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A Review on Design and Fabrication of Fatigue Testing Machine

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Abstract-- Simpler Design arrangement was used and Rotating Bending Fatigue Test (R.B.F) machine was fabricated using local availability of raw materials to make design arrangement simpler, due of its higher market Price. The design approach was simpler in construction .Power was transmitted though vertically arranged pulley arrangement where it is further transferred to load transmitting members through universal joint. Rotating Mandrel for specimen holding , was press fitted inside the two bearings which are separated apart. Bearing Hub was supported on horizontal lever through bolted connection at either side .One end of Lever is hinged and supported on vertical structure. This design helped in understanding Dynamic scatter and their Prevention on these fatigue machines. Frequent Maintenance showed better performance for longer runs. Scatter observed in Machine can help to overcome by the operator itself. This design made a better understanding for correlation with experiment results. Experimental results performed on steel showed acceptable results.

Index Terms--- Machine, Design , Forces, Members, Fatigue

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1 INTRODUCTION

Prevention of fatigue failure in structural members has been an important concern in automotive industrial and aircraft engineering for many years. Technological developments continually bring out new materials, new fabrication processes, improved design concepts, and additional information about service requirements. Hence, engineering procedures for prevention of fatigue need continual review. The initial investigations into this phenomenon was conducted by August Wohler (1819-1914) [2]. A detailed overview is also given by W. Weibull [1]. The first design constitutes the classical high-speed fatigue machine, introduced by Wohler (1871) four point loading rotating bending specimen.[2]. Stringent safety requirements are common in structural design. A design-by-reliability approach is required in ever more situations and accurate design inputs are needed to meet reliability targets while reducing costs [8].

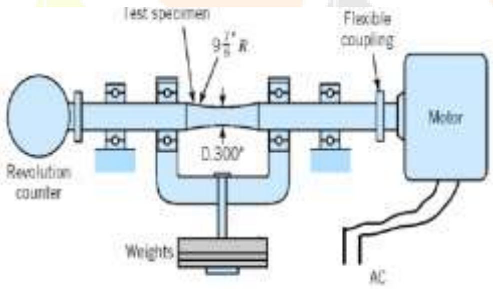


Fig 1 Method of loading and general arrangement of Moore Rotating Bending Fatigue Test Apparatus [9].

FINK and HEMPEL (1951) (1953) found that the accuracy depends upon three different factors [1]

- The design of the machine
- The use of the machine and resulting wear in the bearings
- The proper manipulation of the machine according to established instructions

2 RESEARCH GAP

In designing the fatigue testing machine, we considered that all frame parts are required to be durable and capable of transmitting the necessary forces and performing the necessary motions efficiently and economically without interfering with any other part of the machine [18]. Various number of Industries/author has designed rotating bending fatigue from Wohler to todays Instron .Numerous design modification has been done by various author/ industries to make design simpler [28,30,31,44,45] , Dual spindle additional performance type [25] ,High Speed [32], Combined mode of loading [33] ,efficiency improvement [31] and economical Machine [27] .Indeed, according to [4], in fatigue tests different sources of scatter should be taken into account simultaneously scatter related to, to fatigue test conditions (e.g., test rig) [5,6].Variation of experimental results due to the Fatigue machine scatter has been studied under different mode of loading (uni-axial,3 point,4 point) test set up by various author mostly using electrical Resistance strain gauges [1,21,22,23,24].

Due to the dynamic action involved in RBF machine compared to other fatigue set up, a lesser study was done on modifying the design with simpler arrangement and lack of dynamic scatter calibration was studied [1] .Even fatigue testing machines available in the market are of high cost and complicated in design arrangements. The uncertainty with respect to the scatter of fatigue testing machines has meant that in special cases, where several fatigue testing machines are used in the same study, a statistical comparison of the machines has to be conducted. Laboratory experimentation is a critical final link for a thorough understanding and appreciation of scientific and engineering theories. Every possible effort should be made not to deprive the future engineers or educators from this vital component of their education

[6] It is therefore necessary to continue development of effective and efficient pedagogical methods and techniques for the engineering laboratory experience [7].

The idea behind this thesis work is not to provide answers to the unanswered questions, but to tackle the problem from a different perspective. So here it was planned to design and fabricate a test up at a lower cost, which will help in understanding the problems in a better way.

3 METHODOLGY

Design of Machine elements was procceded by superimposing the Forces and bending moment applied to each element. Design was based on strength and rigidity [12,13,14,19]

3.1 Objective

Design was carried out keeping all the above objective mentioned below

- It should be easily operated
- Machine be capable of withstanding the above mentioned force torque
- It should transmit energy with lower losses
- Easily reapairable
- Design should help in overcoming any dynamic scatter in the machine by the operator itself
- It should help in frequent disassembling and assembly.
- Machine should operate for longer runs without frequent start and stop and monitoring
- Machine should be economical

3.2 Basic Components of Fatigue Testing Machine

Any fatigue testing machine is composed of the following structural components [1]

- Load-producing mechanism
- Load-transmitting members such as grips flexural joints
- Measuring devices for revolutions
- Shut-off apparatus
- Frame work

3.3 General Description

The fatigue-testing machine is of the rotating beam type. The specimen functions as a single beam symmetrically loaded at two points. When rotated one-half revolution the stress in the fibres originally above the neutral axis of the specimen are reversed from compression to tension for equal intensity. Upon completing the revolution, the stresses are again reversed, so that during one complete revolution the test specimen passes through a complete cycle flexural stress.

3.4 Planned Design

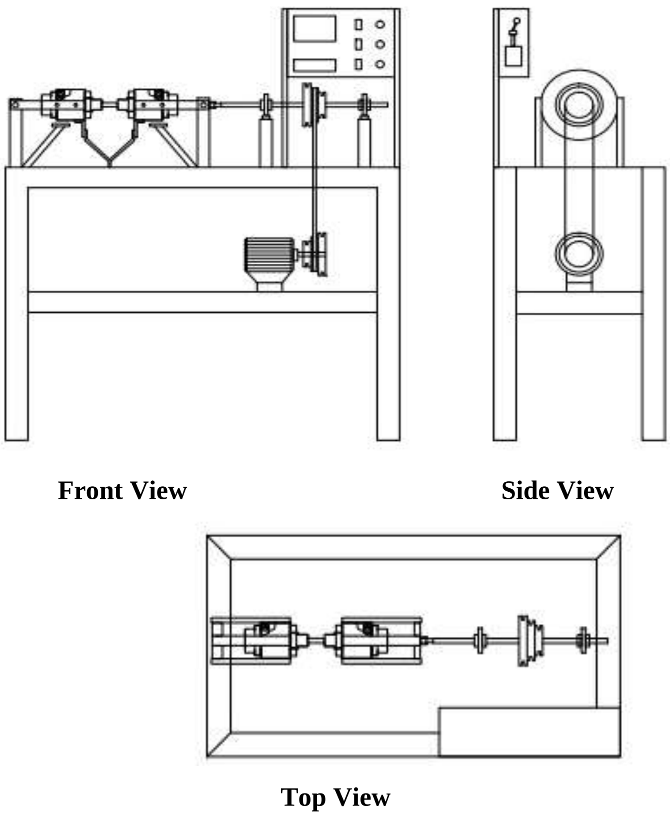


Fig 2 Machine Diagram

Machine parameters taken from Instron fatigue test set up, Model **RRM-A2** conidered as reference [9] for our design of machine elements.

4 DESIGN ANALSYS

Minimum to maximum bending is allowed 25 kg-cm to 230 kg – cm, Maximum applied load 20 kg , Rotational speed 500- 10000 rpm

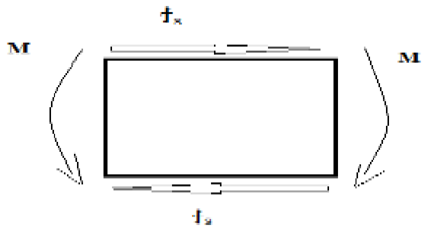


Fig 3 Elemental diagram of a material subjected Treating as Bi-Axial loading [11]

First divide this complex state of force or stress analysis into a elementary type

- Torque applied
- Bending moment due to vertical load

According to **ASTM E 466-82 [10]**

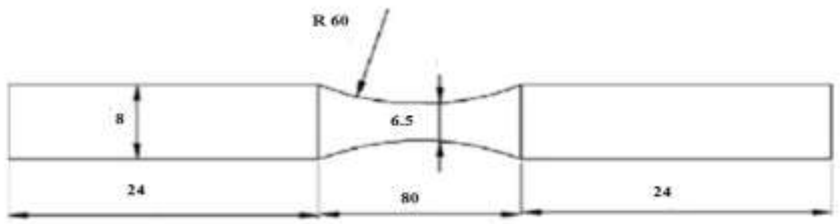


Fig 4 Specimen dimension

D_g =6.5 mm , **L_g** = minimum 3 times diameter= **20 mm** ,**R**= minimum 8 times diameter , 8×6.5 = **60 mm**

4.1 Design Calculation

Superimposing of applied moment and torque for design of machine elements

It is assumed that machine would at times be required to test some mild steel and medium strength steel at the stress level in the neighbour of the yield stress.

The shear stress theory is reasonably good agreement,it is often used by engineer to obtain quick estimates.

Mechanical properties of Steel ,Ultimate strength = **400 Mpa**, yield strength = **240 Mpa**, Г Shear Strength= **200Mpa**

The torque required to start yield at the outer fibers of a specimen of such steel is (T)

$T = Г \times \pi \times d^3/16$ (1)

= **10785.89 N-mm**

4.2 Design of torque transmitting shafts (d)

The torque transmitting shaft

$d^3 = 16 \times t/\pi \times Г$

= **11 mm**

Design of Power transmission shaft considering fatigue load.

Shaft material selected as C30 steel,diameter 10.80mm ,subjected to rotating bending fatigue loading,fatigue factor =1.612 [43].

Assuming

$K_{size} = 0.85$ $K_{surface} = 0.83$ $K_{reliabliity} = 0.896$

$S_e' = 0.5\ \sigma_{ultimate}$

= 245 mpa

$s_e = k_{size} \times k_{surface} \times k_{reliableity} \times 1/k_s \times s_e$ (2)

= 147.6 Mpa

This fatigue strength calculated is less than endurance strength of standard C 30 steel, shows that the design is safe.

Power transmission consists of the following arrangements

- Motor with Standard specification.
- Stepped pulley arrangement.
- Leather trapezoidal section V belt to connect motor driver shaft and driven shaft vertically for better stability.

Considering power transmission shaft as beam

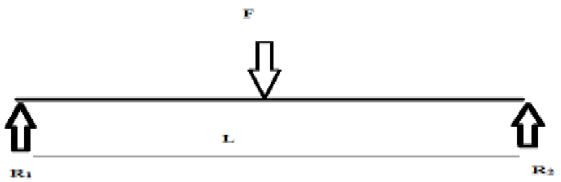


Fig 5 Simply supported beam of shaft

$Y_{max} = FL^3/48EI$ (3)
= **0.00249 mm**
E= Youngs Modulus N/mm^2 , L=40 mm assumed

$\Sigma_{bending} = M/z$ (4) M= Moment,z= Section Modulus mm^3
$\Sigma_{bending}$ = **79.66 N/mm^2**

4.3 Design of motor power required (P)

P = 2×π×n×t/60 watts (5) n=Revoultion
= 379.09 watts
= **0.5 hp**

4.4 Design of tension acting on the belt (P_1,P_2 in Newton N)

Selecting cast iron stepped cone pulley with a leather belt arrangement having a provision of 3 stepped diameters of larger being 125mm,smaller 75 mm and 100 mm respectively.

$(P_1-mv^2/P_2 - mv^2) =^{f\alpha/\sin(\theta/2)}$ (6)
$(P_1-mv^2/P_2 - mv^2)=e^{(o.35\times2.99)/(\sin(40/2))}$

For maximum belt tension condition
(P_1/P_2)=21.54 the belt tension is maximum
$P_1=P_2\times21.54$
Torque T = $F_t\times r$ (7)
10785.89 = $F_t\times$ diameter of pulley/2
$F_t=P_1$=215.27 N (F_t= Tangential Force N)
P_2 = 215.27 /21.54, **P_2 = 9.99 N**

4.5 Power transmission connection & Purpose of the joint

In order to transmit torque from power transmission system to the specimen through load transmitting member there should be an joint/connection required to connect these two assemblies. Various mechanical joints are availiable, as we have decided to select use universal joint which serve the purpose with better efficiencies which used in smaller automobiles.

To connect power transmission system and load transmitting member

It should transmitt power at different angles (sweiling motion is permitted to load transmitting member when specimen fails)

The design up to here ended up with the following dimensions

- Torque required to shear the specimen
- Diameter of shaft for mounting pulley assembly
- An approximate distance between the two bearing
- Safe bending on shaft
- Safe fatigue endurance
- Left section is consider as weakest compare to right

Design and selection of Load transmitting members with an simple arrangment to over come Machine scattering plays a vital role in this machine.

4.6 Load-producing Mechanism

The loads may be produced by various methods: mechanical, electromagnetic. Load produced by dead weight A machine of this type, however, developed by H. F. Moore and KOMMERS (1927). [1] A constant bending moment is maintained by directly applied static loads by using weight hanger [41]

4.7 Design of load transmitting member

Advantages of practical use for supported levers in testing machines, which is a simple and effective means of overcoming difficulties experienced with the usual type of bearing and knife edges. Since a cross- pivot has no sliding parts there is no need of lubrication. The deflection is, however, accompanied by reaction forces, though they are usually very small. This constructional element has been examined by Eastman (1935).It has been studied experimentally by Young (1944) and theoretically by Haringx (1949). [1]

This case can be divided into two sub points:

- vertical external load = **200 N**
- torque transmitted = **10785.89 N-mm**
-

External load of 200 N and Torque of 10785.89 N-mm will be shared by two mandrel at either side,with two bearings placed a distance apart and are fitted in hub.

F_2 & F_3 are the forces distrubuted which are acting on top of the bearings including self weights Bearings are placed at a distance Length of the mandrel is =80 mm

Considering Horizonatal lever as beam

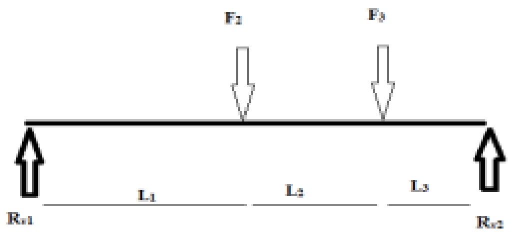


Fig 6 Simply supported beam

$\sum F_y = 0$,$R_{v1} + R_{v2} = 49.5$ N

$\sum m_1 = 0$,- $R_{v1} \times 135 + 24.5 \times 80 + 24.5 \times 130$

$R_{v1} = 11.38$ N, $R_{v2} = 38.11$ N

Static Deflection of shaft

Y_{max} =(-Fba/27EI) × (b+2a) ×(√3b(b+2a)),

At $L_1 > (L_2 + L_3)$ Lever breadth=25mm,t=15mm

Y_{max} **=0 .0000683mm is safe**

4.8 Radial Loads Analysys

Radial force on bearing is when specimen is about to fail after crack propogation.the magnitude of this load will be lesser compare to the external loading.

Treating load supporting member including specimen as an beam for our analysis, due to symmetry in the structure. considering horizontal external forces finding out its reaction

Radial thrust assumed as **50N**

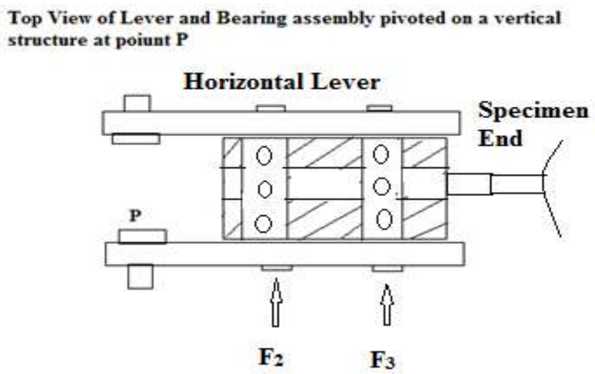


Fig 7 Consider an mandrel and load transmitting member assembly arranged as shown above

Considering Vertical Force of 50 N by rotating it 90^0 treating as radial horizontal force acting on simply supported beam as shown in fig 7

$\sum_H = 0$

$R_{H1} = 11.38$ N,$R_{H2} = 38.11$ N

Considering vertical and horizontal reactions

R_1 $= \sqrt{(Rv_1^2) + (R_{H1}^2)}$ **, = 16.57 N** (8)

R_2 $= \sqrt{(Rv_2^2) + (R_{H2}^2)}$, **= 53.89 N**

Considering this reaction are acting on bearing from either side of holding member

$F_{r1} = 76.57$ N, $F_{r2} = 110.52$ N

Expected bearing life

$L_{10} = 60nl_{10}h/10^6$ (9)

$= 60\times1440\times8000/10^6$

= 691.2 million revolution

Considering load factor

$C_1 = p_1 (l_{10})^{1/3}$ (load factor), (10)

= 1656.02 N Load Factor= 2.5

$C_2 = p_2 (l_{10})^{1/3}$ (load factor), **= 2390.287 N**

4.9 Design of bearing life

There are two bearing on each mandrel so the load carried by each bearing will be approximately 50 N

Select tapered roller bearing because , during crack propogation in there may be sweiling action of the load transmitting member ,and there may be load acting in lateral direction also.

Expected bearing life

$L_{10} = 60nl_{10}h/10^6$, = 864 million revolution ,n = 1400

$C = p (l_{10})^{1/3}$, = 465 N dynamic load

Dynamic load/ cycle = $2390.287/10^6$= **0.002390 N/Cycle**

Consider uniform form force to all member in each cycle may be assumed as 0.002390 N

4.10 Design of Mandrel assembly connection by welding and fasteners arrangement

By knowing the forces acting on the horizontal pivot which is required to hold mandrel assembly, it was planned to have an bolted connection. Initially it was planned to have 2 bolted connection at each side of lever.

Considering an horizontal pivot arrangement with an provision of two holes for bolted connection shown in fig 8

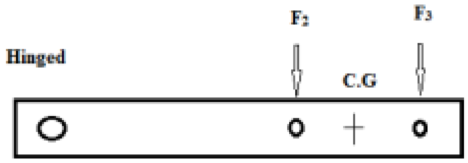


Fig 8 Horizontal lever with bolting provision

Maximum external loading is 200 N, Considering the primary and secondary shear force

Center of gravity lies in between two bolts

Permissible shear stress

$\tau = 0.5\, S_{yt}/F.S$ (11)

= 49 N/mm²

Primary shear force

$P_1^{-}=P_2^{-}=P/2=61.31/2$ (12)

= 30.6 N

Secondary shear force

$P_1^{--}=P_2^{--}= ((P\times e)r_1/(r_1^2+r_2^2))$ (13)

= 55.7N

Resultant shear force

$R.S.F = P_2^{-}+ P_2^{--}$ (14)

= 30.6+55.7

= 86.3 N

Core diameter of bolt is given by

τ = r.s.f/(area of the bolt) (15)

$49 = (86.3/(\pi/4\times d^2))$

d = 2.67 mm diameter of bolt

4.11 Design of welded joint required to connect power transmission shaft and load transmitting member through universal joint.

$\tau = m/2\pi tr^2$ (16)

$140 = 2513.6/2\times\pi\times t\times12.5^2$

t= 0.01828 mm

H = 0.025/0.707

A High strength connection is enough to transmit the torque.

4.12 Design of vertical structure.

There should be a vertical structure in order to with stand the reactions as calculated , on top provided an pin joint connection to the load transmitting member through lever.

4.13 Design of base structure

The machine base structure decides the volume occupied by the machine. Material selected for construction of base is cast iron L shaped section [18] as commonly used . Placing horizontally design of load transmitting member, universal joint and power transmitting system assembly makes an appropriate length of 900 mm, and height of 300 mm, and width of 450mm,where here it is decided to check for safer operation of machine with lesser vibration and high damping.

4.14 Unexpected stresses due to self weight

It is easy to show that in order to limit the error to not more than I per cent of true stress applied to a rectangular specimen of cross-section b x h, the line of load must not deviate from the geometrical axis of the specimen by Nsore than **0-0.002 % h or d (Axial loading)** . This problem has been discussed by Morrison (1940) [1] .

Considering minimum effective weight of the load transmitting member acting on specimen is **7 kg** shown in fig 9

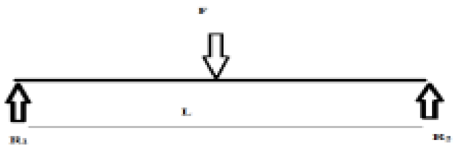


Fig 9 Simply supported beam

$\mathbf{Y_{max}} = Fl^3/48EI$ **L=20 mm** assumed

$= 70\times40^3/48\times200\times10^3\times((\pi\times6.65^4)/64))$

$= \mathbf{0.00437\ mm.}$

4.15 Critical speed of the machine

Although there are many possible causes of vibration in rotating equipment, this technique will deal only with that component of vibration, which occurs at running speed (frequency), and is caused by a mass unbalance in the load transmitting members. The critical speed of the fatigue testing machine design is given by. [12,16]

$\Omega = (\pi/L)^2 \quad (EIg/Ay)$ (17)

Ω = Critical Speed Rad/sec

L=Length of the rotating member= 110mm

$\mathbf{I} = \pi/64\ d^4$ (18)

$= \pi/64 * 7^4$

$= \mathbf{117.874mm^2}$

g = accleration $9.81 m/s^2$

a = area of the shaft

$a = (\pi/4 * d^2)\ mm^2$ (19)

$= (\pi/4 * 0.007^2)$

$= 3*10^{-5}\ m^2$

$\gamma = \rho L \quad \rho$=Density kg/m^3 (20)

$= 7850 * 0.1$

$= 785\ kg/m^2$

If $\varphi = a * \gamma$ (21)

$= 0.00003 * 785$

$= 0.02355$ kg

$\mathbf{\Omega} = (\pi/L)^2 \quad (EIg/\varphi)$ (22)

$= (\pi/100)^2 \quad (200*10^3 * 117.8*9.81/0.0235)$

$= \mathbf{86.89\ Rad/sec}$

First before continuing the vibration.

With L = 110m, F = 293.44N, I = $19174.76mm^4$

= 200 x 103

Static deflection of the shafting is given by

$Y_{max} = FL^3/48EI$

$= (200 * 100^3) / (48*200*10^3*117.8)$

$= \mathbf{0.176mm}$

Frequency of the transverse vibration

$\mathbf{F_n} = 0.4985/\sqrt{Ymax}$ (23)

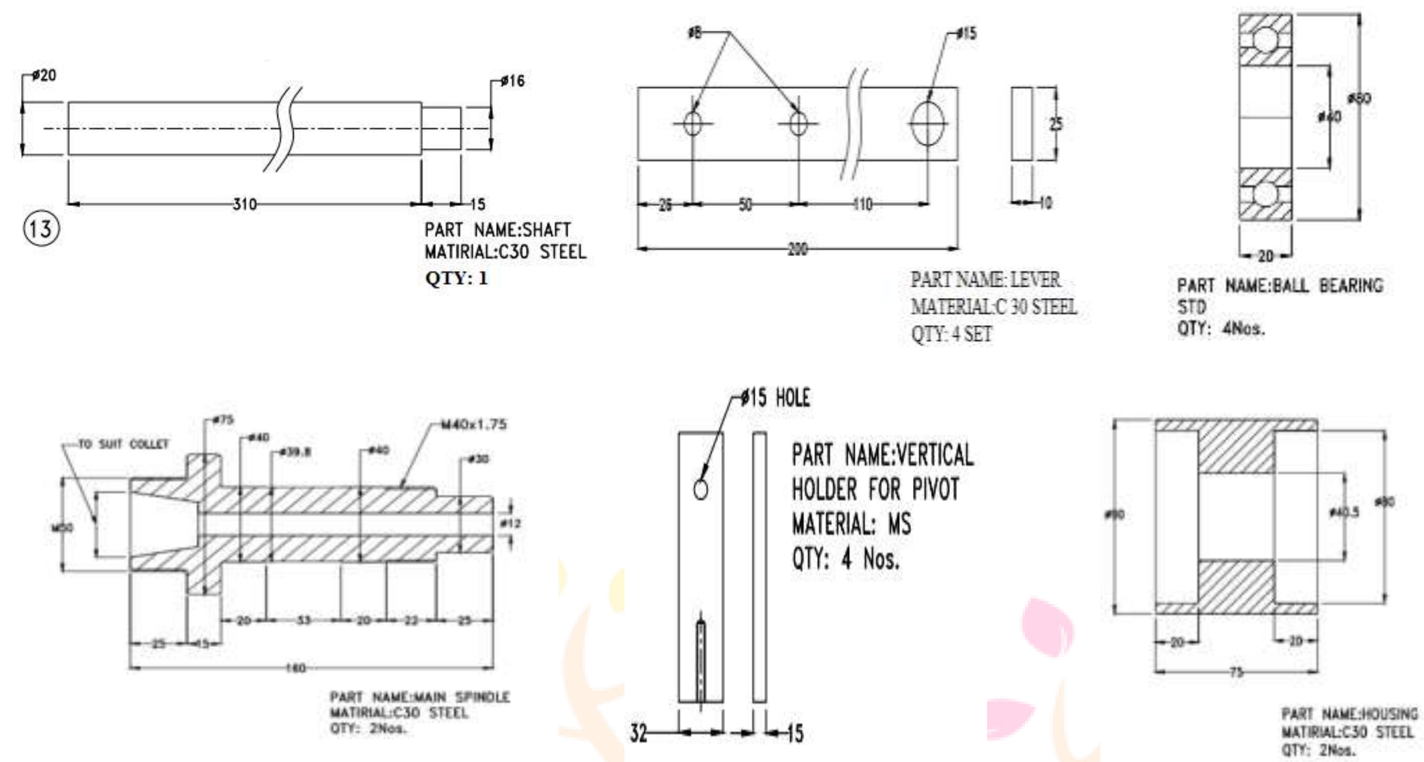
$= \mathbf{1.20\ Hz}$

This is less than the frequency of the rotating speed of the motor. Therefore, the machine is free from much vibration.

4.16 Static Calibration and Dynamic Checking of Testing Machines

The load should be reproduced with sufficient accuracy, and it should be transmitted to the test piece without undue scatter. In machines where the load is generated by dead weights and transmitted to the specimen through a lever system, weights and forces have to be carefully controlled including weighing of all levers and other parts of the loading system, together with an experimental determination of the centers of gravity of these parts.[1,39,40,41]

4.17 2 D Diagram of important machine element



5 FABRICATED MACHINE

A- Horizontal Lever B- Bearing Housing C- Specimen D- Vertical Rest E- weight hanger F- Universal Joint G- Stepped Pulley H- Power Transmission Shaft I- Base M- Phase Motor ,P-Pivot

6 Experimental Results

Material Ductile Iron Austempered at 300^0C

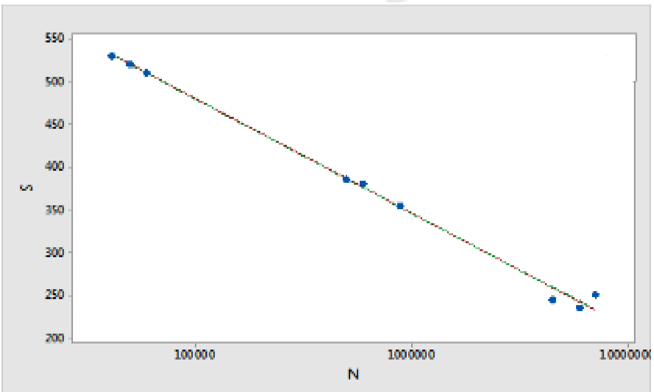


Fig 10 S-N Curve

7 CONCLUSIONS

The design approach was simple, identifying the critical section of the machine, superimposing of forces and moments, followed design based on biaxial loading consideration. Choice of design and material selection was done on local avaliability, so any replacement of parts or machining operations can be done within the workshop itself. Small amount of vibrations was nullified due to the heavy structure and by using rubber pads on four corner. A provision provided at the pivot assembly is that by knowing the lateral difference in distance between the two horizontal pivots, manual adjustment is made by increasing or decrease in washer thickness with machining accuracy so some machine scatter (lateral misalignment / Co- Axiality) can be adjusted which showed a better performance in run, due to this rise in temperature effects can be almost nullified. Gripping methods used in this machine showed that loosening of grips during dynamic conditions can be identified by the movement of unscrewing of locking nut from its locked position which was known before. Due to its symmetry of a load supporting member the choice of design is made such that, use of bolted connection, pin joint and locking its movement by an washer arrangement assembly, will help an operator to easily dismantle the assembly without damage and will help him in calibration of machines whenever necessary. Initial tightening of fastener arrangement required for better and stable operation. Bearing should be lubricated time to time. Simpler Design helped in understanding dynamic scatter in these type of machines. Any misalignments produce in the system will overcome by operator easily. Machine generates acceptable results.

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