of the number of teeth on a gear to the theoretical pitch diameter. It is the index of tooth size when U.S. customary units are used and is expressed as teeth per inch.

Addendum *(a):* the radial distance between the top land and the pitch circle.

Dedendum *(b):* the radial distance between the pitch circle and the *root circle*. Whole depth h_t is the sum of the addendum and dedendum.

Clearance circle *(c):* tangent to the addendum circle of the mating gear. The distance from the clearance circle to the bottom land is called the *clearance c.*

Backlash: the amount by which the width of a tooth space exceeds the thickness of the engaging tooth measured on the pitch circle.

Undercutting (*u):* it occurs under certain conditions when a small number of teeth are used in cutting a gear.

For standard gears; the following conventions are used:

External diameter = $d + 2m$; Tooth thickness = $0.5m$; Tooth height = $2m + c$;

Besides, some more formulas and some standards are given in Table B.1 and B.2 respectively.

Table B.1 Basic formulas for spur gears

Quantity desired	**Formula**
Diametral pitch, p_d	$p_d = N/d$
Module, m	$m = d/N$
Circular pitch, p	$p = \pi m$
Pitch diameter, d	$d = mn$

Table B.2 Standard tooth systems for spur gears

Tooth system	Pressure angle, ϕ	Addendum, a	Dedendum, b
Full Depth	20 22.5 25	1m 1m 1m	1.25m-1.35m 1.25m-1.35m 1.25m-1.35m
Stub	20	0.8m	1m

B.2 Force Analysis in Gear

Based on the theories presented in [2] or [3], definitions and steps needed to design different gears are summarized next. A gear exerts force W against the pinion at pitch point *P* as shown in Fig. B.1. This force is resolved into two components, a radial force W_r, acting to separate the gears, and a tangential component W_t, which is called the *transmitted load.* Equal and opposite to force W is the shaft reaction F. Force F and torque T are exerted by the shaft on the pinion. Note that torque T opposes the force couple made up of W_t and F_x separated by the distance d/2. Thus

$$T = W_t * d/2 \qquad \text{(B.1)}$$

where,

T : torque (N); W_t : transmitted load (N); d : Pitch diameter (m)

Accordingly, the power transmitted is

$$P = W_t\, v \qquad \text{kW} \qquad \text{(B.2)}$$

Where v is pitch-line velocity which is given by

$$v = \Pi d n_p /60 \quad \text{m/s} \qquad \text{(B.3)}$$

in which n_p is the pinion speed in rpm.

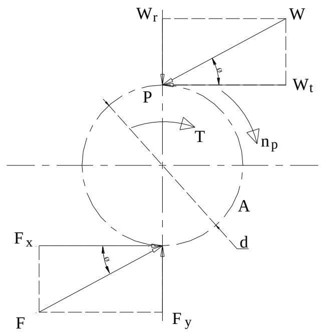

Fig. B.1 Forces on pinion

B.3 Helical Gear

Helical gear in which the teeth are cut at an angle with respect to the axis of rotation is a later development than spur gear and has the advantage that the action is smoother and tends to be quieter. In addition, the load transmitted may be somewhat larger, or the life of the gears may be greater for the same loading than with an equivalent pair of spur gears. Helical gears produce an end thrust along the axis of the shafts in addition to the separating and tangential (driving) loads of spur gears. Where suitable means can be provided to take this thrust, such as thrust collars or ball or tapered-roller bearings, it is of great advantage. When considered in the transverse plane (i.e., a plane perpendicular to the axis of the gear), all helical-gear geometry is identical to that for spur gears. Table B.3 provides some important formulas for helical gears.

Table B.3 Formulas for helical gears

Quantity	**Formula**
Addendum	$1/p_N$
Dedendum	$1.25/ p_N$
Pinion pitch diameter	$N_P/p_N \cos\Psi$
Gear pitch diameter	$N_G/p_N \cos\Psi$
Normal arc tooth radius	$\pi/p_N - B_N/2$
Pinion base diameter	$d \cos\Phi_t$
Gear base diameter	$D \cos\Phi_t$
Base helix angle	$\tan^{-1} (\tan\Psi \cos \Phi_t)$
External gears	
Standard centre distance	$(D+d)/2$
Gear outside diameter	$D+2a$
Pinion outside diameter	$d+2a$
Gear root diameter	$D-2b$
Pinion root diameter	$d-2b$
Internal gears	
Centre distance	$(D-d)/2$
Inside diameter	$d-2a$
Root diameter	$D+2b$

Where,

P_N : normal diametral pitch; Φ_t : transverse pressure angle (degrees);

N_P : number of teeth on pinion; N_G : number of teeth on gear;

Φ_0 : operating transverse pressure angle (degrees); Ψ : helix angle;

B.4 Bending Stresses

Note that a gear is specified by its material, face width, and the module. For its stress analysis, bending stress is calculated as

$$\sigma_b = \frac{W_t}{f\, m\, Y} \tag{B.4}$$

where

W_t is tangential load, f is face width, m is module, and Y is Lewis form factor.

B.5 Surface Fatigue

For a pair of gears, surface fatigue stress can be given by [1, 2]:

$$B = \frac{1}{d_1} + \frac{1}{d_2}$$

$$\sigma_c = \frac{2\ W_t\, B}{\pi\ f\ \cos\varphi\ (m_1 + m_2)}$$

$$m_1 = \frac{1 - \mu_1^2}{E_1} \qquad m_2 = \frac{1 - \mu_2^2}{E_2} \tag{B.5}$$

where,

d_1 and d_2 are the pitch circle diameters of gear and pinion, respectively.

μ_1 and μ_2 are the Poisson's ratio for the gear and pinion, respectively.

E_1 and E_2 are the Young's modulus for the gear and pinion, respectively.

φ is the pressure angle of gear pair.

B.6 Shaft

For the design of shafts, a free-body diagram of shaft 1 is shown in Fig. B.2.

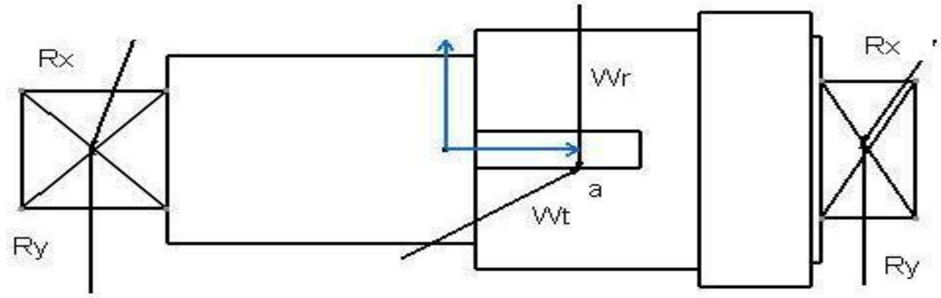

Fig. B.2 Free-body diagram of shaft 1

To find out the FOS of the shaft, the following formulas were used [1,2]:

$$N_f^{\,2} = (\sigma_a / \sigma_e)^2 + (1.73 * \tau_m / \sigma_y)^2 \tag{B.6}$$

$$\sigma_a = k_f * (32 * Ma / \pi * d^3) \ MPa \tag{B.7}$$

$$\tau_m = k_{fs} * (16 * Tm / \pi * d^3) \ MPa \tag{B.8}$$

$$k_f = 1 + q * (k_t - 1),\ k_{fs} = 1 + q * (k_{ts} - 1) \tag{B.9}$$

Where, N_f is factor of safety; σ_a and τ_m are bending and shear stresses, respectively; M_a and T_m are bending and torsion moment, d is the diameter of shaft.

B.7 Bearings

For bearing selections in the gear-box, the equivalent load on bearing is given by (1, 2)

$$P = X*V*F_r + Y*F_a \tag{B.10}$$

where, P: equivalent load; F_r: applied constant radial load; F_a: applied constant thrust load; Y: a thrust factor; X: a radial factor

Life of the bearing is estimated as

$$L = (C / P)^3 \times 10^6 \text{ revolutions} \tag{B.11}$$

where, L is fatigue life expressed in millions of revolutions; C is the basic dynamic load rating for the particular bearing

Appendix C

Computer Programs in MATLAB Software

Matlab programs were prepared to analyze the gear pairs of two kinds, i.e., spur and helical. The folder named "Matlab Gear" would look as shown in Fig. C.1.

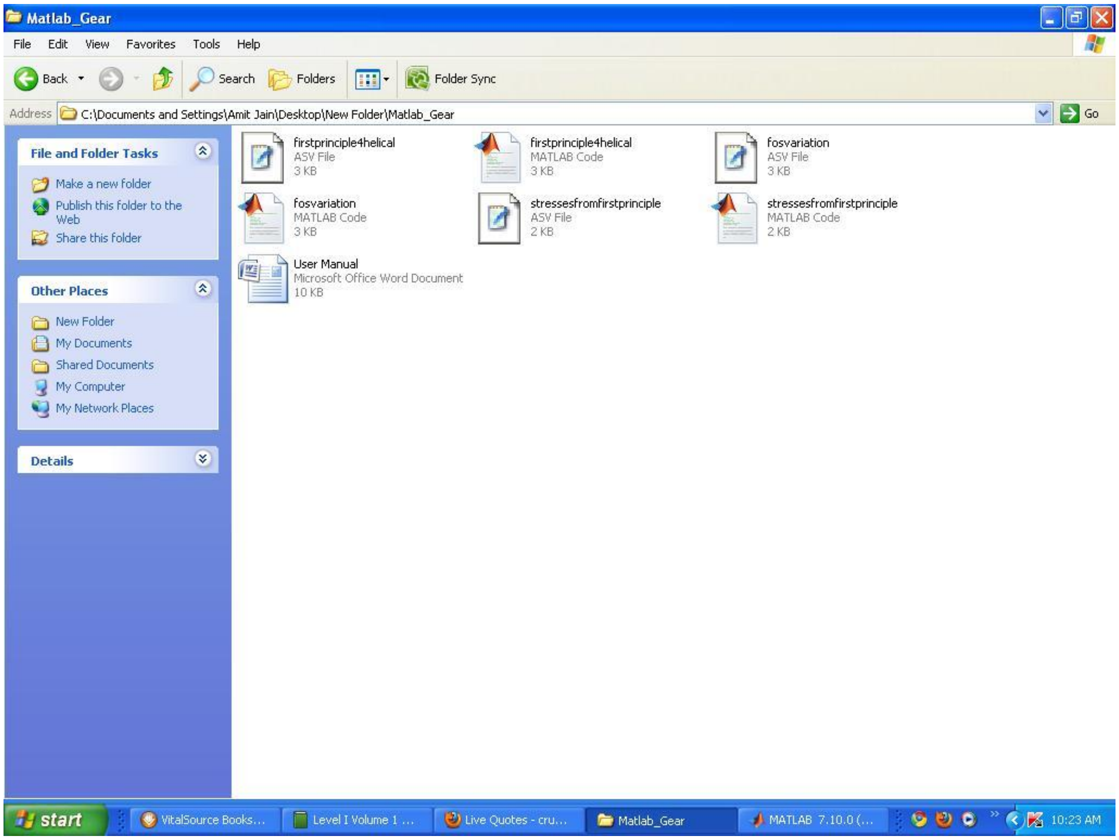

Fig. C.1 Folder "Matlab Gear"

There are three self explanatory programs, namely 'firstprinciple4spur', 'firstprinciple4helical', and 'fosvariation.' Each of these programs contains three parts, namely, the input parameters, analyses, and output parameters. Program for analyses of a helical gear pair looks like

```
***********************Program starts*************************
clear all
clc
%%% This program is made by Amit Jain
%%% Last updated on Feb 22, 2011
%%% Analyses of helical gear pair
%%% Bending fatigue stress from first principle for helical gears
```

%% Input parameters

```
 P=input('P=');                     %Power input in W
N=input('N=');                      %rpm
d=input('d=');                      %diameter of pinion(if power given is at pinion)in mm
zp=input('zp=');                    %pinion no of teeth (35)
zg=input('zg=');                    %gear no of teeth (110)
dg=input('dg=');                    %gear dia in mm (284)
dp=input('dp=');                    %pinion dia in mm (88)
phi=input('phi=');                   %pressure angle 15(0.2617c)
F=input('F=');                      %face width in mm(35)
Y=input('Y=');                      %lewis form factor from fig 14-7 shigley p912
Sb1=input('bending fatigue strength=');
Sc1=input('surface fatigue strength=');
```

%% Analyses

```
T=60*P*1000/(2*pi*N);               %torque in N-mm
Wt=2*T/d ;                          %tangential force in N
 phi1=pi*phi/180;
shi=0;                               %helix angle (corrected after Email from Paschal)
shi1=pi*shi/180;
phin=atan2(tan(phi1)*cos(shi1),1);   %normal pressure angle
Wr=Wt*tan(phi1);                     %radial force in N
Wa=Wt*tan(shi1);                     %axial force in N
W=Wt/cos(phin)*cos(shi1);            %resultant force in N
m=dg/zg;%input('m=');               %transverse module
mn=m*cos(shi1);                     %normal module
Sb=Wt/(F*mn*Y);                     %bending stress in MPa
```

%%%Surface fatigue stress from first principle for helical gears

```
pd=zp/dp;                           %diametral pitch
C=(dp+dg)./2;                       %center distance
a=1./pd;
Z=(((dp/2)+a)^2-(dp*cos(phi1)/2)^2)^0.5+(((dg/2)+a)^2-(dg*cos(phi1)/2)^2)^0.5-C*sin(phi1);
```

```
Mp=pd*Z/(pi*cos(phi1));
px=pi/(pd*tan(shi1));
Mf=F./px;
 nr=Mp-floor(Mp);
na=Mf-floor(Mf);

Lmin=Mp*F-((1-na)*(1-nr)*px)/cos(shi1);
Mn=F/Lmin;

B=((1/dg)+(1/dp));
vg=0.28;%input('vg=');              %poissons ratio for gear(0.28)
vp=0.28;%input('vp=');              %poissons ratio for pinion (0.28)
Eg=210000;%input('Eg=');            %modulous of elasticity for gear in MPa(210000)
Ep=210000;%input('Ep=');            %modulous of elasticity for pinion in MPa(210000)
mg=(1-(vg^2))/Eg;
mp=(1-(vp^2))/Ep;
Sc=(2*W*B*Mn/(pi*F*(mg+mp)))^(0.5);     %surface fatigue stress in MPa
```

%% Output Parameters

```
% Factor of Safety (In bending and Surface Fatigue)
fosb=Sb1./Sb
fosc=Sc1./Sc
```

Program for analyses of spur gear pair look like

```
clear all
clc
%%% This program is made by Amit Jain
%%% Last updated on Feb 22, 2011
%%% Analyses of spur gear pair
%%% Bending fatigue stress from first principle for spur gears
```

%% Input parameters

```
display('Enter motion properties');
P=input('P=');                      %Power input in W
N=input('N=');                      %rpm
display('Enter dimensional properties');
phi=input('phi=');                  %pressure angle in degree
```

```
d=input('d=');                %diameter in mm
F=input('F=');                %face width in mm
m=input('m=');                %module
Y=input('Y=');                %enter value of lewis form factor from table
dg=input('dg=');              %gear dia in mm
dp=input('dp=');              %pinion dia in mm
display('Enter material properties');
vg=input('vg=');              %poissons ratio for gear
vp=input('vp=');              %poissons ratio for pinion
Eg=input('Eg=');              %modulous of elasticity for gear in MPa
Ep=input('Ep=');              %modulous of elasticity for pinion in MPa
Sb1=input('bending fatigue strength=');
Sc1=input('surface fatigue strength=');
```

%% Bending fatigue stress from first principle

```
T=60*P*1000/(2*pi*N);         %torque in N-mm
Wt=2*T/d;                     %tangential force in N
phi1=pi*phi/180;
Wr=Wt*tan(phi1);              %radial force in N
W=Wt/cos(phi1);               %resultant force in N
Sb=Wt/(F*m*Y);                %bending stress in MPa
```

%% Surface fatigue stress from first principle

```
B=((1/dg)+(1/dp));
mg=(1-(vg^2))/Eg;
mp=(1-(vp^2))/Ep;
Sc=(2.*W.*B./(pi.*F.*(mg+mp))).^0.5;%surface fatigue stress in MPa
```

%% Factor of safety

```
fosb=Sb1/Sb
fosc=Sc1/Sc
```

***********************Program ends*************************

Appendix D

**********************Analysis of Spure Gear pair (Cattle Drive Gear)************

Input******************************	Inputs***************************
Power (watt)	650
Rpm of Input gear	2
No. of teeth on Gear	145
No. of teeth on pinion	35
Pressure angle (deg)	20
Pitch diameter of gear (mm)	580
Pitch diameter of pinion (mm)	140
Face width of gear (mm)	45
Face width of pinion (mm)	50
Poision ratio for Gear material	0.28
Poision ratio for Pinion material	0.28
Modulus of Elasticity of Gear material (Mpa)	150000
Modulus of Elasticity of Pinion material (Mpa)	206000
module of gear pair (mm)	4
Bending Strength of Gear (Mpa)	160
Bending Strength of Pinion (Mpa)	207
Surface Fatigue Strength of pinion (Mpa)	592
Surface Fatigue Strength of gear (Mpa)	534

outputs andCalculations*********************************

Gear Ratio	4.142857143
Angular Speed of gear (rad/sec)	0.209333333
Angular Speed of pinion (rad/sec)	0.867238095
Lewis form factor for Gear	0.147710345
Lewis form factor for pinion	0.127942857
Coefficient for gear (/Mpa)	0.000006144
Coefficient for pinion (/Mpa)	4.47379E-06
Curvature Coefficient (/mm)	0.008866995
Pressure angle in radian	0.348888889

Calculation for Torque and Tangential Load****************

Torque acting on the gear (N.m)	3105.095541
Torque acting on the pinion (N.m)	749.5058203
Tangential load on gear (N)	10707.226
Tangential load on pinion(N)	10707.226

Bending Stress***********************

Bending stress in gear (Mpa)	128.2519286
Bending stress in pinion (Mpa)	133.2604517

**Calculation for Surface Fatigue Stress

Surface Fatigue Strength of Gear pair (Mpa)	366.983681

safety consideration **************************

FOS of gear (bending)	1.247544593
FOS of Pinion (bending)	1.553349079
FOS of gear (Surface Fatigue)	1.455105575
FOS of Pinion (Surface fatigue)	1.613150749
All FOS should be greater than one for safe design	SAFE

**********************Analysis of Spure Gear pair (Drive Gear)*************************

Input******************************** **Inputs********************************

Power (watt)	637
Rpm of Input gear	8
No. of teeth on Gear	94
No. of teeth on pinion	22
Pressure angle (deg)	20
Pitch diameter of gear (mm)	235
Pitch diameter of pinion (mm)	55
Face width of gear (mm)	40
Face width of pinion (mm)	40
Poision ratio for Gear material	0.28
Poision ratio for Pinion material	0.28
Modulus of Elasticity of Gear material (Mpa)	150000
Modulus of Elasticity of Pinion material (Mpa)	206000
module of gear pair (mm)	2.5
Bending Strength of Gear (Mpa)	160
Bending Strength of Pinion (Mpa)	207
Surface Fatigue Strength of pinion (Mpa)	592
Surface Fatigue Strength of gear (Mpa)	534

outputs and Calculations

Gear Ratio	4.272727273
Angular Speed of gear (rad/sec)	0.837333333
Angular Speed of pinion (rad/sec)	3.57769697
Lewis form factor for Gear	0.144297872
Lewis form factor for pinion	0.112545455
Coefficient for gear (/Mpa)	0.000006144
Coefficient for pinion (/Mpa)	4.47379E-06
Curvature Coefficient (/mm)	0.022437137
Pressure angle in radian	0.348888889

*************************************Calculation for Torque and Tangential Load************************

Torque acting on the gear (N.m)	760.7484076
Torque acting on the pinion (N.m)	178.0474997
Tangential load on gear (N)	6474.454533
Tangential load on pinion(N)	6474.454533

Bending Stress*****************************

Bending stress in gear (Mpa)	142.8938738
Bending stress in pinion (Mpa)	183.2084826

***Calculation for Surface Fatigue Stress**

Surface Fatigue Strength of Gear pair (Mpa)	481.4837242

safety consideration *******************************

FOS of gear (bending)	1.119712104
FOS of Pinion (bending)	1.129860348
FOS of gear (Surface Fatigue)	1.109071757
FOS of Pinion (Surface fatigue)	1.229532735
All FOS should be greater than one for safe design	SAFE

********************Analysis of Spure Gear pair (Driven Gear)***************

*************************************Input******************************* **Inputs****************************

Power (watt)	624
Rpm of Input gear	34
No. of teeth on Gear	110
No. of teeth on pinion	31
Pressure angle (deg)	20
Pitch diameter of gear (mm)	220
Pitch diameter of pinion (mm)	62
Face width of gear (mm)	22
Face width of pinion (mm)	24
Poision ratio for Gear material	0.28
Poision ratio for Pinion material	0.28
Modulus of Elasticity of Gear material (Mpa)	150000
Modulus of Elasticity of Pinion material (Mpa)	206000
module of gear pair (mm)	2
Bending Strength of Gear (Mpa)	160
Bending Strength of Pinion (Mpa)	207
Surface Fatigue Strength of pinion (Mpa)	592
Surface Fatigue Strength of gear (Mpa)	534

outputs and Calculations********************************

Gear Ratio	3.548387097
Angular Speed of gear (rad/sec)	3.558666667
Angular Speed of pinion (rad/sec)	12.62752688
Lewis form factor for Gear	0.145709091
Lewis form factor for pinion	0.124580645
Coefficient for gear (/Mpa)	0.000006144
Coefficient for pinion (/Mpa)	4.47379E-06
Curvature Coefficient (/mm)	0.020674487
Pressure angle in radian	0.348888889

**************************************Calculation for Torque and Tangential Load********************

Torque acting on the gear (N.m)	175.3465717
Torque acting on the pinion (N.m)	49.41585204
Tangential load on gear (N)	1594.059743
Tangential load on pinion(N)	1594.059743

Bending Stress***********************

Bending stress in gear (Mpa)	79.18366898
Bending stress in pinion (Mpa)	84.89519947

**Calculation for Surface Fatigue Stress

Surface Fatigue Strength of Gear pair (Mpa)	309.2321333

safety consideration *************************

FOS of gear (bending)	2.020618671
FOS of Pinion (bending)	2.438300414
FOS of gear (Surface Fatigue)	1.726858054
FOS of Pinion (Surface fatigue)	1.914419416
All FOS should be greater than one for safe design	SAFE