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Motion Analysis of Torsional Testing Machine to Reduce Impact Load Using SOLIDWORK

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Abstract: Mechanical testing is a standard and essential part of any design and manufacturing process, for ensuring safe working of mechanical component and for ensuring a cost-effective design. Torsion Testing Machine is designed for conducting Torsion and Twist on various metal wires, tubes, sheet materials, torque measurement, in this torque can be applied to testing specimen by geared motor through gear box. But the main difficulty with analytical torsion testing machine is that after test is complete and specimen breaks the trolley on which the test specimen is clamped it can impacted heavily to the rubber stopper mounted on the guide ways of the machine which distort the machine assembly. In this study motion study is done, using SOLIDWORK software by introducing spring at the place of stopper and try to minimise this impact load of the trolley on the machine.
Keywords: Torsion testing, Motion analysis, impact load, spring design, SOLIDWORK software

I. INTRODUCTION

The shear stress-strain response of materials can be extremely important in the design, analysis and manufacture of a wide variety of products and components which are loaded primarily in shear or torsion. These properties are determined from the shear stress-strain diagram which is most commonly measured according to an ASTM torsion test where a material specimen of solid or hollow round cross section is twisted in a torsion testing machine as the applied torque and angle of twist are recorded simultaneously [1].

Torsional testing machine is used for conducting tests in torsion for metal and other materials. The rotating motion is obtained through a speed reduction gear box driven by electric motor. Analytical torsion testing machine is equipped with pendulum dynamometer, a recording device for registering torque. The bar is clamped between two chucks one is driving chuck and other is driven chuck. The drive is given from motor which is connected to speed reduction box and the output shaft, which carries the angle measuring disc and the driving chuck at the end. The driven chuck is fixed on a shaft supported in bearing rigidly clamped to measuring unit. The torque is indicated on dial, a recording instrument automatically registers toque twist diagram [2]. Torsion occurs when any shaft is subjected to a torque. The torque makes the shaft twist and one end rotates relative to the other inducing shear stress on any cross section. Failure occur due to shear alone or because the shear is accompanied by stretching or bending. So in this whole testing one end of shaft fixed and other end is twisted due to the action of torque T. The radius of shaft is R and the length is L. When the end is twisted, the line rotates through an angle θ. G is one of the elastic constants of the material. The equation is only true so long as the material remains elastic.

$$T/J = G\theta/L$$

A Motion Analysis study combines motion study elements with mates in motion calculations. Consequently motion constraints, material properties, mass, and component contact are included in the SOLIDWORKS Motion kinematic solver calculations. A Motion Analysis study also calculates loads that can be used to define load cases for structural analyses. To use the SOLIDWORKS Motion solver in a motion study, select Motion Analysis from the motion studies type list in the Motion Manager [3].

II. PROBLEM DEFINITION

After fracturing the specimen on analytical torsional testing machine, trolley comes back with high speed and impacted on the rubber stopper because of this the whole system gets disturbed. To avoid this, introducing spring by replacing rubber stopper at the end of guide ways.

III.OBJECTIVES

1) First to calculate the impact loading of trolley on the rubber stopper by simulation using Solid works software.
2) From the data obtained, design the spring to reduce the effect of impact loading.
3) Further by using the same software and introducing spring in design the effect impact loading of trolley is observed.

IV. PROCEDURE

First of all calculating the total weight of trolley and tension in the string for that, data obtained by taking measurements from analytical torsional testing machine,

m1 = Mass of total assembly of trolley (Motor, Gearing system, Frame etc.)

m2 = Mass of Weight

The time required by trolley to move 30cm distance from rest is 1.1sec on the platform.

m2 = 1.5 kg, s = 30 cm =0.3 m, t = 1 .1 sec

From above data we find the total mass of assembly m1, acceleration (a) and tension in the string (T)

$s = ut + ½(at^2)$

$0.3 = 0 + ½(a * 1.1^2)$ ………… (u=0)

$a = 0.42\ m/s^2$

Acceleration of assembly (a) = $0.42\ m/s^2$

Therefore mass of the assembly (m1)

a = (m2*g) / (m2 + m1)

0.42 = (1.5*9.81) / (1.5 + m1)

m1 = 35 kg

Tension in the string (T)

T = m1*a

= 35 * 0.42

T = 15 N

From above data we simulate the model and we find the contact force on the rubber stopper when the assembly at the extreme position on the platform

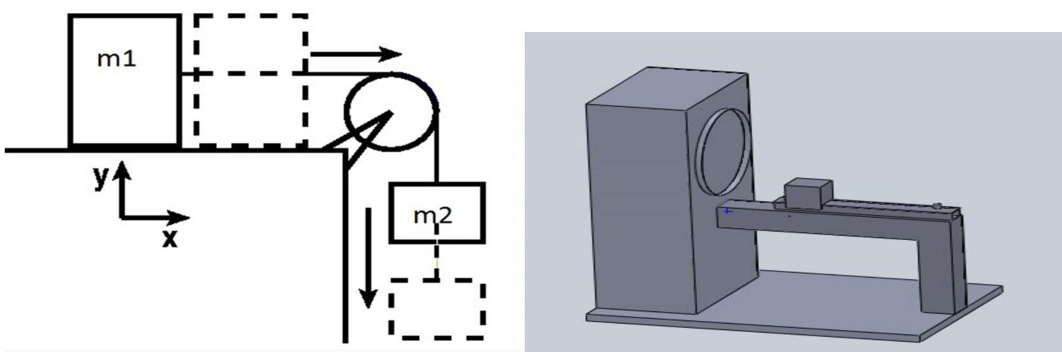

Fig.1 Designed model of torsion testing machine

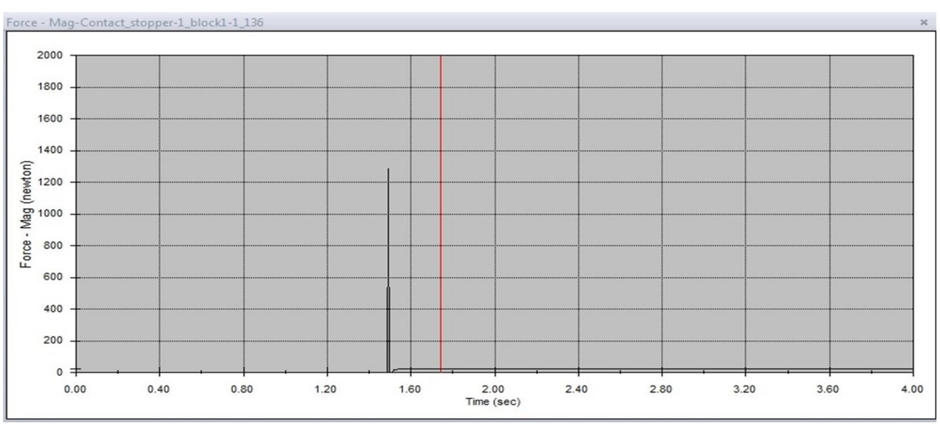

Fig. 2 Graph of force vs. time obtained by impact loading of trolley on the rubber stopper

Above graph shows the impact load between the assembly and the rubber stopper it is 1.3 KN. From that we have to design the spring to reduce the vibration, disturbance and noise produced during the impact.

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A. Design Of spring

1) Selection Of Material

TABLE NO.1
TYPES OF MATERIAL USED FOR SPRING

Material Of Springs	Specific Use
1)Patented And Cold Drawn Steel wire	Used when spring subjected to static forces and moderate fluctuating forces
2)Oil hardened and tempered spring wire	Applications involving fluctuating forces, it is used for application where the stresses are severe
3)Oil hardened and tempered steel wires(alloyed)	Used for applications involving highly stressed springs subjected to shock or impact loading

From above chart we select the 3rd type of material because we have to design the spring for impact loading. There are two popular varieties of it chromium vanadium steel and chromium silicon steel Mechanical Properties of oil hardened and tempered spring steel

TABLE NO.2
WIRE DIAMETER AND ITS MINIMUM TENSILE STRENGTH

Wire diameter(mm)	Minimum Tensile strength(N/mm2)
3.0	1520
4.0	1480
4.5	1440
5.0	1440
6.0	1400
7.0	1360
8.0	1290

2) Selection of Spring Index(C)

When the spring index is low(C<3) the actual stresses in the wire are excessive due to curvature effect. Such a spring is difficult to manufacture and special care in coiling is required to avoid cracking in some wire. When the spring index is high(C>15),it is large variation in coil diameter. Such spring is prone to buckling and also tangles easily during handling. Spring index from 4 to 12 is considered better from manufacturing considerations.

We select for our problem spring index is 6.

By trial and error method for selection of spring diameter and tensile strength

Maximum impact force is P=1.1kN

$$\text{Therefore shear stress}(T) = \frac{(8PC)}{(3.14*d^2)}$$

For P = 1100 N And C = 6

$T = 16806.7/d^2$ And $Td = 0.3*Sut$

Diameter Of wire(d)mm	Shear Stress(T) (N/mm^2)	Sut(N/mm^2)	Permissible Shear Stress(Td)(N/mm^2)	Comparison between T and Td
5	672.68	1440	432	T>Td
6	466.85	1400	420	T>Td
7	342.99	1360	408	T<Td

When T<Td the design is satisfactory and the wire diameter should be 7mm
Then coil diameter(D) = C * d = 6 * 7

D = 42mm

We assume the number of coils (N) = 6
The Spring has Square end then total no. of coils(Nt) = N + 2 = 8

$$\text{Then the deflection in the Spring} = \frac{(8PD^3N)}{Gd^4}$$

G =Modulus of rigidity = 81370 , P = 1100N , D = 42mm , N = 6, d = 7mm putting all these values in above equation

$$\text{Deflection in the spring} = \frac{(8*1100*42^3*6)}{(81370*7^4)}$$

Deflection in the spring = 20mm
Solid length of spring = Nt * d = 8 * 7
= 56mm

Free length of Spring = Solid Length + Total axial gap + deflection
= 56 + 7 + 20 =82mm
The required spring rate is given by
k = P/deflection
= 1100/20 = 53.14 N/mm

The actual Spring Rate is Given by

$$K = \frac{(Gd^4)}{(8D^3N)}$$

$$= \frac{(81370*7^4)}{(8*42^3*6)} = 60 \text{ N/mm}$$

By using the above calculation design of the spring in the SOLIDWORKS and introduce the spring in the previous designed model of analytical torsional testing machine by using the plunger type arrangement.

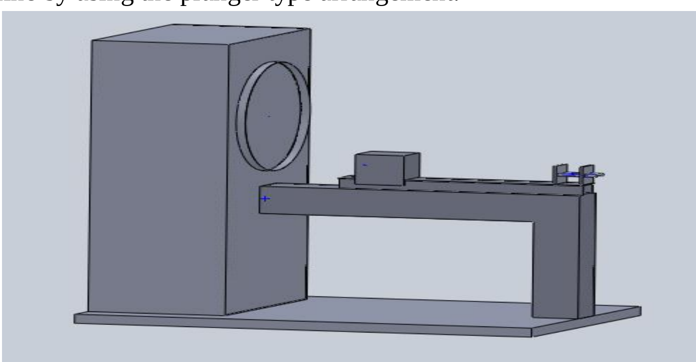

Fig.3 Model after introducing the spring and plunger arrangement

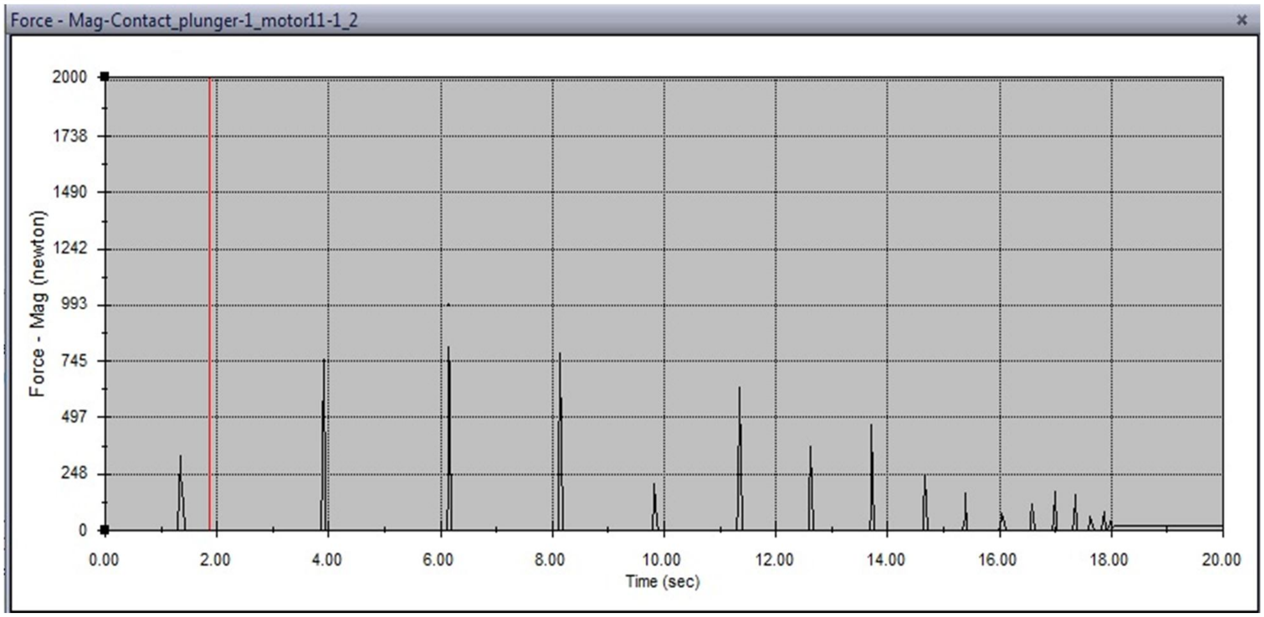

Fig. 4 Graph force vs. time of impact loading of trolley on the spring

V. CONCLUSION

Because of introduction of spring at the end reduction in impact loading from 1.3 KN to 0.745 KN and nearly about 50% impact loading is reduced. But because of introduction of spring large time nearly about 16 seconds required by the trolley to settle down to the rest position.

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