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# Design and Fabrication of Dynamic Testing Machine

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## ABSTRACT
The focus of this paper is on the design and fabrication of a dynamic testing machine capable of generating and testing the fatigue loading created on a steel-based specimen. The primary aim of this work is to adopt a simple design procedure followed by the fabrication with locally available cost-effective materials to enhance the lab facility. A mechanism based on 'four-points-bending' and simple torque balance was suggested out of which both static and dynamic fatigue loading were possible to generate. As a dynamic load generating device, a constant frequency drive motor was employed and then transferred to the specimen held by four rollers under compression. Though it was designed and fabricated based on steel specimens only, it is compatible with other materials also. The machine is also capable of testing the fatigue life cycle of different materials. Finally, a fast design and fabrication process has resulted in a dynamic fatigue testing machine at a lower cost of about 14000 BDT. As a future study, the machine is subjected to frequent testing and capability enhancement.

Keywords: Machine Design, Lever Mechanism, Dynamic Load, Fatigue Crack.

## 1. Introduction
Scientists have studied fatigue for many years and its prevention had been a major concern for years long. With the technological advancement research about fatigue in structural members is also developing. As we still do not understand it properly experimental research plays an important role in getting savvy knowledge of fatigue.

The aim was to design and fabricate a machine that could generate cyclic load on different specimen like stainless steel, iron, etc. It can generate fatigue crack on a rectangular specimen which can be studied further via different methods. Thus, the fatigue life of different materials can also be studied.

Kulkarni et. al. presented work on plane bending fatigue testing where they designed such a machine for composite material [1]. Handrik et. al. undertaken a simulation-based analysis of strain and stress fatigue produced in the specimens by a bi-axial fatigue machine [2]. Montalvão and Wren proposed a methodology to redesign the existing specimens for ultrasonic testing machines [3]. Berchtold and Klopfer in their paper summarize the fatigue testing at high frequency focusing on the consequence of variable loads on the specimen [4]. Ghielmetti et. al. developed a bending fatigue tester. The authors claimed it a simpler approach having a cost-effective scheme [5].

In this machine, both static and dynamic loading is generated. The motor is used for generating the dynamic load. Extra static weight, the weight of the motor, the weight of the upper bar will create the static load. When the motor runs, fluctuating loading will be exerted on the specimen. For the stability of the machine, the static loading will be bigger than the dynamic load, that is the minimum load will also be compressive.

## 2. Governing Equations
Four-point bending on a rectangular bar (stainless Steel) was considered for fatigue loading. Following mathematical expressions were used to determine the value of the required load [6-8]:

$$K_I = F_{IP} \times K_I|_{s/T\to\infty} \quad (1)$$

$$K_I|_{s/T\to\infty} = \frac{3Pl}{WT^2}\sqrt{\pi(a+N)} \times F_{IM}(\alpha) \quad (2)$$

$$F_{IM}(\alpha) = 1.122 - 1.121\alpha + 3.740\alpha^2 + 3.873\alpha^3 - 19.05\alpha^4 + 22.55\alpha^5 \quad (3)$$

$$\alpha = (a+N)/T;\ \alpha \quad (4)$$

$$K_{Imax} = 22\ MPa \cdot m^{1/2} \quad (5)$$

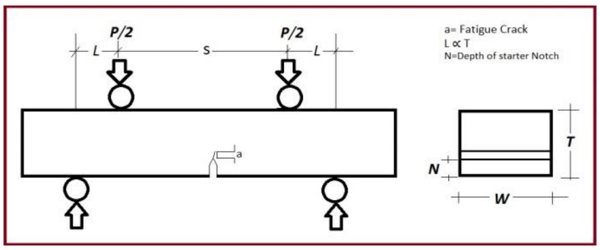


**Fig.1** Four-point bending on the specimen

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## 3. Design Methodology

Durability & efficiency was considered during the design of the machine. The whole machine was designed sequentially as a combination of a range of parts. The design methods have been shown in figure 2 where load requirements of the specimen to be cracked were calculated in such a way that loads should have

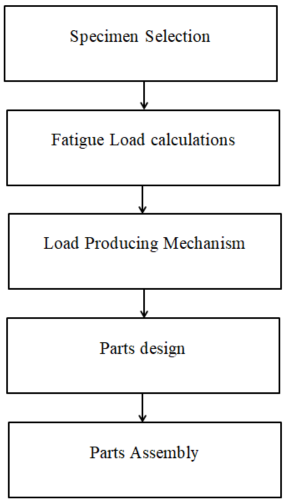


**Fig.2**: Block Diagram of Design Steps

been reconstructed with adequate rigor and conveyed to test specimen accordingly.

Based on the specimen selected as SS bar of dimension (215mm× 34mm×14mm), notch depth (N) of 5 mm, and zero initial fatigue crack the total load to be exerted on the bar was calculated as 21.91 KN according to the above equations. But, to generate such a static load, a huge weight was to be used which is not economical. In that case, dynamic loading of 9.91 KN was taken in such an optimum so that both stability of the machine and faster cracking time could be achieved. The third step was the load producing mechanism where methods of load production were determined with the help of the moment and torque concept.

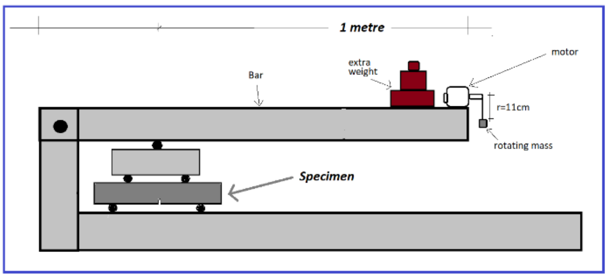


**Fig.3**: Schematic diagram of the machine

Figure 3 and Figure 4 depict the schematic diagram of the machine and load distribution showing the force calculation method at point b to create the desired force at point b considering equilibrium conditions. So, for static loading, the load was produced mechanically having a bar weight of 26 kg, extra static weight of 44.3 kg, and motor weight of 10 kg. The dynamic loading was created by a constant frequency drive motor having the following values:

M=230 gm (rotation mass)

r = 11 cm

rpm = 1400 (rotation per min of the motor)

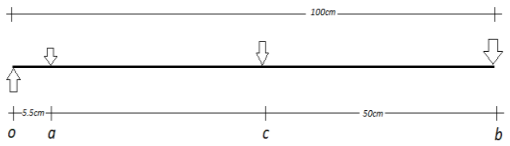


**Fig.4**: Load Distribution

The power requirement of the motor was calculated in the following way [9],

Power = $2 \times \pi \times n \times t/60$ watts n=Revolution

= 750 watts

= 1 hp

The fourth step was designing the parts of the machine. The torque transmitting shaft, upper bar, sidebar, base, and vibration absorber constitute the whole machine. The torque transmission shaft was designed in the following way that $d = 16 \times t/\pi \times \Gamma = 12$ mm [9].

To transmit the load to the specimen simple leaver mechanism was used. That's why the whole extra weight was added at the right end of the upper bar which in turn produced a huge compressive load on the specimen. For better efficacy, the upper bar was pivoted to the main machine body by a shaft. As the load producing upper bar was one meter long it was able to produce a huge compressive load with a leaver mechanism. There was flexibility in positioning the specimen: that is the specimen can be moved left or right to get the desired load. During designing the following dimensions were calculated: Total fluctuating load required to create bending on the specimen, the diameter of the shaft, an approximate distance between the two ends of the upper bar, safe bending on the shaft and upper bar, safe fatigue endurance.

Since the specimen is always in compression, there was no stability concern. The vertical load of 21.91KN will be shared by two rollers which were fitted in grooves. Rollers with 2mm diameter were used with a safety factor.

The stability and volume of the machine depend on the base structure. Cast iron (channel-shaped section) was used here. This is the most robust part of the machine making an appropriate length of 130 cm, and a height of 10 cm, and a width of 20 mm.

Finally, an Elastomeric Bearing pad was used for lesser vibration and high damping under the base structure. After designing all the parts, every one of them was assembled. Fig. 5 demonstrates the final assembly of all the parts designed and assembled with the help of Solidworks software.

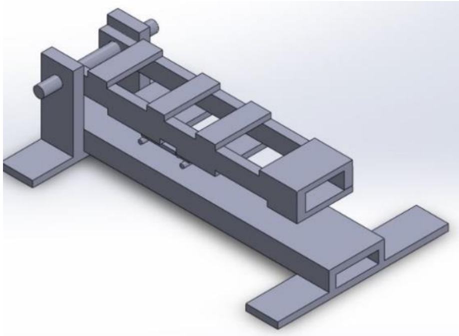

**Fig.5** Final Assembly

## 4. Fabrication of the Machine

4.1 Specimen Setup

During the specimen set up 4-point bending was used instead of 2-point bending. Because it would have deformed the structure of the specimen. So, 4 rollers were used as shown in the figure. The load that is created by the machine is transmitted by the single roller on the topmost position, then it is transmitted to the specimen. As the load fluctuates with time it would create a fatigue crack on the specimen that will help us in future research. The specimen will experience shear force and bending moment because of the fatigue loading.

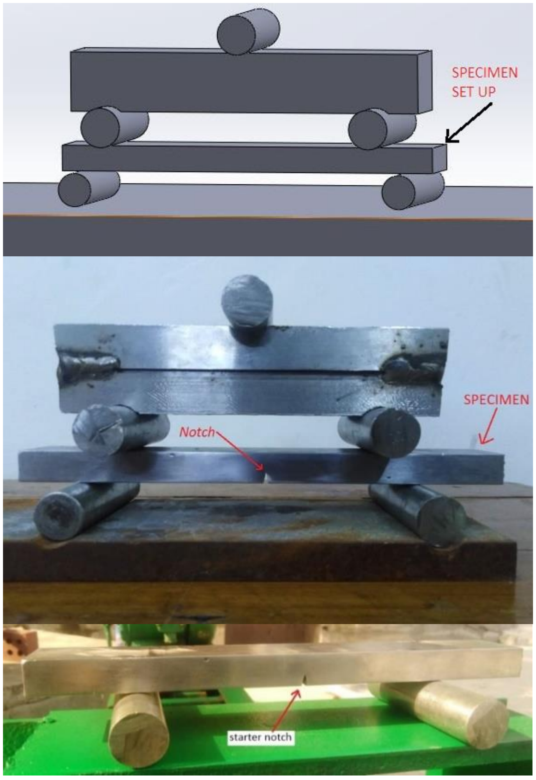


**Fig.6 Specimen Setup**

Fig.6 demonstrates the setup of the specimen labeling the position and depth of the starter notch.

4.2 Fabricated Machine

For the fabrication of the machine, we collected all the materials from the local market. For the base structure, a U channel iron bar of 1.5-meter length was collected. Then sidebars were joint with it via welding joint. Before joining the sidebars two holes were created on them so that the upper bar could be connected with the sidebars by a shaft. During the design, one thing was kept in mind that is decreasing the weight and increasing the mass moment of inertia. While fabricating the upper bar (reciprocating bar) two U channel bar was taken both with the length of 1 meter. Then they were joined by pieces of angle bars in between them by welding. Then this was connected to the sidebars by the shaft. Then according to the

**Fig.7** Closed View of Rotating Weight

Fig.7 demonstrates the closed view of the rotating weight which produces load in fluctuating order. The motor essentially provides this by creating centrifugal motion.


**Fig.8** Machine after Complete Fabrication

Fig.8 demonstrate the complete fabrication of the machine by labeling different parts of it like base, position of the specimen, rotating mass, reciprocating bar, vibration absorber, and static weight

## 5. Cost Analysis

The machine was so designed and developed that it turns out to be cost-effective. The cost list has been depicted in Table 1

**Table 1** Component and Cost List

| Component | Cost (BDT) |
|---|---|
| Motor | 3500 |
| Shaft | 500 |
| Upper Bar | 2500 |
| Base and Side Bar | 2000 |
| Specimen | 2000 |
| Extra weight | 2000 |
| Service Charge and Others | 2000 |
| Total | 14000 |

Thus, the cost of the machine was possible to keep below 14000 BDT, which is much less than that found in the market.

## 6. Results

When the machine operates the subjected specimen will experience harmonic loading which will vary from 21.91KN to 2.09 KN. This type of loading will generate a fatigue crack in the specimen. The specimen will experience both shear force and bending moment which will also vary with the changing load. Fig.9 and Fig,10 demonstrates theoretical results by showing a cyclic load of the specimen and shear and bending moment

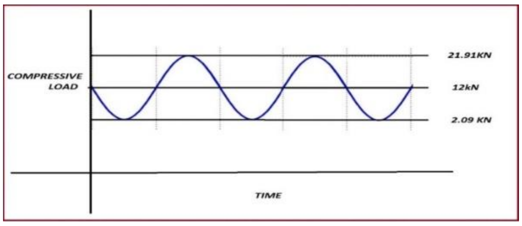


**Fig.9** Cyclic load variation on the specimen

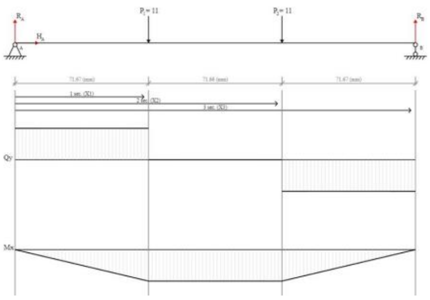

**Fig.10** The shear force and bending moment diagram (when the load in maximum)

## 7. Discussion

The inspection of the specimen is not easy. The motor becomes hot after some time. After running for about 20 minutes, it becomes hot. The machine is very heavy. So, the movement of the machine is very tough.

## 8. Conclusion

The simple design approach was used during the fabrication of the machine. The design materials were collected from domestic markets so cost minimization was achieved. A small number of vibrations was nullified due to the heavy structure and by using rubber pads beneath the base structure.

Inspection problems can be avoided by adding a zoom-in camera connected to pc. A better vibration absorber can be used to reduce vibration and sound. Here we used a constant frequency motor for creating fluctuation load. If one can use a variable frequency motor, the required fluctuating load can be also varied. As the machine is very heavy, a bearing can be added below the base for easy movement of the machine. As the motor becomes hot, a microcontroller-based switch along with a timer can be used. It can maintain operation time and interval time.

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**NOMENCLATURE**

$K_I$ : stress intensity factor, MPa
$K_I|_{S/T\to\infty}$: 4-point pure bending S.I.F, MPa
$\alpha$ : fracture depth to plate thickness
$K_{Imax}$ : maximum stress intensity factor (SS)
$N$ : starter notch, mm
$P$ : load, KN
$d$ : diameter of transmission
$\Gamma$ : torque intensity factor