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Fabrication of Spring Mass System

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Abstract: *A spring testing apparatus is a system to reduce vibration and improve the balancing system. This system has tremendous applications, especially for analysing vibration. Its application is mainly in academic and industrial laboratories. The main aim of this project is to determine the spring characteristics of the spring stiffness for both the compression and tension conditions. This methodology analyses the characteristics of a spring under various loads. The apparatus accommodates different S-type hangers with spring mass span. After applying different masses, we had to analyse the behaviour of the spring. Noted the change of loads with deflection measures the incremental load of the spring under investigation. Two steel rules, attached on each side of the rectangular hole, enable the apparatus to measure the extension of the spring. A Mild Steel base is supported on four rubber bushings, and these give the apparatus a firm stand and balance. This project is a prototype that can be improved for implementation in industrial settings.*

Keywords: *Design, Fabrication, Spring, Vibration, Deflection, Stiffness*

Abbreviations:
SDOF: Single-Degree-of-Freedom
HIS: Human-Structure Interaction
MDOF: Multi-Degree-of-Freedom
FEM: Finite Element Method

I. INTRODUCTION

The spring mass experiment is used worldwide in all laboratory courses because it is both physically and pedagogically rich, and its straightforward content enables students to bridge the gap between theory and practice, to program and master experimental techniques, and to analyse data statistically. In the course, spring-mass is one of the setups required in a sequence of six experiments, as follows: the pendulum, vertical oscillations with the spring-mass, spring- mass with two degrees of freedom, horizontal oscillations with two masses and three springs, vibrations in an elastic string, vibrations in a gas tube and the ripple tank.

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The idea is to study the single-degree-of-freedom spring-mass system, given its importance in various fields, such as electronics, accelerators, and quantum mechanics.

In this context, the mass and spring experiment is connected to the development towards frequency and stiffness. The course organisation is based on the idea of creating groups of approximately twenty students at a time in a laboratory, working together as a research group to study a specific topic. As such, the group head guides the experimental research project, presents the theory in a series of lectures, describes the test strategy, defines steps, and, very importantly, creates opportunities for discussions at crucial points. In the laboratory sessions, sub-groups of four students are helped by assistants. The spring mass system is a fundamental system that plays a vital role in mechanical vibration; hence, it is considered.

II. LITERATURE REVIEW

The study of the movement experienced by a mass suspended from the free end of a spring is a topic discussed in most introductory physics courses, from both theoretical and experimental outlooks. The physics of the spring-mass system oscillations has been widely studied in a great variety of texts, in which the relationship between the period and the oscillation frequency is shown in detail [1]. Similarly, experiments allow for observing the dependence of oscillatory systems on mechanical forces, as established by Hooke's law [2]. Some studies included corrections to consider the influence of the spring mass on the oscillations of the spring-mass system.

The spring-mass experiment is a step in a sequence of six increasingly complex practicals, from oscillations to waves, which cover the first-year laboratory program. Both free and forced oscillations are investigated. The spring-mass is subsequently loaded with a disk to introduce friction. The succession of steps is: determination of the spring constant both by Hooke's law and frequency-mass oscillation law, damping time measurement, resonance and phase curve plots. All cross-checks among quantities and laws are done by applying compatibility rules. The non-negligible mass of the spring and the peculiar physical and teaching problems introduced by friction are discussed. The intriguing waveforms generated in the motions are analysed. We outline the procedures used throughout the experiment to enhance students' ability to handle equipment, select appropriate apparatus, critically apply theory, and practice treating data with statistical methods [1].

Several mass–spring–damper models have been developed to study the response of the human body to the collision with the ground during hopping, trotting, or running. The mass, spring, and damper elements represent the masses, stiffness properties, and damping properties of hard and soft tissues. The masses that

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models are composed of are connected via springs and dampers. This paper reviews various types of mass–spring–damper models, including one-body and multi-body models. The models are further categorised as being either passive or active. In passive models, the mechanical properties (stiffness and damping) of soft tissues remain constant regardless of the type of footwear, ground stiffness, etc. In active models, the mechanical properties adapt to external loads. The governing equations of motion of all models, as well as their parameters, are presented. The specific way the models account for shoe–ground interactions is also discussed. The methods used for the determination of different modelling parameters are briefly surveyed—the advantages and disadvantages of the various types of mass–spring–damper models are also discussed. The paper concludes with a brief discussion of possible future research trends in mass–spring–damper modelling [2].

The exact natural frequencies, mode shapes, and the corresponding orthogonality relations are essential in forced vibration analysis via modal expansion. In the present paper, a free vibration analysis is conducted to determine the exact natural frequencies and mode shapes of an axially loaded beam carrying several absorbers. An explicit expression is presented for the generalised orthogonality relations. These generalized orthogonality conditions are employed along with the assumed modes method to perform forced vibration analysis. The present approach is compared to other approximate methods in the literature with the classical orthogonality relations and a different choice of mode shapes. Results indicate that the use of the generalised orthogonality relation with the exact mode shapes is required for a precise investigation of the dynamic response of a beam with a mass-spring-mass-damper system [3].

This paper presents a deformable model that offers control over the isotropy or anisotropy of an elastic material, independently of how the object is tiled into volume elements. The new model is as easy to implement and almost as efficient as mass-spring systems, from which it is derived. In addition to controlled anisotropy, it contrasts with those systems in its ability to model constant volume deformations. We illustrate the new model by animating objects tiled with tetrahedral and hexahedral meshes [4].

The frequency-response curve is crucial information for structural design; however, the conventional time-history method for obtaining the frequency-response curve of a multi-degree-of-freedom (MDOF) system is time-consuming. Thus, this paper presents an efficient technique for determining the forced vibration response amplitudes of a multi-span beam carrying arbitrary concentrated elements. To this end, the "steady" response amplitudes |Y (x)|s of the above-mentioned MDOF system due to harmonic excitations (with the specified frequencies ωe) are determined by using the numerical assembly method (NAM). Next, the corresponding "total" response amplitudes |Y (x) |t of the same vibrating system are calculated by using a relationship between |Y (x) |t and |Y (x)|s obtained from the single-degree-of-freedom (SDOF) vibrating system. It is noted that, near resonance (i.e., ωe/ω ≈ 1.0), the entire MDOF system (with natural frequency ω) will vibrate synchronously in a particular mode and can be modelled by an SDOF system. Finally, the conventional finite element method (FEM) incorporated with the Newmark's direct integration method is also used to determine the "total" response amplitudes |Y (x) |t of the same forced vibrating system from the time histories of dynamic responses at each specified exciting frequency ωe. It has been found that the numerical results of the presented approach are in good agreement with those of FEM, this confirms the reliability of the given theory. Because the CPU time required by the presented approach is less than 1% of that needed for the conventional FEM, the presented approach is expected to be an efficient technique for the title problem [5].

This paper deals with some dynamical behaviour of a Damped non-homogeneous mass-spring system. Both ordinary differential equations and discrete fractional-order equations are considered. Stability at equilibrium positions is discussed with the analysis of the Jacobian matrix. Additionally, the results are illustrated with numerical examples that incorporate suitable parameters, showcasing the rich dynamical behaviour [6].

The interaction between humans and structures can produce significant dynamic effects. This has been demonstrated on several occasions, including the closure of the Millennium Bridge in London shortly after it was opened to traffic. The scientific community has widely accepted models based on springs, dampers, and lumped masses as a means to model human-structure interaction (HSI) problems. Recently, models of the human body based on control theory have been proposed. This paper provides a comparison between two traditional models, which utilise springs, dampers, and lumped masses, and those employing control theory. The models are updated probabilistically using Bayesian inference. The experimental data used for the comparison were obtained from a laboratory test structure specially designed for HSI studies [7].

Most applications today require dynamic systems modelling and their control. Natural systems that exhibit dynamic behaviour are typically modelled as a mass-spring system, and we can further enhance the system's flexibility by adding a damper. Currently, most systems have multiple Mass Spring Dampers, such as robot manipulators and Vehicle suspension systems, which utilise a Double Spring-Mass-Damper system in a planner and vertical configuration, respectively. In this paper, we will first model a general Double Spring-Mass-Damper System. Then, we will analyse it using different methodologies and control it using various continuous and digital control techniques. A Graphical User Interface is introduced that models, controls, and analyses the system using multiple control techniques, including Closed Loop, P, PI, PID, State Space, Bode Plot, Nyquist, Root Locus, and stability analysis, with applications in both digital and analogue modes [8].

System Dynamics is a well-formulated methodology for analysing the components of a system, including cause-effect relationships, their underlying mathematics and logic, time delays, and feedback loops. It originated in the business and manufacturing sectors, but is now impacting education and numerous other disciplines. Inspired by successful policy changes in

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various fields, system dynamics researchers also targeted the application of the system dynamics approach in the educational field [9].

This paper considers an inverse mixed eigenvalue problem for a damped spring-mass system. The issue of reconstructing the vibration system from its partial physical parameters, including the complex mode and real mode of the corresponding undamped system, is considered. By solving a kind of quadratic inverse eigenvalue problem, the necessary and sufficient condition for the solvability of the problem is obtained, and a numerical method for solving the problem is proposed. The numerical example shows the feasibility of the process [10].

III. DESIGN OF SPRING

A spring is an elastic object that stores mechanical energy. Springs are typically made of spring steel. There are many spring designs. In everyday use, the term often refers to coil springs. They can return to their original shape when the force is released and are defended as an elastic body.

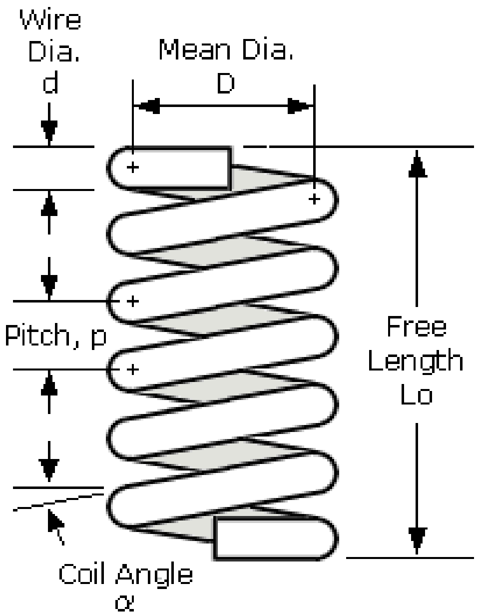

Fig.1: Nomenclature of Spring

A. Design Procedure for Helical Spring

i. Nomenclature of Helical Spring

c = Spring Index d = wire diameter D = Mean diameter of coil D_i = Inner diameter of coil D_o=Outer diameter of coil E = Young's Modulus F = Axial Force G = Modulus of Rigidity k = Wahl Factor L_0=Free Length L_s =Solid Length i'= Total number of coils i = Number of active coils p = pitch y = deflection τ = shear stress τ max = Max shear stress n= additional coils a= clearance

- **Diameter of Wire Shear Stress $\tau = \frac{8FDK}{\pi D_3}$**

Stress factor K $=\frac{4C-1}{4C-4}+\frac{0.615}{C}$ EQ.20.23 [11]

Spring index C $=\frac{D}{d}$

- **Mean Diameter of the Coil**

  Inside diameter of the coil Di = (D- d)

  Outside diameter of the coil Do (D +d)

- **Number of Active Turns or Coils**

  Mean deflection by

  $\frac{8fD^3i}{GD^4}$ EQ. 20.29 [12]

- **Free Length of the Spring**

l_0= (i+ n) d + y +a E q .20.53 [12]

Where a is the clearance between working coils under max load, take a as 25% of max deflection

- **Pitch**

p = $\frac{l_0-2d}{i}$ table 20.14 [12]

- **Stiffness or Rate of spring**

$F_{0=}\frac{F}{Y}$ E q 20.30 [12]

B. Illustration

Design a helical spring to support an axial load of 3000N, the deflection under the load being limited to 60 mm. The spring index is 6. The spring is to be made of chrome vanadium steel & a factor of safety is 2.

i. Given Data:

F=3000N, y=60 mm, c=6, FOS=2, where τe is the elastic limit in torsion for chrome-vanadium steel (Column Nos. 9 & 10). Elastic limit in torsion τe =690 MPa. Rigidity modulus G=78450 MPa. Applying a Factor of safety of 2, the allowable shear stress τ = 345 MPa

- **Diameter of wire shear stress $\tau = \frac{8FDK}{\pi d^3}$ Where c is the (C=$\frac{D}{d}$)**

K Wahl 's Stress factor K $=\frac{4C-1}{4C-4}+\frac{0.615}{C} = 1.2525$

$345=\frac{8\ X\ 3000\ X\ 6\ X\ K}{\pi d^2}$ = d = 12.81mm ≈ 13 $mm.$

- **Mean Diameter of the Coil**

1. D= cd = 6 x 13 =78

   Inside diameter of the coil Di = (D-d) = (78-130=65mm.

   Outside diameter of the coil Do= (D +d) =(78+ 13) =91mm.

- **Number of Active Turns**

Deflection = $\frac{8FD^3i}{Gd^4}=\frac{8\ X\ 3000\ X\ 78^3\ X\ i}{78450X\ 13^4}$ i=11.8≈ 12$mm.$

- **Free Length of the Spring**

  For Squared & Ground Ends, Length $l_{0=}$ (i+2) d +y + a

Where a ≈ clearance is required between working coils under maximum load. Take a 25% of maximum deflection = 25 % of 60 = 15mm

l_O ≥ (12 + 2)13 +60+15=257 mm

- **Pitch**

p = $\frac{l_0-2d}{i}=\frac{257-2x13}{12}$ = 19.25mm

- **Stiffness or Rate of Spring**

$F_{0=}\frac{F}{Y}=\frac{300}{60}$ = 60 N/m

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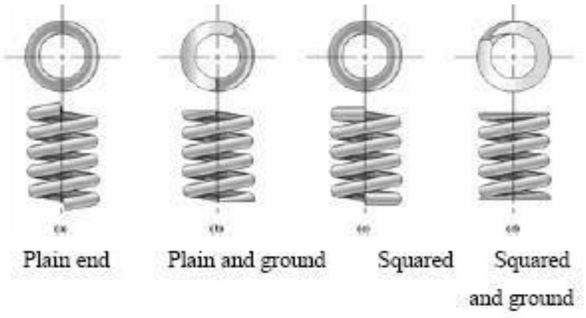

[Fig.2: Types of Coil Ends]

C. Specification

1. Material = chrome, vanadium
2. Wire diameter d=13mm
3. Mean coil diameter D=78mm
4. Free length lo = 257mm
5. Total number of turns I = 14
6. Style of ends: Squared & ground
7. Pitch =19.25mm
8. Spring rate=50N/m

IV. FABRICATION METHODOLOGY

Springs are generally made from Mild Steel because of its versatility, availability, and low cost; therefore, Mild Steel was selected for the fabrication of the spring apparatus. The mild steel was marked using a scriber and then cut to the required dimensions of 751 mm by 356 mm using a Guillotine cutting machine. The sharp end was bent backwards 10 mm to avoid piercing the operator's hand. At the centre of the plate, a rectangular slot of 610 mm by 90mm was drilled to accommodate the load hanger. When loads are placed on the hanger to initiate displacement, one steel meter rule is attached to the side of the rectangular slot cut through the centre of the apparatus to measure the displacement of the spring upon load application. The vertical rectangular member of the apparatus containing the slot is screwed to a mild steel base of dimensions 360mm by 300mm, and the base sits on four rubber bushings to provide the apparatus with a firm stand and support.

[Fig. 3 Experimental Setup]

[Fig.4: Masses for Conduction of Experimental]

Figures 3& 4 represent the experimental setup and masses taken into consideration for the conduct of the Experiment.

Table-I: Design Analysis

Part name	Input	Analysis	Decision
Weight of Apparatus	Mass, m= 6.6kg	W = mg 6.3×9.81	W = 61.8 N
Cross-sectional area of the rectangular body	Height, h_1=750 Base, b1 350	$A_1 = h_1 \times b_1$ 750 × 350	A_1= 262,500mm^2
A_2 = Area where metal is cut off	Height, h_2=600 Base, b_2 = 90	$A_2 = h_2 \times b_2$ 600×90	A_2= 54,000mm^2
Total Area	A= A_1 - A_2	A = 262,500 - 54,000	A=208,500mm^2

V. RESULT

The following results were obtained from a test carried out on four springs of different dimensions, as presented in Table 5 below.

A. Table of Values

Specimen dimension: Spring initial height h_1 = 19.94 mm, Wire diameter (d) = 0.73 mm, Coil diameter (D) = 16.03 mm and Number of coils (n) = 28.

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Table-II: Results Obtained from the Spring Sample A

S.NO	Mass (g)	Force (N)	Initial height h_1 (mm)	New Height h_2 (mm)	Extension $(h_2–h_1)$ (mm)	Shear stress [τ] N/m^2	Angle of Twist [θ]	Spring stiffness [s]	Sprig Deflection [δ]
1	100	981	19.9	67	47.1	2344	1849658.2	6.7×10^{-8}	1.4×10^{7}
2	200	1962	19.9	102	82.1	4688	3715453.3	6.9×10^{-8}	2.9×10^{7}
3	300	2943	19.9	139	1191	7032	5545004.3	7.5×10^{-8}	4.4×10^{7}
4	400	3924	19.9	177	157.1	9376	7430259.6	7.9×10^{-8}	5.9×10^{7}
5	500	4905	19.9	206	186.1	11720	9287824.5	8.5×10^{-8}	7.4×10^{7}
6	600	5886	19.9	247	227.1	14064	11145389.4	8.9×10^{-8}	8.9×10^{8}
7	700	6867	19.9	287	267.1	16409	13005039.7	9.2×10^{-8}	1.0×10^{8}

Specimen B dimensions:

Springinitialheighth1 = 84.20 mm, Wire diameter(d) = 0.75mm, Coil diameter (D) = 9.36 mm Number of coil (n)=120

Table-III: Results Obtained from the Spring Sample B

SNO	Mass (g)	Force (N)	Initial height h_1 (mm)	New Height h_2 (mm)	Extension $(h_2–h_1)$ (mm)	Shear stress [τ] N/m2	Angle of Twist [θ]	Spring stiffness [s]	Spring Deflection δ
1	100	981	84.2	86	1.8	2229.8	31772.8	5.0×10-8	1.9×107
2	200	1962	84.2	95	10.8	4459.1	63545.5	2.5×10-8	3.9×107
3	300	2943	84.2	147	62.8	6688.6	95300.3	1.6×10-8	5.9×107
4	400	3924	84.2	180	95.8	8918.2	127067	1.2×10-8	7.9×107
5	500	4905	84.2	210	125.8	11147.7	158833.5	1.0×10-8	9.9×107
6	600	5886	84.2	241	156.8	13377.3	190600.5	8.3×10-9	1.1×108
7	700	6867	84.2	273	188.8	15606.8	222367.3	7.1×10-9	1.3×108

Specimen C dimension;

Spring initial height h_1 = 95 mm, Wire diameter(d) = 0.50 mm, Coil diameter (D) = 6.85 mm Number of coil (N) =180

Table-IV: Results Obtained from the Spring Sample-C

SNO	Mass (g)	Force (N)	Initial height h_1 (mm)	New Height h_2 (mm)	Extension $(h_2–h_1)$ (mm)	Shear stress [τ] N/m2	Angle of Twist [θ]	Spring stiffness [s]	Spring Deflection δ
1	100	981	87.9	108	19.1	4618.6	11036778.80	0.310-0	1.8×107
2	200	1962	87.9	141	54.1	9237.3	22073557.59	1.5×10-8	3.7×107
3	300	2943	87.9	177	89.1	13855.9	33110336.39	3.0×10-8	5.5×107
4	400	3924	87.9	211	123.1	18474.6	44147115.1	7.9×10-9	7.4×107
5	500	4905	87.9	246	158.1	23093.2	39055570.2	6.3×10-9	9.3×107
6	600	5886	87.9	281	193.1	27711.9	46866684.3	5.3×10-9	1.1×108
7	700	6867	87.9	314	226.1	32330.5	54677798.3	7.6×10-9	1.3×108

Specimen D dimension

Spring initial height h_1 = 87.94 mm, Wire diameter (d) = 0.52 mm, Coil diameter (D)= 5.67 mm Number of coil (N) = 171

Table-V: Results Obtained from the Spring Sample-D

SNO	Mass (g)	Force (N)	Initial height h_1 (mm)	New Height h_2 (mm)	Extension $(h_2–h_1)$ (mm)	Shear stress [τ] N/m2	Angle of Twist [θ]	Spring stiffness [s]	Spring Deflection δ
1	100	981	95.0	149	54	1070.6	61650	9675.5	211151.1
2	200	1962	95.0	210	115	9994.9	123300	19351.1	422302.5
3	300	2943	95.0	273	178	14992.4	184950	29026.2	633453.8
4	400	3924	95.0	337	242	19989.8	246600	38702.1	844605.0
5	500	4905	95.0	400	305	24987.3	308250	48377.7	1055756.3
6	600	5886	95.0	469	374	29984.7	369900	58053.3	1266907.5
7	700	6867	95.0	523	428	34982.2	431550	67728.7	1478058.8

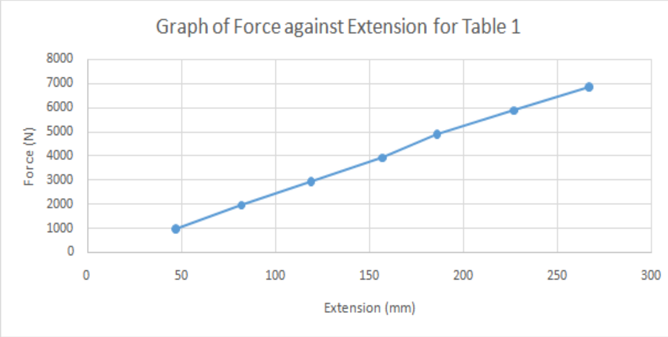

Fig.5: Graph of Force Against Extension for Table 1

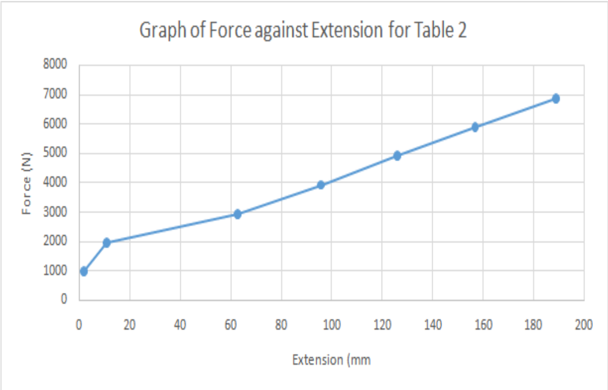

Fig.6: Graph of Force Against Extension for Table 2

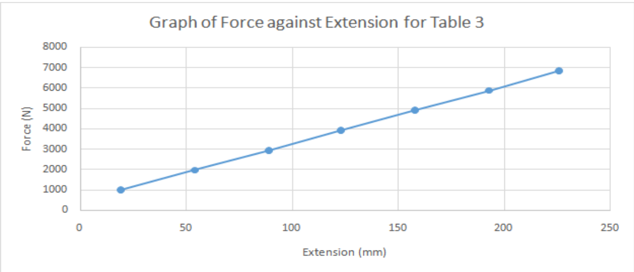

Fig.7: Graph of Force Against Extension for Table 3

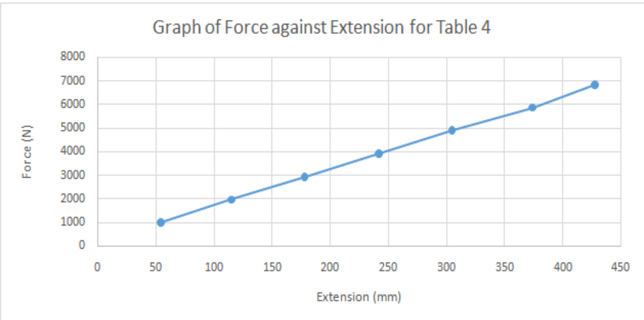

Fig.8: Graph of Force Against Extension for Table 4

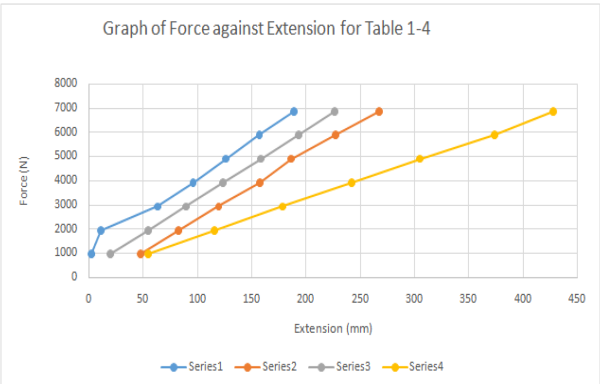

Fig.9: Graph of Force Against Extension for Table 1- 4

VI. DISCUSSION

The spring apparatus was tested with different spring samples. The results were presented in Tables 5.1- 5.4 above. From the test trials on each spring, an increment in load from 100- 700g produced an appreciable increase in length for each of the springs tested. The spring with the highest wire diameter of 0.75mm offers the highest resistance (spring stiffness) to applied load, as opposed to the spring with the lower wire diameters of 0.50mm, 0.52mm and 0.73mm, respectively. This agrees with one of the principles of spring design, which states that „the heavier the wire, the stronger the spring‟.

The computed values of shear stresses for the four springs tested using the apparatus indicated that shear stress is a function of the ratio of applied load to area; therefore, it was observed that the higher the wire diameter (d), the lower the stress acting on the spring. At maximum load of 700g for the sample springs of wire diameters ranging between 0.50 mm and 0.75mm, the corresponding values obtained for shear stresses also range between 32330.51N/m2 and 15606.81N/m2. From these values, it is clear that the spring with the minimum or least value of wire diameter has the highest shear stress value, i.e., the value of shear stress decreases as the wire diameter increases. This clearly shows that the wire diameter affects the value of shear stress in springs.

Similarly, the computed values for an angle of twist for the four springs tested depend on the axial load, mean radius, number of coils and wire diameter. Since an incremental force applied on the four springs was of the same magnitude as an infinitesimal difference in wire diameter as well as mean diameter, the variations in the value of angle of twist is therefore as a result of the significant difference in the number of coils which is n= 28 for sample A, n=120 for sample B, n= 180 for sample C and n= 171 for sample D respectively. Also, the angle of twist provides us with an insight or gives us an idea of the expected value of deflection, by multiplying the value obtained for the angle of twist by the mean radius (Deflection = mean radius x angle of twist).

The computed value of spring deflection for the four springs tested using the apparatus indicates that, since deflection is a function of spring distortion, all the springs were subjected to the same load value, at a maximum load of 700g for the spring tested. It was noted that the spring with the wire diameter and coil diameter of 0.52mm and 5.67mm has the highest deflection (spring travel) to load applied as opposed to the other springs of diameters 0.50mm, 6.85mm, 0.52mm, 5.67mm, and 0.73mm, 16.03mm respectively.

VII. APPLICATION

This apparatus is used to measure the stiffness of coil springs with different diameters. It can also be used by students in the strength of materials or materials testing lab to perform simple tests on springs, thereby deepening their knowledge of the course. Spring manufacturing companies can also use it to determine the strength of their springs. With this apparatus, a comparison can be made of the stiffness of the test spring with that of the standard one.

VIII. CONCLUSION

The most significant performance characteristic of spring is its stiffness, in any area of engineering where spring application is necessary. The results analyzed in this research showed a trend which agrees with the principle of spring design, drawing inferences from tests run with the following spring diameters: 0.75, 0.73, 0.70 and 0.65, respectively. The larger the wire spring diameter, the stronger the spring. From the results of the various test runs using this apparatus, it is seen that to design a spring for any application, for that matter, no assumptions should be made to either underestimate or overestimate this critical characteristic of spring stiffness before selecting it for any typical use.

DECLARATION STATEMENT

After aggregating input from all authors, I must verify the accuracy of the following information as the article's author.

- **Conflicts of Interest/ Competing Interests:** Based on my understanding, this article has no conflicts of interest.
- **Funding Support:** This article has not been funded by any organizations or agencies. This independence ensures that the research is conducted with objectivity and without any external influence.
- **Ethical Approval and Consent to Participate:** The content of this article does not necessitate ethical approval or consent to participate with supporting documentation.
- **Data Access Statement and Material Availability:** The adequate resources of this article are publicly accessible.
- **Author's Contributions:** The authorship of this article is contributed equally to all participating individuals.

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