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# Numerical Analysis and Optimization of Jumping Robot

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***Abstract***

*This paper presents a human jumping mechanism inspired design of a jumping mini robot including the theoretical analysis on jumping dynamics. In the present study, we would focus on optimizing the spring design to attain a theoretical jump height of about 30 cm. The analysis involves spring type, material and specifications selection including fatigue. Also, the currently employed system for jumping robots takes plenty of time to load the springs and to execute the jump. The time could be reduced by optimally selecting a multithreaded power screw in accordance to the raising load and torque needed. Improvement and analysis of existing mechanical systems using meticulous calculations and design theory is extremely important process in today's world. This not only improves the performance but also saves energy and scarce resources. In this paper, we would dwell into optimization and analysis of the jumping mechanism of an indigenously developed. Stress analysis of various parts of robot is also done.*

**Keywords:** Jumping robot, ABAQUS, spring, CAD, Matlab, design

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## INTRODUCTION

Design is an innovative and highly iterative process. It is also a decision-making process. Decisions sometimes have to be made with too little information, occasionally with just the right amount of information, or with an excess of partially contradictory information. Improvement and analysis of existing Mechanical systems using meticulous calculations and design theory is extremely important process in today's world. This not only improves performance but also saves energy and scarce resources. M. Wang *et al.* designed the jumping mechanism inspired from frog.[1] In some past studies, researchers have tried to develop miniature robot which can work in multi degree of freedom.[2]

In the present paper, we would dwell into optimization and analysis of the jumping mechanism of an indigenously developed. This autonomous robot is shown in Figure 1. It follows a particular direction as given by the user and avoids all obstacles that come in its route and to also jump when instructed to do so.

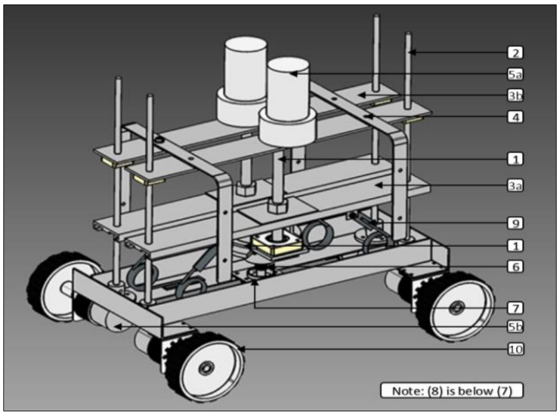



***Fig. 1:*** *CAD Model of Jumping Robot.*

The present mechanical design as shown in Figure 1 comprises of following components:

1. Power Screw Rods: These rods are powered with motors for the up and down motion of the horizontal plates which is connected with these lead screw rods.
2. Vertical Guiding Rods: Mild steel rods are used which will guide the vertical motion of the horizontal plates. It helps to stabilize the system.
3. Horizontal Plates: This design includes two pair of horizontal plates. These plates provide the up and down motion and thus, resulting in the compression of spring.
4. Connecting Plates: These plates ensure that the level of both the horizontal plates remains same.
5. Motors: Six motors are mounted on the top pair of horizontal plates and on the bottom pair of plates. Accordingly, these motors perform two different functions:
   (a) Providing vertical up and down motion to horizontal plates through lead screws (60 rpm).
   (b) Driving the wheels (100 rpm).
6. Latch: It is attached with the lower pair of horizontal plates and it latches with the base as the top plate moves downwards.
7. Conical Structure: This structure ensures that the latch goes vertically inside the hole and gets latched with the base.
8. Limit Switch: It deals with changing direction of motor after the latching takes place.
9. Torsional Spring: Four Torsional springs have been used in this design for providing the required potential energy for jumping.
10. Wheels: To drive the robot.

The design was inspired from human jumping mechanism through knees. The source of energy is from moments applied by human’s knee muscle shown in Figure 2. When human has to jump, the legs are bending at suitable angles, and then sudden moments are applied by knee muscles causing reaction forces from ground to lift up.

The various stages are explained in Figure 2 as:

**Stage 1:** Human starts to bend his knees to suitable angles as per jump requirement.

**Stage 2:** The knee suddenly applies angular impulse, leading to acquiring some velocity by upper body.

**Stage 3:** The body further moves up. The reaction from ground just vanishes as the knees straighten.

**Stage 4:** The body rises up till all the gained kinetic energy gets converted back to potential energy. Instantaneously, the body is at rest and then again starts coming back to ground due to gravity.

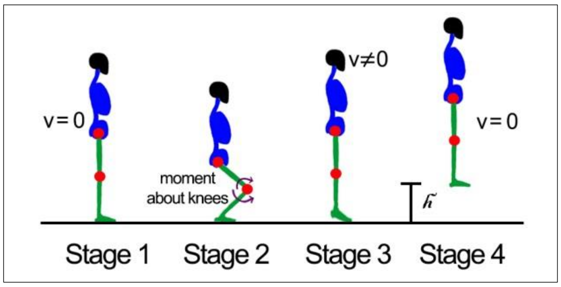



***Fig. 2:*** *Human Jumping Mechanism.*

Driving a mechanical system through the above explained situation goes as follows. In this mechanical design, two high torque motors are attached to two vertical lead screw rods. The motors move these rods up and down which in turn move the horizontal plates. The latch is attached with the middle pair of horizontal plate. The latch moves vertically downwards as the plates moves downwards. Conical structure ensures that the latch goes vertically inside the hole in the base. The latch locks as it passes through the hole on the conical structure. Once latch enters the hole, it presses a limit switch. After limit switch is pressed, it sends a signal to Arduino (a microcontroller) and gives command which reverses the direction of motors. Thus, now the compression of all the four torsional springs starts. Thus this can be compared to the situation where an external force compresses the springs by ‘x’. When springs are fully compressed, the latch de-latches with the help of brackets attached in middle horizontal plates. As soon as, the latch de-latches, the spring retracts. The sudden expansion increases the normal reaction from ground drastically which gives a huge thrust on the robot leading to a jump. And this as we can realize is the situation where the external force is suddenly removed to provide an impulse which creates a finite height ‘h’ jump.

## MATHEMATICAL MODEL OF THE SPRING-MASS SYSTEM

The system is modelled using lumped spring-mass system. It is a semi-definite system consisting of two different masses ($m_1$ and $m_2$) connected using a helical spring as shown in

When a force F is applied to a vertically positioned independent two mass system, which is connected with a spring of force constant k, a compression of x is achieved slowly. When it is released, both the masses start moving upward while constantly vibrating at natural frequency as the system reaches its highest point and strikes back the ground. The various stages are explained in Figure 3.

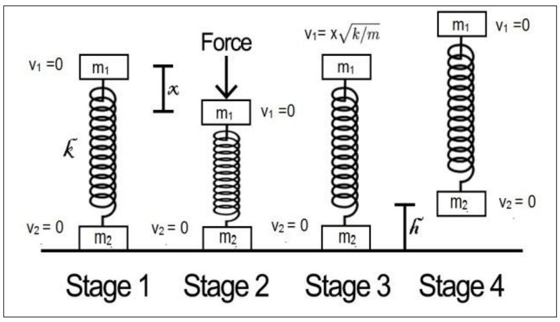



***Fig. 3:** Spring-Mass Model.*

### Governing Equations

The governing equation can be simplified and written in modified form.[3]

$$\begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix}\begin{bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \end{bmatrix} + \begin{bmatrix} k & -k \\ -k & k \end{bmatrix}\begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = \begin{bmatrix} -m_1 g \\ -m_2 g \end{bmatrix} \quad \text{Eq. (1)}$$

### Boundary Conditions

Mass $m_1$ is compressed to 45 mm and then released. Mass $m_1$ attains a velocity $v_1$ when mass $m_2$ is about to leave the ground. The boundary conditions of the given system can be found out at time when the mass $m_2$ just leave the ground. The displacement of $m_2$ and velocity of $m_2$ is zero at this time. The displacement of $m_1$ and velocity of $m_1$ can be found out using force balance and energy balance.

Writing force balance for $m_2$ at time when system is about to leave the ground.

$$\sum F_y = m_2 g - k x_{10} = 0 \quad \text{Eq. (2)}$$

$$x_{10} = \frac{m_2 g}{k} \quad \text{Eq. (3)}$$

With this $x_{10}$ displacement of $m_1$ from the mean position we can found out the velocity of $m_1$ at this point using energy balance.

$\sum$ Energy at point of maximum compression = $\sum$ Energy at point the $m_2$ is about to leave ground

$$\frac{1}{2} k x_0^2 = m_1 g (x_{10} + x_0) + \frac{1}{2} m_1 v_1^2 \quad \text{Eq. (4)}$$

where

$X_0$ is initial compression of the spring, $v_1$ is the velocity of $m_1$ at time when $m_2$ is about to leave ground.

$$v_1 = \sqrt{\frac{k x_0^2 - 2 m_1 g\left(\frac{m_2 g}{k} + x_0\right)}{m_1}} \quad \text{Eq. (5)}$$

So with these initial conditions we can find out the solution of the given differential equation. The governing equations with the initial conditions are solved using MATLAB code (Figures 4, 5).[3]

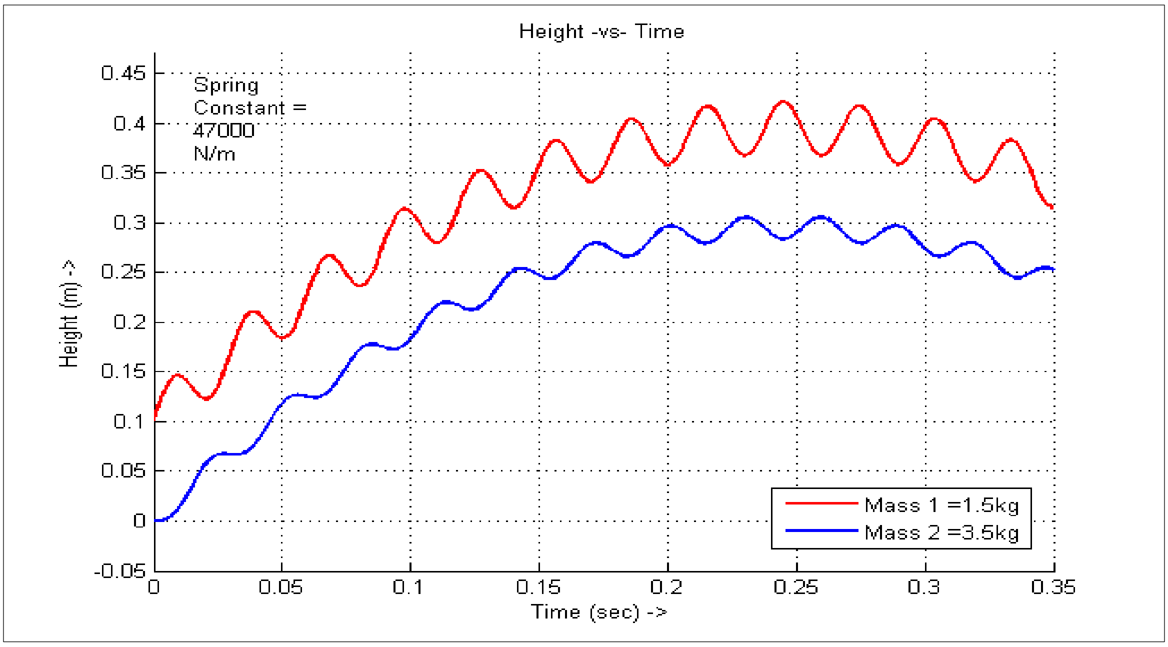



***Fig. 4:*** *Plot of Height Vs Time for $m_1$ = 1.5 kg, $m_2$ = 3.5 kg and k = 47000 N/m.*

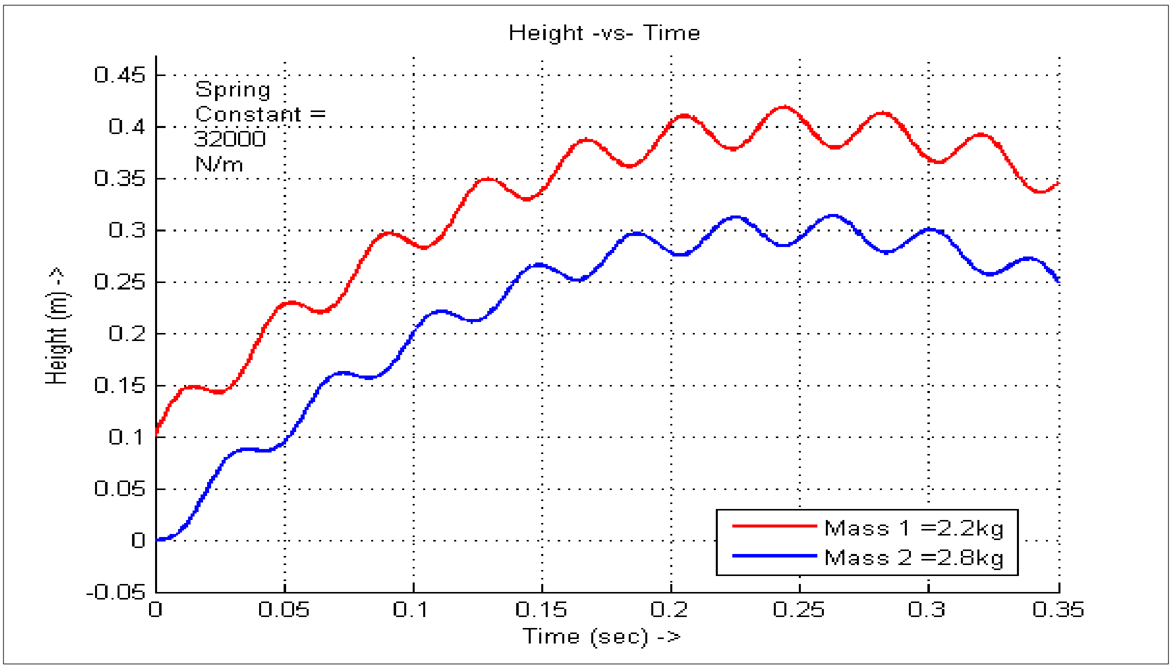



***Fig. 5:*** *Plot of Height Vs Time for $m_1$ = 2.2 kg, $m_2$ = 2.8 kg and k = 32000 N/m.*

**Determining the Spring Type and Constant**

The original system employs four torsional springs to accomplish the task of jumping. But the main disadvantage of torsion spring is that alongside a vertical component of force on the $m_1$, it also exerts a horizontal component due to its obtuse angle which does not contribute to the jumping action of the robot. The horizontal component of the force overall cancels due to the presence of even number of springs, but lead to wastage of force (energy) which could have been used to increase the jump of the system. Hence, we have proposed to replace the four torsional springs with four helical springs and present analysis based on helical springs in the following sections. Helical springs also fit our proposed mathematical model and are easily available in the market.

In this proposed mathematical model, we have combined the four springs into one and find an equivalent spring constant which is four times the spring constant of each spring.

The code requires $m_1$, $m_2$, k and initial compression of the springs as its input. As output it provides us the plot of displacements $x_1$ and $x_2$ *vs* t (time). To find the jump height of the system, we find the maxima of the $x_2$ vs. t plot before $x_2$ becomes zero, as this denotes the time when the lower mass strikes the ground back and our mathematical model becomes invalid beyond this time. Hence, we could iterate the value of k manually till we get the desired value of jump height.

Originally the system has mass distribution as $m_1 = 1.5$ kg and $m_2 = 3.5$ kg and after iteration the value of equivalent k was found out to be 47000 N/m corresponding to 30 cm jump as shown in Figure 6. But after observing the effect of parameters like mass distribution and initial compression on the value of k for the desired height, we noticed that we would be requiring a lower k if we increase $m_1$ and decrease $m_2$ while keeping their sum constant.

As the total mass of our bot was fixed to 5 kg, hence we decided to shift battery and electronics equal to 0.7 kg (maximum possible value) on the upper deck from the lower deck. So our new mass distribution becomes $m_1 = 2.2$ kg and $m_2 = 2.8$ kg and we get a value of spring constant k = 32000 N/m for 30 cm jump as shown in Figure 7. There is a straight decrease of 32% in the value of k.

Finally, the spring constant for each of the helical spring needed becomes:
$k_{each,final} = 8000\ N/m$

## FINALIZING THE SPRING SPECIFICATIONS

We iterated the MATLAB code[4] for 2 mass system and found out the spring constant for the spring should be 32000 N/m. Given system employs 4 spring attached in parallel to each other, so we have to design a spring for k = 8000 N/m. The present system is in such a way that the compression of spring is about 4.5 cm, so we can found out the maximum force at compression. The free length of the spring should be about 10 cm.

With these parameters known we can write the code for spring design. The code we have written takes the values of G (Shear modulus), A (constant) and m (exponent) for estimating minimum tensile strength of material, Force applied and respective displacement as input.

We have employed check on the values of C (spring index, $4 < C < 12$), Na (number of active turns, $3 < N_a < 15$), $n_s$ (safety factor, $n_s > 1.2$) $l_o$ (free length) and $l_{o\,(crit.)}$ ($l_o$ critical for spring to buckle, $l_o < l_{o\,(crit.)}$).

We iterated the given code for values of d ranging from 1 to 7 mm some values of d satisfy all the condition for spring design. These values of d along with various others parameters are presented in Table 1.

***Table 1:*** *Spring Specification Table.*

| **d (mm)** | **3.2** | **3.3** | **3.4** | **3.5** | **3.6** | **3.7** | **3.8** | **3.9** |
|---|---|---|---|---|---|---|---|---|
| D (mm) | 21.090329 | 23.237062 | 25.503063 | 27.891422 | 30.405124 | 33.047072 | 35.820105 | 38.727006 |
| C | 6.5907278 | 7.041534 | 7.500901 | 7.9689778 | 8.4458678 | 8.9316411 | 9.4263434 | 9.9300015 |
| Na | 13.727531 | 11.607927 | 9.8942115 | 8.4938144 | 7.3385835 | 6.3775005 | 5.5718113 | 4.8916935 |
| $L_o$ (mm) | 98.528098 | 93.20616 | 88.840319 | 85.228351 | 82.218901 | 79.696752 | 77.572883 | 75.777605 |
| $L_{o(crit.)}$ (mm) | 110.93513 | 122.22695 | 134.14611 | 146.70888 | 159.93095 | 173.8276 | 188.41375 | 203.70405 |
| $n_f$ | 1.372082 | 1.3880358 | 1.403765 | 1.4192806 | 1.4345928 | 1.4497112 | 1.4646444 | 1.4794005 |
| $n_s$ | 1.2 | 1.2 | 1.2 | 1.2 | 1.2 | 1.2 | 1.2 | 1.2 |
| Ta (MPa) | 358.15407 | 355.26724 | 352.48884 | 349.81178 | 347.22963 | 344.73651 | 342.32707 | 339.99642 |
| Tm (MPa) | 358.15407 | 355.26724 | 352.48884 | 349.81178 | 347.22963 | 344.73651 | 342.32707 | 339.99642 |
| KB | 1.2140144 | 1.1986797 | 1.1851605 | 1.1731547 | 1.1624248 | 1.1527811 | 1.1440699 | 1.1361656 |
| f (Hz) | 187.86028 | 188.73 | 189.39004 | 189.87512 | 190.21327 | 190.42737 | 190.53625 | 190.55557 |

So for the minimizing the cost we will take the spring with d = 3.2 mm and other dimension of spring can be found out using above table.

## DESIGNING

### Shaft Design

The jumping robot contains two vertical power screw shafts for the compression of springs and subsequent downward translation of mass $m_1$. It uses two high torque (38 kg cm) and 60 rpm motors to rotate the power screws. The present shaft specifications are, length 120 mm, diameter 11 mm and single ACME threading of pitch 1.656 mm. An axial 15 mm blind hole is drilled on one side of the power screw and is joined to the motor shaft with a pin joint for torque transmission. The other side is embedded in nylon block in order to provide support to the screw (similar to a collar). With all this, the system takes quite a time to load the spring and finally execute the jump.

Hence, in the subsequent sections we would use the theory of power screw and beam loading to optimize the threaded shaft specs and increase the speed of the spring loading mechanism by using multi-threaded screw.

### Buckling

As a first step, we would determine the minimum possible shaft diameter (d) without changing the original shaft length of 120 mm. We had developed a code which takes input of axial load (P), yield strength ($S_y$), elastic modulus of rigidity (E), shaft length (l) and design factor ($n_d$). The code employs the J. B. Johnson formula to find the minimum d.

For a circular shaft radius of gyration, k

$$k = \frac{d}{4}$$

Critical axial force is given by,

$$\frac{P_{cr}}{A} = S_{y-} \left(\frac{S_y}{2\pi}\frac{l}{k}\right)^2 \frac{1}{CE} \qquad \text{Eq. (6)}$$

If $\frac{l}{k} \le \left(\frac{l}{k}\right)_1$

$$\frac{P_{cr}}{A} = \frac{C\pi^2 E}{\left(\frac{l}{k}\right)^2} \qquad \text{Eq. (7)}$$

If $\frac{l}{k} > \left(\frac{l}{k}\right)_1$

where,

$$\left(\frac{l}{k}\right)_1 = \left(\frac{2C\pi^2 E}{S_y}\right)^{1/2} \qquad \text{Eq. (8)}$$

The end conditions are fixed-fixed; hence the recommended value of end condition constant is taken as C = 1.2

The code calculates d using parabolic formula and then checks it against critical slenderness ratio. If it violates the condition for critical slenderness ratio, then it calculates the diameter (d) using Euler's formula.

For the power screw we have proposed to use the following steel which is in general used in the industry for this application (Table 2):

***Table 2:*** *Properties of Steel AISI 4140.*[5]

| AISI No. | Heat Treatment | Temperature (°C) | Tensile Strength (MPa) | Yield Strength (MPa) |
|---|---|---|---|---|
| 4140 | Quenched and Tempered | 650°C | 758 | 655 |

So, finally we get a minimum possible diameter d = 4.3 mm from the Euler's formula.

## SIMULATION RESULTS

In order to

1. Determine the appropriate dimensions of the shaft
2. Find the Von-Mises stress distribution on the shaft and using the maximum value for calculating fatigue factor of safety of the shaft.
3. Verify the value of Von Mises Stress at the root of the Power Screw which was calculated using power screw approach

ABAQUS[6] software environment was chosen to perform all the required simulation work. So, we started with the initial design of the shaft which was used in the jumping robot and developed its CAD model and then did the analysis part in the following steps:

Firstly, import the CAD model from the Autodesk Inventor Pro, then using ABAQUS

1. Set its Mechanical Properties (E = 210 GPa and Poisson's Ratio = 0.3)
2. Applied the appropriate Boundary Condition (other end of shaft will be ENCASTRE [U1 = U2 - U3 - UR1 - UR2 = UR3 = 0])
3. And finally applied the Axial Load and the Torque on the shaft.
4. Now we moved further into the Mesh Generation Section where we created the fine mesh on the surface of the shaft as shown in Figure 7.
5. And the last step is the simulation step we run the simulation and obtained the result of Von-Mises Stress Distribution as shown below (Figures 8–12):

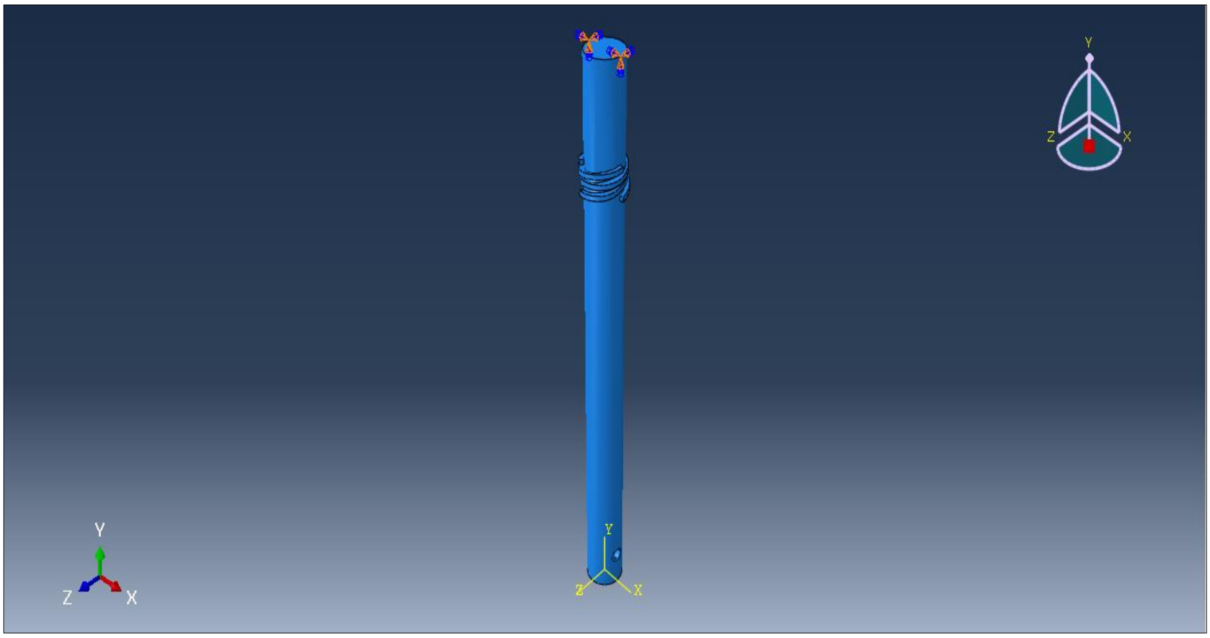



***Fig. 6:*** *Image Showing the Shaft with Appropriate Boundary Condition and Applied Load.*

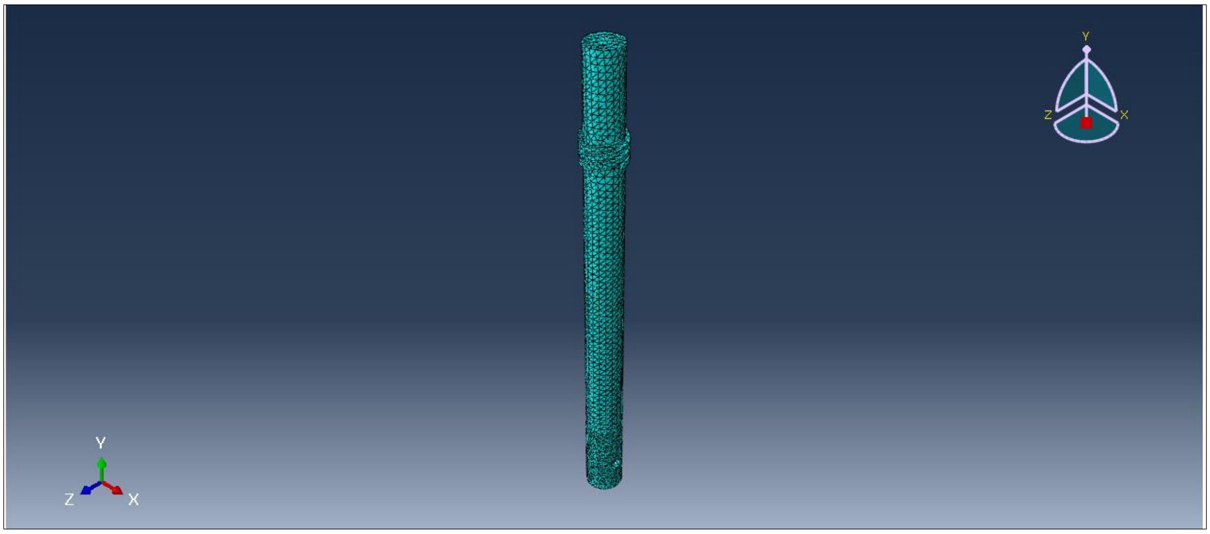



***Fig. 7:*** *Image Showing the Shaft Along with the Mesh.*

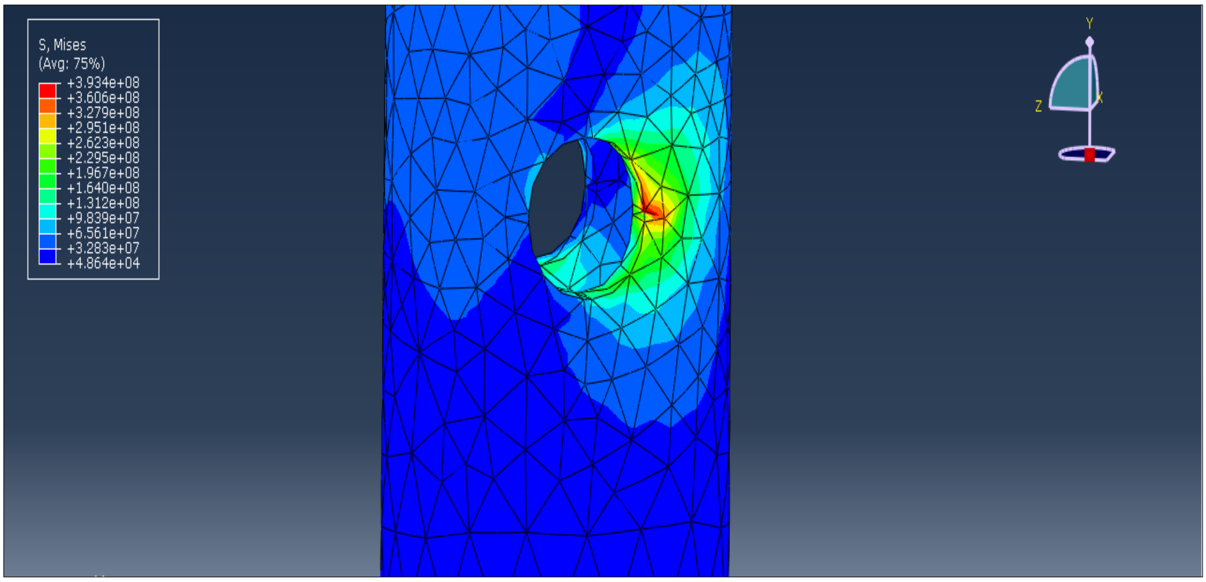



***Fig. 8:*** *Image Showing Von-Mises Stress Distribution on the Bottom Part of the Shaft.*

**Inference**

From the result obtained above, we observed that the critical point on the shaft where the Von-Mises Stress is maximum, is on the hole for key (the region with the red color) with the magnitude of 393.4 MPa. We calculated the factor of safety using Langer Criterion

$$Factor\ of\ Safety = \frac{(Yielding\ Strength)}{(Max\ Von\ Misses\ Stress)}$$

= 1.665

And Stress concentration factor = 2.75

Which exceeds from the theoretical value ($K_{ts}$ = 2.65) calculated using Figure A-15-10 from Shigley's Mechanical Engineering Design 9th Edition.[5]

Therefore, we have decided to modify the shaft geometry in order to avoid the failure in any way. We had iterated the whole process with different dimensions of the shaft and finally ended up finding the correct dimension parameters for the shaft. Following is the analysis part of the final Shaft geometry:

Since, there were no chances of any failure in the threaded region therefore all we needed to care about was the hole for key where motor transfer its torque to the shaft thus, we increased the outer diameter from 10 to 13 mm of the shaft up to the point after which threading starts and inner diameter being equal to the motor's shaft diameter hence remains the same.

6. Repeating first step including setting up mechanical properties, defining boundary condition and load condition.
7. Now we moved further into the Mesh Generation Section where we created the fine mesh on the surface of the shaft as shown below:
8. And the last step is the simulation step we run the simulation and obtained the result of Von-Mises Stress Distribution as shown below:

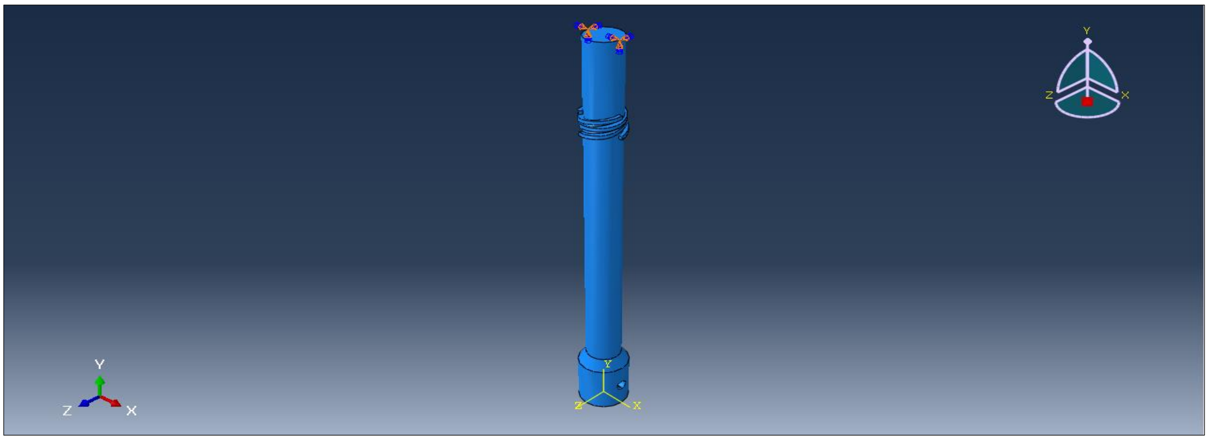



***Fig. 9:*** *Image Showing the Shaft with Appropriate Boundary Condition and Applied Load.*

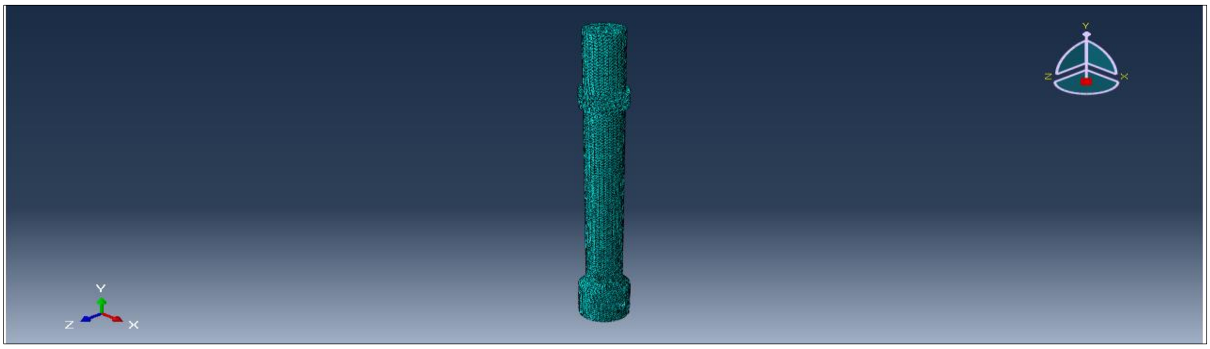



***Fig. 10:*** *Image Showing the Shaft Along with the Mesh.*

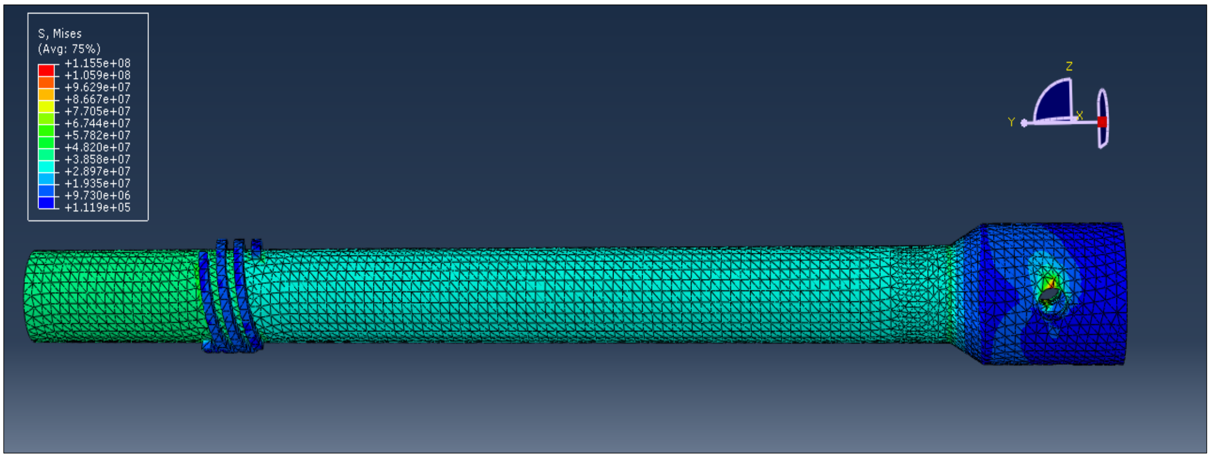



***Fig. 11:*** *Image Showing Von-Mises Stress Distribution on the Shaft.*

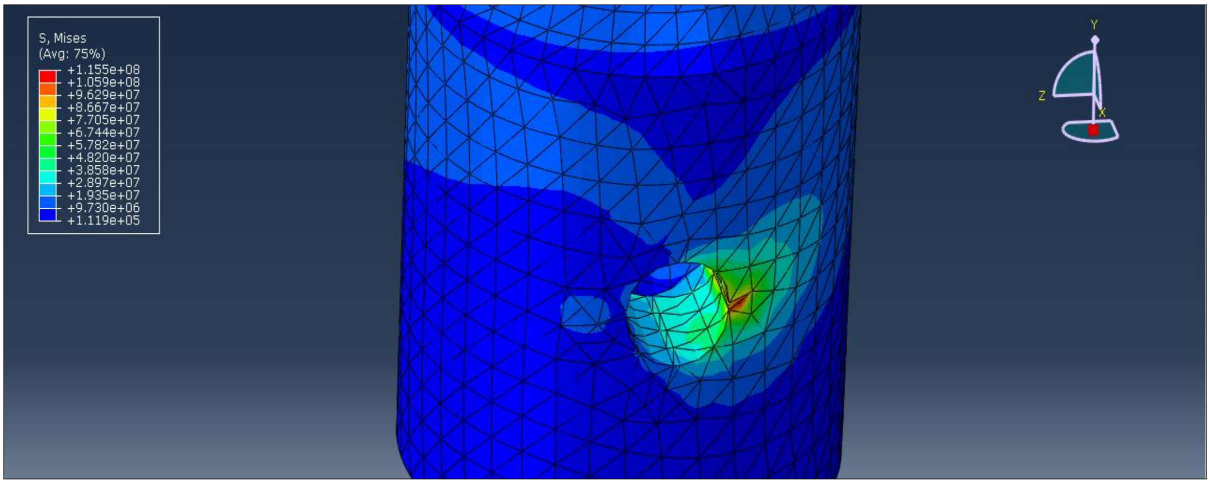



***Fig. 12:** Image Showing Von-Mises Stress Distribution on the Bottom Part of the Shaft.*

**Inference**

From the result obtained above, we observed that the critical point on the shaft where the Von-Mises Stress is maximum, is on the hole for key (the region with the red color) with the magnitude of 115.5 MPa. We calculated the factor of safety using Langer Criterion

$$factor\ of\ safety = \frac{(Yielding\ Strength)}{(Max\ Von\ Misses\ Stress)} = 5.671$$

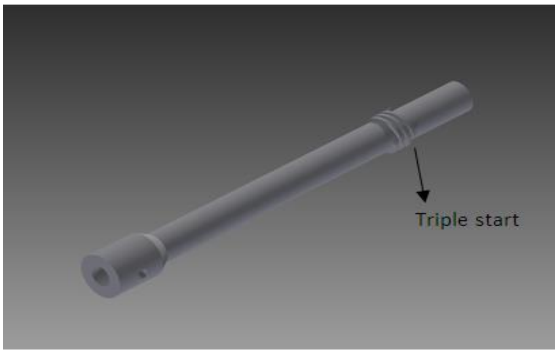
And Stress concentration factor = 2.74

The calculated stress concentration factor matches the theoretical value ($K_{ts}$ = 2.75) calculated using Figure A-15-10 from Shigley's Mechanical Engineering Design 9th Edition.[5]

**Power Screw Design**

The calculation for number of multithread is as follows,

**FORCE ANALYSIS**



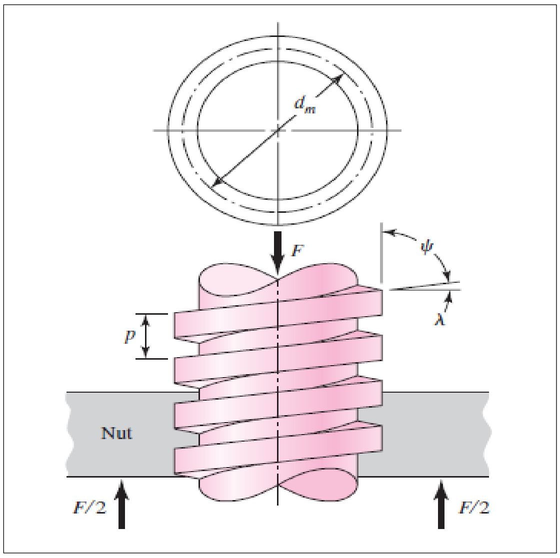




***Fig. 13:** Power Screw.*

In the Figure 13, F is the axial load from springs F = 720 N

p is the pitch of thread

λ is lead angle

l = n×p

n is no. of multithread start

$d_m$ is mean diameter of shaft

The system is right hand (RH) thread and shaft is rotating anticlockwise (Figures 14, 15).

$$T_R = \frac{Fd_m}{2}\left(\frac{l+\pi f d_m}{\pi d_m - fl}\right) \qquad \text{Eq. (9)}$$

f is coefficient of friction between thread and nut.

$P_R$ is load to raise the screw.
N is normal reaction on thread.

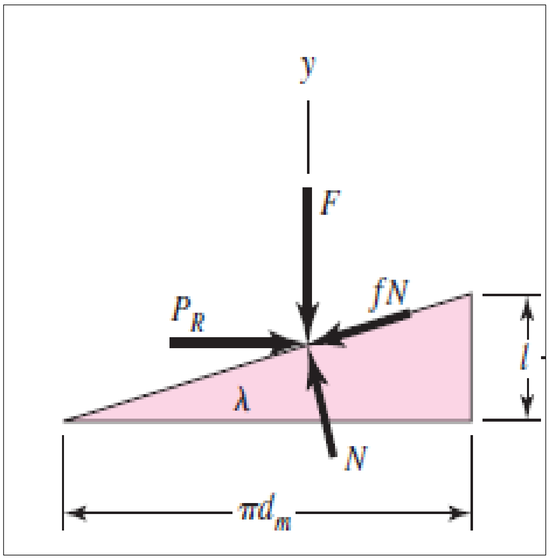



***Fig. 14:*** *Forces on a Power Screw.*

Torque on shaft is product of force and mean radius.

$T_R$ is the torque to overcome thread friction and raise the load
The collar friction is between nylon and steel

$$T_c = \frac{F f_c d_c}{2} \qquad \text{Eq. (10)}$$

Total torque on thread is

$$T_R = \frac{F d_m}{2}\left(\frac{l + \pi f d_m}{\pi d_m - f l}\right) + \frac{F f_c d_c}{2} \qquad \text{Eq. (11)}$$

For power screws, the Acme thread is not as efficient as the square thread, because of the additional friction due to the wedging action and the component of force did not get zero α = 90 degrees for square thread

**STRESS ANALYSIS**
Geometry of square thread useful in bending and transverse shear stress at the thread of the root.

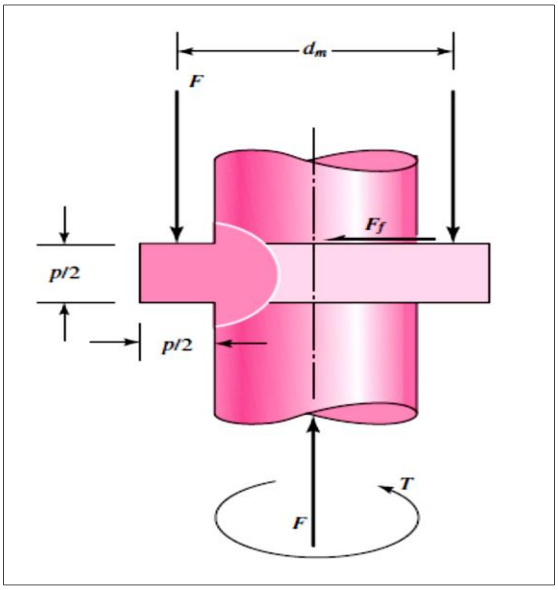



***Fig. 15:*** *Stresses on Screw.*

The stresses on root of thread are calculated is as follows

$$\sigma_x = \frac{6F}{\pi d_r n_t p} \quad \tau_{xy} = 0 \qquad \text{Eq. (12)}$$

$$\sigma_y = -\frac{4F}{\pi d_r^2} \quad \tau_{yz} = \frac{16T}{\pi d_r^3} \qquad \text{Eq. (13)}$$

$$\sigma_z = 0 \quad \tau_{zx} = 0 \qquad \text{Eq. (14)}$$

$d_r$ is minor diameter
$n_t$ is no of thread

Von mises stress ($\sigma'$)

$$\sigma' = \frac{1}{\sqrt{2}}\left[(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 + 6(\tau_{xy}^2 + \tau_{yz}^2 + \tau_{zx}^2)\right]^{1/2} \qquad \text{Eq. (15)}$$

And factor of safety ($n_s$)

$$n_s = \frac{S_y}{\sigma'} \qquad \text{Eq. (16)}$$

The code uses input i.e., axial load (F), coefficient of friction(f), coefficient of friction between collar and screw($f_c$), pitch(p) and design factor of shaft($n_d$) and to give output as Torque($T_r$), Von Mises stress(sv) and no. of multithread start(n). We also need to iterate the code to get the correct values of n.

The output of the code is
N = 2.98 (no. of multithread start)
sv = 58.22 MPa (Von Mises stress)
$T_r$ = 3.77 Nm (Torque)

**FATIGUE**

$S_e$ = Endurance limit at the critical location of shaft
$k_a$ = Surface condition modification factor
$k_b$ = Size factor
$k_c$ = Loading factor
$k_d$ = Temperature factor
$k_e$ = Reliability factor
$k_f$ = Miscellaneous factor
$S_e$' = Endurance limit
$d_m$ = Mean diameter of the shaft
$S_{ut}$ = Minimum tensile strength

$S_e = k_a.k_b.k_c.k_d.k_e.k_f.S_e'$ Eq. (17)

Since, the material of shaft is 650°C, Q&T treated AISI 4140 Steel, and so we use surface finish as Hot-rolled (Table 3):

***Table 3:*** *Tabulated Results for AISI 4140 Steel.*

| Surface Finish | Factor a | | Exponent b |
|---|---|---|---|
| | $S_{ut}$, kpsi | $S_{ut}$, kpsi | |
| Hot rolled | 14.4 | 57.7 | -0.718 |

$k_a$ = 57.7(758) - 0.718 = 0.494
$k_b$ = 0.879(d) - 0.107 2.79 < d < 51 mm.
$k_b$ = 1.24(10) - 0.107 = 0.969
$k_c$ = 1 for torsional and axial loading
$k_d$ = 1 for room temperature (20-25$^0$C)
$k_e$ = 0.620 for 99.9999% reliability
$k_f$ = 1 overlooking the misc. factors
$S_e$' = 0.5($S_{ut}$) for $S_{ut}$ < 1400 MPa

So, we get
Se = 112.7 MPa

Calculating the factor of safety (for fatigue) using Goodman criterion

$$\frac{1}{n_f} = \frac{\sigma'_a}{S_e} + \frac{\sigma'_m}{S_{ut}} \quad \text{Eq. (18)}$$

$n_f$ = Fatigue factor of safety
$\sigma'_a$ , $\sigma'_m$ = Von Mises stresses amplitude and mean

As per the analysis done of shaft done on the *ABAQUS* software (section 4.2) we get the result that the Von Mises stress around the critical location (at the pin joint) is 115 MPa.
Since the load varies from zero to maximum $\sigma'_a = \sigma'_m$= 57.5 MPa

$$\frac{1}{n_f} = \frac{57.5}{112.7} + \frac{57.5}{758}$$

$n_f$ = 1.706

which implies that the life of the shaft is infinite.

**EXPERIMENTAL ANALYSIS**

We want to find out the jump attained by the present jumping robot and the displacement profile of two masses during the jump. We also want to find out the damping constant C in real system and apply the approximate value to earlier MATLAB simulation.[3] The Figure 16 depicts the attached accelerometer on the top mass. The signals from accelerometer are amplified using amplifier and read by oscilloscope.

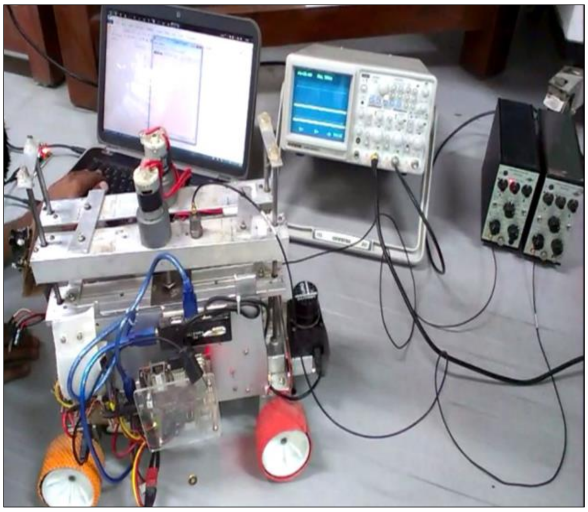


***Fig. 16:*** *Experimental Setup.*

We calibrated the amplifier and oscilloscope and found out the result of the oscilloscope. The above experimental result was obtained with torsional spring attached to two mass systems. It is evident from the oscilloscope that the coupled vibration in torsional spring system is difficult to detect.

The frequency of the displacement graph cannot be obtained from the oscilloscope as with the present experimental setup the oscilloscope does not show sinusoidal vibrations. The jump calculated from the graphs is found out to be 6.1 cm (100 mV = 1 mm). If we improve upon the experimental setup, we can find out the frequency of sinusoidal vibrations $f_{exp}$.

The resolution of oscilloscope needs to be increased largely to observe the oscillating motion of top and bottom mass while jumping. Since, the damping in system is due to friction and other losses, it is independent from the spring chosen (Figure 17).[7,8]

From the frequency of vibration we can find out damping constant C with the help of $\omega_n and\ \varsigma$.

$$f_d = \omega_n\sqrt{1-\varsigma^2} \qquad \text{Eq. (19)}$$

in this case $f_{\exp} = f_d$.

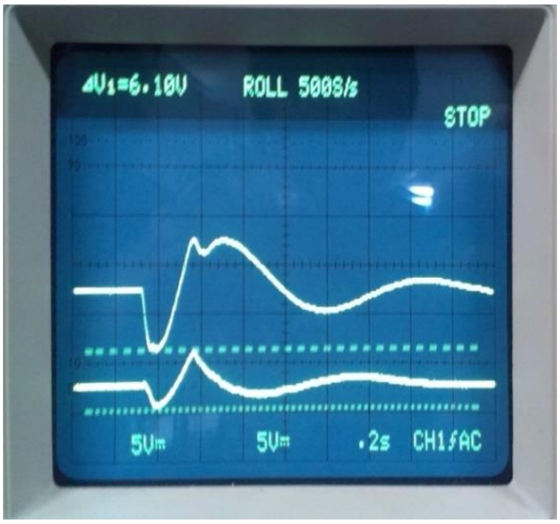



***Fig. 17:*** *Oscilloscope Results*.

If the sensitivity and accuracy of the experimental setup is increased, we can find many other correlations between theoretical result and actual result. These results will help us in better simulation and analysis of the system.

## CONCLUSIONS

We analyzed the jumping robot system and the came up with the following modifications to increase its jump.

1. The 4 torsional spring present in the system should be replaced with 4 linear springs.
2. The designed linear spring is made of 302 stainless steel wire and have dimensions of d = 3.2 mm, D = 21 mm, Na = 13.72, Lo = 105 mm, K = 8000 N/m. The spring has been proved for infinite life.
3. The design and analysis of power screw shaft was completed and 10 mm diameter shaft was found appropriate for given system made from AISI 4140 Q&T (650°C).
4. The power screw design in shaft was modified from single start thread to 3 start thread with square profile and pitch 2 mm. This will make the system faster.

We have described the detailed procedure for doing the experimental analysis and the result from experiments can be further incorporated in design and a better consonance with reality could be achieved.

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