Design and Characterization of a Fatigue Testing Machine

Gbasouzor Austine Ikechukwu, *Member, IAENG*, Okeke Ogochukwu Clementina and Chima Lazarus Onyebuchi

Abstract – **Many engineering machines and mechanical components are subjected to fluctuating stresses, taking place at relatively high frequencies and under these conditions failure is found to occur. This is "fatigue failure". And this led to the invention of a fatigue testing machine. In view of effective design that will not fail accidentally, this research is conceived. This testing machine will determine the strength of materials under the action of fatigue load. Specimens are subjected to repeated varying forces or fluctuating loading of specific magnitude while the cycles or stress reversals are counted to destruction. The first test is made at a stretch that is somewhat under the ultimate strength of the material. The second test is made at a stress that is less that than that used in the first. The process is continued, and results are plotted.**

Keywords: **Fluctuating stresses, fatigue failure, high frequencies, strength of materials, fatigue load.**

I. INTRODUCTION

A fatigue is a failure of material or machine due to the action of repeated or fluctuating stress on a machine member for some number of times.

This failure begins with a small crack. The initial crack is so minute that it cannot be detected by the naked eye and is even quite difficult to locate in a magniflux or x-ray inspection. The crack will develop at a point of discontinuity in the material such as a change in cross section, a keyway, a hole or a notch. Less obvious points at which fatigue failure are likely to begin are inspection or stamp marks, internal cracks or even irregularities caused by machining. Once a crack is initiated the stress concentration effect becomes greater and the crack progresses more rapidly. As the stressed area decreases in size, the stress increases in magnitude until finally, the remaining area fails suddenly. A fatigue failure therefore is characterized by two distinct regions. The first of those is due to the progressive development of crack, while the second is due to sudden fracture.

Unlike other failures, fatigue failure gives no visible warning in advance. It is sudden and totally dangerous. This sudden failure, which is dangerous, and can lead to not just minor accident but fatal accident and loss of lives triggered the quest to invent a machine that can test and give or predict the effect of fatigue on various metals such as aluminium, cast-iron, mild steel, etc.

Gbasouzor Austin Ikechukwu, MIAENG is a Lecturer in the Department of Mechanical Engineering, Anambra State University, P. M. B. 02 Uli, Nigeria. e-mail: unconditionaldivineventure@yahoo.com
Phone: 2348034247458, 2348063896067, 2348028283035

Okeke Ogochukwu Clementina is a lecturer in the Department of Computer Science, Anambra State University, P. M. B. 02 Uli, Nigeria

Chima Lazarus Onyebuchi is a Researcher in the Department of Mechanical Engineering, Nnamdi Azikiwe University, P. M. B. 5025 Awka, Nigeria

In most testing of those properties of materials that relate to the stress-strain diagram, the load is applied gradually to give sufficient time for the strain to fully develop. Furthermore, the specimen is tested to destruction, and so the stresses are applied only once. Testing of this kind is applicable, then, to what are known as static conditions. Such conditions closely approximate the actual conditions to which many structural and machine members are subjected.

The condition frequently arises, however, in which the stresses vary or they fluctuate between levels. For example, a particular fibre or aluminium on the surface of the rotating shaft subjected to the action of bending loads undergoes both tension and compression for each revolution of the shaft. Since the shaft is part of an electric motor rotating at 1400rev/min, the aluminium metal is stressed in tension and compression 1400 times each minute. If in addition, the shaft is also axially loaded (as it would be, for instance, by a helical or worm gear), an axial component of stress is superposed upon the bending component.

In this case, some stress is always present in any one metal, but now the level of stress is fluctuating. These and other kinds of loading occurring in machine members produce stresses that are called variable, repeated, alternating or fluctuating stresses.

Often, machine members are found to have failed under the action of repeated or fluctuating stresses; yet the most careful analysis reveals that the actual maximum stresses were below the ultimately strength of the material, and quite frequently even below the yield strength. The most distinguishing characteristic of these failure is that the stresses have been repeated a very large number of times. Hence, the failure is called "Fatigue Failure".

When machine parts fail statically, they usually develop a very large deflection, because the stress has exceeded the yield strength and part is replaced before fracture actually occurs. Thus, many static failures give visible warning. It is sudden and total, and dangerous. It is relatively simple to design against a static failure because our knowledge is comprehensive. Fatigue is a much more complicated phenomenon, only partially understood, and the engineer seeking competence must acquire as much knowledge of the subject as possible. Anyone who lacks the knowledge of fatigue can double or triple designs factors and formulate a design that will not fail.

Design Objectives

This aims at designing and constructing a fatigue-testing machine that is capable of testing the fatigue life of various samples of specimen from metals, such as mild steel, aluminum, brass, etc.

Also with the result of each test carried out with this machine, the fatigue life of various materials can be obtained and fatigue failure be guarded against in an optimum manner.

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II. METHODOLOGY

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. The fatigue testing machine consists of the following components:

A. AHP electric motor, B. Bearing and its housing assembly
C. Weight hanger assembly D. Dead weight
E. Bearing spindling F. Digital counter
G. Magnetic cyclic pick up (dynamo)
H. Variable speed control I. Switch, J. Specimen
K. Drill chunks, L. The metal desk

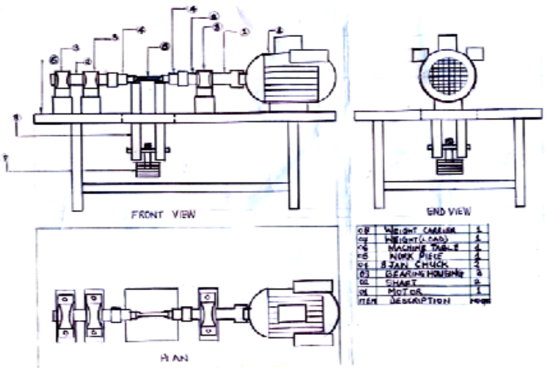

Fig.1: The isometric views of the fatigue-testing machine

Bearing and Its Housing Assembly

The particular bearing used in this project is the single, row, deep; groove bearing that can take both radial load and some thrust load. The bearing is shouldered in a housing made of cast iron to secure adequate support for the bearing and resist the maximum thrust load.

Weight Hanger Assembly

This consists of small bar made of cast iron, with a spring. The head of this bar is designed to be hanged on the loading hardness assembly. The spring is designed to absorb shock preventing any slight vibration of the housings on the dead weight. The choice of cast iron for the material for hanger is due to the fact that cast iron has the modulus of elasticity E as 60 to 90 mPa and can withstand the maximum weight of which the machine can carry

Dead Weight

These are loads of different sizes and weight

Bearing Spindle (Shaft)

The bearing spindle (shaft) are machined from a bar of high grade steel (310 stainless steel) and is thicker than specimen by (5/8 inches) to prevent the machine parts from experiencing fatigue due to the loading of the specimen. The stainless steel can withstand adverse temperature and corrosive environment, due to its high carbon content it does not fail-easily.

Two shafts were employed in this work. One is the driving shaft, which is connected to the motor shaft extension by means of bolting. The other end of this shaft is machined to fit into a drill chuck. The other spindle, the driven shaft is machined to fit into a drill chuck at one end while the other end carries a gear that transmits motion to the dynamo. Both shafts are allowed to pass through the bearing housing which help to check any misalignment and give the shaft rotational support.

Digital Counter

The digital counter is a terminating device that gives out the equivalent value of number of revolution of motor in term of cycles. The digital counter registers 1 for each 140 revolution of the motor, so to obtain the number of cycles of stress, counter readings must be multiplied by 10,000. A total of 1,400,000 revolutions are turned up for each cycle of number. At 1,400 rpm, a complete cycle of numbers takes place in about 166 hours, or 7 days if operating 24 hours per day. In a very long test therefore the number of cycle must be recorded. The speed reduction unit is lubricated from time to time. This easy-to-read digital cycle counter is connected to a magnetic pick up device (dynamo). Push button control is provided to reset the display count at the start of a test.

The Cut-Off Switch

The cut-off switch is employed to set upon the circuit to the motor when the bearing experiences a 3° angle.

Specimens

The standard specimen, a tapered-end specimen has the length of 87.3mm. They are machined to match the tapers within the spindles of the machine and held in place using drill-chuck. Stress as applied to the specimen by direct application of dead weight to ensure precise loading. Maximum fibre stress in a specimen having a 0.300inches (7.62mm) diameter is. By decreasing the diameter, the value of the maximum fibre stress can be increased. An easy-to-use reference table within the operator's manual makes determination of the load weight needed to produce a particular stress a single circulation.

Drill Chucks

The metal desk on which the system is assembled on is made with 3mm angle iron of mild steel and plain pan made of mild steel for the top. The length of the desk is 760mm; the width and the height are respectively 210mm and 190mm. The desk is drilled at each point where parts are to fit with bolt and nut. A rectangular opening is found through which the hanger is passed; this is located at the top of the desk. The length and width of this opening are respectively 14.7mm and 11.8mm. The total weight of the desk is 75kg. A rubber tyre is fixed under each leg of the desk to damp vibration on the desk and moreso to easy locomotion.

Setting Up The Machine

The base should first of all be set leveled so that the weight will hang perpendicularly to the axis of the specimen. Wires in conduct from the power supply are run to the connecting block in the base of the machine and soldered to the lugs provided. The machine is equipped to operate from an AC power source at 120volts, 60 cycles, and single phase. Motor and relay equipment to operate from a power source of different rating can be provided on special order. With mild steel legs, the machine is set on five rubber tyres. The weight hanger is of such length that it will clear the desk since it is mounted on the legs. A shock absorber in the hanger prevents the slight vibration of the housing from

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being impacted to the weight. The base is provided with holes so it can be bolted down if desired.

Assemble the housing with a sample of specimen provided in accordance with the direction given below, and start the machine to see if there is any misalignment. This will be indicated by noise and vibration. Examine the housing wind for oil. It should stand about midway of the windows when the machine is idle.

Adjust the speed to 1,400 RPM by means of a rheostat in the base of the machine. The machine should now be ready for a test.

Speed

The normal speed is 1,400RPM. The speed depends on the voltage. If the voltage of the supply line exceeds the proper amount of sliding rheostat in the base of the machine should be adjusted so that the speed cannot exceed 1,400RPM under ordinary conditions. Speeds in excess of the normal will cause no damage unless continued to long. In adjusting the speed take care that the contraction of the rheostat makes firm contact with the exposed potion of the winding.

If it desired to run at speed below 1,400 rpm, or gradually increase the speed from zero, a variable resistor can be placed in the motor lead wires by utilizing the plug type connector. This will be necessary only in exceptional cases where the slight starting torque exerted on the specimen may be objectionable. The inertia of the motor armature is such in comparison with that of the remote half of the spindle that there is sufficient delay in starting for most purposes

III. ANALYSIS AND DISCUSSION OF RESULT

Specimen Dimension and Machine Capacity

For the solid specimen, the diameter of the test specimen is 7.62mm.It is assumed that the machine would at times be required to test some high strength steel at stress levels in the neighborhood of the yield stress. Using stainless steel of 310 i.e. 45c steel at temperature of about 315°C, ultimate tensile yields stress = 700mpa
Assuming Tresca's Yield Criterion, the shear yield stress is given as 350mpa.
The torque required to start yield at the outer fibres of a specimen of such steel is

$$T = \tau_y \times \pi d^3$$

$$T = 350\text{x}\ \pi\ \frac{(7.62)^3}{16} = 30{,}406.25\text{Nmm}$$

$\therefore$In this design, a machine capacity of 30,400Nmm is assumed.

The torque-transmitting shaft

The torque-transmitting shaft, which comprises of the longer shaft, bolts, the chuck and the short shaft passing through bearing housing sustains the same fatigue load during a testing program as the specimen undergoing the tests.
Where T = torque = 30406.25Nmm
i.e. d = [16 x 30406.25/π x 87.5] = 12.09mm

In the design, the minimum shaft diameter used is 25mm. The reason is to make the shaft more rigid and to make it withstand heat and shock for many number of tests without failing easily due to fatigue. However fig. 3 shows that in some location, circumferential grooves are made for retaining chucks and rings, thereby reducing the effective diameter to approximately 20mm. As stated before, the diameters of the gauge length of the shaft depends on the load range and are thereby designed accordingly.

Check for stress concentration effect.

At the position for the chuck, there is a possible or product stress raiser. It is the shaft shoulder.

Shaft Shoulder

D = 25mm
d =16.2m
r = maximum fillet radius = 5m
$\therefore$ D/d = 25/16.2 = 1.54mm
r/d = 5.0/16.2 = 0.31m

The stress concentration corresponding to the ratio above is 1.17
i.e. kt = 1.17

But the maximum nominal stress in sheer = $16T/\pi d^3$
= 16 x 30406.25/Ti x (16.2)3 =36.42N/m^2
It is designed that no part of this shafting ever fails in service. With the little knowledge of torsion fatigue, we apply therefore factor of ignorance to some considerations.

A stainless steel of 310 or 45c8 steel is recommended for the shafting because of its strength and ability to prevent the machine parts from experiencing fatigue due to the loading of the specimen all through the test. This reason of choosing this material can be traced to the advantage of being highly resistance to fatigue owing to its low notch sensitivity, which is itself, a result of its high ductility.
45c8 steel or 310 stainless steel has these parameters;
Yield strength T_y =350mpa

Ultimate strength S_{ut} = 700mpa Following a generally accepted thumb rule for S_{ut} < 19600 Endurance limit Se' = $0.5S_{ut}$

Endurance limit Se' = 0.5 x 700 = 350mpa

Using Tresca's Yield Criterion and assuming the validity of its application of fatigue loading, the endurance limit in shear is obtained as $T_e = S_e/2 = 175$mpa

At this point a factor of approximately 2 is applied and

$T_e = T_e/2$ = 175/2 = 87.5mpa

The minimum shaft diameter for the full machine load is given by the expression $d = \{16T/\pi T_d\}^{1/3}$

Applying the stress concentration factor,

T_{mass} = 1.17 x 36.42 = 42.616
This value is very much less than the assumed endurance limit of 350N/mm^2 and also than the design shear stress of 87.5N/mm. It is therefore considered safe.

Bolt Design

Two bolts are used to hold the shaft to the motor shaft. The bolt diameter is 4.75mm. They are separated 20mm apart. For a torque of 30406.25Nmm
Shear force on each bolt = ½ x 30406.25/10 = 1520.31N
Average shear stress = 1520.317 [π/4(4.75)^2] = 85.79N/mm^2
To bear this load safely, SAE grade 2 bolts are recommended and shear stress of bolt is less than the design shear stress, which shows that it is safe.

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Chucks

Each chuck consists of two major parts, the hub made of mild steel and a circular plate of hardened medium carbon low alloy steel machined to take the end of the specimen. The two parts are fitted together with two aligning pins and four 5mm diameter SAE grade 2 screws. The hub is shrink-fitted onto the shaft whose diameter is stepped up to 40mm through a gene ales radius as shown in order to improve the fatigue strength of the joint.

Analysis of shrink fit.

When cylinder is assembled by shrinkage as shown above, a contact pressure is developed at the interface. The pressure is a function of the digital interference and for materials of the same modulus E, is given by:

$P = E\delta/b\ [(a^2-b^2)/2b^2-(b^2-a^2)/c^2-a^2]$

For the case of shrinkage a hub onto a solid shaft, (a = 0) the expression reduces to:

$P = E\delta/2b\ [1-b^2/c^2]$

In the design b = 10mm, c = 16.2

$P = 200 \times 10^3\delta /2 \times 10\ [1-10/16.2]$

$= 10{,}000[1-0.617] \qquad = 3830\delta N/mm^2$

An interface of approximately 0.00762 or 7.62×10^{-3}mm is recommended. This will result in an interface pressure of

$P = 3830 \times 7.62 \times 10^{-3}mm \quad = 29.19N/mm.$

This value of stress is safely below the endurance limit and the design stress,

Total area on which interface pressure acts $= \pi \times 40mm$

$= 125.66mm^2$

$\therefore$ Total effective normal force = 125.66 x 29.19 = 3668.02N

Assuming a low value of the coefficient of static friction say μ = 0.08, Limiting tangential force = 0.08 x 3668.02 = 293.44N.

Where 1 and d are in millimeters. The angular twists of the various segments of the shafting are computed, using the approximate effective length. The results are expressed in the table below.

Shaft diameter d (mm)	Length, l (approx) mm	Twist angle θ (rads)
25 longer shaft	150	1.4861×10^{-3}
25 short shaft	70	6.93504×10^{-4}
16.2(gauge length for long shaft)	40	2.248×10^{-3}
16.2(gauge length for short shaft)	40	2.248×10^{-3}
7.62 specimen	87	9.986×10^{-2}

From the table, angular twist is obtained by summing the angular twist of the separate segments.

$\sum\theta$ approximate = 0.106 = 0.11 rads

Determination of Bending Stress

2nd moment of inertia $I = \pi d^4/64 = \pi/64 \times (25)^4 = 19174.76mm^4$

Using the relation $\delta = (M/I)\ y$

Where $y = \frac{d}{2}$ = 12.5mm

M = 30406.25Nmm

The maximum bending stress is safe for the machine.

Deflection under Peak Load

The deflection is given by

$\delta_b = M L^2 /3EI$

Where L is the effective length = 250mm

$\delta_b = \dfrac{30406.25 \times (250)^2}{3 \times 200 \times 10^3 \times 19174.76} = 0.165mm$

The corresponding angular displacement is given by

$0.165/250 \qquad = 6.607 \times 10^{-4}$ rads

This displacement is very negligible compared with the relative shaft under maximum torque 0.1 1rads.

Short shaft or driven shaft

Determination of bending stress

2nd moment of inertia $I = \pi d^4/64 = \pi/64 \times (25)^4 = 19174.76mm^4$

Using the relation $\delta_b = (M/I)\ y$

Where $y = \frac{d}{2}$ = 12.5mm

M = 30406.25Nmm

The maximum bending stress is obtained as

$\delta = \dfrac{30406.25 \times 12.5}{19174.76} = 19.82N/mm^2$

The maximum bending stress is also safe for the machine.

Deflection of driven shaft under Peak Load

The deflection is given by

$\delta_b = M L^2 /3EI$

Where L is the effective length = 150mm

$\delta_b = \dfrac{30406.25 \times (150)^2}{3 \times 200 \times 10^3 \times 19174.76} = 0.0575mm \approx 0.06mm$

Therefore angular displacement = 0.06/180 $= 3.96 \times 10^{-4}$rads

This displacement is also negligible compared with the relative displacement of the ends of the torque-transmitting shaft under peak load. The analysis shows that the setting of the double eccentric shaft is accurately reflected by the angular twist of the shafting with negligible error; introduced by the elastic bending of both the torque arm and the crank. Where 1 and d are in millimeters. The angular twists of the various segments of the shafting are computed, using the approximate effective length. The results are expressed in the table below.

Shaft diameter d (mm)	Length, l (approx) mm	Twist angle θ (rads)
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25 short shaft	70	6.93504×10^{-4}
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Using the relation $\delta = (M/I)\ y$

Where $y = \frac{d}{2}$ = 12.5mm

M = 30406.25Nmm

The maximum bending stress is safe for the machine.

Deflection under Peak Load

The deflection is given by

$\delta_b = M L^2 /3EI$

Where L is the effective length = 250mm

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$$\delta_b = \frac{30406.25 \times (250)^2}{3 \times 200 \times 103 \times 19174.76} = \frac{1900390625}{1.1504856 \times 10^{10}} = 0.165\text{mm}$$

The corresponding angular displacement is given by

$0.165/250 = 6.607 \times 10^{-4}$ rads

Maximum torque transmitting capacity

$= 1.0 \times 293.44 = 2934.4$N/mm

This value is far more than the machine shall ever encounter in service.

Screws

Four 5mm diameter screw in 48.26mm PCD.

Shear load on each screw $= \frac{1 \times 30406.25}{4 \times 48.26} = 157.51$N

Average shear stress $= \frac{157.51}{\frac{\pi}{4} \times 5^2} = 8.02\text{N/mm}^2$

Estimation of the angular twist of the entire torque-transmitting shaft under full capacity load

This exercise is necessary in order to be certain that the maximum angular displacement obtainable from the double eccentric shafts is adequate for providing the twist corresponding to maximum load. The analysis is made assuming. The use of the use of the largest shaft diameter of (25mmϕ) and that the, specimen steel remains elastic unto the full load, The general idea is to ensure that adequate allowance is made for the case of plastic deformation where strains are larger.

The angular twist on a length L of a shaft of diameter d subjected to a torque T is given by the expression:

$$\theta = \frac{T}{G} \times L \times \frac{32}{\pi d^4} \theta$$

Where G is the modulus of rigidity = 80 x 10^3N/mm

Substituting for T and G, and the expression reduces to

$$\theta = \frac{30406.25}{80x10^3} \times L \times \frac{32}{\pi d^4} = 3.87 \frac{L}{d^4} \text{ rads}$$

Bearing

A total of three bearing are incorporated into the design. They are of the light series. They are chosen for rigidity and to reduce deflection. The details are given below:

3 bearings, 25mm bore, single row, deep groove ball bearing, self-aligning for the bearing housing for the three bearings.

The bearing in housing No 1 sustains a maximum load of

$$\frac{M}{L} = \frac{30406.25}{250} = 121.625\text{N}.$$

Hence average shear stress $= \frac{\frac{1}{2}(121.625) \times 4}{\pi \times (25)^2} = 0.12\text{N/mm}^2$

The bearing in housing No. 2 and 3 sustains a load of

$$\frac{M}{L} = \frac{30406.25}{150} = 202.71\text{N}$$

Also average shear stress $= \frac{\frac{1}{2}(202.75) \times 4}{\pi \times (25)^2} = 0.21\text{N/mm}^2$

Bearing Housing

The three main bearing housing Nos 1 - 3 are robust construction and are perfectly made of cast steel together with the weds as flanges. The casting are then reborned and forced rigidly onto the horizontal metal plate after which the final counter boring and reaming operators are performed with a high degree of accuracy. This ensures the maintenance of the axial symmetry of the bearing that was mounted in them.

Retaining rings are chosen according to availability and size, no emphasis is being placed on axial load capacity since in the design, and there are no axial loads.

Analysis of dynamic forces

Dynamic force of vary large magnitude are called into play by the rotation and acceleration of the unbalanced masses of the shafts. The effect here is to develop approximate analytical solution from which maximum dynamic forces are then computed.

Design of weight hanger

The weight hanger is designed to carry dead weight that will induce stress on the specimen. The weight hanger is designed to gain support from the specimen. This implies that the weight hanger is on the specimen. The weight hanger has these components:

Brass buchion: That has a direct contact with the specimen. It is simply on it that the hanger is simply supported from. It is two in number and separated apart. It is of brass buchion to create room for rotation without loosing its parts.

Arms: The hanger has arms that are 200mm long. This arm is extended to a flat pan, which is the base for the weight hanger assembly. The arm is bolted firmly to the flat pan.

The spring and leg of hanger: This is the load carried. It has a spring to absorb shocks or impact force from getting to the specimen. Even when the machine is switched on there are some vibrations that are generated due to critical speed of these elements do not have an effect on the weight applied.

Analysis of the Design Calculations of the Weight Hanger

The weight hanger is designed under some assumed specifications. The analysis begins from determining the critical speed.

Critical speed of the machine

Critical speed is the speed the shaft attains and becomes unstable with deflection increasing without upper bonds. When as shaft is turning eccentricity causes a centrifugal force deflection which can be resisted to some extent by the shaft flexural rigidity EL As long as the deflection are small, no harm is done in the cause of the design, we are determining the critical speed based on the shaft and ensembled attachments mass. The critical speed of the fatigue testing machine design is given by:

$$\omega_c = (\pi / L)^2 \sqrt{\frac{EI \times g}{Ay}}$$

L= 110mm = 0.11m

E = 200KN/mm^2 = 200 x 10^3N/mm^2

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$I = \frac{\pi}{64} \times d^4 \quad = \frac{\pi}{64} \times 25^4 = 19174.76\text{mm}^4$

$g = 9.8\text{m/s}^2$

a = area of shaft $= \frac{\pi}{4} \times \delta^2 \quad = \frac{\pi}{4} \times 25^2 = 490.9\text{mm}^2$

γ = p x L
= 7850 x 0.11 = 863.5kg/m^2

If $\phi = A\gamma$
γ = 863.5kgm3
A = 0.491m2
ϕ = 423.87kg

$$\omega_c = (\pi / L)^2 \sqrt{\frac{EI \times g}{\phi}}$$

$$\omega_c = (\pi / 110)^2 \sqrt{\frac{200 \times 10^3 \times 9.8 \times 19174.76}{423.87}}$$

= 8.1567 x 297767.067 = 242.88 rad/s = 14572.8rad/mm

This is greater than the natural speed of speed of the motor. This now approves the critical speed to be acceptable, to reduce its effect on the machine, the deflection is calculated first before continuing the vibration.

With L = 110m, F = 293.44N, I = 19174.76mm^4
= 200 x 10^3

Static deflection of the shafting is gotten by

$$\delta = \frac{FL^2}{3EI} \quad = \frac{293.44 \times (110)^2}{3 \times 200 \times 10^3 \times 19174.76} = 0.034\text{mm}$$

Frequency of the transverse vibration

$$F_n = \frac{0.4985}{\sqrt{\delta}} \quad = \frac{0.4985}{\sqrt{0.034}} \quad = 2.71\text{Hz}$$

This is less than the frequency of the rotating speed of the motor.

Therefore, the machine is free from much vibration.

Determination of spring for damping Vibration

Taking Minimum weight to be carried to 100N
Maximum weight to be carried to 500N
Spring index C = 6, Factor of safety, Fs =1.25,
Yield strength τ_y = 700mPa, Endurance limit, τ_e = 350mpa, Modulus of rigidity G = 80KN /mm^2 , Maximum deflection δ_f = 30mm

Determination of size of spring

Let d = diameter of wire
D = mean diameter of spring D = C.d
We have that
Mean load W_m = $W_{max} + W_{min}$ / 2
=500+100/2 = 300N
Variable load Wr = max - min / 2
= 500-100/2 = 200N

Shear stress factor k_s= I + ½ c
= I + '/2 x 6 = 1.083

Wahls stress factor, $k = \frac{4c-1}{4c-4} + \frac{0.615}{6}$

$$= \frac{4 \times 6 - 1}{4 \times 6 - 4} + \frac{0.615}{6}$$

$$= \left(24 - \tfrac{1}{24} - 4\right) + \left(0.615/5\right)$$

$$= 1.252\text{s}$$

We know that menu shear stress τ_m is given by the equation
τ_m = k x 8W_m x D / πd^3
=1.083 x 8 x 300 x 6d/πd^3 = (4964.1/d^2)N/mm^2
Variable shear stress is gotten from the equation
τ_v = k x 8W_V x D / πd^3
= 1.2525 x 8 x 200 x 6d/πd^3 = (3827.36 / d^2) N/mm^2

Also we know that factor of safety (F.s) is equal to:
1 /F.s (τ_m - τ_v / τ_y) / (τ_y 2τ_v / τ_c)

$$\frac{1}{1.25} = \frac{\frac{4964.1}{d^2} - \frac{3827.36}{d^2}}{700} + 2 \times \frac{\frac{3827.36}{d^2}}{350}$$

$$\frac{1}{1.25} = \frac{\frac{1136.74}{d^2}}{700} + \frac{\frac{7654.72}{d^2}}{350}$$

$$\frac{1}{1.25} = \frac{1.6239}{d^2} + \frac{21.8706}{d^2}$$

$$\frac{1}{1.25} = \frac{23.4945}{d^2}$$

d^2 = 23.4945 x 1.25
d = 5.42mm

Diameter of Spring

a) Mean diameter D = c.d
= 6.d = 6 x 5.42 = 32.52mm
b) Outer diameter of the spring D_o = D + d
D_o = 32.52 – 5.42 = 37.94mm

c) Inner diameter of the spring D_i = D + d
D_i = 32.52 – 5.42 = 27.1mm

Number of turns of spring

Let n = number of turns of spring and deflation of spring

$$\delta = \frac{8.W.D^3.n}{G.d^4}$$

$$30 = \frac{8 \times 500 \times (32.52)^3.n}{8 \times 10^3 \times (5.42)^4}$$

$$n = \frac{30 \times 80 \times 103 \times (5.42)^4.}{8 \times 500 \times (32.52)^3} \quad = 15.06$$

n = 15turns.

Using squared and grounded ends of spring, the total number of turns of spring for the weight hanger will be:
n1 = n + 2
= 15 + 2 = 17 turns

Free length of the spring

The free length (4) of a spring is the length of the spring in the free or unloaded condition. It is given by:
L_f = solid length + maximum compression + clearance between adjacent coils.
n^{1}d +δ_{max} + 0.15d_{max}
17 x 5.42 + 30 + 0.15(3D)
92.14 + 30 + 4.5 = 126. 64mm.

Stiffness of spring

This is the load required per unit deflection of the spring.

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$$K = \frac{W_{max}}{\delta} = \frac{500}{30} = 16.67\text{N/mm}$$

Pitch of The Coil Spring P

The pitch of a spring coil is defined as the axial distance between adjacent coils in uncompressed state.

$$P = \frac{Free\,length}{n^1 - 1} = \frac{126.64}{17 - 1} = 7.915\text{mm}$$

Hanger Rod

Diameter of the hanger rod will be equal to the internal diameter of the spring minus clearance or allowances.

Di =27.1mm

Rod diameter = Di - 0.1 Di

= 27.1 - 0.1(27.1) = 27.1 - 2.71 =24.39mm

Hanger Rod Head (Top)

This is a circular shape at the top of the rod to mesh on the top of spring to room for compression of the spring. Diameter of the hanger rod top is equal to outer diameter plus allowance.

D_o = 37.94mm

Rod head diameter = $D_o + 0.1\, D_o$

= 37.94 + 3.794 = 41.734 mm

Test for Brittle Aluminium

For a test conducted on the machine using brittle aluminium specimen, the following result on table below was obtained and when plotted on S - N graph, the graph below was obtained.

Stress (mpa)	320 300 260 200 150
Life (cycle)	10 10^2 10^3 10^4 10^5

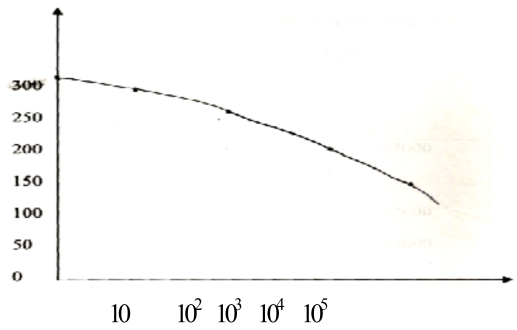

Fig. 2: An S-N graph of a test conducted on the machine using brittle aluminium

IV. CONCLUSION

When fatigue stress is induced on a material due to the action of force reversing and fluctuating, a failure known as fatigue failure takes place. The study and test conducted so far shows that fatigue failure cannot be predicted accurately since material failure under fatigue are affected not by just reversal loading alone but also the number of revolution (cycle per minute) and fluctuating stress and other factors such as temperature, atmospheric condition, both internal and external defect on material subjected under fatigue stress. Such defect includes notch, inclusion, stress concentration and non-homogeneity.

At such, fatigue failure is sudden and total, hence dangerous and leads to major accident characterized by loss of lives, valuable goods and devices. Thus all precautions and measures should be taken to checkmate this failure since it cannot be curbed entirely or predicted in-to-to.

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