DESIGN, MODIFICATION AND PERFORMANCE EVALUATION OF SELF PROPELLED BUSH CUTTER

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PROJECT REPORT

Submitted in partial fulfilment of the requirement for the degree of

BACHELOR OF TECHNOLOGY IN AGRICULTURAL ENGINEERING

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KELAPPAJI COLLEGE OF AGRICULTURAL ENGINEERING AND TECHNOLOGY
Tavanur - 679573, Malappuram

1998

DECLARATION

We hereby declare that this project report entitled "**DESIGN, MODIFICATION AND PERFORMANCE EVALUATION OF SELF-PROPELLED BUSH CUTTER**" is a bonafide record of project work done by us during the course of project and that this report has not previously formed the basis for the award to us of any degree, diploma, associateship, fellowship or other similar title to us, of any other University or Society.

Tavanur
30-4-1998

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CERTIFICATE

Certified that this project report entitled "**DESIGN, MODIFICATION AND PERFORMANCE EVALUATION OF SELF-PROPELLED BUSH CUTTER**" is a bonafide record of project work done jointly by Biju K. Varghese, Rajiv Aravindan and Rema. P.P under my guidance and supervision and that it has not previously formed the basis for the award of any degree, diploma, associateship or fellowship to them.

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LIST OF ABBREVIATIONS USED

Agric.	-	Agriculture
Agrl.	-	Agricultural
ASAE	-	American Society of Agricultural Engineers
BSW	-	British Standard Whitworth
c/c	-	Centre to centre
cm	-	centimetre
Dept.	-	Department
dg.	-	degree(s)
dia.	-	diameter
Ed.	-	Edition
Eff.	-	effective
eg.	-	example
Engg.	-	Engineering
et al.	-	and others
etc.	-	etcetra
Fig	-	Figure
ha	-	hectare
HP	-	Horse power
hr	-	hour
KCAET	-	Kelappaji College of Agricultural Engineering and Technology
kg	-	kilogram
km	-	kilometre
KW	-	kilowatt
lit.	-	litre
m	-	metre
mm	-	millimetre
MS	-	Mild Steel

No.	-	Number(s)
pp.	-	page(s)
PTO	-	Power Take Off
rpm	-	revolution per minute
Rs.	-	Rupees
s	-	second(s)
Trans.	-	Transactions
o	-	degrees
/	-	per
"	-	inches
%	-	percentage
@	-	at the rate of

INTRODUCTION

Bush clearing, both in the farm as well as in human inhabited areas is a labour intensive operation and hence an expensive affair. Reclamation of the bush infested land becomes all the more difficult with the number of days it remains unattended. It is estimated that about 8 man-days are required for clearing a hectare of the bush infested land. Soon after the withdrawal of the monsoon, there is enough moisture in the soil for the bush growth and the problem becomes much more alarming. In this era of labour shortage and considering the expense it involves, it becomes necessary to develop a suitable machinery to meet the beforesaid requirements at reasonable cost of operation.

The bush cutter is a versatile machinery. It can be used for land reclamation and even for landscaping works. In Kerala scenario context, the bush cutter is extremely useful when worked between the widespaced perennials like coconut. The bush cutter has an additional advantage over the lawn mower in that it can deal with even a heterogenous mass of bush. Keeping these points in view a self propelled bush cutter was developed at the KCAET, Tavanur. The bush cutter had a field capacity of 0.105ha/hr and a field efficiency of 58.33%. However the operation of the bush cutter proved tedious and the cutting was not efficient. Further, the cutting system was not fabricated as per design considerations. To solve the problems involved in operating the machine and to evaluate the working of the bush cutter, certain modifications of the existing model were done. As it stands, the major objectives of the work are

i) to modify and improve the bush cutter, and

iii) to evaluate the performance parameters of the improved bush cutter.

REVIEW OF LITERATURE

A brief review of the various methods adopted in cutting shrubs, forage crops etc. are dealt in this chapter.

2.1 Manual Methods

Manual harvesting is done by a hand sickle or hand dropper. Studies conducted on smooth and serrated edged sickles indicate that both are equally effective in cutting the plants. Sickles with serrated edges were light in weight and required less or no cutting force (Michael and Ojha, 1978). Manually operated push type harvester called the hand dropper, was developed and used in Japan. It consists of a pair of serrated blades mounted at an angle of 30° to each other, a mechanism to retain the harvested plant and a lever to drop them. Its capacity is reported to be about 0.20 ha/day. In India, this equipment was not successful. Manual methods are outdated mainly because of irritation caused to the human skin while doing the cutting operation.

2.2 Mechanical Methods

Harvesting machines are usually equipped with two types of cutting mechanism, the reciprocating and rotary type cutter. Reciprocating cutters are mostly used for harvesting cereals, pulses and oil seeds while the rotary type mechanism is mostly used in harvesting forage crops and grasses. The advantage of rotary type movers lies in its simplicity in construction, sturdiness and less wearing parts and therefore less frictional power loss, high working speed and higher field capacity.

2.2.1 Impact Cutters

Impact cutting principle is applied in two types of implements described as rotary cutters and flail shredders. The rotary cutters have knives rotating in a horizontal plane where as in flail shredders the knives rotate in vertical planes parallel to the direction of travel (Kepner et al., 1978). These were first developed for cutting up stalks, small bush, cover crops, weeds and other similar jobs. In the early 1950's, stalk cutters and shredders were adapted to chopping forage crops as a low-priced alternative for conventional, shear bar-type field choppers. In the rotary cutters, knives were attached to support arms through vertical hinge axes so that they could swing back if an obstruction was hit. However, these were hazardous because of the tendency to throw solid objects outward in a violent manner. Flail shredders had free swinging knives attached through a loop or chain link rather than through pivot axes to provide greater flexibility in rocky condition.

2.2.2 Rotary Grass Cutters

The mowers included in this category are of two types: drum mowers driven from the top, by a shaft and gears or by V-belts; and disc mowers which have cutting mechanism driven by spur gears in a slim casing below the knives (Culpin, 1978). The knife carriers of drum types tend to be of larger diameter than those of disc mowers.

2.2.3 Orchard Mowers

Rotary mowers with vertical spindles are widely used but the special requirements call for specialized types; capable of working beneath the trees in an offset position and taking up a cut of 3.6m wide (Culpin, 1978). In the past, machines with a swinging wing which could work right upto the tree trunks were used; but the guarding required by the current safety regulations interferes with efficient cutting by the wing, so growers tend to rely increasingly on use of herbicides for controlling growth near tree trunks.

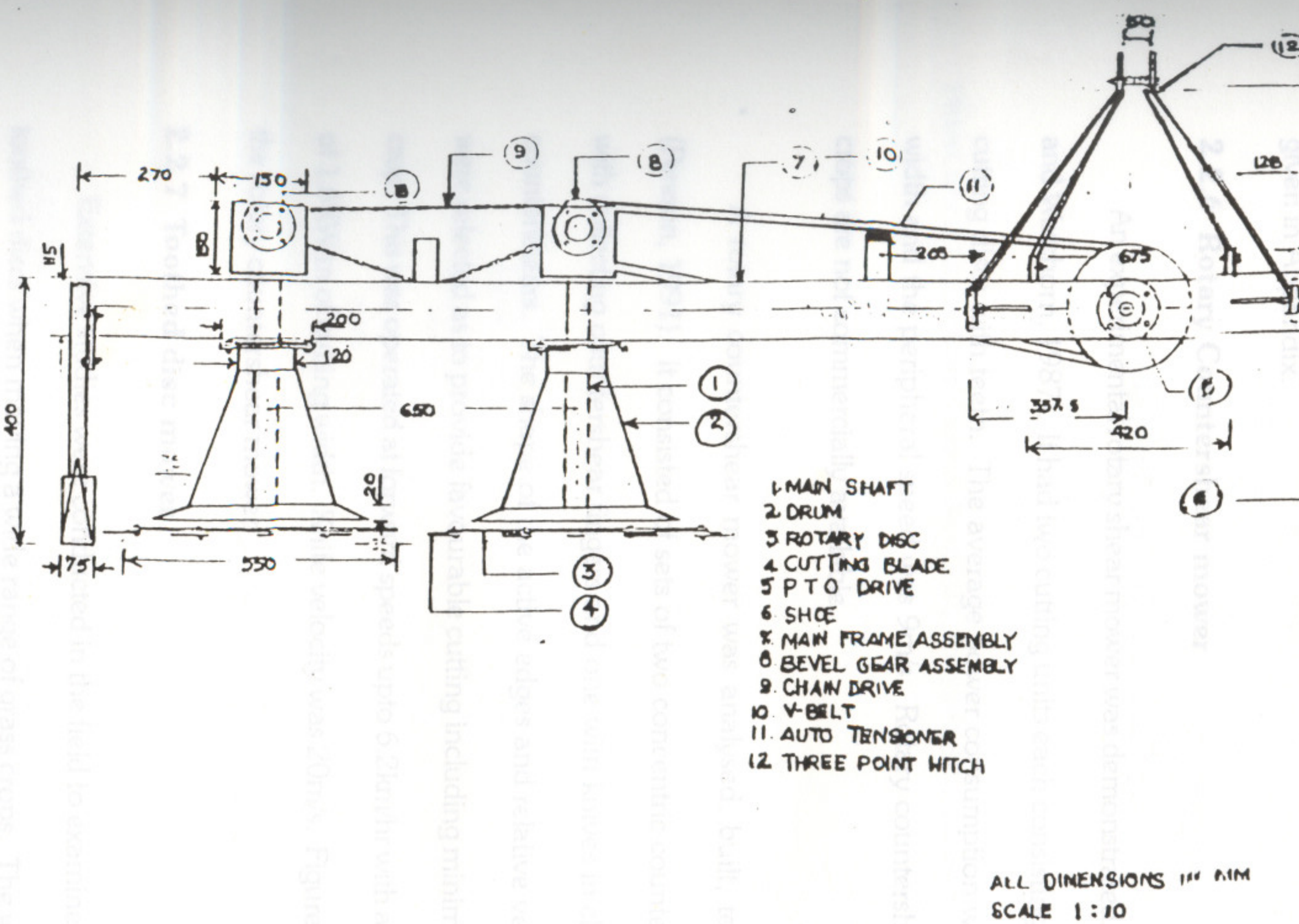
2.2.4 Flail Mowers

The term 'flail mower' is usually applied to mowers employing a horizontally arranged shaft with swinging cutting blades rotating at high speed (Culpin, 1978). Knives may be mounted on the cylinder in a variety of ways, the most common being a simple pivot mounting which allows the blades to fold inwards on meeting an obstruction or overload.

2.2.5 Rotary disc mower

The cutting principle of rotating disc mower is that knife like edge travelling at high speed, cuts through the grass on impact (Hawker and Keenlyside, 1971). In practice this is done by having a series of blades mounted on a disc which rotates at a high speed (3000 rpm) parallel to the ground. The number of blades vary from one machine make to another, generally two, three or four. The shape of the blades may be triangular, or rectangular but nearly in all cases they can be removed and replaced to expose another cutting edge.

A tractor operated rotary disc mower was designed, developed and tested for harvesting berseem, lucerne and oats (Chattopadhyay and Jai Singh, 1982). The effective field capacity was 0.35 ha/hr with 74.4 per cent field efficiency. The cost



ROTARY DISC MOWER

of harvesting of berseem by the machine including gathering were reduced to Rs. 160/ha which is about 50 percent with reference to cost of manual harvesting by sickle. The machines were found positive in level fields only. The front view of the rotary disc mower is shown in the figure. The specifications of the machine are given in Appendix.

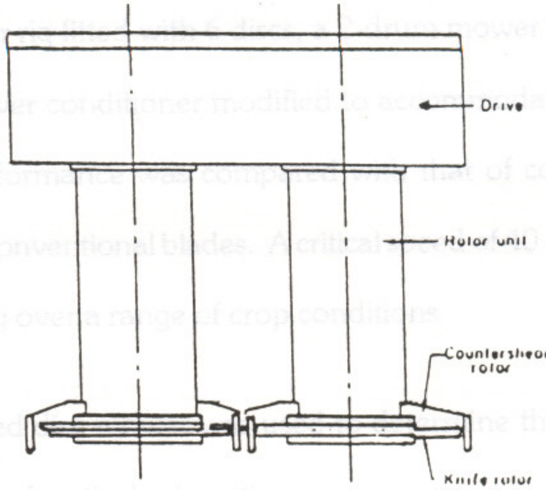
2.2.6 Rotary Countershear mower

An experimental rotary shear mower was demonstrated in England (Copland and Watchorn, 1987). It had two cutting units each consisting of 2 counter rotating cutting discs with teeth. The average power consumption was 3KW/m of cutting width and the peripheral speed was 9m/s. Rotary countershear mowers for field crops are not commercially available.

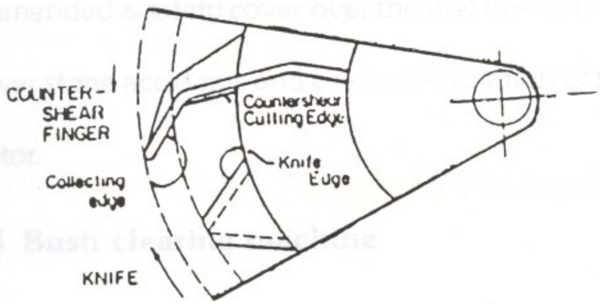
A rotary countershear mower was analysed, built, tested and evaluated (Persson, 1991). It consisted of sets of two concentric counter rotating discs, one with collecting countershear fingers and one with knives in close contact with the countershears. The shape of the active edges and relative velocities of the rotors were selected as to provide favourable cutting including minimum deflection of the crop. This was operated at forward speeds upto 6.2km/hr with a power consumption of 1.6KW/m of cutting width. Knife velocity was 20m/s. Figure shows the details of the rotary countershear mower.

2.2.7 Toothed disc mower

Extensive studies were conducted in the field to examine the performance of toothed discs when moving a wide range of grass crops. The work was performed



PRINCIPAL ELEMENTS OF A ROTARY COUNTERSHEAR MOWER



COUNTERSHEAR FINGER AND KNIFE, TOP VIEW

DETAILS OF ROTARY COUNTERSHEAR MOWER

using a modular rig fitted with 6 discs, a 2-drum mower fitted with toothed annuli and 5-disc mower conditioner modified to accommodate toothed discs (Baker et al, 1992). Performance was compared with that of commercial drum and disc mowers using conventional blades. A critical speed of 40 m/s is suggested to achieve effective cutting over a range of crop conditions.

A modified disc mower was used to determine the optimum rotational and forward speeds of vertical axis cutting components when operating in grass pastures (Fernandez et al, 1992). A hydraulic motor was fitted to allow the drive speed to be varied and a tensiometer to allow measurement of energy consumption; optimum rotational and forward speeds were 90-95 and 1.29-1.58 m/s respectively.

A field study of farmers using portable rotary mowers or orchard slopes investigated methods of accident prevention (Yonemura. J, 1991). The investigation recommended a safety cover over the disc blade, the use of circular saw type blades for fewer stone accidents and clock wise rotation of the rotor shaft for a right handed operator.

2.2.8 Bush clearing machine

A machine for the mechanical cutting of bushy plants like parthenium, which may be allergic to human body, was developed as an attachment to the prime mover of the self propelled paddy harvesting machine in the Zonal Research Centre of the College of Agricultural Engineering, Tamilnadu Agricultural University (Tajuddin et al, 1992).

The attachment consisted of one metre long shaft carrying 6mm size 250mm long mild steel chain pieces with circular MS flats as cutting elements at the tip. The power is transmitted from the 5.4 HP diesel engine of the prime mover to the shaft through V-belt and chain sprocket mechanism. When the shaft rotates at 1000 RPM, the plants are cut by impact. A dog clutch assembly is provided to cut off the shaft rotation irrespective of forward travel of the machine. The field capacity is 0.2 ha/hr and costs Rs.6,300/-. The operational cost of clearing one hectare of land by this machine works out to Rs.102/- as compared to Rs.313/- by conventional method of manual clearing.

2.2.9 Self propelled Bush Cutter

A self propelled bush cutter was developed and its performance was evaluated at the Kelappaji College of Agricultural Engineering and Technology, Tavanur (Venugopal et al, 1997). The vertical axis rotary cutter was propelled by a 5.18 HP diesel engine with a maximum speed of 1800 rpm. The speed of the vertical axis was measured to 480 rpm. It was observed that the bush cutter had an effective field capacity of 0.105ha/hr and with 58.33% field efficiency. The cost of clearing of the land was Rs.402.44 per hectare compared to a manual labour charge of Rs.640/ha. The details of the self propelled bush cutter is shown in figure.

MATERIALS AND METHODS

This chapter includes the general description of the existing bush cutting assembly developed during 1996-1997 in K.C.A.E.T Campus, the problems encountered in its operation, the modification details of the existing bush cutting assembly for improving its load taking capacity and efficiency and the methods involved in the performance evaluation of the modified bush cutting assembly.

3.1 Details of existing bush cutting assembly

The existing bush cutting assembly consists of the following important components:

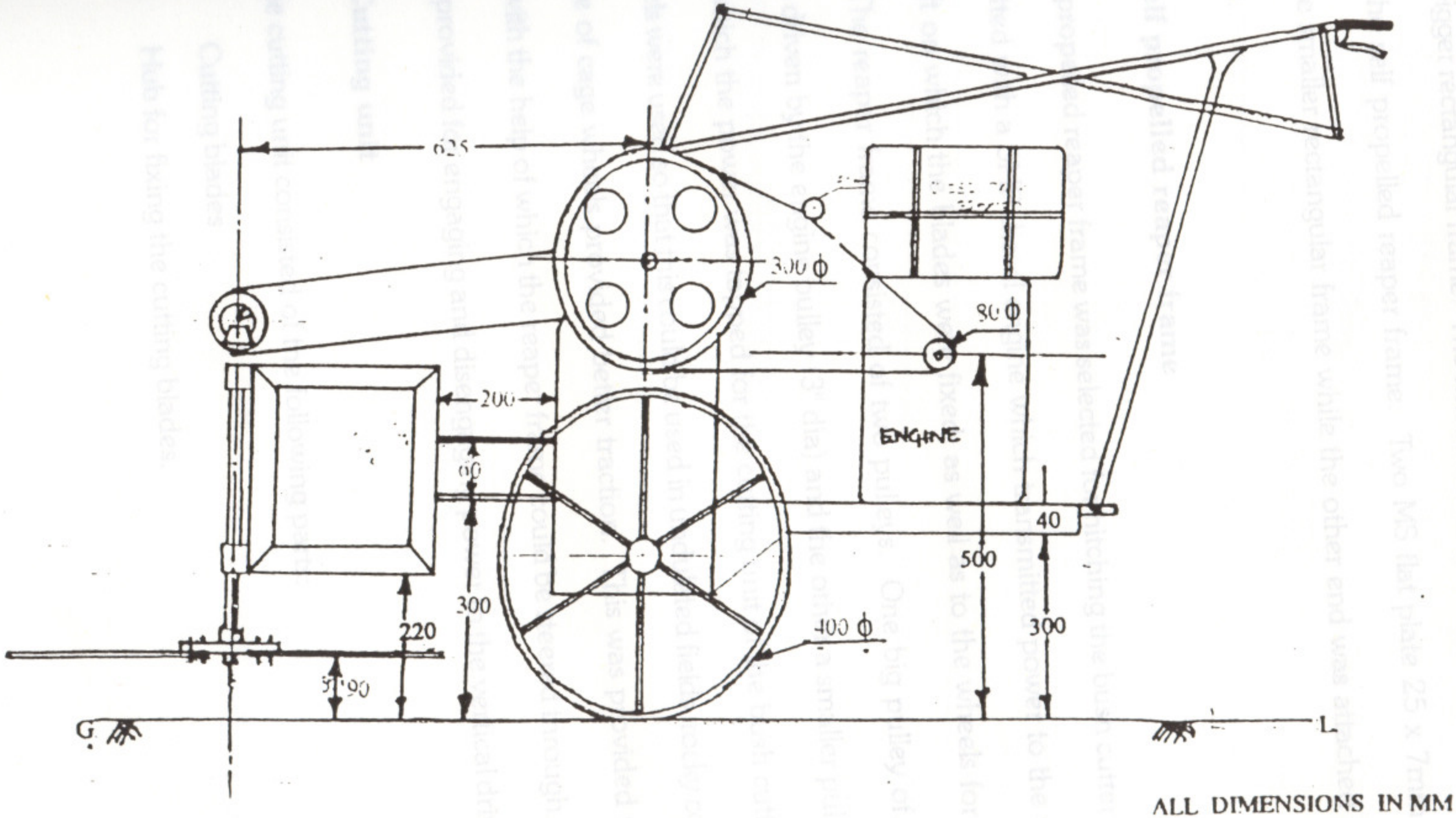
1. Main frame
2. Cutting unit
3. Power transmission unit
4. Prime mover

3.1.1 Frame

The frame consists of two parts namely bush cutter frame for mounting the cutting unit and self propelled reaper frame for mounting the engine.

3.1.1.1 Bush cutter frame

Figure 3.1 shows the details of the bush cutting unit. A rectangular frame of 350 x 280mm was made out of 4MS angles of 35 x 35 x 7mm welded them together. Four MS angles of 25 x 25 x 7mm were welded to each corner of this rectangular frame in such a manner that they projected out from the frame. To these projecting ends a bigger rectangular frame 700 x 280mm was welded. This frame was made



SIDE VIEW OF THE BUSH CUTTER

out of 35 x 35 x 7mm MS angles by welding them together. On this rectangular frame the vertical driven shaft was fixed. For fixing the horizontal driving shaft, a bracket (150mm long) was made out of 50 x 50 x 7mm MS angle iron and welded to top of the bigger rectangular frame. The smaller rectangular frame was hitched to the front of the self propelled reaper frame. Two MS flat plate 25 x 7mm were welded to the smaller rectangular frame while the other end was attached to the reaper frame.

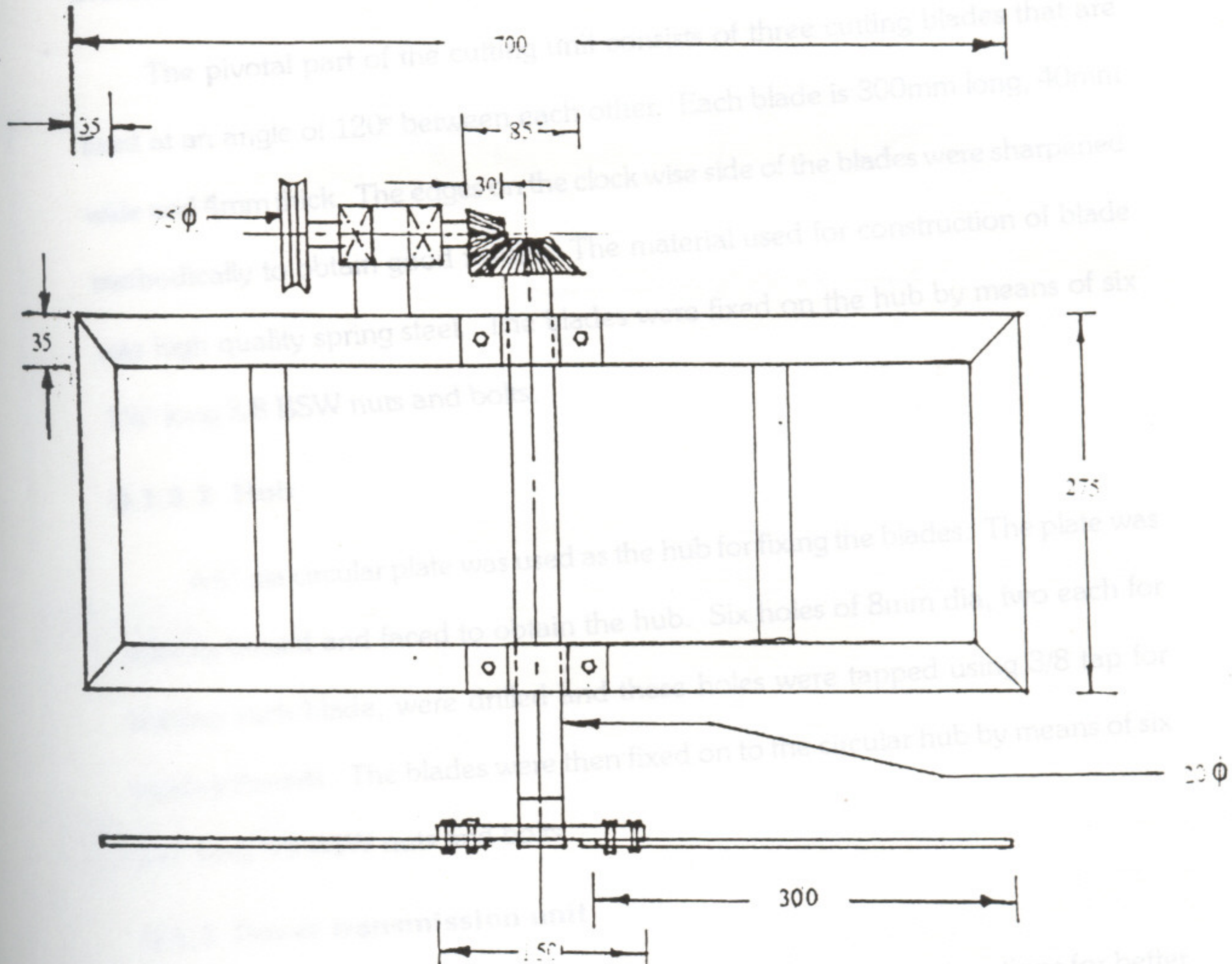
3.1.1.2 Self propelled reaper frame

A self propelled reaper frame was selected for hitching the bush cutter frame. This was flitted with a 5HP diesel engine which transmitted power to the vertical driven shaft on which the blades were fixed, as well as to the wheels for proper traction. The reaper frame consisted of two pulleys. One big pulley of 12" dia which was driven by the engine pulley (3" dia) and the other a smaller pulley of 5" dia from which the power was tapped for the cutting unit of the bush cutter. Two cage wheels were used so that this could be used in undulated fields, rocky conditions etc. Usage of cage wheels provided better traction. This was provided with side clutches with the help of which the reaper frame could be steered through. A clutch was also provided for engaging and disengaging power to the vertical driven shaft.

3.1.2 Cutting unit

The cutting unit consisted of the following parts:

1. Cutting blades
2. Hub for fixing the cutting blades.



ALL DIMENSIONS IN MM

SCALE: 1:5

FRONT VIEW OF THE BUSH CUTTER

3.1.2.1 Cutting blades

The pivotal part of the cutting unit consists of three cutting blades that are fixed at an angle of 120° between each other. Each blade is 300mm long, 40mm wide and 4mm thick. The edges on the clock wise side of the blades were sharpened methodically to obtain good results. The material used for construction of blade was high quality spring steel. The blades were fixed on the hub by means of six 1½" long 3/8 BSW nuts and bolts.

3.1.2.2 Hub

A 6" dia circular plate was used as the hub for fixing the blades. The plate was suitably turned and faced to obtain the hub. Six holes of 8mm dia, two each for holding each blade, were drilled and these holes were tapped using 3/8 tap for making threads. The blades were then fixed on to the circular hub by means of six 1½" long 3/8 BSW nuts and bolts.

3.1.3 Power transmission unit

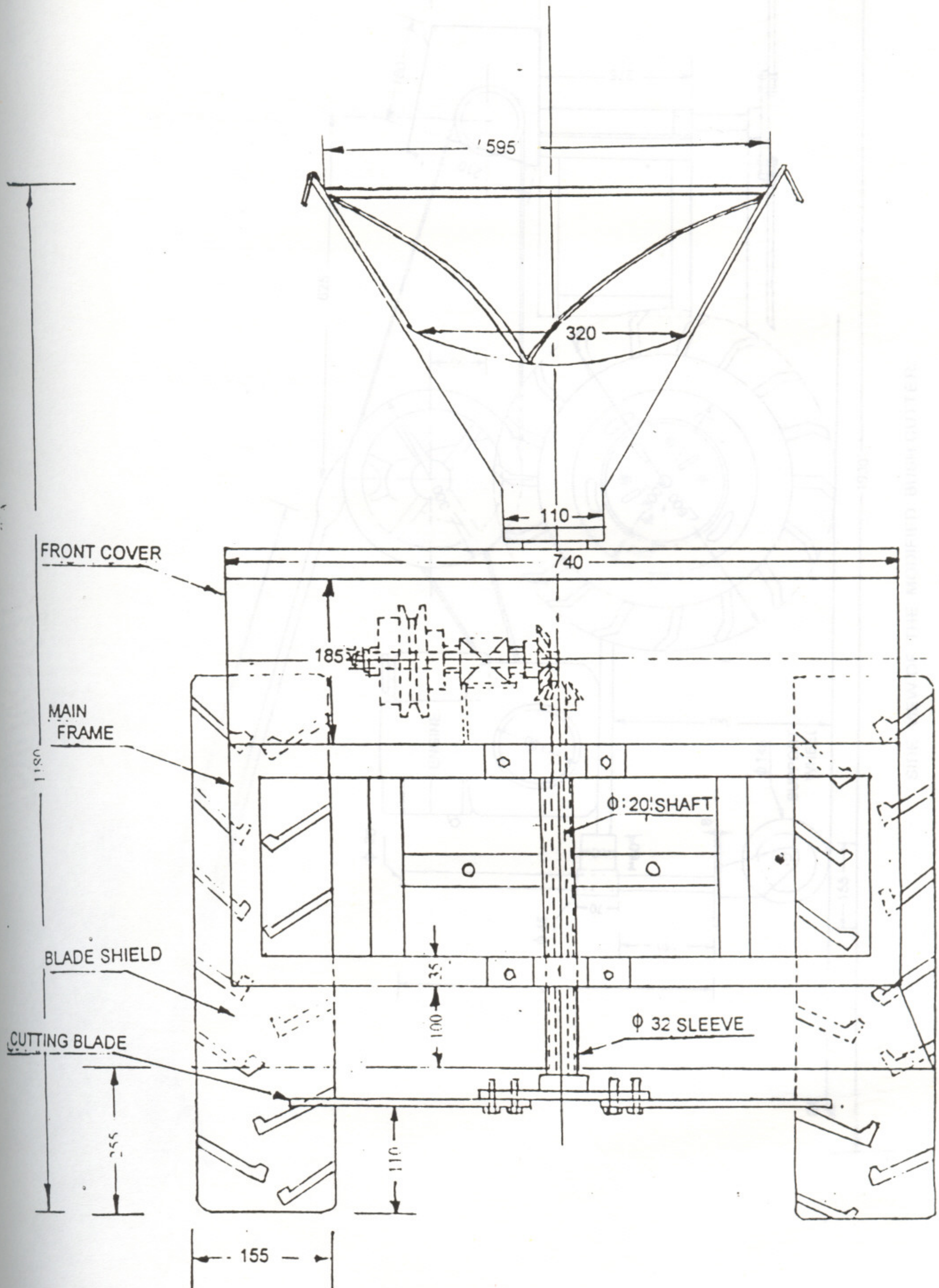
The power transmission unit can be divided into different sections for better assimilation.

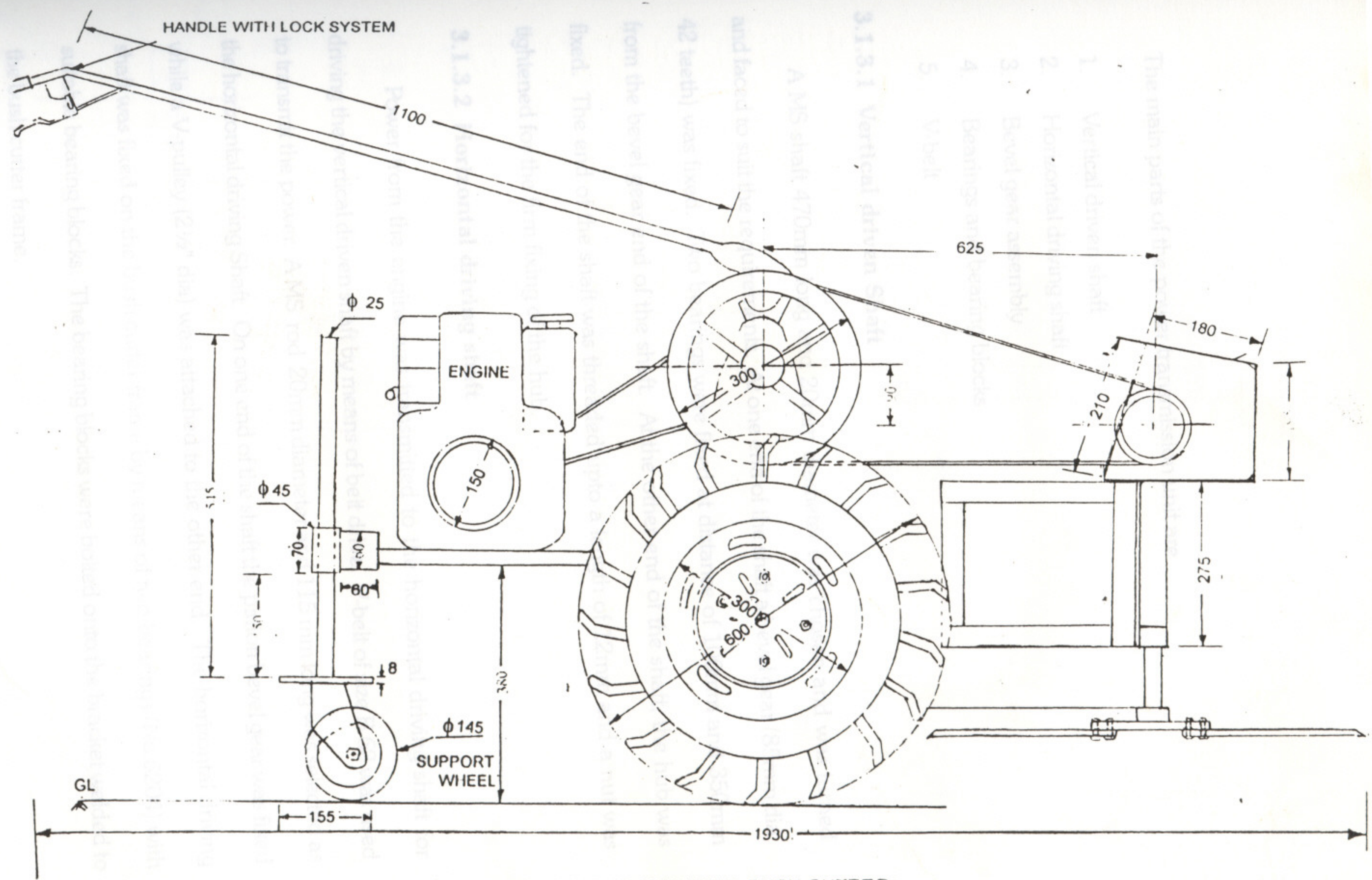
The differents stages of power transmission are

1. First stage power transmission - between engine and reaper horizontal shaft.

2. Second stage power transmission - between reaper horizontal shaft and horizontal driving shaft.

3. Third stage power transmission - between the individual gears of the bevel assembly.


595
320
110
FRONT COVER
740
185
MAIN FRAME
Φ 20 SHAFT
1180
35
BLADE SHIELD
100
Φ 32 SLEEVE
CUTTING BLADE
255
110
155



SIDE VIEW OF THE MODIFIED BUSH CUTTER.

The main parts of the power transmission unit are

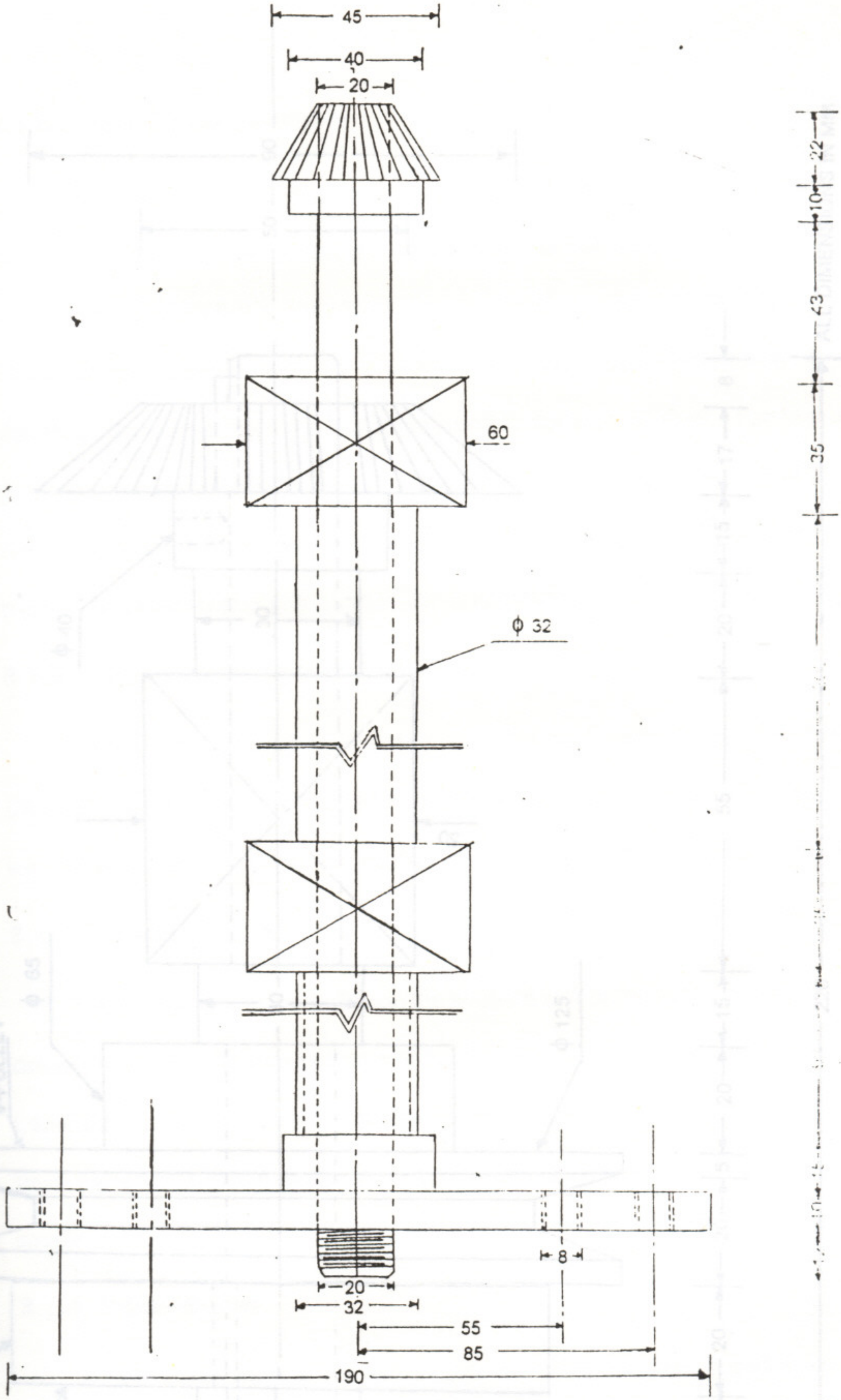
1. Vertical driven shaft
2. Horizontal driving shaft
3. Bevel gear assembly
4. Bearings and bearing blocks
5. V-belt

3.1.3.1 Vertical driven Shaft

A MS shaft 470mm long and 20mm diameter was chosen and was turned and faced to suit the requirements. At one end of the shaft a bevel gear (85mm dia, 42 teeth) was fixed. Two bearings were fixed at distance of 100mm and 350mm from the bevel gear end of the shaft. At the other end of the shaft, the hub was fixed. The end of the shaft was threaded upto a length of 12mm and a nut was tightened for the firm fixing of the hub.

3.1.3.2 Horizontal driving shaft

Power from the engine was transmitted to the horizontal driving shaft for driving the vertical driven shaft by means of belt drive. V-belt of size B 62 was used to transmit the power. A MS rod 20mm diameter and 115 mm long was selected as the horizontal driving Shaft. On one end of the shaft the pinion bevel gear was fixed while a V-pulley (2½" dia) was attached to the other end. The horizontal driving shaft was fixed on the bush cutter frame by means of two bearings (No.6204) with suitable bearing blocks. The bearing blocks were bolted onto the bracket welded to the bush cutter frame.



VERTICAL DRIVEN SHAFT

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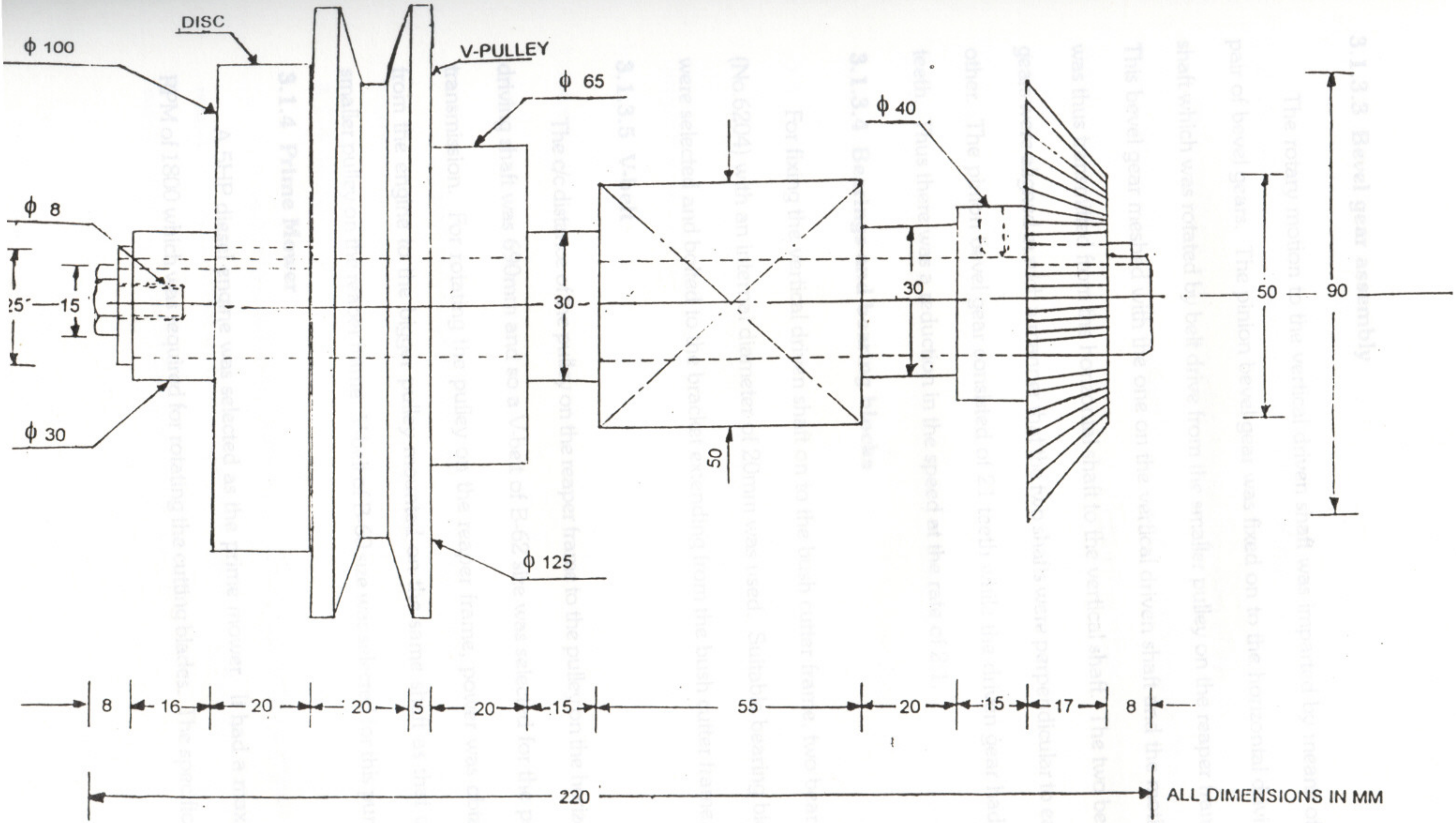


HORIZONTAL DRIVING SHAFT

3.1.3.3 Bevel gear assembly

The rotary motion to the vertical driven shaft was imparted by means of a pair of bevel gears. The pinion bevel gear was fixed on to the horizontal driving shaft which was rotated by belt drive from the smaller pulley on the reaper frame. This bevel gear meshed with the one on the vertical driven shaft and the motion was thus transmitted from the horizontal shaft to the vertical shaft. The two bevel gears were aligned in such a manner that the two shafts were perpendicular to each other. The pinion bevel gear consisted of 21 teeth while the driven gear had 42 teeth. Thus there was a reduction in the speed at the rate of 2:1.

3.1.3.4 Bearings and bearing blocks

For fixing the vertical driven shaft on to the bush cutter frame, two bearings (No.6204) with an internal diameter of 20mm was used. Suitable bearing blocks were selected and bolted to the bracket extending from the bush cutter frame.

3.1.3.5 V-belt

The c/c distance of the pulley on the reaper frame to the pulley on the horizontal driving shaft was 690mm and so a V-belt of B-62 size was selected for the power transmission. For rotating the pulley on the reaper frame, power was obtained from the engine to the bigger pulley mounted on the same shaft as that of the smaller pulley on the reaper frame. V-belt of B-60 size was selected for this purpose.

3.1.4 Prime Mower

A 5HP diesel engine was selected as the prime mower. It had a maximum RPM of 1800 which was required for rotating the cutting blades. The specifications of the engine are given in Appendix.

3.2 Drawbacks of the existing bush cutting assembly

Many problems were encountered in the operation of the developed bush cutter which called for several modifications to improve its performance. The various drawbacks were first identified by test drive and then their remedies are suggested.

3.2.1 Cutting speed

The cutting speed available at the blades was insufficient to cause impact cutting. This was due to the speed reductions at the first, second and third stages of power transmission.

3.2.2 Idler clutch

The idler system used in the first stage power transmission did not function properly leading to the forward movement of the unit immediately after starting. This required two operators, one to hold on the side clutches and the other to start the engine.

3.2.3 Cutting blade rotation

A separate clutch system was absent to prevent the rotation of the cutting blades while transporting. This is not advisable from the safety point of view.

3.2.4 Weight Balance

Since the cutting unit of the bush cutter had very little weight compared to the reaper, this replacement led to a condition in which a major portion of the engine weight had to be borne by the operator himself.

3.2.5 Clogging of Bevel gears

The existing unit had no protective cover for the bevel gears which caused the clogging of the cut materials between the gears leading to their damage and reduced life.

3.2.6 Blade shield

The developed bush cutter lacked a shield for the cutting blades which questioned the safety of the operator.

3.2.7 Types of Blades

The testing of the bush cutter was done with only one set of blades. Studies were not conducted for different types of blades for various terrain conditions.

3.2.8 Vibration of mainframe

Excessive vibration of the mainframe was encountered in its operation. The mainframe was not properly secured to the reaper frame.

3.2.9 Unprotected Vertical shaft

The vertical shaft was not protected by any method leading to the entangling of weeds over it and gradual build up of load in operation.

3.3 Modification of the bush cutter

The different problems in the operation of the existing unit were dealt one by one and the solutions are suggested here.

3.3.1 Cutting speed

The 12" diameter V-pulley in the first stage power transmission was replaced by a 10" dia V-pulley. The 5" dia and 2½" dia V-pulleys of the second stage power transmission were replaced by a 12" dia and a 5" dia V-pulley respectively. Also the bevel gear assembly was reversed in position. This resulted in a maximum speed of 2880 rpm at cutting blades with 1800 rpm derived from the engine.

3.3.2 Locking system for side clutches

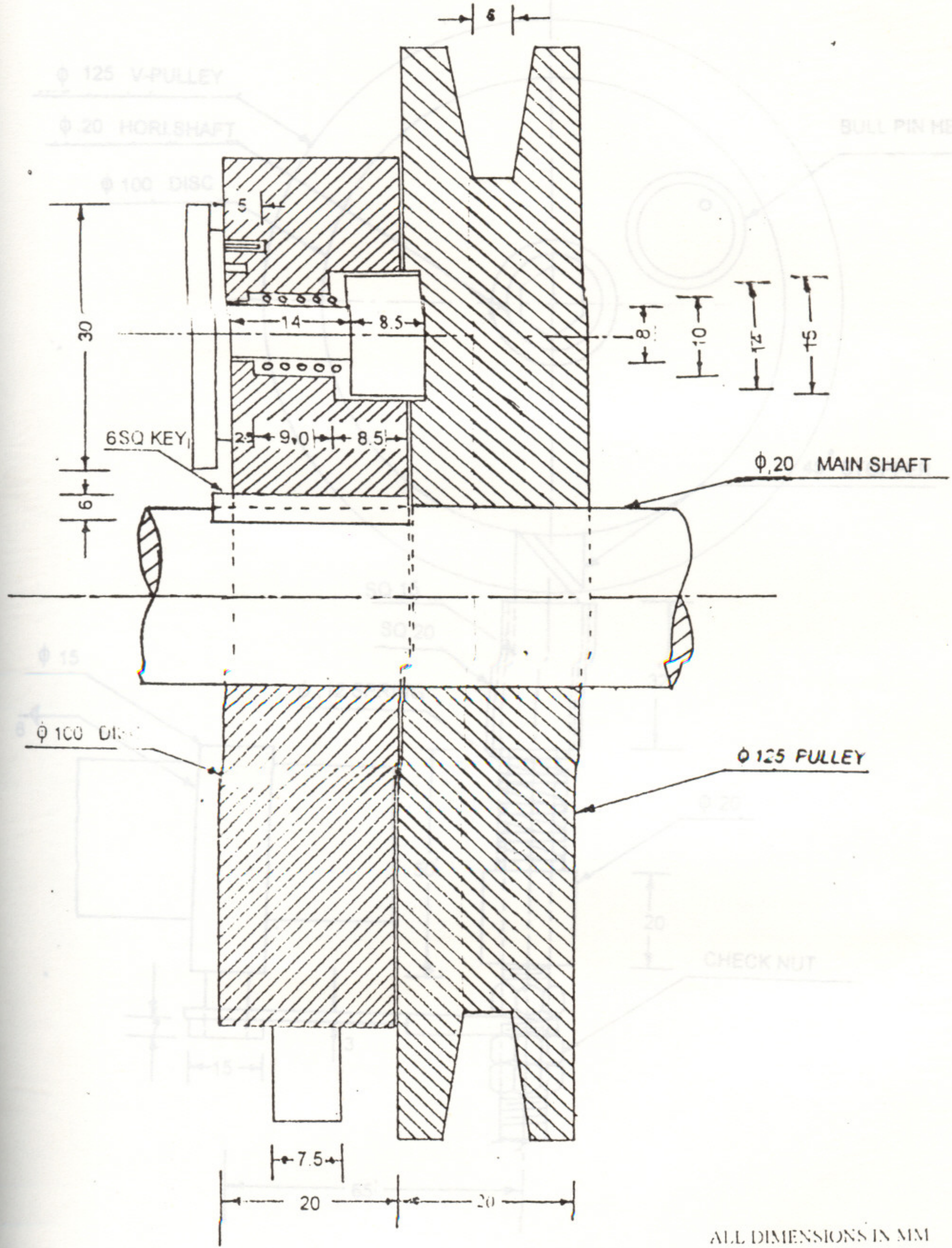
The idler system was removed and a locking system was introduced in the clutch mechanism. This facilitated the starting of the machinery after locking the hand clutches thus solving the problem of idler system failure.

3.3.3 Bull-pin system

To prevent the rotation of the cutting blade while transport, a bull pin mechanism was introduced in the horizontal shaft. A *M.S plate was turned to* 100mm dia and faced to 20mm thickness. The bull pin was machined to the required dimensions and the details of the assembly are shown in the figure. A separate stopper mechanism was also made to ensure fool-proof working of the bull pin system. The details of the stopper mechanism is shown in the figure.

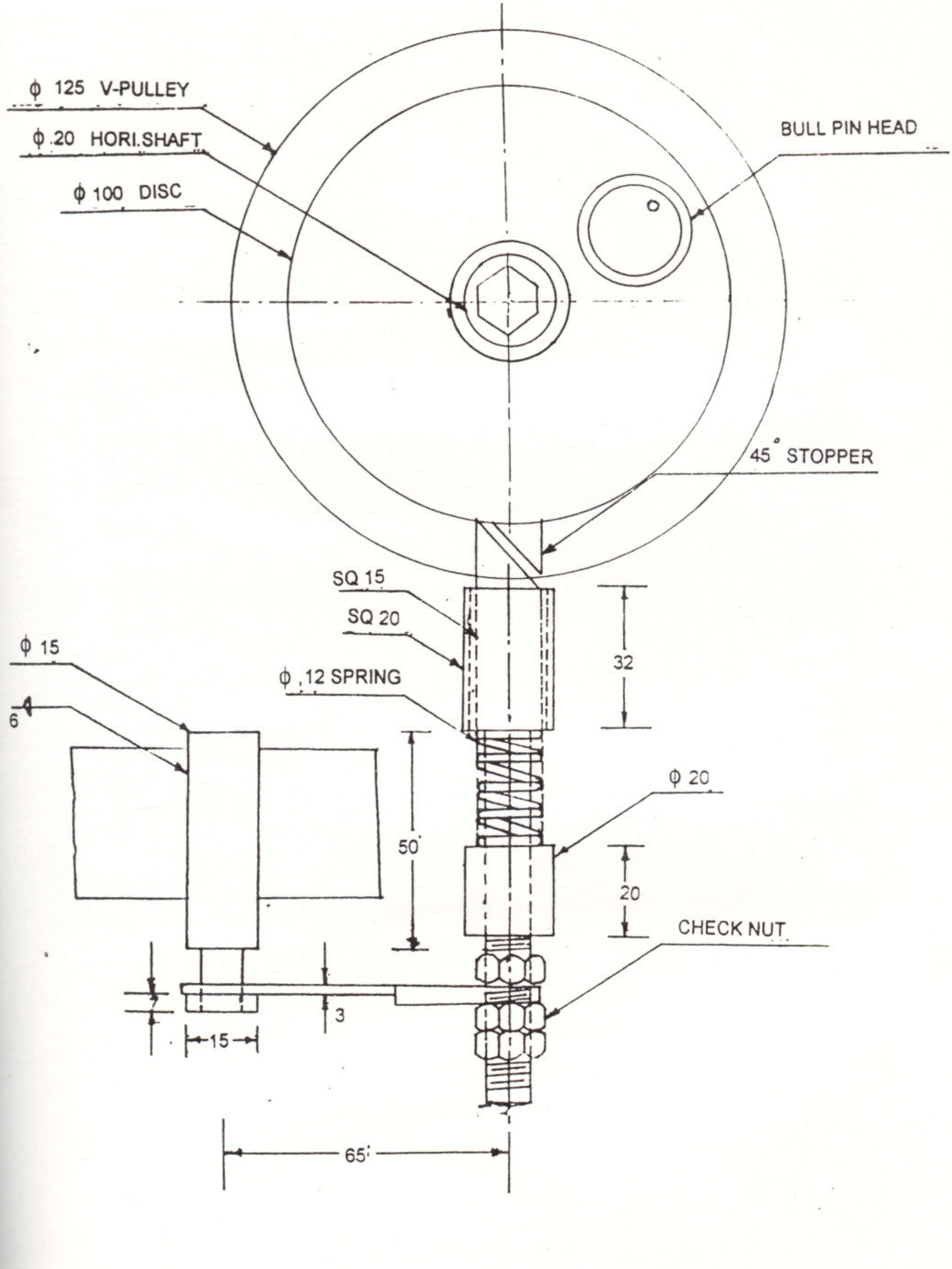
3.3.4 Support Wheel

A 5 inch diameter support wheel was introduced in the bush cutting unit. This provided for supporting the weight of the engine partly, so also short trips of the bush cutter even without the operator became possible. The solid vertical stem



ALL DIMENSIONS IN MM

SECTIONAL VIEW OF BULL PIN SYSTEM



STOPPER MECHANISM

of the supporting wheel facilitated positioning at different heights to adjust the height of cutting.

3.3.5 Front cover

A 14 gauge MS sheet of size 70x45cm was used as the protective cover, the details of which are seen in the front and side views of the bush cutter. M.S flats were spot-welded to the sheet to give additional strength to the cover. The cover can be easily removed when need arises, since the cover was bolted to the mainframe. The clogging of bevel gears was thus prevented.

3.3.6 Blade shield

A semicircular sheet of 35cm radius was placed beneath the frame. The cover was provided with 3/8 BSW nuts and bolts and the check nut aided proper locking. The details of the blade shield are visible in the photographs.

3.3.7 Types of blades

In addition to the rigid blades, a 30cm dia serrated disc and a set of 3 chains were also tested in the field. The photographs of the different cutting systems reveal the details.

3.3.8 Mainframe vibration

The mainframe was securely bolted to the reaper frame by providing two additional legs to the mainframe. This prevented the excessive vibration of the mainframe.

3.3.9 Sleeve for vertical shaft

The vertical shaft was provided with a 32cm dia pipe sleeve to prevent the entangling of weeds while operating. The rotation of the sleeve was prevented by providing two lock legs inserted at the side of the bearings.

3.4 Performance evaluation

Measures of agricultural machine performance are the rate and quality with which the agricultural operations are accomplished. Farmers usually are aware of the need for complete and speedy operation but they often ignore the economic penalties resulting from poor quality operations. It is therefore necessary to consider the quality as well as the quantity while computing machine performance.

The various parameters analysed were:

3.4.1 Field capacity

The field capacity gives an exact idea of the rate at which the operation is finished. Knowing the width of cut and operating speed the theoretical field capacity could be calculated using the formulae:

$$\text{T.F.C.} = \frac{W \times S}{10}$$

Where W = Width of cut in metres

S = Operating speed in km/hr

T.F.C. = Theoretical field capacity in ha/hr.

The theoretical field capacities using the three sets of blades were found out.

In order to find out the effective field capacity plots of (10 x 7) infested by bushes and weeds were selected in the college campus. The bush cutter was operated at a speed of 2.5 km/hr and the time taken to clear of the bushes in the selected plots were noted. From this the effective field capacity for the three sets of blades were found out.

3.4.2 Field efficiency

After calculating the actual and theoretical field capacities, the field efficiency was worked out using the relation

$$\text{Field efficiency (\%)} = \frac{\text{Effective field capacity}}{\text{Theoretical field capacity}} \times 100$$

3.4.3 Cutting efficiency

Cutting efficiency refers to the amount of weed cut by the cutter.

The cutting efficiency was found out as follows. Five sets of plots, each of size 1mx1m infested with bushes were selected for one type of blade. The number of weeds in each plot before operation were counted. Then the machine was operated in each plot at different speeds. The number of weeds left in the field after operation was noted. The cutting efficiency was found out using the relation,

$$\text{Cutting efficiency} = \frac{B-A}{B} \times 100$$

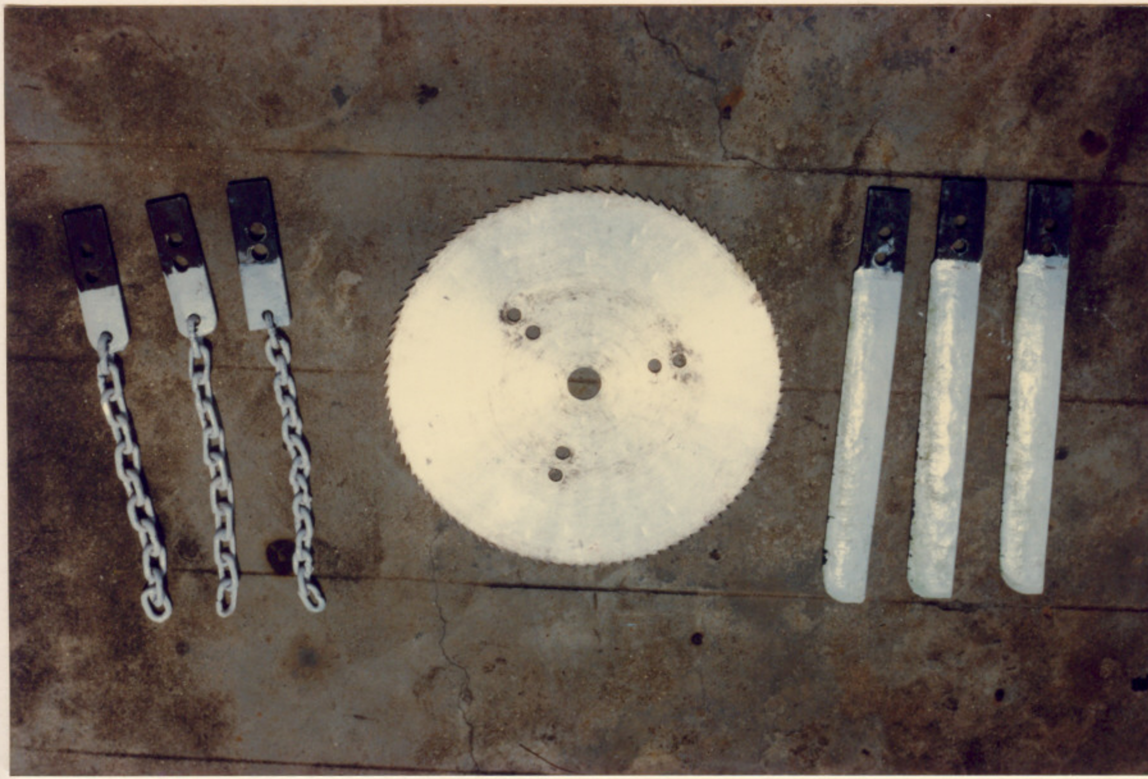
Where B = No. of weeds present *before operation*

A = Number of weeds left in the field after operation

The same procedure was repeated for the remaining two sets of blades and cutting efficiency was thus noted.

3.4.4 Cost of operation

The economic feasibility of any farm machinery is as important as any other aspect related to it. The total cost for manual bush clearing is calculated at the rate of Rs 120/- per person for a hectre of land. The total cost for bush clearing a hectre of land with the bush cutter is calculated and the results are compared.



RESULTS AND DISCUSSION

The results of the modifications brought into effect on the existing bushcutter are discussed in this chapter. The performance of the modified bushcutter was analysed for the parameters like field capacity, field efficiency, cutting efficiency and cost of operation. The details of the results are mentioned below.

4.1 Cutting devices

Three types of cutting devices are used for testing. This included the rigid blade, the chains and the circular saw. The results showed that the rigid blade was most efficient for grass cutting; the chains were especially useful in stony and undulating conditions and the circular saw blade was effective for stem dia upto 2.5cm. However the circular saw type didn't yield good results for grass cutting.

4.2 Locking system for side clutches

The locking system introduced at the hand lever for the side clutches facilitated the stopping the forward motion of the unit with ease; also circular trips were possible with locking of one of the side clutches.

4.3 Rotation of cutting blades at transport

The bull pin system introduced in the second stage power transmission prevented the blade rotation at transport. At operation, the bull pin can be engaged, but only after arresting the drive. The stopper mechanism aided in the efficient working of the bull pin system.

On field testing, it was observed that the blade shield protected the operator from the stones and stalks thrown by the cutter and the front cover was successful in preventing the stalks interfering with the gear meshing. The pipe sleeve provided on the main shaft ensured trouble free rotation of the main shaft without the entangling of weeds around it.

4.5 Support wheel

The adjustable height support wheel carried a major portion of the unbalanced load and aided easy manouvering of the unit. The provision of both the cage wheels and pneumatic tyres facilitated the use on various terrain conditions.

4.6 Main frame vibration

The main frame vibration at operation was greatly reduced by providing additional supports to the frame. All the bolted joints subject to vibration were provided with lock nuts for checking the vibration.

4.7 Performance Evaluation

The performance of the bush cutter was evaluated by its field capacity, field efficiency, cutting efficiency and cost of operation.

4.7.1 Field Capacity and Field Efficiency

The plots of suitable sizes were selected and then the unit was operated with the three types of cutting devices. The actual field capacities, theoretical field capacities and field efficiencies were found out. All the types of blades were tested at a forward speed of 2.5 km per hour. The bar representation of the field capacity and the field efficiency of the bush cutter using the different types of cutting devices are shown in figure.

Table.1 Field Capacities and Field Efficiencies of the bush cutter with different cutting devices

Cutting device	Area of plot (m^2)	Total time taken for cutting (s)	Actual field capacity (ha/hr)	Width of cut (m)	Theoretical field capacity (ha/hr)	Field efficiency (%)
Rigid Blade	70	196	0.1285	0.7	0.175	73.43
Chain	70	224	0.1125	0.7	0.175	64.28
Circular saw	42	354	0.0427	0.3	0.075	56.95

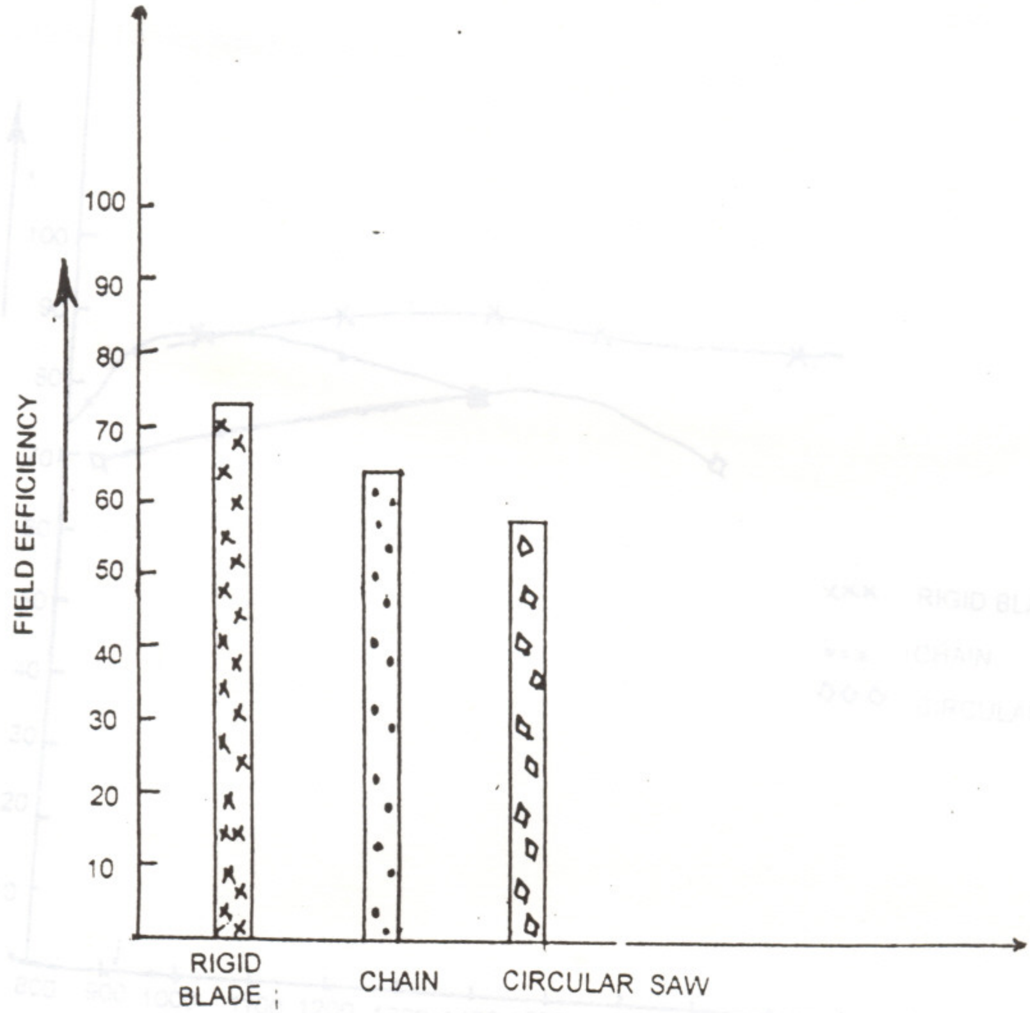
4.7.2 Cutting Efficiency

The cutting efficiency of the unit employing the different types of cutting devices was calculated and the results are shown in the table. The graphical representation of the cutting efficiencies at varying cutting speeds for different cutting devices are shown in the figure

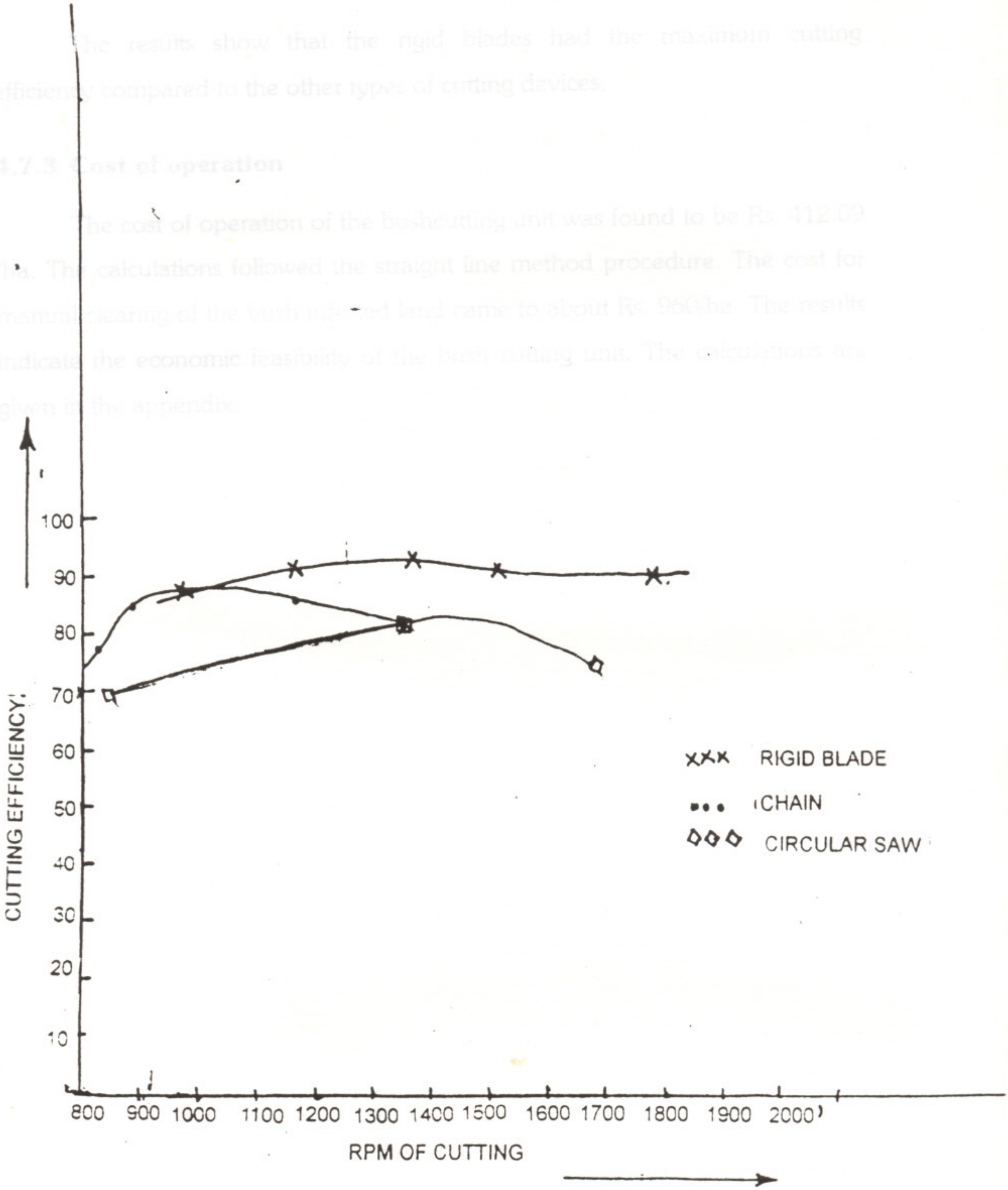
Table 2. Cutting efficiencies of the bushcutter with different cutting devices.

Cutting Device	Sl. No.	Cutting speed (RPM)	No. Of Weeds before operation	No. Of Weeds after operation	Cutting Efficiency
Rigid Blade	1	975	62	7	88.70
	2*	*1165*	170	14	*91.76*
	3	1370	320	20	93.75
	4	1510	114	9	92.10
	5*	1785	410	32	92.11
Chain	1	885	410	60	85.36
	2*	825	184	40	78.26
	3	1165	380	52	86.31
	4	1350	268	45	83.21
	5*	1510	86	34	60.46
Circular saw	1	845	87	26	70.11
	2*	1085	155	106	31.61
	3	1340	56	10	82.14
	4	1675	64	15	76.56
	5*	1850	210	130	38.09

* The test was conducted for grasses



FIELD EFFICIENCY OF DIFFERENT CUTTING DEVICES



CUTTING EFFICIENCY V/S RPM OF CUTTINGFOR

DIFFERENT CUTTING DEVICES

The results show that the rigid blades had the maximum cutting efficiency compared to the other types of cutting devices.

4.7.3 Cost of operation

The cost of operation of the bushcutting unit was found to be Rs. 412.09 /ha. The calculations followed the straight line method procedure. The cost for manual clearing of the bush infested land came to about Rs. 960/ha. The results indicate the economic feasibility of the bush cutting unit. The calculations are given in the appendix.

SUMMARY AND CONCLUSION

The mechanical methods of bush clearing as already been adopted in many parts of the world. The main reason behind the adoption of the mechanical methods is the alarming rise of labour charges. This problem is particularly severe in our state.

The bush cutter developed in the K.C.A.E.T, Tavanur, though proved versatile, had many operating difficulties and drawbacks. The effort of this work has been to modify the bush cutter to bring it under safe working standards, to check the design of the various components and also to evaluate the performance parameters. Considering the design aspects, the forces acting on the main shaft were studied and components of the unit were within design considerations. The cutting speed of the blades were enhanced by replacing a set of pulleys at the different stages of power transmission. The safety aspects were kept inmind while fabricating the blade shield; also the cover and pipe speed on the main shaft ensured trouble free operation of the bush cutter unit. The bull pin system for preventing the rotation of the blades at transport proved successful in operation. The introduction of the locking system for side clutches and the provision of an 'adjustable height' support wheel enhanced the ease of working with the bushcutter. The cutter was tested with three different types of cutting devices.

The performance parameters studied were the field capacity, field efficiency, cutting efficiency and cost of operation. The field capacities were 0.1285 ha/hr, 0.1125 ha/hr and 0.0488 ha/hr for the rigid blades, chains and circular saw blade respectively. The respective field efficiencies were 73.43%, 64.28% and 56.95%. The maximum cutting efficiency was observed for rigid lades. The cost of operation was found to be Rs. 412.09/ha compared to manual charges of Rs. 960/ha.

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APPENDIX - I

Specifications of the tractor operated rotary disc mower

Type of prime mover	:	Tractor
Recommended size of prime mover	:	35 HP
Type of drive	:	PTO
Power transmission system	:	V-belt and chain sprocket drive
No. of discs	:	2
Dia of disc (mm)	:	555
No. of blades in each disc	:	6
Length of each blade (mm)	:	75
Bevel angle of the blade, dg	:	25
Maximum RPM of disc	:	1100
Maximum cutting speed of blade (m/s)	:	40
Eff. cutting width (mm)	:	1300
Cutting height (mm)	:	50 - 150
Overall dimensions (mm)		
Length	:	2600
Width	:	640
Height	:	630
Weight	:	300

APPENDIX - II

Specification of the Engine

Make	:	Greaves
Model	:	1523, Diesel engine
Engine number	:	D.C 02751
Fuel	:	Diesel
Output	:	5.18 H.P
Speed	:	1800 RPM

APPENDIX - III

Cost Analysis

Cost of labour for working 8 hrs per day - Rs.120/-

Cost of manual operation - Rs.15.00/hr

An average of 8 man days is required for bush clearing operation in 1 hectare.

thus cost of the bush clearing operation manually = 15 x 64= 960 Rs/ha

Cost Analysis for the developed bush cutter

- Self propelled repair frame

Initial cost (C)	-	Rs. 30,000
Fuel consumption	-	1.00 lt/hr
Oil cost	-	1/3 rd of fuel cost.
Life period (L)	-	10 Yrs.
Operating hrs/annum (H)	-	500 hrs
Salvage value (S)	-	10% of initial cost

Fixed cost

Depreciation - $\frac{C-S}{L} x \frac{1}{H}$

$$= \frac{30000-3000}{10} x \frac{1}{500} = 5.4 / hr$$

Interest (10%)per annum $= \frac{C+S}{2} x \frac{I}{100} x \frac{1}{H}$

$$= \frac{30000+3000}{2} x \frac{10}{100} x \frac{1}{500} = 3.3 / hr$$

Insurance = 1% of initial cost = 300/500 = Rs. 0.60/ hr

Taxes = 1% of initial cost = 300/500 = Rs. 0.6/ hr

Housing = 1% of initial cost = 300/500= Rs.0.6/ hr

Total fixed cost = Rs.10.5/ hr

Variable Cost

Fuel cost @ Rs. 10/l = Rs.10/l x 1l/hr
= Rs. 10\hr

Oil cost = 1/3x10 = Rs. 3.33/hr

Repairs & maintenance = 10% of initial cost

$$= \frac{10}{100} x \frac{30000}{500} = Rs.6 / hr$$

Operators wages = 120/8 = Rs.15.00/hr

Total \variable cost = Rs.34.33/hr

Total cost = 10.5+34.33 = Rs.44.83/hr

Bushcutter assembly

Initial cost (c) = Rs. 3000
Life period = 10 yrs
Salvage value (s) = 10% of initial cost
Operating hrs per annum = 500

Fixed cost

Depreciation $= \frac{C-S}{L} x \frac{I}{H}$

$$= \frac{3000-300}{10} x \frac{1}{500} = Rs.0.54 / hr$$

Interst (10%)per annum $= \frac{C+S}{2} x \frac{I}{100} x \frac{1}{h}$

$$= \frac{(3000+300)}{2} x \frac{10}{100} x \frac{1}{500}$$

= Rs. 0.33/hr

Housing (1%) of initial cost = $\frac{1}{100} x \frac{3000}{500}$

= Rs. 0.06/hr

Taxes = Nil

Insurance = Nil

Total fixed cost = Rs. 0.93/hr

Variable cost

Repairs & Maintenance = 10% of initial cost

= $\frac{1}{100} x \frac{3000}{500} = Rs.0.6 / hr$

= F.C+V.C

Total cost = 0.93+0.6

= Rs. 1.53/hr

Total cost of operating the bush cutter = 44.83+1.53 = Rs. 46.36/hr

Field capacity = 0.1125 ha/hr

Cost of operating in a land of 1 ha area = $46.36x \frac{1}{0.1125} = Rs.412.09 / ha$

APPENDIX-IV

I Force and Moment Distribution on intermediate shaft

Load at the point A = $\dfrac{5.4 x 4500 x 1000}{2 x \pi x 427 x \dfrac{225}{2}} = 80.50 kgf$

A) Vertical force diagram

Vertical upward load at A due to belt drive and driven pulley

F_{VA} = $80.5 \text{ x } \sin 15^0$

= 20.83kgf

Vertical upward load at B due to belt drive

F_{VB} = 60.38 x sin10

= 10.48kgf

vertical downward load at M due to sprocket drive

$= \dfrac{5.4 x 4500 x 1000}{2 \pi x 427 x \dfrac{50}{2}} = 362.3 kgf$

Taking moment about point P to get the force at points P&Q

$20.83\text{x}70+362.3\text{x}75-F_Q\text{x}150-10.48\text{x}220=0$

F_{VQ} = 175.5kgf

F_{VP} = 155.49kgf

B) Vertical Bending Moment (M_V) Diagram

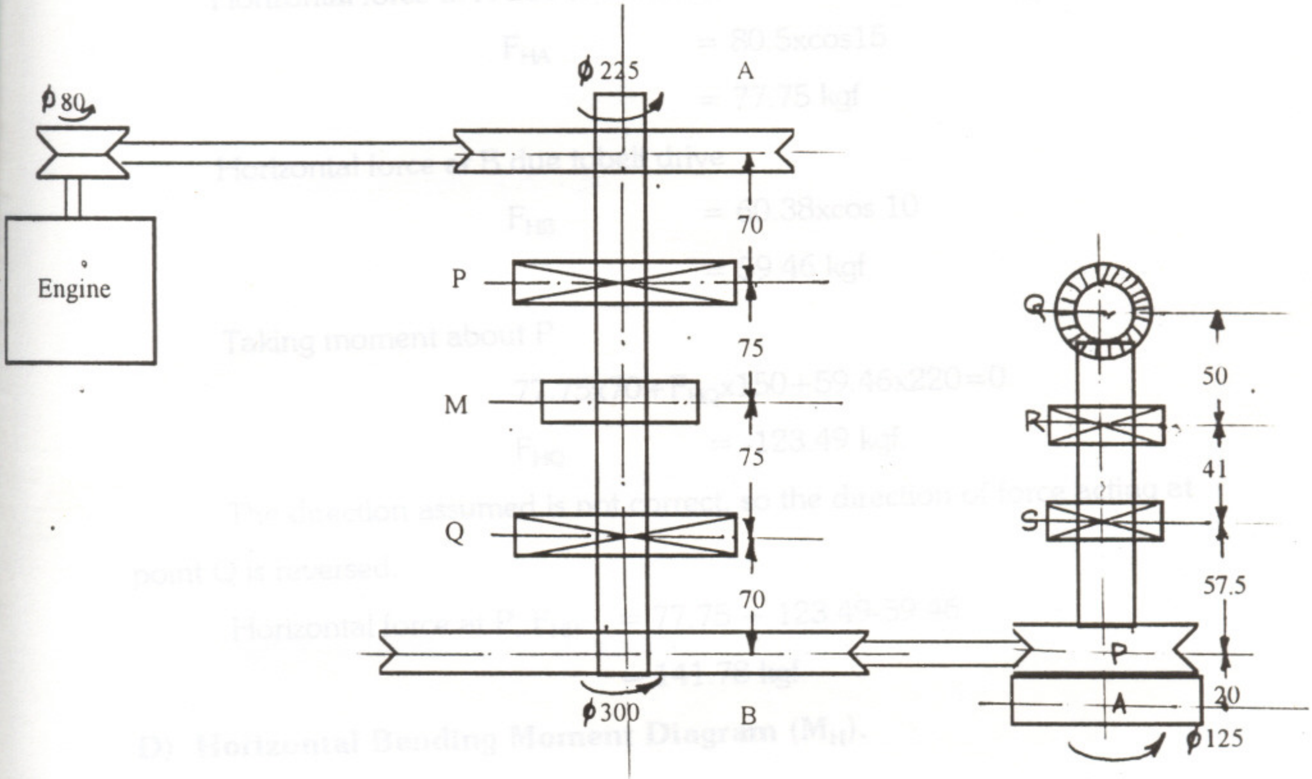
Moment at P =20.83 x 70 = 1458.1 kgf mm

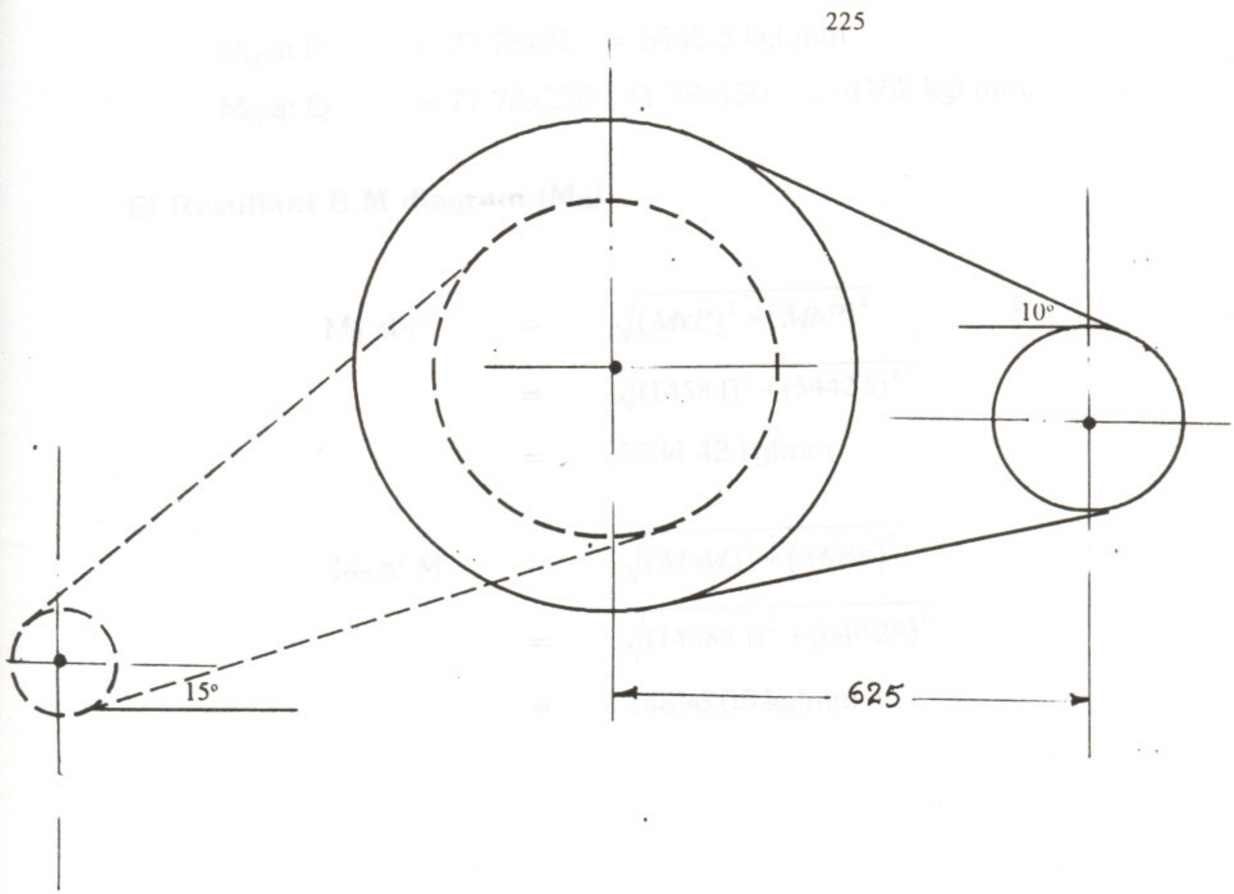
Momemt at M = 20.83 (70+75) +155.49x75

= 14682.1kgf mm.

Moment at Q = -10.48x70

= -733.6 kgfmm.


ϕ80
Engine
ϕ225
A
70
P
75
M
75
Q
70
ϕ300
B
Q
50
R
41
S
57.5
P
A
20
ϕ125


225
10°
15°
625

C) Horizontal force diagram (F_H)

Horizontal force at A due to belt drive

F_{HA} = 80.5xcos15

= 77.75 kgf

Horizontal force at B due tobelt drive

F_{HB} = 60.38xcos 10

= 59.46 kgf

Taking moment about P

$$77.75 \times 70 + F_{HQ} \times 150 + 59.46 \times 220 = 0$$

F_{HQ} = -123.49 kgf.

The direction assumed is not correct, so the direction of force acting at point Q is reversed.

Horizontal force at P, F_{HP} = 77.75 + 123.49-59.46

= 141.78 kgf.

D) Horizontal Bending Moment Diagram (M_H).

M_H at P = 77.75x70 = 5442.5 kgf mm

M_H at Q = 77.75x220-141.78x150 = -4162 kgf mm.

E) Resultant B.M diagram (M_R)

$$M_R \text{ at P} = \sqrt{(MvP)^2 + (MhP)^2}$$

$$= \sqrt{(1458.1)^2 + (5442.5)^2}$$

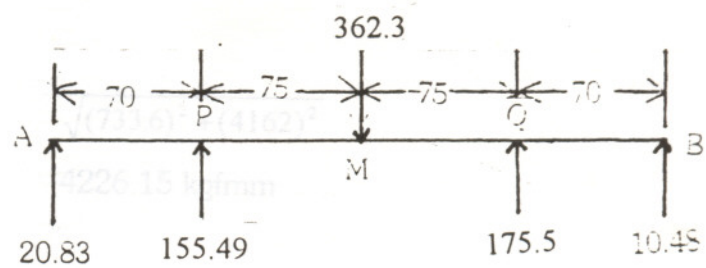
= 5634.43 kgfmm.

$$M_R \text{ at M} = \sqrt{(MvM)^2 + (MhM)^2}$$

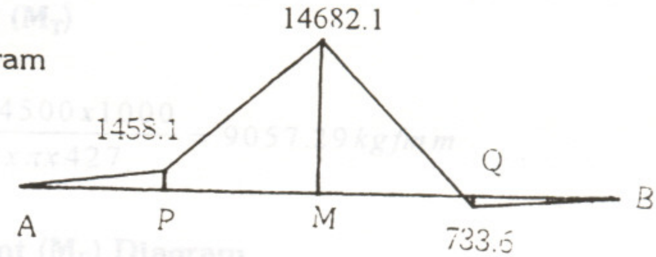
$$= \sqrt{(14682.1)^2 + (640.29)^2}$$

= 14696.05 kgfmm

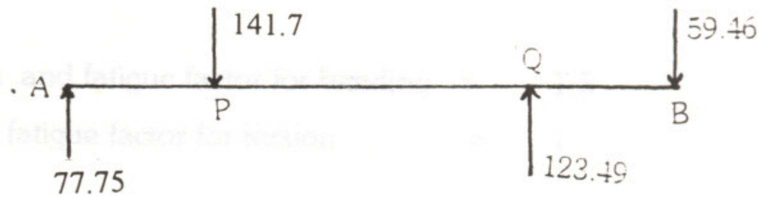
Vertical force diagram



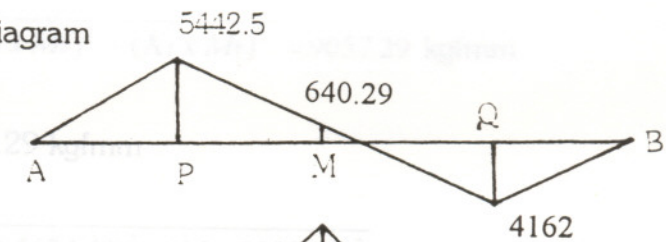
Vertical bending moment diagram



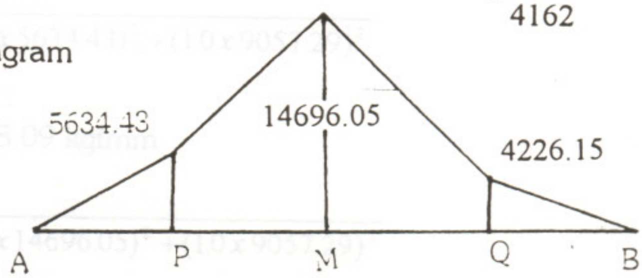
Horizontal force diagram



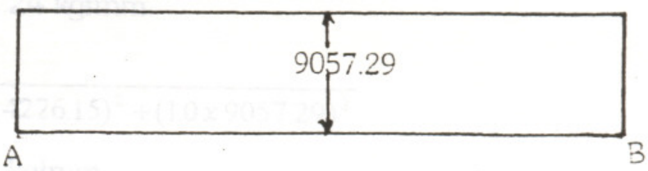
Horizontal bending moment diagram



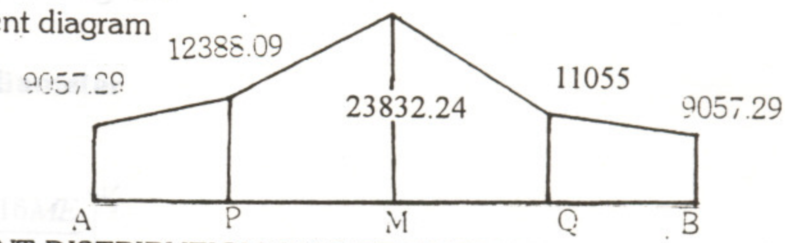
Resultant bending moment diagram



Twisting moment diagram



Equivalent bending moment diagram



FORCE AND MOMENT DISTRIBUTION ON INTERMEDIATE SHAFT

$$M_R \text{ at } Q = \sqrt{(733.6)^2 + (4162)^2}$$

$$= 4226.15 \text{ kgfmm}$$

F) Twiisting moment diagram (M_T)

$$M_T = \frac{5.4 x 4500 x 1000}{2 x \pi x 427} = 9057.29 \, kgfmm$$

G) Equivalant binding momemt (M_E) Diagram

Assuming K_M; combined shock and fatigue factor for bending = 1.5

and K_T; Combined shock and fatigue factor for torsion = 1

$$M_EA = \sqrt{(K_M \, x \, M_R)^2 + (K_T \, x \, M_T)^2} = 9057.29 \text{ kgfmm}$$

$$M_EB = 9057.29 \text{ kgfmm}$$

$$M_EP = \sqrt{(1.5 x 5634.43)^2 + (1.0 x 9057.29)^2}$$

$$= 12388.09 \text{ kgfmm}$$

$$M_EM = \sqrt{(1.5 x 14696.05)^2 + (1.0 x 9057.29)^2}$$

$$= 23832.24 \text{ kgfmm}$$

$$M_EQ = \sqrt{(1.5 x 4226.15)^2 + (1.0 x 9057.29)^2}$$

$$= 11055 \text{ kgfmm}$$

H) Selecion of shaft diameter

For C55 steel,

$$d = \left[\frac{16ME}{\pi Fs}\right]^{1/3}$$

$$Fs = 66x\ 0.3 Kgf/mm^2$$

Minimum shaft diameter at $P=\left[\frac{16x23832.24}{\pi\, x 66\, x\, 0.3}\right]^{\frac{1}{3}} = 18.3mm$

Keeping this requirement in consideration, the shaft diameter is selected as 20mm

II Force and moment distribution on horizontal during shaft with bevel gear.

A) Design of bevel gear

		Pinion	Gear
1.	No. of teeth	21	42
2.	Material	semisteel	semisteel
3.	Allowable static stress	$8.5 N/mm^2$	$8.5 N/mm^2$
4.	RPM	2050	1025
5.	Tooth Profile	$F=14\ 1/2^0$ composite	$F=14\ 1/2^0$ composite
6.	Diameter	Dp=43.5mm	Dg=89mm

Slant height of pitch cone (cone distance)

$$L = \sqrt{\left(\frac{Dg}{2}\right)^2 + \left(\frac{Dp}{2}\right)^2}$$

$$= \sqrt{\left(\frac{89}{2}\right)^2 + \left(\frac{43.5}{2}\right)^2} = 49.53mm$$

$$\text{Sin}\theta\ p1 \text{ for gear} = \frac{\underline{Dg/2}}{L}$$

$$\theta\ p1 = \text{Sin}^{-1}\ \frac{89}{2x49.53}\Big) = \underline{\underline{63°57'}}$$

Torque acting on the gear $= \dfrac{5.4x4500x1000}{2\pi\ 1025}$

$= \underline{\underline{3773.12Kg/mm}}$

Tangential force (W_T) acting at the mean radius Rm of the gear

Rm $=$ 89/2 $= \underline{\underline{44.5}}$

$M_T = \dfrac{T}{Rm} = \dfrac{3773.13}{44.5} = \underline{\underline{84.78Kgf}}$

Axial force (W_{RH}) acting on the gear

$= W_t \tan\theta \sin\theta\, p_1$

$= 84.78 \text{ x } \tan 14.5 \text{ x } \sin 63° 57'$

$= \underline{\underline{19.69 \text{ Kgf}}}$

Radial force (W_{RV}) acting on the gear

$= W_t \tan\theta \cos\theta\, p_1$

$= 84.78 \text{ x } \tan 14.5 \text{ x } \cos 63° 57'$

$= \underline{\underline{9.6287 \text{ Kgf}}}$

B) Vertical force diagram

Force acting on point A = Self weight of the disc = 1.75 kg.

The tension side of the belt is at an angle of 10° with the horizontal.

Force acting on point P $= \dfrac{5.4\,x\,4500\,x100}{2\pi\,x1025\,x\dfrac{125}{2}}$

$= \underline{\underline{60.38 \text{ Kgf}}}$

Vertical upward load at point P $=$ 60.38 x Sin10

$= \underline{\underline{10.48 \text{ Kgf}}}$

Vertical load on the Gear $= \underline{\underline{9.6287 \text{ Kgf}}}$

Taking moment about points.

$$-1.75\ (77.5) + 10.48 \times 57.5 - Fv_R \times 41 - 9.63 \times 91 = 0$$

$$Fv_R = -9.98\ Kgf$$

Assumed direction is to be reversed.

Vertical load on point S $= Fv_S = (10.48+9.63)-(1.75+9.98)$

$$Fv_S = \underline{\underline{8.38\ Kgf}}$$

C) Vertical bending moment diagram

B.M. at A = 0

B.M. at P = $-1.75 \times 20 = \underline{\underline{-35\ Kgfmm}}$

B.M. at S = $-1.75(77.5) + 10.48 \times 57.5$

= $\underline{\underline{466.975\ Kgfmm}}$

B.M. at R = $-9.63 \times 50 = \underline{\underline{-481.5\ Kgfmm}}$

B.M. at Q = 0

D) Horizontal force diagram

Horizontal force at the point P $= 60.38 \times \cos 10$

$= \underline{\underline{59.46\ Kgf}}$

Axial load on the bevel gear $= \underline{\underline{19.69\ Kgf}}$

Taking moment about points $-59.45 \times 57.5 + Fr \times 41 - 19.69 \times 91$

$F_HR = \underline{\underline{127.07 Kgf}}$

E) Horizontal Bending moment (M_H) diagram

B.M. at P = 0

B.M. at S = -59.45 x 57.5 = -3418.375 kgfmm

B.M. at R = -19.69 x 50 = -984.5 kgfmm

F) Resultant Bending moment (M_R) diagram

M_R at Q = 0

M_R at P = 35

M_R at S = $\sqrt{(-3418.375)^2 + (467)^2}$

= <u>3450.127 Kgfmm</u>

B.M. at R = $\sqrt{(-984.5)^2 + (-481.5)^2}$

= <u>1095.939 Kgfmm</u>

G) Twisting moment diagram (M_T)

$$M_T = \frac{5.4 x 4500 x 1000}{2\pi \, x \, 1025}$$

= <u>3773.14 Kgfmm</u>

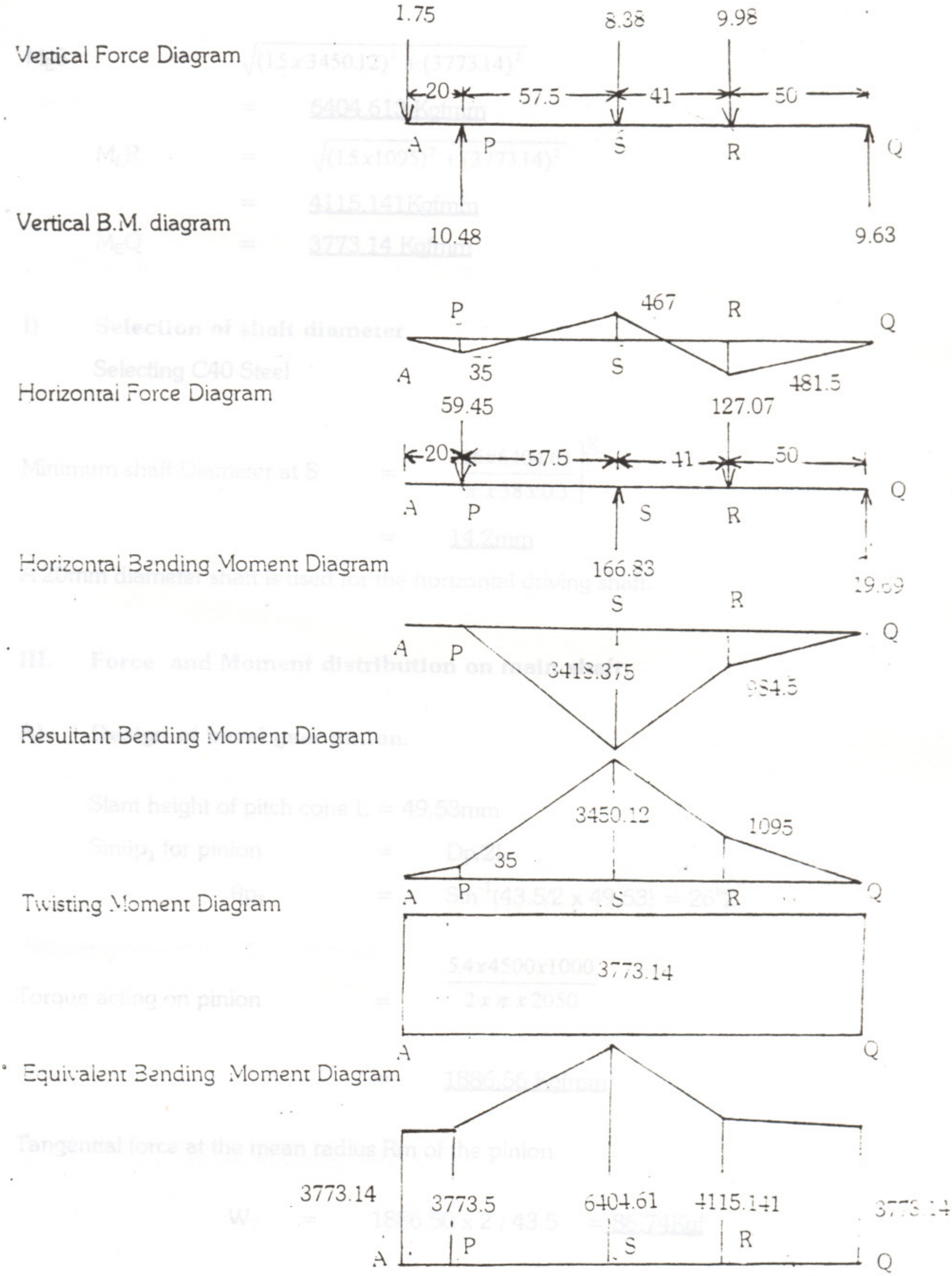
H) Equivalent bending moment diagram (M_E)

Assume $K_M = 1.5$ $K_T = 1.0$

at $M_E A$ = 3773.14 Kgf mm

$M_E P$ = $\sqrt{(1.5 \, x \, 35)^2 + (3773.14)^2}$

= <u>3773.50 Kgfmm</u>



FORCE AND MOMENT DISTRIBUTION ON HORIZONTAL DRIVING SHAFT.

$M_ES \quad = \quad \sqrt{(1.5\, x\, 3450.12)^2 + (3773.14)^2}$

$= \quad \underline{6404.613 \text{ Kgfmm}}$

$M_ER \quad = \quad \sqrt{(1.5 x 1095)^2 + (3773.14)^2}$

$= \quad \underline{4115.141 \text{Kgfmm}}$

$M_EQ \quad = \quad \underline{3773.14 \text{ Kgfmm}}$

I) Selection of shaft diameter

Selecting C40 Steel

$$\text{Minimum shaft Diameter at S} = \left[\frac{16 x 6404.61}{\pi\, x\, 38\, x\, 0.3}\right]^{1/3}$$

$= \quad \underline{14.2\text{mm}}$

A 20mm diameter shaft is used for the horizontal driving shaft.

III. Force and Moment distribution on main shaft.

A) Design of bevel gear pinion.

Slant height of pitch cone L = 49.53mm

$\text{Sin}\theta p_1$ for pinion = Dp/2L

$\theta p_1 = \text{Sin}^{-1}(43.5/2 \times 49.53) = 26°2'$

$$\text{Torque acting on pinion} = \frac{5.4 x 4500 x 1000}{2\, x\, \pi\, x\, 2050}$$

$= \quad \underline{1886.56 \text{ Kgfmm}}$

Tangential force at the mean radius Rm of the pinion

$W_T = 1886.50 \times 2 / 43.5 = \underline{86.74 \text{Kgf}}$

Axial force W_{RH} = $W_T \tan 14.5^0 \sin 26^0 2'$

= $86.74 \tan 14.5^0 \sin 26^0 2'$

= <u>9.84 Kgf</u>

Radial force W_{RV} = $86.74 \tan 14.5 \cos 26^o 2'$

= <u>20.15 Kgf</u>

Force acting at point S (Maximum force at a distance of 50mm from centreline of shaft)

= 1886.56 / 50 = <u>37.73Kgf</u>

B) Vertical force diagram

Vertical force at point S = 37.73 Sin 45o = <u>26.67 Kgf</u>

Vertical force at pont P = <u>9.84 Kgf</u>

Taking moment about R = - 26.67x127-Fva x 235-9.84(235+81)

FvQ = - 27.64 kgf

Assumed direction is to be reversed

FvQ = <u>27.64 Kgf</u>

FvR = 27.67 +27.64 -9.84

= <u>44.47 Kgf</u>

C) Vertical bending moment (Mv) Diagram

B.M at S = 0

B.M at	P	=	0	
B.M at	R	=	-26.67x127	= <u>- 3387.09Kgfmm</u>
B. M at	Q	=	-9.84x81	= -797.04 Kgfmm

D) Horizontal force diagram (F_H)

Horizontal force at S = 37.75 Cos 45 = <u>26.67 Kgf</u>

Horzontal force at P = <u>20.15 Kgf</u>

Taking moment about R = $26.67 \times 127 - F_{HQ} \times 235 + 20.15(235+81)$

F_{HQ} = 41.50 Kgf

F_{HR} = 26.67+41.50-20.15 = <u>48.02 Kgf</u>

E) Horizontal bending moment diagram (M_H)

M_H at S = 0

M_H at R = 26.67 x 127 = <u>3387.09 Kgfmm</u>

M_H at A = 20.15 x 81 = <u>1632.15 Kgfmm</u>

F) Resultant bending moment diagram (M_R)

M_R at S = 0

M_R at P = 0

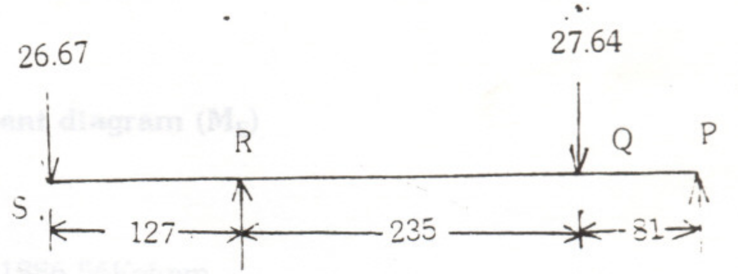
M_R at R = $\sqrt{(3387.09)^2 + (3387.09)^2}$ = <u>4790.13Kgfmm</u>

M_R at A = $\sqrt{(-797.04)^2 + (1632.15)^2}$ = <u>1816.37 Kgfmm</u>

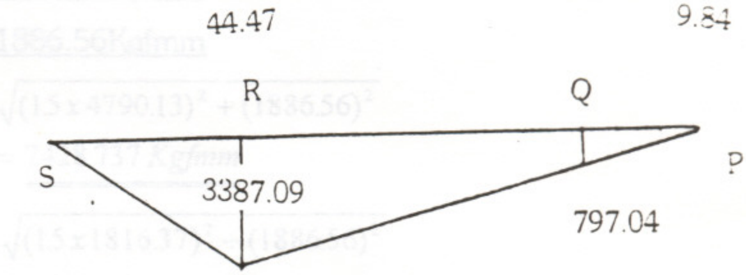
G) Twisting moment diagram M_T

$$M_T = \frac{5.4 \, x \, 4500 \, x \, 1000}{2 \, x \, \pi \, x \, 2050} = \underline{\underline{1886.56 \, Kgfmm}}$$

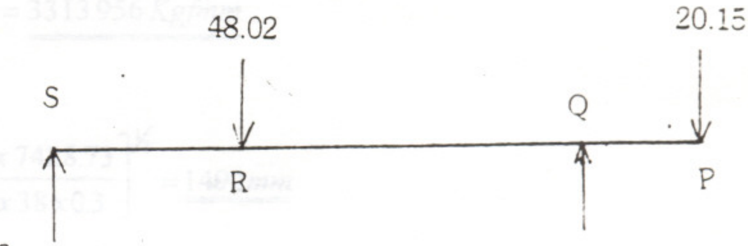
Vertical force diagram



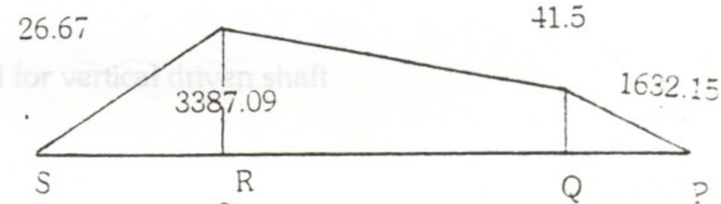
Vertical bending moment diagram



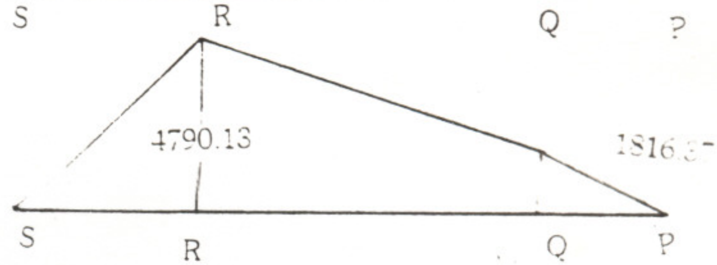
Horizontal force diagram



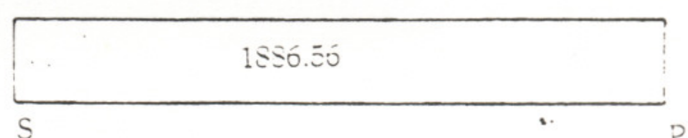
Horizontal bending moment diagram



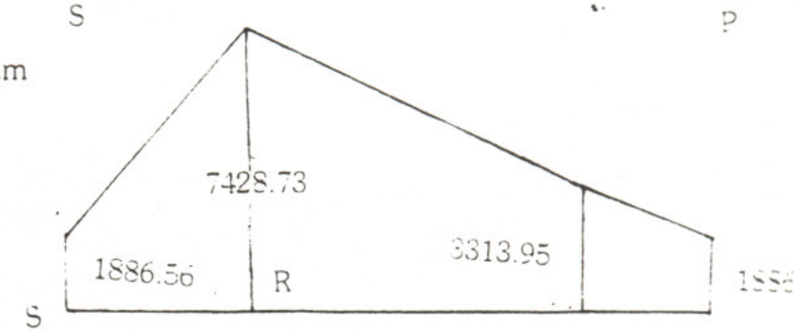
Resultant bending moment diagram



Twisting moment diagram



Equivalent bending moment diagram



FORCE AND MOMENT DISTRIBUTION ON VERTICAL SHAFT Q P

H) Equivalent bending moment diagram (M_E)

$K_M = 1.5$ & $K_T = 1.0$

M_E at S = $\underline{1886.56 Kgfmm}$

M_E at P = $\underline{1886.56 Kgfmm}$

M_E at R = $\sqrt{(1.5 x 4790.13)^2 + (1886.56)^2}$

$= \underline{\underline{7428.737\ Kgfmm}}$

M_E at Q = $\sqrt{(1.5 x 1816.37)^2 + (1886.56)^2}$

$= \underline{\underline{3313.956\ Kgfmm}}$

I) Selection of shaft diameter

Minimum shaft diameter at R= $\left[\frac{16 x 7428.73}{\pi\ x 38 x 0.3}\right]^{1/3} = \underline{\underline{14.92 mm}}$

A shaft diameter of 20mm is selected for vertical driven shaft

DESIGN, MODIFICATION AND PERFORMANCE EVALUATION OF SELF PROPELLED BUSH CUTTER

By

BIJU. K. VARGHESE
RAJIV ARAVINDAN
REMA. P. P

ABSTRACT OF THE PROJECT REPORT

Submitted in partial fulfilment of the requirement for the degree of

BACHELOR OF TECHNOLOGY IN AGRICULTURAL ENGINEERING

Faculty of Agricultural Engineering and Technology
KERALA AGRICULTURAL UNIVERSITY

Department of Farm Power Machinery and Energy
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1998

ABSTRACT

A self propelled bushcutter was developed at the K.C.A.E.T as a part of the project work keeping in mind, ever increasing labour charges. The initial fabrication of the bushcutter didn't fulfil the field working demands and it called modification of the unit. The variety of problems encountered was solved and the performance of the bushcutter was evaluated and the design aspects were checked under the project Design Modification & Performance Evaluation of Self propelled Bushcutter conducted at KCAET, Tavanur.

The cutting speed available at the ground was increased by effecting replacements of the pulleys in the different stages of power transmissions and the reversal of the individual gears of the bevel gear assembly. A separate bull pin system with a stopper mechanism was incorporated to prevent the blade rotations at transport. The front cover and pipe sleeve for the main shaft ensured long life working parts by preventing the clogging of stalks and weeds upon them. The locking system for side clutches and the adjustable height support wheel ensured easy handling of the cutting unit also three types of cutting devices were tested and recommended for different conditions.

The performance parameters studied included the field capacity, field efficiency, cutting efficiency and cost of operation. The field capacities were 0.1285 ha/hr, 0.115 ha/hr and 0.0488 ha/hr for the rigid blades, chains and circular saw blade respectively. The respective field efficiencies were 73.43%, 64.28% and 56.95%. The maximum cutting efficiency was observed for rigid blades. The cost of operation was found to be Rs. 412.09/ha compared to manual charges of Rs. 960/ha. all these indicate an essential replacement of human labour by the modified bushcutter for clearing vast tracts of bush infested land.