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Design and Development of Fatigue Machine: Rotating Bending Fatigue Testing on different Materials

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Abstract

Fatigue fracture is the most common type of failure as more than 75-90% of documented materials' failure in engineering components due to cyclic loading. So, it is preeminent to test the materials before use. This research work has been done in order to design and fabricate "Rotating Bending Fatigue Testing Machine" which is simple to understand, perform, portable, safe for use and more economical than others. A rotating bending fatigue testing machine was developed by keeping in mind the basic concepts of technical theory of bending of elastic beams. Different specimens of 6mm to 8mm diameter of different materials were tested under various loads to analyse the performance of this machine. The experimental readings and theoretical calculations indicate that the results were promising to agreement of the objective had been achieved. It was detected that the machine has the abilities of generating reliable bending stress – number of cycles data; and the cost of design (31,500 PKR) was lower in evaluation to that of rotating bending machines from abroad.

Keywords

Analysis, Bending, Fracture, Failure, Fatigue, Fatigue Machine, Stress,

I. DESIGN OF EXPERIMENTAL SETUP

The phenomenon of premature failures originating from repeated stress was first noted in the railway industry in the 1840s as railroad axles constantly failed at the shoulders [1]. The rounding of sharp corners did not obviate this type of failure which was termed fatigue and earmarked to failure under repeated stresses[2]. The earliest systematic surveys into this phenomenon were conducted by August Wohler 1819-1914 who presented the results of his fatigue tests at the Paris exhibition of 1867 which were also reviewed in the same year in engineering [3]. In 1870 Wohler concluded that the stress range plays a most significant role in fatigue failures. Wohler was the first to introduce the concept of the so-called s-n diagram which relates applied stress "s" to life "n" i.e. to number of cycles to failure. He could thus show that fatigue life decreases with higher stress amplitudes and material reaches a fatigue limit. The consideration of fatigue failure of materials in industrial use is thus important for the safety of components, particularly under the oscillating loads that leads to fatigue[4]–[7]. The fatigue life of a component is defined by the total number of stress cycles required to cause failure. Fatigue is generally understood as the gradual weakening of a material which is exposed to cyclic loads.

In fatigue testing a specimen is exposed to periodically fluctuating constant amplitude stress. The applied stresses may different between equal positive and negative value from zero to maximum positive or negative value or between equal positive and negative values or between unequal positive and negative values [8]. Therefore it is important to consider the experimental fatigue failure of materials as the theoretical equation do not always fit to the real life fatigue behavior of components [9]. So, it is important to carry out mechanical fatigue tests in different conditions and on a wide variety of materials to investigate the principal causes of failure in engineering components subjected to uniaxial and multiaxial loading[10]–[12].

Over the years, there have been many testing machines and methods developed for fatigue testing by many authors[13]–[15]. In regard of the fatigue machines, the first prototype was developed in the nineteen century[16] , in this machine load was induced by mechanical deflection and inertia forces. The systematic studies on fatigue have been carried out in the second half of the twenty century on aluminum alloys[17].The servo – hydraulic machines are currently the fastest having features that allow for a wide range of test to be performed on it [18], [19]. The high price of these machines is a downside to its use. This research work addresses the problem by developing a low-cost rotating bending fatigue testing machine. The theory governing the design of the fatigue machine is based on elastic beam bending theory [20]. Experimental values of fatigue life are confirmed using theoretical calculations.

A. Design Objectives

The aim is to design and fabricate a rotating bending fatigue testing machine that is capable of testing the fatigue life of various samples of specimen of different minimum diameters and capable of calculating the numbers of cycles required to fail specimen. The machine must be cost effective, must have minimized time of experiment and enhanced loading capacity.

B. General Description

The fatigue-testing machine is of the rotating beam kind. The specimen functions as a single cantilever beam loaded at one end. When interchanged one-half revolution the stress in the fibers initially above the neutral axis of the specimen are inverted from compression to tension for equal intensity. Upon finishing the revolution, the stresses are again inverted, so that during one complete revolution the test specimen passes through a complete cycle flexural stresses i.e the specimen is subjected to the tension and compression cycles. The cycles are recorded by mean of proximity sensor and output is displayed on a revolution counter. The machine is equipped with tacho-sensor to show the revolution per minute(rpm). Specimen is clamped by mean of chucks. The machine features two shafts, a driving shaft rotated by mean of motor and a driven shaft for supporting the specimen and application of load.

C. Component Selection and Design Calculations

a. Electric Motor

The electric motor used is of specifications as shown in table 1.

Table 1. Motor specifications

Voltage	Phase	Frequency	Rated Power	RPM
220 V	Single AC	50 Hz	0.75 KW	2880

b. Shaft

A high carbon alloy steel that is EN8/ AISI 1045 was selected as material for shaft. The main function of the shaft is to interchange the specimen while it is under the action of bending moments from the dead weights hanged to overhang bearings. The shaft is subjected to bending moment only so: We know that

$$\frac{M}{I} = \frac{\sigma}{y}$$

Where:

M= bending Moment

I= Moment of inertia $= \frac{\pi}{64} \times d^4$

$\sigma = Bending\ Stress$

y = Distance from neutral axis = $\frac{d}{2}$

Final form of equation is $\sigma = \frac{32M}{\pi d^3}$

For applied load of 300N (30Kg) and a shaft of 1-inch diameter and 11.5 inch in length, the maximum bending stress is 69.73 MPa

M = W x L = 293.3 x 0.381 = 112.12 Nm

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$\sigma = \frac{32x112.12}{\pi 0.0254^3}$ = 69.73 MPa

Which is less than the yield strength (310 MPa) of AISI 1045/EN8 Steel.

So, selected parameters of shaft are:

Material= AISI 1045/EN8

Diameter= 1 inch

Length= 11.5 inch

c. Specimen

The specimen selected are of 6mm and 8mm neck diameter. The maximum bending stress in rotating specimen can be modeled as max bending stress of simply supported beam as:

$$S_f = \frac{8WL}{\pi d_n^3}$$

Where

W= Load applied

L= Length of Specimen = 150mm

D= Neck Diameter= 6mm & 8mm

And the number of cycles is given as:

$$N = [\frac{S_f \times S_e}{(S_u \times 0.9)^2}]^{-3/\log(0.9^{S_u}/S_e)}$$

Where

Sf= max bending stress

Se= Endurance limit of material

Su= ultimate strength of material

d. Drill Chuck

A three-jaw drill chuck with taper fittings was selected as the specimen clamp for the fatigue machine. The specimen fasten is expected to firmly hold the specimens without allowing for extraneous bending moments during operation of the machine. Also, the specimen must not rotate from the grip or be displaced vertically or horizontally.

e. Pedestral Bearing

The bearings carefully chosen for the design were self-aligning roller bearings, which have high load carrying capacity and it can accommodate misalignment and shaft deflections. Bearings are inserted into Plummer blocks. Using NHBB Hi-tech Ball bearings Design Guide 2010, according to the diameter of shaft, bearing of following parameters is selected.

Table 2. Bearing specifications

Series	Metric
Type	Cylindrical Roller
Code	TP-305
Alignment	Self
Deflection	0.5^0
Bore	25 mm
Load bearing capacity	9300 DYN

f. Jaw Coupling

Jaw coupling made up of grey cast iron is used to transfer power from motor shaft to the shaft. Jaw coupling is used because the spider tolerates the angular misalignment if any and adjust accordingly. A coupling of Jaw type, size L/Al090 is selected using “Love Joy Coupling Solution Catalog 2010”. They need no lubrication and deliver highly reliable service for light, medium, and heavy-duty electrical motor.

Table 3. Coupling Specifications

Size	Max Bore	SOX (NBR) Torque
L/AL090	25 mm	16.3 Nm

g. Rubber Pads and Packing

A rubber pad of 8 x 8 x 0.5 inch is used as a packing beneath the motor to damp vibrations that are coming from the motor and isolate it from the table hence from the specimen. A metal packing is placed below the Plummer blocks so as to serve the vertical alignment.

h. Weight Hanger and U-Clamps

U-clamps and weight support arrangement is used to apply load on overhang bearings. U clamps are bolted to overhang bearings and weight hanger is hanged on it. Dead weight is added to the hangers.

i. Frame

Frame is the foremost supportive element in the system. The frame has to tolerate all the weight of the experimental setup. The force exerted on the system is distributed to the four legs. The material used is GI rectangular pipe.

II. Design Of Experimental Setup

i. Schematic Diagram

The schematic diagram was made on AutoCAD software.

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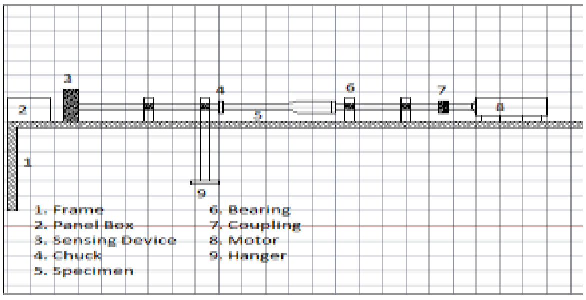


***Fig.1** Schematic Diagram of Rotating Bending Fatigue Testing Machine*

The 3D models were generated using two software:

Microsoft Paint 3D with Mixed Reality

SolidWorks

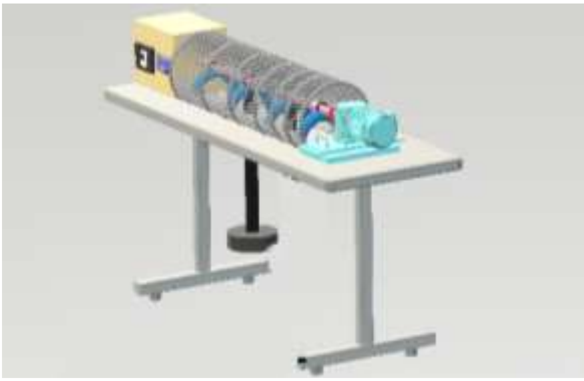


***Fig. 2** 3D Model*

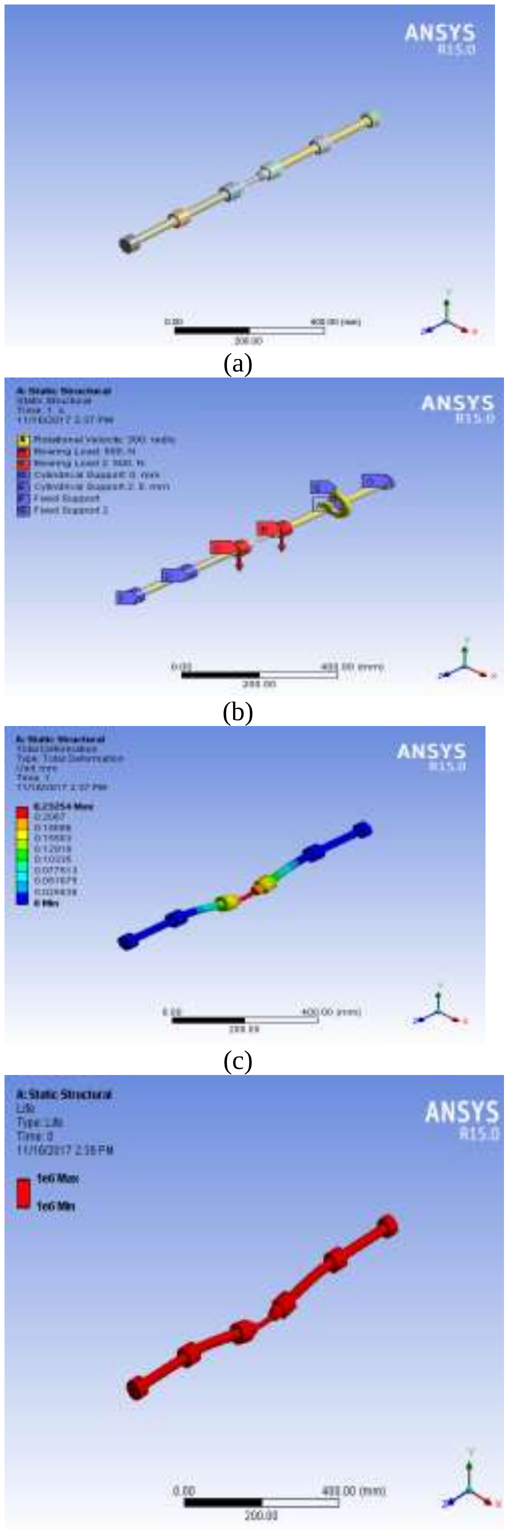
III. Finite Element Analysis

A simple setup of shaft and specimen is modelled in Solid Works and then it is imported into ANSYS WORKBENCH. Fig 3a Various loading situations like bending, torsional moments and cyclic speed are applied. Fig 3b

1. Cyclic Speed: 300 rad/s
2. Bending Force: 500N on both ends
3. Fixed Support at both end

And solved for solution of following parameters,

a) Total deformation-it shows that maximum deformation occurs at midpoint of specimen as shown in Fig 3cand deformation is 0.2854 mm before specimen undergoes failure.
b) Safety factor (in fatigue tool)-this shows that failure will occur at minimum cross-sectional area of specimen as it has zero safety factor as shown in Fig 3d.



(a) (b) (c) (d)

Fig. 3 Ansys simulation result (a) Assembly, (b) boundary Conditions (c) deformation (d) safety Factor

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IV. Fabrication Process

After finalizing design of rotating bending testing set up materials were obtained from local market and assembly was carried out.

Angular bars were cut according to dimension and welded to make table frame. Metallic sheet was cut and was welded into the frame which was used as base for mounting. The layout of whole setup was sketched on table according to dimensions of components. Motor was mounted on platform and rubber pads were used as packing below the motor. Coupling was tapped so that motor shaft fixes into one of the flanges and other flange holds shaft. Shafts were cut and tapered at one end and supported by block bearings. The tapered end was fixed and tightened into drill chuck. Slot was made using jigsaw machine in table for overhanging of U clamps. After that weight hanger was hanged to U clamps and loaded with dead weights. Electrical connections were made for the motor and main panel board. The sensors were positioned near the coupling. Fig. 4 shows the experimental setup.

Fig. 4 ***Fabricated Machine***

V. Results and Discussion

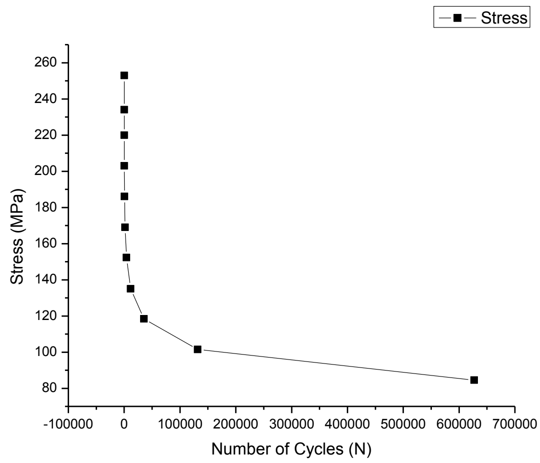
Fatigue tests were performed and analyzed on four different materials: 4.5% cast carbon iron, aluminum 6061-T6, mild steel 1080 and stainless steel 1020. The tests were obtained at room temperature (22°C) and with environmental humidity comprised between 45% and 60%. The four materials were subjected to fatigue tests at different levels of applied load. All tests were performed at 50 Hz of frequency.

A. Fatigue Life

a. Cast Carbon Iron

Table 4. 4.5% Cast Carbon Iron

Sr No	Load (N)	Max Bending Stress (S_f)(MPa)	Number of Cycles (N) Theo.	Number of Cycles (N) exp.
1	100	84.5	627099	□
2	120	101.5	131667	□
3	140	118.4	35439	23,970
4	160	135	11590	9,439
5	180	152.3	4150	3,801
6	200	169	1710	1,201
7	220	186.1	753	559
8	240	203	359	239
9	260	220	181	103
10	280	234	107	60
11	300	253	55	-------



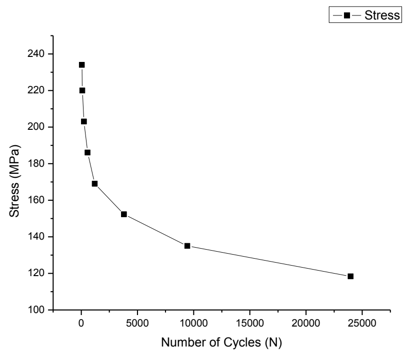


***Fig. 5**S-N Curve of Cast Carbon Iron*

From the values obtained TABLE 4 during the testing of the specimen of 4.5% cast carbon iron, S-N graph is plotted with stress (MPa) on y-axis and number of cycles to failure on x-axis. From the above graph it can be noted that with reduction in load the number of cycles to failure rises. Below a certain load limit the number of cycles to failure rises and approaches infinity, this limit is known as endurance limit.

b. T6 6061 Aluminum

Table 5. T6 6061 Aluminum

Sr No	Load (N)	Max Bending Stress (S_f)(MPa)	Number of Cycles (N) Theo.	Number of Cycles (N) Exp.
1	100	84.5	129351	-----------
2	120	101.5	71232	65,913
3	140	118.4	57017	49,827
4	160	135	44941	37,796
5	180	152.3	33901	26,731
6	200	169	21338	17,631
7	220	186.1	13829	9,671
8	240	203	7503	5,630
9	260	220	4286	3,336
10	280	234	2534	1,673
11	300	253	1303	879

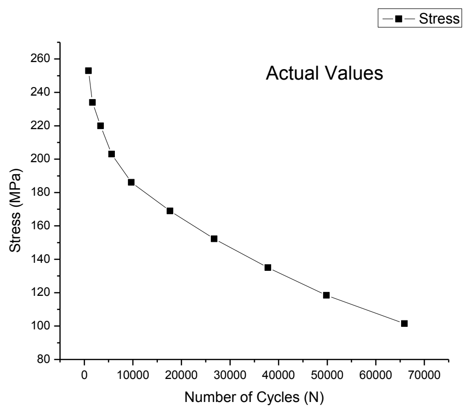


***Fig. 6** S-N Curve of T6 6061 Aluminum*

From the values obtained TABLE: 5 during the testing of the specimen of T6 6061 Aluminum, S-N graph is plotted with stress (MPa) on y-axis and number of cycles to failure on x-axis. From the above graph Fig 6., the endurance limit of material is the obtained.

c. Mild Steel 1080

Table 6. Mild Steel 1080

Sr No	Load (N)	Max Bending Stress (Sf)(MPa)	Number of Cycles (N) Theo	Number of Cycles (N) exp
1	100	176.84	7509	7019
2	120	212.3	5336	4900
3	140	247.6	4812	4498
4	160	283	3799	3371
5	180	318.47	2888	2109
6	200	353.8	1733	1311
7	220	389	1184	927
8	240	424.6	424	302
9	260	460	166	91
10	280	495	70	54
11	300	530.7	31	18

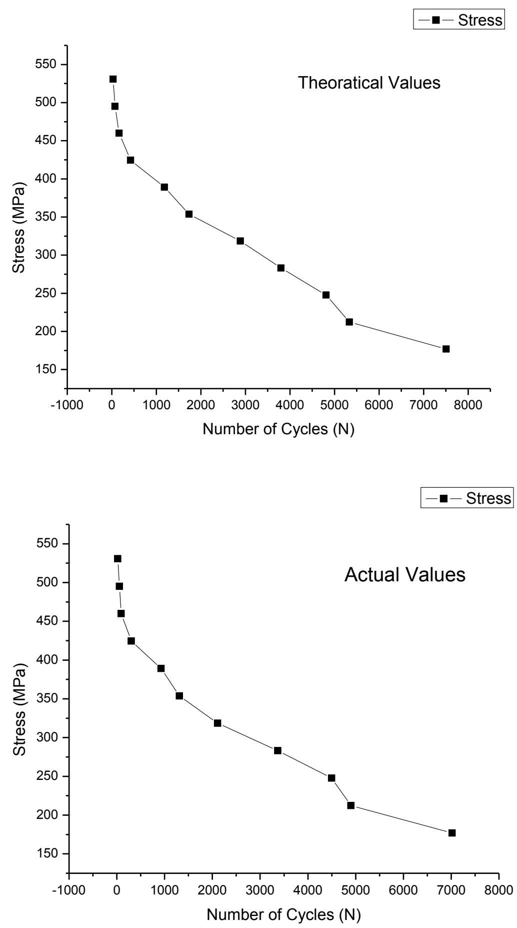


***Fig. 7**S-N Curve of Mild Steel 1080*

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From the values obtained from TABLE: 6 during the testing of the specimen of Mild Steel, S-N graph is plotted with stress (MPa) on y-axis and number of cycles to failure on x-axis. From the above graph Fig.7, the endurance limit of mild steel is obtained.

d. Stainless Steel 1020

Table 7. Stainless Steel 1020

Sr No	Load (N)	Max Bending Stress (S_f)(MPa)	Number of Cycles (N) Theo.	Number of Cycles (N) Exp.
1	100	176.84	368,797,728	---------
2	120	212.3	43,110,770	---------
3	140	247.6	7,088,677	---------
4	160	283	1,471,278	----------
5	180	318.47	367,716	----------
6	200	353.8	106,737	----------
7	220	389.04	34,959	29331
8	240	424.6	12,516	9,271
9	260	460	4,878	3,201
10	280	495	2,062	1,931
11	300	530.7	911	607

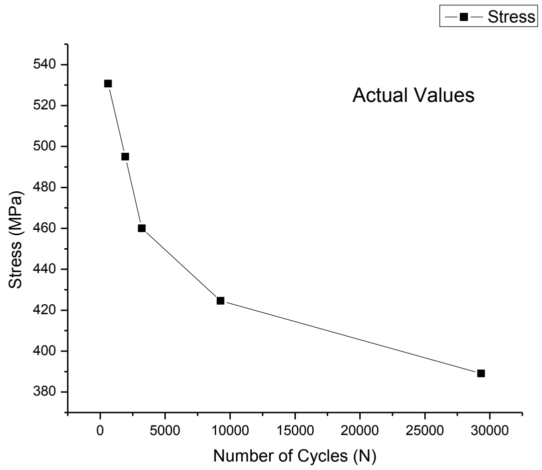

Fig 8.*S-N Curve of Stainless Steel 1020*

From the values obtained TABLE: 7 during the testing of the specimen of Stainless Steel 1020, S-N graph is plotted with stress (MPa) on y-axis and number of cycles to failure on x-axis. From the above graph Fig 8, the endurance limit of stainless steel is obtained.

From the test conducted on these four different materials it can be concluded that the stainless steel of grade 1020 has better fatigue strength than the mild steel of grade 1080, t6 6061 aluminum and 4.5% cast carbon iron.

B. Cost Analysis

The materials and equipment used in the design are locally obtained, and the overall cost of designing the machine is approximately 31,500 PKR ($228.28). The machine is economical in comparison to similar designs from abroad.

Table 8. Cost Analysis

Sr. No.	Description (Materials)	Quantity	Our Design Cost	Nigerian Design Cost [21]
			Price (PKR)	Price (₦)
1	Electric Motor	1	2,850	30,000
2	Proximity	2	2,200	17,200
3	Speed Counter	1	2,300	24,500
4	Chucks	2	1,000	4,000
5	Bolts and Nuts	---	500	2,450
6	Ball Bearings	4	1,600	2,400
7	Flat Metal Plate	1	3,800	6,000
8	Shaft	2	500	5,000
9	Cycle Counter	1	1,800	17,200
10	Switch	3	550	1,400
11	Emergency Switch	1	175	800
12	Wood press sheet	--	1,600	7,500
13	Frame	--	4,600	15,000
14	Electrical Connections	--	2,000	5,000
15	Coupling	1	700	6,000
16	Paint	--	2,000	6,500
18	Fabrication	--	3,500	30,000
19	Total		PKR 31,500 USD 228.28$	180,950 Naira USD 504.04$

VI. CONCLUSION

The testing machine was fabricated and tested. The experimental results were compared with theoretical calculations and it was found that there was little variation in results. It was also observed that the machine has the potentials of generating reliable bending 'stress-number of cycles' data. The study and test directed so far shows that fatigue failure cannot be predicted correctly since material failure under fatigue are exaggerated not by just reversal loading alone but also by factors such as fluctuating stress, temperature, atmospheric condition, both internal and external defect on material subjected under fatigue stress. Such flaws include notch, inclusion, stress concentration and non-homogeneity. The advantage of the present design is the easiness of its modeling and ease of understanding. By using simple mechanism, the cost of this machine has been decreased to Rs. 31,500 (228.28$) without affecting the experimental results. On the other hand, market price of this machine used by various institutes' ranges from 75,000 to 2.5 lakhs PKR. This machine is helpful for the institutions and students to perform various materials testing in less time. This machine is currently in use in mechanics of materials lab of an engineering institute.

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