# TORSIONAL VIBRATION IN NILE IRRIGATION LOW PRESSURE CENTRIFUGAL PUMPS.

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1.1- **ABSTRACT:**

This work deals with the investigation of torsional vibration for the low pressure centrifugal Nile irrigation pumps (L.P.C.N.I.P), using two types of prime movers, electric and diesel with variable speeds, coupled rigid and flixable, with and without non-return valve, using rubber and steel pipes on the suction side. The water pureness was varied.

1.2- **INTRODUCTION:**

The vibration phenomena in centrifugal pumps largely depends on:

a - Mechanical Sources:
- Type of prime mover,
- Speed of rotation 2*,
- Construction of pump set (shafts, bearings, coupling, impeller, valves and pipes), 2*,
- Type of suction and delivery sides,
- Foundation and fixation.

b - Hydraulic Sources:
- Water head, 8*
- Water pureness,
- Cavitation, 3*, 4*,
- Turbulence, 2*, 3*,
- Separation 2*, 3*.

It must be taken into consideration the effect of; the primemover, speed of rotation, the coupling, the non-return valve, the material of suction side and water pureness.

All other acting factors were neglected. The type of the investigated (L·P·C·N·I·P) was double bearings. All variable values were measured and recorded on 12 multichannel recorder-PHILPS Type (PR 9034).

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1.3- EXPERIMENTAL WORK:

The arrangement of the station used for these investigations shown in Fig. (1a, 1b). This station was designed and constructed at the Faculty of Engineering & Technology, Shebin EL-Kom Menoufia University. It consists of the tested centrifugal pump which have the following specifications; Motor horsepower 10 H.P., suction and delivery pipes diameter 6 inches, main supply tank, a delivery calibrated tank, five speeds conical pulleys system, a calibrated torque transducer on the impeller shaft - PHILPS type (PR 9380 R/50) Fig. (2). a calibrated force transducer on the electric motor - PHILIPS type (PR 6205/23) Fig. (3), a calibrated R.P.M. transducer fixed beside the impeller shaft - PHILIPS type (PR 9373) Fig.(4) and vibration measuring system has the following elements;

- Multi-channel recorder- PHILIPS type (PR 9034) Fig. (7).
- Transduces type AV 100 C Serial B.A.S. E.E.F.A. No. Ex. 71080 (B) ENGLISH - which fixed on the measured parts as shown in Fig. (5).
- Vibration measuring system- ENGLISH type - 2000 Series 2 channels system A, V, D in each channel shown in Fig. (6).

The transducers were fixed accurately in vertical and horizontal positions Fig. (5), the calibration for every transducer is shown in Fig. (8).

The calibration of force transducer was made as shown in Fig. (3), the torque acting on the shaft was measured by a torque transducer which was calibrated as shown in Fig. (2), the calibration chart is shown in Fig. (9).

The rate of flow was measured by a calibrated orifice meter, the suction head was measured by a calibrated vaccume meter, another three calibrated pressure gauges were used for measuring; delivery head, up and down stream pressure of orifice meter as shown in Fig. (10).

## 1.4- THEORETICAL ANALYSIS:

In order to estimate the torsional vibration on the system must be determine the stiffness of each element, the equivalent system and equation of motion 1*.

### 1.4.1. Stiffness:

#### 1.4.1.1. Shaft

The stiffness of the shaft can be generally determined from the following equation:

$$C = \frac{G}{32} \cdot \frac{\pi \cdot d^4}{L}$$

Where:

G : Torsional elastic modulus ...........Kg/Cm$^2$,

L : Uniform shaft length ................ Cm,

d : Shaft diameter .................... Cm,

The stiffness can also be determine by the following equation:

$$e = \frac{1}{C} = \frac{32}{\pi . G} \cdot \frac{K_{key} \cdot L}{d^4}$$

where:

e : Nomber of rad. of cylinderical shape per unit of torsion moment 1* ,

K : Key factor ($K_{key} = 1.0 + 10 \frac{h}{d}$),

h : Key height in Cm.,

d : Shaft diameter in Cm.

#### 1.4.1.2 Coupling Stiffness:

The stiffness of coupling can be calculated with the same notations as follows:

$$e = \frac{1}{C} = \frac{32}{\pi G} \cdot \frac{L}{D^4} \cdot k_f .$$

where:

$k_f$ : Flange factor,

$$k\ f : \frac{1}{0.5\ n\ (\frac{d_B}{D})}$$

$d_b$ : Bolt diameter .................... Cm,
n : Nomber of bolts ......,
D : Flange diameter .................... Cm.

1.4.1.3. Equivalent Stiffness:

The equivalent stiffness of the system can be calculated by the following equation:

$$\frac{1}{C_{eq.}} = \frac{1}{C_1} + \frac{1}{C_2} + \dots\dots\dots\dots + \frac{1}{C_n}$$

Table (1): represents the calculated stiffness for the parts of the system used.

Table (1)

| Part | Equivalent stiffness Kg.Cm/rad. |
|---|---|
| Shaft part (1) | $0.0257 \times 10^{-5}$ |
| Shaft part (2) | $0.0514 \times 10^{-5}$ |
| Shaft part (3) | $0.0310 \times 10^{-5}$ |
| Shaft part (4) | $0.0899 \times 10^{-5}$ |

So, simplest equivalent stiffness of the system equals to:

$C_{equivalent}$ = 505050.50 Kg.Cm/rad.

1.5- EQUIVALENT MASS SYSTEM:

In order to determine the equivalent mass system, the moment of inertia of each part must be determined. In what follows the determination of the moment of inertia for the system shown in Fig. (11a and 11b) which represented in Table (2) for different speeds of the system and electric motor.

Table (2)

| moment of inertia / R.P.M. | $J_1$ Kg. Cm. $Sec^2$ | $J_2$ Kg. Cm. $Sec^2$ |
|---|---|---|
| 1350 | 1185 | 645 |
| 1550 | 1185 | 1270.65 |
| 2350 | 1185 | 2057.55 |
| 2700 | 1185 | 2573.55 |
| 3300 | 1185 | 3276.60 |

The system was simplified as shown in Table (2) to two equivalent masses considering moment of inertia for rotor is $J_1$, the another parts (impeller, shafts and coupling) is $J_2$. By the same steps Table (3) gives the moment of inertia of the system with diesel engine.

Table (3)

| R.P.M \ Moment of Inertia | $J_1$ Kg. Cm. $Sec^2$ | $J_2$ Kg. Cm. $Sec^2$ |
|---|---|---|
| 1350 | 17820 | 645 |
| 1550 | 17820 | 1270.65 |
| 2350 | 17820 | 2057.55 |
| 2700 | 17820 | 2573.55 |
| 3300 | 17820 | 3276.60 |

1.6- <u>CALCULATION OF THE MAX. DYNAMICS LOADS OF THE SYSTEM:</u>

The equation of motion for the equivalent system is:

$$J_1 \ddot{\varphi}_1 + C_{12} (\varphi_1 - \varphi_2) = M_1 \quad \ldots\ldots\ldots\ldots \quad 1^*$$

$$J_2 \ddot{\varphi}_2 + C_{21} (\varphi_2 - \varphi_1) = -M_2 \quad \ldots\ldots\ldots\ldots \quad 1^*$$

where:

$J_1$ - equivalent moment of inertia of the driving part, Kg.Cm.$Sec^2$,

$J_2$ - equivalent moment of inertia of the driven part, Kg.Cm.$Sec^2$.,

$C_{12}$- equivalent stiffness, Kg.Cm./rad.,

$\varphi_1, \varphi_2$- angular displacement,

$M_1$ - equivalent driving torsional moment, Kg.Cm,

$M_2$ - equivalent driven torsional moment, Kg.Cm.

These equations can be simplified as follows:

$$M_{12} + B_{12}^2 \cdot M_{12} = \frac{C_{12}}{J_1 \cdot J_2} (M_1 \cdot J_2 + M_2 \cdot J_1) \quad \ldots\ldots\ldots (1)$$

where:

$M_{12}$ - the elastic torsional moment between $J_1$ & $J_2$,

$$M_{12} = C_{12} \quad (\varphi_1 - \varphi_2)$$

$B_{12}$ - free frequencies of the system,

$$B_{12} = \sqrt{C_{12}\left(\frac{1}{J_1} + \frac{1}{J_2}\right)} \qquad \text{..........(2)}$$

From the previous equations, we can written in the form of general solution of the equations finding the max. torsional moment as follows;

$$M_{12} = \frac{M_1 J_2 + M_2 J_1}{J_1 + J_2}\left[1 + \frac{\sin \pi\lambda}{\pi\lambda}\right]$$

where:

$$\frac{M_1 J_2 + M_2 J_1}{J_1 + J_2} = \text{Static torsional moment,}$$

$$\lambda = \frac{T \cdot B}{2\pi}$$

In the case of diesel engine $J_1 = R_R$, can be considered relativilly very large compared with $J_2$. Then the system well be studied as one mass system ($J_1 = \infty$). From equation No. (2). The free frequencies ($B_{12}$) for both types of prime-movers was obtained. Table (4) represents the equivalent stiffness ($C_{12}$) and free frequencies ($B_{12}$) of the system for different speeds and both types of prime-movers.

Table (4)

| Items / R.P.M. | $C_{12}$ Kg.Cm./rad. | | $B_{12}$ | |
|---|---|---|---|---|
| | Electric | Diesel | Electric | Diesel |
| 1350 | $5.051 \times 10^5$ | $5.051 \times 10^5$ | 34.74 | 28.48 |
| 1550 | $9.95 \times 10^5$ | $9.95 \times 10^5$ | 40.72 | 28.96 |
| 2350 | $1.61 \times 10^6$ | $1.61 \times 10^6$ | 46.27 | 29.54 |
| 2700 | $2.02 \times 10^6$ | $2.02 \times 10^6$ | 49.85 | 30.02 |
| 3300 | $2.6 \times 10^6$ | $2.60 \times 10^6$ | 54.68 | 30.64 |

From the analysis of the graphic chart for the torque transuducer reading, the frequencies of the system taken a small values compared with the theoretical results because the damping of liquied and parts takes a large effect on this item.

The following tables (5 and 6) gives the $\lambda$ values for different times of startes, for different speeds and for various types of prime-movers.

(electric motor) Table (5)

| R.P.M. \ T sec. | 0.01 | 0.02 | 0.03 | 0.04 | 0.05 | 0.06 | 0.07 |
|---|---|---|---|---|---|---|---|
| 1350 | 0.06 | 0.11 | 0.17 | 0.22 | 0.28 | 0.33 | 0.39 |
| 1550 | 0.065 | 0.13 | 0.19 | 0.26 | 0.32 | 0.39 | 0.45 |
| 2350 | 0.07 | 0.15 | 0.22 | 0.29 | 0.39 | 0.44 | 0.52 |
| 2700 | 0.08 | 0.16 | 0.24 | 0.32 | 0.40 | 0.48 | 0.56 |
| 3300 | 0.09 | 0.17 | 0.26 | 0.35 | 0.44 | 0.52 | 0.61 |

(Diesel engine) Table (6)

| R.P.M. \ T sec | 0.01 | 0.02 | 0.03 | 0.04 | 0.05 | 0.06 | 0.07 |
|---|---|---|---|---|---|---|---|
| 1350 | 0.045 | 0.090 | 0.138 | 0.180 | 0.230 | 0.270 | 0.320 |
| 1550 | 0.046 | 0.092 | 0.140 | 0.184 | 0.230 | 0.280 | 0.322 |
| 2350 | 0.047 | 0.094 | 0.142 | 0.190 | 0.236 | 0.282 | 0.330 |
| 2700 | 0.048 | 0.096 | 0.143 | 0.191 | 0.240 | 0.286 | 0.334 |
| 3300 | 0.050 | 0.100 | 0.150 | 0.200 | 0.244 | 0.290 | 0.340 |

## 1.7- DIFFERENT PROPOSED METHODS OF VIBRATION MINIZATION:

Vibration in irrigation pumps can be minimized using the following three proposed methods:

### 1.7.1- A study of using different materials for suction system:

Vibration measurements were done using two types of suction pipes, one of steel which is used in the project of electrification of irrigation means in Menoufia Governorate and the other is of a rubber material which is usually used in irrigation pumps.

Fig. (12) illustrates the measuring points in the pump under study using the measuring station of vibration which is used in our investigation.

The effect of using suction pipes of different material on vibration was carried out using two types of primemovers (electric and diesel motors) and using a conical pullyes system of five speeds (1350, 1550, 2350, 2700 and 3300 R.P.M.). The vibration was measured at different values of the capacity (Q) to obtain its effect on vibration at suction side.

1.7.2- Using of flixable coupling between the motor and the irrigation pump.

The rigid couplings were used in the research to study the effect of motor vibration on the system. In this work a flixable coupling is used to study its effect on the vibration of the pump. The study was carried out using two types of prime movers (electric and diesel motors) and using a conical pulleys system of five speeds (1350, 1550, 2350, 2YOO and 3300 R.P.M.). The vibration was measured at different capacities (Q) to obtain its affect on the vibration at bearing No. I besides the motor.

1.7.3- Release of the Non-Return Valve:

The part of worke aims to studying the effect of releasing the non-return valve on the vibration during the non-off operation. The study was carried out on the same system using a pulleys system of five speeds and different capacities to study their affect on the vibration of the system when releasing the non-return valve.

The problem that faces us in this study is the preparation processes which takes along time using the preparation pump. For ideal preparation this unit must have no leakage in any part of it.

1.8 <u>EXPERIMENTAL RESULTS</u>:

1.8.1- Torsional vibration results:

Table (7) gives the torsional moment using different prime-movers at different speeds and different types of water.

Table (7)

| Torsion Moment | SPEEDS R.P.M. | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1350 | | | 1550 | | | 2350 | | | 2700 | | | 3300 | | |
| | D.W. | M.W. | | D.W. | M.W. | | D.W. | M.W. | | D.W. | M.W. | | D.W. | M.W. | |
| | | $\frac{1}{40}$ | $\frac{1}{20}$ | | $\frac{1}{40}$ | $\frac{1}{20}$ | | $\frac{1}{40}$ | $\frac{1}{20}$ | | $\frac{1}{40}$ | $\frac{1}{20}$ | | $\frac{1}{40}$ | $\frac{1}{20}$ |
| Diesel Motor | 8.9 | 9.9 | 10.9 | 9.8 | 11.3 | 12.2 | 18.2 | 18.9 | 19.9 | 25.0 | 25.6 | 26.6 | 30.8 | 32.9 | 35.6 |
| Electric Motor | 6.5 | 7.1 | 7.5 | 9.4 | 10.0 | 10.7 | 17.2 | 17.4 | 18.2 | 24.1 | 25.0 | 25.8 | 29.2 | 32.2 | 34.7 |
| Torsional moment factor* | 1.37 | 1.39 | 1.45 | 1.04 | 1.13 | 1.14 | 1.06 | 1.06 | 1.09 | 1.04 | 1.02 | 1.03 | 1.05 | 1.02 | 1.03 |

* Torsional Moment Factor = $\frac{\text{Torsional moment of diesel motor}}{\text{Torsional moment of electric motor}}$

1.8.2- Vibration results of the suction system:

Table (8) gives the vibration values at different speeds, average capacity and different prime-movers for the rubber pipes with respect to steel pipes (at suction side).

Table (8)

| Prime movers \ R.P.M. | | | 1350 | 1550 | 2350 | 2700 | 3300 |
|---|---|---|---|---|---|---|---|
| Electric Motor | $110\ m^3/hr.$ | Amp. | 18.5 | 22.4 | 26.8 | 14.0 | 9.2 |
| | | Vel. | 2.0 | 2.4 | 3.4 | 1.7 | 1.2 |
| Diesel Engine | | Amp. | 22.0 | 26.2 | 31.0 | 17.5 | 11.4 |
| | | Vel. | 2.30 | 2.7 | 4.2 | 2.2 | 1.5 |
| Electric Motor | $150\ m^3/hr.$ | Amp. | 25.2 | 30.5 | 36.5 | 19.1 | 12.5 |
| | | Vel. | 2.7 | 3.3 | 4.6 | 2.3 | 1.6 |
| Diesel Engine | | Amp. | 30.0 | 35.7 | 42.3 | 23.9 | 15.5 |
| | | Vel. | 3.1 | 3.7 | 5.7 | 3.0 | 2.0 |

1.8.3- Vibration results for changing the stiffness (coupling):

Table (9) gives the percentage increasing ratio of vibration values at different speeds, average capacity and different types of couplings for the rigid coupling with respect to flexible coupling (horizontal and vertical measurements for bearing No. 1) using electric motor. Bearing No. 1 was chosen to be measured because it is lay near to the motor and consequantly it is largely affected by the coupling type.

Table (9)

| Measure position \ R.P.M. | | | 1350 | 1550 | 2350 | 2700 | 3300 |
|---|---|---|---|---|---|---|---|
| Horizontal | 110 $m^3$/hr. | Amp. | 5.0 | 4.0 | 1.6 | 4.0 | 4.0 |
| | | Vel. | 8.0 | 8.0 | 24.0 | 7.0 | 8.0 |
| Vertical | | Amp. | 4.5 | 4.2 | 15.9 | 4.0 | 4.1 |
| | | Vel. | 7.5 | 8.1 | 23.0 | 7.1 | 8.2 |
| Horizontal | 150 $m^3$/hr. | Amp. | 6.8 | 5.5 | 21.8 | 5.5 | 5.5 |
| | | Vel. | 10.9 | 10.9 | 32.7 | 9.5 | 10.9 |
| Vertical | | Amp. | 6.1 | 5.7 | 21.7 | 5.5 | 5.6 |
| | | Vel. | 10.2 | 11.0 | 31.4 | 9.7 | 11.2 |

**1.8.4- Vibration results of the releasing of non-return valve.**

Table (10) gives the amplitude for the release of the non-return valve with respect to up release of it for different prime-movers and different speeds. (For average capacity).

Table (10)

| Prime mover \ R.P.M. | | 1350 | 1550 | 2350 | 2700 | 3300 |
|---|---|---|---|---|---|---|
| Electric Motor | 110 $m^3$/hr | 0.38 | 0.31 | 0.71 | 0.35 | 0.58 |
| Diesel Motor | | 0.39 | 0.28 | 0.20 | 0.33 | 0.59 |
| Electric Motor | 150 $m^3$/hr. | 0.52 | 0.42 | 0.23 | 0.47 | 0.79 |
| Diesel Motor | | 0.53 | 0.38 | 0.27 | 0.45 | 0.80 |

1.9.- CONCLUSION:

The torsional vibration mainly depends upon the constructive values of the system (moment of inertia and stiffness of considerable shafts), the form of starting and working moment, and the transidient time/starting (on) and braking (off).

From Figs (13a and 13b) and work 1* to minimize the dynamic loads near to the static, we must reach $\lambda = (8 : 10)$, that means the controlled switching time T must be within (1.46) second for E.M. and (1.76) second for diesel engine.

In formula (1) $M_{12}$ dynamic = (1.03 : 1.037) $M_{12}$ static, when $\lambda = 8.5 : 10.5$.

At the working conditions the driving moment of the electric motor is steady, so the torsional vibration is steady too. But the driving moment of the diesel engine is sine wave and that cases (37 : 45%) dynamic loading at speed (1350-1550) R.P.M. By increasing the R.P.M. that ratio decreases to(4:14%).

It may be recommended that it is preferable to design the (H.P.C.N.I.P.) works of 2500:3000 R.P.M.

The torsional vibration increases by the use of diesel engine because of the transedient conditions, the form of the working driving torsional moment, and circumstances.

The torsional vibration in irrigation low head pumps is affected by changing the water pureness. The torsional (vibration increases with the increase of the mud ratio. When the mud ratio is 1/40 the torsional vibration increases by 3% than that of drinkage water and by 9% for 1/20 mud water when using electric motor as a prime mover as shown in Fig. (13d). When the diesel motor is used this ratio becomes 14% and 22% for 1/40 and 1/20 mud water respectively, that is larger than what had obtained using electric motor - as shown in Fig. (14). So it is clear that the torsional vibration using diesel engine is larger than using electric motor by 0.8%, 12% and 13% for drinkage water, 1/40 and 1/20 mud water respectively which illustrated in Fig. (15).

The increase in torsional vibration for different types of water is due to changing the specific gravity of water resulting from increasing the mud ratio which increases the water moment of inertia and consequantely increases the torsional vibration. 8*.

In the case of using steel suction pipe, the dynamic of the system is take a steady values during working, But its take a variable values when using rubber types.

The relation between the capacity (Q) and amplitude (Amp) using steel and rubber pipes in suction system is shown in Fig. (16). From the figure it is clear that the amplitude in case of rubber pipes is higher than that of steel pipes at the same capacity by a ratio of (1.3) approximately. This increase in amplitude is approximately constant for the different speeds even when the speed reaches 2350 R.P.M.. When the speed exceeds 2350 R.P.M. the amplitude begins to decrease but the amplitude of rubber is still higher than that of steel with the same ratio for speeds (2700 and 3300 R.P.M.).

The relation between the capacity (Q) and vibration velocity (v) is given in Fig. (17) which has the same trend as that obtained for the (Q - Amp.) relation.

The relation between the capacity (Q) and acceleration ($A_{cc}$) has the same trend as that obtained for the (Q-V) relation.

The increase in the capacity (Q) up to 180 $mt^3$/hr. leads to an increase in the vibration and then its beigin to decrease for discharges more than 180 $mt^3$/hr.

The increase in vibration resulting from using rubber pipes is due to the low rigidity of rubber pipes which makes it easy to vibrate, the rubber pipes is fixed from the pump side only and its have a waveness surfaces.

In the case of using a both type of couplings (Rigid and flixable) and from the analytic part, its clear that the dynamic of the system increases with using rigid coupling. So in the experimental work this phenomena become more clear specially at $B_1$ beside the coupling.

The relation between the capacity (Q) and the amplitude (Amp) for bearing (1) using rigid and flexible coupling is shown in Fig. (18). The figure shows that the amplitude in case of flexible coupling decreases by 30% than that for rigid coupling which is approximately constant for each speed. Fig. (19) represents the relation between the capacity and vibration velocity which indicates that the vibration velocity increases with the same ratio when using rigid coupling.

The relation between the capacity (Q) and acceleration ($A_{cc}$) takes the same behivour as (Q - V) relations.

The increase in the capacity (Q) up to 180 $mt^3$/hr. leads to an increase in the amplitude and vibration velocity and then the amplitude begins to decrease for the discharge above 180 $mt^3$/hr. at all speeds nearly.

The effect of capacity on the amplitude for both cases on suction pipe at releasing and unreleasing of the non-return valve is shown in Figure (20). The figure shows that the amplitude decreases at the starting operation by 5% when releasing the non-return valve and 40% at the stopping moment. This decreasing percent is approximately constant when using electric or diesel motors. This phenomena is due to get red of part of the hammer resulting from stopping the pump.

The relations of (Q - V) and (Q - $A_{cc}$) takes the same trend as the (Q - Amp.) relations.

1.10- REFERENCES:

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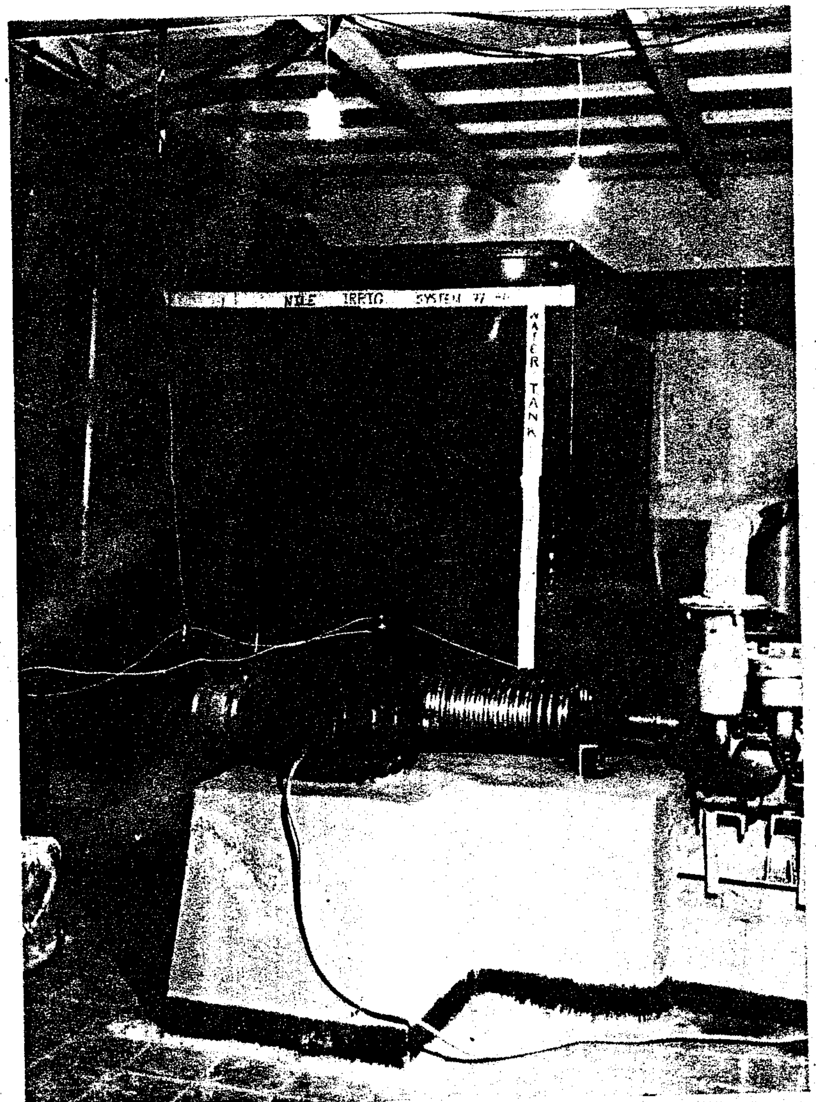
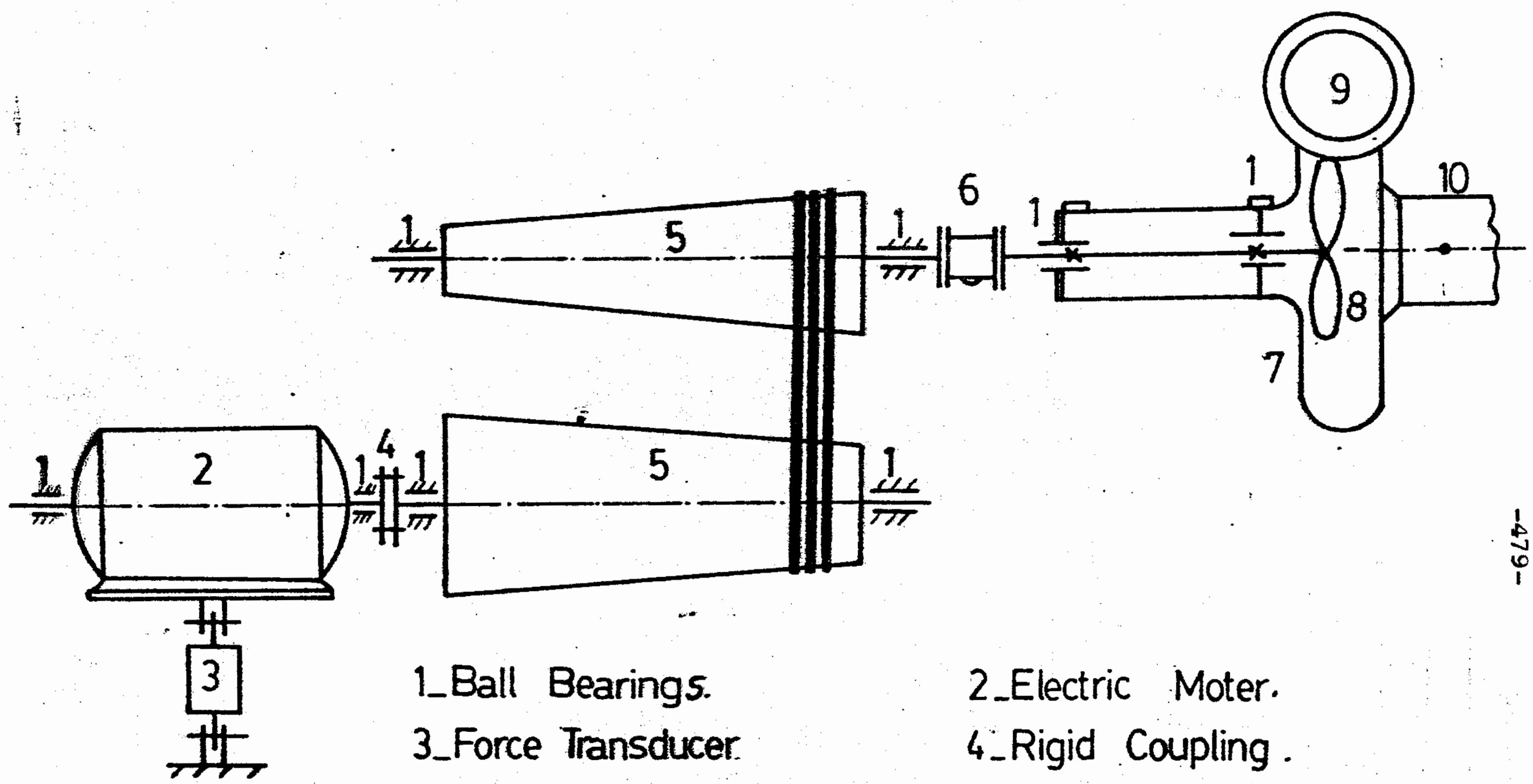



*Fig.(1a). Nile Irrigation Station Under Study.*



1_Ball Bearings.
2_Electric Moter.
3_Force Transducer.
4_Rigid Coupling.
5_Conical Pulleys.
6_Torque Transducer.
7_Casing.
8_Impeller.
9_Delivery Pipe.
10_Suction Pipe.

*Fig.(1 b ) Skitch of the System Used.*

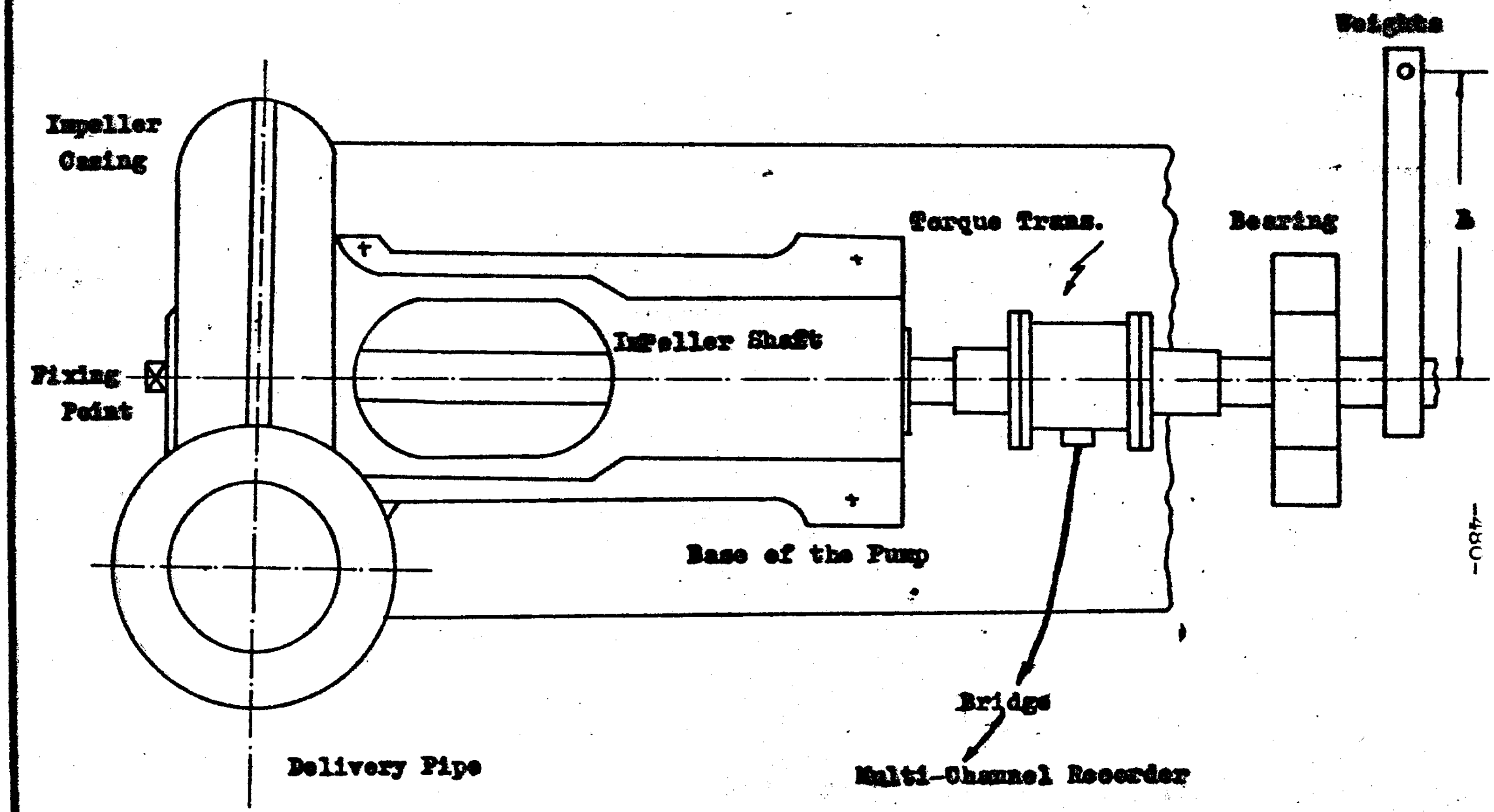



Fig.(2) Calibration of Torque Transducer, METHOD,

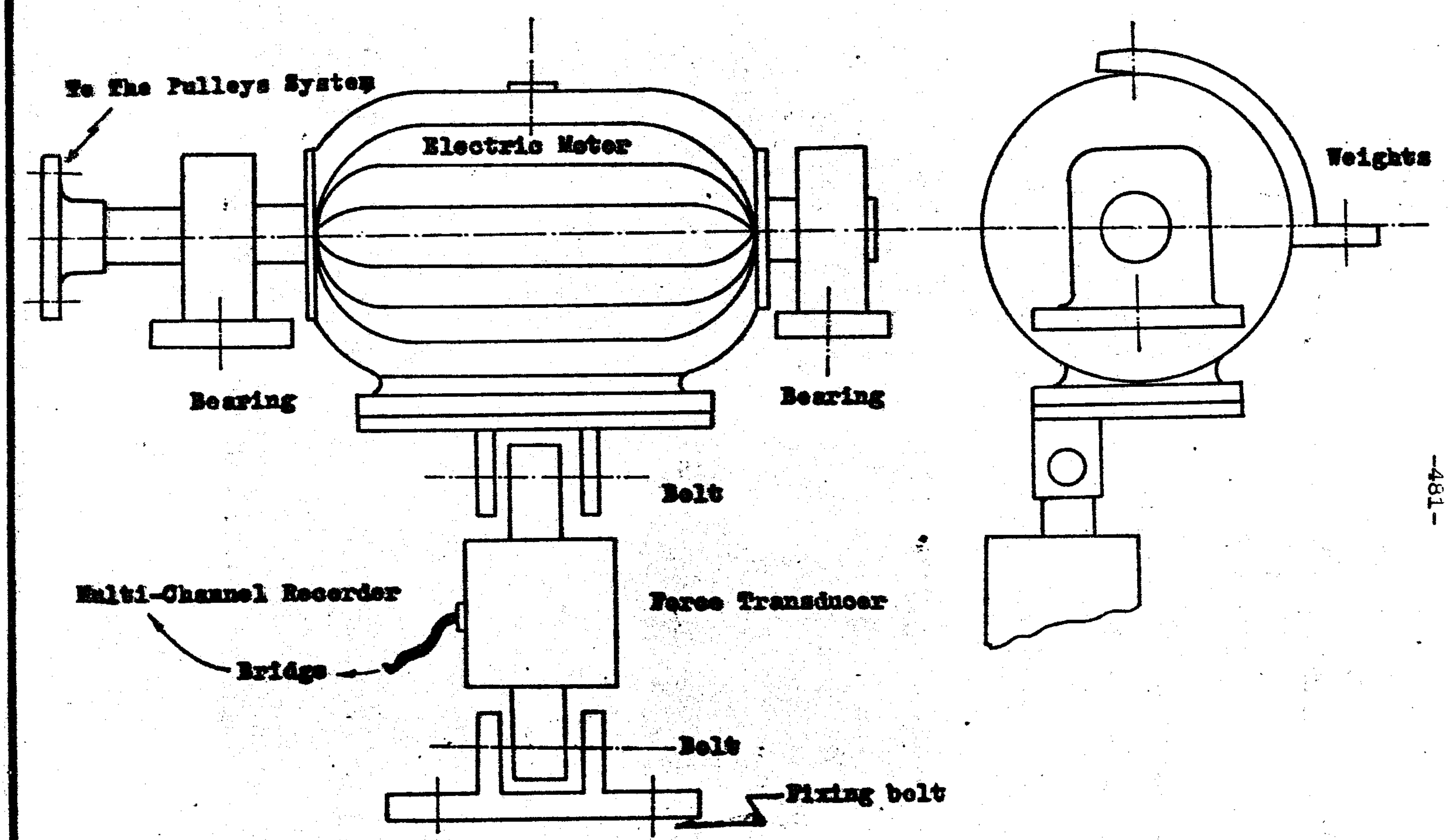



Fig.(3) Calibration of Force Transducer. METHOD.

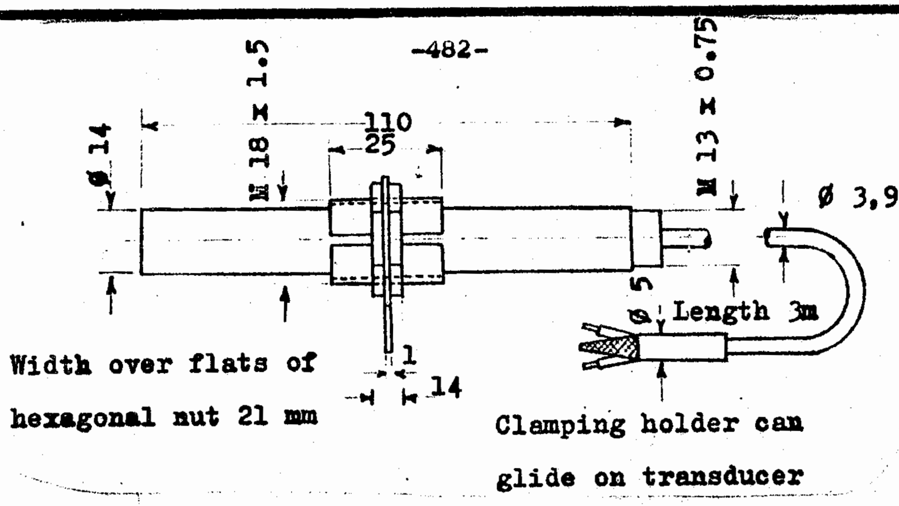
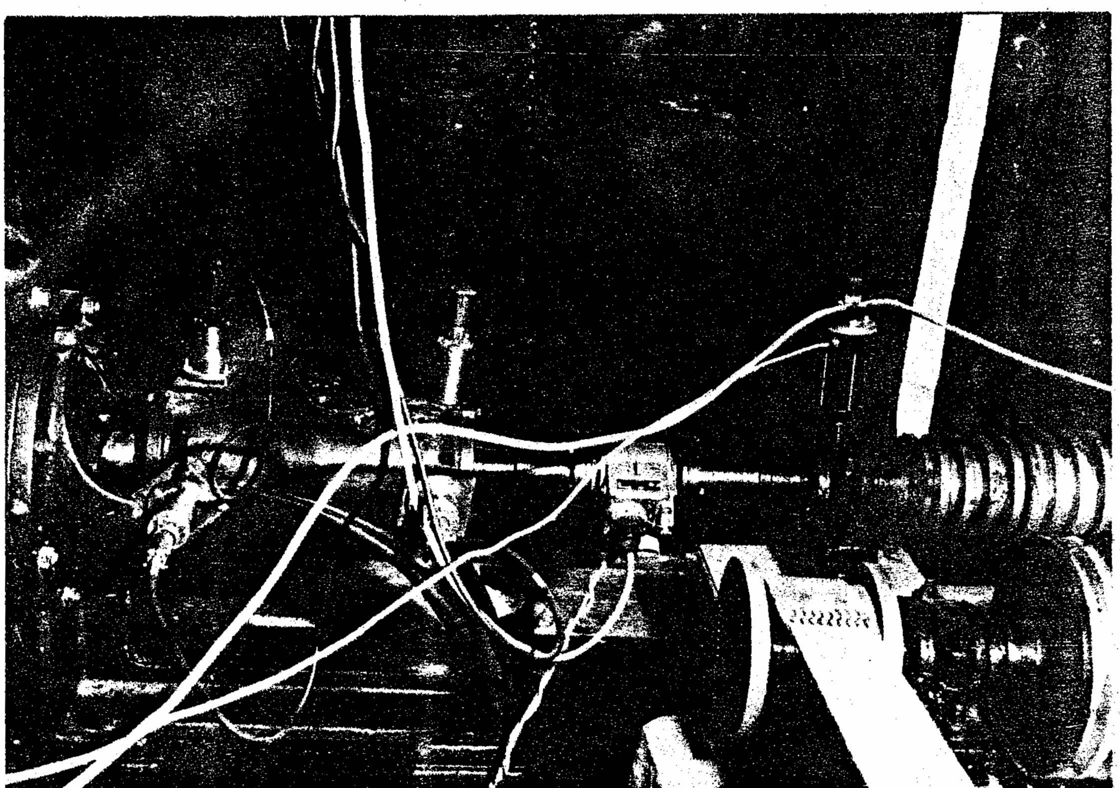
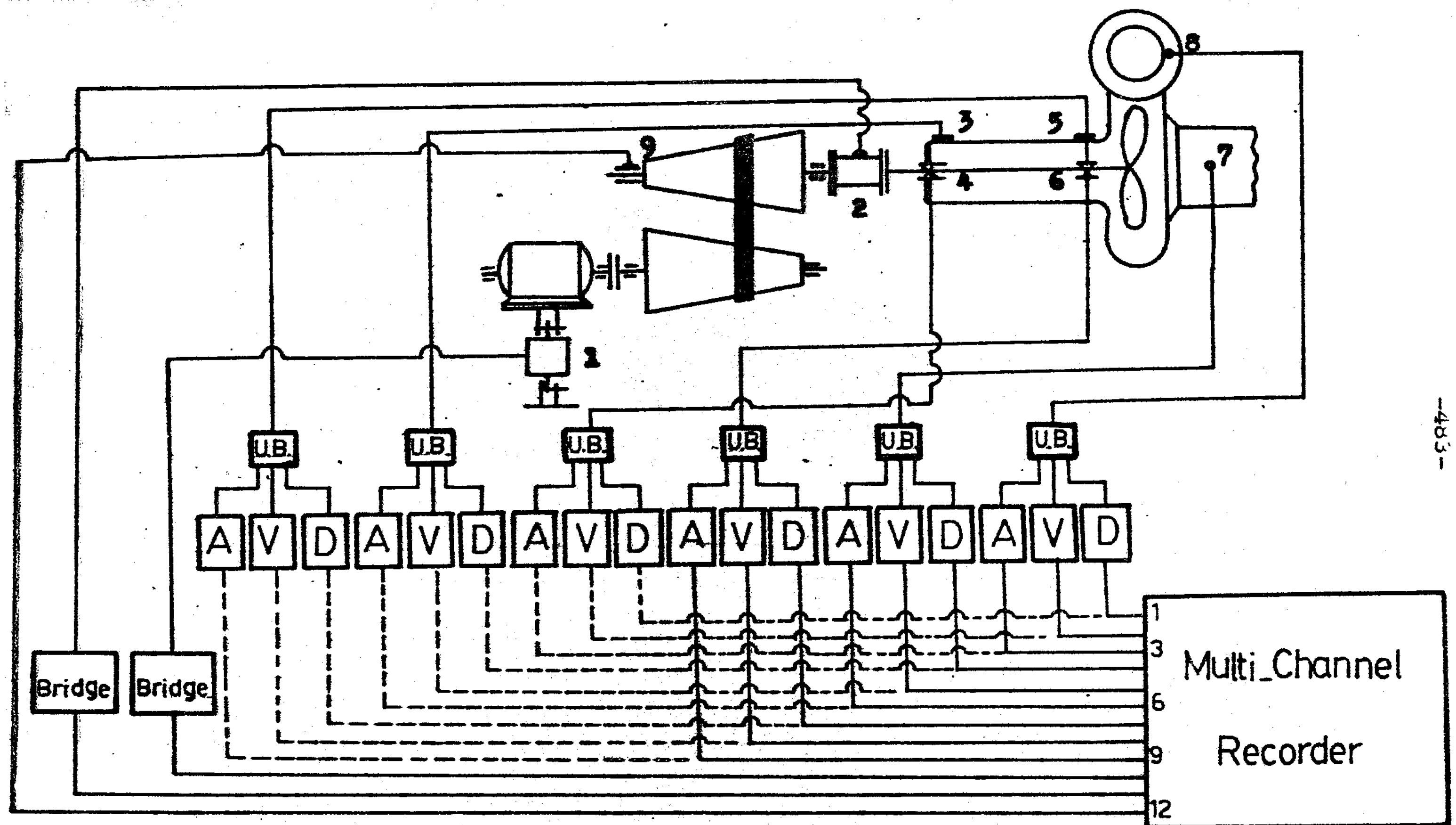



Fig.(4) R.P.M. Transducer PR 9373.



Fig(5) Vibration Transducer on Measuring Parts of the Pump.




1. Force Trans.
2. Torque Trans.
3. Horz. Trans.
4. Vertical Trans.
5. Horz. Trans.
6. Vertical Trans.
7. Sus. Trans.
8. Delivery Trans.
9. RPM Trans.

Fig.( 6 ) Block Diagram of Vibration Measuring System.

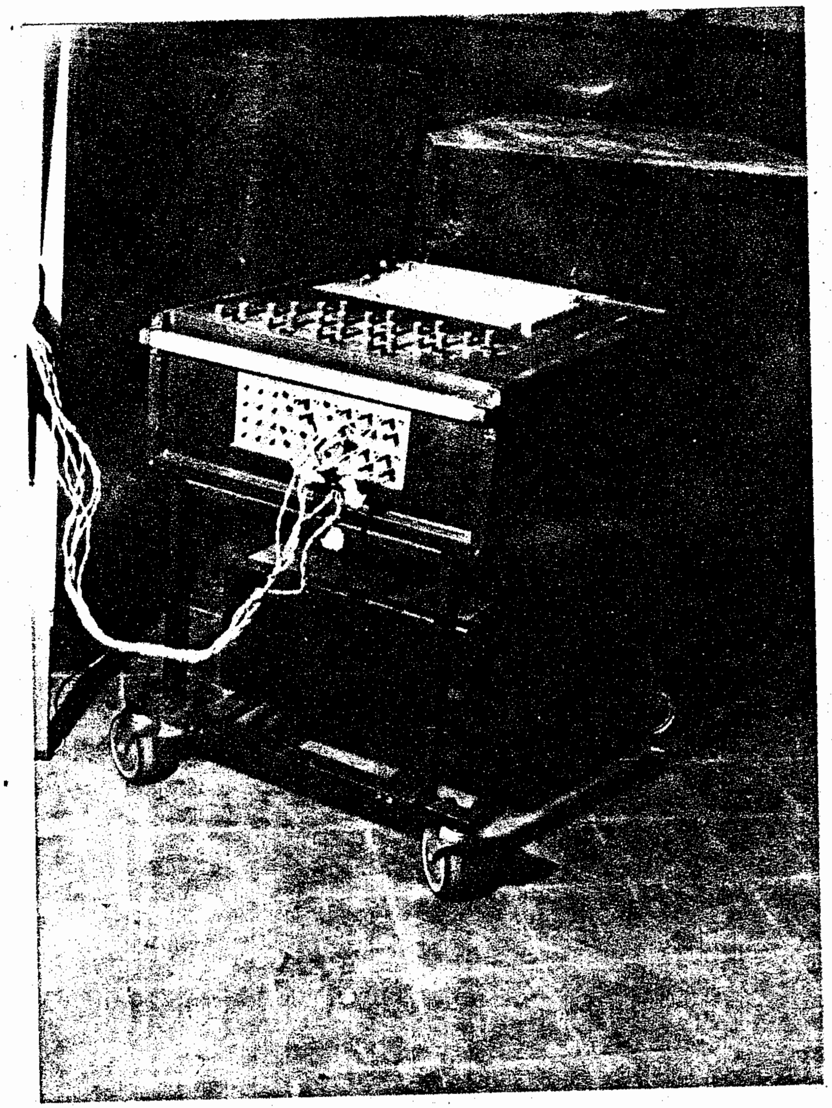


Fig.(7) Multi_Channel Osciloscript PR 9335.

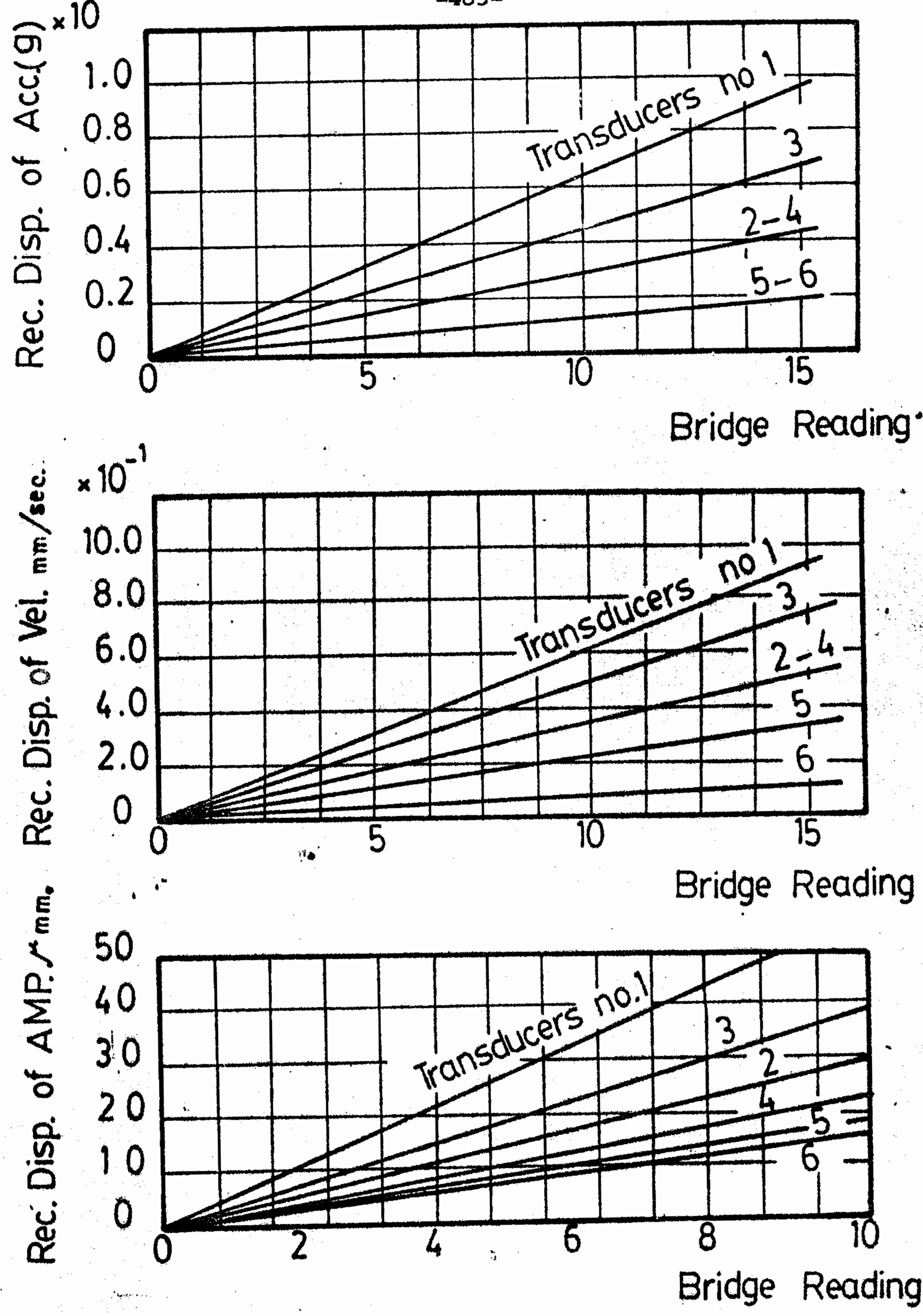



Fig.(8 )Calibration of the Six Vibration Trans.

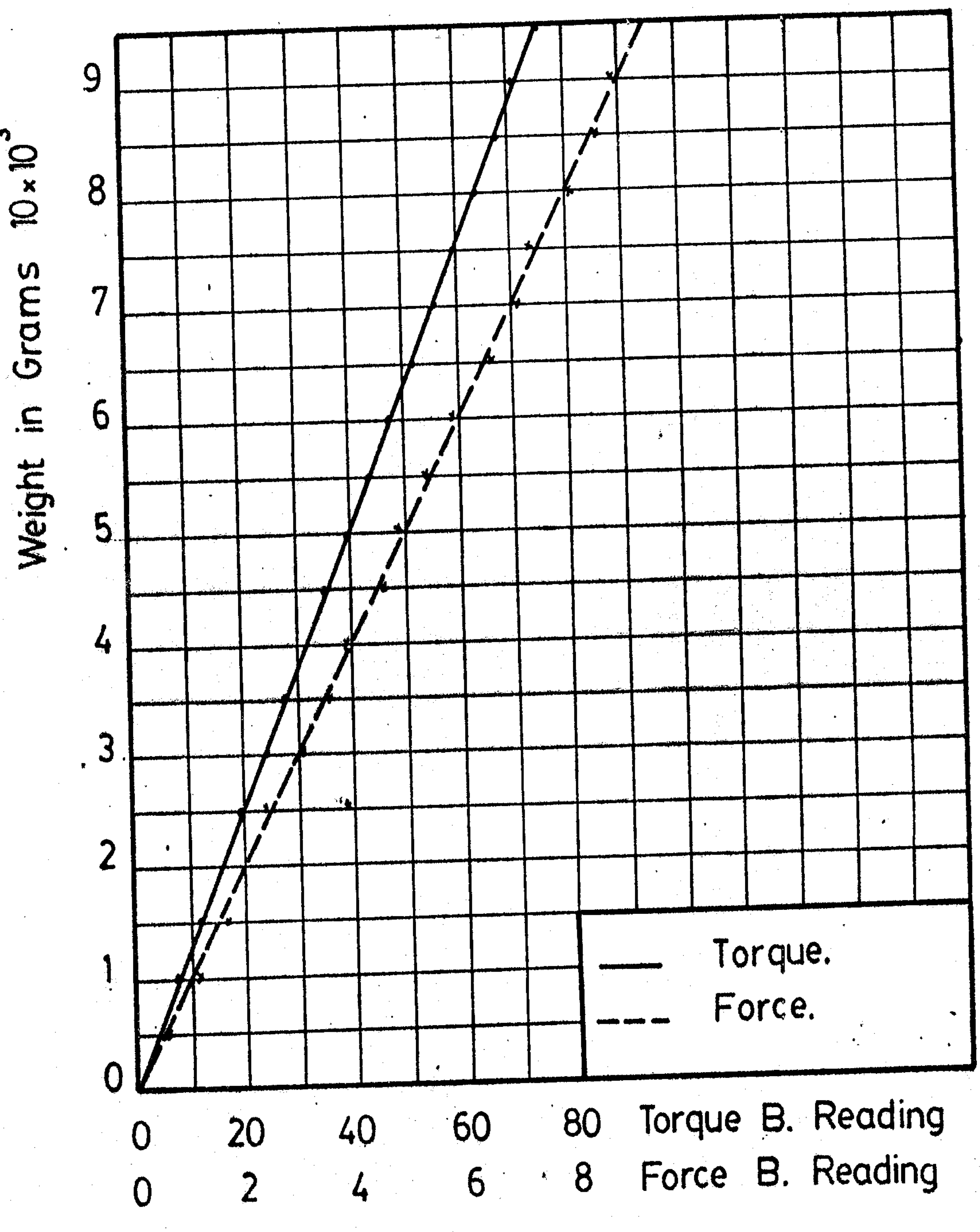



*Fig.(9) Calibration of Force and Torque Transducers.*

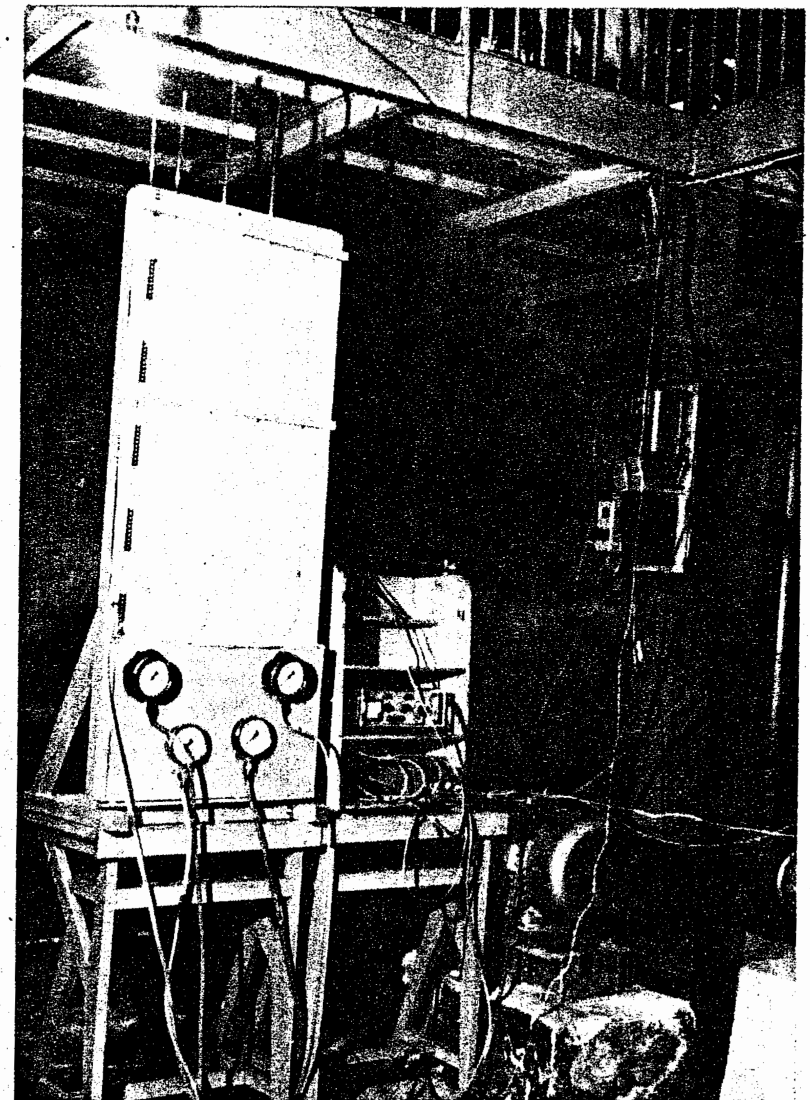


*Fig.(10) Pressure Gauges For Different Pressure Measurements.*

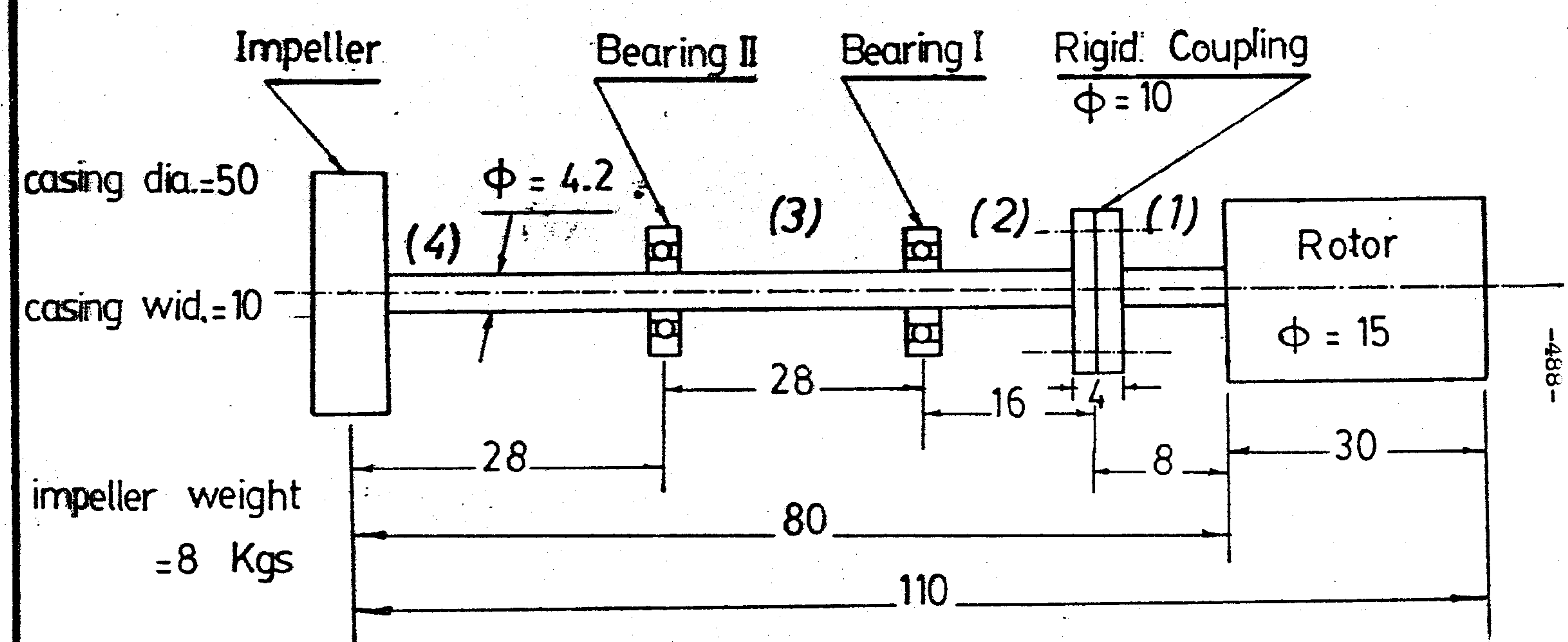



Fig.($11_a$) Mechanical System of Pump.

Dims. in cm.

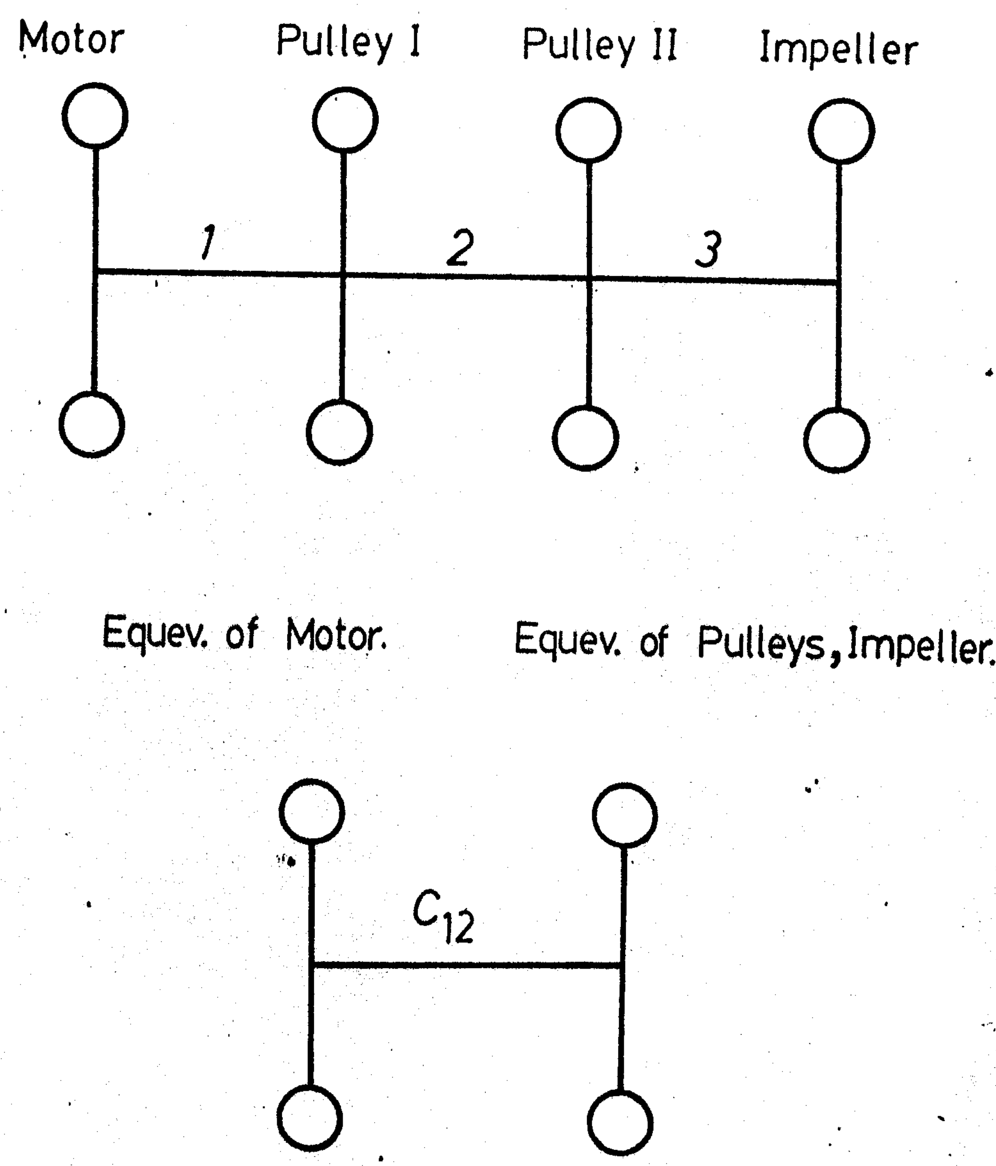



*Fig.(11b) Equevilant Mass System.*

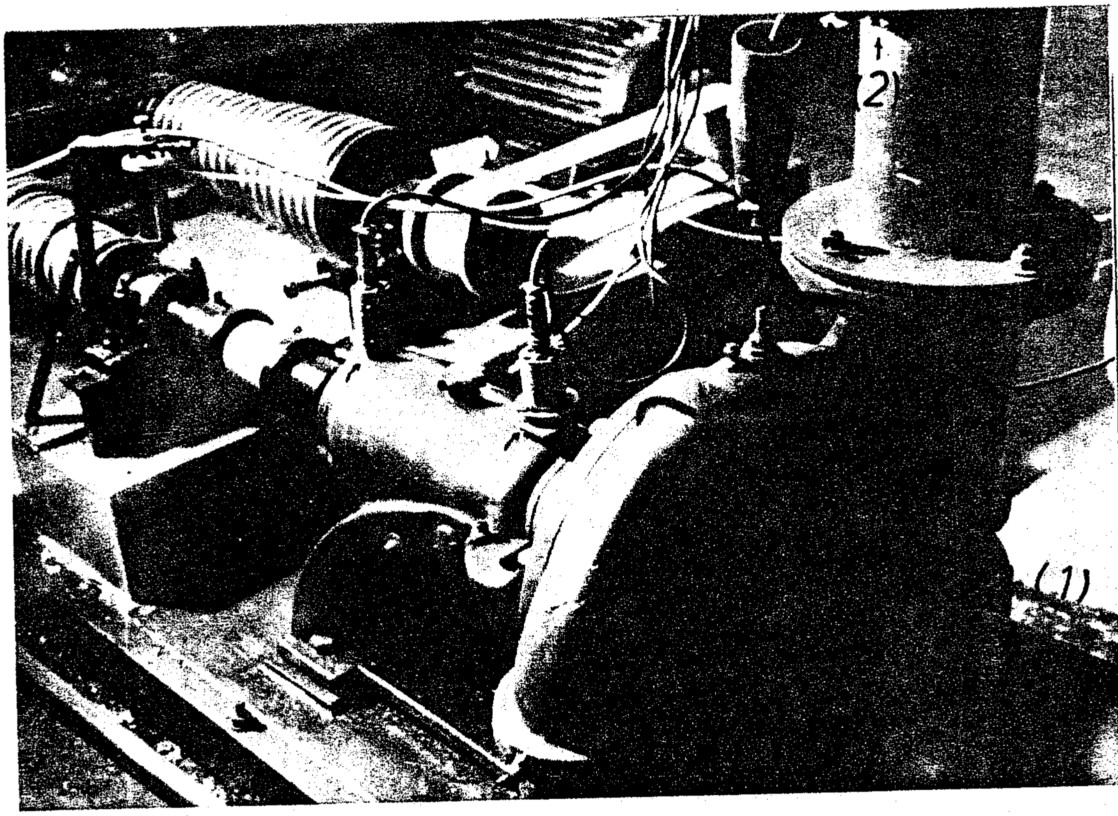
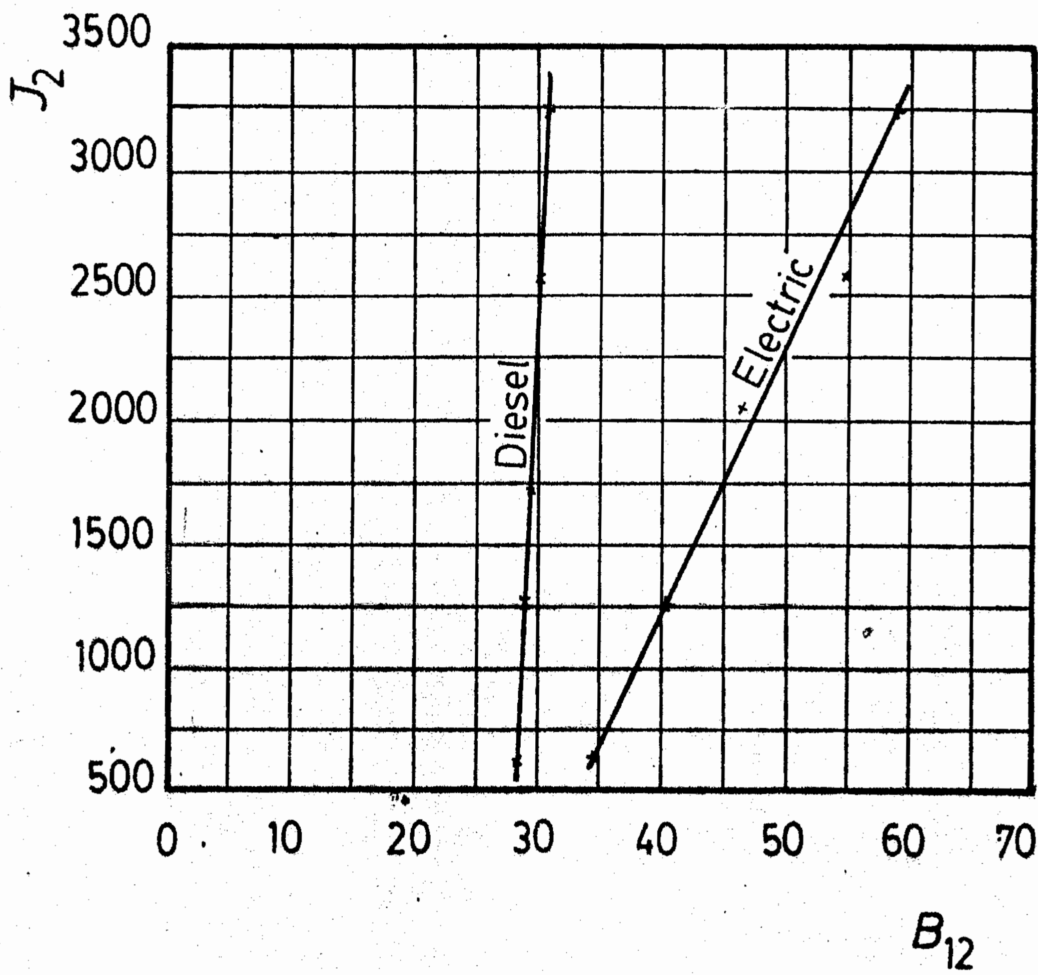



*Fig.(12) Measuring Points at Suction and Delivery Sides.*




*Fig(13a) [Moment of Inertia _ No. of Free Freq.] Relation For Tow Types of Prime _ Movers and Different Speeds.*

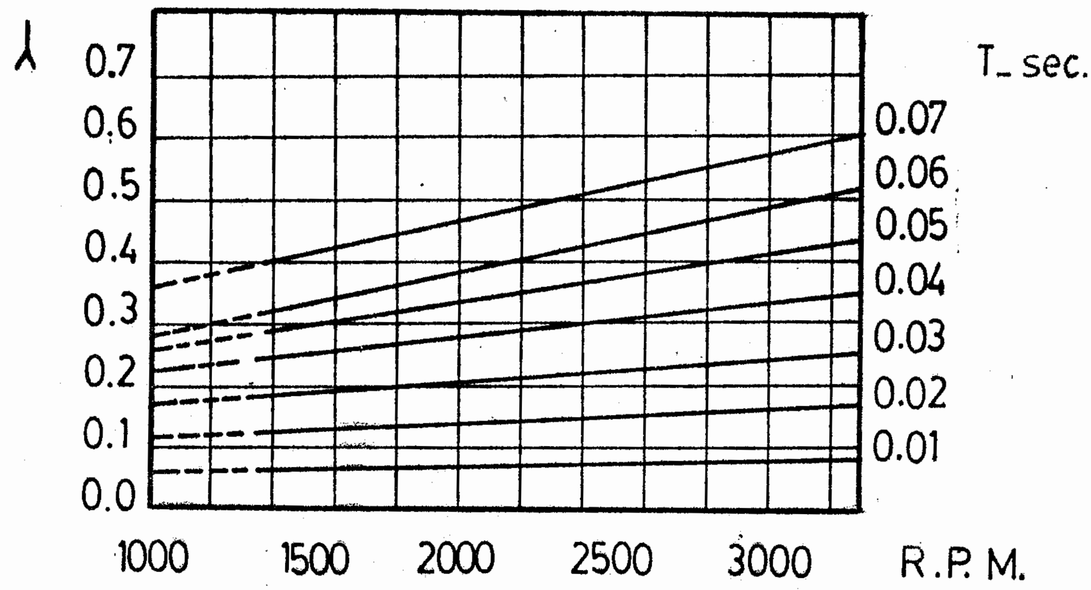



*Fig.($13_b$)[λ_R.P.M.] Relation For Different Speeds and Electric Motor.*

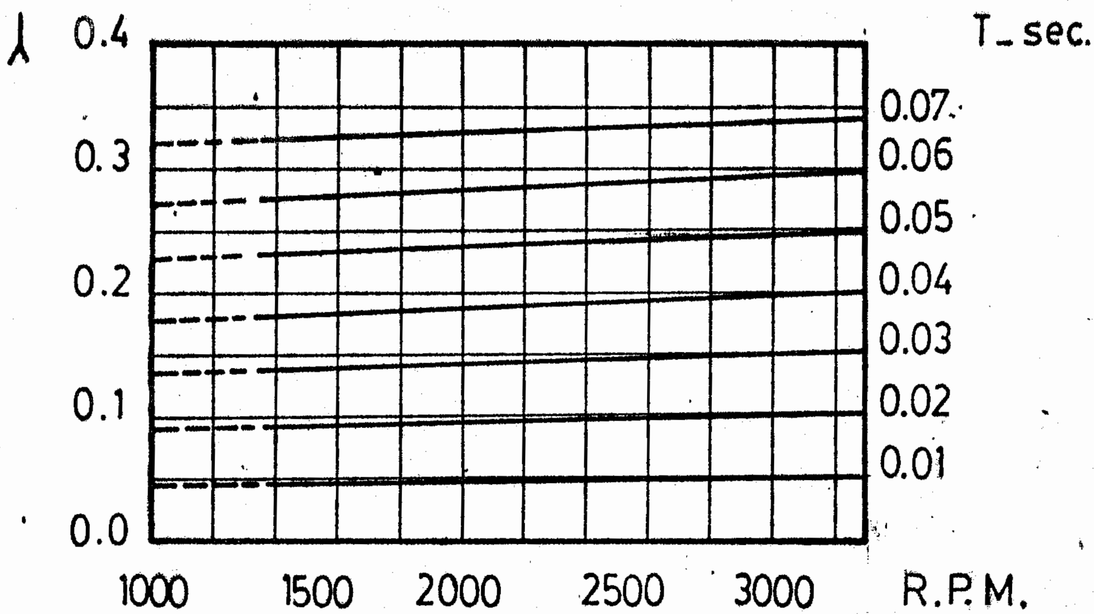



*Fig.($13_c$)[λ_R.P.M.] Relation For Different Speeds and Diesel Engine.*

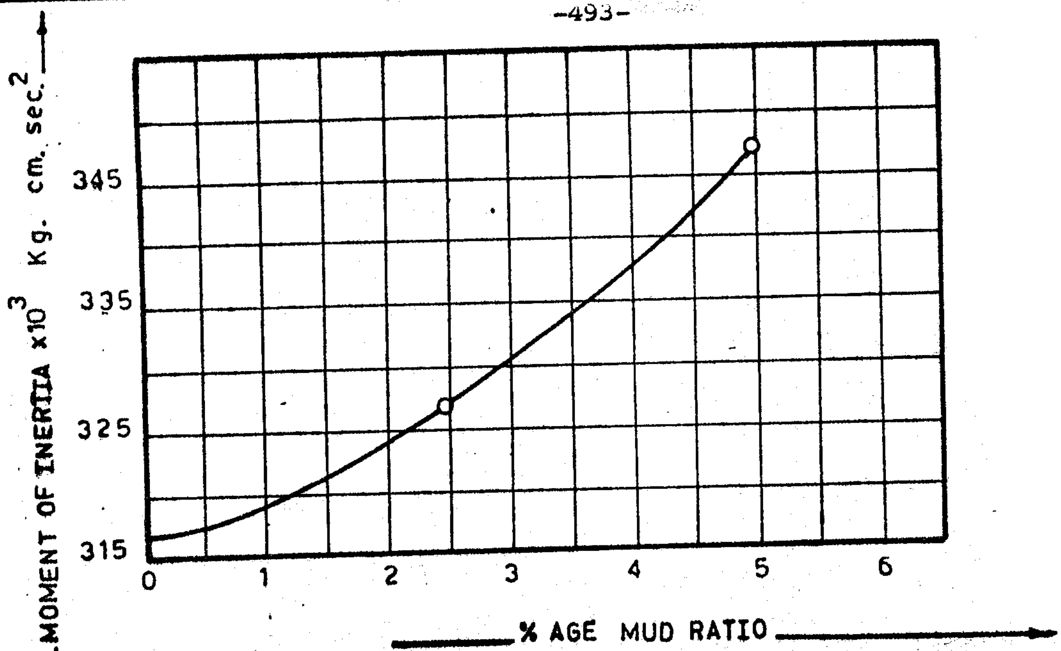



Fig.(13$_d$) Effect of Mud Ratio on the Moment of Inertia Using Electric Motor.

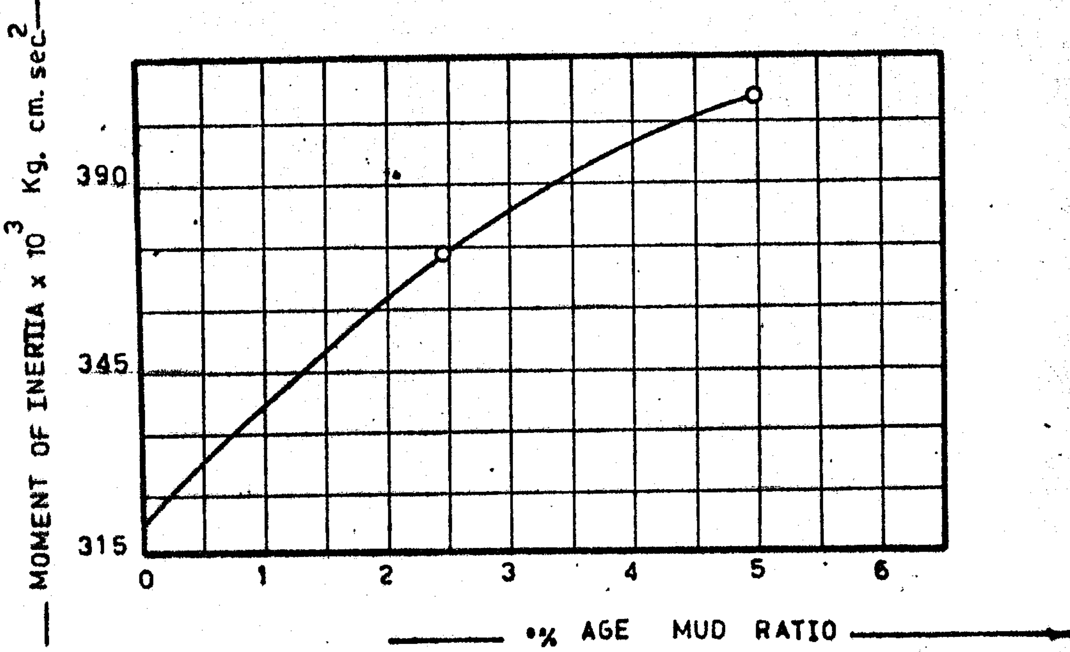



Fig.(14) Effect of Mud Ratio on the Moment of Inertia Using Diesel Motor.

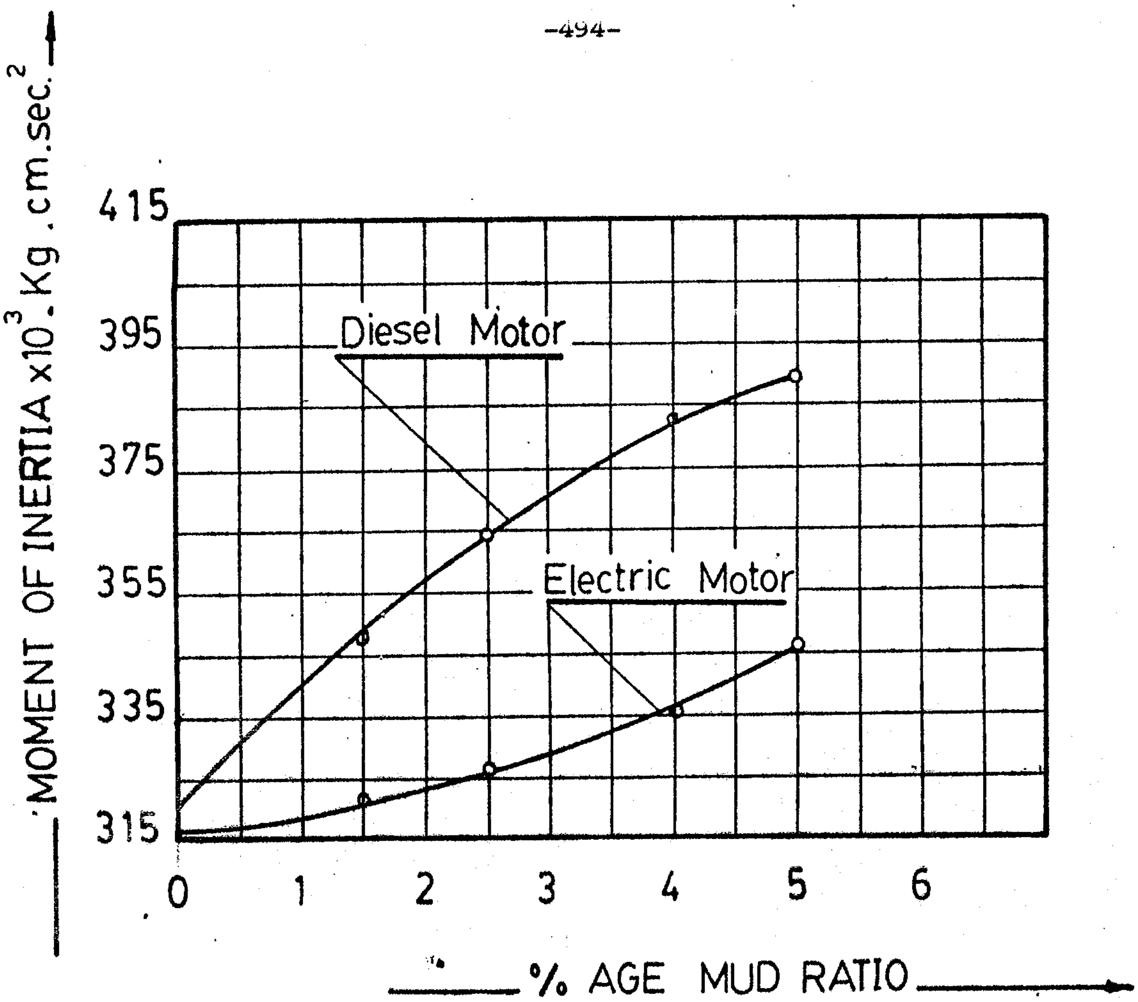



*Fig (15) Comparison Between Two Prime-movers [ Diesel & Electric Motor ].*

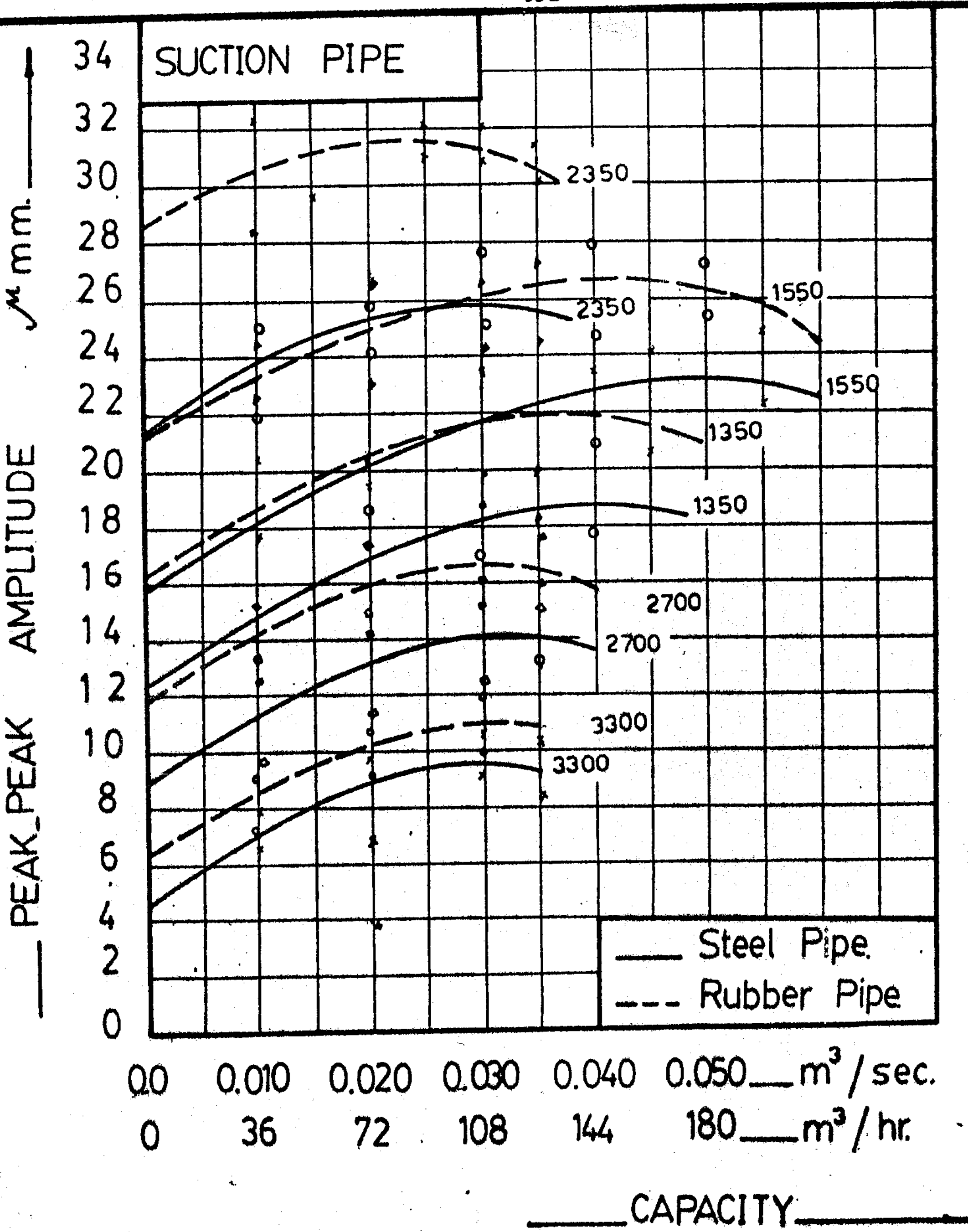



*Fig (1-6) Effect of Capacity (Q) on the Amplitude (Amp) Using Both Types of Suction Pipes and at Different Speeds.*

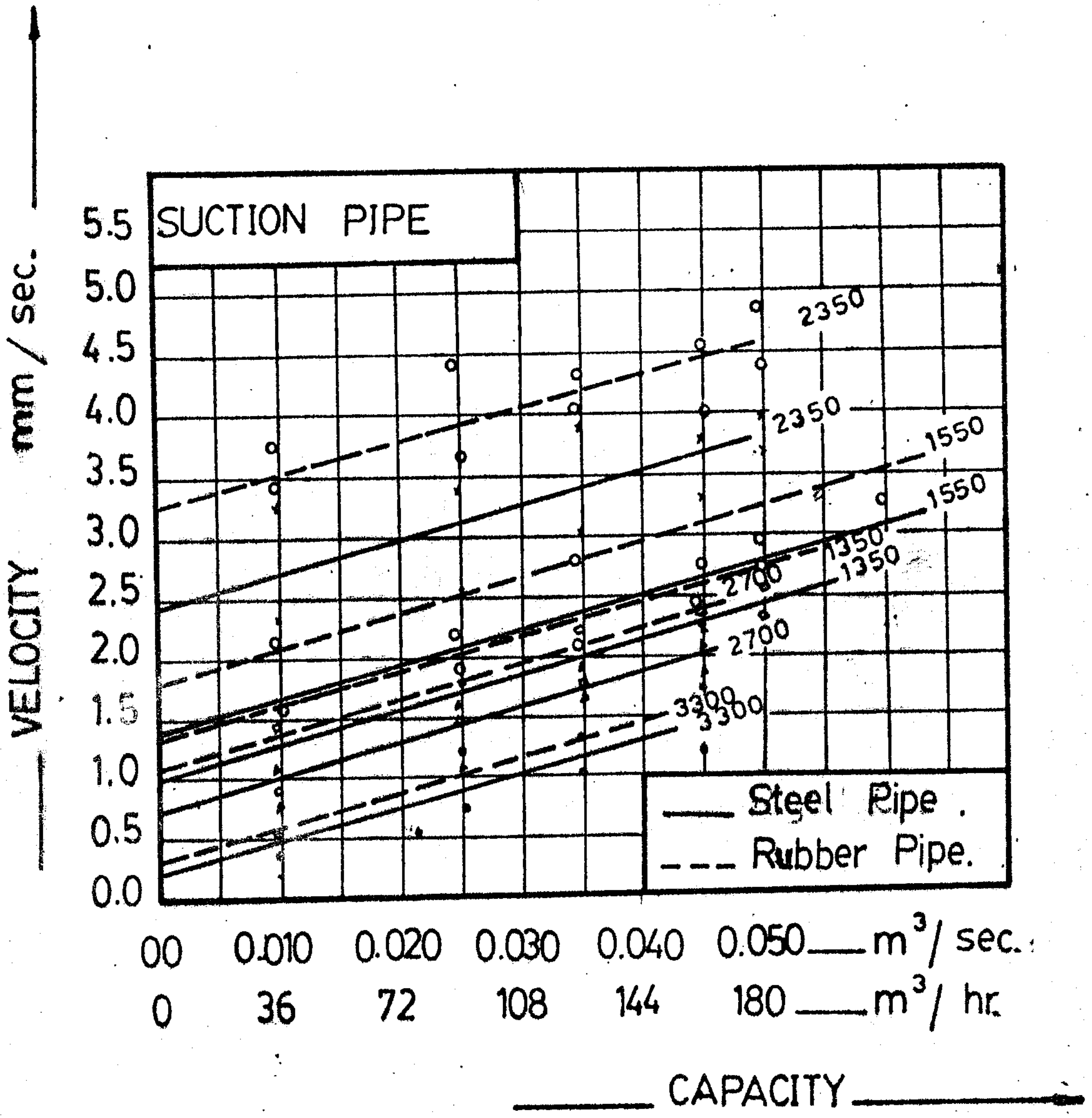



*Fig.(17) Effect of Capaity (Q) on the Velocity (V) Using Both Types of Suction Pipes and at Different Speeds.*

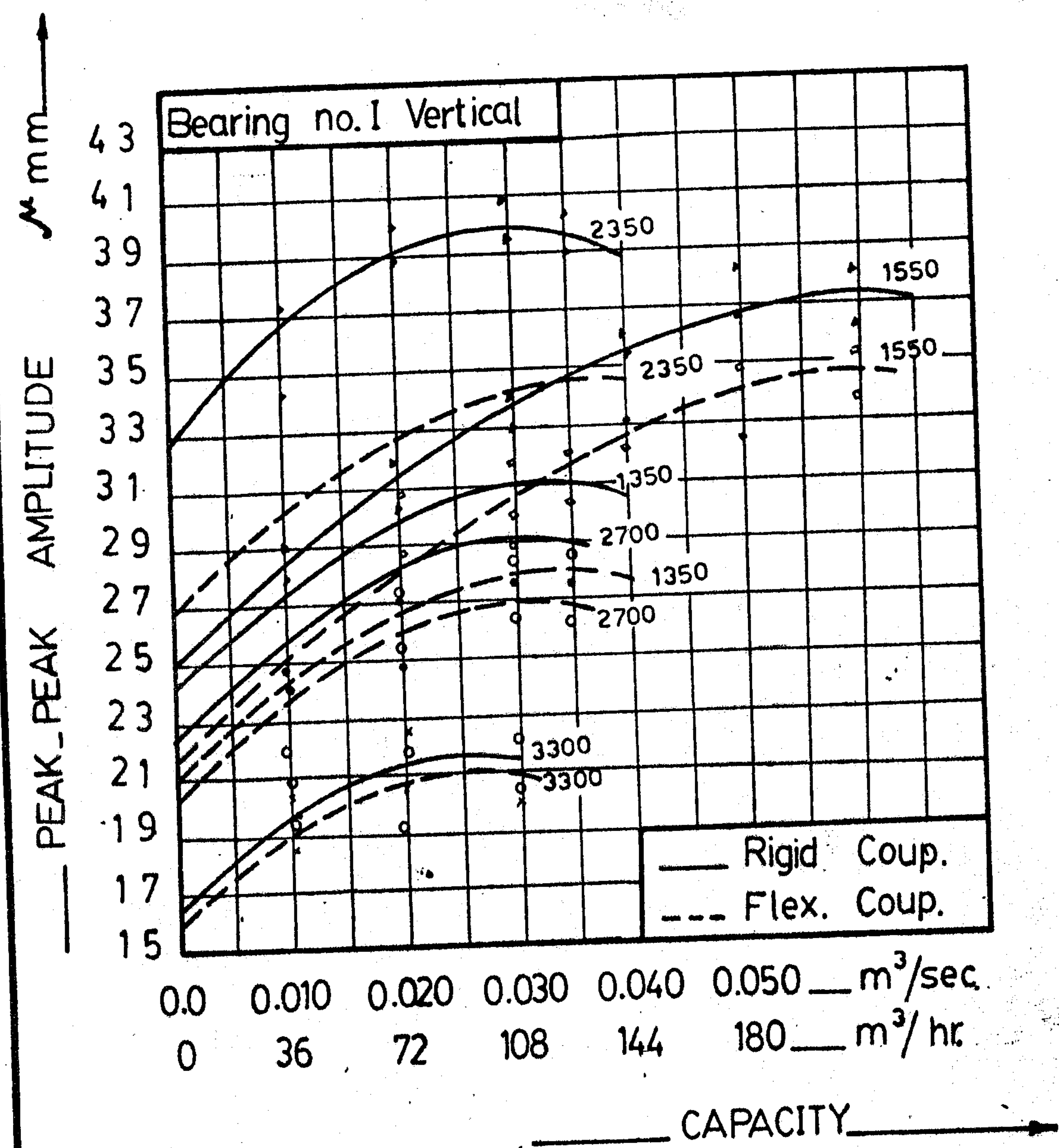



*Fig.(18) Effect of Capacity (Q) on the Amplitude (Amp$_h$) Using Both Types of Couplings and at Different Speeds.*

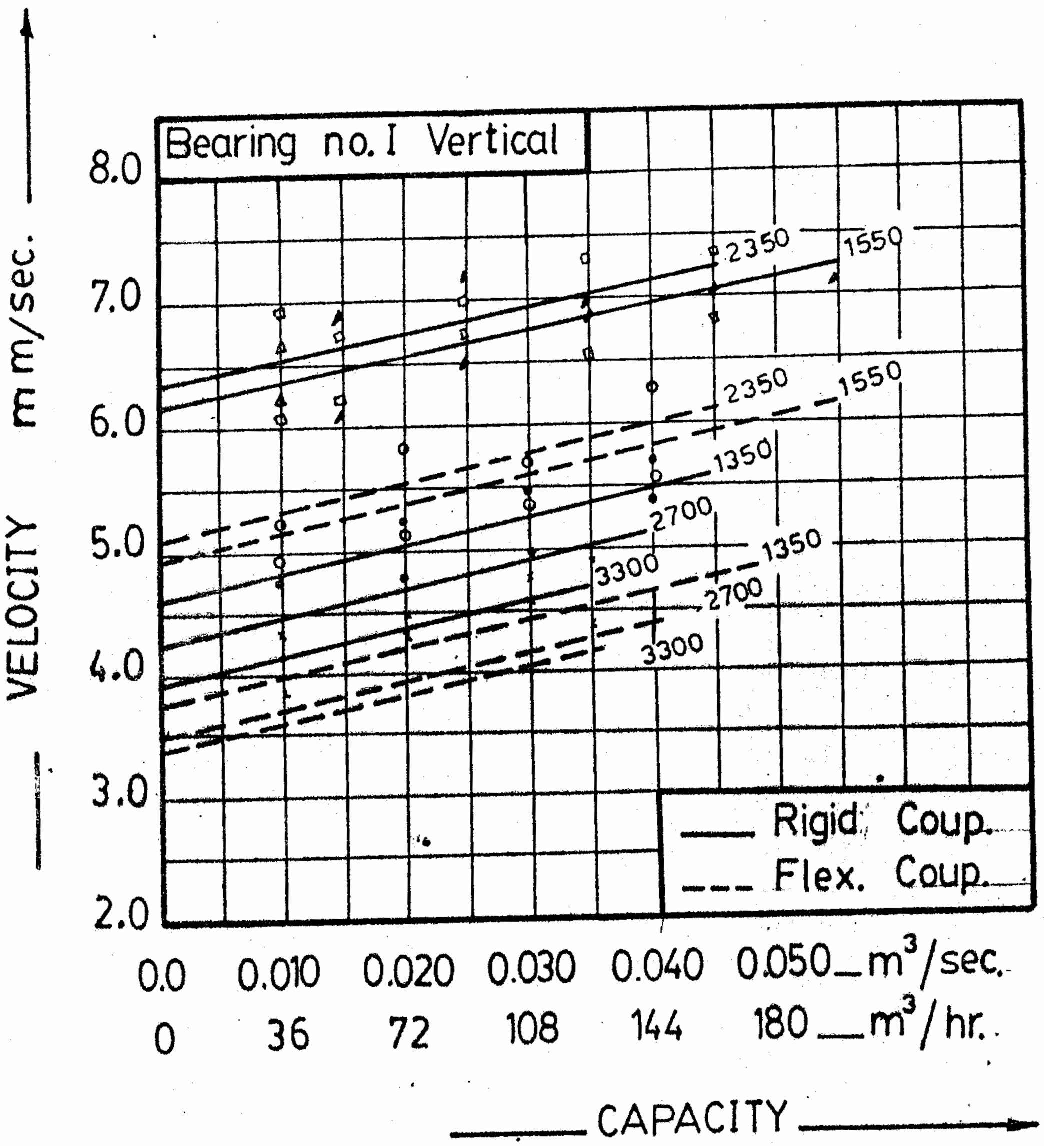



*Fig.(19) Effect of Capaity (Q) on the Velocity ($V_r$) Using Both Types of Couplings and at Diff_erent Speeds.*

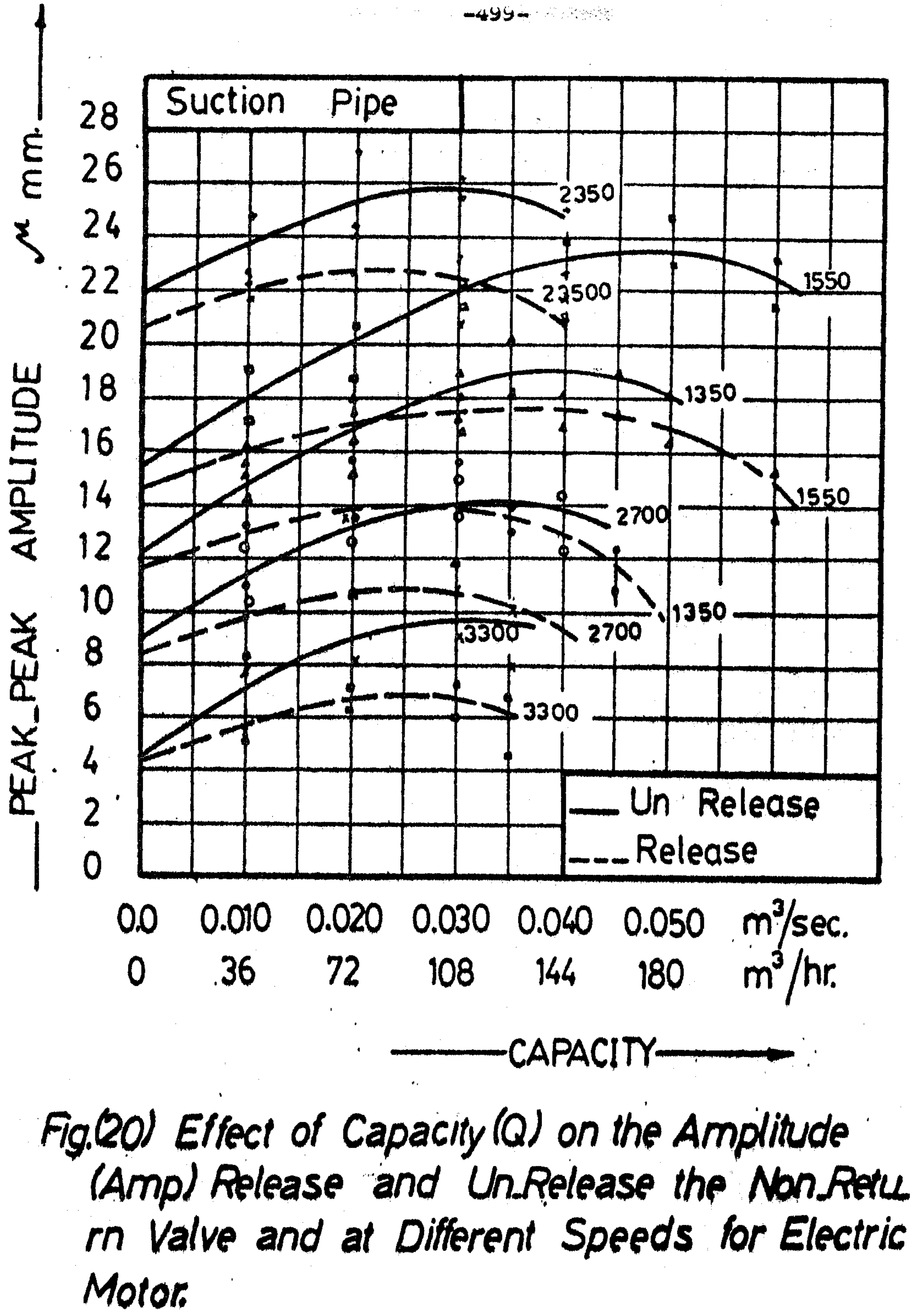



*Fig.(20) Effect of Capacity (Q) on the Amplitude (Amp) Release and Un.Release the Non.Return Valve and at Different Speeds for Electric Motor.*

# الاهتزازات الالتوائية فى مضخات الرى النيلى المنخفضة العلو

أ.د. عبدالهادى ناصر ، أ.د. سعد محمد وهيبه ، أ.د. عصام أحمد سالم
م. أحمد محمود عيسى

هذه المقالة تشتمل على دراسة نظرية وعملية عن الاهتزازات الالتوائية فى مضخات الرى النيلى المنخفضة العلو — ولقد تمت الدراسة باستخدام مصادر طاقة مختلفة واثناء تغيير مواد مواسير السحب (مطاط وحديد) وتغيير القارنة المرنة باخرى جسيئة وايضا تم رفع صمام عدم الرجوع — وذلك لمعرفة اثر كل ذلك على الاهتزازات الناتجة على جسم المضخة (الشاسية)، المضخة ، القارنة ، المحرك ، مجموعة السحب والطرد ).

ولقد تمت القياسات باستخدام مسجل متعدد القنوات (١٢ قناه ) ليتسنى لنا تسجيل مجمل القراءات او اغلبها فى نفس اللحظة .

ولقد ظهر من نتائج هذه الدراسة مايلى :

١ — زيادة قيمة الاهتزازات اثناء استخدام مواسير مطاط للسحب .
٢ — زيادة قيمة الاهتزازات اثناء استخدام قارنة جديدية .
٣ — إن قيمة الاهتزازات تقل برفع صمام عدم الرجوع خاصة لحظتى البدء والايقاف .
٤ — تزداد قيمة الاهتزازات اثناء استخدام محرك ديزل عنه اثناء استخدام موتور كهربى له نفس القدرة .
٥ — يجب زيادة زمن نمو العزم اثناء استخدام موتور كهربى حتى يصل الى ٨ — ١٠ ثانية مقابل ٥ر. ثانية وأيضا يجب زيادة زمن نمو العزم اثناء استخدام محرك ديزل من ١١ — ١٣ ثانية بدلا من ٨ر. ثانية حاليا .