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Development of a Fatigue Strength Testing Machine

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Abstract
A profound engineering design and modeling of fatigue strength machine for testing the failure which occurs when a metallic material is subjected to cyclic stresses is presented. The main goal is developing a locally made fatigue strength machine that can be used in material science laboratory. A major direction in this development of the machine is the ability to thread on the path of technology transfer and dominance by concentrating on the building of indigenous capacity. In carrying out the machine development, engineering design formulae were applied and Computer Aided Design modeling was developed. The developed fatigue strength testing machine had a designed machine capacity of 18875Nmm that sustained a motor power of 5.93kw.The machine shaft diameter determined to be 9.88mm developed an equivalent torque and bending moment of 45799.8Nmm and 40890Nmm respectively. The machine designed critical speed of 23.52rad/s had belt tensions on the tight and slack sides to be 139.02N and 44.65N respectively. The fabricated machine has the ability to record low and high fatigue cycles.

Keywords: Fatigue strength machine, failure, design and modeling

1.0 INTRODUCTION

Over the years engineering materials have been noticed to undergo repeated cyclic stresses which culminates into initiation of cracks and then total failure [1]. Fatigue brings about failures which are preceded by little plastic deformation and propagation of crack. The failure begins with a crack which might be very small to be detected. Over time such cracks have been known to develop in spots like keyway and holes where a localized stress concentration occurs [2]. Cracks initiation increases stress concentration effect which in turn promotes crack propagation. Fatigue failure has brought about colossal waste of resources across the entire globe. Major engine components and bridges have been known to face this huge challenge. The inability to predict fatigue failure is very remote as it does not give sign before an outright fracture [3]. The fatigue strength machine is applied in testing the fatigue life of materials before been deployed to construction site. Research conducted on scraps metal shows that 90% of failures are known to occur at stresses less than the yield strength of the material [4]. The untold technical and human loss caused by fatigue failure has made it mandatory to carry out proper consideration during material selection.

The fatigue research is timely as it stands the chance of acquainting us the technical know-how on tackling the associated challenges. Several researches have been conducted on material technology in ascertaining how to deploy materials that can withstand the menace or catastrophe associated with fatigue failure [5]. Recently, equipment made of Servo-hydraulic materials had been adjudged best for design facilities that withstand fatigue failure. A major drawback of this inference is the high cost of the servo-hydraulic mechanism [6].

In this present design of fatigue strength machine, there will be a focus of applying steel materials in constructing framework, load producing and transmitting links [7]. Some important components that were considered are frame, electric motor, digital counter and sensors that can aid the fracture of specimen and record the number of cycles to failures[8-14]. A prior knowledge of fatigue strength of materials will promote choice and material application in the area of construction, equipment, material handling and fabrication [9]. This study intends to bring to the fore the necessity of testing fatigue strength of metallic materials before application.

The realization of our true potential in this advent global technology will remain a mirage if attention is not paid to development of local content [10][16]. The shortcoming involved in developing local content for the production of laboratory equipment has been a major hindrance in science and engineering research in Nigeria.

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As a result some of the imported laboratory facilities come with incomprehensive manuals and numerical codes which are difficult for Nigerian technologists to comply with. This is known to impede the much talked-about technology transfer. This study is geared towards developing a local content fatigue strength testing machine that can be used in Nigerian material science laboratories.

2.0 METHODOLOGY

The basic fatigue strength testing machine consist mainly of the framework, load producing and transmission mechanisms as structural components. It has measuring devices such as counter for recording number of cycles to failures. The Fatigue strength testing machine was fabricated in the welding workshop of Auchi Polytechnic, Auchi.

2.1 Components of the Fatigue strength machine

2.1.1 Electric Motor

Electric motor is one of the commonest prime mover used in energizing machines. It converts electrical energy to mechanical energy.

2.1.2 Digital counter

High and precise Data recording is necessary for fatigue strength testing machines. A six digital counter was used to record the number of stress cycles attained by the specimen during the test.

2.1.3 Sensor

Signals from the rotating motion of the machine shaft are received by the sensor. The sensor sends signal to the digital counter.

2.1.4 Frame

The structural support of the fatigue strength testing machine is offered by the frame. The frame bears a good portion of the machines weight on its four legs. The material applied in the construction of the frame of this machine was made of steel angle bar.

2.1.5 Specimen

The machined test specimen made up of AISI 1080 steel was used for the experimentation. It was machined to a width and neck diameter of 9.5mm and 6.5mm respectively. The schematic diagram of the standard specimen is shown in Figure 1.

2.2 Design specifications

The design specifications used in this study are:

Shear stress of specimen(AISI 1080) =350Mpa
Maximum applied load = 25kg
Rotational speed in m/s = 500
Shear stress of AISI 1020 steel used for shaft=240Mpa
Speed in rpm = 3000
Coefficient of friction = 0.3
Density of mild steel = 7850kg/m^3.
Ultimate shear strength of steel = 480Mpa
Diameter of bolt = 5mm

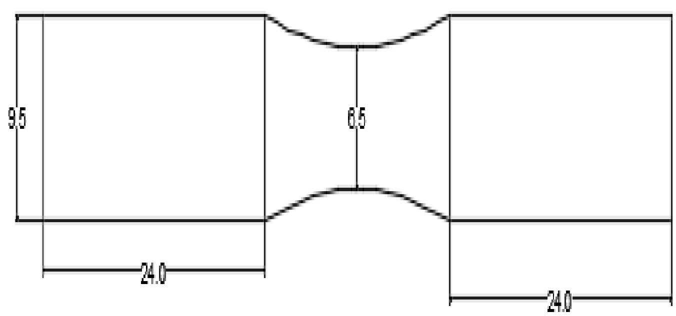

Figure 1: Schematic of Fatigue specimen

2.3 Design for Machine Capacity

The torque to cause yielding of the outer fibre of standard fatigue specimen of diameter of 6.5mm as prescribed by ASME E 466-82 code[7] was determined by applying equation (1) obtained from [7]

$$T = \pi \times \delta \times \frac{d^3}{16} \tag{1}$$

Where T= torque in Nmm
δ = Shear stress of specimen(AISI 1080) in Mpa
d= diameter of standard specimen in mm

Applying a specimen diameter of 6.5mm and a shear stress of 350Mpa yielded a machine capacity of 18875.3Nmm.

2.4 Design of Motor power

The motor power was determined by applying equation (2) obtained from [8]

$$P = \frac{2\times\pi\times T\times N}{60} \tag{2}$$

Where P= motor power in kw
N =Speed of motor r.p.m
Substituting the torque value of 18875.3Nmm and a speed of 3000rpm yielded a motor power of 5.93Kw.

2.5 Design of Shaft torque

The equivalent torque induced in the shaft is determined by applying equation (3) obtained from [8]

$$T_e = \sqrt{[K_b M_b]^2 + [K_t T]^2} \tag{3}$$

Where T_e=Equivalent torque induced in shaft
K_b =Fatigue shock factor for bending moment which is 1.5
K_t = Fatigue shock factor for twisting moment taken as 1.5
M_b=Maximum bending moment
T=Torque to cause yielding of the specimen outer fibre

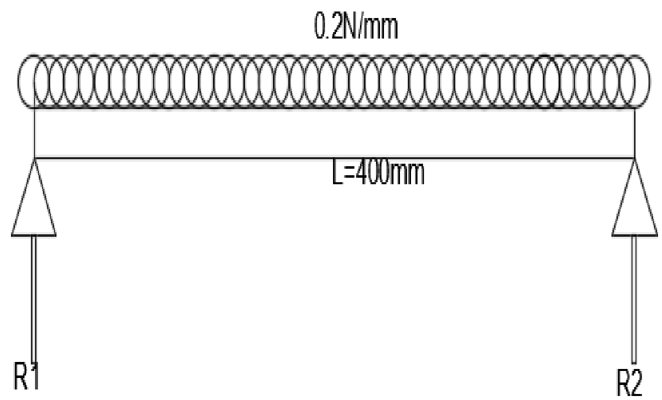

Figure 2: Uniformly distributed load on the shaft

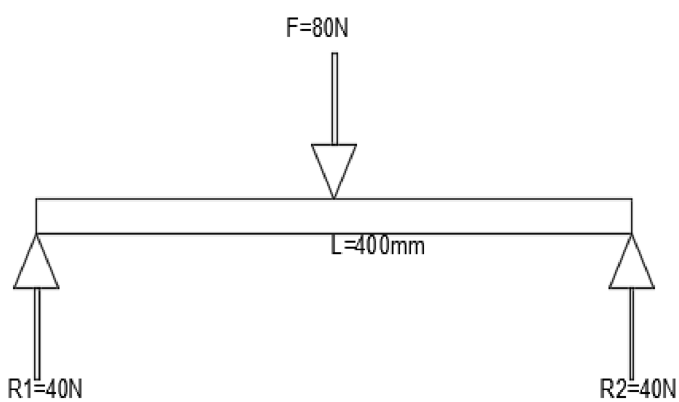

Figure 3: Point load on the Simply supported beam

Considering the power transmission shaft to be a simply supported beam with reactions (R1 and R2) at both ends and uniformly distributed load as shown in Figure 2 being reduced to a point load as shown in Figure 3. Applying a uniform load of 0.2N/mm over a distance of 400mm yields a point load of 80N. Using the principles of moments the reactions at both ends were determined to be 40N each. The maximum bending moment occurs at the centre of the beam. By Sectioning the beam at the centre and replacing the point load as 0.2×200=40N yielded a maximum bending moment of 24000Nmm as shown in equation (4)

$$M_b = 80 \times 400 - (40 \times 200) = 24000Nmm \qquad (4)$$

The equivalent torque is obtained by substituting in equation (3)

$$T_e = \sqrt{(1.5 \times 24000)^2 + (1.5 \times 18875.3)^2} = 45{,}799.8Nmm \qquad (5)$$

2.6 Design of Machine Shaft Diameter

The machine shaft diameter was determined by using equation (6).

$$T_e = \frac{\pi}{16} \times \tau_s \times d_s^3 \qquad (6)$$

Where τ_s= Shear stress of AISI 1020 steel used for machine shaft in Mpa
d_s =diameter of shaft

By substituting for equivalent torque, with shear stress of AISI 1020 steel used for shaft taken to be 240Mpa in equation 6 yielded a shaft diameter of 9.88mm.

2.7 Design of Bending Stress

The bending stress was determined by first getting the equivalent induced bending moment of the loaded shaft. The induced bending moment was calculated by using equation (7) obtained from [9]

$$M_e = \frac{1}{2}\left[K_b M_b + \sqrt{[K_b M_b]^2 + [K_t T]^2}\right] \qquad (7)$$

Where M_e= equivalent bending moment

$$M_e = \frac{1}{2}[1.5 \times 24000 + 45779.8] = 40890\text{mm}$$

The bending stress was determined by using equation (8) obtained from [9]

$$\sigma_b = \frac{32 \times M_e}{\pi \times d_s^3} \qquad (8)$$

Where σ_b= bending stress

The induced bending moment and shaft diameter were substituted into the equation (8) to produce a bending stress of 431N/mm^2. The ultimate shear strength of steel used in this study is 480Mpa. This shows that the design is safe.

2.8 Design of Bolt

The shear force on each of the two bolts used were determined using equation (9) obtained from [10]

$$Shear\ force\ on\ each\ bolt = \frac{1}{2} \times \frac{machine\ capacity}{10} \qquad (9)$$

The shear force on each bolt is 943.76N.
The area of the bolt was calculated using equation (10)

$$Area\ of\ bolt = \frac{\pi}{4} \times d_b^2 \tag{10}$$

Where d_b = diameter of bolt mm.

In this study, the bolt diameter was taken to be 5mm. By substitution of bolt diameter into equation (10) produced a bolt area of 19.63mm^2. The shear stress of the bolt is determined to by using equation (11) obtained from [10]

$$shear\ stress = \frac{shear\ force}{area\ of\ bolt} \tag{11}$$

The shear stress is calculated to be 48.08N/mm^2.

2.9 *Design of Critical speed of the machine*

The critical speed of the fatigue strength testing machine was determined by using equation (12) obtained from [11]

$$C_s = \left[\left(\frac{\pi}{l}\right)^2 \left(\frac{EgI_m}{A_s \rho l}\right)\right] \tag{12}$$

Where Cs=Critical speed in rad/s
l= length of shaft
E=Modulus of Elasticity
g = acceleration due to gravity
A_s=Shaft area
ρ =density of mild steel (7800Kg/m^3)
I_m =moment of inertia
The moment of inertia was determined using equation (13)

$$I_m = \frac{\pi}{64} d_s^4 \tag{13}$$

Moment of inertia was calculated to be 467.79mm^4.
The critical speed of the machine was calculated to be 23.52rad/s by substituting values into equation (12)

$$C_s = \left[\left(\frac{\pi}{400}\right)^2 \left(\frac{200000 \times 9.81 \times 467.79}{76.676 \times 7850 \times\ 400}\right)\right] = 23.52 rad/s$$

2.10 *Design of Belt tensions*

The belt tensions were determined using equation (14) obtained from [8]

$$\frac{T_1}{T_2} = \ell^{\mu\theta} \tag{14}$$

Where T_1= Tension in tight side
T_2=Tension in slack side
μ= Coefficient of friction taken as 0.3
θ=angle of wrap in radians

$$T_1 - T_2 = \frac{P}{v} \tag{15}$$

Where v =speed of belt in m/s
The speed of the belt was determined by using equation (16) obtained from [12]

$$v = \frac{\pi DN}{60} \tag{16}$$

The speed of belt was calculated to be 10.997m/s by using a smaller pulley diameter of 70mm and motor speed of 3000rpm. The difference between the tensions was determined to be 94.37N by substituting into equation (15).

Angle of contact was calculated by using equation (17)

$$\theta = 180 + 2\alpha \tag{17}$$

Where α=angle of lap
The angle of lap is determined by applying equation (18)

$$\alpha = Sin^{-1}\frac{R + r}{x} \tag{18}$$

Where x= distance between centres of the two pulleys
R=radius of large pulley
r =radius of smaller pulley

For a centre distance of 300mm, large and small pulley diameters of 120mm and 70mm respectively, α is 18.46°. Substituting for α in equation (17) yields a value of 216.92° for θ.

The angle of contact in radians is:
$\theta = 180 + 2(18.46)\frac{\pi}{180} = 3.786 rad$

The belt tension ratio was obtained to be 3.1136 by the substitution of the angle of contact in radians and the coefficient of friction in equation (14).

Solving equations (14) and (15) together yielded belt tensions T_1 and T_2 to be 139.02N and 44.65N respectively.

3.0 RESULTS AND DISCUSSIONS

3.1 Modelling of the Fatigue Strength Machine

The Fatigue Strength Machine was modeled using the AutoCAD 2016 software. The isometric drawing and the First Angle orthographic projection are shown in Figures 4 and 5 respectively. The pictorial views are similar to that designed in [13] and [14].

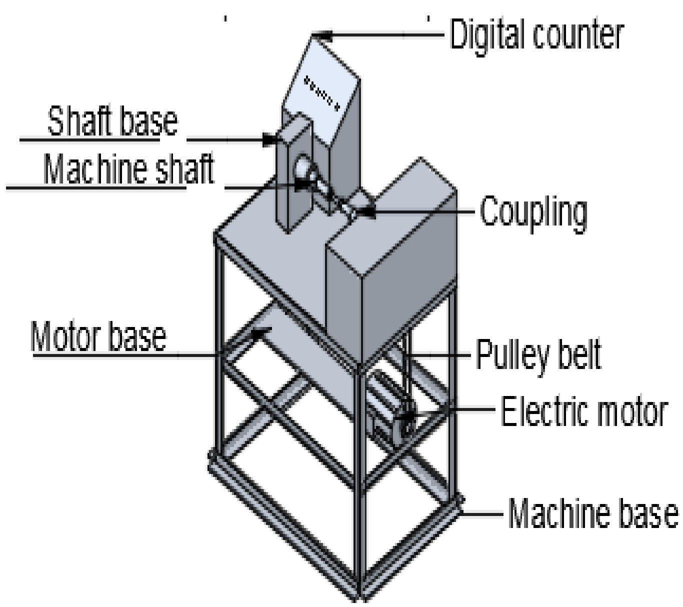

Figure 4: Isometric view of the fatigue strength machine

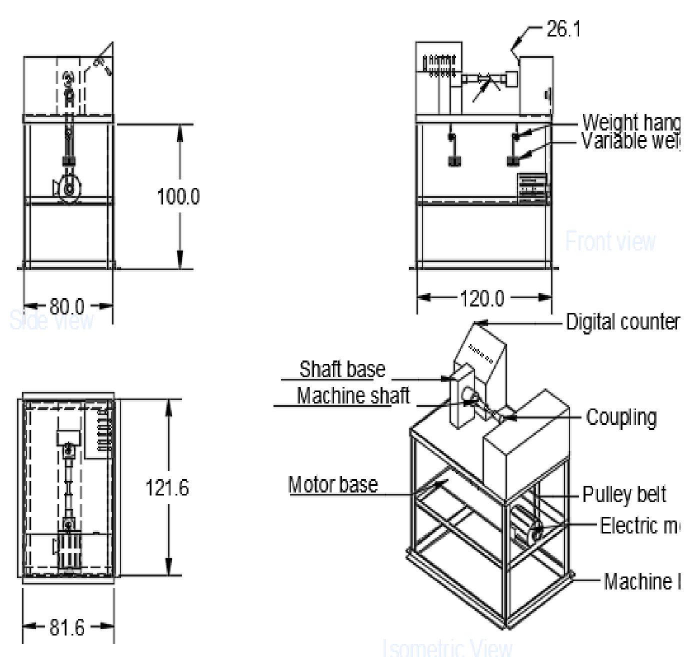

Figure 5: First Angle Orthographic Projection of the machine

3.2 Summary of the designed Parameters of Fatigue Strength Machine

The designed values of the machine parts are presented on Table 1.

Table 1: Designed data of the Fatigue Strength Machine

S/N	Parameters	Designed values
1	Machine Capacity	18875.3Nmm
2	Motor power	5.93Kw
3	equivalent torque	45799.8Nmm
4	Machine Shaft Diameter	9.88mm
5	equivalent bending moment	40890Nmm
6	shear stress of the bolt	48.08N/mm^2
7	Critical speed of the machine	23.52rad/s
8	Tension on tight side	139.02N
9	Tension on slack side	44.65N

The designed values calculated in this study are observed to be very close to values obtained in [14] and [15].

3.3 Fabrication of the Machine

The various components of the fatigue strength testing machine were technically marked out and fixed to the mild steel constructed machine base structure. The shaft which accommodated the specimen holder and pulley for power transmission was machined using the Computer Numerical Control Lathe machine. The locally fabricated Fatigue strength machine shown in Figure 6 was designed to be very simple in attaining low and high fatigue cycles for metallic alloys.

Figure 6: Fabricated Fatigue Strength machine

A scattered plot developed for the fatigue stress against number of cycles shown in Figure 7 portrays the conventional Wohler's S-N curve. It shows that the higher the fatigue stress applied the lower the number of cycles to failure.

3.4 Stress to Number of Cycles(S-N) Experimentation

The experimented result carried out with the fatigue strength machine is presented on Table 2.

Table 2: Fatigue strength Experimental Result

Test number	Load (N)	Stress(Mpa)	Number of cycles
1	240	520.5	1500
2	200	460.6	1805
3	160	420.8	2296
4	140	360.8	6800
5	120	310.6	12040
6	100	222.0	40040
7	80	180.5	60300
8	60	140.6	80000
9	40	106.0	96000
10	20	102.6	99010

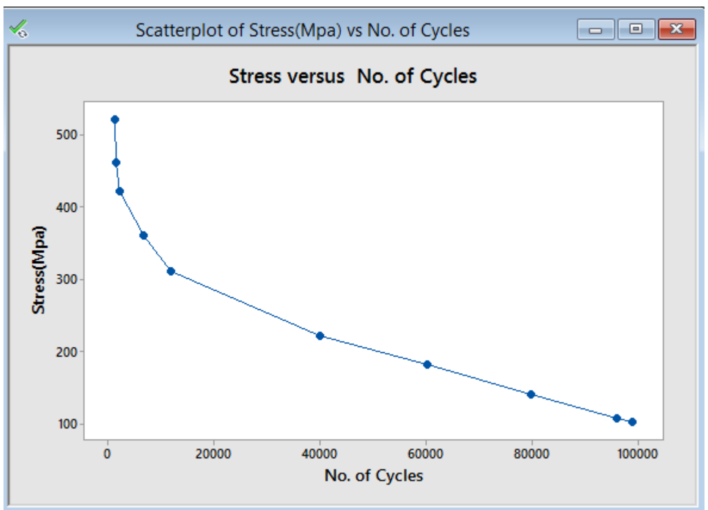

Figure 7: Scattered plot of Stress against No. of cycles

4.0 CONCLUSION

The existing challenge of importing workshop and laboratory equipment is a major drawback in our technological development. To make the Nigerian technology relevant in this dispensation all efforts should be geared towards building a local content in the development of machines that are deployed in our schools and industries. A boost in the Nigerian economy and conservation of our scarce import duties is likely to be imminent if the idea of our local content development is put into perspective. Although some of our researchers have done a lot in design of workshop equipment in material engineering there is still some appreciable gap to be filled. This study has brought to the fore the crave to develop material testing machine that can be applied in academics and the industries. The Fatigue strength testing machine which is the major focus of this study was accomplished by the creation of design calculation and graphical modeling that was applied in the fabrication of the machine in the workshop. The developed fatigue strength testing machine had a designed machine capacity of 18875Nmm that sustained a motor power of 5.93kw.The machine shaft diameter determined to be 9.88mm developed an equivalent torque and bending moment of 45799.8Nmm and 40890Nmm respectively**.** The development of the machine is key to the filling of the gap created by the importation of the material testing machine. This study successful designed a locally made fatigue strength testing machine for our material science laboratory.

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