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Investigation on inertia propulsion concept

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This thesis is submitted in partial submission for M. Sc. degree.

In 1963, R.B. Lindsay remarked,

> "..... if history shows anything at all,
> it is that man has always striven to
> enlarge the scope of his experience"
> never content with what he already knows.

this piece of work demonstrates that his remark is equally valid in 1980.

SUMMARY.

The objective of the study was to investigate the possibility of using the Inertia. Force as a basis for propulsion rather than the conventional ground/fluid traction.

A model inertia equipment consisting of contra-rotating m?,3es mechanism was developed and tested.

The model dragged itself on different surfaces and also pulled external load demonstrating that the concept of inertia propulsion is feasible.

However, the analysis of the system performance indicates need for further investigatory study on the concept before designing bigger scale model for studying its application.

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And above all, God for everything.

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LIST OF SYMBOLS

SYMBOL	QUANTITY
d :	shafts diameter (m)
E :	Youngs Modulus of Elasticity ($N.\ m^{-2}$)
	(mild steel $E = 207 \times 10^7\ N.m^{-2}$)
f :	Linear acceleration ($m.S^{-2}$)
F, P, R, :	force (N)
g :	gravitational acceleration ($g = 9.81\ m.s^{-2}$)
G :	Modulus of Rigidity ($N\ m^{-2}$)
	(mild steel $G = 82.8 \times 10^9\ n.\ m^{-2}$)
I :	Second Moment of Area (M^4)
Ir, Is :	Mass moment of rotor and shaft respectively ($kg.\ m^2$)
Kt :	Torsional stiffness (N.m)
L :	Beam Length
M, m :	Mass (kg)
r :	eccentric distance (m)
t :	time (sec)
v :	Linear velocity ($m.s^{-1}$)
w :	angular velocity ($rad.\ s^{-1}$)
Z :	section modulus (m^3)
θ :	angular displacement (rad)
λ :	dimension less constant
μ :	coefficient of frictions
σ :	stress ($N.\ m^{-2}$)
ρ :	density ($kg.\ m^{-3}$)
	(ρ = 7840 kg/m^3 for mild steel)

1. INTRODUCTION

In terms of the concept of energy, what the growth of Science and Technology has done is to increase the number of different transformations of energy available to man in the usable state and place. These have involved the development of both conversion and transmission system of energy of which propulsion devices form only a part.

Normally, propulsion devices react against media such as soil, water, air in order to produce work or motion to vehicle. For example an aeroplane or boat is moved by a propeller by traction in a fluid medium, while an automobile is propelled by traction between the wheels and the ground; most of the agricultural propulsion devices belong to this latter group. The Draw Bar Pull is dependent on the wheel slippage. Lower wheel slip values reduce the pull developed, while higher values lead to higher pull but reduced traction efficiency.. Generally, the maximum Draw Bar Power developed for an average conventional tractor occurs at slip value of 15 - 20 per cent (DWYER, 1974). Unfortunately, farming operations are often carried out in conditions of high slip values, so high that only half of the available engine power is converted to Draw Bar Power. Several methods are in use in order to increase the traction efficiency such as ballasting i.e. adding weights especially in frictional soils. This method like others has its limitations such as:

i) the manufacturer's recommended tyre/wheel load which must not be exceeded.

ii) The physical soil conditions after the operations may be undesirable for optimum plant growth and development.

Addition of weights increases soil compaction which may result in poor aeration, lower water infiltration and drainage rates, reduced water holding capacity and greater mechanical impedance to

plants. (HENDRICK et al 1963). On the other hand it may not be economical to acquire a bigger tractor with larger wheels so as to avoid the above problems arising from ballasting.

Another alternative method of power transmission to soil engaging tool such as vibratory and rotary tools is via the power-take-off shaft of the tractor. Works done on these powered tools so far indicated that although there was reduction in both soil compaction and draw bar pull, the total energy requirements were either identical or higher than with non-vibrating tools. Venter (1968) reported that reductions in draw bar pull of up to 80 per cent were possible but that the total energy requirements could be increased by as much as 300 per cent.

In view of the serious limitations of these existing propulsion devices, the benfits of a more efficient means of propulsion can be appreciated. One such device is the Inertia Propulsion system.

The Inertia Propulsion device does not use the traditional media for its propulsion thereby eliminating wheel slippage loss. In this device a repeated force is produced for propulsion through the production of a force between the inertia member and the body and the inertia member reacts directly with the body.

Available literatures indicate that although some works had been done on inertia propulsion system (FARRALL 1965, 1966, LAWYER 1973, CHANCELIOR 1974) there is still more work to be done on the topic before the concept is either rejected or commercialised.

Therefore, this project was concerned with investigating the possibility of using the force of inertia as a basis for propulsion system.

This involved constructing an inertia propulsion device which would be simple in construction, positive in operation and economical in manufacture.

In order to effectively carry out this study the work has been done in the following sequence:

i) a critical appraisal of the works already done on the subject.

ii) design and construction of model propulsion equipment.

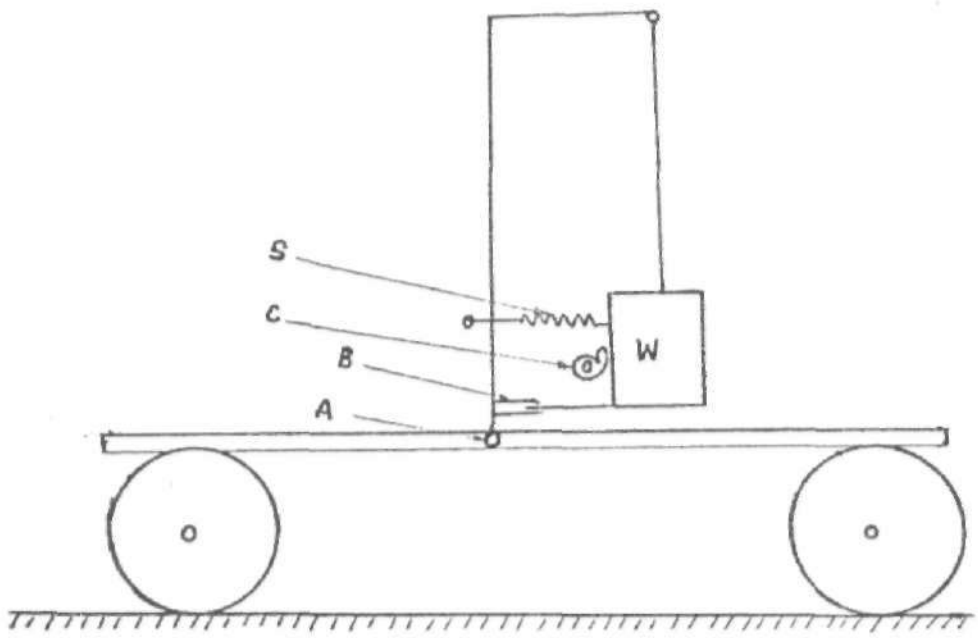
iii) testing of the equipment.

2. REVIEW OF LITERATURES

Since the patency of Inertia Propulsion by Farrell (1966) some efforts have been made to apply the concept on soil engaging implements (LAWYER 1973, and CHANCELLOR 1974). Brief summary of the works by these authors is given below.

2.1 Invention of Inertia Propulsion Device

Farrall, (1966), one of the inventors of Inertia Propulsion Device; termed his invention "Reaction Impulse Device". He demonstrated it with a heavy pendulum as the inertia mass connected to the main body by springs in which energy was intermittently stored by tension/compression when the pendulum was moved from the vertical position. Figure 2.1.



A: Vehicle centre of gravity.

W: Inertia member

S: Spring or Hydraulic device for rapidly moving W toward A.

C: Cam for moving W away from A

B: Snubber for decellerating movement of W toward A

Fig. 2.1 An Inertia Propulsion System. (Farrall 1966)

Any means could be provided for moving the Inertia member, and Farrall proposed the use of trigger mechanism operated by either of the following:

i) an electric motor through a cam system

ii) pressurised fluid in a cylinder on to a piston.

iii) or energy produced from an exloding gas using compression ignition system.

When the pendulum was released the weight of it became a free body so that energy in the spring could react between the pendulum and the body to provide force in the whole body. By this arrangement, any object could be moved in any direction through the inertia force on the pendum which would be substantially independent of the support of the body. The amount of the work which the device could perform depended on the weight of the inertia member, the weight of the body and the velocity and acceleration of the inertia member to produce a sharp impulse of short duration to the body. The use of plurality of pendulums, severally released would produce a relatively smooth force in the direction of, or opposite to the movement of the released pendum.

Farrall's demonstration showed that Inertia Propulsion would work. He did not suggest the device as a substitute for conventional propulsion systems when they could be used effectively; however, the inertia propulsion device could be used as a auxilliary unit, providing propulsive effort under conditions of poor traction for wheel vehicles.

Later works by Lawyer et al (1973) indicated that the efficiency of energy transfer that could be expected with such a system was in the range of 20 to 40 per cent.

2.2 Application of Inertia Propulsion System.

The application of Inertia Propulsion system directly to soil-engaging implement was done by Bodine (Lawyer et al 1973), followed by Lawyer

et al (1973). The design used by the latter author consisted of a hammer, mounted on one leg of a tuning fork and an oscillator on the other side, Fig. 2.2. The tuning fork was then attached to the soil-engaging blade assembly with the hammer biased against an anvil mounted on the rear of the blade just above the level reached by the soil surface during operation.

The operation of the system is similar to a system in which a person stands on a block of wood using a heavy hammer, hammers the block along the floor. The propulsion force/torque was generated by the oscillatory motion of the oscillator, delivering repeated impulses at frequent intervals on the subsoiler via the hammer.

With the assembly, Fig 2.2, mounted on tractor; its performance tests indicated that a very large proportion of the motion of the soil cutting blade involved movement back and forth in previously cut soil. The system, therefore, functioned as an oscillating tillage tool rather than a pure inertia propulsion tool.

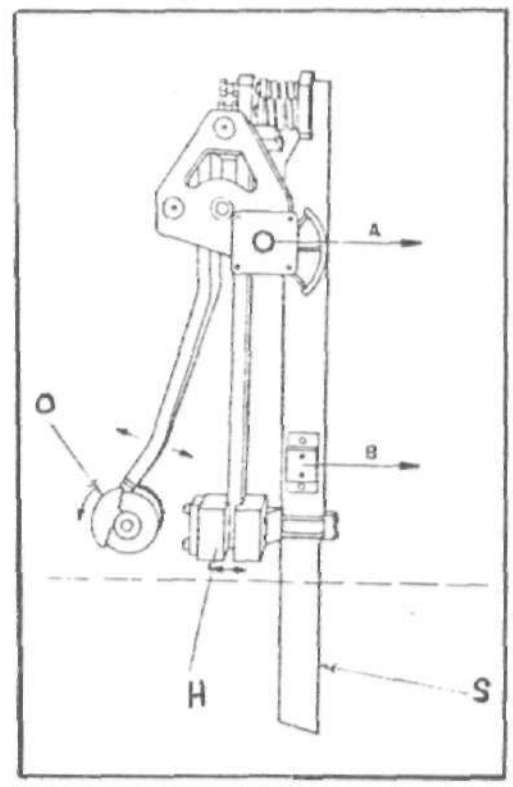
Later Chancellor (1974) modified the above Lawyer's design and used it as both :

i) a resonant oscillating tool system and

ii) an inertia propelled tool system.

He then did comparative performance and efficiency studies of these two modified forms as against a non-oscillating tool system. Chancellor's concluding remarks included the following:

> "Under circumstances in which the oscillator power applied were equal or higher than non oscillating draughtpower, the inertia propulsion system achieved greater draughtreduction and slightly higher power efficiency than the resonant tool system. These were attributed to the fact that the portion of the proportion of the period of each cycle during which



O: oscillator

H: Hammer

S: Subsoiler

Fig. 2.2. Sketch of the subsoiler blade and inertia propulsion drive (Lawyer et al 1973)

soil cutting was actually taking place, tended to be smaller in the inertia propulsion system than resonant oscillating system because of the very high tool velocities achieved by hammer impact by the former. Also the inertia propulsion tool did not waste any power in any rearward movement into the previously cut soil. The energy lost due to impact by the inertia system seemed to be less than the frictional energy lost by the oscillating tool in its rearward movement".

However the inertia propulsion system was more complex in construction and was more subject to fracture

All the above works indicated that inertia propulsion concept is workable. The system achieved greater draught reduction and power efficiency than both non-oscillating and oscillating tools, but there was no information on the soil physical conditions after the operations and the eventual effects on plant growth, development and yield.

The above were the relevant publications on Inertia Propulsion Concept that could be retrieved using all the college facilities with the able assistance of the Librarian; and from the references quoted by the various authors, it seemed that at least most of the important relevant paper were obtained.

Therefore from this rather limited information on inertia propulsion concept and from Chancellor's concluding remark, it was apparent that more works should be done on both investigation and application of Inertia Propulsion Concept.

This literature review, therefore, further strengthe the purpose for which this project was set up, i.e. to investigate the possibility of using inertia as a basis for propulsion system, through construction of a simple economical, and positive acting model equipment.

3. DESIGN PROCEDURE.

3.1 THEORY

3.1.1 Definition of the basic design concept.

The basic design concept involved a propulsion system mounted in a simple vehicle. The propulsion would consist of mass which oscillate but constrained in such a way that the acceleration over part of a cycle would greatly exceed that in the remaining part of the cycle - See fig. 3.1.

In this way a large inertia force capable of driving the vehicle forward in one direction only would be developed.

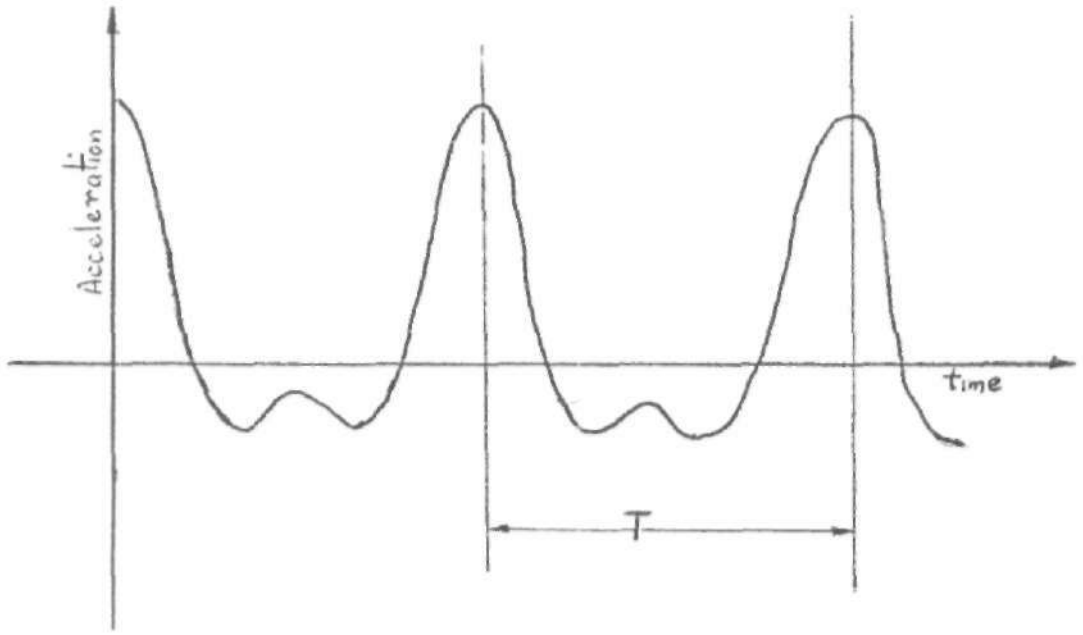


Fig. 3.1 Hypothetical Required Acceleration Waveform.

3.1.2 Alternative Systems Considered.

With the above basic design concept in mind; the following systems were considered:

a) Trigger Mechanism. This could be in different forms such as a cam operated compression/tension spring loaded inertia member, which when released would become a free body; providing the required inertia force. Alternatively the inertia member could be propelled by air pressure to produce the same required inertia force. Farrall (1966) adopted this system. Fig. 2.1.

b) Quick Return Mechanism. It was initially thought that the Quick Return Mechanism would give the required inertia force. However a detailed analysis of this mechanism revealed that the system would give equal acceleration force in opposite directions and it would not meet the requirement.

c) Contra-Rotating Masses Mechanism. This mechanism would consist of series of pairs of eccentric masses; each pair rotated at same speed but opposite direction, while the speed of each pair were in the ratios 1 : 2 : 3 : 4 :------ Analysis indicated that the continued net centrifugal force of the system would produce the required inertia force.

Out of the above alternatives, the author used the Contra-Rotating Masses system for designing the model apparatus. The details of this system now follows.

3.1.3 Fourier Analysis of basic design concept.

From section 3.1.1, the basic design required that the acceleration/force over part of a cycle should greatly exceed that in the remaining part of the cycle; such that a large force capable of driving the whole vehicle in one direction only would be developed. An hypothetical required wave form is as shown on fig. 3.1.

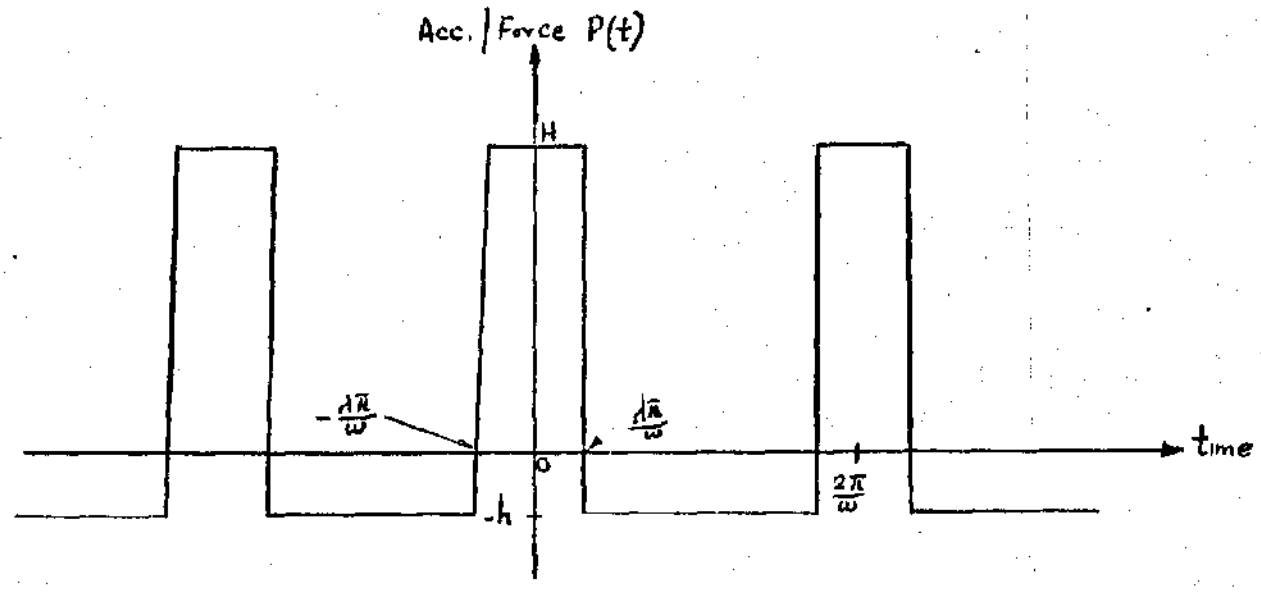


Fig. 3.2. Hypothetical Force/Acceleration Wave Form.

Now any wave form can by Fourier analysis be represented as the summation of a series of Cosine and sine waves. Based on the above design criteria, consider an hypothetical force/acceleration wave form shown in fig. 3.2.

From the above wave form

$$P(t) = H \text{ for } -\frac{\lambda\pi}{w} \leq t \leq \frac{\lambda\pi}{w} \qquad (1)$$

$$P(t) = -h \text{ for } \frac{\lambda\pi}{w} \leq t \leq \left(\frac{2\pi}{w} - \frac{\lambda\pi}{w}\right) \qquad (2)$$

The Fourier equation of the above wave for and as defined by equations (1) and (2) is

$$P(t) = \frac{a_o}{2} + \sum_{n=1}^{n=\infty} \frac{2(H+h)\ \mathrm{Sin}\,(n\lambda\pi)\ \mathrm{Cos}\,nwt}{n\pi}$$

$$= \frac{a_o}{2} + \frac{2(H+h)\ \mathrm{Sin}(\lambda\pi)}{\pi}\ \mathrm{Cos}\ wt$$

$$+ \frac{2(H+h)\ \mathrm{Sin}\,(2\lambda\pi)}{2\pi}\ \mathrm{Cos}\ 2\ wt$$

$$+ \frac{2(H+h)\ \mathrm{Sin}\,(3\lambda\pi)}{3\pi}\ \mathrm{Cos}\ 3\ wt$$

$$+ \frac{2(H+h)\ \mathrm{Sin}\,(4\lambda\pi)}{4\pi}\ \mathrm{Cos}\ 4\ wt$$

$$+ \text{- - - - - - - - - - - - - -} \qquad (4)$$

See Appendix A1 for alculation of above fourier coefficients. Now for the above Fourier equation to be convergent, one of the conditions that should be satisfied is that P(t) should be periodic with period 2π (Spencer, 1977) i.e. the net displacement of a mass suffering force P(t) over one cycle should be zero. The acceleration is proportional to the force P(t) in the absence of external force, although this is not true in practice. However, under zero external force; the conditions for zero net displacement is:

$$\frac{H}{h} = \frac{1-\lambda}{\lambda} \text{ - - - - - - - - - - - - - - - - - -} \qquad (5)$$

See detail of derivation of (5) in Appendix A2.

Then (4) becomes :

$$P(t) = \frac{a_o}{2} + \frac{2h}{\lambda\pi}\left[\mathrm{Sin}\,(\lambda\pi)\ \mathrm{Cos}\ wt + \frac{\mathrm{Sin}(2\lambda\pi)}{2}\ \mathrm{Cos}\ 2\ wt\right.$$

$$\left. + \frac{\mathrm{Sin}\,(3\lambda\pi)}{3}\ \mathrm{Cos}\ 3wt + \frac{\mathrm{Sin}\,(4\lambda\pi)}{4}\ \mathrm{Cos}\ 4\ wt\right.$$

$$+ \text{- - - - - - - - - - - - - - - - - -} \qquad (6)$$

Assigning different values to λ, the value of λ that gave the best waveform was 0.064, and its corresponding Fourier equation from (b) was

$$P(t) = \frac{a_o}{2} + \frac{2h}{\lambda\pi}(0.199\ \mathrm{Cos}\ wt + 0.195\ \mathrm{Cos}\ 2\ wt + 0.188\ \mathrm{Cos}\ 3\ wt$$

$$+ 0.179\ \mathrm{Cos}\ 4\ wt + \text{- - - - - - - - - - - -}) \qquad (7)$$

and

$$a_o = 0$$

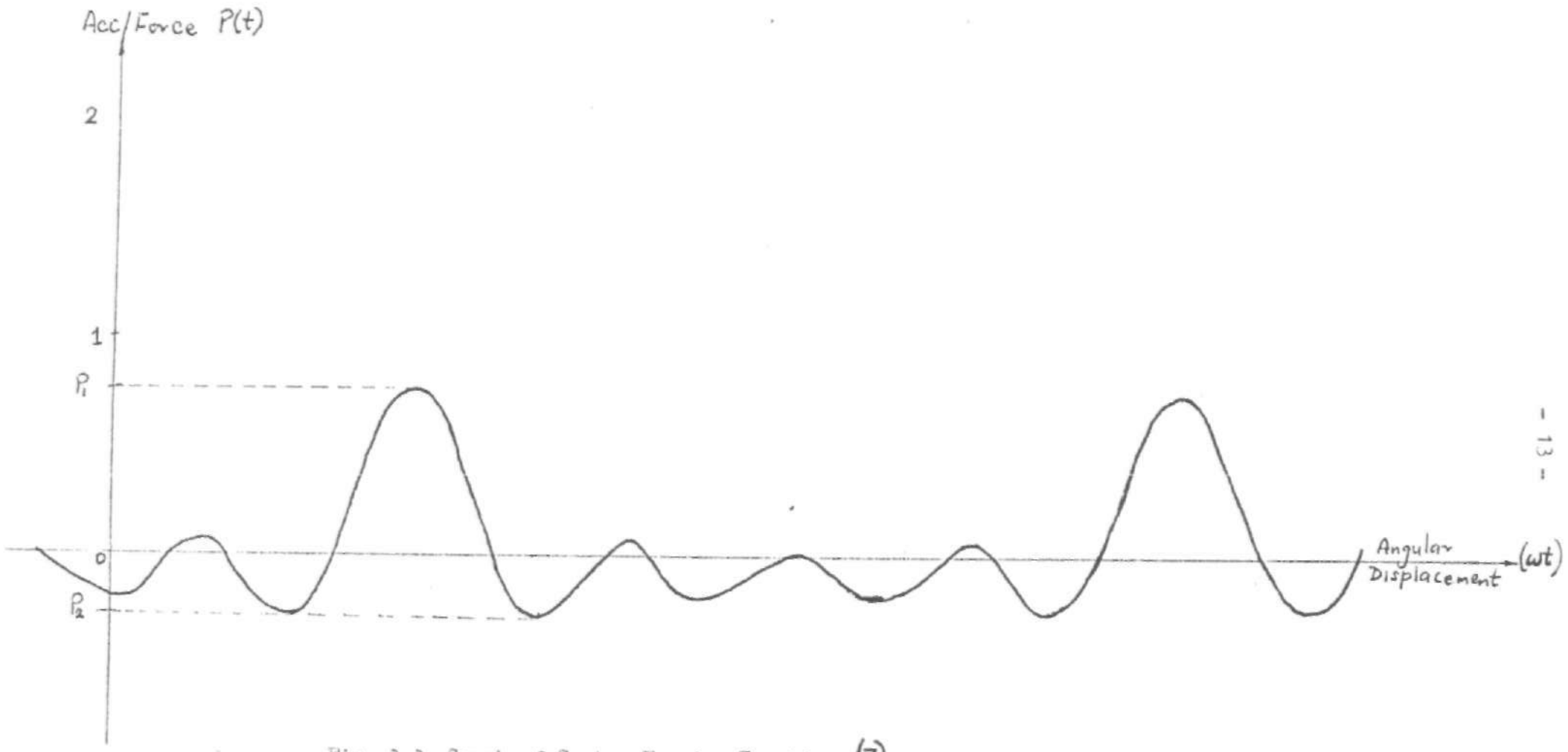


Fig. 3.3 Graph of System Fourier Equation (7)

3.1.4. Analysis of system of Contra-Rotating Masses Mechanism

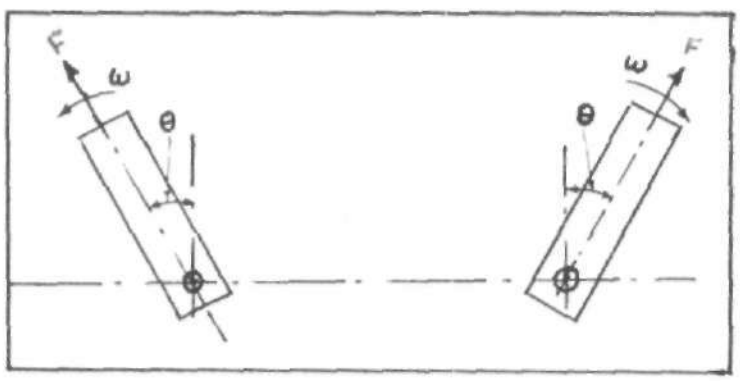


Fig. 3.4 Pair of Contra-Rotating Masses.

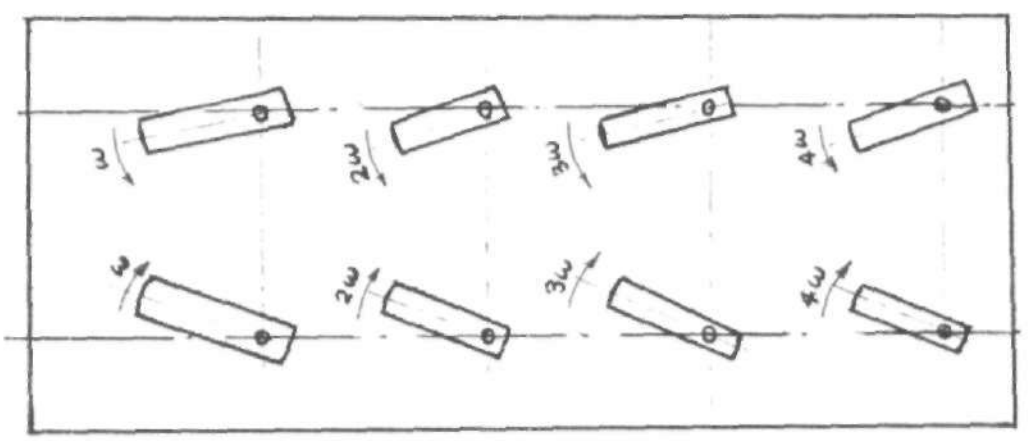


Fig. 3.5 Series of Contra-Rotating Masses.

Consider a pair of eccentric masses of equal mass; equal eccentric distance r; and equal but opposite angular velocity w, rotating as shown in Fig. 3.4
If the axis remain stationary, the combined effect of the rotating pair of masses will generate a net force given by

$$2 F \cos \theta$$

where $F = mrw^2$

$\theta = wt$

The direction of this force acting on the bearings will be perpendicular to the line joining the shafts if masses are in line when perpendicular to it.

Now consider a series of pairs of contra-rotating masses as shown in fig. 3.5. If the angular speed of the pairs are in the ratio 1: 2: 3: 4: ---- then the combine net force at any time t will be

$$F(t) = m_1 r_1 w^2 \cos wt + m_2 r_2 (2w)^2 \cos 2wt + m_3 r_3 (3w)^2 \cos 3wt + m_4 r_4 (4w)^2 \cos 4wt + - - - \quad (8)$$

$$F(t) = m_1 r_1 w^2 \cos wt + 4 m_2 r_2 w^2 \cos 2wt + 9 m_3 r_3 w^2 \cos 3wt + 16 m_4 r_4 w^2 \cos 4wt + - - - - - - - - - - - - \quad (9)$$

Comparing (4) and (8) then the net force of a system of series of pair of contra-rotating masses can, therefore, be represented by a Fourier equation of the form in (4). Then using (7) and (8) and equating the coefficients :

$$\left.\begin{aligned} m_1 r_1 w^2 &= K \times 0.199 \\ 4m_2 r_2 w^2 &= K \times 0.195 \\ 9m_3 r_3 w^2 &= K \times 0.188 \\ 16m_4 r_4 w^2 &= K \times 0.179 \end{aligned}\right\} \quad (9)$$

from (9)

$$\left.\begin{aligned} m_2 r_2 &= 0.245\, m_1 r_1 \\ m_3 r_3 &= 0.105\, m_1 r_1 \\ m_4 r_4 &= 0.07\, m_1 r_1 \end{aligned}\right\} \quad (10)$$

3.2 DESIGN OF MODEL INERTIA EQUIPMENT.

3.2.1. Assumptions and material selection

A draft sketch and calculation on the model inertia equipment indicated that, the greater the mass of the model, the greater would be

the size of the rotating masses and also the power required to drive the rotating masses. The mass of the model equipment exoluding the rotating masses was estimated as 25 kg. The eccentric distance r, as contained in (8) - (10) were estimated to be as following:

$$\left.\begin{aligned} r_1 = r_2 = r_3 &= 10 \text{ mm} \\ r_4 &= 4 \text{ mm} \end{aligned}\right\} \quad \text{- - - - - - - - - - (11)}$$

The detail of calculation of the eccentric distance is contained in Appendix B.

then from (10) and (11)

$$\left.\begin{aligned} m_2 &= 0.245\ m_1 \\ m_3 &= 0.105\ m_1 \\ m_4 &= 0.14\ m_1 \end{aligned}\right\} \qquad (12)$$

and the total rotating masses interm of m_1 would be

$$m_1 (1 + 0.245 + 0.105 + 0.14) = 1.49 m_1$$

and the total mass of the model became

$$m = (25 + 1.49\ m_1) \quad \text{- - - - - - - - - - - - - - (13)}$$

Based on the available electric motor and reduction gear box; the estimated range of the operating relating speed (w) was 7 - 15 revs per second.

The coefficient of friction of the soil in the proposed test site was estimated to be 0.414.

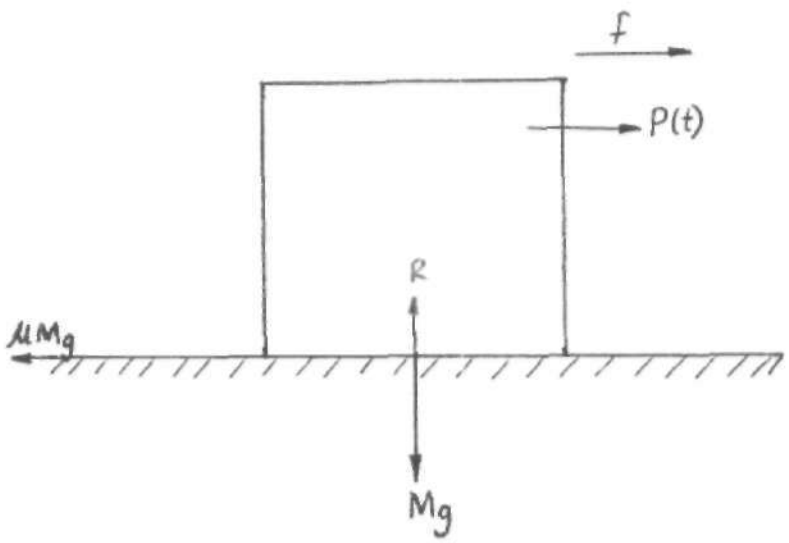
The following factors were considered in material selection.

(i) WEIGHT the two materials used were mild-steel and aluminium; aluminium generally preferred where strength permitted.

ii) STRENGTH as much as system analysis permitted, the selected materials had sufficient margin of safety. See Section 3.2.3.

iii) COST To reduce costs available materials were always first considered, e.g. the available electric motor was used; and the most economical methods of manufacture were specified.

3.2.2 Estimation of the size of the Eccentric Masses.



f: direction of motion
P (t): Inertia Force
Mg: Weight of model Inertia equipment
μMg: Frictional Resistance
R: Reaction

Fig. 3.6 Model Inertia Equipment in Motion.

Consider a model Inertia equipment shown in Fig. 3.6 of total mass M, suffering inertia force P(t) and frictional resistance μMg. Fig. 3.3 is the waveform of the external force P(t) and

$$\frac{P_1}{P_2} = 3 \text{ - - - - - - - - - - - - - - - - (14) from fig 3.3}$$

Now for motion to occur in one direction only.

$$P_{max} = P_1 > \mu mg \quad \text{- - - - - - - - - - - - (15)}$$

$$P_{min} = P_2 \leq \mu mg \quad \text{- - - - - - - - - - - - (16)}$$

From (8), (10, (11) and (12)

$$P_{max} = m_1 r_1 w^2 + 9m_3 r_3 w^2 + 16 m_4 r_4 w^2 + 4m_2 r_2 w^2 \quad \text{- - - - (17)}$$

$$= m_1 r_1 w^2 (1 + 0.98 + 0.945 + 0.896)$$

$$\therefore \quad P_{max} = P_1 = 0.03821 \, m_1 w^2 \quad \text{- - - - - - - - - - (17)}$$

Using (13), (15) and (17)

$$0.03821 \, m_1 w^2 > \mu(25 + 1.49 \, m_1) \times 9.81$$

Using $\mu = 0.414$; and $w = 45$ rad/sec.
then $m_1 > 1.37$ kg

Similarly using (13), (14), (16) and (17)

$$\text{Then } 1/3 \, (0.03821 \, m_1 w^2) \leq (0.414)(25 + 1.49 m_1) 9.81$$

$$m_1 \leq 5.4 \text{ kg}$$

$$\therefore \quad 1.37 \text{ kg} < m_1 \leq 5.40 \text{ kg}$$

within the limits of the space available, the chosen rotating masses were as follows :

$$m_1 = 2.884 \text{ kg}$$
$$m_2 = 0.245 \, m_1 = 0.7066 \text{ kg}$$
$$m_3 = 0.105 \, m_1 = 0.3028 \text{ kg}$$
$$m_4 = 0.14 \, m_1 = 0.403 \text{ kg}$$

Each of these rotating masses were divided into 4 equal parts i.e. 2 pairs. The detail of each of the four rotating masses are as contained on drawing number IPC - 12/5. The arrangement of these masses are as shown on general assembly drawing number IPC - 12/1 and plates 2 and 3.

3.2.3 Description of the Model Inertia Equipment

Plates 1, 2 and 3 show the different views of the model inertia equipment. The details of the orthogonal views are on the assembly drawing number IPC - 12/1, while the part-drawing of the workshop manufacturered parts are contained on drawings IPC - 12/2-9.

Referring to IPC - 12/1, the model equipment had a base plate 1, brackets 6, for carrying the vertical mounting plates 3, on the base plate. The mounting plates 3, carried both the Drive Unit and Inertia members.

The Drive Units consisted of a Universal electric motor 17, of maximum voltage 110V at speed of 13,000 r.p.m., a reduction gear box 19, of ratio 1 : 15 and a train of spur gears 18 (A-E) of ratios 1 : 2 : 3 : 4.

The inertia unit consisted of 8 pairs of eccentric masses 4 A-D mounted at the ends of shafts 5 A-D. This arrangement of the eccentric masses should give better stability to the system.

When the electric motor was energised, the shafts drove the eccentric masses so as to set up pulrality of centrifugal force as described in section 3.1.4. The net force imparted motion to the whole equipment. As long as the motor was energised, continual impulses were applied to the whole unit and the equipment was propelled.

The total mass of the equipment was 24.289kg.

The equipment is self contained and could be attached to any other equipment e.g. a soil engaging implement; which is to receive the impulses for performing work without the use of any substantial contact medium such as the ground.

The equipment could be mounted so that it could turn in any direction to conform to the direction in which the impulses were to be applied.

Plate 1: The Model Inertia Equipment.

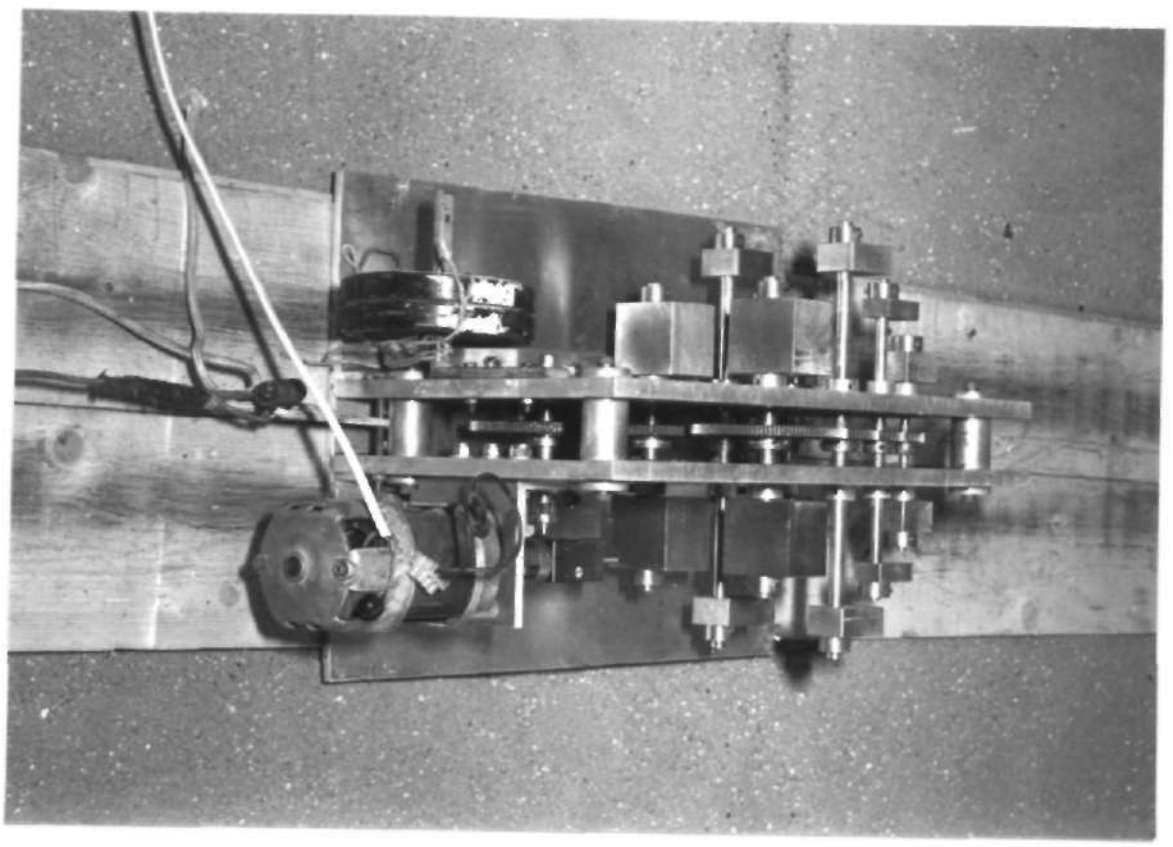
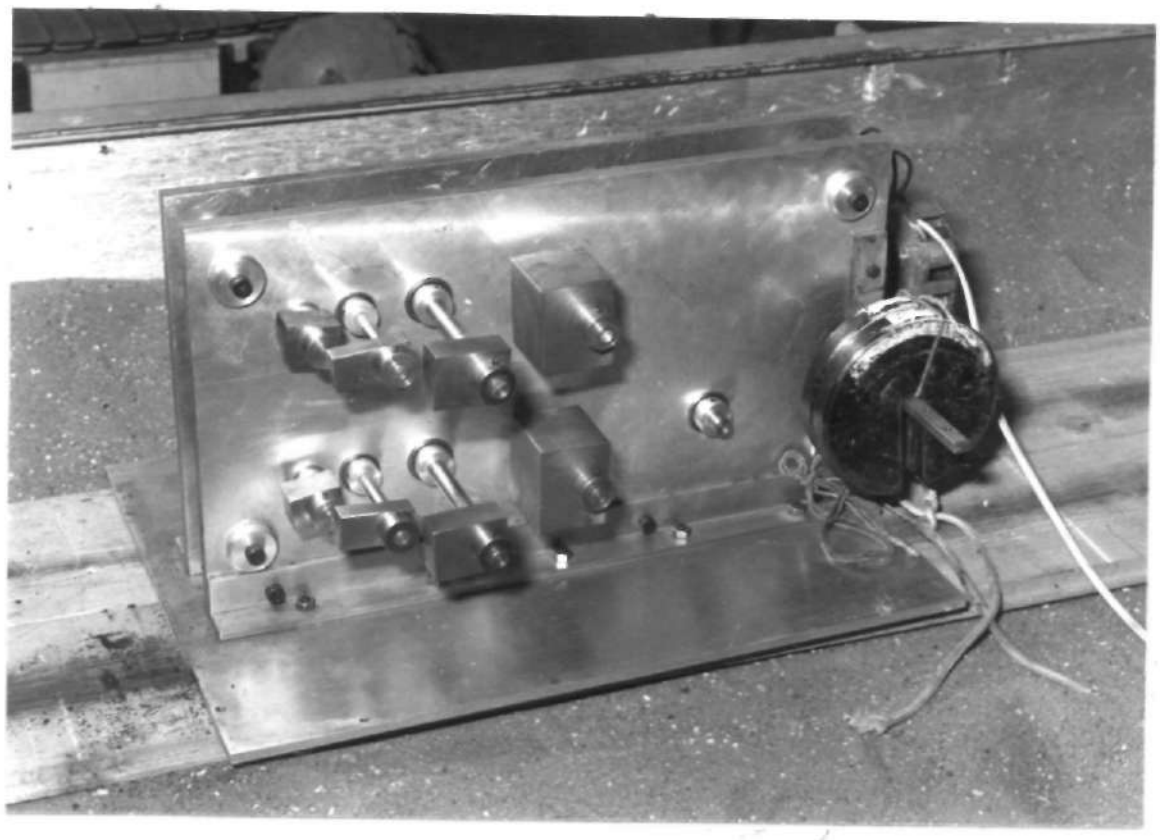


Plate 2: Side View (A) of Model Inertia Equipment.



Plate 3. Side View (B) of Model Inertia Equipment



3.2.4 System Design Safety.

The system design was checked for:

i) Maximum bending stress at critical points
ii) Maximum deflection
iii) Resonant frequency.

Sample calculation of the above are as shown in Appendix C.

All the above were found to be within sufficient margin of safety.

Another design safety which the author considered to be of importance was the loading of the gear teeth. The nature of the system made the force analysis on the gear teeth difficult to estimate; and this could not be checked. However, since the gears were designed to operate for 100's of hours and the equipment would not be run for such a long time; it was hoped that no serious problem would occur. Gear lubrication would be intermittently hand applied. The cover IPC -12/2 was provided mainly for safety purposes; for protection from the rotating masses; and it should therefore be on wherever the system is in operation.

3.3 INSTRUMENTATION.

Ideally, the equipment should be tested to show the extent the experimental results agreed with the theoretical prediction as represented by (7). It was therefore proposed to instrument the equipment to measure acceleration/force at varying rotating speeds using an accelerometer. However, time did not permit this and a simpler instrumentation was adopted.

The behaviour of the equipment was observed under varying rotating speed, draw bar pull, and surface conditions; using a tachometer, stop watch, and rule to measure speed of rotation of the eccentric masses, time and distance moved by the equipment respectively.

4. PERFORMANCE OF MODEL INERTIA EQUIPMENT

4.1 Experimental Procedure

The reference system performance parameter against which other parameters were measured was the speed of rotation of the rotating masses (w). This speed was controlled by regulating the input voltage using a rotary megavolt; and each position on it's dial corresponded to a particular speed of rotation. In order to facilitate the measurment of the speed of rotation, it was calibrated against the megavolt position.

In order to find the system linear velocity, the test procedure adopted was to place the equipment on the selected surface, apply the required draw bar pull, set the speed of rotation and measure the time taken to travel a measured distance. Tests were run on three different surfaces (concrete, loose soil and wood) at zero draw bar pull and also at different draw bar pull; the latter on the wood surface only.

Similarly, to measure the maximum pull, the equipment was placed on the selected surface, and at set speed, and loaded through a pulley system dynamometer until a point of no-motion was reached.

Tests with draw bar pull could only be performed on the wood surface since time did not permit the construction of dynamometer system at different test sites for different surfaces. Successful tests could not be performed on loose sand available in the soil bin where the dynamometer system is available because of the bull-dozing effects as explained below.

The coefficient of friction of the different surfaces was measured with the aid of spring dynamometer. The dynamometer was attached to the equipment on one end; and hand-pulled at the other end of the dynamometer. The indicated pull at the point of limiting friction was noted. The results obtained by this simple method approximately agreed with those

obtained by other more sophisticated method.

All measurements were replicated and average values were used for final calculations.

4.2. Experiemental Results.

Typical test results are presented in Appendix D, and also figure D.1, 4.1, 4.2, and 4.3.

It was found that the relationship between the megavolt position (% input voltage) and the speed of rotation (w) was linear Fig. D.1. The operating range of the speed was found to be:

7 - 15 revolutions per secs.

For each of the surfaces, the relationship between the Linear Velocity (v) and the speed of rotation (w) was approximately parabolic Fig. 4.1; and the maximum Linear velocity occured at approximately the same speed.

Similarly for the different draw bar pulls, the relationship between the Linear velocity and speed of rotation was approximately parabolic but maximum Linear velocity occured at different values of rotating speed. Fig. 4.2. The lighter load occured at higher speed.

It was equally noted that the equipment exhibited only forward motion i.e. one way motion, for rotating speed less than the maximum Linear velocity rotating speed; and it exhibited both forward and rearward motion; i.e. two-way motion at other higher speed and the rearward speed increased with speed.

The transition from one-way motion to two-way motion seemed to coincide with the point of maximum Linear velocity.

The maximum pull and speed also appeared to indicate a parabolic relationship fig. 4.3. The pull initially increased, with speed, before gradually decreasing with further increase in speed.

The following features in addition were noted during the system operation. There was vibration which increased with increase in the rotating speed, and draw bar pull. Secondly the equipment bulldozed when it was operated in non compacted soil. This bulldozing effect was very high in loose sand so much that the equipment performance test could not be performed in the soil available on the test site. Part of the equipment got buried before it travelled any appreciable distance. As this would permit easy-entrance of sand into the equipment which could be trapped and cause damage, these tests were discontinued.

At the initial stage of the testing, the equipment exhibited a sideways steering effect. This was due to the centre of mass not coinciding with the geometric centre of the equipment; whereas the generated inertia force passed through the geometric centre of the system; due to the symmetrical mounting of the eccentric masses. The electric motor and reduction gear box caused this "out of balance effect". However, mounting 1.82 kg, the equivalent of the total mass of the electric motor and reduction gear box: on the opposite side of the electric motor restored the system to steer in a straight line. This additional weight was added during the testing of the equipment; to enable it to move in a straight line.

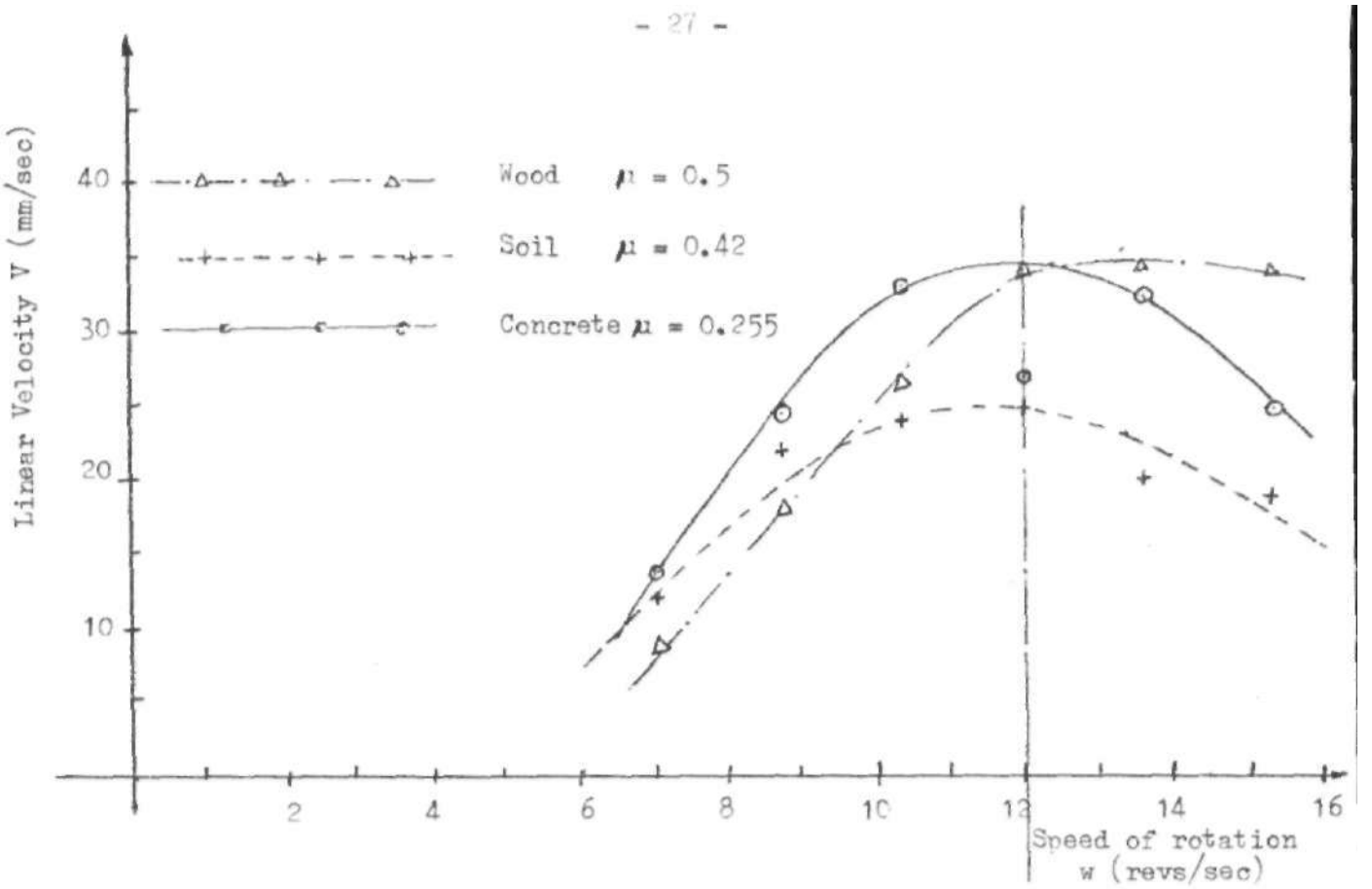


Fig. 4.1. System Performance Parameters for different frictional surfaces at zero draw bar pull

Fig. 4.2 System Performance Parameters for different Draw Bar Pulls (L) (Wood surface only)

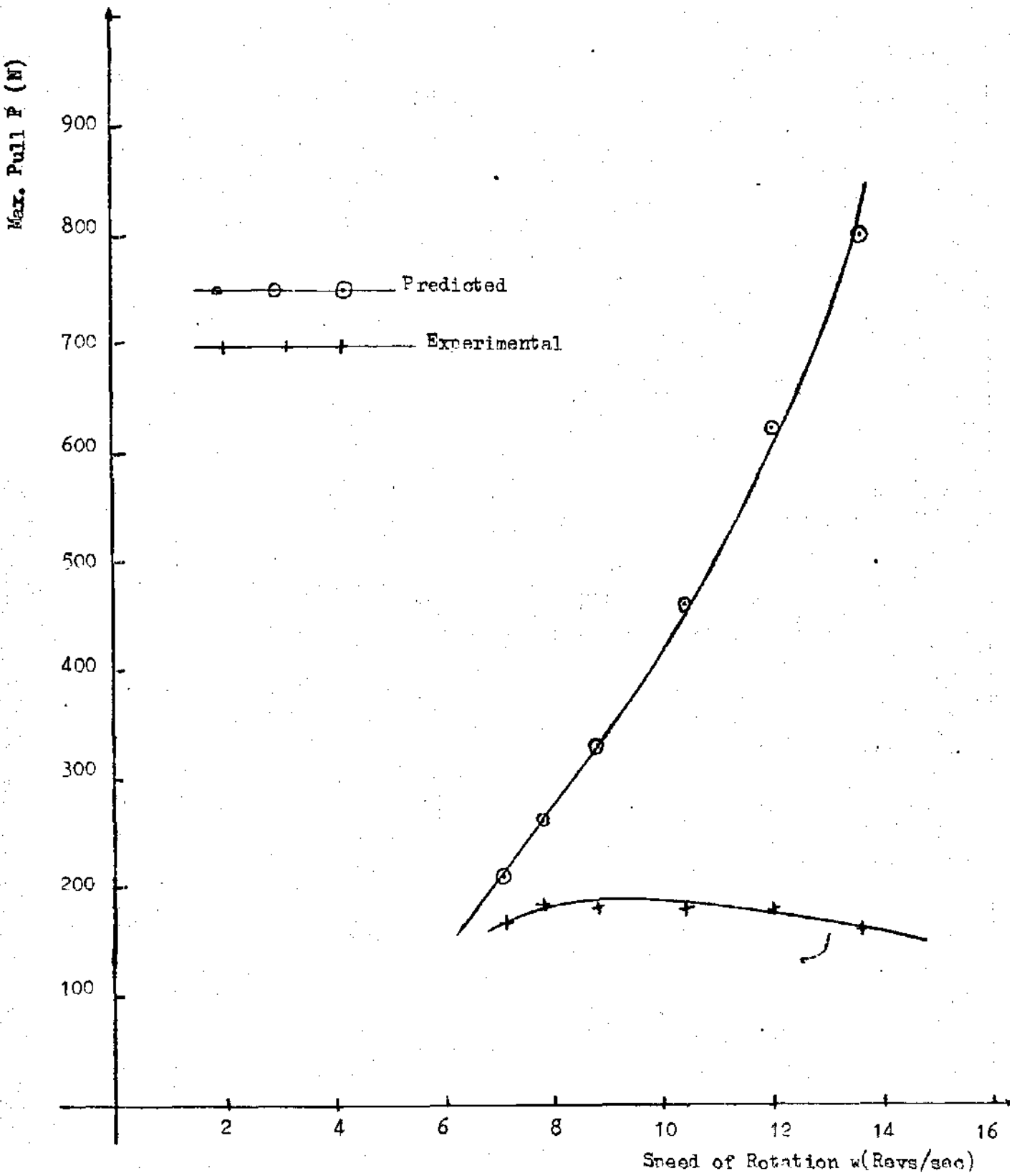


Fig. 4.3. Max. Pull of Model Inertia Equipment.

5. DISCUSSION

It would appear that at the lower speed range when the equipment exhibited one-way motion, the frictional resistance was sufficient to prevent motion in the opposite direction i.e. $P_2 < \mu Mg$, where as the higher speeds $P_2 > \mu Mg$, fig 3.3 and 3.6, and therefore the rearward motion occured in addition to the forward motion. The transition point corresponded to when $P_2 = \mu Mg$, when the frictional resistance was just sufficient to prevent motion in rearward direction. The occurence of the maximum linear velocity i.e. the point of system optimum performance is consistent.

As for the draw bar pull, fig. 4.2, the occurance of the transition point at different rotating speed is equally significant. During every revolution, the effect of the draw bar pull (L) was to increase P_2, and the greater the external load, the greater the resultant $(P_2 + L)$ and the greater the possibility of reaching the transition stage at which $(P_2 + L) \geq \mu Mg$, and then the occurence of transition point at lower values of rotating speed for higher loads.

Also the smaller the external load, the faster the motion of the equipment, and therefore the results in fig. 4.2 seem consistent.

The system performance for the three different surfaces at zero draw bar pull, Fig. 4.1., does not seem to follow any predictable pattern.

If frictional resistance was the only external force acting this would suggest that the lower the coefficient the better the performance. The results obtained does not follow this prediction as wood with coefficient of friction of 0.5 gave a much better performance than Soil concrete with coefficient of friction of 0.42. This result therefore seems to suggest that there were possibly other factors affecting the mode of operation that were not considered. Similarly the disparity between the predicted and experimental maximum pull values, fig. 4.3 suggest the same thing.

In both the introduction and literature review, it was reported that although the vibratory tools had reduction in both soil

compaction and draw bar pull compared with the conventional non-vibrating tools, the vibratory tool consumed rather more power. One of the reasons given for this high power consumption was that its rearward movement into the already cut soil constituted a wastage of power/energy loss. It therefore followed that if the implement could be constrained to have forward motion only, the energy in this rearward movement would be avoided. It was in view of this fact that the definition of the basic design concept in this study was set as a propulsion system mounted on a simple vehicle consisting of a moving mass, but constrained in such a way that acceleration over part of a cycle would greatly exceed that in the remaining part of the cycle, so that the developed driving force would move the vehicle in one direction only provided that the resistance to rearward motion was sufficiently high.

From the above results the developed model inertia equipment exhibited this positive motion. At lower speeds the equipment exhibited forward motion only, while on high speeds it exhibited both forward and rearward motion.

This indicates that the equipment is capable of operating both as a pure inertia propulsion system or vibrating system depending on the operating speed. This makes the system a dual purpose tool.

It's usage either as a pure inertia system or a vibratory system depends on the end results desired; since vibratory tools have their advantages too.

However, it should be remembered that the higher the operating speed, i.e. as a vibratory system, the higher the power consumption. Therefore when the system operates at lower speed, i.e as a pure inertia propulsion system, it conserves power; compared to when it operates as a vibratory tool.

The transition point corresponds to the point of system optimum performance.

6. CONCLUSIONS AND RECOMMENDATION.

The project served to test the concept of Inertia Propulsion; using a developed contra-rotating masses mechanism.

Although time allowed only limited study of the performance of the model equipment, the results demonstrated that inertia propulsion is feasible.

With its capability to pull in one direction only, it is possible that its performance in terms of draught reduction and power efficiency will exceed the performance of vibratory system.

In view of the limited tests that had been done on the equipment, due to time limit, no logical conclusion can be drawn on some of the system performance parameters, such as the effect of the coefficient of friction on system mode of operation, and the disparity between the predicted and experimental values of maximum pull. It is therefore recommended that further detailed extensive equipment performance tests should be carried out using more sophisticated instrumentation such as accelerometer as earlier intended.

It is hoped that this further investigatory study will generate enough data for designing a bigger scale model equipment for studying the application of the concept.

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APPENDICES

A: Fourier Analysis

A1 Calculation of the Fourier Coefficients.

From Fig.3.2 of Fourier Analysis of Basic Design Concept (3.1.3) wave form P(t) is defined over a complete period

$$-\frac{\alpha\pi}{w} \quad \text{to} \quad \left\{\frac{2\pi}{w} - \frac{\alpha\pi}{w}\right\}$$

and $P(t) = H$ for $-\frac{\alpha\pi}{w} < t < \frac{\alpha\pi}{w}$ ---------(1)

$P(t) = -h$ for $\frac{\alpha\pi}{w} < t < \left\{\frac{2\pi}{w} - \frac{\alpha\pi}{w}\right\}$ ------(2)

For a periodic function P(t), the Fourier equation is given by

$$P(t) = \tfrac{1}{2} Ao + \sum_{n=1}^{n=\infty} (An \cos nwt + b_n \sin nwt) \text{ - - -(3)}$$

where Ao, An, bn are the Fourier Coefficients and defined as follows:

$$\left.\begin{aligned} An &= \frac{w}{\pi} \int_{\tau}^{(\tau + \frac{2\pi}{w})} P(t) \cos nwt \, dt \\ bn &= \frac{w}{\pi} \int_{\tau}^{(\tau + \frac{2\pi}{w})} P(t) \sin nwt \, dt \end{aligned}\right\} \text{ - - - (Lawden 1959)}$$

From (1) and (2)

$$\text{then } An = \frac{w}{\pi} \int_{-\frac{\alpha\pi}{w}}^{\frac{\alpha\pi}{w}} H \cos nwt \, dt + \frac{w}{\pi} \int_{\frac{\alpha\pi}{w}}^{\left[(2-\alpha)\frac{\pi}{w}\right]} (-h) \cos nwt \, dt$$

$$= \frac{Hw}{\pi nw} (\sin nwt) \Big|_{-\frac{\alpha\pi}{w}}^{\frac{\alpha\pi}{w}} - \frac{wh}{\pi wn} (\sin nwt) \Big|_{\frac{\alpha\pi}{w}}^{\left[(2-\alpha)\frac{\pi}{w}\right]}$$

$$= \frac{2 \sin(n\alpha\pi)}{\pi n} (H+h)$$

$\therefore A_o = 0$

Similarly

$$bn = \frac{w}{\pi}\int_{-\frac{d\pi}{w}}^{\frac{d\pi}{w}} H \sin wt\, dt + \frac{w}{\pi}\int_{\frac{d\pi}{w}}^{(2-d)\frac{\pi}{w}} (-h) \sin nwt\, dt$$

$$= o + \frac{hw}{\pi nw}\left(\cos nwt\right)\Bigg|_{\frac{d\pi}{w}}^{(2-d)\frac{\pi}{w}}$$

$$= \frac{hw}{\pi n}\left[\cos\frac{nw\,d\pi}{w} + \cos\left(\frac{nw}{w}\pi\right)(2-d)\right]$$

$$= \frac{h}{n\pi}\left(\cos nd\pi - \cos n\pi d\right)$$

$$= 0$$

then (3) becomes

$$P(t) = \tfrac{1}{2}Ao + \sum_{n=1}^{n=\infty}\frac{2(H+h)}{n\pi}\sin(dn\pi)\cos nwt \quad \text{---- (4)}$$

A.2 Condition for Convergence of Fourier Equation of P(t)

For the Fourier equation to be convergent, one of the conditions that must be satisfied is that P(t) should be periodic with period 2π (Spencer, 1977) i.e. the net displacement of a mass suffering P(t) over a cycle should be zero.

i.e. $$x = \int_{o}^{2\pi} v\, dt = o$$

but $v = \int P(t)\, dt$ since acceleration is proportional to force.

from Fig. 3.2 considering P(t) over a cycle

then $$v = \int_{o}^{2\pi} P(t)\, dt = \frac{H2d\pi}{w} - h\left(\frac{2\pi}{w} - \frac{d\pi}{w} - \frac{\pi d}{w}\right)$$

$$v = \frac{2d\pi H}{w} - \frac{2h\pi}{w}(1-d)$$

Now for Zero displacement.

$$x = \int_0^{2\pi} v \, dt = 0$$

i.e. $$\frac{2\lambda\pi}{w} Ht \Big|_0^{\frac{2\pi}{w}} - \left[\frac{2\pi h}{w} (1-\lambda)\right] t \Big|_0^{\frac{2\pi}{w}} = 0$$

$$\frac{2\lambda\pi H}{w} \times \frac{2\pi}{w} - \frac{2\pi h}{w} (1-\lambda) \frac{2\pi}{w} = 0$$

$$\frac{4\pi^2}{w^2} \left[H\lambda - h(1-\lambda)\right] = 0$$

then $$\frac{H}{h} = \frac{1-\lambda}{\lambda}$$

B: ESTIMATION OF ECCENTRIC DISTANCE r.

The general formula of the Centrifugal force on a rotating mass expressed as $mr\,w^2$ is based on a particle mass m at radius r.

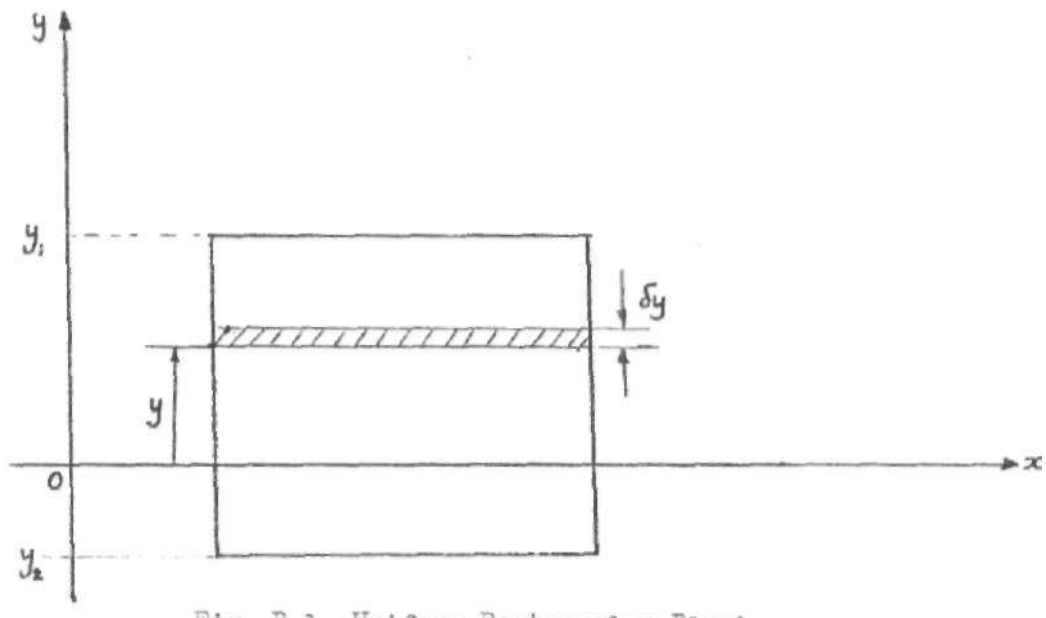


Fig. B.1 Uniform Rectangular Block.

For a uniformly distributed mass, consider a rectangular block as shown in Fig. B1 rotating about the x - axis. If the mass of the block is m, then the mass of the element δy is given by

$$\delta m = m \frac{\delta y}{(y_1 + y_2)}$$

where $(y_1 + y_2)$ is the length of a side

Centrifugal force on element $\delta m = y.\,\delta m\,w^2$

$$= y.\left(\frac{m\,w^2}{y_1+y_2}\right)\delta y$$

then Centrifugal force on whole block

$$= \int_{y_2}^{y_1} \left(\frac{m\,w^2}{y_1+y_2}\right) y.\,dy$$

$$= \frac{m\,w^2}{2}\,(y_1 - y_2)$$

then from the general formula, $r = \left(\frac{y_1 - y_2}{2}\right)$

Within the limits of the space available for rotating masses

4A - C IPC - 12/1

$y_1 = 30$ mm, $y_2 = 10$ mm, then $r = \frac{(y_1 - y_2)}{2} = 10$ mm

For rotating mass 4D

$y_1 = 18$ mm, $y_2 = 10$ mm, then $r = \frac{(y_1 - y_2)}{2} = 4$ mm

C. CHECKING FOR SYSTEM DESIGN SAFETY

With reference to system assembly as on IPC - 12/1; the loading of the shafts were considered to be equivalent to a cantilever system; each shaft constituted a uniformly loaded beam, and a point load of the centrifugal force of the eccentric mass.

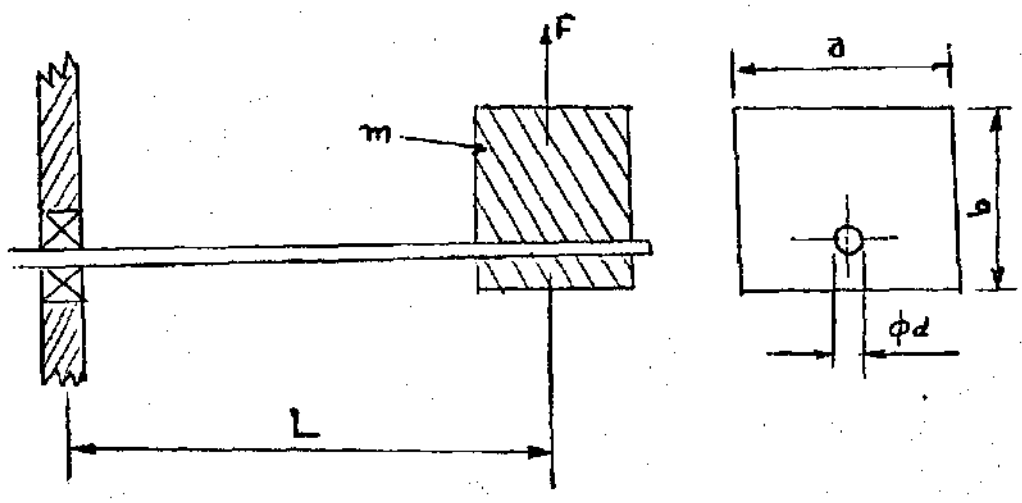


Fig. C.1 Shaft Loading Considered as Cantilever System.

The possible maximum centrifugal force (F) that could be developed by each of the eccentric masses will be at the maximum operating speed w = 90 radi/sec.

for mass 4A, F = 59N
for mass 4B, F = 57N
for mass 4C, F = 55N
for mass 4D, F = 52N

C.1 Checking for maximum bending stress

From fig. C.1

Max. Bending moment = FL

Section modulus $Z = \frac{\pi d^3}{32}$

System bending stress $\sigma = \frac{FL}{Z}$

Max. allowed bending stress (σ_{max}) for mild steel
$= 280 \times 10^6 \ N/m^2$

For shaft 5A

System bending stress $= \frac{FL}{Z}$

$= \frac{59 \times 35 \times 10^9 \times 32}{10^3 \times \pi \times 10^3}$

$= 2.103 \times 10^6 \ N/m^2$

∴ safety factor ≏ 133

Similarly safety factor for 5B, C and D are 5, 5, and 6 respectively.

C.2. Checking for Maximum deflection.

From fig. C.1.

Max. deflection $\delta_{max} = \frac{FL^3}{3EI}$

where $I = \frac{\pi d^4}{64}$

for Shaft 5A, $\delta_{max} = \frac{59 \times 35^3 \times 64 \times 10^{12}}{10^6 \times 3 \times 207 \times 10^9 \times \pi \times 10^4}$

$= 0.23$ mm

Similarly for 5B, C, D the maximum deflections are

1.51 mm, 1.33mm, 0.53 respectively.

C.3 Checking for Resonant Frequency.

(i) Torsional Vibration.

From fig. C.1. the natural frequency w_n of the system if given by

$$w_n = \sqrt{\frac{K_t}{Ir + 1/3\ Is}} \qquad \text{(Harris 1976)}$$

where $K_t = \frac{\pi G d^4}{32 l}$

$$Ir = \frac{1}{12} m_r (a^2 + b^2)$$

$$Is = \frac{1}{2} m_s \left(\frac{d}{2}\right)^2 = \frac{\pi d^4 l \rho}{32}$$

for shaft 5A

$$Ir = \frac{1}{12} (.721) (50^2 + 40^2) \times 10^6 = 14.78 \times 10^{-4}$$

$$Is = \frac{\pi \times 10^4 \times 35 \times 7840}{32 \times 10^{15}} = 2.694 \times 10^{-7} \quad \text{(very small)}$$

then $K_t = \frac{\pi \times 10^4 \times 82.8 \times 10^9 \times 10^3}{32 \times 35 \times 10^{12}} = 0.23225 \times 10^4$

then $w_n = \sqrt{\frac{0.23225 \times 10^4}{14.78 \times 10^{-4}}} = 0.125 \times 10^4$ rad/sec.

(ii) Bending Vibration.

The natural frequency of the system is also given by

$$w_n = \sqrt{\frac{3EI}{l^3 (m_r + 0.23\ m_s)}}$$

$$m_r = \frac{\pi d^2 l \rho}{4} = \frac{\pi}{4} \times \frac{10^2}{10^6} \times \frac{35}{10^3} \times 7840 = 215.3 \times 10^{-4} \text{ kg}$$

$$I = \frac{\pi d^4}{64} = \frac{\pi \, 10^4}{10^{13} \times 64} = 0.049 \times 10^{-8}$$

Then

$$w_n = \sqrt{\left(\frac{3 \times 307 \times 10^9 \times 0.049 \times 10^9}{35^3 \times (0.7425) \, 10^8}\right)}$$

$$= 0.030916 \times 10^5 \text{ rad/sec.}$$

Similarly the torsional and bending natural frequencies were calculated for 5B, C and D.

Shaft Identification	Maximum working frequency (rad/sec)	Natural frequencies	
		torsional (rad/sec)	Bending (rad/sec)
5A	90	1250	3092
5B	180	4961	1445
5C	270	6945	3059
5D	360	7796	7020

Table C.1 Frequencies of Eccentric Masses.

D. PERFORMANCE DATA FOR MODEL INERTIA EQUIPMENT.

Total mass of the model inertia equipment = 24.289 kg.

Table. D.1 Calibration of speed of rotation (w)

RegaVolt Position (% Input Voltage.)	Average speed of Rotation (w)	
	revs/min	revs/sec.
20	306.7	5.1
25	413.3	6.9
30	530	8.8
35	633.3	10.6
40	720	12.0
45	813.3	13.6
50	906.7	15.1

Fig. D.1. System Rotating Speed Calibration Curve.

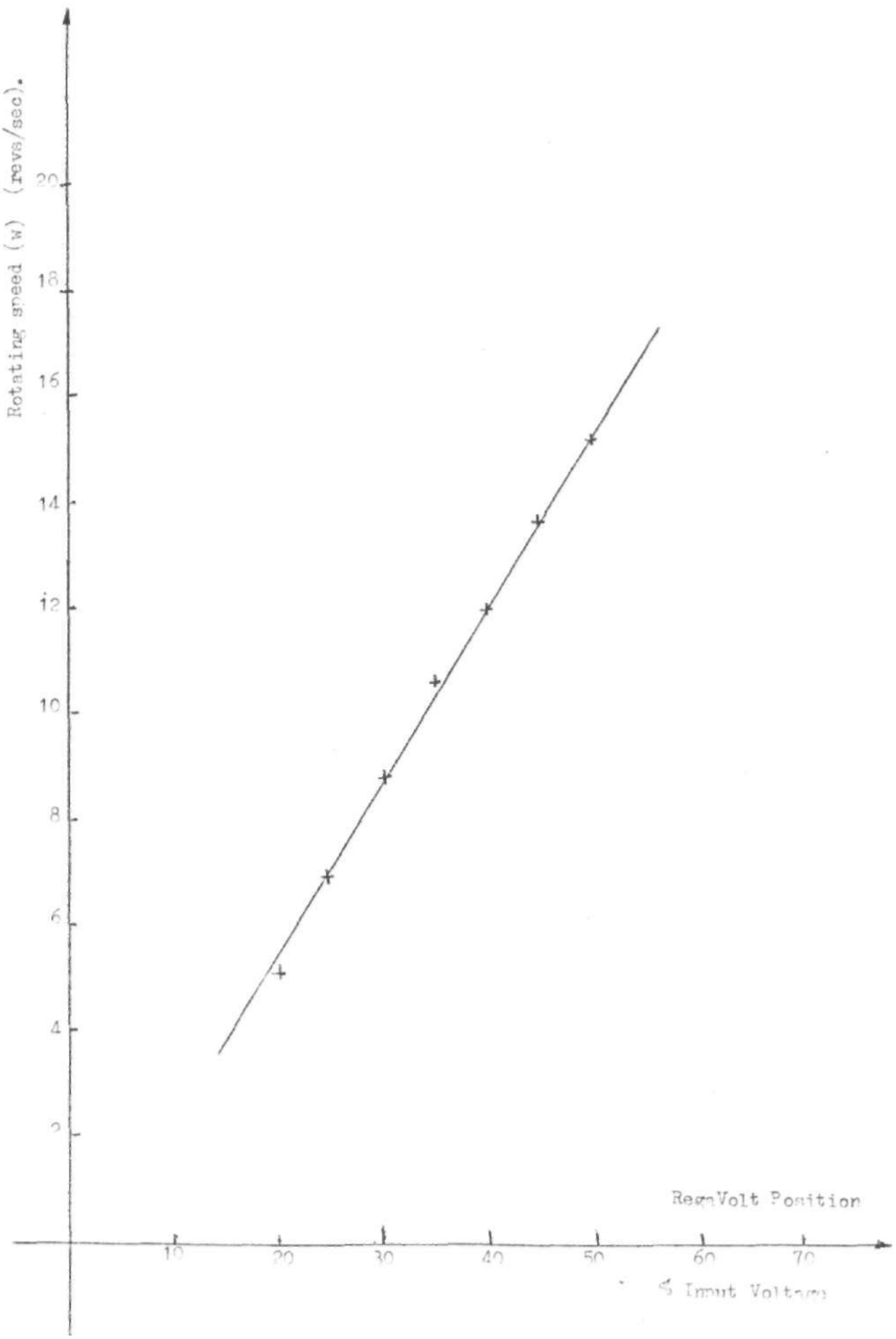


TABLE D.2. System Performance Data for different frictional Surfaces at Zero Draw Bar Pull.

Surface Type	Speed of Rotation		Average Time to travel 1500 mm	Linear Velocity v (mm/sec)
	% input Voltage	w.revs/sec		
CONCRETE	25	7.1	105	14.3
	30	8.8	61.1	24.5
	35	10.4	45.0	33.3
	40	12.0	56.1	26.7
	45	13.6	46.2	32.5
	50	15.3	60	25.0
SOIL	25	7.1	125	12.0
	30	8.8	68.3	22.0
	35	10.4	62.0	24.0
	40	12.0	61.0	25.0
	45	13.6	74.5	20.0
	50	15.3	78.3	19.0
WOOD	25	7.1	161.5	9.3
	30	8.8	81.5	18.4
	35	10.4	56.0	26.8
	40	12.0	43.0	34.9
	45	13.6	43.0	34.9
	50	15.3	43.5	34.5

Coefficient of friction (μ) for the different surfaces.

Weight of the equipment = 24.289Kgf = 238.3N

1. Concrete.

Maximum pull at point of limiting friction = 60N.

$\mu = \frac{60}{238.3} = 0.25$

2. SOIL

Maximum pull at point of limiting friction = 100 N

$\mu = \frac{100}{238.3} = 0.42$

3. Wood

Maximum pull at point of limiting friction = 120N

$\mu = \frac{120}{238.3} = 0.50.$

TABLE D.3 System performance data for different Draw Bar Pulls, L
(Wood surface only.)

Draw Bar Pull L (N)	Speed of rotation		Average time to travel 1500 mm (sec)	Linear velocity (mm/sec)
	% input voltage	w. Revs/sec		
0	25	7.1	161.5	9.21
	30	8.8	81.5	18.4
	35	10.4	56.0	26.8
	40	12.0	43.0	34.9
	45	13.6	43.0	34.9
	50	15.3	43.5	34.48
14	25	7.1	183.5	8.20
	30	8.8	92.33	16.3
	35	10.4	83.0	18.1
	40	12.0	95.5	15.7
	45	13.6	64.10	23.4
	50	15.3	68.0	22.1
24	25	7.1	209.0	7.2
	30	8.8	134.5	11.2
	35	10.4	153.5	9.8
	40	12.0	205.0	7.3
	45	13.6	219.3	6.8

TABIE D.4 : System Data for Maximum Pull (Wood Surface Only.)

Frictional Resistance = μMg = 120N

See Section D.2

Speed of rotation		MAXIMUM PULL (N)	
% RegaVolt position	w (revs/sec)	Experimental (Draw bar Pull and μMg)	Predicted
25	7.1	171	216
27	7.8	185	260
30	8.8	180	335
35	10.4	176	468
40	12.0	174	624
45	13.6	162	801

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