DESIGN AND CONSTRUCTION OF TILLING MACHINE FOR SMALL SCALE FARMING

By

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ABSTRACT

The paper covers the design and construction of a soil cultivator for small scale farming. One of the very important assemblies is the drive assembly which is designed to make possible suitable tine design for soil cutting. An optimum tines attack angle of 20 degree is determined for the final design. It is shown that the final design is relatively insensitive to tines attack angles below 20 degrees and above 20 degrees. The tines tends to lock up in the soil at tine attack angles below 20 degrees while at tines attack angle above 20 degrees the tines act like a bulldozer. Local sourcing of materials and indigenous technique were employed. The machine is powered by a 1.5 kw petrol engine and is fabricated at a cost of thirty one and ninety naira, (₦31, 090:00) which is presumed affordable for small scale farming.

KEYWORDS: *Soil cultivator, v-belt drive, Chain drive, Tine, Reel, Assembly, Frame*

NOTATIONS

H_{max} = the maximum resisting shear force

A = the area of soil sheared

W = the normal loading on the soil

c = the soil/tine cohesion interface constant

γ = the tine's angle of attack during cutting.

F_{max} = the tight side span tension

F_{min} = the slack side span tension

ρ = the belt linear density

v = the pitch lne velocity of the belt

f = the effective coefficient of friction.

d = the center distance between pulleys

C = center distance between sprockets

L = the tangent length of belt

Lc = chain length

D_1, D_2 = the diameter of the driving and driven pulleys respectively

ϕ = the belt angle

α = contact arc.

r = the RPM of the small sprocket

R = the RPM of the large sprocket

n = the number of teeth on the small sprocket

1.0 INTRODUCTION

Land tilling is one of the important agricultural tasks facing small scale farming in Nigeria especially in the area of vegetable farming. Manual tilling methods' using spades, hoes, rakes and shovel are often laborious and limits the overall area cultivated by farmers. The need therefore exist for the development of equipment to improve he overall performance of the farmer with serious consideration given to the economic constraints of the design for cost effectiveness. The cases of animal drawn implement to do the tilling and weeding of farmland have existed in many years especially in the Asian countries. The capacity of the animals to provide the required tilling of the soil had also affected performance.

The tilling machine also known as soil cultivator is a power driven farming tool which is used for spading and turning over of the soil in preparation for farming. During tilling the cultivator break up the soil to preserve moisture and destroy the weeds. The ability of the cultivator to do the job effectively is a function of its weight, strength, type of tines and the soil characteristics. Heavy, powerful tilling equipment is most effective on stony clay soils, while smaller tilling equipment is more appropriate in a small garden or a garden with light soil. Kelpas et.al., (1994). Some equipment relating to soil tillage had been designed for incorporation of herbicides in soils; these include tandem disk harrow, drag harrows, and power take-off tillers, flexible shank cultivators and ground driving tillers as discussed in Barren tine, (1987). In comparative trials of these equipment by Juzwik et. al., (1997), maximum incorporation depth did not exceed 10 cm and uniformity of incorporation varied by equipment.

Design Description

The soil cultivator shown in figure 1 consists mainly of the tines unit, chain drive unit, belt drive unit and the power supply unit. The power supply unit is a petrol engine which supplies power to the belt drive unit. The torque from the belt drive unit is transmitted through a drive shaft to the chain drive unit which in turn provides the tines with rotational motion required for turning over the soil.

The objective of the project was to develop a soil cultivator for the purpose of tilling farmland. Thus the paper discusses the design and construction of the soil cultivator, which is suitable for small scale farming.

2.0 MATERIALS AND METHODS

The performance of the soil cultivator depends on the following:

- Soil characteristics
- Interface between the soil and the tines
- Tillage design parameters

The important tillage design parameters are:

- The tine angle of attack
- The tine support configuration

In order to limit space, the following sizes were decided.

- The pulley center distance of the v-belt drive unit should not exceed 350 mm.
- The drive shaft length with enough clearance between the shaft elements should not exceed 360 mm
- The input power range 1.0 – 1.5kw of a 4-cycle petrol engine.
- The soil cultivator is designed to occupy a space of 250 x 340 x 770 mm^2

The soil cultivator consist of the following sub-assemblies

(i) The Reel

(ii) Drive assembly

(iii) Body frame

2.1 **Reel Assembly**

The reel assembly is the part that does the actual cultivating. The reel comprises the tines, tine brackets, reel pipe, tine shaft, and reel bracket. Figure 2 shows the reel assembly. The tines are mounted on the tine brackets which in turn are welded on a reel pipe. The tines and bracket are made of cold rolled steel, the reel is a black iron pipe and the reel bracket is made from hot rolled steel.

The tines are the major feature responsible to subject the soil to cutting which causes shear force in the soil system due to the resistance of the soil as a result of its cohesion and internal friction between the soil particles. The resisting force required to shear a soil system is expressed in equation (1) as discussed in Sims, (2000).

$$H_{max} = cA + W\tan\gamma \qquad (1)$$

Where H_{max} is the maximum resisting shear force

A is the area of soil sheared

W is the normal loading on the soil

c is the soil/tine cohesion interface constant

γ is the tine's angle of attack during cutting.

Equation (1) shows that the tine surface area (A), and the tine attack angle (γ) are responsible for a lower shear force requirement to cut through the soil. The less the surface area (A) the less the shear force, and the lower the attack angle (γ) the lower the normal load (W) acting on it and consequently the lower the frictional component of the resisting shear (H). A study carried out by Smith and Sims, 1992 shows that the suitable range for the attack angle is between 10 – 50 degrees as shown in figure 3.

2.2 Drive Assembly

The drive assembly is made up of the V-belt drive unit, the belt tensioner, the drive shaft and pillow block, and the chain drive unit as shown in figure 4.

Belt Drive unit

The general process for designing the belt drive consists of selecting a safety factor of 1.3 for medium duty operation as discussed in Oliver, (1976), calculating the design power, and selecting the belt pitch, center distance, pulley combination, belt length, and belt width. The V-belt drive consists of the V-belt made of leather material, the driving and driven pulleys of cast iron materials. The driving pulley is keyed to the output shaft of a petrol engine while the driven pulley is keyed to the drive shaft mounted on a top plate with the aid of pillow blocks. The V-belt is mounted on the pulleys to transmit motion from the driven pulley to the chain drive unit. For the V-belt drive, an initial tension maintains the contact force between the belt and the surface of the pulley grooves so that the resulting friction assist in the transmission of the power. The initial tension also known as preload is provided in the belt drive by an idler which imposes a side load on one of the spans. In order for the drive to transmit the required torque, the belt wedged into the groove tightly enough to transmit the applied force.

The required initial tension is developed from three factors;

(a) the maximum torque to be transmitted by the drive
(b) the minimum arc of contact, and
(c) the pitch line velocity of the belt

The maximum torque to be transmitted is calculated from the resisting shear H_{max} From the defined maximum power and the physical dimensions shown in the sketch (figure 5) the relevant belt load can be calculated. The maximum tension ratio which the drive can support without slip on either of the two pulleys is expressed in equation (2) as in Oliver et. al., (1976).

$$(F_{max} - \rho v^2) / (F_{min} - \rho v^2) \leq \exp(f\theta)_{min} \qquad (2)$$

Where F_{max} is the tight side span tension

F_{min} is the slack side span tension

ρ is the belt linear density

v is the pitch line velocity of the belt

f is the effective coefficient of friction.

The power transmitted by the drive is expressed as

$$P = (F_{max} - F_{min})\, v \qquad (3)$$

F_{max} of the belt for a given slip limited power P is expressed as

$$F_{max} = \frac{P}{k_\theta v} + \rho v^2 \qquad (4)$$

Where $k_\theta = 1 - \exp(-f\theta)_{min}$ is the drive property.

In addition to the direct stress in the belt due to the tight side section tension, F_{max}, the belt pulley causes bending stresses which is inversely proportional to the pulley diameter. The overall effect of the belt loading is expressed by an equivalent damaging force F in equation (5) as discussed in Firbank, (1998) and South and Mancuso, (1994)

$$\left.\begin{aligned} F_1 &= F_{max} + \frac{M}{D_1} \\ F_2 &= F_{max} + \frac{M}{D_2} \end{aligned}\right\} \qquad (5)$$

Where M is a constant of proportionality, D_1, D_2 are the driving and driven pulley diameter respectively. If the life of the belt under the combined loading F_1, F_2 is Nc, the drive life, T of the belt in unit of time is T = NcL/v. the final design equation for the drive is thus expressed in equation (6).

$$\Sigma_{i=1,2}\left(\frac{P}{k_\theta v} + \frac{M}{D_i \rho v^2}\right) = F_{max}\left(\frac{L}{vT}\right) \qquad (6)$$

The tension in the slack side of the belt could be determined from equation (7) as in Raji, (2005).

$$\left.\begin{aligned} F_o &= \frac{1}{2}(F_{max} - F_{min}) \\ \frac{F_{max}}{F_{min}} &= \exp(f) \end{aligned}\right\} \qquad (7)$$

Where

f is the effective coefficient of friction defined as $\mu\theta$.

μ is the coefficient of friction between the belt and the pulley.

θ is the belt contact angle on the pulley.

The geometrical relationships applied to the design of the belt drive is expressed as follows

$$\left.\begin{aligned} \alpha_2 &= \pi + 2\phi \\ \alpha_1 &= \pi - 2\phi \\ \phi &= \sin^{-1}\left(\frac{D_2 - D_1}{2d}\right) \\ L &= [d^2 - (D_2 - D_1)] \end{aligned}\right\} \qquad (8)$$

Where d is the center distance between pulleys

L is the tangent length of belt

D_1, D_2 is the diameter of the driving and driven pulleys respectively

ϕ is the belt angle

α is contact arc.

Properties found by curve fitting to belt capacities cited in the code, are given in Table1.

An A1750 v-belt of horsepower range 1 – 5 is selected. The pulleys of D_1 = 90 mm and D_2 = 230 mm are set at center distance 270 mm apart as calculated. A 20 mm diameter shaft is designed for the drive shaft using the ASME codes for shaft design. (Hamrock et.al., 1999). The drive shaft is assembled on the top plate with the use of two pillow blocks and keyed to the driven pulley of the belt – drive used to transmit motion to the chain drive.

Chain Drive Unit

The chain drive is used to transmit power from the v-belt unit output pulley via the drive shaft to the tines shaft. This chain drive system is selected for the transmission in order to avoid slip because of the effect of the resistive force from the soil on the reel assembly. A vertical drive as shown in figure 4 is chosen for the design. When the chain engages the sprocket, the centers of the chain joints lies on the pitch circle of the sprocket. The center line of each link form a chord of the pitch circle. As the trailing roller enters the sprocket, it follows the leading roller along the cordial line. However, since the trailing roller is forced to follow the pitch circle of the sprocket, it is raised relative to its entry position. This cordial action initiates pulsations in the following links. The following procedure is considered for the chain drive design selection.

(a) The class of driven load is determined to be moderate shock loading.

(b) The safety factor, (SF) is selected from table 2 for the moderate shock application

(c) The design horsepower is calculated from equation 9

$$\text{Design Horsepower (DHP)} = \text{HP} \times \text{SF} \qquad (9)$$

Where HP is the motor horsepower

SF is the service factor

(d) A tentative chain selection is made from a selector chart (Roller chain manufacturer's handbook).

(e) The number of teeth for the small sprocket is selected. The minimum number of teeth required on the small sprocket in order to transmit the design horsepower is 10.

(f) The number of teeth for the large sprocket is determined from equation (10)

$$N = \frac{r}{R} \times n \qquad (10)$$

Where N is the number of teeth on large sprocket

r is the RPM of the small sprocket

R is the RPM of the large sprocket

n is the number of teeth on the small sprocket

(g) The minimum shaft distance is determined from equation (11).

$$C_{min} = \frac{2N + n}{6} \qquad (11)$$

(h) The chain length in pitches is determine from equation (12)

$$Lc = \frac{1}{2}(N + n) + 2C + \frac{K}{C} \qquad (12)$$

The value of K is selected from table 3.

2.3 Body frame Assembly

The body frame assembly consists of the main frame, the mounting rail, the top plate, the handle, and the wheel bracket and wheel. The body frame is made from a 38 mm angle iron cut to sizes and welded for permanent fastening. The mounting holes for the engine were drilled and positioned with the aid of a cardboard used to layout the engine mounting holes. Figure 6 shows the body frame assembly. The reel bracket and wheel bracket are fabricated from hot rolled steel, while the handles are fabricated from a thin-walled steel conduit.

3.0 PERFORMANCE ANALYSIS

The technical evaluation of the cultivator was carried out on a soil system and the performance compared using six sets of tines attack angles, 10^{O}, 15^{O}, 20^{O}, 25^{O}, 30^{O}, 40^{O}. Emphasis was laid on the moisture content of the soil system using the procedure discussed in Smith and Sims,(1992). An optimum attack angle of 20^{O} is preferred for the cultivator for optimum performance. This is evident as shown in figure 7. The optimum angle is considered ideal because at greater angle the reel tends to act like a bulldozer while at lower tines angle of attack the tines tends to lock in the soil. The depth of cut measured by a graduated rule in the soil system for the different tines attack angle tabulated in table 4 is depicted in figure 7.

The work time, field capacity and field efficiency are evaluated as discussed in RNAM, 1983 and Sims, (2000). Total work time is the time measured between the start of the first tilling and the end of the last tilling. It should include time for any breakdown and adjustment. The effective field capacity is expressed in equation (13).

$$FC_E = \frac{\text{(Total area cultivated in hectares)}}{\text{(Total work time)}} \quad (13)$$

The theoretical field capacity is expressed in equation (14) as,

$$FC_T = \text{(Working width)} \times \text{(Mean speed)} \quad (14)$$

Thus, the field efficiency is expressed as

$$FE\,(\%) = \frac{FC_E}{FC_T} \times 100 \quad (15)$$

The durability and reliability of the cultivator is currently being investigated.

Economic Appraisal

Two basic concepts of the economic appraisal was considered.

- Cost of construction of the cultivator
- Cost of Operation and Maintenance

Table 5 shows the type and cost of materials used for the machine.

The materials were sourced locally. The equipment operates efficiently at 7 litres of fuel per hectare of farm area at tines attack angle of 20^O.

CONCLUSION

The cultivator is ideal for tilling light soil and previously worked ground or allotments.

The equipment is powered by 1.5kw engine which delivers maximum torque to the tines when needed .The tines were evaluated for soil cutting. Incorporation of all the tines except that at attack angle of 20^O did not give a complete control of the soil tilling when compared to fuel consumption. The cultivator has been designed and constructed from locally sourced materials. The cultivator is suitable for use for small scale vegetable farming. The total cost of fabrication of the machine is thirty one thousand and ninety naira (N31,090:00) and is presumed affordable for small scale farming.

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Table1: BELT PROPERTIES (Corresponding to f = 0.512) (Source: Oliver, 1976)

Section	SPZ	SPA	SPB	SPC	Y	Z	A	B	C	D
F (N)	3730	6235	9057	16585	318	734	3216	5535	9842	20080
M (Nm)	34.04	87.48	182.0	555.1	0.587	3.045	23.93	62.72	174.4	618.5
ρ (kg/m)	0.0728	0.1287	0.2037	0.412	0.0396	0.0468	0.0968	0.16666	0.2958	0.604
m	12.8	13.0	13.4	13.8	11.11	11.11	11.11	11.11	11.11	11.11
Lo (mm)	1592	2278	3204	5070	466		1717	2266	3653	6112
Δ (mm)	-	-	-	-	6	8	11	14	19	24

Table 2: Service Factor for Chain Drive. (Source: Gates fact. Technical Information Library)

Type of Input Load				
Class of Driven Load	**Machines**	Internal Combustion engine with Hydraulic Drive	Eletric Motor or Turbine	Internal Combustion Engine with Mechanical Drive
Uniform	Belt conveyors with small load fluctuation, chain conveyors, centrifugal blowers, general textile machines, Machines with small load fluctuation.,	1.0	1.0	1.2
Moderate	Centrifugal compressors, marine engine, conveyors with some load fluctuation, automatic furnaces, dryers, pulverizes, general machine tools, compressors, general work machines, general paper mills	1.2	1.3	1.4
Heavy	Press, construction or mining machines, vibration machines, oil well rigs, rubber mixers, rolls, general machines with reverse or large impact loads.	1.4	1.5	1.7

Table 3: "K" values for calculating Chain Length (L)
(Source: Gates fact. Technical Information Library)

N - n	K	N - n	K	N - n	K	N - n	K	N - n	K	N - n	K	N - n	K
1	0.03	21	11.17	41	42.58	61	94.25	81	166.19	101	258.39	121	370.86
2	0.10	22	12.26	42	44.68	62	97.37	82	170.32	102	263.54	122	377.02
3	0.23	23	13.40	43	46.84	63	100.54	83	174.50	103	268.73	123	383.22
4	0.41	24	14.59	44	49.04	64	103.75	84	178.73	104	273.97	124	389.48
5	0.63	25	15.83	45	51.29	65	107.02	85	183.01	105	279.27	125	395.79
6	0.91	26	17.12	46	53.60	66	110.34	86	187.34	106	284.67	126	402.14
7	1.24	27	18.47	47	55.95	67	113.71	87	191.73	107	290.01	127	408.55
8	1.62	28	19.86	48	58.36	68	117.13	88	196.16	108	295.45	128	415.01
9	2.05	29	21.30	49	60.82	69	120.60	89	200.64	109	300.95	129	421.52
10	2.53	30	22.8	50	63.33	70	124.12	90	205.18	110	306.5	130	428.08
11	3.06	31	24.34	51	65.88	71	127.69	91	209.74	111	312.09	131	434.69
12	3.65	32	25.94	52	68.49	72	131.31	92	214.40	112	317.74	132	441.36
13	4.28	33	27.58	53	71.15	73	134.99	93	219.08	113	323.44	133	448.07
14	4.96	34	29.28	54	73.86	74	138.71	94	223.82	114	329.19	134	454.83
15	5.7	35	31.03	55	76.62	75	142.48	95	228.61	115	334.99	135	461.64
16	6.48	36	32.83	56	79.44	76	146.31	96	233.44	116	340.84	136	468.51
17	7.32	37	34.68	57	82.30	77	150.18	97	238.33	117	346.75	137	475.42
18	8.21	38	36.58	58	85.21	78	154.11	98	243.27	118	352.70	138	482.39
19	9.14	39	38.53	59	88.17	79	158.09	99	248.26	119	358.70	139	489.41
20	10.13	40	40.53	60	91.19	80	162.11	100	253.30	120	364.76	140	496.47

Table 4: Performance Evaluation Result

Attack angle of tines, Degrees	Depth of cut (mm)	Fuel Consuption/Hectares (ltrs)
10	2	0.5
15	6	0.5
20	11	7
25	17	20
30	30	35
40	60	65

Table 5: Material Cost of Equipment

S/n	Item	Size	Qty	Cost/unit (#)	Total Cost (#)
1	38 mm angle iron	370 mm	2	680	1360
2	Cold rolled steel	3 x 25 x 50 mm	4	1010	4040
3	Hot rolled steel	6 x 100 x 500	1	930	930
4	Cold rolled steel	ϕ25 mm x 400 mm	2	980	1960
5	Bore bearing	25 mm	2	450	900
6	Sprocket	25 teeth, 10 teeth	2	410	820
7	Roller Chain	685 mm long	1	1400	1400
8	Machine screws &nuts	various	56	40	2240
9	Gasoline engine	1.0 – 1.5 kw.	1	13500	13500
10	v-belt (leather)	A1750	1	840	840
11	v- grooved pulleys	2	2	350	700
12	Rubber tires	ϕ 25 mm	2	1200	2400
				Total	**31090**

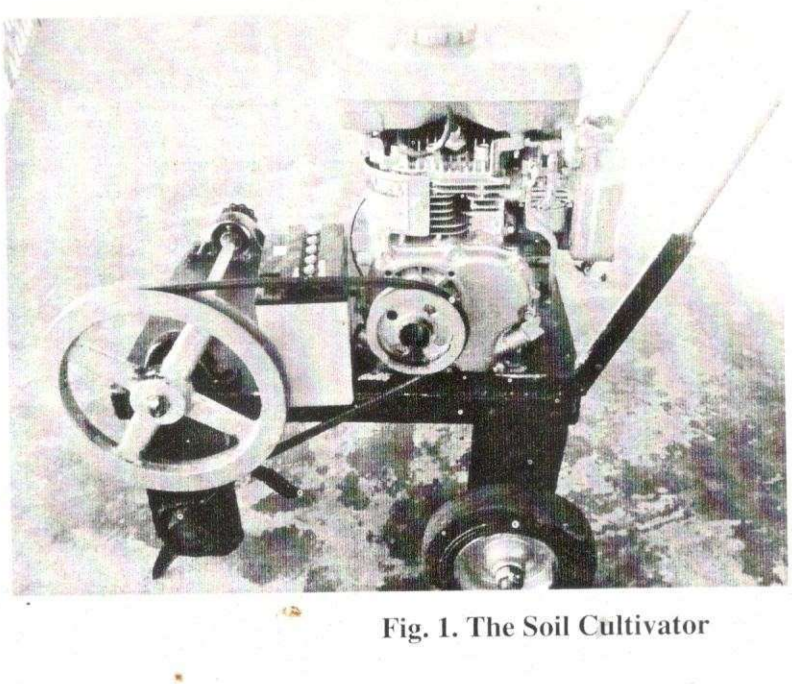


Fig. 1. The Soil Cultivator

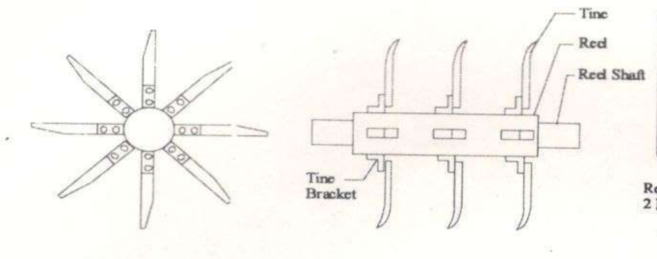
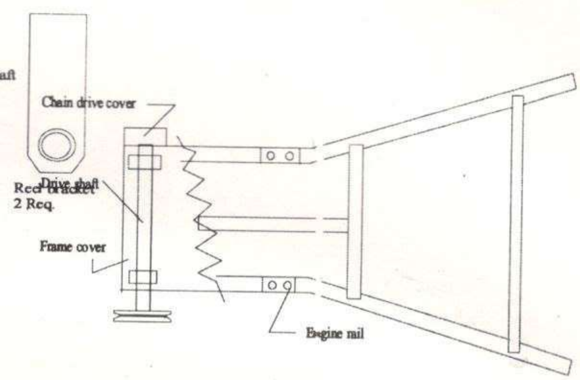


Fig. 2: Schematic Drawing of Reel Assembly



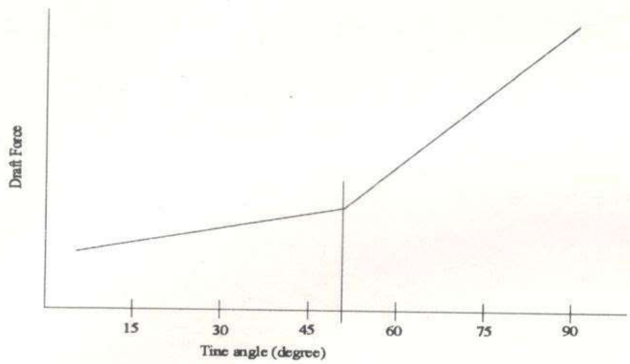


Fig. 3. Effect of tine angle on draft force.
Source: Sims, 1992.

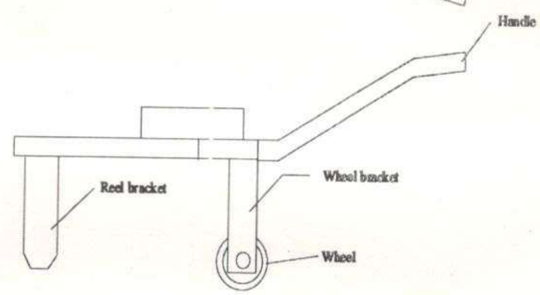
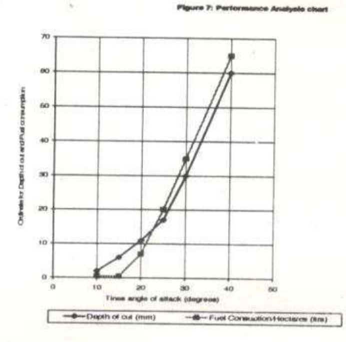


Fig. 6. Body Frame Assembly

Figure 7: Performance Analysis chart



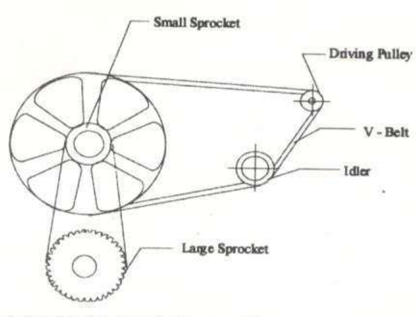


Fig. 4. Schematic Drawing of the Drive Assembly

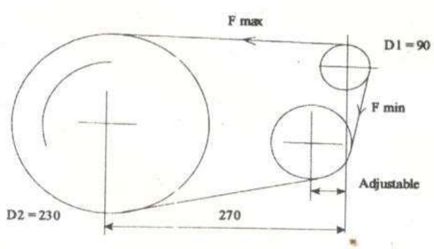


Fig. 5. V-bet drive at maximum power