DEVELOPMENT OF A LOW COST GARDEN TRACTOR

BY
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THESIS
SUBMITTED IN PARTIAL FULFILMENT OF
THE REQUIREMENTS FOR THE DEGREE
MASTER OF SCIENCE IN AGRICULTURAL ENGINEERING
FACULTY OF AGRICULTURE
KERALA AGRICULTURAL UNIVERSITY

DEPARTMENT OF AGRICULTURAL ENGINEERING
COLLEGE OF HORTICULTURE
VELLANIKKARA - 680 654
TRICHUR
1982

Dedicated to my

beloved teacher

Late Dr. JOSE SAMUEL

DECLARATION

I hereby declare that this thesis entitled "Development of a low cost garden tractor" is a bonafide record of research work done by me during the course of research and that the thesis has not previously formed the basis for the award to me of any degree, diploma, associateship, fellowship or other similar title of any other University or Society.

(M. SIVASWAMI)

Vellanikkara,
10th November , 1982.

CERTIFICATE

Certified that this thesis entitled "Development of a low cost garden tractor" is a record of research work done independently by Shri. M. SIVASWAMI under my guidance and supervision and that it has not previously formed the basis for the award of any degree, fellowship, or associateship to him.

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We, the undersigned, members of the advisory committee of Shri. M. SIVASWAMI, a candidate for the degree of Master of Science in Agricultural Engineering with major in Farm Power and Machinery, agree that the thesis entitled, "Development of a low cost garden tractor" may be submitted by Shri. M. SIVASWAMI in partial fulfilment of the requirements for the degree.

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ACKNOWLEDGEMENTS

I have immense pleasure to express my deep sense of gratitude, indebtedness and respect to the late Dr. Jose Samuel, Professor and Head, Department of Agricultural Engineering, who was the Chairman of the Advisory Committee during the first phase of the investigation, for his valuable guidance, constant encouragement and constructive criticism.

I have boundless pleasure in recording my indebtedness and profound thanks to Shri. C.P. Muhammad, Associate Professor, Department of Agricultural Engineering and the Chairman of the Advisory Committee for fabrication of the first prototype and for his valuable guidance and keen interest during the investigation and in preparation of the thesis.

I wish to express my thanks to Prof. T.P. George, Head, Department of Agricultural Engineering, Dr. R. Vikraman Nair, Professor of Agronomy and Shri. K. John Thomas, Department of Agricultural Engineering for their advice and suggestions during the investigation.

I am extremely grateful to Shri. K. Madhavan Nair, Associate Professor, Instrumentation Centre and the technicians of the Agricultural Engineering Research Workshop, Mannuthy for their unstinted help and assistance.

I acknowledge my gratitude to the staff members of the College of Horticulture, for providing necessary infrastructure facilities for conducting the research work.

Thanks are also due to the Kerala Agricultural University for awarding me the research fellowship for the post-graduate programme.

On a personal note, I am indebted to my beloved parents for their constant help and encouragement.

(M. SIVASWAMI)

Vellanikkara,
10th November, 1982.

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SYMBOLS AND ABBREVIATIONS

Agric.	Agricultural
ASAE	American Society of Agricultural Engineers
cc	cubic centimetre(s)
CIAE	Central Institute of Agricultural Engineering
cm	centimetre(s)
Co.	Company
Coeff.	Coefficient
contd.	continued
C.R.R.	coefficient of rolling resistance
C.T.	coefficient of traction
deg	degree
Dept.	Department
edn.	edition
et al.	and other people
FAO	Food and Agriculture Organization
Fig.	Figure
gm	gram(s)
ha	hectare(s)
hp	horse power
hr	hour(s)
ICAR	Indian Council of Agricultural Research
IRRI	International Rice Research Institute
ISAE	Indian Society of Agricultural Engineers
J.	Journal
kg	kilogram(s)
kgf	kilogram force
kmph	kilometres per hour
kw	kilowatt(s)
lit	litre(s)
log	logarithm

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log	logarithm

Ltd.	Limited
m	metre(s)
MA	main assembly
max	maximum
min	minimum
mm	millimetre(s)
MS	Mild Steel
N.A.	not applicable
NCAER	National Council of Applied Economic Research
No.	Number
OD	outer diameter
P.	page
pp.	pages
Proc.	Proceedings
Pvt.	Private
Res.	Research
RNAM	Regional Net Work on Agricultural Machinery
rpm	revolutions per minute
Rs.	Rupees
SA	sub-assembly
sec	second
TNAU	Tamil Nadu Agricultural University
UNIDO	United Nations Industrial Development Organization
Unpubl.	Unpublished
USA	United States of America
/	per
%	per cent
π	pi, (22/7)
Σ	sum of

Introduction

INTRODUCTION

Agriculture is the main occupation in most developing countries, yet many of them find it difficult to grow sufficient food for their population. An increase in food production is achieved by the combined effect of increased power input, improved seeds and the modified production technology in agriculture. The entire farm operation can be completed efficiently in time, only if sufficient power is available in all farms. And thus the farm power is the prerequisite for all agricultural developments in any country.

1.1. Farm Power Status in India

The total geographical area of India is 328.78 million hectares and out of this only 161.10 million hectares were under cultivation in 1980. The National Commission on Agriculture (1976) assessed the power available in Indian farms as 0.30 hp per ha as evident from the Table 1. Fig. 1 shows the percentage distribution of the various sources of farm power in India.

1.2. Minimum Power Requirements and Availability

With the introduction of high yielding varieties and new irrigation practices the cropping pattern has considerably changed, resulting in a tremendous increase in the demand for power input. According to Giles (1967) a minimum

Table 1. Power availability for Indian agriculture

Sl. No.	Source	Quantity (million Nos.)	Total hp available (million)
1	Human labours	81.40	5.8
2	Draught animals	63.30	25.3
3	Mechanical and electrical:		
	i) Tractors and power tillers	0.09	1.9
	ii) Diesel engines and electric motors	2.41	15.7
	iii) Power sprayers and dusters	0.24	0.5
	iv) Electrically operated sugarcane crushers	0.10	0.6
	Total	147.54	49.8

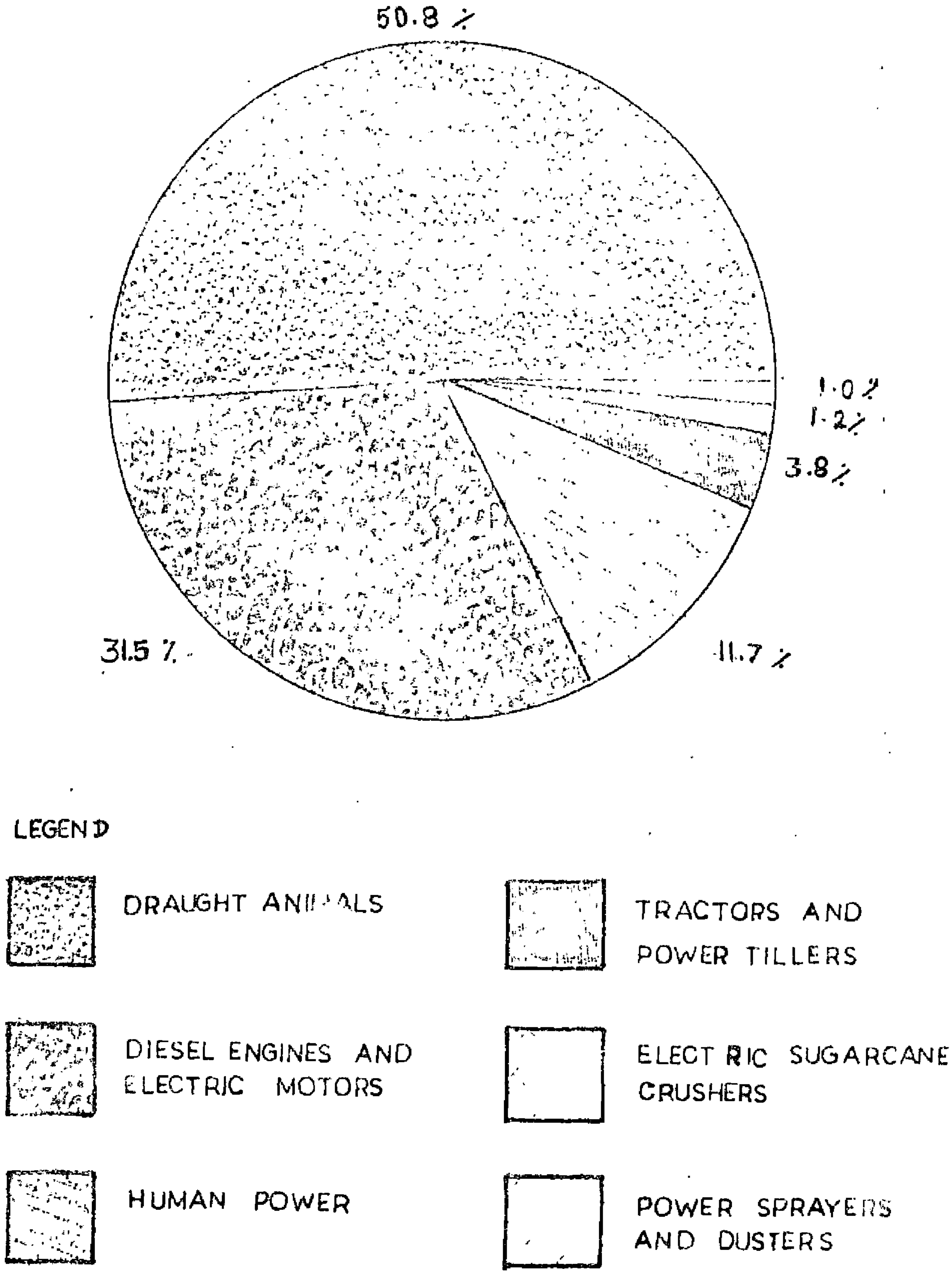


FIG. 1 PERCENTAGE DISTRIBUTION OF FARM POWER SOURCES IN INDIA

power input of 0.50 hp per ha is necessary to achieve sustained agricultural growth in developing countries.

A study conducted by the Food and Agriculture Organization (1969) also revealed that the power input in the developing countries might be increased atleast to 0.50 hp per ha.

The total power availability in the developing countries is in the range of 0.10-0.30 hp per ha as compared to the 1.40 hp per ha in the USA and 4.70 hp per ha in West Germany, according to the Expert Consultation on the Mechanization of Rice Production (1974).

Ministry of Agriculture and Irrigation, Government of India (1977) stated that, with the huge population of human labour and animal power, India suffered from power famine and this affected agricultural production based on scientific farming, and it has recommended a doublefold growth of the available 0.36 hp per ha, which agrees with the recommendation of 0.80 hp per ha by the National Commission on Agriculture (1976) for an efficient agriculture.

1.3. Case for Mechanization of Agriculture

Man by himself can produce a little only, but with the help of machinery he can easily produce more and he can get relieved of the heavy work. While increasing the output per ha and per man, it will render the large cattle population as surplus, which otherwise will compete subsistence from land.

Nunn and Balis (1973) reported that India has the resources, manpower, facilities and desire to obtain a mechanized agriculture. Khan (1973) observed in Philippines that more mechanized farms have an increased labour utilization. In highly mechanized rice farms of Japan, labour utilization is as high as 1400 man-hours per ha. Also tractors are reported to be five times fast and forty per cent cheaper than animal power in Bangladesh (Haq, 1975).

Utilization of tractors in India conferred definite benefits in terms of greater efficiency, economy, higher productivity and additional employment upto 2644.5 man-hours per tractor, (ISAE, 1978). Tractors in all selected regions in India showed positive results with an increased average yield and increased employment due to timely and effective farm operations, (RNAM, 1980). It also increased the energy input and net profit. It has been analysed that for berseem cultivation the demand of tillage energy under tractor (210.76 hp hr per ha) was more than that of bullock farming (142.45 hp hr per ha). Irrigation consumed maximum energy (tractor farms - 1050 hp hr per ha and bullock farms - 1065 hp hr per ha), followed by seed bed preparation, and the net profit under tractor farming was Rs.8848.50 per ha and under bullock farming Rs.7875.60 per ha, (Ram et al., 1980). It is in confirmation with the results obtained by the National Council of Applied Economic Research, which found that tractor farms in India have 20 per cent higher yield and 5 per cent higher overall employment. And tractor owners earn 4.5 times more profit than bullock farmer and the holding size is clearly not relevant for establishing a viability for running a tractor, (Baig, 1980; ISAE, 1981).

Besides, the agricultural machines can be used where it is very difficult to engage animals.

1.4. Case against Mechanization of Agriculture

Another school of thought is that mechanization has limited scope in India because of the small holdings, average being less than 2.3 ha, scattered in tiny plots. Also, Agarwal (1979) observed that abundant animal and human power, illiteracy and lack of trained personnel are the constraints in mechanizing the Indian farms.

Khan and Duff (1972) while explaining India as an example, observed that nearly twenty years of efforts to introduce western mechanization technology in Asia, the ownership and the use of such equipment remains beyond the means of majority of farmers. The benefits accruing from modern mechanization technology are concentrated in a subsector composed of a few, larger and prosperous farmers and have not available to the majority of small farmers in India.

1.5. Two Wheel Tractor

It is suggested that, for a country like India, most suitable kind of machine would be a two wheel tractor, which unlike a large tractor, would replace animals but not people and would be consistent with the country's objectives of promoting economic development, employment and better income distribution, (Hamid, 1973; Datt and Sundharam, 1976). The two wheel tractor stands between the animal traction and four wheel tractor and so forms the first phase of mechanization.

Although the power machinery are being produced in the country with foreign collaboration, these are found to be too sophisticated and costly for the average Indian farmers, (Policarpio, 1973; Reddy, 1973).

Japan has been able to mechanize its agriculture through a variety of small equipments designed and developed to suit local resources and economic condition without replacing labour. On the same line of appropriate technology, we have to provide sufficient mechanical power source with less capital for agricultural machinery production.

The requirements of small farmers and local agricultural machinery fabricators must be the important criteria in selecting a mechanization strategy for this country. Pellizzi and Turrini (1973) suggested that indigenous manufacture of small and low powered equipment which will be compatible not only with the agricultural, social and economical conditions but also with the manufacturing process, repairing facility and materials that are available in the region. Kherdekar (1973) reported that the Government of India have guided to develop farm machinery and implements having durability and simplicity. It is very much stressed that the immediate necessity of developing countries is promoting the local manufacture of agricultural machinery and designing, developing and testing of simple power machinery based upon an intermediate technology, (Jinasena, 1973; UNIDO, 1974).

Samuel (1970) felt that intermediate technology could be of assistance and that a light weight, low cost garden tractor if designed and developed could serve as a link between animal power and the factory made power tiller and tractor. Reddy (1973) and the Ministry of Agriculture and Irrigation, Government of India (1977) emphasized the need to take up design and development of a simple, low cost and light weight 5-7 hp two wheel garden tractors, to continue in the line of the International Rice Research Institute, Philippines for the rapid acceptance by the Indian farmers. It should bring better employment generation, high foreign exchange saving, adhere to horizontal integration using existing manufacturing facilities and development of small scale industries.

It was therefore decided to undertake the development of a garden tractor with the following objectives.

1.6. Objectives

a) To select suitable design criteria for development of a low cost garden tractor.

b) To select a promising unit from the available engines for using in the proposed garden tractor.

c) To adapt promising design concept for optimizing the various components of the low cost garden tractor.

d) To carry out an adaptive design work and to fabricate a prototype unit with maximum use of standard off-the-shelf components to facilitate local manufacture.

e) To conduct the tests and its performance in the field and road conditions to examine the design snag between the performance and functional requirements for further modifications.

Review of Literature

REVIEW OF LITERATURE

A brief review of history, classification and work done on design, development and testing of low cost garden tractors are presented in this chapter.

2.1. History and Classification of Garden Tractors

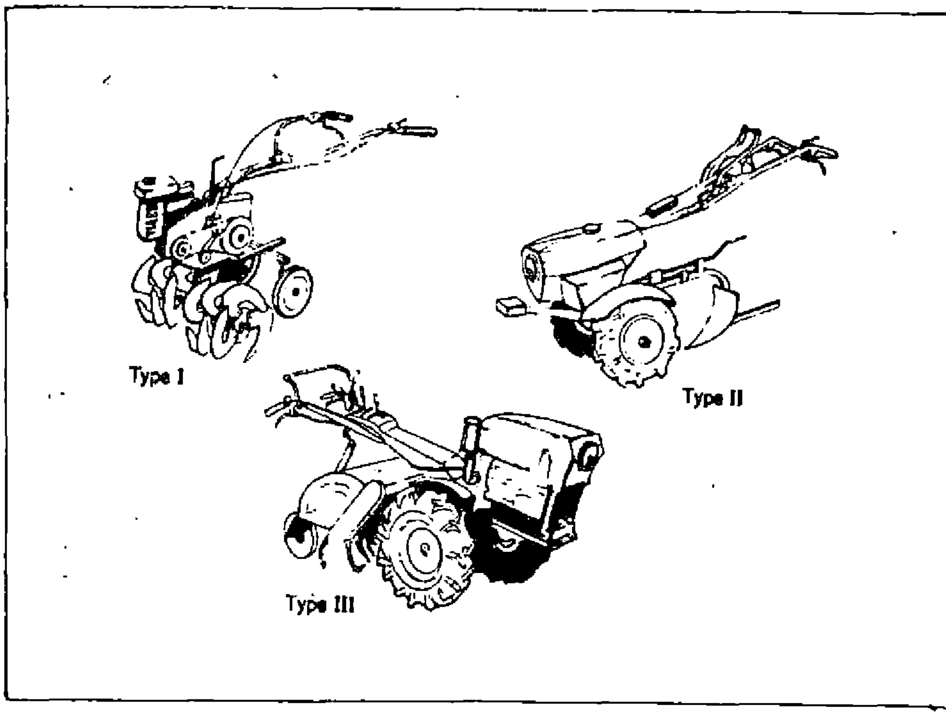
The garden tractors with different basis of classifications have a history of about sixty years old as seen from the literature.

2.1.1. History

Tsuchiya (1965) and Sakai (1968) reported that around 1923 Japan imported 'Utiliter' and 'Simar' type garden tractors from Europe for remaking to suit Japanese farming systems. According to Policarpio (1973) and Manalili (1974) these two wheel tractors are called as walking or garden tractors in the United States, motoolteurs in France, rotovators in England and power tillers in Japan.

2.1.2. Classification

The garden tractors have been classified with respect to the power output and the type of general construction, such as (i) light duty single axle, 4-6 hp garden tractor (ii) medium duty single axle garden tractors, upto 8 hp and

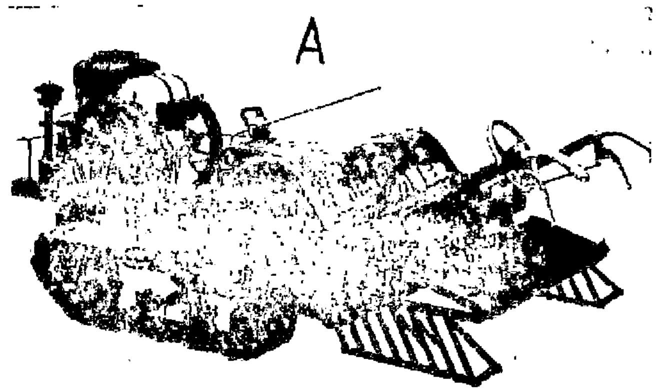


Type I Light duty single axle 4-6 hp garden tractor

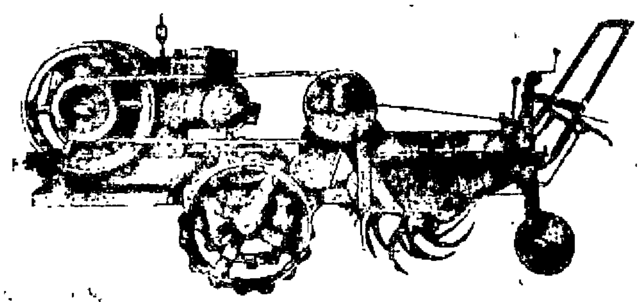
Type II Medium duty single axle garden tractor upto 8 hp

Type III Heavy duty double axle 8-14 hp garden tractor

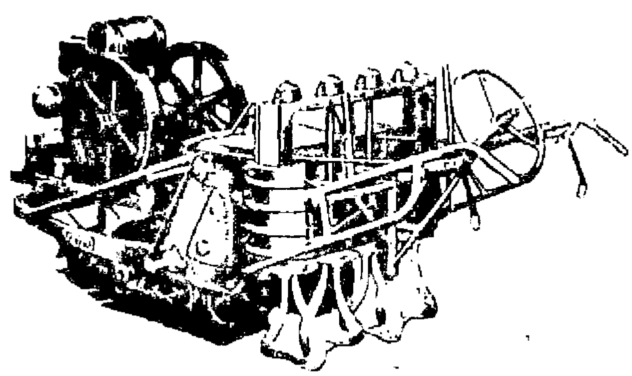
Fig.2 CLASSIFICATION OF GARDEN TRACTORS
(Policarpio, 1973)



AKIYAMA, CRANK TYPE



KUBOTA, ROTARY TYPE



FURUKAWA, SCREW TYPE

Fig.3 PRINCIPAL TYPES OF GARDEN TRACTORS
(Tsuchiya, 1965)

(iii) heavy duty double axle, 8-14 hp garden tractors, (Fig.2); and the mode of soil cutting, such as (i) crank, (ii) rotary and (iii) screw types, (Fig. 3). Mahmud (1974) classified them into (i) large, drive type garden tractors, (ii) medium dual purpose garden tractors and (iii) small traction type garden tractors, (Fig. 4, Fig.5 and Fig.6).

2.2. Garden Tractors in Indian Agriculture

Field trials of Japanese power tillers or garden tractors were carried out in India at the Indian Agricultural Research Institute, New Delhi during 1955 to 1966, with a view of introducing it in India and found its suitability for Indian Agriculture. The growth of demand for garden tractors has not been encouraging in India, since their introduction, although the Planning Commission agreed that development of garden tractors will help in transforming the rural economy, (ISAE, 1975).

There are six firms producing garden tractors of power ranging from 5-12 hp in India, which were designed and developed abroad to meet the specific requirements there, (Table 2 and Table 3). They are not gaining popularity in India, because their construction is complex, weight is higher, cost is prohibitive, not repairable locally and are mass produced in factories. Hence the Government of India is considering the question of making two wheel tractors

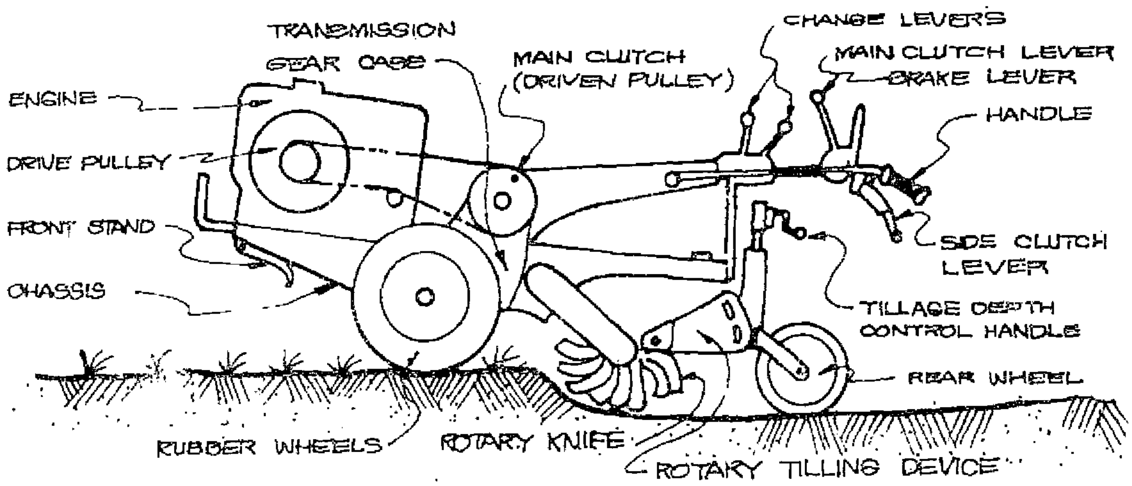


Fig. 4 LARGE DRIVE TYPE GARDEN TRACTOR
(Mahmud, 1974)

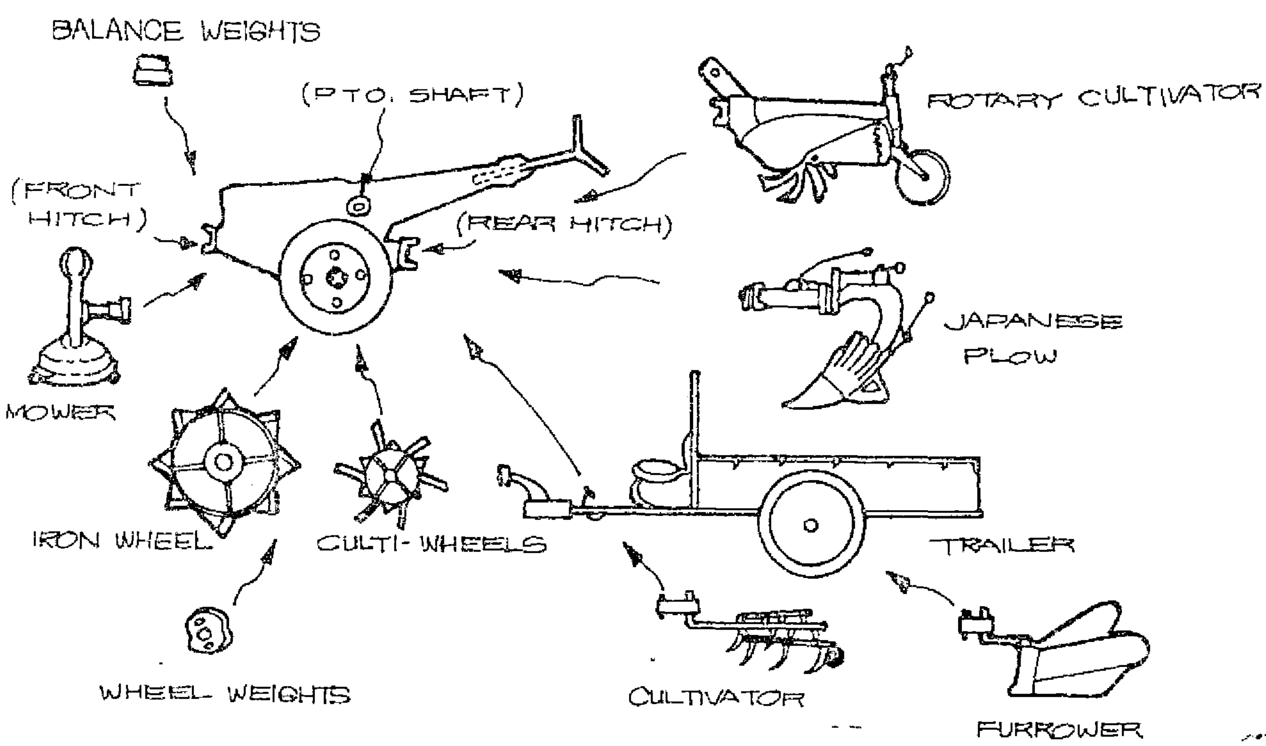


Fig.5 MEDIUM DUAL TYPE GARDEN TRACTOR
(Mahmud, 1974)

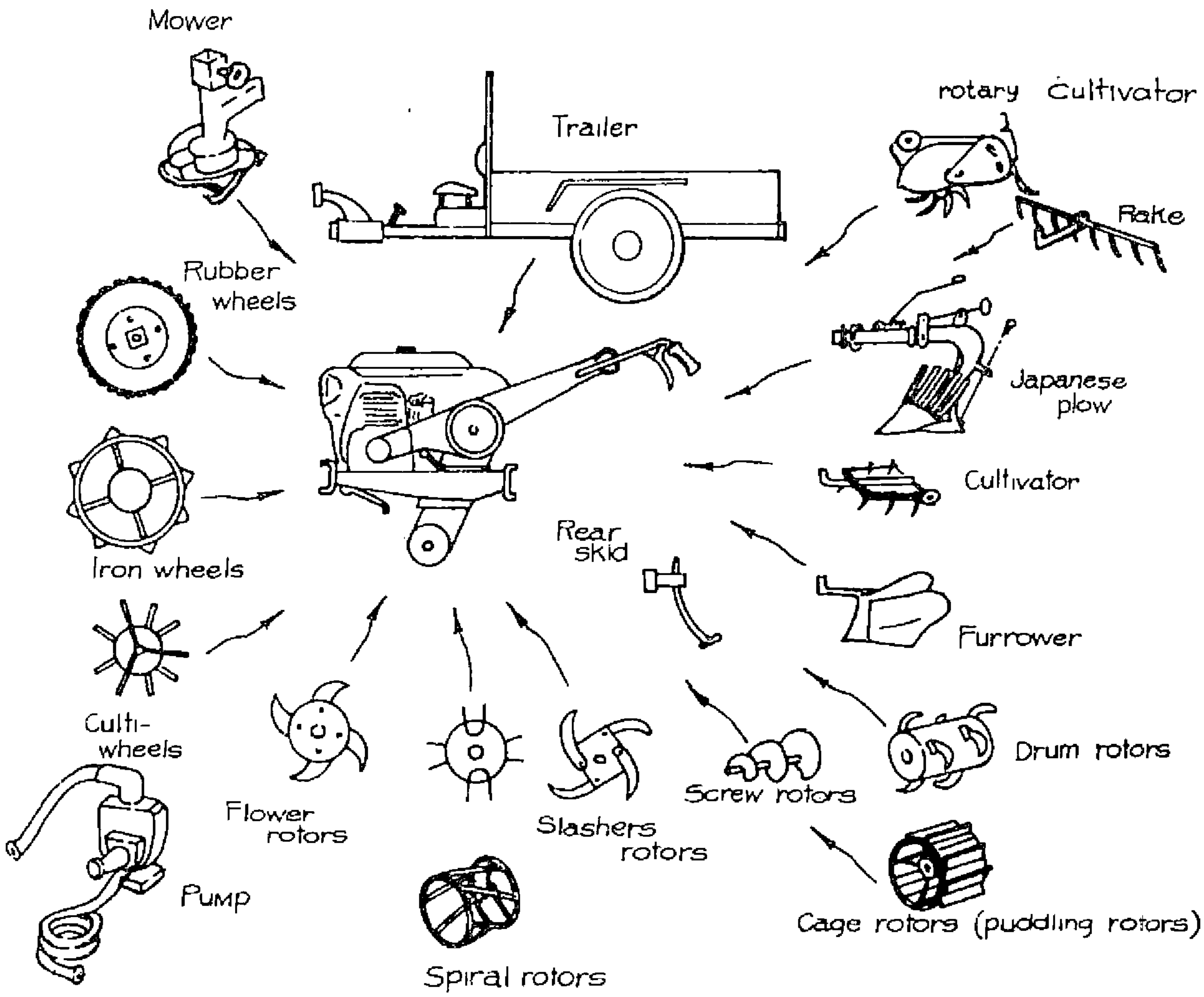


Fig.6 SMALL TRACTION TYPE GARDEN TRACTOR (Mahmud, 1974)

Table 2. Details on the garden tractors produced in India

Sl. No.	Name of the firm	Make	hp	Capacity per annum		Price as on 31-1-1982
				Licensed	Installed	
1	M/S. Indequip Engg.Ltd., Baroda, Ahmedabad	ISEKI	5-7	10000	(revoked)	N.A.
2	M/S. J.K. Satoh Agrl. Machines Ltd., Kanpur	SATOH	5-7	6000	5000	19,410
3	M/S. Kerala Agromachinery Corpn. Ltd., Angamali, Alwaye	KUBOTA	9-12	12000	3000	29,000
4	M/S. Krishi Engines Ltd., Hyderabad	KRISHI	5-7	3000	3000	17,520
5	M/S. Maharashtra Co-op. Engg. Society Ltd., Kolhapur	YANMAR	8-12	4000	(cancelled)	N.A.
6	M/S. VST Tillers Tractors Ltd., Bangalore	MITSUBISHI SHAKITI JANATA	8-10	5000	4000	22,000

Table 3. Production of indigenous garden tractors

Sl. No.	Make / Year	ISEKI	SATOH	KUBOTA	KRISHI	YANMAR	MITSUBISHI	Total
1	1965-67	-	-	-	906	-	-	906
2	1967-69	-	-	-	357	-	-	357
3	1969-71	-	-	450	629	113	509	1701
4	1971-73	-	-	411	254	262	1098	2025
5	1973-75	255	500	748	306	-	1762	3571
6	1975-77	249	453	1002	563	165	2015	4447
7	1977-79	100	230	1389	298	-	1814	3831
8	1979-81	-	128	553	212	-	1537	2430
	Total	604	1311	4553	3525	540	8735	19268

available at reduced prices for the benefit of the small farmers of the country.

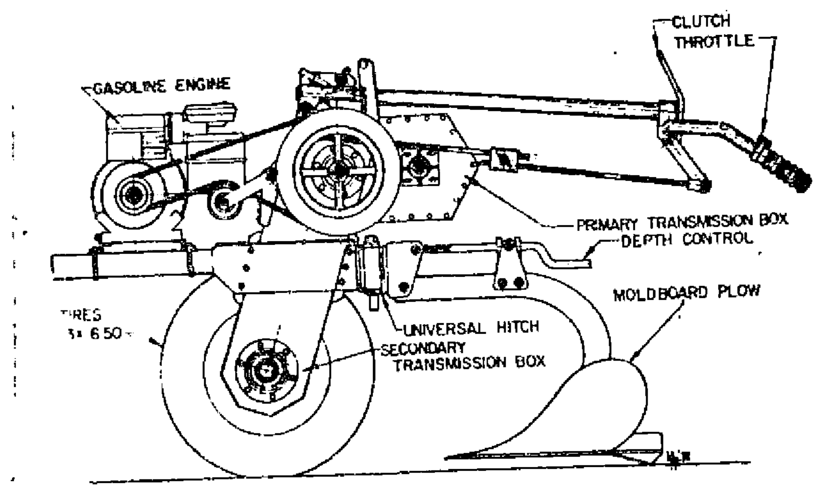
2.3. Low cost Garden Tractors

Although considerable work has been done abroad for the development of a cheap and efficient garden tractor, the work done in India for a purely indigenous version is limited.

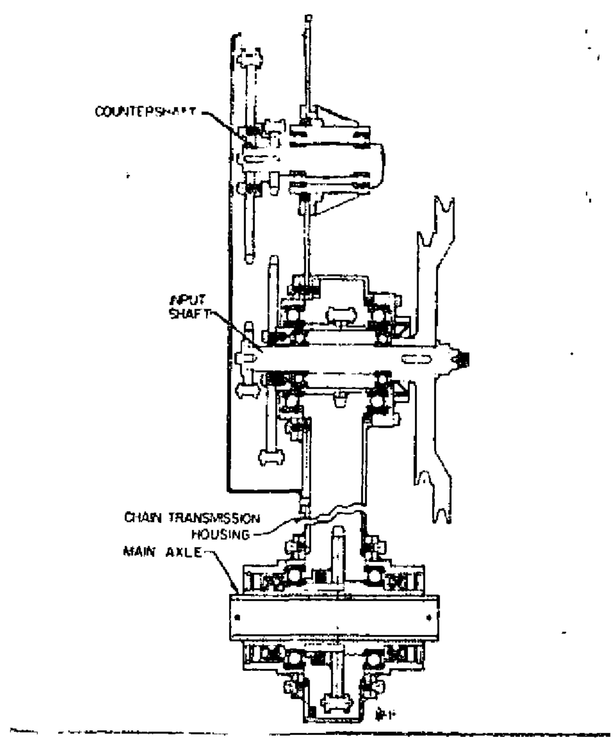
2.3.1. Work done in Abroad

Design, development and manufacture of low cost garden tractors were taken up in Japan, Federal Republic of Germany, Britain, Philippines, Tailand, Korea and in Sri Lanka with prime movers of output power ranging from 3.5 hp to 19.0 hp for performing all field operations without depriving of labour, (Mukumoto, 1959; Kishida, 1969; Devakul, 1971; Khan, 1973; Nicholas, 1974; Dias, 1975; Lee, 1975).

The International Rice Research Institute, Philippines developed a low cost 5-7 hp garden tractor and conducted a series of tests to determine the performance and suitability for small farms of the developing countries. In the IRRI garden tractor (Fig.7-a), clutch and first step speed reduction have been combined by using a belt idler with a cone V pulley and it has a three step speed reduction with standard



(a)



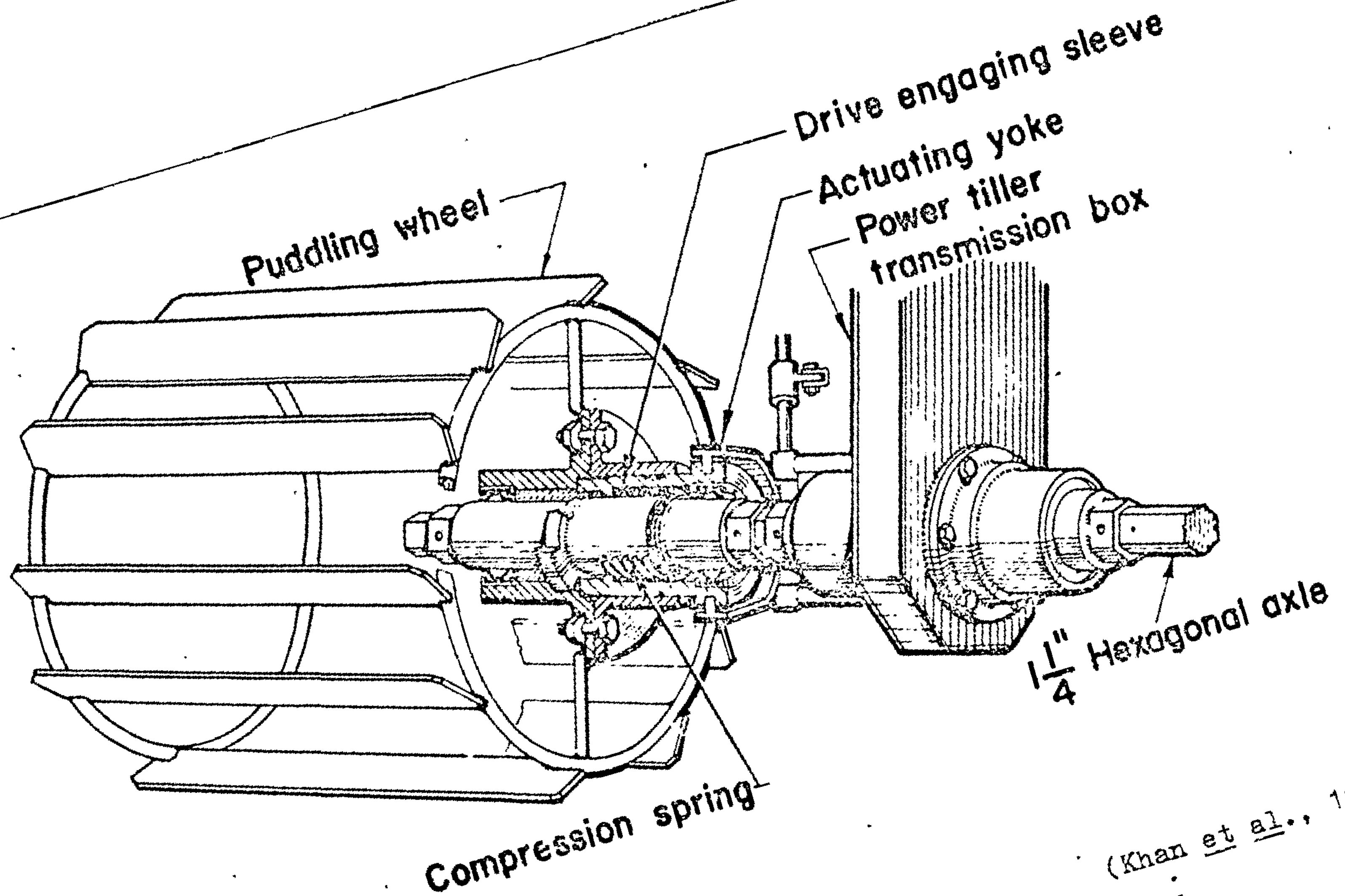
(b)

Fig.7 SCHEMATIC DRAWINGS OF (a) IRRI 5-7 hp GARDEN TRACTOR AND (b) ITS POWER TRANSMISSION SYSTEM

(Policarpio,1973)

chain and sprocket transmission (Fig. 7-b). It has been found that this type of transmission system reduced the cost of production, repair and total weight of the unit, (Policarpio, 1973). IRRI (1979) reported that as an improvement, the idler pulley type clutch was eliminated by mounting a pivotted countershaft where the clutching is effected by swinging the countershaft with the clutch lever which changes the centre distance between engine shaft and the countershaft. This could be operated as a safety device against overload and sudden shock other than merely for transmission of power.

IRRI first developed an externally mounted steering clutch as shown in the Fig.8 and noticed improper functioning due to mudclogging. Hence a pair of totally enclosed jaw clutch sliding on the hexagonal axle as shown in the Fig.9 was tried and found that disengaging was difficult under full load condition. For avoiding that, a more compact steering clutch was designed and fabricated as shown in the Fig.10. Here the drive-engaging sleeve slides on the axial ball bearings during engagement and disengagement. Under normal straight travel, spring pressure holds the clutch in the engaged position. However, these types of clutches are not functioning satisfactorily under field conditions and the garden tractors with this arrangement



(Khan *et al.*, 1975)

Fig.8 IRRI EXTERNALLY MOUNTED STEERING CLUTCH

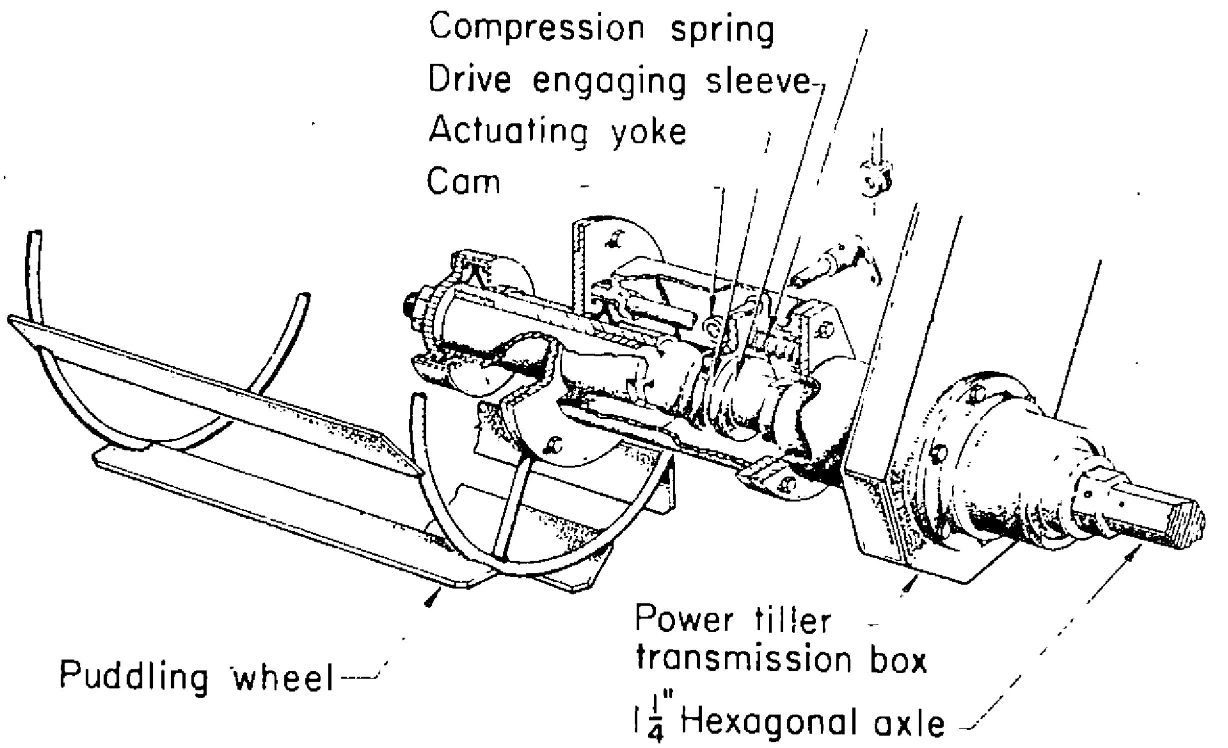


Fig.9 IRRI TOTALLY ENCLOSED STEERING CLUTCH
(Khan et al., 1975)

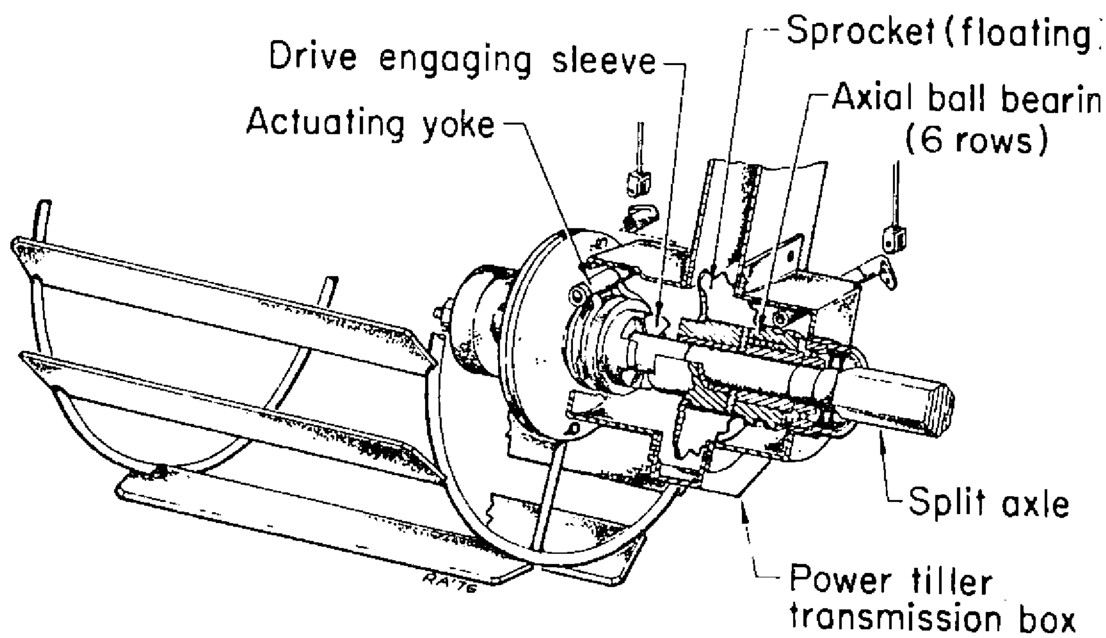


Fig.10 IRRI TOTALLY ENCLOSED STEERING CLUTCH WITH AXIAL BALL BEARING (Khan et al., 1975)

are not popular in the developing countries, (Khan et al.,1975).

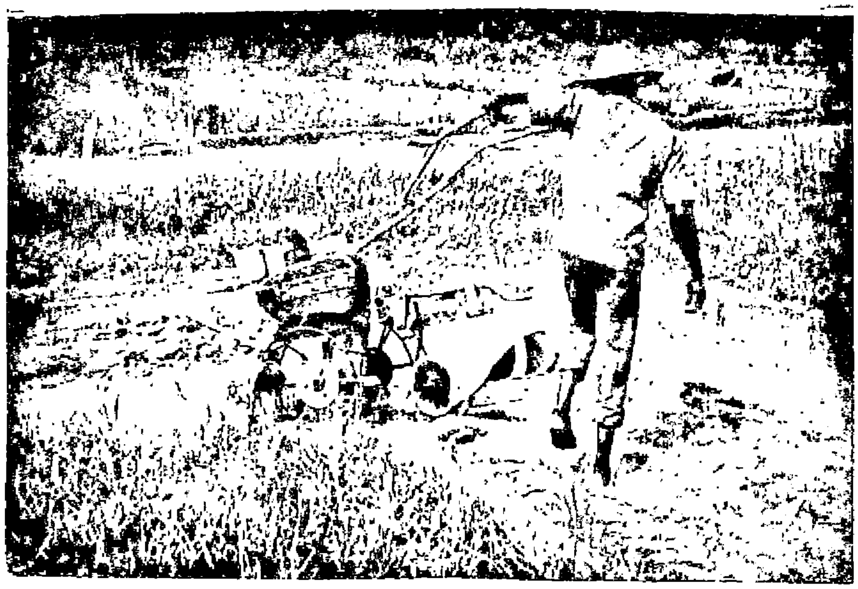
Wijewardena (1976) reported that 'Landmaster' 5-7 hp hand tractor for upland and lowland farming is having very simple means for turning the tractor at headlands, (Fig.11).

2.3.2. Work done in India

Samuel (1970) fabricated a light weight garden tractor with a 3.5 hp Indian built Villiers engine and a modified 'Economy' model imported chassis, which had been originally built for Briggs and Stratton engine and reported for further improvements.

In 1980, India received five units of 6-7 hp diesel powered 'Self Helper' from USA through Action for Food Production Scheme for extensive evaluation in laboratory and in farmers fields and the results are awaited, (ISAE, 1981).

Manian (1980) designed and fabricated a low cost power tiller with 5.4 hp Lombardini diesel engine and studied the structural and transmission systems. Satisfactory results have been obtained for the garden tractor with transmission ratios of 3.11:1, 4.5:1 and 4.0:1 for single stage belt and double stage chain drives. It has separate steering clutches and the total weight is 178 kg. Tamil Nadu Agricultural University, Coimbatore developed a 5.4 hp diesel powered low cost power tiller (Fig.12) and reported its suitability for



TURNING AT HEADLAND

STRAIGHT TRAVEL

Fig.11 LAND MASTER GARDEN TRACTOR IN FIELD OPERATIONS (Wijewardena, 1976)

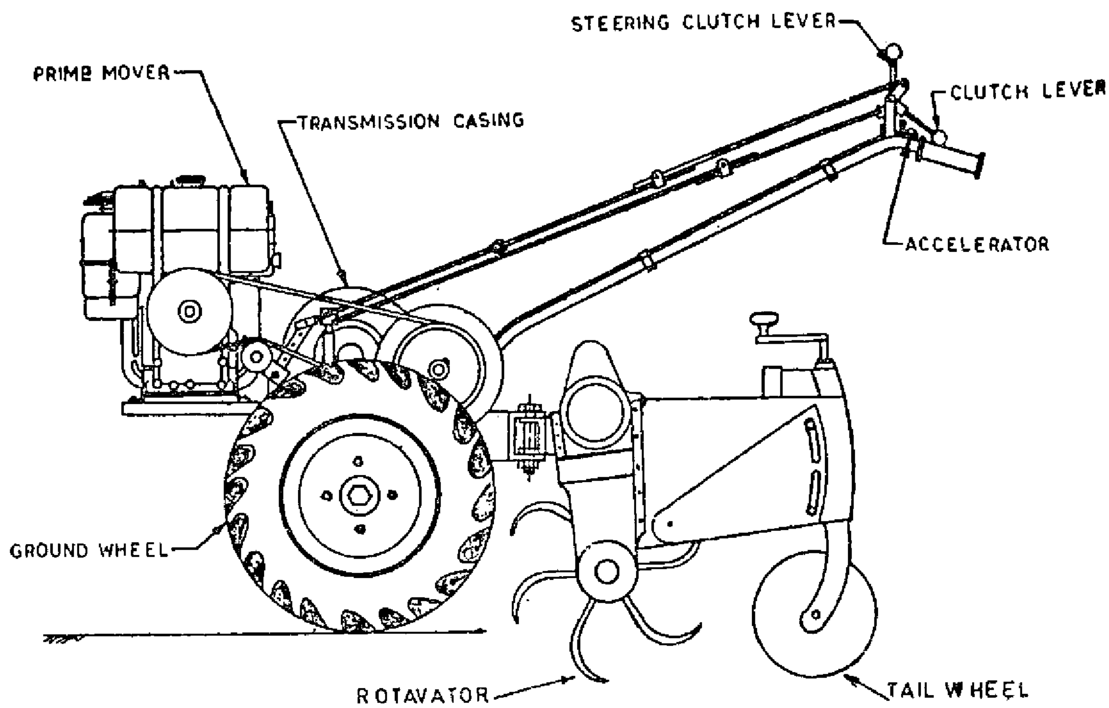


Fig.12 5.4 hp POWER TILLER DEVELOPED AT TNAU (TNAU, 1980)

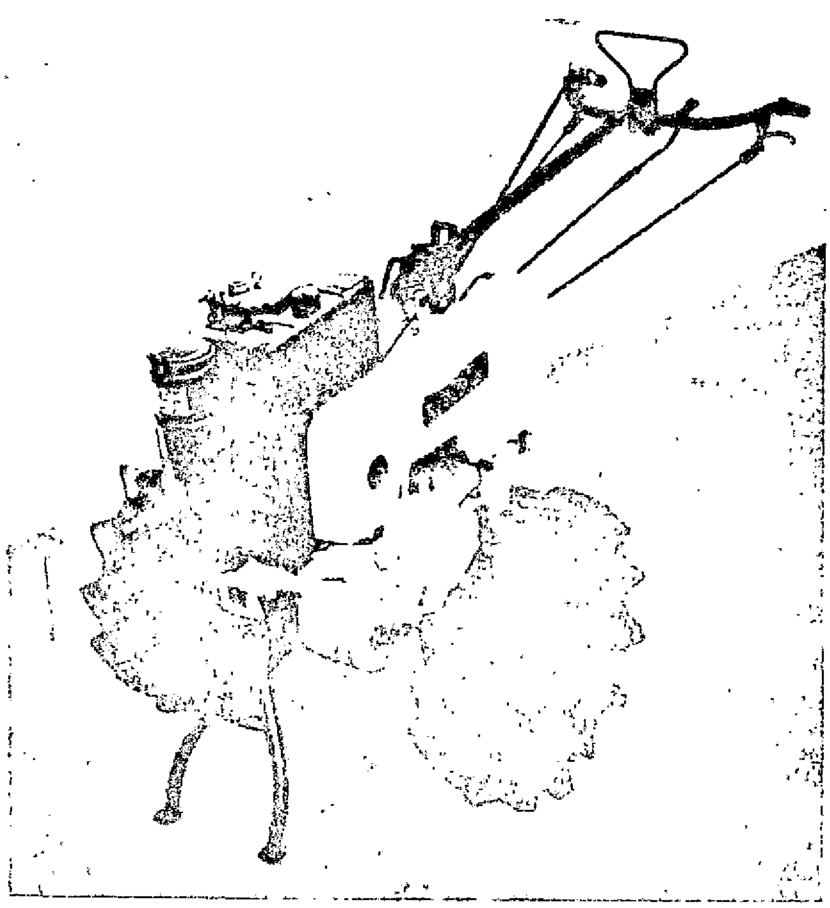


Fig.13 IRRI TYPE 5.4 hp NATIONAL POWER TILLER

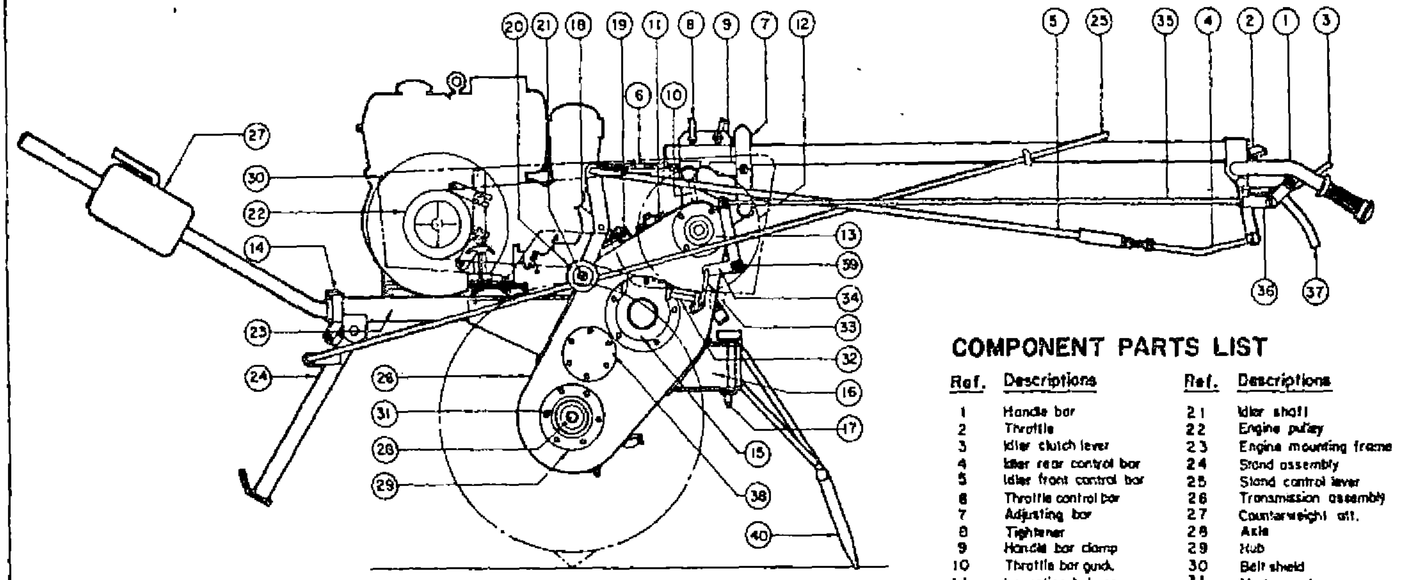
hilly regions with a field capacity of 0.41 ha per day of 8 hours of actual ploughing operation, (TNAU, 1980; CIAE, 1980). Yadav et al. (1980) reported that a prototype 8-10 hp garden tractor had been designed and developed to serve as an ideal source of power for small and medium sized Indian farms in comparison with the existing power-tillers.

Recently M/S. National Engineering Co. (Madras) Pvt. Ltd., Madras, started manufacturing the IRRI type 5.4 hp power tillers (Fig.13 and Fig.14) but it has not gained much popularity in Indian Agriculture. The exploded view of these IRRI type power tiller transmission parts along with the steering clutch arrangement is given in the Fig.16.

2.4. Traction Studies

Southwell (1963) investigated the comparative performance of various forms of multiaxle and single axle tractors and reported that the single axle tractor had 1.25 times greater pull, 1.30 times maximum average drawbar hp, 1.02 times higher pull to weight ratio and 1.72 times higher power to weight ratio than the conventional four wheel tractors.

Chang and Cooper (1969) reported that the slip loss was much higher than the rolling resistance loss at slips above 25 per cent.



COMPONENT PARTS LIST

Ref.	Descriptions	Ref.	Descriptions
1	Handle bar	21	Idler shaft
2	Throttle	22	Engine pulley
3	Idler clutch lever	23	Engine mounting frame
4	Idler rear control bar	24	Stand assembly
5	Idler front control bar	25	Stand control lever
6	Throttle control bar	26	Transmission assembly
7	Adjusting bar	27	Counterweight att.
8	Tightener	28	Axle
9	Handle bar clamp	29	Hub
10	Throttle bar guid.	30	Belt shield
11	Inspection hole cover	31	Mud guard
12	Transmission pulley	32	Shifting lever
13	Cover	33	Steering clutch rod
14	Counterweight bracket pin	34	Steering linkage
15	Clutch housing	35	Clutch control bar
16	Implement hitch	36	Clutch control clamp
17	Hitch pin	37	Steering clutch lever
18	Idler linkage	38	Hand hole cover plate
19	Support assembly	39	Steering linkage pin
20	Idler pulley	40	Harrow attachment

Fig.14 SCHEMATIC DIAGRAM OF IRRI TYPE NATIONAL POWER TILLER

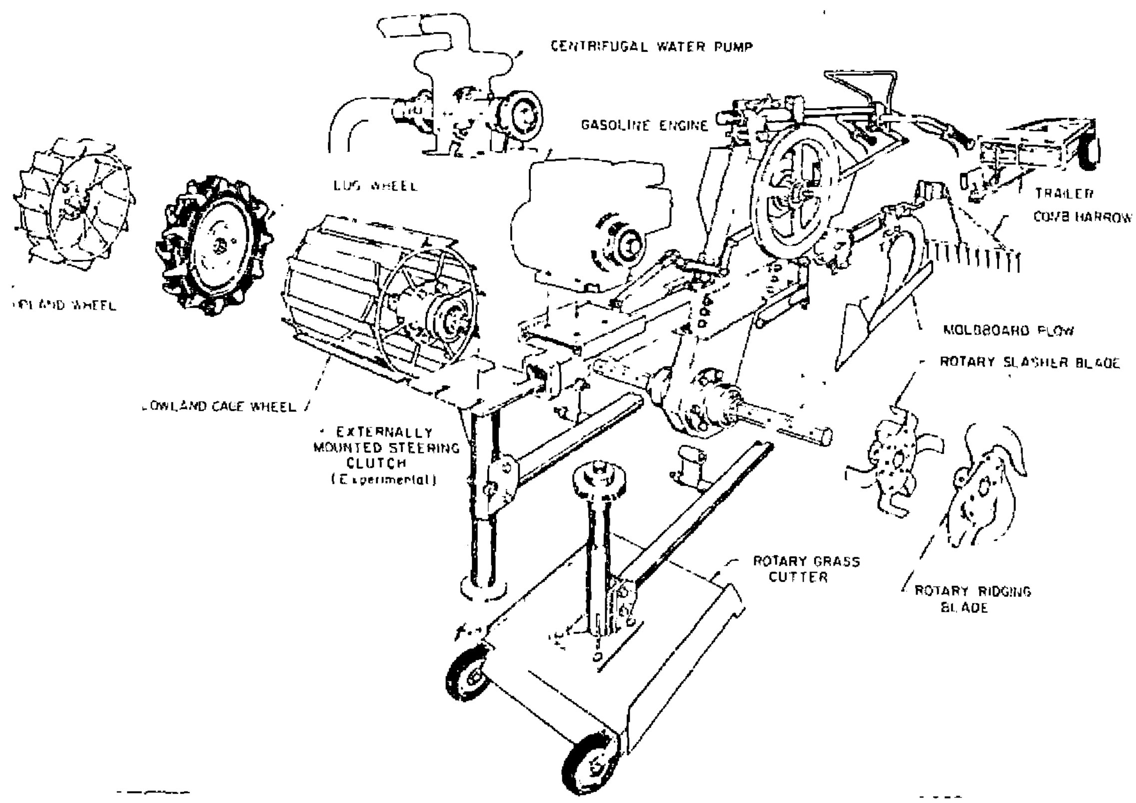
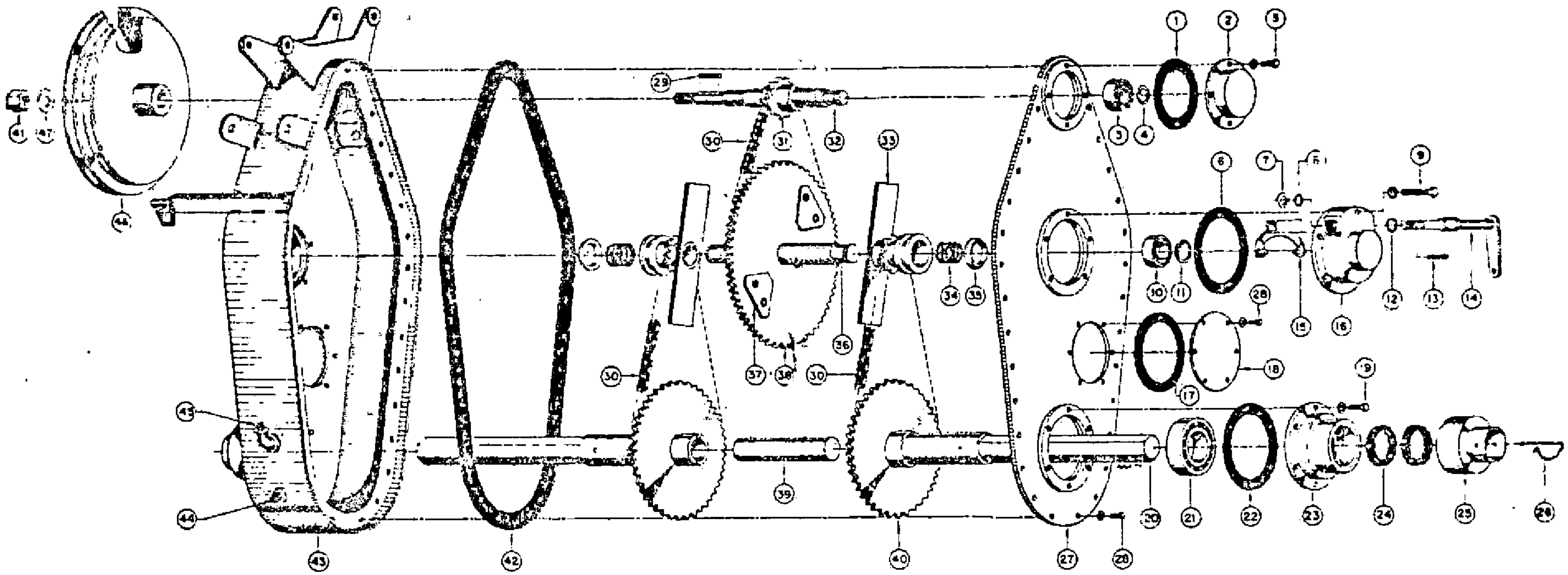


Fig.15 TEN ATTACHMENTS WITH IRRI 5-7 hp POWER TILLER (IRRI, 1973)



REF.	QTY/UNIT	DESCRIPTION
1	2	Gasket, input shaft
2	2	Cover
3	2	Bearing, input shaft # 6204
4	1	External retainer ring
5	18	Bolt, hexagonal head w/ lock washer
6	2	Gasket, steering clutch shaft
7	2	Lock washer
8	2	'O' ring
9	2	Bolt, hexagonal head w/ lock washer
10	2	Bearing, steering clutch shaft # 6204
11	2	External retainer ring
12	2	'O' ring
13	2	Cotter pin
14	2	Shifting lever
15	2	Fork
16	2	Clutch housing
17	2	Gasket, hand hole cover plate
18	2	Hand hole cover plate
19	12	Bolt, hexagonal head w/ lock washer
20	2	Axle
21	2	Bearing, final drive # 6209
22	2	Gasket, final drive
23	2	Hub
24	4	Oil seal
25	2	Mudguard
26	2	Lock, pin
27	1	Transmission housing cover
28	36	Bolt, hexagonal head w/ lock washer
29	1	Key, input shaft
30	3	Roller chain, RC 50, 5/8 Pitch, 66 links
31	1	Input drive sprocket
32	1	Input shaft
33	2	Clutch bar sub-assembly
34	2	Steering clutch spring
35	2	Retainer
36	1	Steering clutch shaft
37	4	Clutch plate
38	1	Intermediate drive sprocket
39	1	Pin
40	2	Final drive sprocket
41	1	Nut, castle
42	1	Gasket, transmission housing
43	1	Transmission housing
44	1	Plug, oil drain
45	1	Plug, oil check
46	1	Transmission pulley
47	1	Plain washer, wide series

Fig. 16 EXPLODED VIEW OF IRRI TYPE POWER TILLER TRANSMISSION PARTS

Oroino and Duff (1973) informed that small tractors have the advantage of more mobility and higher level of field efficiency than large units. Bhole and Tiwari (1977) found that a power tiller of nominal output 5 hp is able to transmit a maximum of 3 hp to the axle.

Reed (1966) reported that oversizing the tyre, increased the load carrying capacity and had indicated it as an aid to increase traction. Taylor et al. (1967) stated that at the same normal load and inflation pressure, an increase in tyre diameter, increased both pull and efficiency of traction for pneumatic tyres. Domier (1978) observed that increase in diameter by 10 and 18 per cent resulted in decrease in travel reduction by 0.6 to 1.1 per cent and increase in tractive efficiency by 1.0 to 1.6 per cent. Ali and McKyer (1978) concluded that tyre lug having a length of 250 mm or less offered better tractive force per unit width when the lug depth was 50 mm. Narang and Ram (1980) compared the performance of steel wheel, solid rubber wheel and pneumatic wheel under various loads on various surfaces and reported that the solid rubber wheel and pneumatic wheel had less towing force and performed better on the roads comparing the steel wheel. Manian (1980) recommended four ply 6.00 x 12 size pneumatic tyres for 5.4 hp garden tractor because it performed better comparing 5.00 x 14 and 4.00 x 12 size tyres.

The single axle garden tractors when used to develop drawbar pull exhibit a weight transfer to the rear. Hence a tail wheel or a front counterweight is used to counteract it. But a downward force at handle is to be given when there is no load at its drawbar (IRRI, 1978).

2.5. Ergonomic Studies

Ehrlich (1958) reported that man is acclimatized to vibration with a frequency of the order of 1.7 cycles per second with a 75 mm amplitude and the design of the vehicle should be within these limits.

Moens et al. (1974) made a somotometric analysis on position, range, movement of handles and controls and reported that the tolerance pulse rate of two wheel tractor for 8 hours duration should not exceed 30 pulses per minute above the normal pulse rate.

2.6. Economics of Garden Tractor

Khan (1970) reported that the simplicity, lightness and increase in yield due to timliness of land preparation were the major factors behind the selection of single axle garden tractors in the Philippines.

A well designed garden tractor, with proper implements and attachments can effectively be used for as many operations as possible including land levelling, seeding, planting, weeding, fertilizing, spraying, ridging, pumping and threshing. A transport cart can be hitched to the tractor to form a four wheel vehicle for transport purpose. Thus it increased the internal rate on investment upto 45 per cent comparing the larger units (Hamid, 1973).

Manian (1980) found that 5.4 hp garden tractor has the breakeven point of 500 hours as against 800 to 1130 hours for other available garden tractors and hence the ownership of these type of garden tractors will be profitable even to the small farmers.

Materials and Methods

MATERIALS AND METHODS

The design details and selection of individual components of the garden tractor are presented in this chapter.

The functional requirements of the low cost garden tractor is defined as follows.

a) The garden tractor with its attachments should be able to operate in the field at a forward speed range of 3.0-3.5 kmph and on the road at a forward speed range of 6.0-7.0 kmph.

b) The total weight of the unit should not be more than 150 kg so that it can be lifted by two persons.

c) The cost should be around Rs.10,000/- so that an average farmer can own a unit.

d) The number of components should be least for easy manoeuvrability and maintenance by the farmers.

e) The position of the controls and handle and the vibration level should be within the allowable ergonomic limits.

f) It should be repairable by local mechanics.

g) All the standard components should easily be available from local markets and other components can be fabricated from local workshops with readily available materials.

3.1. Prime mover

The horsepower of the prime mover should be of the most economical size. It should be of low weight, easily available, compact diesel engine with higher efficiency and the performance should match with the field requirements.

The optimum hp of the garden tractor prime mover is determined by the formula given by Chancellor (1967).

$$\text{Optimum hp} = \frac{\left[LW(C+LD)\right]^{\frac{1}{2}}}{(AK)^{\frac{1}{2}}}$$

where,

L = Land area which can efficiently use a tractor per year, 2.5 ha

W = Rated hp-hr required to operate the area, 350 hp-hr/year for South East Asia.

C = Cost of operation, primarily of driver's wage, Rs.6/hr.

D = Average penalty in cost for the delay from sowing to harvest, 0.1.

A = Annual cost coefficient, 0.2 for South East Asia.

K = Initial cost of tractor per rated hp, Rs.2000.

The optimum hp arrived by this method is 3.98. Manian (1980) using the formula given by Abrosimov, calculated the power of the prime mover of a garden tractor to pull a mould board plough at an adverse field condition as 4.54 hp, which comprises of drawbar hp to overcome soil resistances with plough and for self propulsion at a field speed of 3 kmph.

A critical review also indicated that an engine in the range of 5-6 hp is ideal for a low cost garden tractor. Increase in weight of prime mover will increase the power needed for self propulsion, other than increasing the weight of the garden tractor which in turn may affect the balancing. Therefore, a light weight engine in the range of 25-40 kg is to be preferred and hence an aircooled, 38 kg dry weight diesel engine producing a maximum of 5.4 hp at 1800 rpm (type 523, Greaves Lombardini engine) which is readily available in the market is selected.

It has a maximum torque of 2960 kg mm at a speed of 1200 rpm. The specific fuel consumption corresponding to this is 200 gm per hp-hr, (Fig.17). Also the engine is having adequate power generating potentials to meet the field demands. It is suitable to combat the stationary and mobile operations, because it is light in weight and it can be operated continuously upto an angularity of 35 degrees

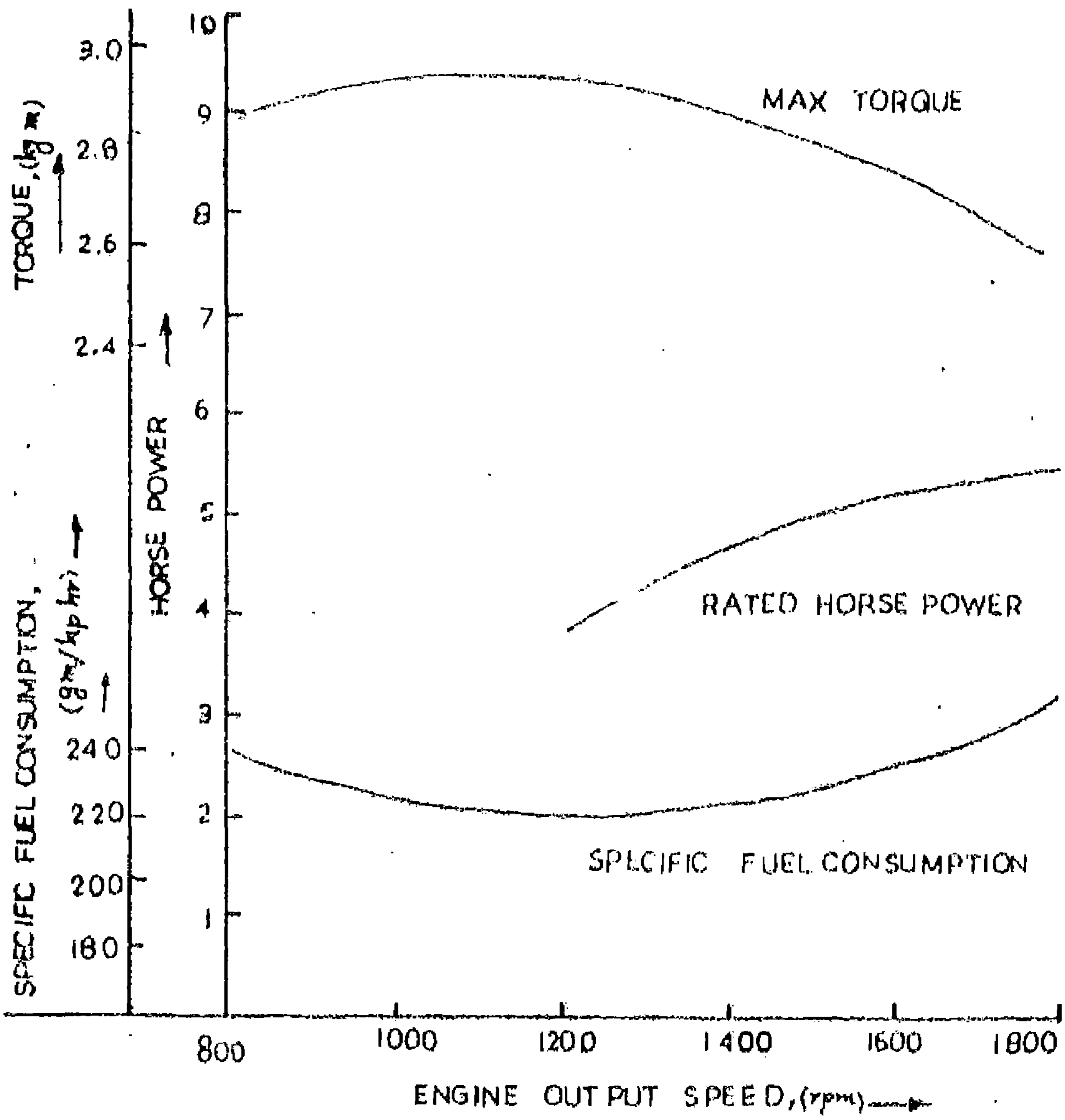


FIG. 17 CHARACTERISTICS OF TYPE 523 LOMBORDINI DIESEL ENGINE

(Appendix I) which enables the garden tractor to suit for hilly tracts also. The engine is having an added advantage of directly operating a majority of centrifugal pumps.

3.2. Power Transmission System

In order to reduce the cost, weight and power loss of the garden tractor, the gear transmission has been eliminated. A simple V belt and pulley with chain and sprocket transmission have been chosen, as these are standard components and are readily available, and moreover it is easier to repair and maintain.

The V belt transmission has been used as a pivotted countershaft clutch which also reduces overload and sudden shock. Separate steering clutch assembly is eliminated to make the unit more simpler by providing a pair of new type of overrunning clutches in the final drive.

3.2.1. Mechanical Design of Machine Members

Entire machine members of the low cost garden tractor have been designed with the help of the Design Data Book and the Agricultural Engineer's Hand Book. Correction factors and factor of safety have been taken into account while designing the machine members. The dimensions arrived at, by calculations are corrected to the next standard size for optimizing the material cost. Only minimum number of simple components which can easily be fabricated with local technology have been selected. Standard machine elements such as V pulleys, V belts, bearings, chain, sprockets and circlips are adapted to avoid any higher production cost and to aid easy replacement. While designing the shafts the bending and twisting moments as well as the tensile, compressive, shear, bending and torsional stresses are taken into account at all stages.

The proven type pneumatic tyres of 6.00 x 12 size, four ply, 25 mm lug height with 55 degree lug angle and suitable cage wheels have been directly utilized to serve as traction devices.

3.2.2. Design of Power Transmission Elements

The selected engine is having a rated speed of 1800 rpm. The field speed of garden tractor should not exceed the average walking speed of a man in the field, as the operator has to walk behind the garden tractor. Thus it is decided to allow the field speed of garden tractor upto 3.5 kmph. Corresponding to this field speed with 6.00 x 12 size pneumatic wheel, the speed of the wheel (N) is given by,

$$N = \frac{3.5 \times 1000 \times 1000}{\pi \times 600 \times 60}$$

$$= 30.947 \text{ rpm}$$

To have a speed, below 3.5 kmph, the wheel speed is taken as 30 rpm. Total speed reduction needed,

$$= \frac{1800}{30}$$

$$= 60$$

In single and double stage speed reduction the size of the pulleys or sprockets will be larger, resulting in improper weight distribution and thereby problems of vibration and balancing. Hence a three stage speed reduction is selected. The first reduction is through a belt drive and the other two reductions are by chain drives. Three reductions such as 2.75, 4.00 and 5.50 are selected for the belt, first and second stage chain drives respectively to achieve a total reduction ratio of 1:60.5.

The detailed calculations and other design aspects of V belt drive for field operation is given in the Appendix II. Hence by selecting a pair of double groove 80 mm and 220 mm diameter V pulleys and B section V belts of B-1067/42-41 specification, a speed reduction of 61.3636 is achieved. The centre distance between engine shaft and counter shaft is 311.515 mm which is illustrated in the Fig.18.

In the Appendix III, the design of V belt drive for transportation is given and hence a pair of 160 mm and 220 mm dia V pulleys and a B-1168/46-45 V belt are selected. The speed reduction achieved is 30.6818 and the centre distance

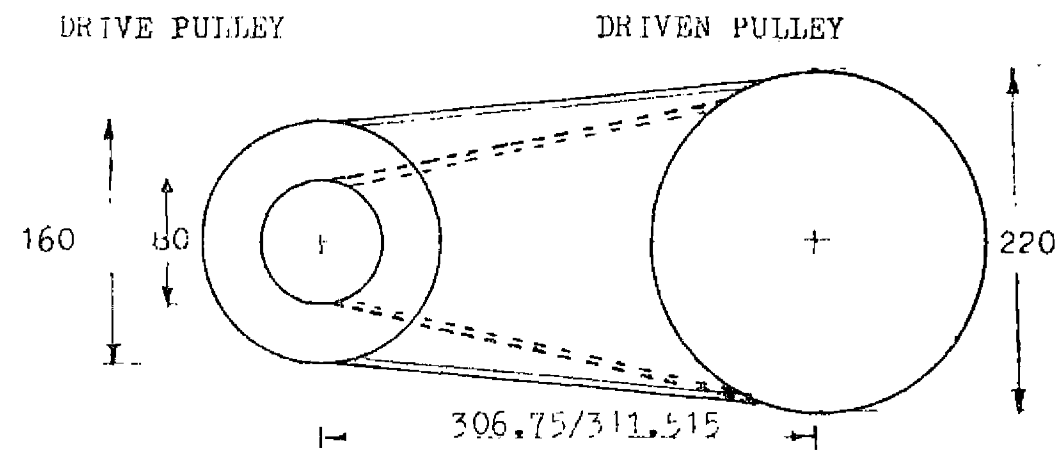
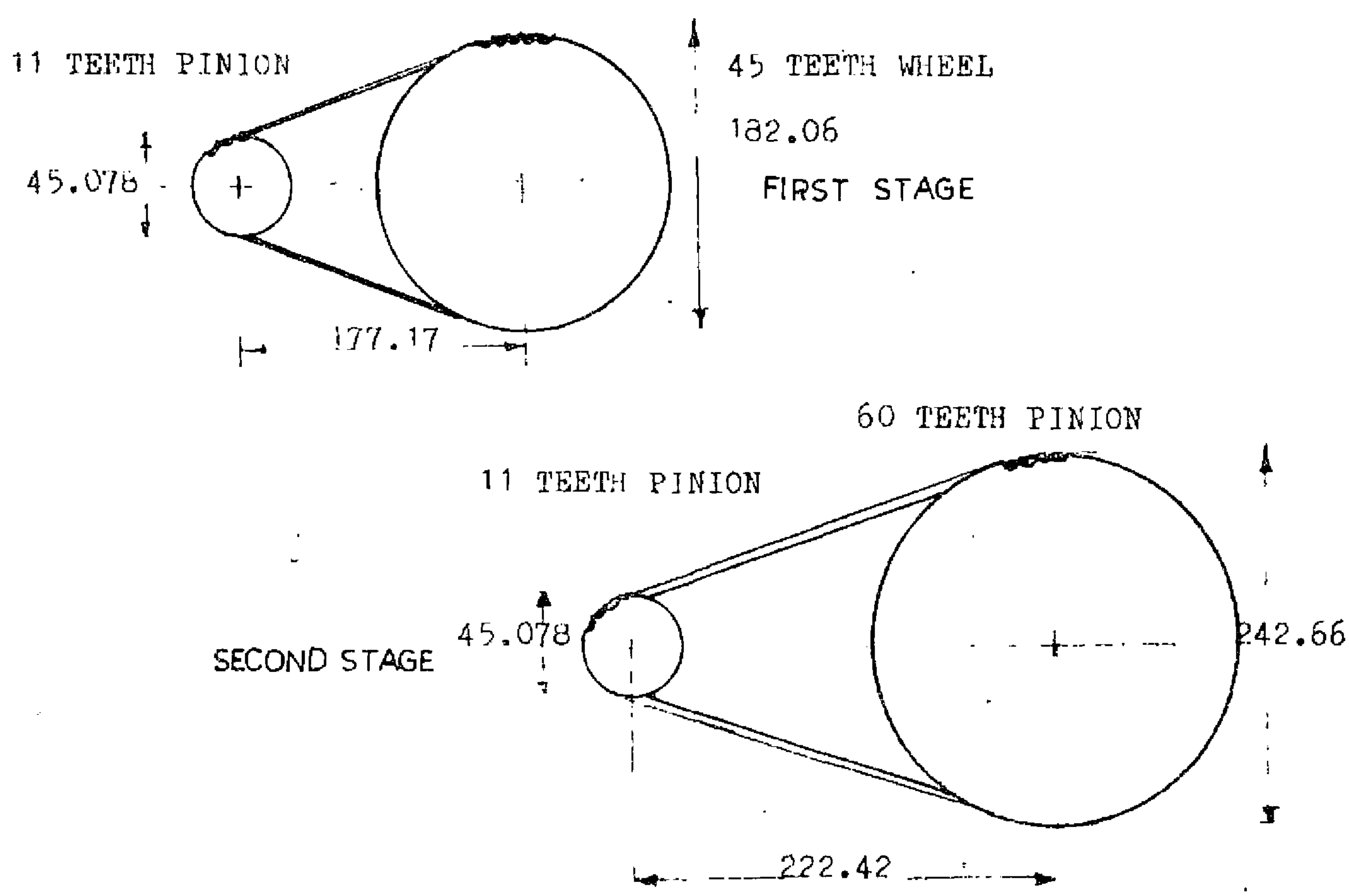


FIG. 18 DETAILS OF V BELT DRIVE



ALL DIMENSIONS IN mm. SCALE 1:5

FIG.19 DETAILS OF FIRST AND SECOND STAGE CHAIN DRIVES.

between the shafts is 306.075 mm and as illustrated in the Fig.18.

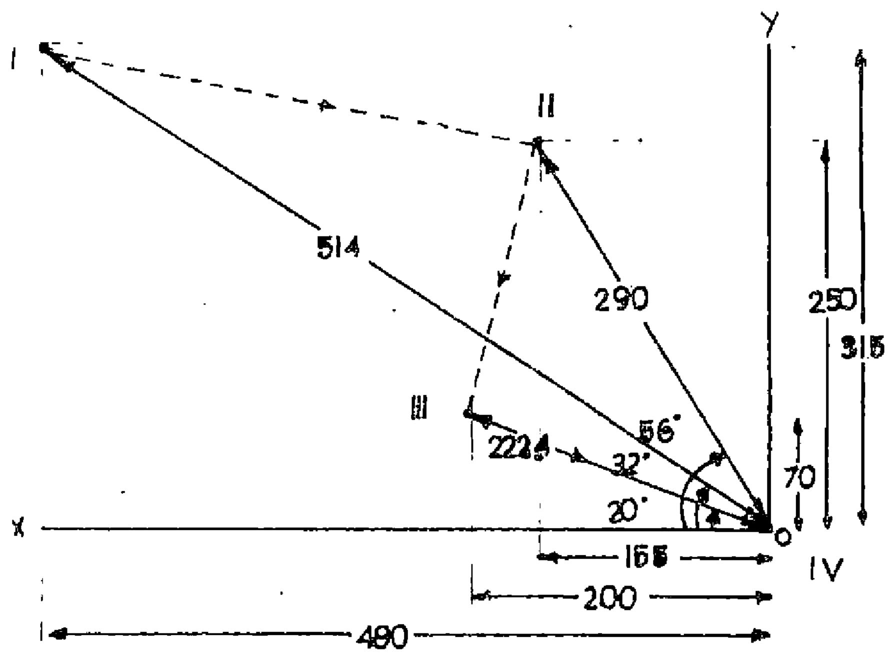
The easy changing of speed for field and road is achieved by integrating the two engine drive pulleys into a cone V pulley and by providing a pivotted countershaft clutch arrangement in the belt transmission.

The first stage chain and sprocket transmission is designed with 12.700 mm pitch 11 teeth sprocket pinion and 45 teeth sprocket wheel. The centre distance between the countershaft and intermediate shaft is 177.1715 mm (Fig.19). The detailed design procedure is given in the Appendix IV. The specifications of the selected ISO/DIN 084-1 R 1248 H chain is given in the Appendix V and the teeth specifications is given in the Appendix VI.

The second stage chain and sprocket transmission is designed on the same lines (Appendix VII) with the same type of 12.700 mm pitch, 11 teeth sprocket pinion and 60 teeth sprocket wheel with centre distance between the intermediate shaft and the final drive axle, 222.424 mm as illustrated in the Fig.19.

3.2.3. Position of Shafts

Power is transmitted through a countershaft and an intermediate shaft to the final drive axle. The three shafts



I ENGNIE SHAFT

II COUNTER SHAFT

III INTERMEDIATE SHAFT

IV FINAL DRIVE AXLE

ALL DIMENSIONS IN mm

FIG. 20 POSITION OF ENGINE SHAFT, COUNTER SHAFT, INTERMEDIATE SHAFT AND FINAL DRIVE AXLE (VIEW FROM SHAFT END) are arranged as in the Fig.20, so that the weight is uniformly distributed on the chasis and also to bring down the centre of gravity close to the final drive axle for avoiding any problem in balancing. The schematic diagram of power transmission is shown in the Fig.21.

3.2.4. Design of Countershaft

Fig.22 shows the various loads coming on the countershaft.

The load on the countershaft due to belt drive acting at C, at an inclination of 13 deg to horizontal,

$$Q_{oc} = \frac{2 \times 4500 \times P}{n\ dp\ \pi}$$

$$= \frac{2 \times 4500 \times 5.4 \times 1000}{654.545 \times 220 \times \pi}$$

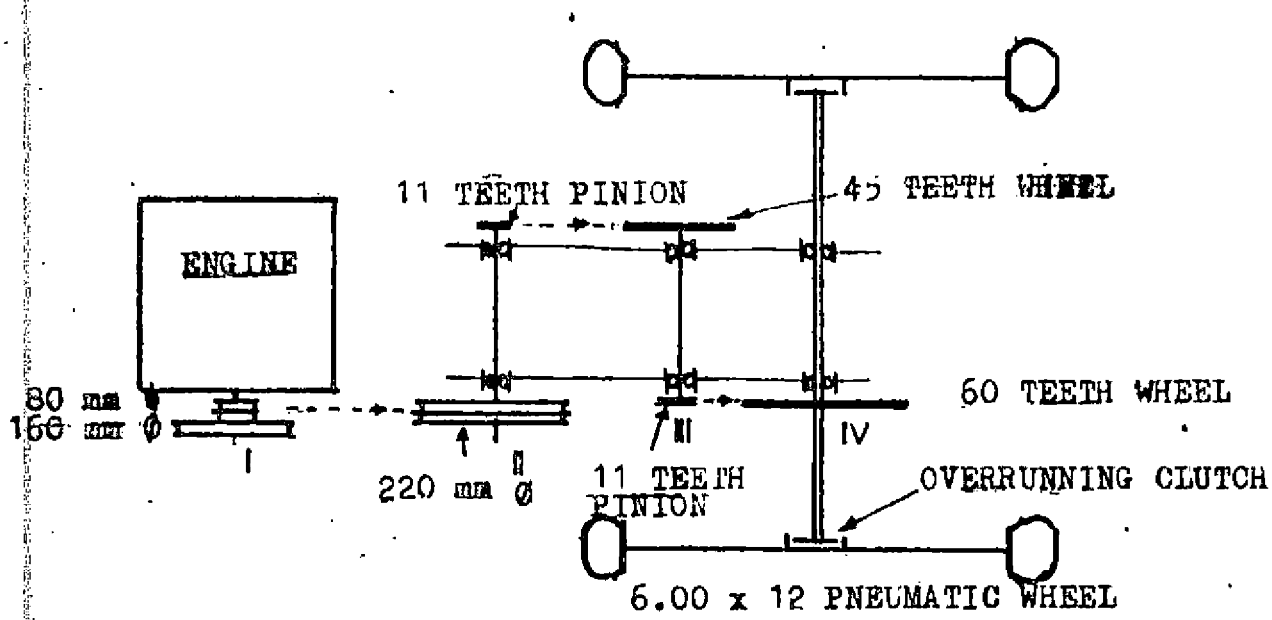
$$= 107.429 \text{ kgf}$$

The load on the shaft due to chain drive acting at D, at an inclination of 84 deg to the horizontal,

$$Q_{od} = \frac{1.15 \times 4500 \times 5.4 \times 1000}{654.545 \times \underset{160}{\cancel{220}} \times \pi}$$

$$= 301.997 \text{ kgf}$$

The detailed design calculations of vertical force, vertical bending moment, horizontal force, horizontal bending moment, resultant bending moment, twisting moment and equivalent bending moment acting on the countershaft



SHAFT	FIELD OPERATION		ROAD OPERATION	
	SPEED (rpm)	SPEED REDUCTION	SPEED (rpm)	SPEED REDUCTION
I ENGINE SHAFT	1800		1800	
		1 : 2.75		1 : 1.375
II COUNTERSHAFT	654.54		1309.09	
		1 : 4.09		1 : 4.09
III INTERMEDIATE SHAFT	160.00		320.00	
		1 : 5.45		1 : 5.45
IV FINAL DRIVE AXLE	29.33		58.66	

FIG.21 SCHEMATIC DIAGRAM OF POWER TRANSMISSION OF GARDEN TRACTOR

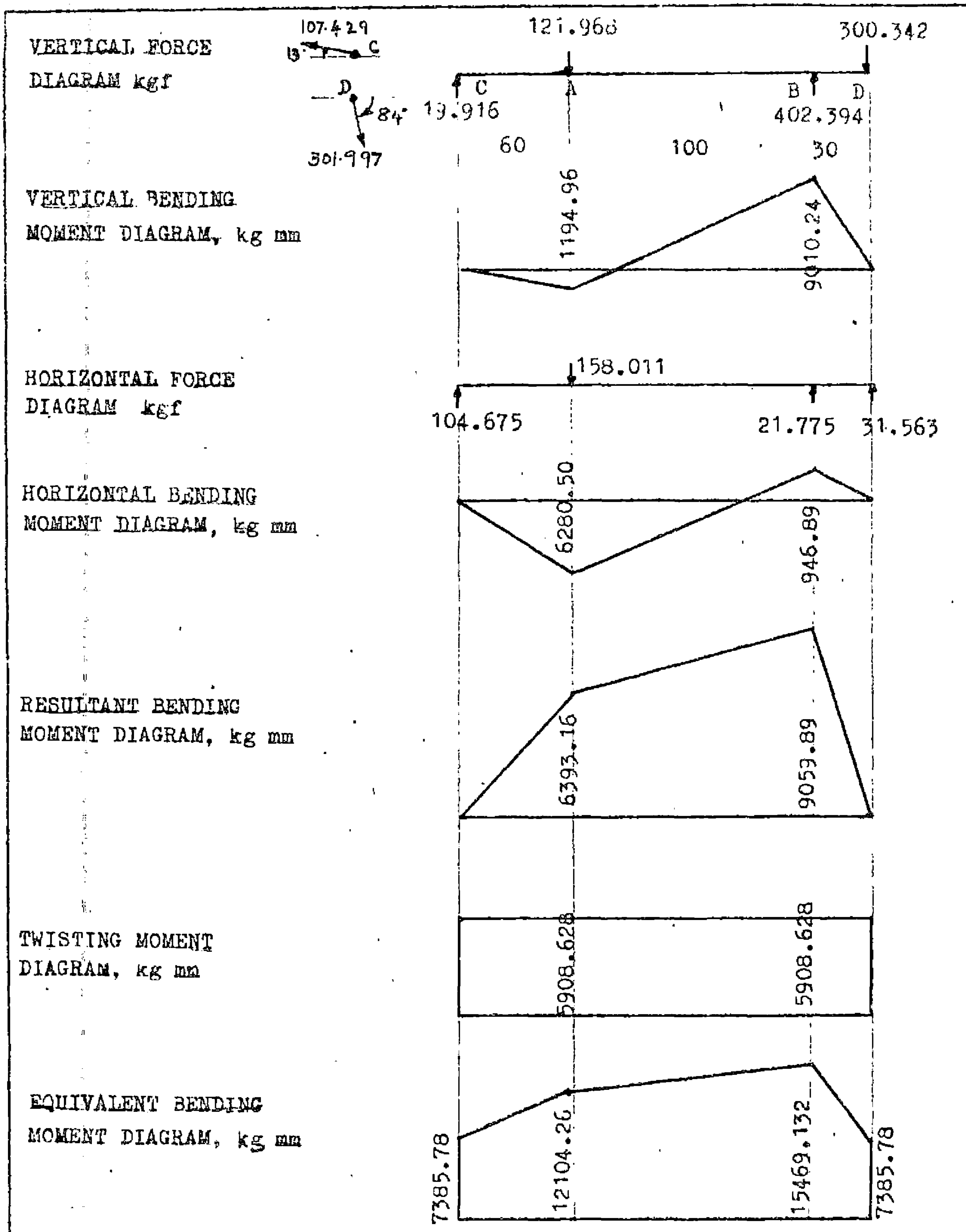


FIG. 22 FORCE AND BENDING MOMENT DISTRIBUTION ON COUNTERSHAFT

are given in the Appendix VIII and the values are illustrated in the Fig.22.

The driven pulley and the sprocket pinion are fixed at points C and D on the shaft and the bearings are fixed at points A and B to support the countershaft with the intermediate shaft.

3.2.4.1. Selection of Shaft Diameter

For C 40 steel, minimum diameter at points C and D for repeated load in shear,

$$d_1 = \left[\frac{16\ M_e}{\pi\ \tau}\right]^{\frac{1}{3}}$$

$$= \left(\frac{16 \times 7385.787}{\pi \times 6.5}\right)^{\frac{1}{3}}$$

$$= 17.947 \text{ mm}$$

Hence a diameter of 20 mm is selected.

Diameter at points A and B,

$$d_2 = \left(\frac{16 \times 12104.265}{\pi \times 6.5}\right)^{\frac{1}{3}}$$

$$= 21.160 \text{ mm}$$

Hence a diameter of 28 mm is selected to suit the bearings selected under 3.2.4.2.

3.2.4.2. Selection of Bearings

Radial load at point A,

$$F_{ra} = (F_{va}^2 + F_{ha}^2)^{\frac{1}{2}}$$
$$= (121.968^2 + 158.011^2)^{\frac{1}{2}}$$
$$= 199.608 \text{ kgf}$$

The radial load at point B,

$$F_{rb} = (402.394^2 + 21.773^2)^{\frac{1}{2}}$$
$$= 402.983 \text{ kgf}$$

Since there is no axial load at the countershaft the ball bearings of 62 series is selected.

Equivalent load at bearings,

$$F_e = (XF_r + YF_a)S$$

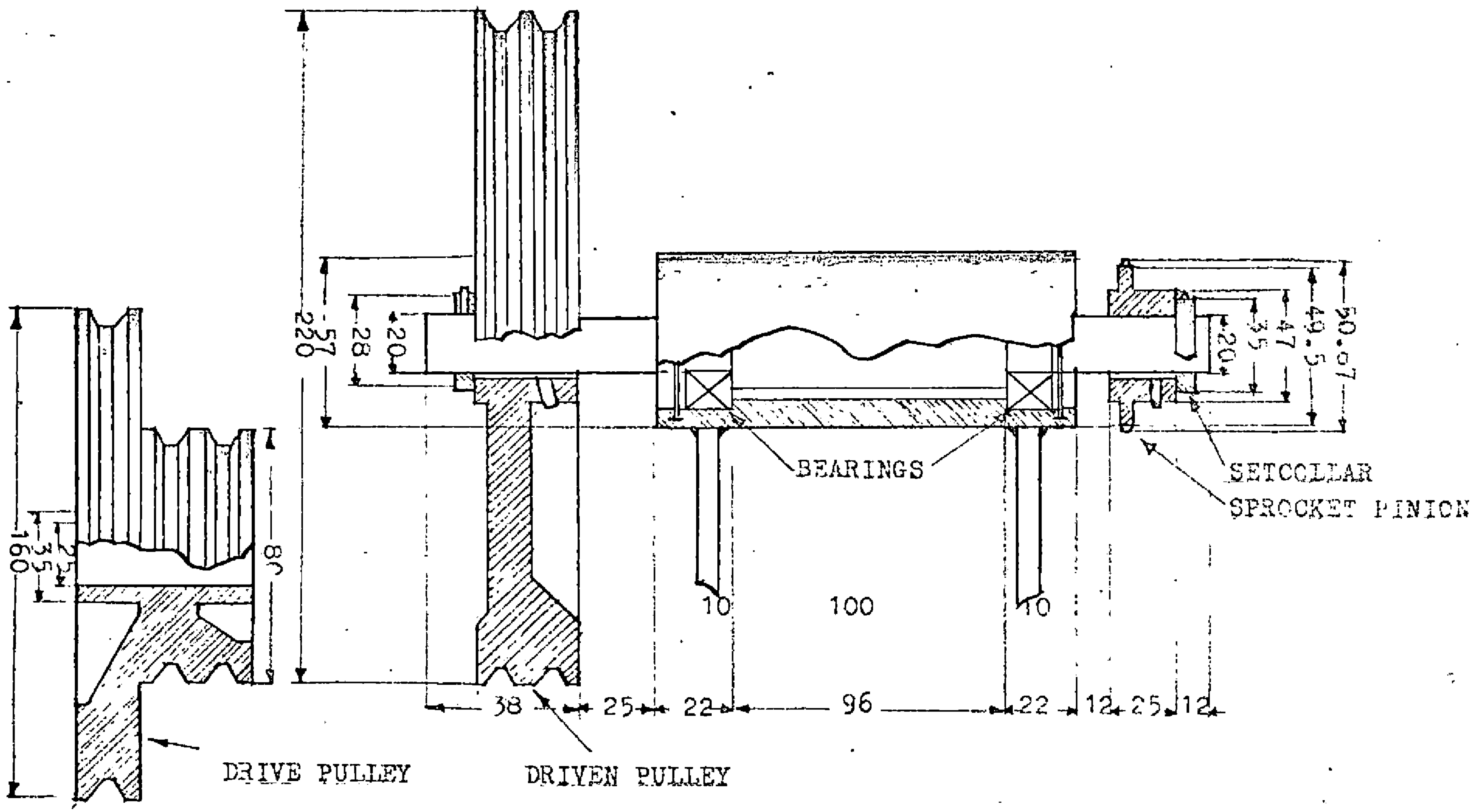
where the values of X, Y and S at points A and B are 1, 0 and 1.1 respectively. Substituting these values the equivalent load (F_e) at point A,

$$F_{ea} = 219.569 \text{ kgf}$$

and at point B,

$$F_{eb} = 443.281 \text{ kgf}$$

Hence SKF 6204 2Z ball bearings are selected (The specifications of the ball bearings are given in the Appendix XI). For retaining these self sealed bearings, a pair of



SA 01 DETAILS OF COUNTER SHAFT

ALL DIMENSIONS IN mm

SCALE 1:2

light B, 47 IS:3075-1965 internal circlips (Appendix XII) are used in the casing. The detail of assembly is given in the Drawing SA 01. Standard light series set collars (Appendix XIII) along with slotted head grub screws and standard keys are used for fastening the driven pulley and the sprocket pinion with the countershaft.

3.2.5. Design of Intermediate Shaft.

Fig.23 shows the various loads acting on the intermediate shaft. The load acting at point C on intermediate shaft due to chain drive at an inclination of 20 deg to the horizontal,

$$Q_{oc} = \frac{1.3 \times 4500 \times 5.4 \times 1000}{\pi \times 160 \times 45}$$

$$= 1396.585 \text{ kgf}$$

The load on the shaft at point D due to chain drive acting at 84 deg to horizontal,

$$Q_{od} = \frac{1.15 \times 4500 \times 5.4 \times 1000}{\pi \times 160 \times 182}$$

$$= 305.466 \text{ kgf}$$

The detailed calculation of vertical force, vertical bending moment, horizontal force, horizontal bending moment, resultant bending moment, twisting moment and equivalent bending moment acting on the intermediate shaft are given in the Appendix IX and in the Fig.23.

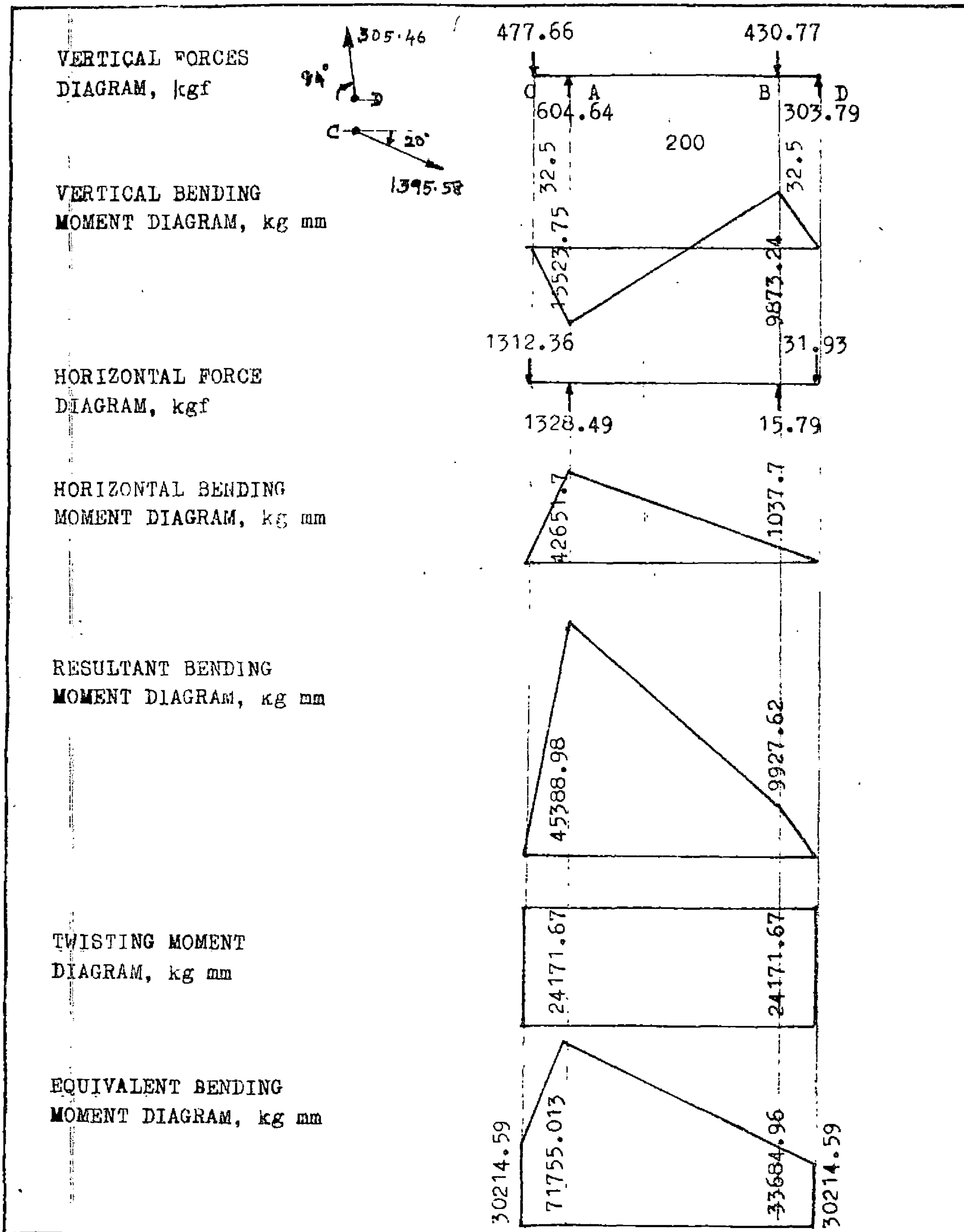


FIG. 23 FORCE AND BENDING MOMENT DISTRIBUTION ON INTERMEDIATE SHAFT

At points A and B bearings and at points C and D sprocket pinion and 45 teeth sprocket wheel are respectively fixed.

3.2.5.1. Selection of Shaft Diameter

For axle steel, minimum diameter at points C and D,

$$d_1 = \left(\frac{16 \times 30214.5914}{\pi \times 25.0}\right)^{\frac{1}{3}}$$

$$= 18.320 \text{ mm}$$

Hence a diameter of 20 mm is selected.

The minimum diameter at points A and B,

$$d_2 = \left(\frac{16 \times 71755.013}{\pi \times 25.0}\right)^{\frac{1}{3}}$$

$$= 24.442 \text{ mm}$$

Hence a diameter of 30 mm is selected as it should suit the bearings selected under 3.2.5.2.

3.2.5.2. Selection of Bearings

Radial load at point A,

$$F_{ra} = (604.646^2 + 1328.497^2)^{\frac{1}{2}}$$

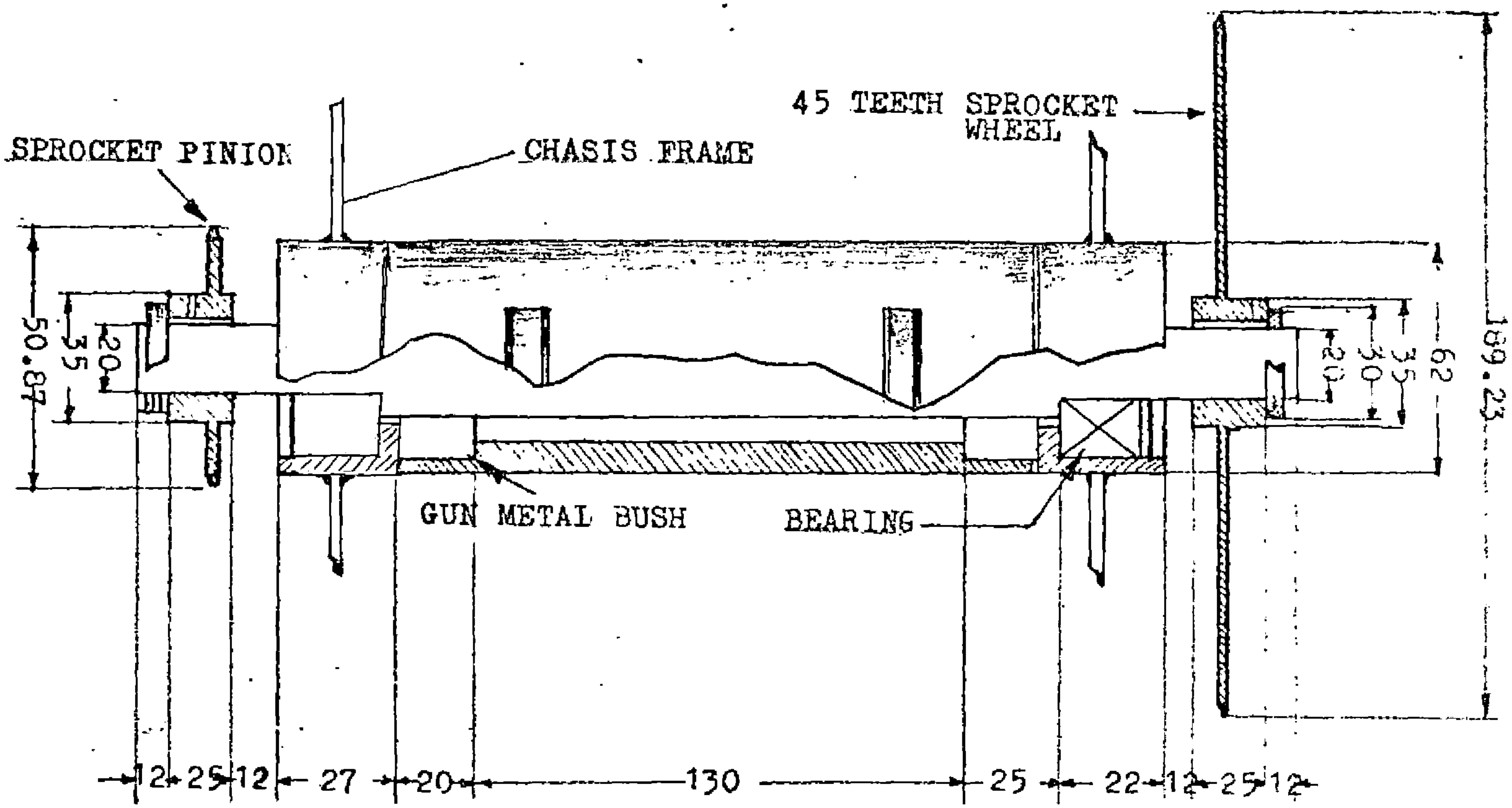
$$= 1459.623 \text{ kgf}$$

Radial load at point B,

$$F_{rb} = (430.778^2 + 15.793^2)^{\frac{1}{2}}$$

$$= 431.067 \text{ kgf}$$

As there is no axial load at the intermediate shaft, the ball bearings of 33 series are selected.



SH . 02 DETAILS OF INTERMEDIATE SHAFT

ALL DIMENSIONS IN mm
SCALE 1 : 2

Equivalent load at points A and B are calculated as 1605.585 kgf and 474.174 kgf respectively and hence SKF 3304 2Z ball bearings are selected (Appendix XI). In order to retain the self sealed bearings, one pair of light B, 52 IS:3075 - 1965 internal circlips (Appendix XII) are used in the bearing housings. Two numbers of standard gunmetal bushes with inner diameter 30 mm, outer diameter 42 mm and width 20 mm are used in between the intermediate shaft and its casing for functioning as a pivotted countershaft clutch Suitable standard set collars with slotted head grub screws (Appendix XIII) are provided along with keys as shown in the Drawing SA 02 for fixing sprocket pinion and 45 teeth sprocket wheel with the intermediate shaft.

3.2.6. Design of Final Drive Axle

The details of loads on the final drive axle are shown in the Fig.24.

Introduction of a pair of overrunning clutch in the final drive axle eliminated the need of split shaft or hexagonal shaft and thus it reduced the production cost. An ordinary shaft with 680 mm length is selected to have a track width of 780 mm for easy ploughing operations along with the previous furrow. A higher track width than the axle length is achieved due to the offset wheel discs.

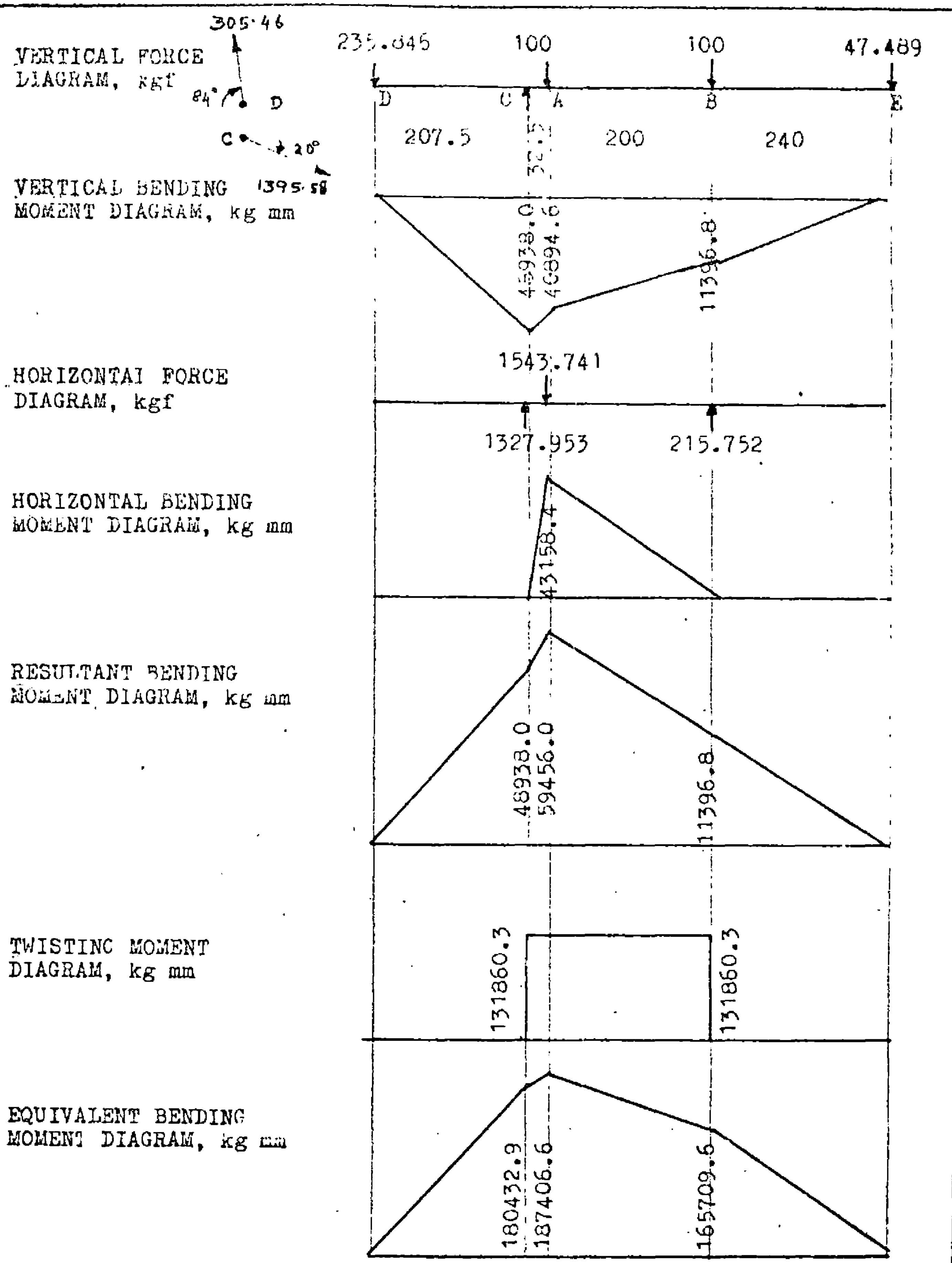


FIG. 24 FORCE AND BENDING MOMENT DISTRIBUTION ON FINAL DRIVE AXLE

Load acting on the axle due to chain drive at point C at an inclination of 20 deg to the horizontal,

$$Q_{oc} = \frac{1.3 \times 4500 \times 5.4 \times 1000}{29.33 \times \pi \times 242.6}$$

$$= 1413.178 \text{ kgf}$$

The final drive axle experiences the maximum load when it is used for ploughing operation rather than hauling or pumping.

The maximum vertical loads at the points A and B, for a maximum pull of 100 kgf acting at the drawbar at an inclination of 38 deg to the horizontal along with the additional load at handle frame is taken as 100 kgf each.

The detailed design calculations of vertical force, vertical bending moment, horizontal force, horizontal bending moment, twisting moment and equivalent bending moment acting on the final drive axle are given in the Appendix X and the values are given in the Fig.24.

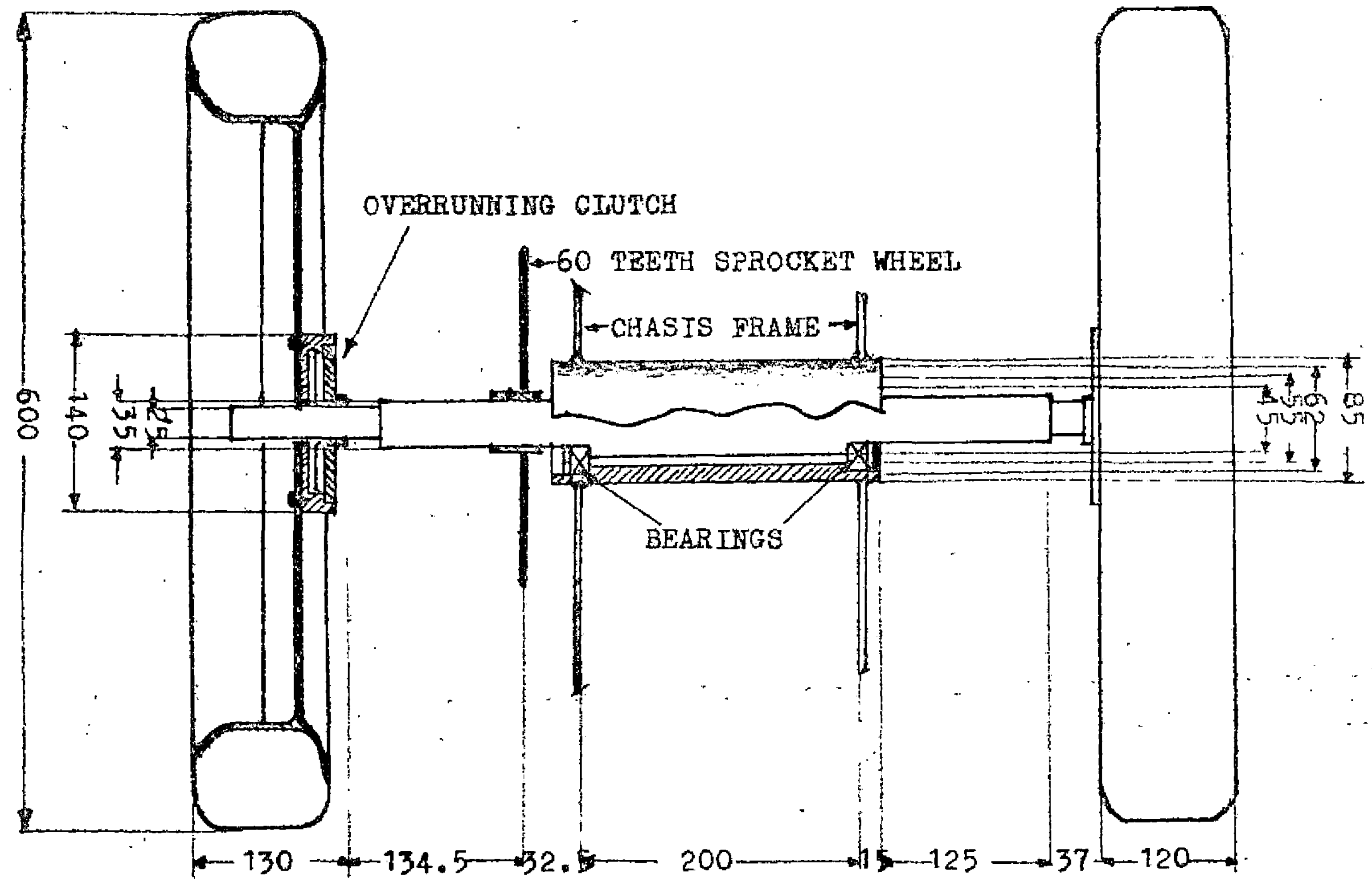
3.2.6.1. Selection of Axle Diameter

In order to accommodate the overrunning clutch a shaft end diameter of 25 mm with axle steel is selected.

Minimum diameter at points A and C where the maximum equivalent bending moment is acting,

$$d_1 = \left(\frac{16 \times 187406.634}{\pi \times 25}\right)^{\frac{1}{3}}$$

$$= 33.671 \text{ mm}$$



SA 03 DETAILS OF FINAL DRIVE AXLE

ALL DIMENSIONS IN mm

SCALE 1:5

Hence a diameter of 35 mm is selected.

3.2.6.2. Selection of Bearings

Radial load at point A,

$$F_{ra} = (100.000^2 + 1534.741^2)^{\frac{1}{2}}$$
$$= 1538.000 \text{ kgf}$$

Radial load at point B,

$$F_{rb} = (100.000^2 + 215.792^2)^{\frac{1}{2}}$$
$$= 237.836 \text{ kgf}$$

The ball bearings will take 25 per cent of additional side thrust and hence it has been decided to use the ball bearings from 62 series. Equivalent load at bearings at points A and B are calculated as 1691.800 kgf and 261.620 kgf respectively and hence SKF 6207 2Z ball bearings (Appendix XI) are selected. A pair of light B-72 IS:3075-1965 internal retainer circlips (Appendix XII) and light series set collars with slotted head grub screws (Appendix XIII) are used for fixing the 60 teeth sprocket wheel and overrunning clutches respectively which is illustrated in the Drawing SA 03.

3.2.7. Design of Overrunning Clutch

The overrunning clutch is a very simple and compact device, which can provide the differential effect by free wheeling action at turnings and to fasten the wheel rim with the final drive shaft. The overrunning clutch which is also known as oneway clutch or free-wheel clutch will transmit power between two in-line shafts as long as the two shafts revolve at the same speed in the same direction. When the driving shaft lags in speed or reverses it cannot transmit power to the driven shaft. Under these conditions, this unidirectional clutch 'uncouples' the two shafts and the driven shaft overruns. The force applied to the pawls of the driving shaft push the driven shaft through ratchets and thus the power is directly transmitted to the driven shaft, (Greenwood, 1959; Stone and Gulvin, 1967; Nash, 1979). Hence a pair of simple and more compact steering device which can be operated smoothly without separate steering clutches, controls or differential gears is designed and developed, (Drawing SA 04). It was then evaluated for its performance and endurance.

The maximum torque of 2960 kg mm is obtained when the prime mover runs at 1200 rpm. Hence the maximum torque transmitted by the final drive axle at the speed of 20 rpm,

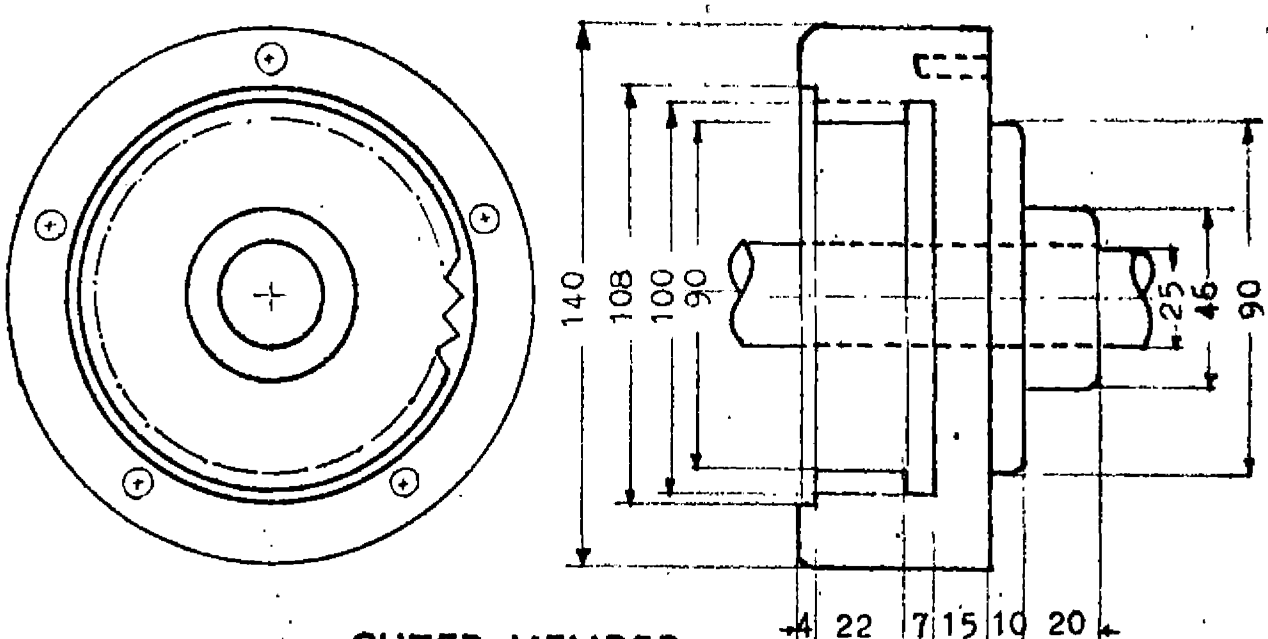
$$= \frac{2960 \times 1200}{20}$$

$$= 177600 \text{ kg mm}$$

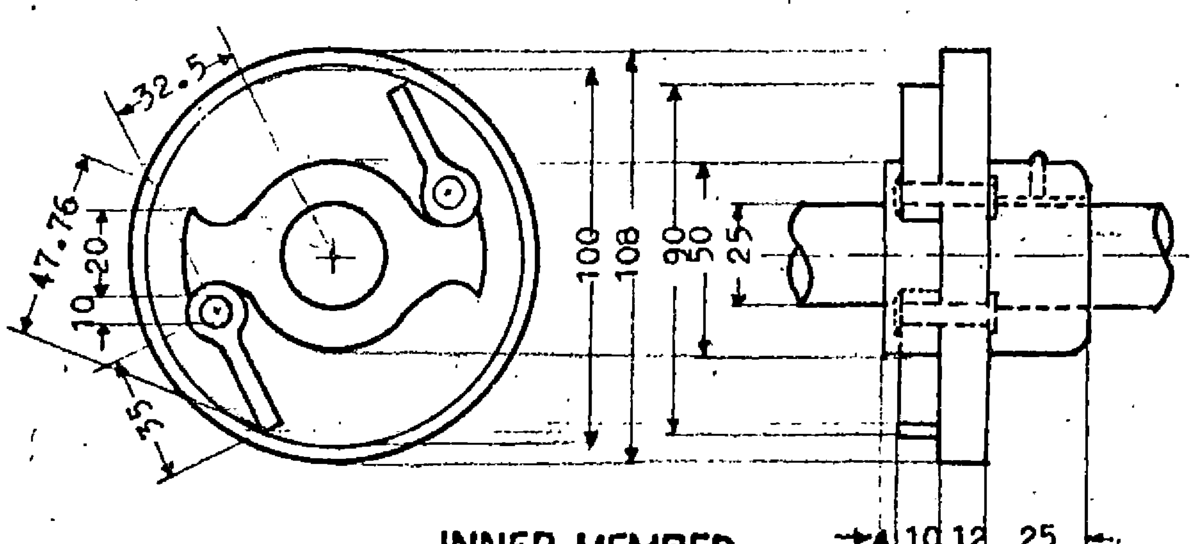
Then the torque transmitted by each side,

$$= 88800 \text{ kg mm.}$$

At each side, inside the inner member of the over-running clutch, two pawls or dogs are provided to transmit



OUTER MEMBER



INNER MEMBER

SA 04 DETAILS OF OVERRUNNING CLUTCH

ALL DIMENSIONS IN mm

SCALE 1:2

the torque to the ratchets of the outer member, which is directly connected to the wheel rim. The two pawls are pivotted by two pawl pins located 32.5 mm from the centre in opposite directions. Let the length of the pawl be 35 mm, hence it transmits the load in the direction perpendicular to the radial line joining the centre of the axle to the pawl pin at 47.76 mm from the centre of the axle. Hence the load transmitted by each of the pawl,

$$= \frac{88800}{47.76 \times 2 \times 2}$$

$$= 464.82 \text{ kgf}$$

For C 14 steel, the minimum cross section of the pawl in bending,

$$= \frac{464.82}{6.00}$$

$$= 77.47 \text{ mm}^2$$

The compression load on the pawl,

$$= \frac{88800}{37.5 \times 2 \times 2}$$

$$= 592.00 \text{ kgf}$$

Hence the minimum cross section of the pawl in compression,

$$= \frac{592.00}{6.00}$$

$$= 98.66 \text{ mm}^2$$

Hence an area of cross section of 100 mm^2 is selected for the pawl. The entire load is transmitted from the axle to the pawls through the semicircular projection of the inner member, and the pawlpin is not taking any load. Even then the pawl pin is checked for failure in shear at its fixed end. The minimum diameter of the pawl pin for C 14 steel,

$$= \left(\frac{592.00 \times 4}{8.0 \times \pi}\right)^{\frac{1}{2}}$$

$$= 9.71 \text{ mm}$$

Hence a 10 mm dia is selected. A standard key with 8 mm x 7 mm x 36 mm size and standard grub screws of dia 6.5 mm are used for the overrunning clutch.

The pawls of the inner member directly transmit the power to the ratchets of the outer member. Hence a minimum 120 mm^2 contact area for the ratchet face is allowed. For easy fabrication of the outer member a regular type 24 teeth ratchet is designed (Drawing SA 04).

3.2.7.1. Design of Wheel Bolts

Through five bolts the outer members of the overrunning clutch are directly fitted with the wheel rims. The minimum diameter of the bolts is checked for failure in shear.

Torque transmitted

$$= 177600 \text{ kg mm}$$

Shear force acting at bolts,

$$= \frac{177600}{65}$$

$$= 2732.31 \text{ kgf}$$

Hence the minimum diameter for C 15 steel bolts,

$$= \left(\frac{2732.31 \times 4}{6.0 \times 5 \times \pi}\right)^{\frac{1}{2}}$$

$$= 10.76 \text{ mm}$$

Hence 15 mm dia wheel bolts are selected to suit the wheel disc.

3.2.8. Design of Pivotted Countershaft Clutch

Pivotted or swing type countershaft clutch consists of a link rod connecting the countershaft casing and hinged hand lever. The countershaft can swing over the intermediate shaft as the casings of both the shafts are linked by two members of 30 mm x 10 mm flats as in the Drawings SA 01, SA 02 and SA 05.

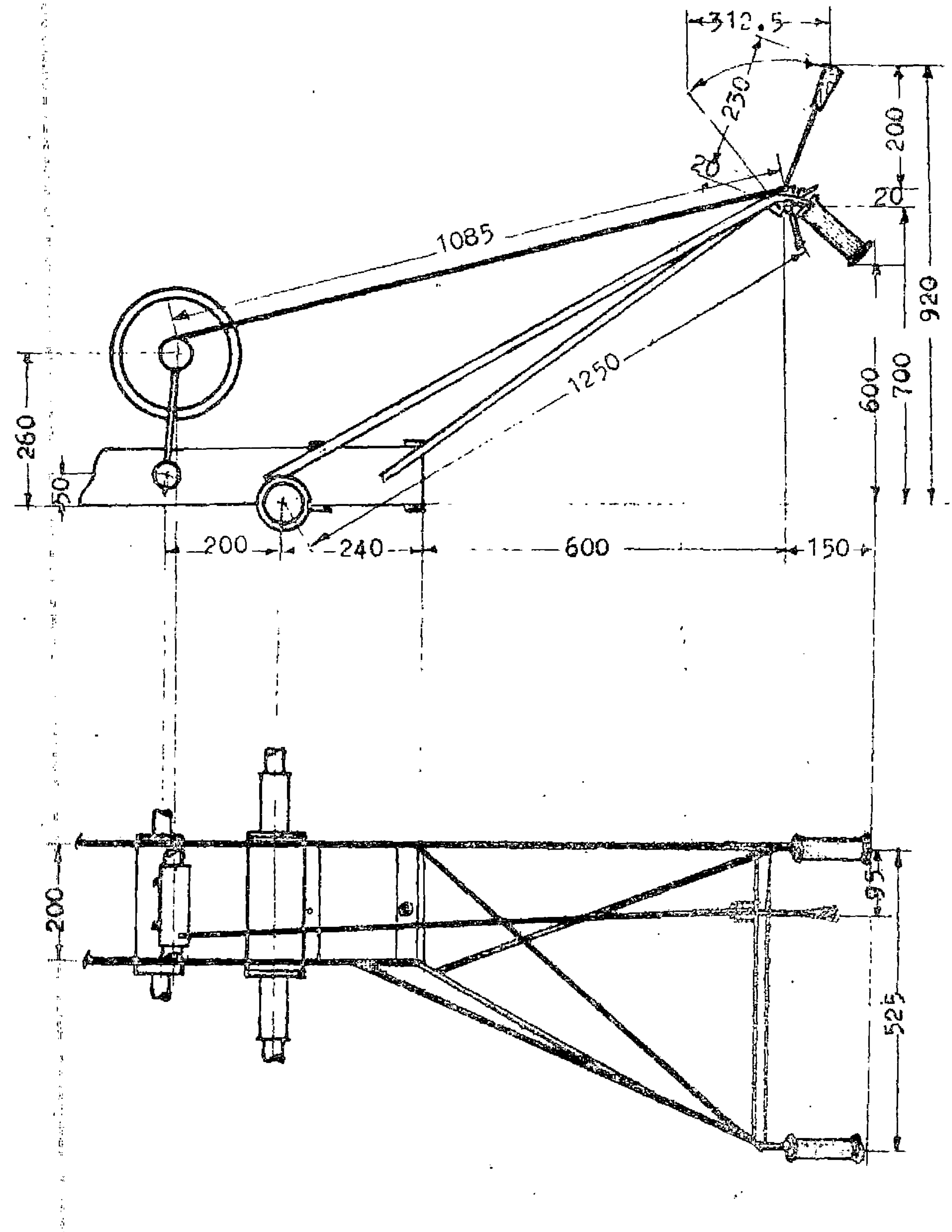
It is calculated in detail under Appendix VIII, that a horizontal force of 104.675 kgf acting on the countershaft due to belt drive. This force should be counteracted by the operator through the hand lever. The clutch links are fabricated as in the drawing SA 05.

The load at the hand lever grip,

$$= \frac{104.675 \times 20}{250}$$

$$= 8.37 \text{ kgf}$$

which is within the allowable maximum limit of 8.9 kgf for the hand operated lever (Mathews and Knight, 1971). It is



SA 05 SWING OR PIVOTTED COUNTER SHAFT CLUTCH AND HANDLE

ALL DIMENSIONS IN mm

SCALE 1 : 10

decided to have a minimum of 25 mm swing at the driven pulley centre for effective disengagement of power. Hence the swing needed at the handle,

$$= \frac{25}{20} \times 250$$

$$= 312.5 \text{ mm}$$

which is also within the allowable range of hand operation, (Fig.25).

3.2.8.1. Check for Component Dimensions

Minimum diameter of the link rod, which can fail by tension for the maximum tensile load of 107.429 kgf for C 15 steel,

$$= \left(\frac{107.429 \times 4}{\pi \times 4.5}\right)^{\frac{1}{2}}$$

$$= 5.51 \text{ mm}$$

Hence a 10 mm dia rod is used and 9.15 mm dia C 15 steel rod is selected for hand lever.

The pins at countershaft, hand lever fulcrum and at link rod may fail by shear, hence the minimum diameter for C 40 steel pins,

$$= \left(\frac{107.429 \times 4}{2 \times \pi \times 6.5}\right)^{\frac{1}{2}}$$

$$= 3.243 \text{ mm}$$

Hence 5 mm dia pins are selected for all clutch linkages.

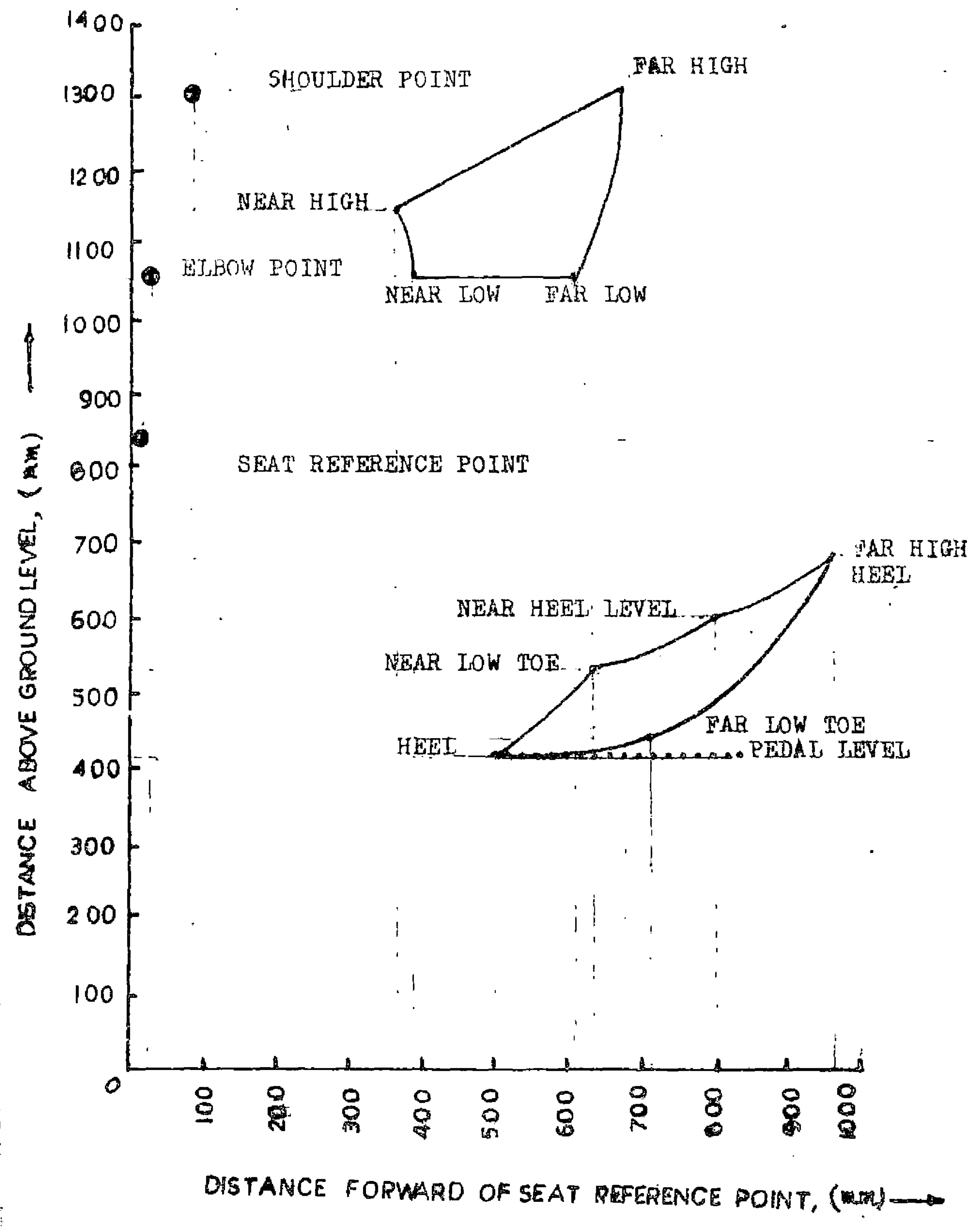


FIG. 25 OPTIMUM HAND AND PEDAL CONTROL AREA FOR GARDEN TRACTOR

A spring is employed in between the countershaft and chasis to keep the clutch always in the power engaged position and a belt retainer is also provided to prevent the belts running out of the pulleys when the power is disengaged.

3.3. Chasis

Vertical chasis is liable to unstability, particularly when mounted with different implements. Hence a horizontal chasis of rectangular shape is selected for further development to suit the prime mover and the power transmission system, to which simple front mounted attachments can also be hitched. The front mounted countershaft or tail-wheel which is common in many garden tractors is eliminated by distributing the weight of the components uniformly so that it will be balanced by itself when the engine is at front and the implement is hitched at the back.

The basic beams of the chasis are constructed by two C 15 steel flats of size 100 mm x 850 mm x 5 mm. It has to take the weight of the engine and a pump (if attached) and the failure may be by shear at the points of axle bearing housings, as it is the least area that takes the maximum load of 300 kgf. The shear stress produced at the point,

$$= \frac{300}{2 \times 100 \times 5}$$

$$= 0.3 \text{ kg/mm}^2$$

which is within the allowable shear limit for the selected C 15 steel.

Six cross bars of 32 mm x 12 mm flats are welded in between the chasis for additional rigidity and also to act as a hitch bracket with a 16 mm dia vertical drawbar pin for hitching the implement or trailer with the garden tractor (Drawing SA 06).

For a maximum hitching load of 450 kgf, the hitch bracket and drawbar pin may fail by shear. The shear stress produced on one of the hitch bars at an adverse field condition,

$$= \frac{750}{16 \times 12 \times 2}$$

$$= 1.95 \text{ kg/mm}^2$$

Hence a C 15 steel which can withstand upto 2.5 kg/mm^2 is selected.

The shear stress at the 16 mm dia drawbar pin,

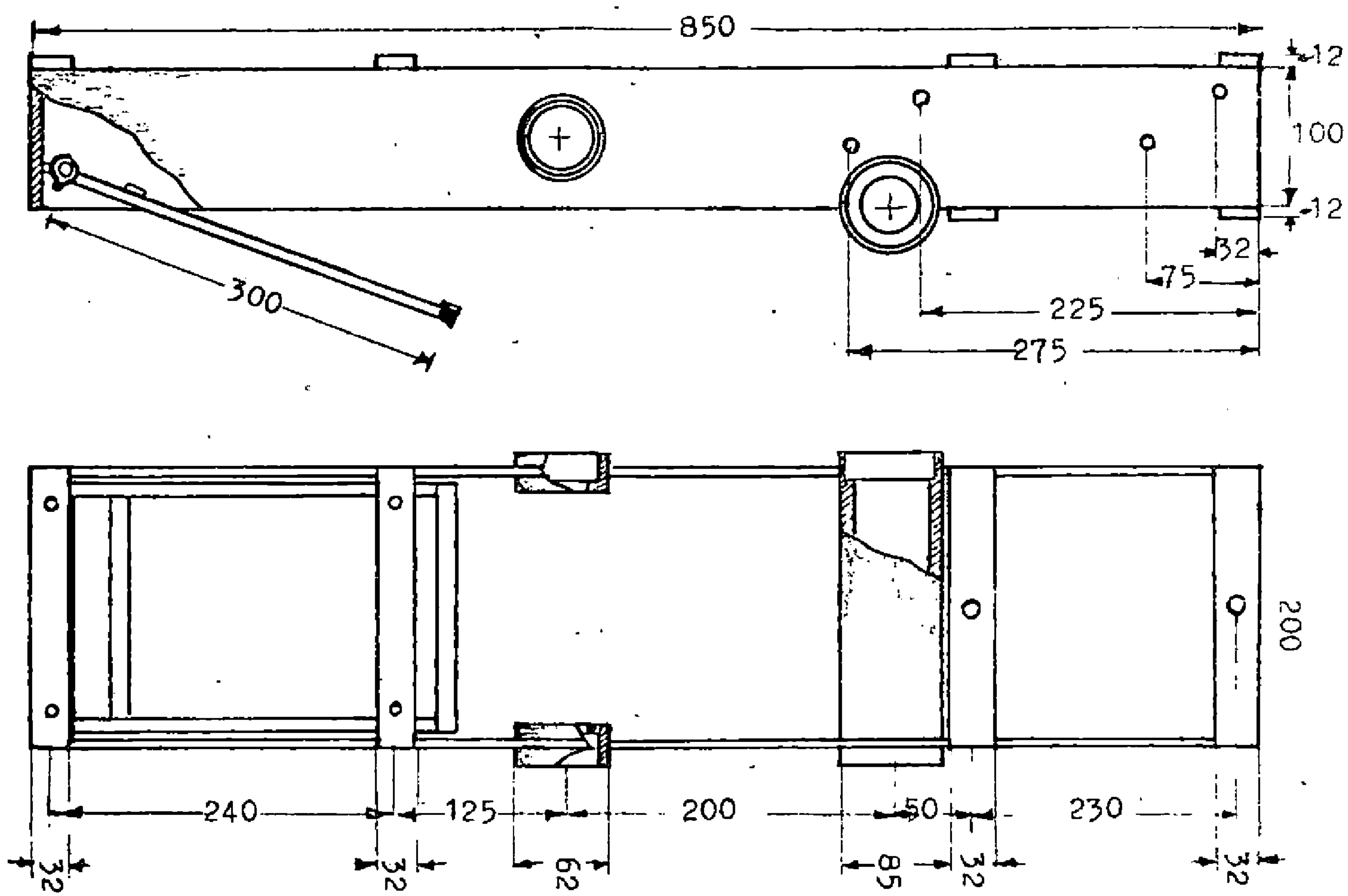
$$= \frac{750 \times 4}{\pi \times 16^2 \times 2}$$

$$= 1.87 \text{ kg/mm}^2$$

Hence the selected C 15 steel drawbar will be safe.

3.4. Handle

The design of handle is done by considering mechanical strength as well as ergonomical limits. At the worst situation when the implement noses into the soil, the handle may be utilized to lift the implement with a maximum of 200 kgf



SA06 CONSTRUCTION DETAILS OF CHASIS

ALL DIMENSIONS ARE IN mm

SCALE 1:5

at the drawbar. In order to take this load two main handle bars along with two tie rods are provided from the chasis and also to reduce the transmission of vibration from the engine. The material selected is C 15 steel flats of 31 mm x 8 mm size and the stress at the handle,

$$= \frac{200}{4 \times 31 \times 6}$$

$$= 0.27 \text{ kg/mm}^2$$

which is within the allowable limit and hence the handle selection is safe. Cross rods are also provided to increase the rigidity of the frame. The handle frame is fitted with the main beams at 30 deg inclination. The load at the handle grip when the distance between main axle and the handle grip is 1250 mm,

$$= \frac{200 \times 250}{1250}$$

$$= 40 \text{ kgf}$$

which is ergonomically within the bearable limits by two hands at an adverse condition.

In the bilaterally symmetrical handle the operator has to walk either on the inverted soil or in the furrow when the mould board plough is used, which is very inconvenient. Hence an offset of 162 mm is given to the handle frame and is found convenient.

Plate II POWER TRANSMISSION DETAILS OF THE GARDEN TRACTOR

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Plate II POWER TRANSMISSION DETAILS OF THE GARDEN TRACTOR

The distance between the two handle grips, that is the width of control area for manual control is taken as 525 mm and the main clutch lever is provided at 95 mm from the right handle grip. The accelerator, fuel cut off and main clutch levers, trailer seat and trailer brake pedal are ergonomically designed with the aid of the graph shown in the Fig.25, which was prepared from the data given by Mathews and Knight (1971). Drawing SA 05 shows the handle arrangement. The accelerator and fuel cut off cables are so connected with the engine controls for efficient operations, (Drawing MA 01).

3.5. Stand

The maximum load in addition to the part of the garden tractor weight the stand has to withstand, is the weight of a pump which may be loaded at front of the chasis. A maximum of 300 kgf is taken as the total load inclusive of pump and its accessories and hence the stress produced in two parallel pipes of 26 mm outer dia and 3 mm thickness,

$$= \frac{300 \times 4}{2 \times \pi (26^2 - 20^2)}$$

$$= 0.69 \text{ kg/mm}^2$$

which is within the allowable compressive stress for C 15 steel.

3.5.1. Stand Pin

The pins pivotting the stand with the chasis may fail by shear, hence the minimum diameter of the pins for C 40 steel,

$$=\left(\frac{300 \times 4}{4 \times \pi \times 6.5}\right)^{\frac{1}{2}}$$

$$= 3.83 \text{ mm}$$

Hence a 5 mm dia pin is selected. An over centre spring is also provided to keep the stand in its position at all times.

3.6. Assembling Details

Fabrication work of the garden tractor has been completed at the Research Workshop, Department of Agricultural Engineering, Kerala Agricultural University, Mannuthy, after finishing the drawings.

The intermediate shaft is first passed through its bearing housings and bushes of the shaft casing. After locking the bearings with circlips, sprocket pinion and 45 teeth sprocket wheel are fitted by keys, set collars and grub screws. The final drive axle is pressed through casing and bearings of the chasis. The 60 teeth sprocket wheel and two inner members of the overrunning clutch are positioned by keys and grub screws after locking the bearings with the circlips. Outer and inner members of the overrunning clutch are assembled by pressing the pawls of the inner members. Correct assembling of this clutch is checked by the uniform sound when it overturns. The pneumatic or cage wheels are fitted by wheel bolts with the outer members of the over-running clutch and locked by set collars and grub screws.

The handle frame is assembled with the chasis by means of bolts. Countershaft, bearings, driven pulley and sprocket pinion are assembled and locked with circlips, set collars and grub screws. Countershaft spring is fastened with the chasis and the swing arm. Then the main clutch link rod is connected with the countershaft casing and the hand lever.

The engine is mounted on the crossbars of the chasis and the accelerator and fuel cut off cables are connected from the engine controls to the respective hand levers. The stand and its over centre spring are fixed with the chasis.

The V belts and chains are wound around the respective pulleys and sprockets then the implement or trailer is hitched at the hitch bracket using the drawbar pin.

Results and Discussion

RESULTS AND DISCUSSION

The results of the evaluation of the low cost garden tractor, the dynamics and stability aspects involved in the construction of the unit, traction studies and economics are discussed in this chapter.

4.1. Determination of Centre of Gravity

In order to determine the centre of gravity, the garden tractor was dismantled into three groups as (1) prime mover, (2) the chasis with handle and power transmission elements and (3) the wheels. The c.g. of each of these was determined by hanging individually in different positions and locating the points of intersection of the lines of suspension. The c.g. of the garden tractor from three reference lines is found out analytically from the c.g. of the above three groups. It was checked by hanging the garden tractor totally in two different positions and finding out the point of intersection of the vertical lines from the point of suspension. The c.g. of the unit with implement is found out separately by hanging and the shifting of c.g. at working condition is also determined. The relative positions of individual c.g. from three reference lines are given in the Fig. 26 and the analytical method is as follows:

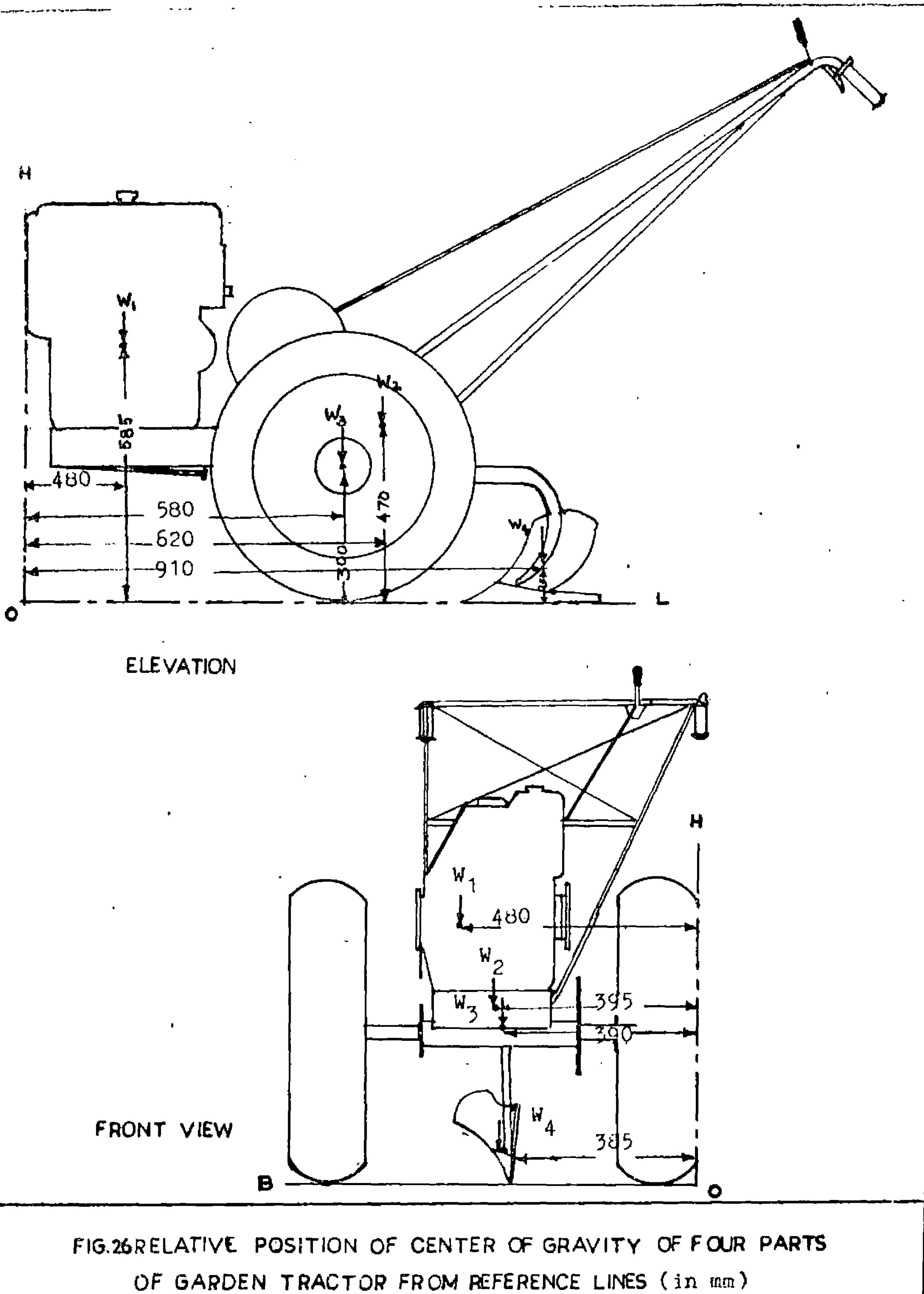

FIG.26 RELATIVE POSITION OF CENTER OF GRAVITY OF FOUR PARTS OF GARDEN TRACTOR FROM REFERENCE LINES (in mm)

$$140 \times L = (43 \times 480) + (56.5 \times 620) + (40.5 \times 580)$$

$$L = 565.42 \text{ mm}$$

$$140 \times H = (43 \times 585) + (56.5 \times 470) + (40.5 \times 300)$$

$$H = 456.14 \text{ mm}$$

$$140 \times B = (43 \times 480) + (56.5 \times 395) + (40.5 \times 390)$$

$$B = 419.66 \text{ mm}$$

Hence the c.g. of the garden tractor without any implement is found on calculation to lie at (570.00-565.42), 4.58 mm in front of the final drive axle, 456.14 mm above the ground level and (419.66-390.00), 29.66 mm right side of the central longitudinal line of the chasis. The location of the c.g. when the garden tractor was hanged without any implement in two different positions is at 8 mm in front of the axle, 454 mm above the ground level and 23 mm right side of the central longitudinal line of the chasis.

In the same method, the c.g. of the garden tractor with a mould board plough is analytically calculated by knowing the position of c.g. of the individual groups, as follows:

$$145.75 \times l = (43 \times 480) + (56.5 \times 620) + (40.5 \times 580) + (5.75 \times 910)$$

$$l = 579.02 \text{ mm}$$

$$145.75 \times h = (43 \times 585) + (56.5 \times 470) + (40.5 \times 300) + (5.75 \times 20)$$

$$h = 438.93 \text{ mm}$$

$$145.75 \times b = (43 \times 480) + (56.5 \times 395) + (40.5 \times 390) + (5.75 \times 385)$$

$$b = 418.29 \text{ mm}$$

The c.g. of the garden tractor with a mould board plough is found to lie at (579.02-570.00), 8.98 mm rear of the final drive axle, 438.93 mm above the ground level and (418.29-390.00), 28.29 mm right of the central longitudinal line of the chasis. The c.g. with the implement is found directly by hanging in two different planes to lie at 10 mm rear of the axle, 435 mm above the ground level and 25.5 mm right of the central longitudinal line.

It is found that by hitching a mould board plough the c.g. of the garden tractor moves approximately 18 mm towards rear, 19 mm towards ground level and 2.50 mm away from the central longitudinal line of the chasis. This transformation of c.g. is desirable for stability point of view at field operation.

By analytical method the c.g. of the garden tractor along with a trailer weighing 240 kg is found to lie at 850 mm rear of the final drive axle, 498.8 mm above ground level and 19.8 mm right of the centre longitudinal line.

4.2. Limiting Stable Angles

The maximum inclination for continuously operating in longitudinal and traverse slope is 35 deg for the selected, type 523, Greaves Lombardini engine. And the limiting stable angles of the garden tractor with trailer (Fig.27) are determined by employing the formulae given by Mahmud (1969) when,

- a = horizontal distance between c.g. and rear axle, 1490 mm
- l = horizontal distance between the two axles, 2340 mm
- h = vertical distance of c.g. from ground level, 498.8 mm
- B = track width of tractor trailer combination, 1500 mm
- b = tyre width, 120 mm
- e = distance between the c.g. and the vertical line passing through the centre of wheel track, 19.8 mm.

The limiting stable angle in upward slope,

$$\alpha_1 = \tan^{-1}\left(\frac{a}{h}\right)$$

$$= \tan^{-1}\left(\frac{1490}{498.8}\right)$$

$$= 71 \text{ deg } 30 \text{ min}$$

The limiting stable angle in downward slope,

$$\alpha_2 = \tan^{-1}\left(\frac{l-a}{h}\right)$$

$$= \tan^{-1}\left(\frac{2340-1490}{498.8}\right)$$

$$= 59 \text{ deg } 35 \text{ min}$$

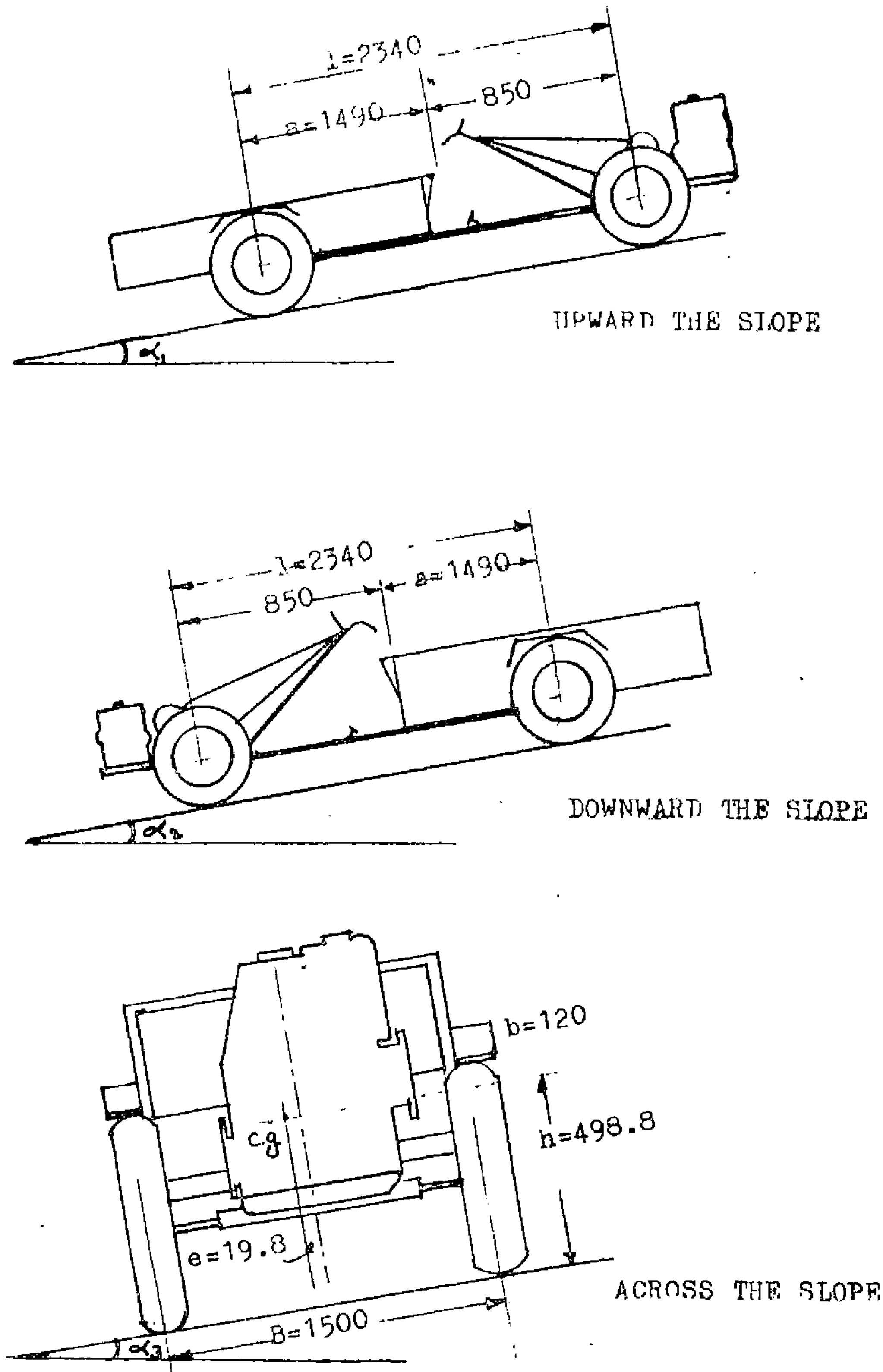

FIG. 27 DETAILS OF LIMITING STABLE ANGLES OF THE GARDEN TRACTOR

ALL DIMENSIOS IN mm

The limiting stable angle while running across the slope,

$$\alpha_3 = \tan^{-1} \frac{0.5(B+b)-e}{h}$$

$$= \tan^{-1} \frac{0.5\ (1500+120)-19.8}{498.8}$$

$$= 57 \text{ deg } 44 \text{ min}$$

Hence the unit is stable for transportation purpose at a maximum intermittent slope of 57 deg 44 min.

4.3. Starting Torque Requirement

The prime mover of the garden tractor is loaded additionally at the time of starting from rest due to the extra torque required to overcome the inertia of the moving components. Hence the prime mover is checked to overcome the inertial forces.

Moment of gyration $\frac{GD^2}{4\ g}$ of all revolving components is calculated as given below.

$$GD^2 = G_1D_1{}^2\ (\frac{N_1}{N})^2 + G_2D_2{}^2\ (\frac{N_2}{N})^2 + \ldots\ldots + G_nD_n{}^2\ (\frac{N_n}{N})^2$$

where G, G_1 .... G_n are weights, D, D_1 .... D_n are dia of gyration and N, N_1 .... N_n are the speeds of driving pulley, driven pulley, countershaft, sprocket pinion, 45 teeth sprocket wheel, intermediate shaft, sprocket pinion, 60 teeth sprocket wheel, final drive axle, overrunning clutches and the pneumatic wheels respectively.

$$GD^2 = \left[3.8 \times 0.0215^2\right] \left(\frac{1800}{1800}\right)^2 + \left[(4.250 \times 0.0552^2) + (0.75 \times 0.006^2) + (0.15 \times 0.0108^2)\right] \left(\frac{645.545}{1800}\right)^2 + \left[(1.80 \times 0.0279^2) + (0.80 \times 0.006^2) + (0.15 \times 0.0108^2)\right] \left(\frac{160}{1800}\right)^2 + \left[(2.10 \times 0.0368^2) + (3.00 \times 0.076^2) + (5.20 \times 0.035^2) + (36.50 \times 0.150^2)\right] \left(\frac{2933}{1800}\right)^2$$

$$= 0.0048331 \text{ kg m}$$

$$= 4.8331 \text{ kg mm}$$

Dynamic torque required at an acceleration time of 1.5 sec,

$$T_d = \frac{4.831 \times 1800}{375 \times 1.5}$$

$$= 15.459 \text{ kg mm}$$

Maximum starting torque required to overcome the power arrived under 3.1,

$$T_s = \frac{71620 \times 4.54 \times 10}{1800}$$

$$= 180.642 \text{ kg mm}$$

Hence the hp required for the prime mover to overcome the inertia force and starting resistance,

$$= \frac{(15.459 + 180.642) \times 1800}{71620}$$

$$= 4.929 \text{ hp}$$

which is less than the rated available 5.4 hp for the selected prime mover, and thus the unit will overcome the operational resistances.

4.4. Acceleration Studies on the Garden Tractor

When the garden tractor is operated in the field or road it will have either acceleration or retardation which contributes to the inertial forces of the unit. Fig.28 shows the forces acting on wheels and on plough at working condition. Assuming the centre of resistance acting at 6 cm above the bottom and 5 cm behind the share point of the plough, the acceleration analysis is done.

Let b = horizontal distance between wheel and centre of resistance of plough, cm

F = the difference between the soil thrust and rolling resistance, kgf

f = acceleration, cm/sec^2

F_s = resultant soil reaction on the plough, kgf

h = height of c.g. of the unit above the ground level, cm

l = the distance of c.g. of the unit from the centre of resistance of the plough, cm

V_1 and V_2 = the total normal reactions at wheels and plough respectively, kgf

W = weight of the garden tractor with implement, kgf

x = the centre of resistance from the ground level, cm

μ = coefficient of friction between the tyres and the soil surface.

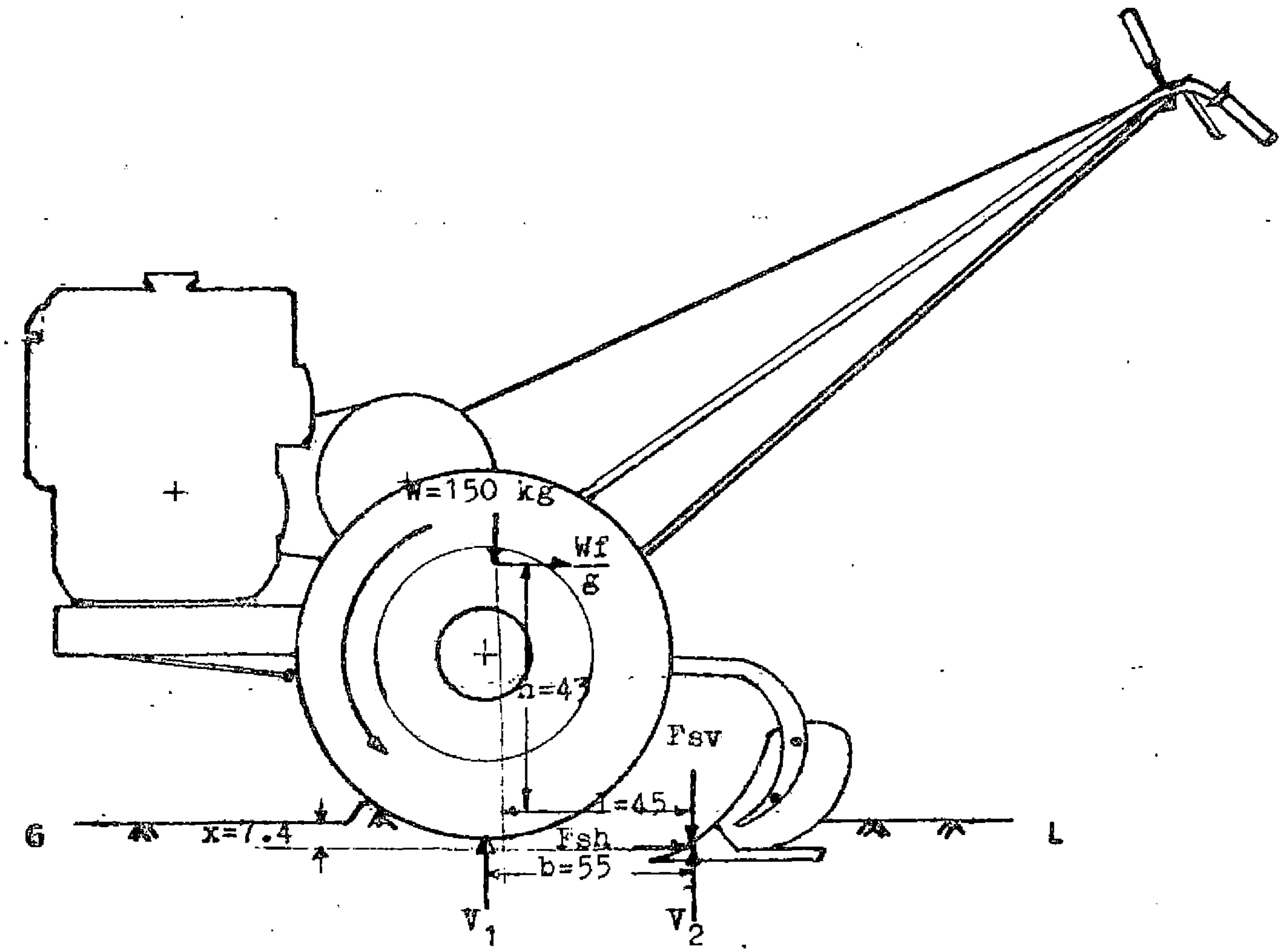

FIG.28 MECHANICS OF GARDEN TRACTOR AT PLOUGHING
ALL DIMENSIONS IN cm

Then $\Sigma V = 0$; $\quad F_{sv} + W = V_1 + V_2$

$\Sigma H = 0$; $\quad F_{sh} + \frac{W}{g}h = F$

Taking moment about the centre of resistance,

$$\Sigma M = 0; \quad V_1 b + \frac{W}{g}f(h+x) = Wl + Fx$$

$$\text{but} \quad F = F_{sh} + \frac{W}{g}f$$

$$(F_{sh} + \frac{W}{g}f)\ \frac{b}{h} + \frac{W}{g}f(h+x) = Wl + (F_{sh} + \frac{W}{g}f)x$$

$$f = \frac{g}{W}\left[\frac{W\mu l - F_{sh}(b-\mu x)}{b + \mu h}\right]$$

substituting the values from the Fig.28 and simplifying, $f = 244.570 - 47138\ F_{sh}$ where F_{sh} is the horizontal component of the soil reaction, which is equal to draft. For different draft the acceleration is calculated and is given in the Table 4.

4.5. Field Tests

The field tests on the garden tractor has been done at the Research Station and Instructional Farm, Mannuthy and at College of Horticulture, Vellanikkara for evaluating the following performance: (a) drawbar pull, (b) drawbar power, (c) slip, (d) acceleration, (e) coefficient of traction, (f) coefficient of rolling resistance, (g) rolling resistance, (h) soil thrust, (i) power efficiency and (j) tractive efficiency. These tests were conducted for both 6.00 x 12 size

Table 4. Relationship between draft and acceleration of the garden tractor

Sl.No.	Draft (kgf)	Acceleration (cm/sec^2)
1	9.78	198.469
2	19.56	152.368
3	29.34	106.267
4	39.12	60.166
5	48.90	14.065
6	58.68	-32.035
7	68.46	-78.136
8	79.24	-124.230
9	88.02	-170.339
10	97.80	-216.440
11	107.58	-248.399

Plate III GARDEN TRACTOR WITH CAGE WHEELS AT FIELD TESTS

Plate IV GARDEN TRACTOR WITH ITS TRAILER pneumatic tyre and cage wheel in the clay loam fields with the static weight of garden tractor (W), 140 kg. The depth of operation of the plough was changed by using the plough depth adjustment screw to get different drawbar pull (P) from 10 kgf to 110 kgf which was measured by a hydraulic dynamometer. Drawbar power (D) is the product of drawbar pull and vehicle velocity which represents the potential productivity of the vehicle. The wheel slip (S) has been estimated from the velocity at no load (V_o) and the velocity with load (V), (Wong, 1978).

$$S = \frac{V_o - V}{V_o}$$

By using slip (S) and the static weight of the garden tractor (W) the coefficient of rolling resistance (C_r) and soil thrust (F) are calculated from the following equations, (Manian, 1980).

$$S = 2 + 220\ C_r^2$$

$$F = P + W\ C_r$$

The power efficiency (PE) is determined from the equation, $PE = \frac{P}{F}(1-S)$. Tractive efficiency is the ratio of drawbar power to the power delivered by the prime mover. And the coefficient of traction is the ratio of drawbar pull to dynamic load on the wheels of the garden tractor, (Wong, 1978). The values obtained are given in the Table 5 and Table 6.

Table 5. Traction characteristics of garden tractor for pneumatic wheel

Sl. No.	Drawbar pull (kgf)	Draft (kgf)	Vehicle speed (kmph)	Slip (%)	Coeff. of traction	Coeff. of rolling resistance	Rolling resistance (kgf)	Drawbar power (hp)	Soil thrust (kgf)	Power efficiency (%)	Tractive efficiency (%)
1	10	9.78	3.178	4.2	0.0714	0.100	14.00	0.117	24.00	39.9	2.17
2	20	19.56	3.139	5.4	0.1428	0.124	17.36	0.240	37.36	50.6	4.44
3	30	29.34	2.926	11.8	0.2142	0.211	29.40	0.325	59.54	44.4	6.02
4	40	39.12	2.903	12.5	0.2856	0.218	30.52	0.430	70.52	49.5	7.96
5	50	48.90	2.896	12.7	0.3570	0.221	30.94	0.536	80.94	53.3	9.93
6	60	58.68	2.760	16.8	0.4284	0.259	36.26	0.613	96.26	51.8	11.35
7	70	68.46	2.588	22.0	0.4998	0.302	42.28	0.671	112.28	44.6	12.43
8	80	78.24	2.471	25.5	0.5712	0.305	42.70	0.732	122.70	48.6	13.55
9	90	88.02	2.319	30.1	0.6426	0.357	49.98	0.773	139.98	44.9	14.31
10	100	97.80	2.093	41.8	0.7140	0.427	59.50	0.715	159.50	36.4	13.24
11	110	107.58	1.840	43.3	0.7858	0.440	61.60	0.744	171.60	35.7	13.78

Table 6. Traction characteristics of garden tractor for cage wheel

Sl. No.	Drawbar pull (kgf)	Draft (kgf)	Vehicle speed (kmph)	Slip (%)	Coeff. of traction	Coeff. of rolling resistance	Rolling resistance (kgf)	Drawbar power (hp)	Soil thrust (kgf)	Power efficiency (%)	Tractive efficiency (%)
1	10	9.78	2.930	11.7	0.0714	0.210	29.40	0.109	39.40	22.4	2.02
2	20	19.56	2.688	19.0	0.1428	0.278	38.92	0.199	58.92	27.5	3.68
3	30	29.34	2.578	22.3	0.2142	0.304	42.56	0.286	72.56	32.1	5.30
4	40	39.12	2.429	26.8	0.2856	0.352	49.28	0.360	89.28	32.8	6.67
5	50	48.90	2.323	30.0	0.3570	0.357	49.98	0.430	99.98	35.0	7.96
6	60	58.68	2.074	37.5	0.4284	0.402	56.28	0.461	116.28	32.2	8.54
7	70	68.46	2.017	39.2	0.4998	0.411	57.54	0.523	127.54	33.4	9.96
8	80	78.24	1.834	44.7	0.5712	0.441	61.74	0.543	141.74	31.3	10.06
9	90	88.02	1.709	48.5	0.6426	0.460	64.40	0.570	154.40	30.0	10.56
10	100	97.80	1.643	50.5	0.7140	0.470	65.80	0.609	165.80	29.8	11.30
11	110	107.58	1.593	52.0	0.7858	0.477	66.78	0.649	176.78	29.8	12.02

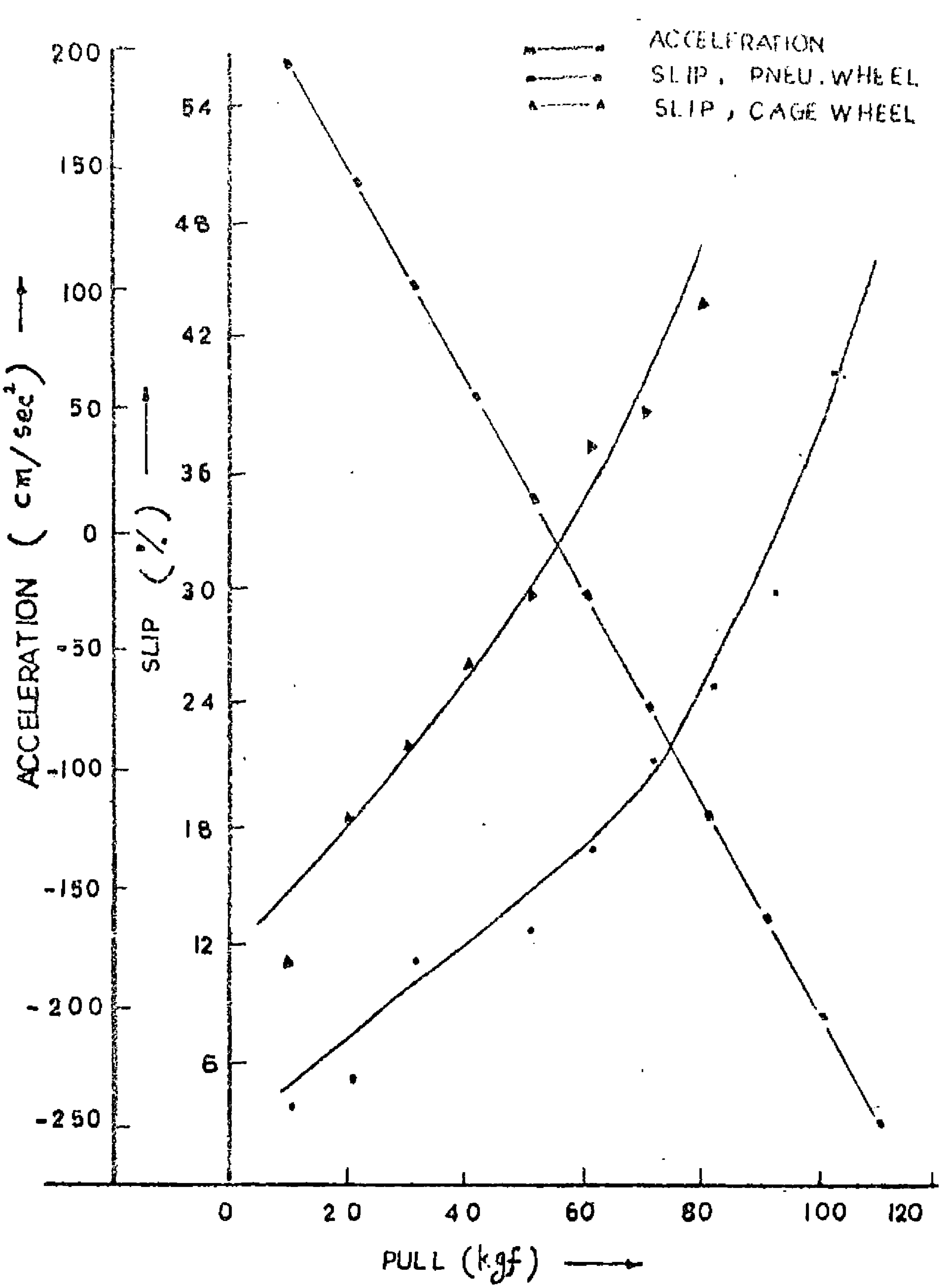

FIG. 29 INFLUENCE OF PULL ON SLIP AND ACCELERATION

Fig. 29 shows the influence of pull over slip and acceleration for pneumatic and cage wheel at upland ploughing. It is seen that the rate of increase of slip with respect to the pull is high beyond 20 per cent slip for pneumatic wheels and beyond 30 per cent slip for cage wheels. Taylor and Burt (1975) reported that the steep part of the pull curve generally occurred before 15 per cent slip whereas Manian (1980) found the rate of increase of slip was higher beyond 20 per cent slip, which was obtained for a pull of 50 kgf approximately.

The higher values of slip is prevailing for cage wheel at dry ploughing conditions comparing with the pneumatic wheels. Similar trend was also noticed for the IRRI 6-7 hp garden tractors, (IRRI, 1977).

It is observed that the value of pull, from which the higher increment of slip begins is at 50 kgf and 70 kgf for pneumatic and cage wheels respectively. It is also noticed that there exists an inverse relationship between pull or draft with acceleration for both the traction devices. The value of pull when the acceleration reaches zero is near about 55 kgf. Hence the garden tractor can be recommended for a ceiling pull of 55 kgf, which is equivalent to that of an average pair of bullocks.

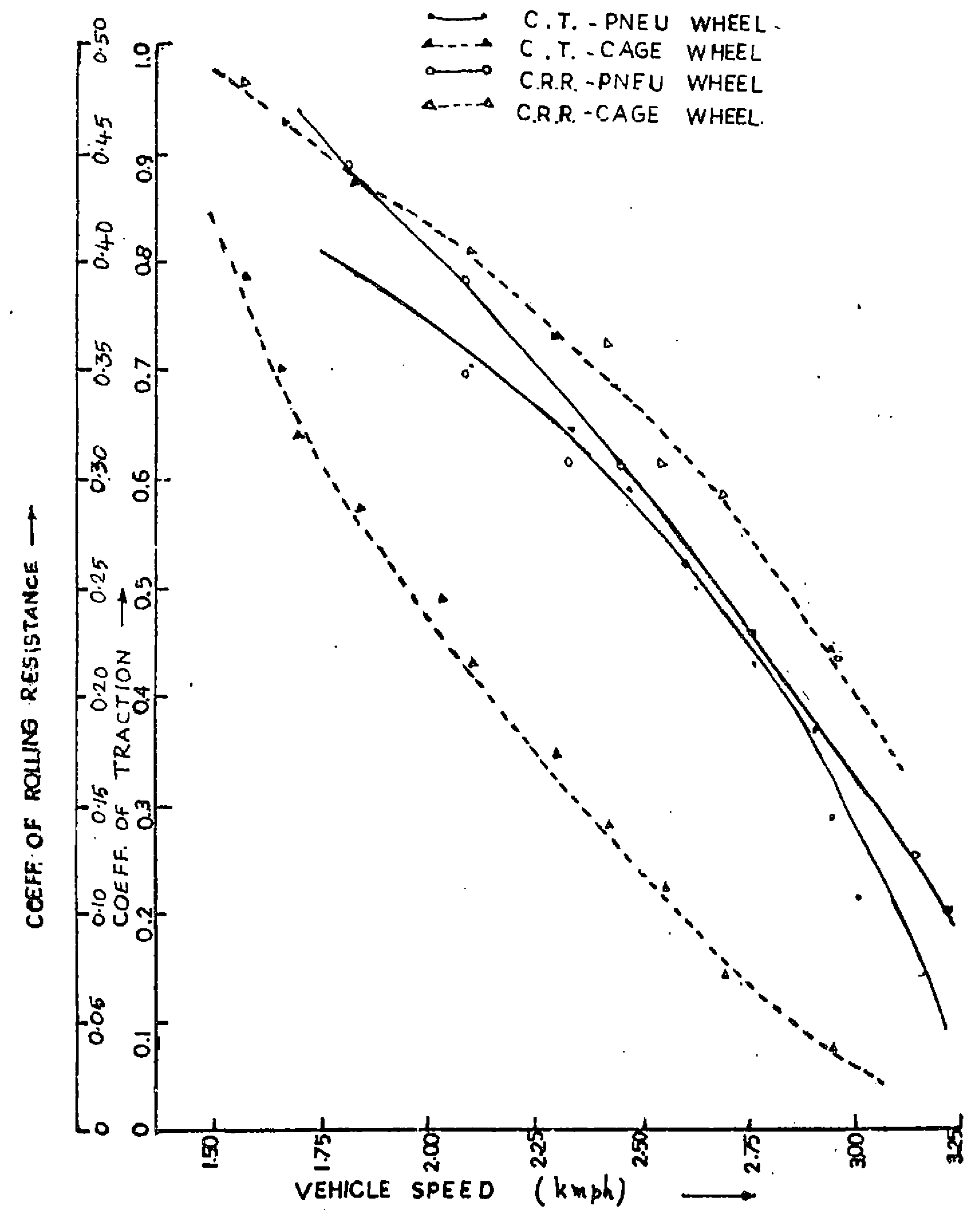

FIG-30 RELATIONSHIP BETWEEN COEFF. OF TRACTION AND COEFF. OF ROLLING RESISTANCE WITH SPEED

The relationship between coefficient of traction and coefficient of rolling resistance with respect to vehicle speed is shown in the Fig.30. It is noticed that the increase in vehicle speed reduces the coefficient of traction and coefficient of rolling resistance. The coefficient of rolling resistance shows higher values for cage wheels comparing the pneumatic wheels where as the pneumatic wheels have higher coefficient of traction than the cage wheels with respect to increase of speed. Chang and Cooper (1969) observed similar relationship between the coefficient of traction and slip. The rate of decrease of coefficient of traction and coefficient of rolling resistance are gradual upto a speed of 2.6 kmph. Manian (1980) confirmed the change point as at 30 per cent slip where as by superimposing the values of speed and slip, the change point is arrived approximately at 20 per cent slip for both the types of wheels of the low cost garden tractor.

The change of power efficiency and tractive efficiency with respect to the pull of the unit is illustrated in Fig.31. The power efficiency for the pneumatic wheel and cage wheel have the peak values of 53 per cent and 35 per cent respectively occurring at a pull of 50 kgf which is very close to the pull recommended already for the garden tractor namely 55 kgf after considering its acceleration characteristics.

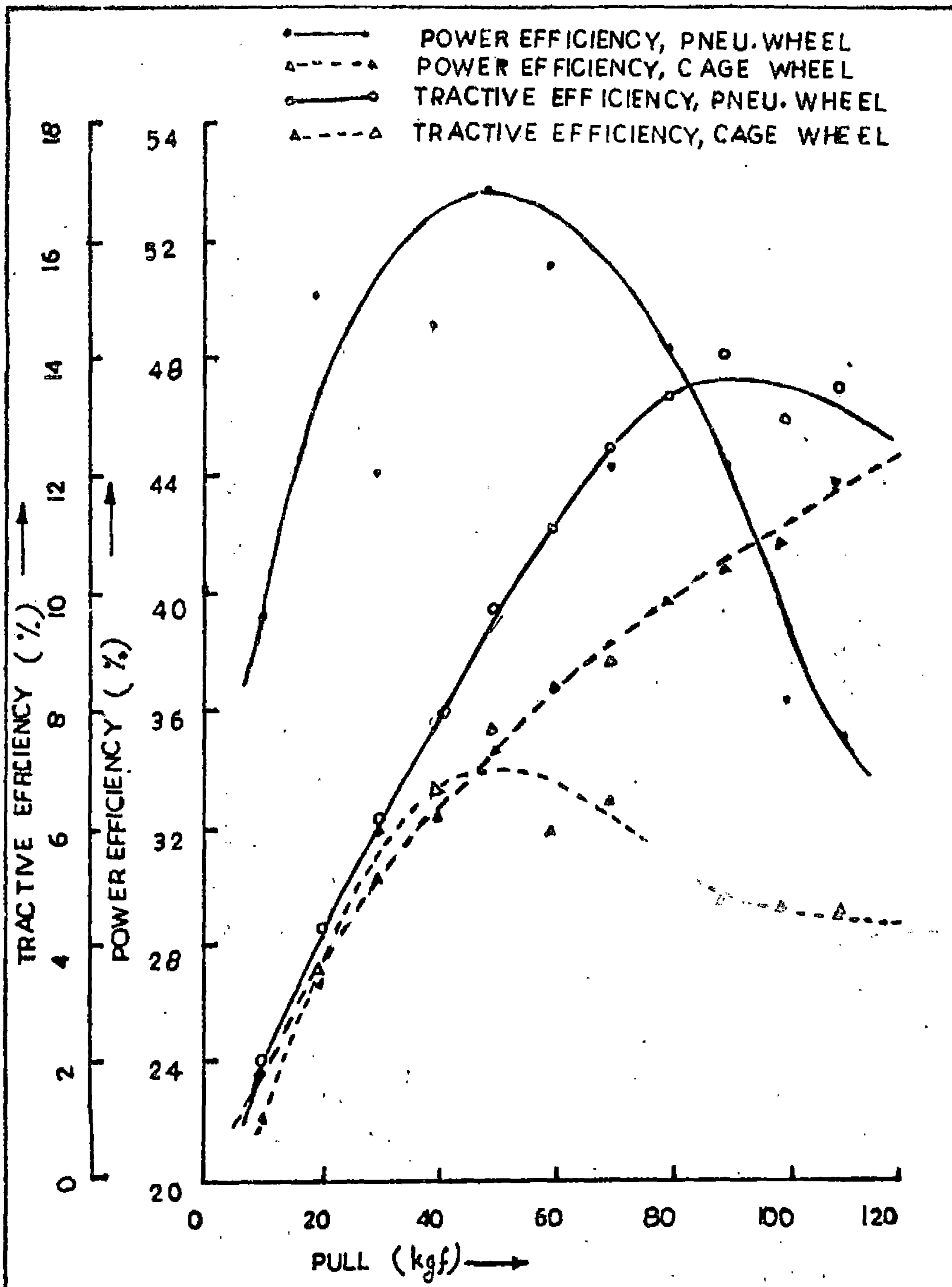

FIG.31 CHANGE OF POWER EFFICIENCY AND TRACTIVE EFFICIENCY WITH PULL

The tractive efficiency indicates an increasing trend with the increase of pull and the rate of increase of tractive efficiency is higher for the pneumatic wheel than the cage wheel for upland operations. From the power efficiency and tractive efficiency points of view, the garden tractor is recommended for using it at a pull range of 50 kgf to 80 kgf with pneumatic wheels for the upland operations. The average field capacity of the garden tractor with a mould board plough is 0.05 ha/hr and with a four tyne cultivator is 0.18 ha/hr respectively.

4.6. Road Tests

The alternate use of the garden tractor is for transportation. Hence the following performance of the garden tractor is studied.

4.6.1. Brake Test

The severity of braking of the garden tractor-trailer unit is measured with a nominal load of 600 kg at a maximum road speed of 6.25 kmph on a levelled tar paved road by the skid of tyre (K) which is given by Wong (1978) as,

$$K = \left(\frac{V-V_o}{V_o}\right)100$$

where V is the velocity without applying brake, and V_o is the velocity after the application of brake and before coming to rest, hence,

$$K = \left(\frac{6.25-2.15}{6.25}\right)100$$

$$= 65.60 \text{ per cent}$$

which is within the limit of application.

4.6.2. Slip Angle

The effectiveness of the steering alignment is measured by the slip angle, which is defined as the angle formed between the direction of wheel travel and the line of intersection of wheel plane with road surface. The garden tractor is operated at different speeds and the slip angle is found to be approximately equal to 1 deg 15 min which is considered as better steering ability and efficient functioning of the overrunning clutches.

4.6.3. Tractive Power in Climbing up a Slope

The tractive horse power (Q) in climbing up a slope (α) in deg for the garden tractor-trailer unit with a total static weight (W) in kgf which covered an actual distance (L) in meters with a time (t) in seconds, is given by Mahmud (1969) as,

$$Q = \frac{WL \sin\alpha}{75\ t}$$

$$= \frac{795 \times 275 \times \sin 35}{75 \times 155}$$

$$= 0.195 \text{ hp}$$

The following characteristics of the garden tractor-trailer unit was found out by using the formulae given by Taborek (1975) as they indicate the general mechanics of vehicle performance.

a. Static drive axle weight,

$$W_a = \frac{WL_r}{L}$$

where,

W = vehicle weight, kgf

L_r = distance of c.g. from rear axle, mm

L = wheel base, mm

substituting,

$$W_a = \frac{380 \times 1490}{1500}$$

$$= 377.47 \text{ kgf}$$

b. Weight distribution factor,

$$W_f = \frac{L_r + C_r H}{L + \mu H}$$

where,

C_r = coefficient of rolling resistance

H = height of c.g. from ground level, mm

M = coefficient of road adhesion

substituting,

$$W_f = \frac{1490 + (0.02 \times 498.80)}{1500 + (0.75 \times 498.80)}$$

$$= 0.80$$

c. Maximum transeferable tractive force

$$f_t = W_f \mu W$$

substituting,

$$f_t = 0.80 \times 0.75 \times 380$$

$$= 228.00 \text{ kgf}$$

d. Maximum gradability,

$$s = \tan^{-1} (W_f \mu - C_r)$$

substituting,

$$s = \tan^{-1} \left[(0.80 \times 0.75) - 0.02\right]$$

$$= 30 \text{ deg } 6 \text{ min } 45 \text{ sec}$$

e. Maximum drawbar pull on level ground,

$$P = W \left(\frac{\mu L_f - C_r L}{L - H_d}\right)$$

where,

L_f = distance of c.g. from front axle, mm

H_d = height of hitch point from ground level, mm

substituting,

$$P = 380 \left[\frac{(0.75 \times 850)-(0.02 \times 1500)}{1500-(0.75 \times 35)}\right]$$

$$= 156.64 \text{ kgf}$$

f. The minimum turning radii of the garden tractor with implement and with trailer were studied and are 425 mm and 2150 mm respectively where as the Kubota Power Tiller is having the minimum turning radii as 600 mm and 3300 mm with implement and with trailer respectively.

These values of the garden tractor suit the functional requirements for adoption of the low cost garden tractor for field and transport operations.

4.7. Endurance Tests

The performance of the fabricated and purchased components were studied under the endurance tests. The garden tractor was put under use intermittently totalling 150 hours duration with implements namely, a mould board plough, a four tyne cultivator, a disc harrow, a bund former and a trailer with a load of 750 kgf. The following observations were made from the tests.

a) Due to the vibration of the engine the ordinary type foundation bolts of engine were loosened frequently, hence additional locknuts were provided for the engine foundation bolts.

b) At sudden release of main clutch, the belts are sliped away from their pulley grooves and it was prevented by welding a belt retainer with the chasis.

c) Improper engagement of pawls with the ratchets in the overrunning clutch made the inner and outer members loose, and hence the grease was oozed out. As there was no seating of pawl with its semicircular back, bending of pin was noticed. Spring action was arrested and wearing out of the pawl face-corners was identified. These faults in the overrunning clutch were due to the improper alighment and seatings of pawl, pawl pin and springs. Hence in the second model fabricated, all these defects were rectified by providing correct seating for pawls at its semicircular back and by providing seating, step and guide for the spring in the pawl itself. Hence any lateral movement was arrested and thus bending and wearing of pawl face were rectified.

4.8. Material and Economical Analysis of Garden Tractor

The materials required for fabricating the garden tractor is grouped under the headings of 'fabricated components' and 'purchased components'. A detailed material list with specification, cost and quantity is prepared and is presented in the Table 7 and Table 8. The total cost of the power tiller is arrived by adding 10 per cent overhead charges on cost of materials and 200 per cent overhead charges on cost of labour, over the actual cost of fabrication. The cost of operation is compared with that of Kubota 9-12 hp power tiller at different working hours per year. The rate of change of cost of operation with increase of yearly working hours is also studied.

Table 7. Details of fabricated components and their cost

Sl. No.	Part	Drawing No.	Specification	Weight (kg)	Quantity (No.)	Cost (Rs.)
1	Drive pulley	SA 01	Casting, B section V pulley 80 mm and 160 mm dia	3.800	1	27.50
2	Driven pulley	SA 01	Casting, B section V pulley 220 mm dia	4.250	1	43.70
3	Countershaft	SA 01	C 40 steel, 28 mm dia	0.760	1	12.30
4	Countershaft casing	SA 01	C 15 steel, 57 mm dia	1.100	1	9.60
5	Sprocket pinion	SA 02	C 15 steel, 50.8 mm dia	0.350	2	16.90
6	45 teeth sprocket wheel	SA 02	C 15 steel, 189.2 mm dia	1.800	1	78.10
7	60 teeth sprocket wheel	SA 03	C 15 steel, 249.9 mm dia	2.100	1	85.10
8	Intermediate shaft	SA 02	Axle steel, 30 mm dia	0.850	1	31.50
9	Intermediate shaft casing	SA 02	C 15 steel, 52 mm dia	1.350	1	9.60
10	Countershaft clutch lever	SA 05	C 15 steel, 30 mm x 10 mm flats	2.350	2	11.50
11	Countershaft clutch lever	SA 05	SWG 15 sheet	0.270	1	2.25
12	Countershaft link rod	SA 05	C 15 steel, 10 mm dia	1.510	1	7.60
13	Hand lever	SA 05	C 15 steel, 12 mm dia	0.620	1	3.75

(Table 7 continued)

Sl. No.	Part	Drawing No.	Specification	Weight (kg)	Quantity (No.)	Cost (Rs.)
14	Chasis main beam	SA 06	C 15 steel, 100 mm x 5 mm	7.850	3	38.75
15	Chasis cross bars	SA 06	C 15 steel, 32 mm x 12 mm	3.725	6	16.20
16	Bearing housing	SA 06	C 15 steel, 62 mm OD	0.870	2	3.45
17	Final axle	SA 03	Axle steel, 45 mm dia	1.050	1	41.20
18	Final axle casing	SA 03	C 15 steel, 85 mm OD	2.125	1	13.75
19	Overrunning clutch assembly	SA 04	Casting and C 14 steel pawl and pawl pins	5.200	2 sets	570.00
20	Handle main flat	SA 05	C 15 steel, 31 mm x 8 mm	5.920	4	30.25
21	Handle cross rods	SA 05	C 15 steel, 10 mm dia	2.630	2	12.50
22	Fuel cut off lever	SA 05	SWG 15 sheet	0.150	1	2.00
23	Stand	SA 06	C 15 steel pipe, 26 mm OD	1.260	1	7.35
24	Drawbar pin	MA 01	C 15 steel, 16 mm dia	0.265	1	1.40
25	Pivot pins	MA 01	C 40 steel, 5 mm dia	0.125	14	3.00
			Cost of fabricated components		=	Rs.1079.25

Table 8. Details of purchased components and their cost

Sl.No.	Part	Specification	Quantity (No.)	Cost (Rs)
1	Diesel engine	Lombardini, 5.4 hp, type 523, 1800 rpm	1	5201.75
2	V belt	B-1062/42-41	2	69.45
3	V belt	B-1168/46-45	1	35.80
4	Chain	ISO/DIN 084-1 R 1248 H	1676.40 mm	75.10
5	Chain clips	12.70 mm pitch	3	2.60
6	Bush	Gunmetal liner, 42 mm OD 30 mm ID, 20 mm length	2	56.50
7	Bearings	SKF 6204 2Z	2	63.00
8	Bearings	SKF 3304 2Z	2	173.70
9	Bearings	SKF 6207 2Z	2	87.50
10	Circlips	Light B-47 IS: 3075-1965	2	4.40
11	Circlips	Light B-52 IS: 3075-1965	2	5.15
12	Circlips	Light B-72 IS: 3075-1965	2	7.35
13	Keys	- -	6	8.25
14	Set collars with slotted head grub screws	Light series, Type -A:		
		20 mm bore	4	24.55
		25 mm bore	2	18.90

(Table 8 continued)

Sl.No.	Part	Specification	Quantity (No.)	Cost (Rs.)
15	Slotted head grub screws	6.5 mm dia and 8.0 mm dia	7	6.75
16	Springs	-	2	3.75
17	Accelerator cables	Kubota spare part	2	62.00
18	Cable release clamp	Kubota spare part	1	9.70
19	Pneumatic tyre wheel	Tyre: 6.00 x 12 size	2	1689.20
		Tube: 6.00 x 12 size	2	148.30
		Rim : 6.00 x 12 size	2	633.50
20	Primer and paint		600 ml	30.50
21	Hexogonal head bolts, nuts and washers		32	15.25
		Cost of purchased components	=	Rs.8432.95

Cost of fabricated components	=	Rs. 1079.25
Cost of prime mover	=	Rs. 5201.75
Cost of other purchased components	=	Rs. 3231.20
Cost of material	=	Rs. 9512.20
Cost of labour	=	Rs. 850.00
Cost of the unit	=	Rs.10362.20
10 per cent overhead charges on cost of material	=	Rs. 951.22
200 per cent overhead charges on cost of labour	=	Rs. 1700.00
Total cost of the unit	=	Rs.13013.42

This is comparable with the National power tiller of the same power output costing Rs.13555/- (excluding sales tax) and Kubota power tiller of 9-12 hp costing Rs.32190/-. As most of the farmers own a diesel engine for pumping, additional amount of only Rs.5160.45 is needed to own this low cost garden tractor, which is much comparable with a pair of bullocks.

Cost of operation in Rs/year is expressed by,

$$C_o = \frac{(C_f+R)}{100}\frac{P}{h} + (L+F+O)$$

where,

C_o = cost of operation per hr, Rs.

C_f = fixed cost in percentage of capital cost

P = capital cost, Rs.

h = hours of operation per year

R = repair and maintenance per year in percentage of capital cost

L = cost of labour, Rs./hr

F = cost of fuel, Rs./hr

O = cost of oil and lubricant, Rs./hr

The fixed cost includes the following components.

a. depreciation, 10 per cent of P

b. interest, 4.4 per cent of P

c. tax, 1.25 per cent of P

d. insurance, 0.25 per cent of P and

e. housing, 1.00 per cent of P

Hence the fixed cost is 16.90 per cent of P. Repair and maintenance cost is taken as 6 per cent of P and the life period of the low cost garden tractor is assumed as 10 years.

Cost of fuel	=	Rs.3.40/lit.
Cost of oil and lubricant	=	1/3 cost of the fuel
Cost of labour	=	Rs.2.50/hr

Let the fuel consumption be, x lit./hr. Then the cost of operation per hr,

$$C_o = \left(\frac{16.90+6.00}{100}\right)\frac{P}{h} + 2.50+(3.40x)+\left(\frac{3.40x}{3}\right)$$

$$= 0.2290\,\frac{P}{h} + (4.73x) + 2.50$$

Hence the cost of operation per hr for the low cost garden tractor, when P = 13100 and x = 1.00,

$$C_o = \frac{2999.9}{h} + 7.23$$

and the rate of decrease of cost of operation,

$$\frac{dC_o}{dh} = - \frac{2999.9}{h^2}$$

The cost of operation per hr for the Kubota power tiller, when P = 32190 and x = 1.75,

$$C_o = \frac{7371.51}{h} + 10.78$$

and the rate of decrease of cost of operation,

$$\frac{dC_o}{dh} = - \frac{7371.51}{h^2}$$

The cost of operation and the rate of decrease of cost of operation for different hours of yearly use are given in the Table 9 and is illustrated in the Fig.32 for the low cost garden tractor and the Kubota power tiller. It is observed that an increase in yearly use for both the units will reduce the cost of operation per hour. But the cost of operation per hour is changing less gradually from the break even point of 600 hr/year for the newly developed low cost garden tractor where as for the Kubota power tiller the break even point is 1000 hr/year. It justifies the

Table 9. Cost of operation and rate of decrease of cost of operation in relation to the yearly use

Sl. No.	Hours of operation	P = 13100; x = 1.00		P = 32190; x = 1.75	
		Cost of operation per hour (Rs.)	Rate of decrease of cost of operation ($\times 10^{-3}$)	Cost of operation per hour (Rs.)	Rate of decrease of cost of operation ($\times 10^{-3}$)
1	200	22.23	75.00	47.64	184.29
2	400	14.73	18.75	29.21	46.07
3	600	12.23	8.33	23.07	20.48
4	800	10.98	4.69	19.99	11.52
5	1000	10.23	2.00	18.15	7.38
6	1200	9.73	2.08	16.92	5.12
7	1400	9.37	1.53	16.05	3.76
8	1600	9.10	1.17	15.59	2.88

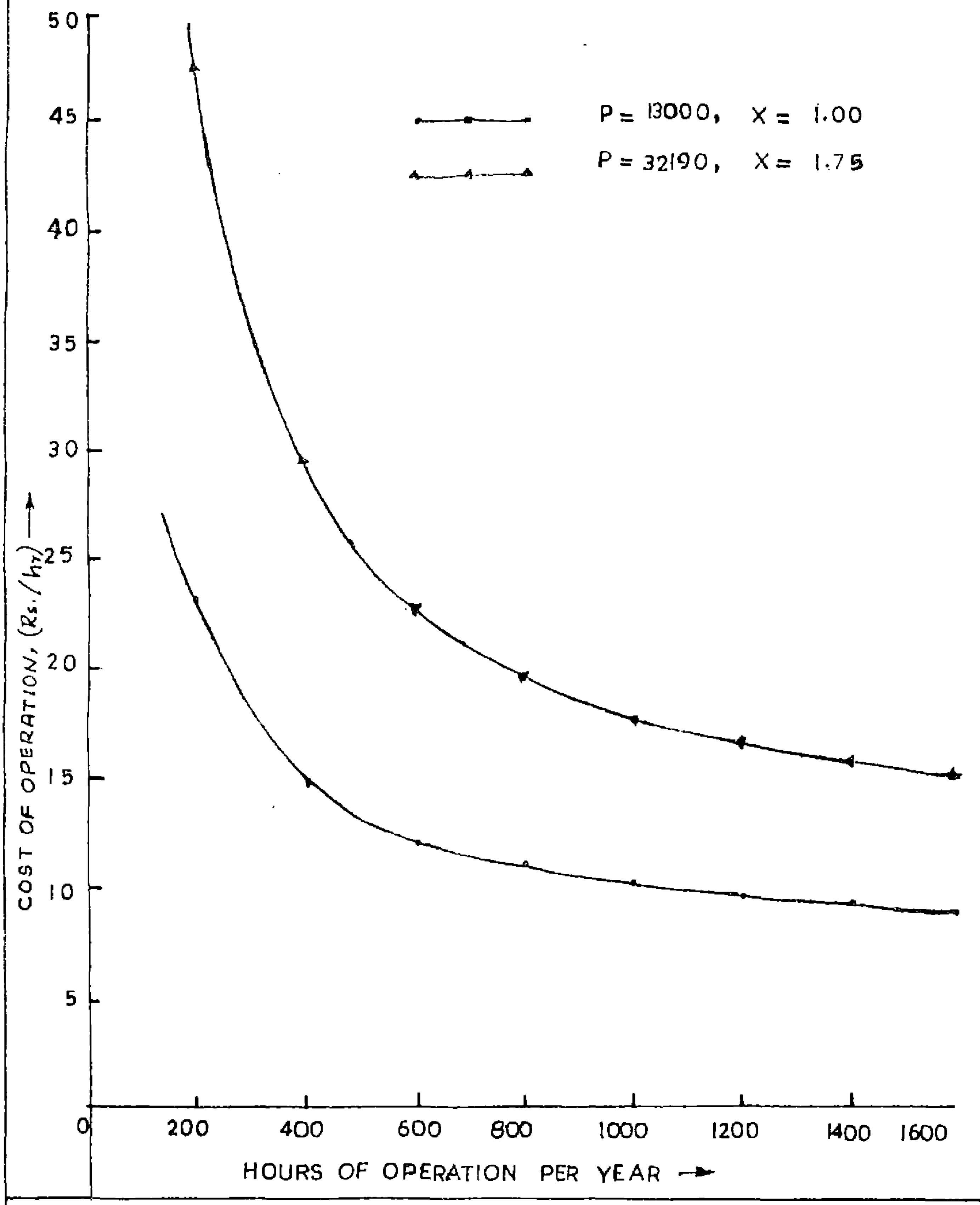

FIG.32 RELATIONSHIP BETWEEN ANNUAL UTLIZATION AND COST OF OPERATION

advantage in socio-economic aspects for having an own unit of the low cost garden tractor than the factory made power tillers by the average Indian farmers.

Summary

SUMMARY

The two wheel garden tractor stands between the animal traction and four wheel tractor which would replace only animals but not people. It is the machine most suitable for a country like India, for promoting economic development, employment and better income distribution. Hence the development of a low cost garden tractor was taken up for optimizing the power needed, selecting suitable design criteria and finding out the operational limits.

5.1. After a critical analysis, 5.4 hp Greaves Lombardini, type 523 diesel engine with 1800 rpm which is compact and weighing only 38 kg is selected to serve as the prime mover. Total speed reduction of 30.6314 and 61.3629 have been achieved for operating at a speed of 6.635 kmph and 3.317 kmph for road and field conditions by using a simple cone pulley and V belt arrangement, along with chain and sprockets. The first step speed reduction is ultimately utilized as an effective pivotted countershaft clutch.

5.2. The countershaft, intermediate shaft and the final drive axle are designed by considering proper service factors, adequate factor of safety and to suit the various power transmission elements. The three shafts are so positioned, that the weight is uniformly distributed over the final drive axle and a perfect balancing is achieved.

5.3. One pair of overrunning clutches are designed, fabricated and preliminary field tests are carried out. The defects are rectified and further improvements have been incorporated and are again successfully field tested. Manual steering is made very simple by the successful application of the overrunning clutches which eliminated separate steering clutch assembly, wheel flanges and split or hexogonal final axle.

5.4. A pivotted countershaft clutch with 25 mm swing at the driven pulley is designed and successfully tested for its functional, mechanical and ergonomical aspects. Chasis, handle, stand, drawbar and all other components of the unit are designed and fabricated by taking care in integrating the parts and using maximum number of standard components to reduce the total cost and for easy maintenance. The controls of main clutch, acceleration and fuel cut off are located at handle frame to fall within the ergonomical limits for easy operation. The garden tractor has a length of 1750 mm, width of 780 mm and height of 1220 mm and weighs 140 kg without any implement.

5.5. From the dynamics and kinematics of the garden tractor the following aspects are studied.

a) The centre of gravity of the unit without any implement is found to be 8 mm in front of the final drive axle, 454 mm above the ground level and 23 mm right of the central longitudinal line of the chasis. When a mould board plough is mounted the centre of gravity occupies a new position at 10 mm in rear of the final drive axle, 435 mm above the ground level and 25.5 mm right of the central longitudinal line. These values have been compared by analytical method.

b) To overcome the dynamic and other inertial torque at the time of starting, a minimum of 4.929 hp is found to be necessary, which is less then the rated hp of the selected prime mover.

c) The limiting stable angles of the garden tractor in upward, downward and across the slope are calculated to be 71 deg 30 min, 50 deg 35 min and 57 deg 44 min respectively. The severity of braking is found out by the skid which is equal to 65.60 per cent and the steering effectiveness is determined by the slip angle which is equal to 1 deg 15 min for the garden tractor-trailer combination. The maximum transferable tractive force is found to be 228 kgf, the maximum drawbar pull on level ground is found as 156.64 kgf and the maximum gradability, which is the slope, that the vehicle can negotiate at its constant speed is calculated to be 30 deg 7 min. The minimum turning radii with and without trailer is found to be 2150 mm and 425 mm respectively.

5.6. By 150 hours endurance test, modifications and developments have been incorporated in the overrunning clutch design and no breakdown of any other designed or purchased components have been observed.

5.7. In the field tests by changing the pull which was measured by a hydraulic dynamometer, the characteristics of slip, coefficient of traction, coefficient of rolling resistance, drawbar power, power efficiency and tractive efficiency are found out for the 6.00 x 12 size pneumatic tyre and cage wheels. The rate of increase of slip is higher beyond 20 per cent and 30 per cent for pneumatic and cage wheels when the pull is around 50 kgf and 70 kgf respectively. The cage wheel shows comparatively higher values of slip for upland ploughing. The acceleration of the garden tractor has on inverse relationship with pull or draft and the value of pull when the acceleration reaches zero is around 55 kgf. The rate of decrease of coefficient of traction and coefficient of rolling resistance with respect to speed of the vehicle are found to be gradual upto a speed of 2.6 kmph. The peak values of the power efficiency namely 53 per cent for the pneumatic wheel and 35 per cent for the cage wheel exist when the pull is around 50 kgf.

5.8. The designed low cost garden tractor can easily be dismantled and assembled into three main parts namely, prime mover (43.0 kg), transmission system with chasis (56.5 kg) and the ground drive components (40.5 kg) which facilitates easy transportation to any type of fields. The cost of materials for fabrication of the garden tractor is Rs.9512.20 and the cost of labour is Rs.850. When a 10 per cent overhead charges on the material and 200 per cent overhead charges on the labour is imposed, the total cost comes to Rs.13013.42. Out of this 9.12 per cent is cost of fabricated components, 71.28 per cent is cost of purchased components and 19.60 per cent cost of labour.

a) The cost of operation of the garden tractor can be expressed as given below in terms of cost of the garden tractor (P), fuel consumption in litres per hour (x) and hours of operations per year (h),

$$\text{Cost of operation (Rs/hr)} = 0.2290 \left(\frac{P}{h}\right) + (4.73x)+2.50$$

b) The cost of operation per hour is changing less gradually from an yearly use of 600 hours for the garden tractor and 1000 hours for the Kubota power tiller which costs Rs.32,190. It clearly indicates the adaptability of the low cost garden tractor over the factory made power tillers.

5.9. Further improvements on the garden tractor may be attempted by (a) providing a brake at the outer member of the overrunning clutch which can be operated along with the main clutch lever, (b) employing vibration dampers at engine foundation bolts, (c) substituting tubular handle frame for further restriction of vibration transmission, and (d) providing a simple type of separate stand control lever which can be operated from the handle frame.

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Appendices

APPENDIX I

SPECIFICATIONS OF THE PRIME MOVER

1.	Make	: Greaves Lombardini
2.	Model	: Type 523 (Four stroke, aircooled, direct injection on piston, gear pump lubrication, automatic governor, rope crank starting)
3.	No. of cylinders	: 1
4.	Bore	: 78 mm
5.	Stroke	: 68 mm
6.	Displacement	: 325 cc
7.	Compression ratio	: 18:1
8.	Rated speed	: 1800 rpm
9.	Power	
	a) N DIN 70020	: 6.5 hp
	b) NB DIN 6270	: 6.0 hp
	c) NA DIN 6270	: 5.4 hp
10.	Maximum torque	: 2960 kg mm
11.	Specific fuel consumption	: 220 gm/hp hr
12.	Fuel tank capacity	: 4.5 lit.
13.	Oil consumption	: 0.013 kg/hr
14.	Oil sump capacity	: 1 lit.
15.	Air cleaner oil bowl capacity	: 0.15 lit
16.	Maximum angularity	: 35 deg
17.	Dimensions, mm	: 427 x 352.5 x 476
18.	Dry weight	: 38kg

APPENDIX II

DESIGN OF V BELT DRIVE FOR FIELD OPERATION

The drive pulley diameter (d), is selected as 80 mm, to accommodate in the available space below the fuel tank of the engine.

The dia of driven pulley,

$$D = 80 \times 2.75$$

$$= 220 \text{ mm, which is a standard size}$$

Selection of centre distance, (C)

$$C_{min} = 0.55\ (D + d) + T, \quad \text{when } T = 11$$

$$= 0.55\ (220 + 80) + 11$$

$$= 176 \text{ mm}$$

$$C_{max} = 2\ (D + d)$$

$$= 2\ (220 + 80)$$

$$= 600 \text{ mm}$$

Adopting a centre distance of 300 mm, the belt pitch length,

$$L = 2C + \frac{\pi}{2}(D+d) + \frac{(D-d)^2}{4C}$$

$$= (2 \times 300) + \frac{\pi}{2}(220+80) + \frac{(220-80)^2}{4 \times 300}$$

$$= 1087.572 \text{ mm}$$

The corrected standard standard nominal pitch length is 1110 mm and the nominal inside length is 1067 mm.

(Appendix II continued)

Corrected centre distance, C:

$$L = 2C + \frac{\pi}{2}(D+d) + \frac{(D-d)^2}{4C}$$

$$1110 = 2C + \frac{\pi}{2}(220+80) + \frac{(220-80)^2}{4C}$$

C = 311.515 mm, which is within the allowable range of 176 mm to 600 mm.

Arc of contact angle,

$$\theta = 2 \cos^{-1}\left(\frac{D-d}{2C}\right)$$

$$= 2 \cos^{-1}\left(\frac{220-80}{2x311.515}\right)$$

= 154 deg 1 min 40 sec

Correction factor for service (F_a) for light duty upto 10 hr of working is 1.00. The small dia factor (F_b) to account for variation of arc of contact is 1.13, when D/d is 2.75. Correction factor for length of standard V belts (F_c) is 0.85, and the correction factor for arc of contact (F_d) for 154 deg is 0.93.

Equivalent pitch dia,

$$d_e = d \times F_b$$

$$= 80 \times 1.13$$

$$= 90.40 \text{ mm}$$

Belt speed, $S = \frac{\pi \times 80 \times 1800}{1000 \times 60}$

$$= 7.539 \text{ m/sec}$$

(Appendix II continued)

Corresponding to the drive power, P = 3.97 kw, S = 7.539 m/sec and d_e = 90.40 mm, the rating of V belts,

$$R = 1.125 \text{ for A section}$$
$$= 1.991 \text{ for B section}$$

No. of B section V belts

$$= \frac{P \times F_a}{R \times F_c \times F_d}$$

$$= \frac{3.97 \times 1}{1.991 \times 0.85 \times 0.93}$$

$$= 2.5237$$

As normally 75 per cent of the maximum available power is utilized, only two numbers of B section V belts are selected.

The specification of the belt is B-1067/42-41. A pair of standard B section pulleys of 80 mm dia and 220 mm dia is selected for field operation.

APPENDIX III

DESIGN OF V BELT DRIVE FOR TRANSPORTATION

The dia of engine pulley, d $= 160$ mm

The dia of driven pulley, D $= 220$ mm

$$C_{min} = 0.55\ (220+160)+11$$
$$= 220 \text{ mm}$$
$$C_{max} = 2\ (220+160)$$
$$= 760 \text{ mm}$$

To accommodate the pivotted countershaft clutch in the belt drive, the centre distance of the belt drives for field and road speeds should approximately be the same. Hence C is taken as 300 mm, which is also within the allowable range for transportation.

The belt pitch length, $L = 2 \times 300 + \frac{\pi}{2}(220+160)+\frac{(220-160)^2}{4 \times 300}$

$$= 1199.90 \text{ mm}$$

The corrected standard nominal pitch length is 1212 mm and the nominal inside length is 1168 mm

$$1212 = 2C + \frac{\pi}{2}(220+160)+ \frac{(220-160)^2}{4 \times C}$$

Hence the corrected centre distance, $C = 306.075$ mm, which is very close to the centre distance arrived for field operation, namely 311.515 mm and the deviation of 5.440 mm is adjusted by providing double positions in the clutch lever retainer.

(Appendix III continued)

Arc of contact angle, $\theta = 2 \cos^{-1}\left(\frac{220-160}{2 \times 306.075}\right)$

$= 168$ deg 45 min

Correction factors, $F_a = 1.00$, $F_b = 1.10$, $F_c = 0.81$ and $F_d = 0.97$

Equivalent pitch dia, $d_e = 160 \times 1.10$

$= 198.00$ mm

Belt speed, $S = \frac{\pi \times 160 \times 1800}{1000 \times 60}$

$= 15.079$ m/sec

Corresponding to $P = 3.97$ kw, $S = 15.079$ m/sec, and $d_e = 198.00$ mm, the rating of B section V belt (R) is 4.63.

No. of B section V belts required $= \frac{3.97 \times 1.0}{4.63 \times 0.87 \times 0.97}$

$= 1.016$

Hence a single B section V belt is selected as the normal power transmitted is less than the available rated power. The specification of the belt is B-1168/46-45. A pair of standard B section pulleys of 160 mm dia and 220 mm dia is selected for the transportation purpose.

APPENDIX IV

DESIGN OF CHAIN AND SPROCKET IN FIRST STAGE

Let the transmission ratio, (i) be 4 as discussed under 3.2.2. The number of teeth in the sprocket pinion (Z_1) is taken as 11 which is higher than the minimum required. Then, the number of teeth in sprocket wheel,

$$Z_2 = 4 \times 11$$
$$= 44$$

It is corrected to the next odd number 45. Hence the actual transmission ratio,

$$i = \frac{45}{11}$$
$$= 4.0909$$

Speed of rotation of sprocket pinion,

$$n_1 = \frac{1800 \times 80}{220}$$
$$= 654.545 \text{ rpm}$$

Speed of rotation of sprocket wheel,

$$n_2 = \frac{654.545}{4.0909}$$
$$= 160 \text{ rpm}$$

A standard chain with pitch (p), 12.700 mm is selected. Then the pitch dia of sprocket pinion,

$$d_{p1} = \frac{p}{\sin\left(\frac{180}{Z_1}\right)}$$
$$= \frac{12.700}{\sin\left(\frac{180}{11}\right)}$$
$$= 45.078 \text{ mm}$$

(Appendix IV continued)

Pitch dia of sprocket wheel,

$$d_{p2} = \frac{12.700}{\sin(\frac{180}{45})}$$

$$= 182.060 \text{ mm}$$

Tip dia of sprocket pinion,

$$d_{a1} = \frac{p}{\tan(\frac{180}{11})} + 0.6\ p$$

$$= \frac{12.700}{\tan(\frac{180}{11})} + 0.6\ (12.700)$$

$$= 50.871 \text{ mm}$$

Tip dia of sprocket wheel,

$$d_{a2} = \frac{12.700}{\tan(\frac{180}{45})} + 0.6\ (12.700)$$

$$= 189.237 \text{ mm}$$

Centre distance,

$$C_{min} = 1.3\ (\frac{d_{a1} + d_{a2}}{2})$$

$$= 1.3\ (\frac{50.871 + 189.237}{2})$$

$$= 156.0708 \text{ mm}$$

$$C_{max} = 80\ p$$

$$= 80 \times 12.700$$

$$= 1016 \text{ mm}$$

Let the initial centre, (C_o) be 175 mm. Then the approximate centre distance in multiple of pitch,

(Appendix IV continued)

$$C_p = \frac{C_o}{p}$$

$$= \frac{175}{12.700}$$

$$= 13.7795$$

The length of continuous chain in multiples of pitch,

$$l_p = 2\ C_p + \left(\frac{Z_1 + Z_2}{2}\right) + \left[\frac{(Z_2 - Z_1)/2\pi}{C_p}\right]^2$$

$$= 2 \times 13.7795 + \left(\frac{11+45}{2}\right) + \left[\frac{(45-11)/2\pi}{13.7795}\right]^2$$

$$= 57.6840$$

It is corrected to the next number, 58

The centre distance,

$$C = \frac{p}{h}\left[l_p - \frac{(Z_1+Z_2)}{2} + \sqrt{\left\{l_p - \left(\frac{(Z_1+Z_2)}{2}\right)\right\}^2 - 8\ \frac{(Z_2-Z_1)}{2\pi}}\right]$$

$$= \frac{12.700}{4}\left[58 - \frac{(11+45)}{2} + \sqrt{\left\{58 - \frac{(11+45)}{2}\right\}^2 - 8\ \frac{(45-11)}{2\pi}}\right]$$

$$= 177.1715 \text{ mm}$$

And hence the actual centre distance between counter shaft and intermediate shaft is 177.1715 mm.

For variable load with or without mild shock, $k_1 = 1.15$; for fixed centre distance, $k_2 = 1.25$; when $\frac{l_p}{Z_1+Z_2} = \frac{57.6840}{56}$ which is more than 1, $k_3 = 1.15$; for position of chain drive at more than 60 deg, $k_4 = 1.25$; for periodic lubrication

(Appendix IV continued)

k_5 = 1.5; for single shift of 8 hr duration, k_6 = 1.0 and hence the service factor $k_s = k_1 \times k_2 \times k_3 \times k_4 \times k_5 \times k_6$

$$= 3.662$$

Chain velocity, $v = \frac{\pi d p_1 \, n_1}{60}$

$$= \frac{\pi \times 45.078 \times 1309.09}{60 \times 1000}$$

$$= 3.0898 \text{ m/sec}$$

Hence ISO/DIN 084-1, R 1248 H chain is selected, for which the minimum breaking load, (Q) is 1600 kgf.

Factor of safety, $n = \frac{Qv}{102 \ P \ k_s}$

$$= \frac{1600 \times 3.0898}{102 \times 3.97 \times 3.662}$$

$$= 3.334$$

Check for actual factor of safety:

Tangential force due to power transmission when P is 5.4 hp,

$$P_t = \frac{75 \ P}{v}$$

$$= \frac{75 \times 5.4}{3.0898}$$

$$= 131.07644 \text{ kgf}$$

When the weight of chain, (w) is 0.58 kg/m, then centrifugal tension,

$$P_c = \frac{w \ v^2}{g}$$

$$= \frac{0.58 \ (3.0898)^2}{9.81}$$

$$= 0.5644 \text{ kgf}$$

(Appendix IV continued)

Coefficient of sag (k) is 1 when the chain is approximately vertical. Tension due to sagging of chain,

$$p_s = \frac{kwC}{1000}$$

$$= \frac{1 \times 0.58 \times 177.1715}{1000}$$

$$= 0.10276$$

Hence the actual factor of safety,

$$n = \frac{Q}{p_t + p_c + p_s}$$

$$= \frac{1600}{131.0764 + 0.5644 + 0.1028}$$

$$= 12.1448$$

The actual factor of safety, (n) is greater than the minimum required 9.50. Hence the design is safe.

Length of chain,

$$l = l_p \times p$$

$$= 58 \times 12.700$$

$$= 736.600 \text{ mm}.$$

APPENDIX V

SPECIFICATION OF ISO/DIN 084-1 ROLLEN R 1248 H CHAIN

Pitch	=	12.70 mm
Roller dia, max	=	7.75 mm
Width between inner plates	=	4.90 mm
Pin body dia, max	=	4.09 mm
Plate depth, max	=	11.10 mm
Bearing area	=	36.00 mm^2
Weight per meter	=	0.58 kgf
Breaking load, min	=	1600 kgf

APPENDIX VI

TEETH SPECIFICATIONS OF SPROCKET PINION AND SPROCKET WHEEL

Pitch	=	12.70 mm
Roller dia, max	=	7.75 mm
Width between inner plates	=	3.30 mm
Tooth width, min	=	2.79 mm
Tooth width, max	=	2.97 mm
Tooth side radius	=	12.70 mm
Side relief	=	0.89 mm
Shroud depth, min	=	2.16 mm
Shroud radius, max	=	0.76 mm

APPENDIX VII

DESIGN OF CHAIN AND SPROCKET IN SECOND STAGE

Let the transmission ratio (i) be 5.5 as discussed under 3.2.2. The number of teeth in the sprocket pinion, (Z_1) is taken as 11, which is higher than the minimum required. Hence the number of teeth in the sprocket wheel,

$$Z_2 = 5.5 \times 11$$
$$= 60.5$$

It is corrected to the even number of 60. The actual transmission ratio,

$$i = \frac{60}{11}$$
$$= 5.4545$$

The speed of rotation of sprocket pinion, n_1 is 160 rpm. Then the speed of rotation of sprocket wheel,

$$= \frac{160}{5.4545}$$
$$= 29.33 \text{ rpm}$$

Chain with pitch (p) 12.70 mm is selected.

Pitch dia of sprocket pinion,

$$d_{p1} = \frac{12.700}{\sin(\frac{180}{11})}$$

$$= 45.078 \text{ mm}$$

Pitch dia of sprocket wheel,

$$d_{p2} = \frac{12.700}{\sin(\frac{180}{60})}$$
$$= 242.663 \text{ mm}$$

(Appendix VII continued)

Tip dia of sprocket pinion,

$$d_{a1} = \frac{12.700}{\tan(\frac{180}{11})} + (0.6 \times 12.700)$$

$$= 50.871 \text{ mm}$$

Tip dia of sprocket wheel,

$$d_{a2} = \frac{12.700}{\tan(\frac{180}{60})} + (0.6 \times 12.700)$$

$$= 249.949 \text{ mm}$$

Centre distance for the transmission ratio 5.4545,

$$C_{min} = 1.3 \left(\frac{50.871 + 249.949}{2}\right)$$

$$= 195.533 \text{ mm}$$

$$C_{max} = 80 \times 12.7$$

$$= 1016 \text{ mm}$$

Let the initial centre distance (C_o) be 220 mm. Then the approximate centre distance in multiples of pitch,

$$C_p = \frac{220}{12.700}$$

$$= 17.3228$$

The length of continuous chain in multiples of pitch,

$$l_p = 2\,(17.3228) + \frac{(11+60)}{2} + \frac{\left(\frac{60-11}{2\pi}\right)^2}{17.3228}$$

$$= 73.6554$$

It is approximated to the even number, 74.

(Appendix VII continued)

Actual centre distance,

$$C = \frac{12.700}{4}\left[74-\frac{(11+60)}{2}+\sqrt{\left\{74-\left(\frac{11+60}{2}\right)\right\}^2-8\left(\frac{60-11}{2\pi}\right)^2}\right]$$

$$= 222.4240 \text{ mm}$$

And hence the actual centre distance between the intermediate shaft and final drive axle is 222.4240 mm

Service factor for the operating condition,

$$k_s = 2.92968$$

Chain velocity,

$$v = \frac{\pi \times 45.078 \times 1309.0909}{0.6 \times 1000}$$

$$= 3.0898 \text{ m/sec}$$

Hence the chain with specification, ISO/DIN 084-1 R 1248 H is selected, for which the minimum breaking load (Q) is 1600 kgf.

Factor of safety,

$$n = \frac{1600 \times 3.0898}{102 \times 3.97 \times 2.92968}$$

$$= 4.167$$

Check for actual factor of safety,

$$p_t = 131.07644 \text{ kgf}$$

$$p_c = 0.5644 \text{ kgf}$$

Coefficient of sag (k) is 4 when the chain is inclined 20 deg to the horizontal.

(Appendix VII continued)

Tension due to sagging of chain,

$$p_s = \frac{4 \times 0.58 \times 222.4240}{1000}$$

$$= 0.5160 \text{ kgf}$$

Hence the actual factor of safety,

$$n = \frac{1600}{131.07644 + 0.5644 + 0.5160}$$

$$= 12.106$$

It is higher than the minimum required factor of safety of 9.50.

Hence the design is safe.

Length of chain,

$$l = 74 \times 12.700$$

$$= 939.8 \text{ mm}.$$

APPENDIX VIII

FORCE AND MOMENT DISTRIBUTION ON COUNTERSHAFT

a. Vertical Force Diagram (Fig.22)

Vertical upward load at C due to belt drive and driven pulley,

$$F_{vc} = (107.429 \sin 13) - 4.250$$
$$= 19.916 \text{ kgf}$$

Vertical downward load at D due to chain drive,

$$F_{vd} = 301.997 \sin 84$$
$$= 300.342 \text{ kgf}$$

Taking moment about the point A, to get the vertical upward force acting at B, F_{vb}:

$$(19.916 \times 60) - (F_{vb} \times 100) + (300.342 \times \overset{130}{\cancel{300}}) = 0$$
$$F_{vb} = 402.394 \text{ kgf}$$

Hence the vertical downward force acting at A,

$$F_{va} = 19.916 + 402.394 - 300.342$$
$$= 121.968 \text{ kgf}$$

b. Vertical Bending Moment (BM_v) Diagram

BM_v at point A $= -19.916 \times 60$

$= -1194.960$ kg mm

BM_v at point B $= (-19.916 \times 160) + (121.968 \times 100)$

$= 9010.240$ kg mm

c. Horizontal Force Diagram

Horizontal force at C towards front end due to belt drive,

$$F_{hc} = 107.429 (\cos 13)$$
$$= 104.675 \text{ kgf}$$

(Appendix VIII continued)

Horizontal force at D towards front end due to chain drive,

$$F_{hd} = 301.997\ (\cos 84)$$
$$= 31.563 \text{ kgf}$$

Taking moment about the point A, to get the horizontal force acting towards front end at B, F_{hb}

$$(104.675 \times 60) - (F_{hb} \times 100) - (31.563 \times 130) = 0$$
$$F_{hb} = 21.773 \text{ kgf}$$

Hence the horizontal force acting towards rear end at A,

$$F_{ha} = 104.675 + 21.773 + 31.563$$
$$= 158.011 \text{ kgf}$$

d. Horizontal Bending Moment (BM_h) Diagram

BM_h at point A $= -104.675 \times 60$

$= -6280.50$ kg mm

BM_h at point B $= 31.563 \times 30$

$= 946.890$ kg mm

e. Resultant Bending Moment (M_r) Diagram

M_r at point A $= (1194.960^2 + 6280.500^2)^{\frac{1}{2}}$

$= 6393.1689$ kg mm

M_r at point B $= (9010.240^2 + 946.890^2)^{\frac{1}{2}}$

$= 9059.8579$ kg mm

f. Twisting Moment (M_t) Diagram

M_t at point C $= \dfrac{4500 \times 5.4 \times 1000 \times 220}{\pi \times 220 \times 654.545 \times 2}$

$= 5908.628$ kg mm

(Appendix VIII continued)

M_t at point D $= \dfrac{4500 \times 5.4 \times 1000 \times 45}{\pi \times 45 \times 654.545 \times 2}$

$= 5908.628$ kg mm

g. Equivalent Bending Moment (M_e) Diagram

M_e at point A $= \left[(1.5 \times 6393.169)^2 + (1.25 \times 5908.628)^2\right]^{\frac{1}{2}}$

$= 12104.265$ kg mm

M_e at point B $= \left[(1.5 \times 9059.858)^2 + (1.25 \times 5908.628)^2\right]^{\frac{1}{2}}$

$= 15469.132$ kg mm

M_e at point C $= 1.25 \times 5908.6$

$= 7385.787$ kg mm

M_e at point D $= 1.25 \times 5908.628$

$= 7385.787$ kg mm

APPENDIX IX

FORCE AND MOMENT DISTRIBUTION ON INTERMEDIATE SHAFT

a. Vertical Force Diagram (Fig.23)

Vertical downward force at C, due to the second chain drive,

$$F_{vc} = 1396.585 (\sin 20)$$
$$= 477.660 \text{ kg mm}$$

Vertical upward force at D due to the first chain drive,

$$F_{vd} = 305.466 (\sin 84)$$
$$= 303.792 \text{ kg mm}$$

b. Vertical Bending Moment (BM_v) Diagram

BM_v at point A $= 477.660 \times 32.5$

$= 15523.950$ kg mm

BM_v at point B $= 303.792 \times 32.5$

$= 9873.24$ kg mm

C. Horizontal Force Diagram

Horizontal force at C towards rear end,

$$F_{hc} = 1396.585 (\cos 20)$$
$$= 1312.360 \text{ kg mm}$$

Horizontal force at D towards rear end,

$$F_{hd} = 305.466 (\cos 84)$$
$$= 31.930 \text{ kg mm}$$

Taking moment about the point A, to get the horizontal force acting towards the front end at B, F_{hb}:

$$(-1312.360 \times 32.50) - (F_{hb} \times 200) + (31.930 \times 232.5) = 0$$
$$F_{hb} = 15.793 \text{ kg mm}$$

Hence, $F_{ha} = 1328.497$ kg mm which also acts towards the front end.

(Appendix IX continued)

d. Horizontal Bending Moment (BM_h) Diagram

BM_h at point A $= 1312.360 \times 32.5$

$= 42651.700$ kg mm

BM_h at point B $= 31.930 \times 32.5$

$= 1037.725$ kg mm

e. Resultant Bending Moment (M_r) Diagram

M_r at point A $= (15523.950^2 + 42651.700^2)^{\frac{1}{2}}$

$= 45388.991$ kg mm

M_r at point B $= (9873.240^2 + 1037.725^2)^{\frac{1}{2}}$

$= 9927.625$ kg mm

f. Twisting Moment (M_t) Diagram

M_t at point C $= \dfrac{4500 \times 5.4 \times 1000 \times 45}{\pi \times 160 \times 45 \times 2}$

$= 24171.675$ kg mm

M_t at point D $= \dfrac{4500 \times 5.4 \times 1000 \times 182}{\pi \times 160 \times 182 \times 2}$

$= 24171.675$ kg mm

g. Equivalent Bending Moment (M_e) Diagram

M_e at point A $= \left[(1.5 \times 45388.991)^2 + (1.25 \times 24171.675)^2\right]^{\frac{1}{2}}$

$= 71755.013$ kg mm

M_e at point B $= \left[(1.5 \times 9927.625)^2 + (1.25 \times 24171.675)^2\right]^{\frac{1}{2}}$

$= 33684.960$ kg mm

M_e at point C $= 1.25 \times 24171.675$

$= 30214.675$ kg mm

Appendix X

FORCE AND MOMENT DISTRIBUTION ON FINAL DRIVE AXLE

a. Vertical Force Diagram (Fig.24)

Vertical force acting at C upward due to the chain drive,

$$F_{vc} = 1413.178 (\sin 20)$$
$$= 483.335 \text{ kgf}$$

Vertical loads acting at the points A and B,

$$F_{va} = F_{vb} = 100 \text{ kgf}$$

To know the reactions acting vertically at the points D and E, moments are taken about the point D,

$$(-483.335 \times 207.5) + (100 \times 240) + (100 \times 440) + (F_{ve} \times 680) = 0$$

$$F_{ve} = \frac{(483.335 \times 207.5)-(24000)-(44000)}{680}$$
$$= 47.48825 \text{ kgf}$$

Hence,

$$F_{vd} = 235.846 \text{ kgf}$$

b. Vertical Bending Moment (BM_v) Diagram

BM_v at point C $= -(235.846 \times 207.5)$

$= -48938.045$ kg mm

BM_v at point A $= -(235.846 \times 240) + (483.335 \times 32.5)$

$= -40894.650$ kg mm

(Appendix X continued)

BM_v at point B $= -(235.846 \times 440) + (483.335 \times 232.5) + (100 \times 200)$

$= -11396.852$ kg mm

c. Horizontal Force Diagram

Horizontal force at C towards front end,

$$F_{hc} = 1413.178 (\cos 20)$$
$$= 1327.953 \text{ kgf}$$

The resultant horizontal force acting towards the rear end at the points A and B are taken as 40 kgf each.

Taking moments about the point B, to get the horizontal force acting at the point A,

$$-(1327.953 \times 232.5) + (F_{va} \times 200) = 0$$
$$F_{va} = 1543.745 \text{ kgf}$$

Hence,

$$F_{vb} = 215.792 \text{ kgf}$$

d. Horizontal Bending Moment (BM_h) Diagram

BM_h at point A $= 1327.953 \times 32.5$

$= 43158.472$ kg mm

e. Resultant Bending Moment (M_r) Diagram

M_r at point A $= (40894.650^2 + 43158.472^2)^{\frac{1}{2}}$

$= 59456.085$ kg mm

(Appendix X continued)

Similarly M_r at points B and C are calculated to be 11396.852 kg mm and 48938.045 kg mm respectively.

f. Twisting Moment (M_t) Diagram

$$M_t \text{ at point C} = \frac{4500 \times 5.4 \times 1000 \times 242.6}{29.33 \times \pi \times 242.6 \times 2}$$

$$= 131860.385 \text{ kg mm}$$

g. Equivalent Bending Moment (M_e) Diagram,

$$M_e \text{ at point A} = \left[(1.5 \times 59456.085)^2 + (1.25 \times 131860.385)^2\right]^{\frac{1}{2}}$$

$$= 187406.637 \text{ kg mm}$$

$$M_e \text{ at point B} = \left[(1.5 \times 11396.852)^2 + (1.25 \times 131860.385)^2\right]^{\frac{1}{2}}$$

$$= 165709.649 \text{ kg mm}$$

$$M_e \text{ at point C} = \left[(1.5 \times 48938.045)^2 + (1.25 \times 131860.385)^2\right]^{\frac{1}{2}}$$

$$= 180432.915 \text{ kg mm}$$

Appendix XI

SPECIFICATION OF BALL BEARINGS USED IN THE GARDEN TRACTOR

Sl. No.	Details	SKF 6204-2Z	SKF 3304-2Z	SKF 6207-2Z
1	Series	62	33	62
2	Inner dia, mm	20.0	20.0	35.0
3	Outer dia, mm	47.0	52.0	72.0
4	Abutment dia on shaft, mm	26.0	27.0	42.0
5	Abutment dia on housing, mm	47.0	45.0	65.0
6	Width, mm	14.0	22.2	17.0
7	Corner radii on shaft and housing, mm	1.5	2.0	2.0
8	Basic static capacity, kgf	655	1400	1370
9	Basic dynamic capacity, kgf	1000	1930	2000
10	Permissible speed, max. rpm	16000	8000	10000

Appendix XII

SPECIFICATION OF TYPE B LIGHT SERIES INTERNAL RETAINER CIRCLIPS

Sl. No.	Details	47 IS:3075-1965	52 IS: 3075-1965	72 IS: 3075-1965
1	Bore dia, mm	47.00	52.00	72.00
2	Seating dia, mm	49.50	55.00	75.00
3	Compressed dia, mm	33.20	37.60	55.40
4	Uncompressed dia, mm	50.50	56.20	76.50
5	Width, mm	4.40	4.70	6.40
6	Thickness, mm	1.75	2.00	2.50
7	Bore groove width, mm	1.85	2.15	2.65
8	Axial force, kgf	4720	6300	8650

Appendix XIII

SPECIFICATION OF SET COLLARS

(Type: Light series-A, Structural steel St 37)

Sl. No.	Specification	Counter-shaft	Intermediate shaft	Final axle drive
1	Nominal bore, mm	20.0	20.0	25.0
2	Outer dia, mm	32.0	32.0	40.0
3	Width, mm	14.0	14.0	16.0
4	Grub screw dia, mm	6.5	6.5	6.5
5	Grub screw length, mm	8.0	8.0	10.0

DEVELOPMENT OF A LOW COST GARDEN TRACTOR

BY
M. SIVASWAMI

ABSTRACT OF THESIS
SUBMITTED IN PARTIAL FULFILMENT OF
THE REQUIREMENTS FOR THE DEGREE
MASTER OF SCIENCE IN AGRICULTURAL ENGINEERING
FACULTY OF AGRICULTURE
KERALA AGRICULTURAL UNIVERSITY

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1982

ABSTRACT

The study was conducted with the objectives of systematic analysis of components with respect to their kinematics, dynamics and ergonomics and evaluation of traction performance and economics of low cost garden tractor.

A Greaves Lombardini 5.4 hp diesel engine with 1800 rpm was selected and a simple three step speed reduction system having a single stage V belt drive and double stage chain drives with a pivotted countershaft clutch have been designed. A road speed of 6.635 kmph and field speed of 3.317 kmph were achieved by using a cone pulley arrangement with 6.00 x 12 size wheel. Correct position of various components brings the centre of gravity of the unit with and without implement at very close to the final drive axle for easy balancing.

A simple overrunning clutch for differential action had been designed, developed and successfully field tested and all the controls were ergonomically located. Entire machine members had been mechanically designed and the unit was fabricated and successfully field tested for its functional and endurance requirements. The selected prime mover was found to be suitable even for starting the unit with all resistances.

The limiting stable angles in upward, downward and across the slope had been computed. The maximum gradability, drawbar pull and the minimum turning radius were studied and found acceptable for field and road requirements. The unit was found to work efficiently for upland operation in the pull range of 50 to 65 kgf.

The cost of the garden tractor was worked out to be around Rs.10,000/- and it has a break even point of 600 hr/year comparing that of 1000 hr/year for Kubota power tiller, which indicates its adaptability over the factory made garden tractors. The total weight of the garden tractor is 140 kg and it can easily be dismantled and assembled as prime mover (43.0 kg), transmission system (56.5 kg) and ground drive components (40.5 kg).

The unit can be fabricated by local technology from the readily available standard components and can successfully be maintained by small farmers.