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Faculty of Mechanical Engineering

Design of System Ensuring Defined Cable Tension under Dynamic Load

Master Thesis

Study programme: N2301 Mechanical Engineering
Study branch: Machines and Equipment Design

Author: **Murali Prasath Balu**
Thesis Supervisors: Ing. Petr Žabka, Ph.D.
Department of Design of Textile Machine

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Master Thesis Assignment Form

Design of System Ensuring Defined Cable Tension under Dynamic Load

Name and surname:	**Murali Prasath Balu**
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Rules for Elaboration:

1. Perform a search for automatic reel systems.
2. Design several possible solutions.
3. For selected variants, create mathematical models describing their dynamic behavior and compare the proposed solutions.
4. Elaborate drawing documentation for the selected solution.

Scope of Graphic Work: drawing documentation
Scope of Report: 35 pages
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[1] MEERKAMM, Harald, ed. *Technical Pocket Guide*. Germany: Schaeffler Technologies GmbH & Co. KG, 2014.
[2] AMBEKAR, Ashok G. *Mechanism and machine theory*. Delhi: PHI Learning Private Limited, 2007. Eastern economy. ISBN 978-81-203-3134-1.
[3] BAUCHAU, Olivier Andre. *Flexible multibody dynamics*. Dordrecht: Springer, 2011. Solid mechanics and its applications, volume 176. ISBN 978-94-007-0334-6.

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Liberec October 30, 2019

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Abstract

This thesis deals with designing a mechanical system that ensures defined cable tension under dynamic load conditions. In the current situation, the machine uses two servo motors to ensure proper tension and the wire position. The primary aim is to propose an alternate mechanical system that uses one less servo motor than the current situation—at the same time, ensuring proper cable tension on the wire during operation.

Multiple ideas were proposed based on the storage of potential energy, and on analysis, two are mainly considered. The primary constraint is the mechanical system's compactness as it is to be accommodated inside a box where the servo motor exists in the current situation. One variant uses a constant torque torsional spring which is coupled to the primary shaft. The second variant is a gravitational potential energy system using a mass connected to the primary shaft by using pulleys.

Simulations and measurements are performed for the wire and the two variants. For simulations, mathematical model using lumped parameters are created and compiled in MATLAB. Measurements such as identification of natural frequencies, stiffness, damping of the wire and measurements on the variants under dynamic load conditions were done to verify the simulation results.

A significant amount of data has been collected from the simulation and measurements. The torsional spring system's mathematical model shows desirable performance, but it is not suitable for this specific application, so it is not tested. The mathematical model and the experiments of the gravitational potential system show similar and desirable performance compared with the current situation.

The new proposed variant can significantly reduce the cost of the machine. The system can also operate at different speeds and ensure constant tension at the same time. When the constant tension can be achieved on the wire, it means the wire will have no or minimal vibration, which is desirable.
Key words

constant tension, potential energy, dynamic load conditions, vibration, cost, compactness

Abstrakt

Tato práce se zabývá návrhem mechanického systému, který zajišťuje definované napětí lanka při dynamickém zatížení. V současné situaci stroj využívá dva servomotory k zajištění správného napětí a polohy lanka. Primárním cílem je navrhnout alternativní mechanický systém, který používá o jeden servomotor méně, než je současná situace a současně zajišťuje správné napětí lanka během provozu.

Bylo navrženo několik řešení založených na uchování potenciální energie a na základě analýzy jsou zvažovány dva hlavní systémy. Primárním omezením je kompaktnost mechanického systému, protože má být umístěn uvnitř skříně, kde se v současné situaci nachází servomotor. Jedna varianta používá torzní pružinu s konstantním momentem, která je spojena s primárním hřídelem. Druhou variantou je systém gravitační potenciální energie využívající hmotu připojenou k primární hřídeli pomocí kladek.

Simulace a měření jsou provedeny pro lanko a pro dvě varianty. Pro simulace je v MATLABu vytvořen a zkompilován matematický model využívající soustředěné parametry. K ověření výsledků simulace byla provedena měření, jako je identifikace vlastních frekvencí, tuhosti, tlumení lanka a měření variant při dynamickém zatížení.

Ze simulace a měření bylo shromážděno značné množství dat. Matematický model systému torzních pružin vykazuje požadovaný výkon, ale není vhodný pro tuto konkrétní aplikaci, takže není testován. Matematický model a experimenty systému gravitační potenciální energie vykazují ve srovnání se současnou situací podobné vhodné chování.

Nová navrhovaná varianta může výrazně snížit náklady na stroj. Systém může také pracovat při různých rychlostech a současně zajistit konstantní napětí. Dosažení stálého napětí na lanku vede k tomu, že lanko bude mít minimální vibrace, nebo nebude vibrovat vůbec, což je žádoucí.

Klíčová slova

konstantní napětí, potenciální energie, dynamické zatížení, vibrace, náklady, kompaktnost

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CHAPTER 1: Introduction

This thesis deals with designing a mechanical system that ensures defined cable tension under dynamic load conditions. The mechanical systems proposed in this thesis are designed only for a specific application. However, these proposals can also work for other similar applications.

1.1 Current situation

As in figure 1-1, servo motor driven drums govern the steel cable's position in the current situation. They move the cable by winding and unwinding them alternatively. The drums are threaded to guide the cable during winding (Figure 1-2 & Figure 1-3). The system needs two servo motors, each for one drum to ensure motion and proper tension on the wire. One servomotor, which defines the position of the cable, operates in position mode. At the same time, the other ensuring constant tension on the cable operates in torque mode. However, the servo motors switch modes during a change in the rotational direction.

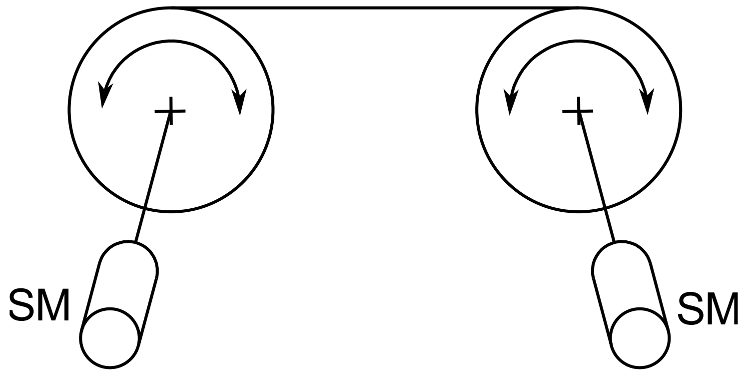


Figure 1-1 scheme of the current setup

From the above Figure 1-1, it can be seen that a scheme is a simplified form of the actual setup. In the real situation, the motor drives the drum employing a belt and pulley. The system described above has already been manufactured and in operational condition (Figure 1-2). Figure 1-2 shows that the drum is connected to a pulley by a shaft. A servo motor drives the pulley through a belt. The servomotor that drives the pulley is hidden under the plate as per the technological requirement. As the technology operates at high voltages, it is desirable to have the servomotor below the plate.

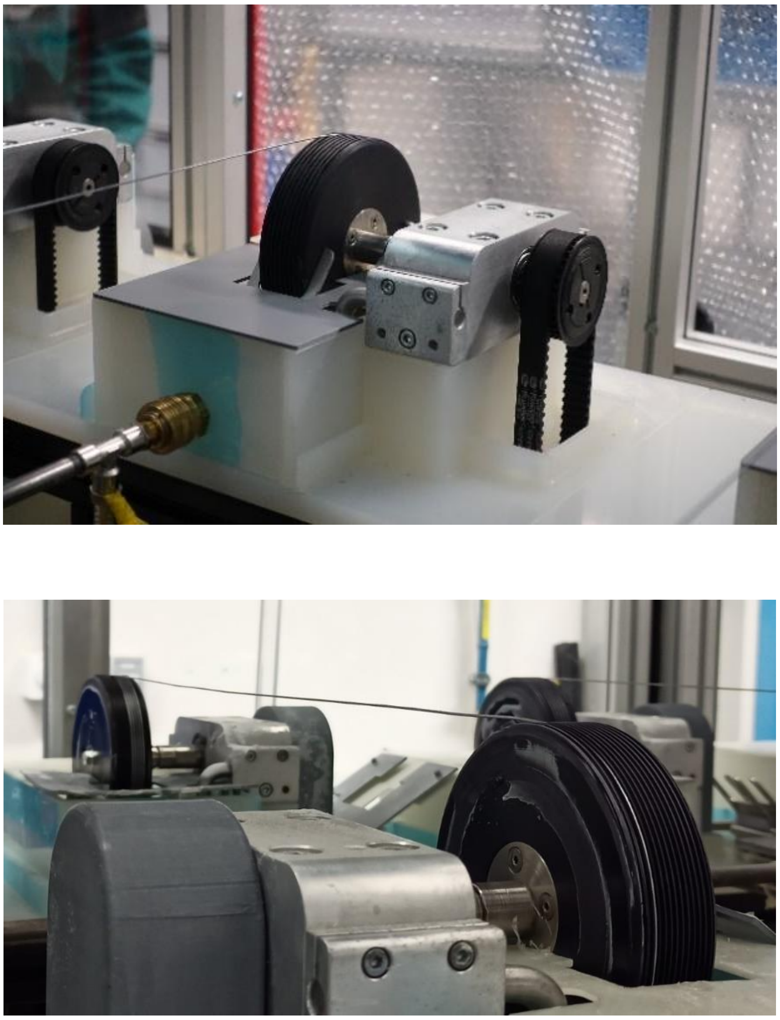


Figure 1-2 existing setup at one end (top) and both ends (bottom)

1.2 Problem definition

The results of the current system are satisfying. However, the cost of the servomotors and their subsystems are high and can be reduced. Maintaining constant tension on the wire or within the desired range is the main problem. A wire without constant tension might cause vibration in the wire. In this specific application, the vibration should be kept as minimal as possible because a fluid layer will be coated on the wire during operation. If the vibration occurs on the wire, then the fluid might drip away from the wire. Tuning of tension needs to be done to avoid such vibrations in some tension zones. However, it is impossible to maintain constant tension but can be brought down to the desired range.

The picture (Figure 1-3) shows the fluid droplets on the wire. This picture was taken when the machine was in a static condition. The fluid looks like droplets and is almost equally spaced due to adhesion, cohesion, and surface tension properties.

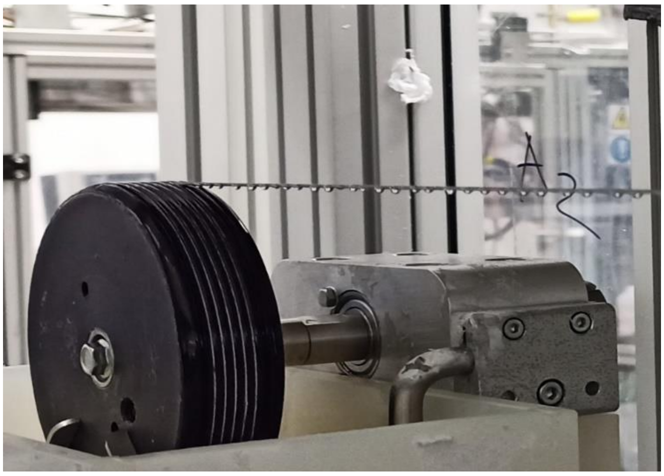


Figure 1-3 fluid on the wire (taken when the machine is static)

Figure 1-4 shows us the actual excitation curve, which the current system uses (velocity = 22.5 m/min, frequency = 1/12 Hz).

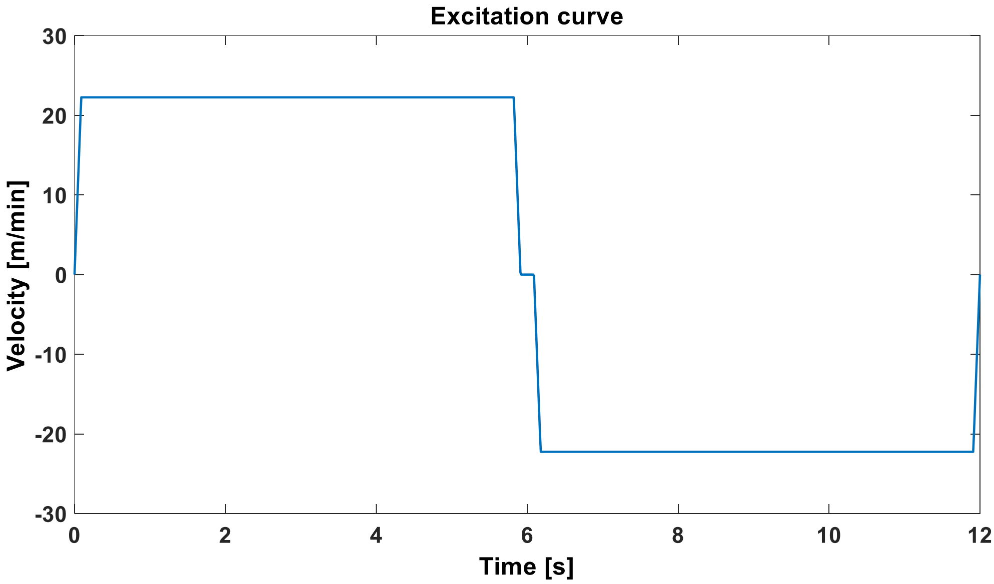


Figure 1-4 excitation curve used in a real-world situation

1.3 Aim

The primary aim is to reduce the cost of the system. The total cost of the system can be reduced by eliminating one servomotor. So, it is necessary to design an alternative mechanical system that compensates for the servomotor.

Maintaining the wire's tension within a small desired range even at high speeds is also a priority. The vibration on the wire is avoided by tuning the tension properly to avoid dripping of fluid away from the wire.

The work also focuses on testing the system's limits by adjusting some parameters under dynamic load conditions. The parameters include various designs, mass, tension, length of the wire. The work is also to design several variants, which enable us to change the parameter more easily.

The steps followed in the work are as follows:

- Research for various possible mechanical systems.
- Selection of variants.
- Design of various possible solutions.
- Create mathematical models describing their dynamic behavior for selected variants.
- Comparison of proposed solutions.

The excitation curve from Figure 1-4 will be used in the experiment. Furthermore, some other curves like sin, square and trapezoidal shape are also tested using mathematical models, and the mentioned curves are as follows (Figure 1-5) (velocity = 22.5 m/min, frequency = 1/12 Hz). More details on the excitation functions can be found in the supplements (A1 , A2 & A3).

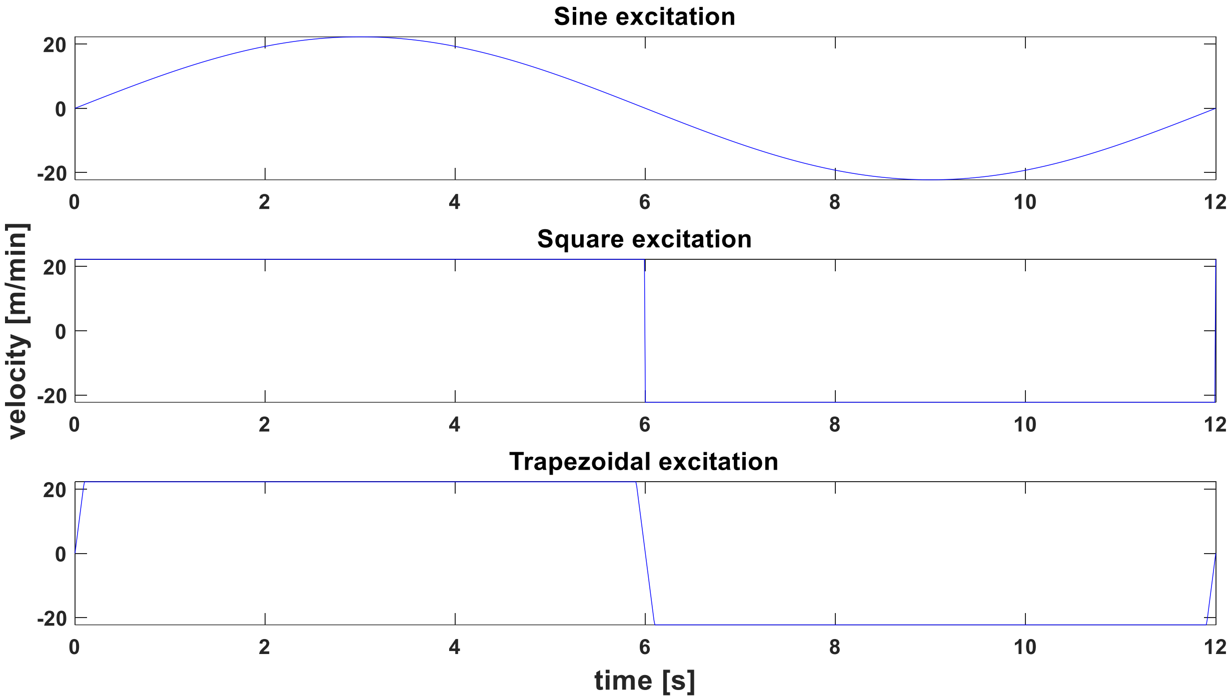


Figure 1-5 excitation function (sin, square, trapezoidal) various shapes

CHAPTER 2: Research

The research work is done mostly with the help of patent inspiration [12]. Simultaneously, the search for patents was based on the keywords automatic reels, self-retracting, winding, torsion spring, potential energy system, and more. On considering the search for the system that stores potential energy, the ideas were inspired by the various fields in the field analysis (Figure 2-1). Each field is taken individually, and research brainstorming sessions were done. The significant fields considered were gravitational, mechanical, magnetic, and pressure.

Figure 2-1 field analysis

2.1 Inspiration

Some of the ideas are inspired by real-world applications. They are as follows,

2.1.1 Measuring tape

Spring return measuring (Figure 2-2) tape comprises a flexible metal strip with linear measurement markings inside the casing. The flexible metal strip is connected to a pre-tensioned flat torsional spring attached to the casing. So, this system helps in a compact design and helps the metal strip to self-retract after use. It also noted that this mechanism does not provide constant tension.

Figure 2-2 measuring tape

2.1.2 Magnetic spring

A system with servomotors performing vertical motion requires additional force other than the required force to counterbalance the gravity. In such cases, the usage of magnetic springs [21] (Figure 2-3) comes in handy as they have the unique property of maintaining constant force over the entire working range. This property also reduces the load on the motor and smoothening the start and stop of the cycle. These systems help in reducing the stresses created on the moving parts during dynamic motion.

Figure 2-3 magnetic spring

2.1.3 Constant torque spring

Constant torque spring comprises a pre-stressed strip of metal, which is wound like a coil and stored in a small storage drum [2][3][4]. The other end of the strip is connected to a large output drum. When the output drum rotates, the strip is back wound onto the large drum. The strip returns to the initial natural shape, resulting in the strip's retraction to the small drum. This type of spring has a unique property of giving constant torque output. The torque comparison plot between the wire spring, flat spiral spring, and constant torque spring is given below (Figure 2-4).

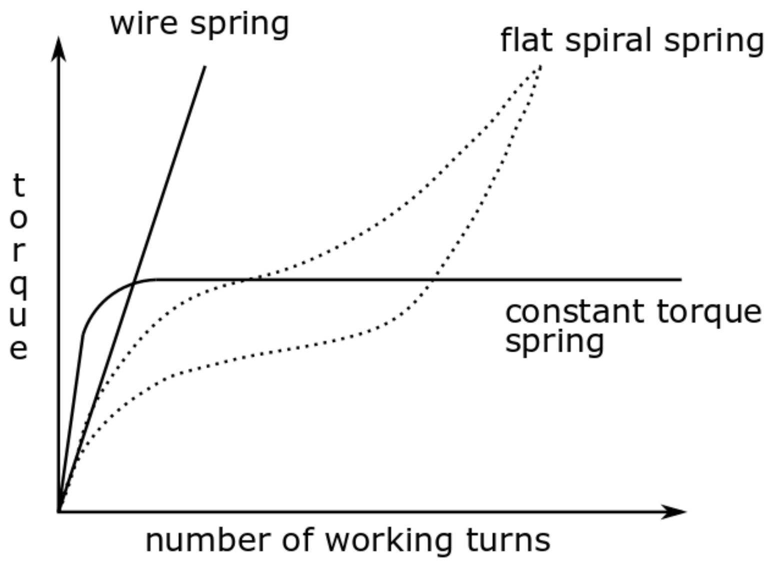


Figure 2-4 displacement vs torque [3]

2.2 Proposal of ideas

The fundamental idea is to design a mechanical system that stores potential energy. The stored potential energy can be used again to wind the cable. There are two distinct types of motion, and they are,

- Rotary conservation
- Linear conservation

Based on the two conservations mentioned, the proposal of some mechanical systems is made. The proposed mechanical systems are as follows,

2.2.1 Rotary conservation

2.2.1.1 Torsion spring

Figure 2-5 torsion springs. ordinary flat torsion spring (left), constant torque spring (right)

A torsion spring is a mechanical device that stores potential energy when twisted [18][2]. It exerts torque in the opposite direction of a twist when twisted. The torque is proportional to the angle of twist.

Firstly, in the proposal, two types of torsion spring (Figure 2-5) were considered. One is the ordinary flat torsion spring, and the other one is the constant torque spring. From the torque characteristics observed from the plot (Figure 2-4), constant torque spring was mainly considered.

The arrangement of the spring is as follows. One end of the torsion spring is attached to the shaft connecting the drum. At the same time, the other end to be attached to the frame. So, when the pulley is rotated in the first half of the cycle, the spring is twisted. The spring's potential energy stored by the twist will be used in the second half of the cycle. In the second half, the torque exerted by the spring rotates the drum which winds the cable.

2.2.1.2 Compressed air

A rotary actuator (Figure 2-6) is a device that operates under the influence of compressed air. The device comprises a vane present inside a sealed cylinder. The vane is connected directly to the output shaft. When rotary actuation is required, compressed air is fed in, which pushes the vane, and rotation is achieved. Pneumatic valves control the supply of air.

In the proposal, the output shaft of the actuator is connected to the shaft connecting the drum. When actuation is required, the valve is triggered, and the supply of air starts. The air supply actuates the drum, resulting in the winding of the wire.

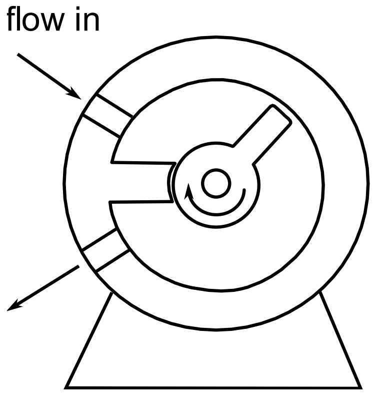


Figure 2-6 compressed air

2.2.1.3 Gravitational potential energy

The potential energy is directly proportional to the displacement (height) is a known fact. So, if there is an increase in height of a mass, there is an increase in potential energy and vice versa. In the proposal (Figure 2-7), the system uses a lever, gear train, and potential energy conservation technique. The lever's rigid rod holds a mass at one end, while the other end is pivoted to the gear train's output shaft. The input of the gear train is from the shaft connecting the drum. The mechanism has a large gear ratio so that even the smallest displacement of mass results in some rotations of the drum. In the first half of the cycle, the drum rotation causes the mass to rise, storing potential energy. Then in the second half of the cycle, the raised mass, which comes down because of gravity, rotates the drum. The drum rotation causes the cable to wind, and the process is continued.

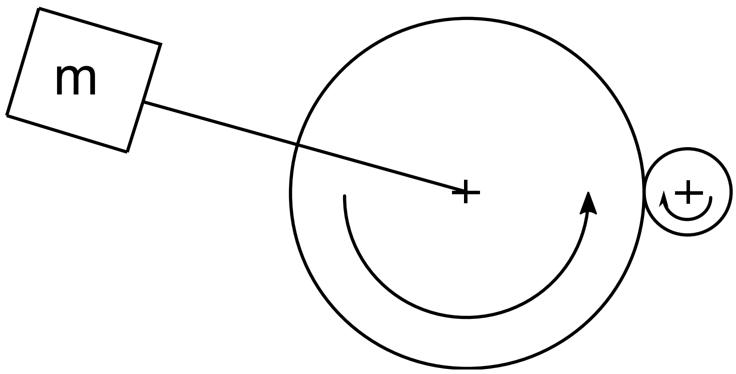


Figure 2-7 gravitational potential energy (rotatory conservation)

2.2.2 Linear conservation

2.2.2.1 Linear spring

Linear spring is a mechanical device that can store potential energy by altering its dimensions. In a linear spring, the applied load is directly proportional to the displacement in the spring.

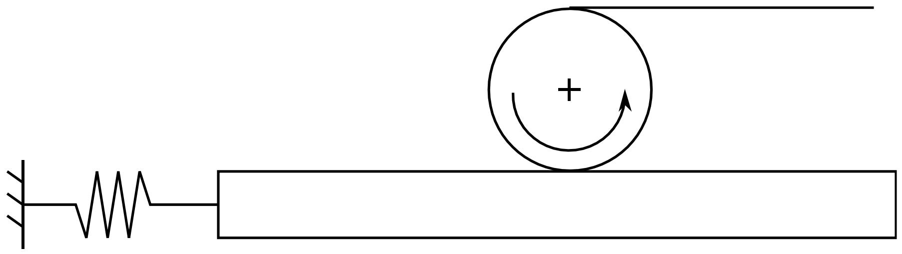


Figure 2-8 simplified version of linear spring connected to rack and pulley

In the proposal (Figure 2-8), the linear spring is connected between the frame and the rack. In the first half of the cycle, the rotational energy conversion into linear energy is done by the rack and pinion. The displacement of the rack compresses the spring, enabling it to store potential energy. The stored potential energy is used to rotate the drum in the second half of the cycle.

2.2.2.2 Gravitational potential energy

In the proposal (Figure 2-9), raising the mass's position enables us to store potential energy in the system. The system uses a mass connected on one end of the wire. Another end of the wire is connected with the shaft through a pulley. When the main shaft is rotated, the mass will be raised storing potential energy. Potential energy is stored proportionally to the rise in height. The stored potential energy is used to actuate the drum in the second half of the cycle.

Figure 2-9 gravitational potential energy system (linear conservation)

2.2.2.3 Magnetic spring

The significant difference between a linear spring and a magnetic spring is the exertion of force along with its operating range. In linear spring, force is proportional to the displacement, whereas the magnetic spring exerts a constant force (Figure 2-10) throughout their operating range. The magnetic spring [21] comprises a stator and a slider, which comprises neodymium magnets. The stator supports the slider by using a sliding bearing.

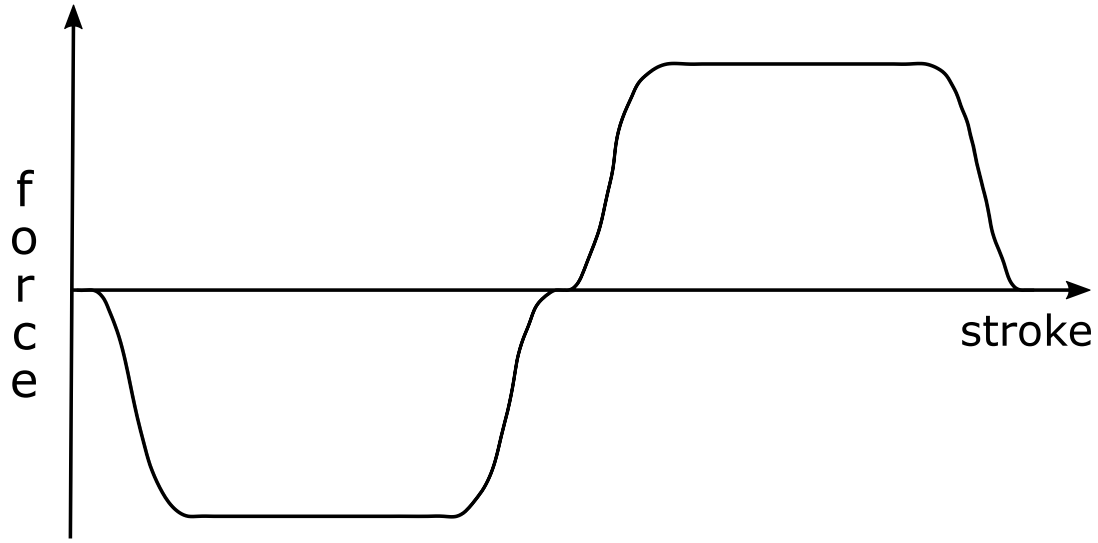


Figure 2-10 force vs displacement mag spring [5]

2.2.2.4 Compressed air

A pneumatic cylinder, a piston sealed airtight inside a cylinder, is actuated by air's regulated flow.

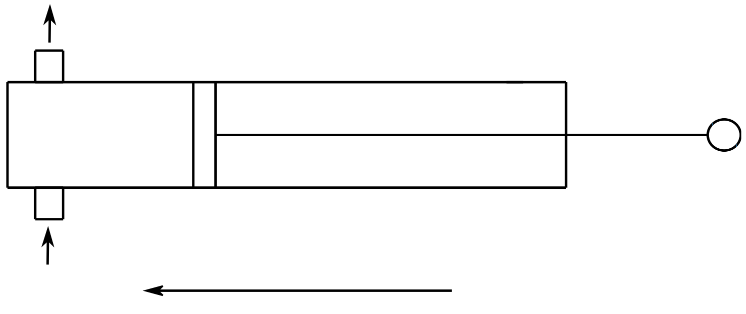


Figure 2-11 piston and cylinder

In the proposal (Figure 2-11), the end of the piston is connected to the rack. In the first half of the cycle, the drum's rotation motion is converted into the rack's linear motion. The rack actuates the piston from one side to another. In the second half of the cycle, the compressed air is fed into the cylinder, which actuates the rack. That, in turn, rotates the drum, and the cycle is continued.

2.2.3 Energy transfer

The conservation techniques mentioned above require a means of energy transfer. Moreover, they do it by using the following mechanisms:

- Rack and pinion
- Gear train
- Lever
- Drum

2.2.3.1 Rack and pinion

Rack and pinion (Figure 2-12) combination that converts rotary motion to linear motion and vice versa. It comprises two different gears: linear gear (rack), and the other is circular gear (pinion). Rotating the pinion moves the rack linearly, and translating the rack rotates the pinion. The force delivered depends mainly on two parameters, and they are tooth pitch and pinion size.

Figure 2-12 rack and pinion

2.2.3.2 Gear train

A mechanical system with a sequence of gears mounted in a way that their teeth engage is known as a gear train (Figure 2-13). They help in providing transmission without slippage. They provide the system with various mechanical advantages such as variable speed, torque multiplier.

Figure 2-13 gear set

2.2.3.3 Lever

A simple mechanism comprising a rigid body pivoted by a fulcrum is a lever (Figure 2-14). This mechanism's unique advantage is its capability to rotate the rigid body on a point (fulcrum). So, if the fulcrum position is changed, the torque required to rotate also changes. The force required to rotate also depends on the point of application of load. If the load application point is closer to the pivot point, it requires an enormous amount of force to rotate the rigid body. Moreover, farther, the point of application is less force is enough to rotate the body.

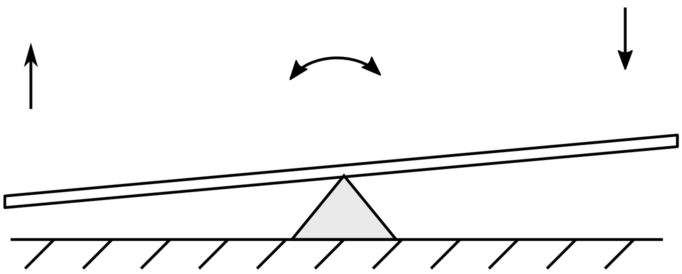


Figure 2-14 lever

2.2.3.4 Drum

The drum is a small cylindrical onto which wire, thread and tapes can be wound. It is often used as a temporary storage device (Figure 2-15).

Figure 2-15 drum

2.3 Analysis of ideas

From the ideas discussed and their energy transfer methods, the best ideas are selected. The best ideas were shortlisted by considering the following factors:

- low cost
- simplicity in modifications from the current situation
- compactness
- acceleration of the moving parts should be low
- ability to provide constant tension
- maintenance

The two ideas selected are as follows:

- Gravitation potential energy system (linear conservation) is one of the variants. This variant is selected due to its low cost and simple modifications from the current situation and low moving parts' acceleration. The system was also considered for its compactness. The system should be designed to be placed inside the box where the servo was placed in the current situation. The space available inside the box was considered, and calculations were made to check whether the system's mass can be accommodated.
- The torsion spring system is the second variant. This idea was selected due to its low cost and compactness. This idea was selected because the spring gives constant torque output and it is compact. It is easy to accommodate the spring in the current situation with very few modifications.

Furthermore, the two variants are considered for designing and simulations.

CHAPTER 3: Vibration on the wire

Vibration is a mechanical phenomenon whereby oscillations occur about an equilibrium point [14][16][19]. In other words, vibration in a system is due to unbalanced forces in the system. Vibration in a system should be considered as one of the most critical parameters to be aware. The resonance condition can cause vibration. Resonance is the phenomenon that occurs when the frequency of the excitation matches the natural frequency of the system. The vibration occurs in the lateral and longitudinal direction. Since the longitudinal natural frequencies are generally higher, it is ignored, and the lateral vibrations are concentrated more in work.

3.1 Properties of wire

This chapter explains the measurement of the general properties of the wire.

3.1.1 Stiffness of wire

The stiffness of the wire is one of the essential material properties. Stiffness is found by using the universal testing machine. A small sample of wire is placed in the universal testing machine, and constant incremental tension is given. The setup of the wire on the universal testing machine can be seen in Figure 3-1. The wire stretches until the elastic limit and deforms or breaks permanently beyond the elastic limit due to the tension force. The tension force applied and displacement of the wire are recorded. The data recorded are plotted, as shown below in Figure 3-2.

Figure 3-1 cable of length 200mm placed in a Universal testing machine

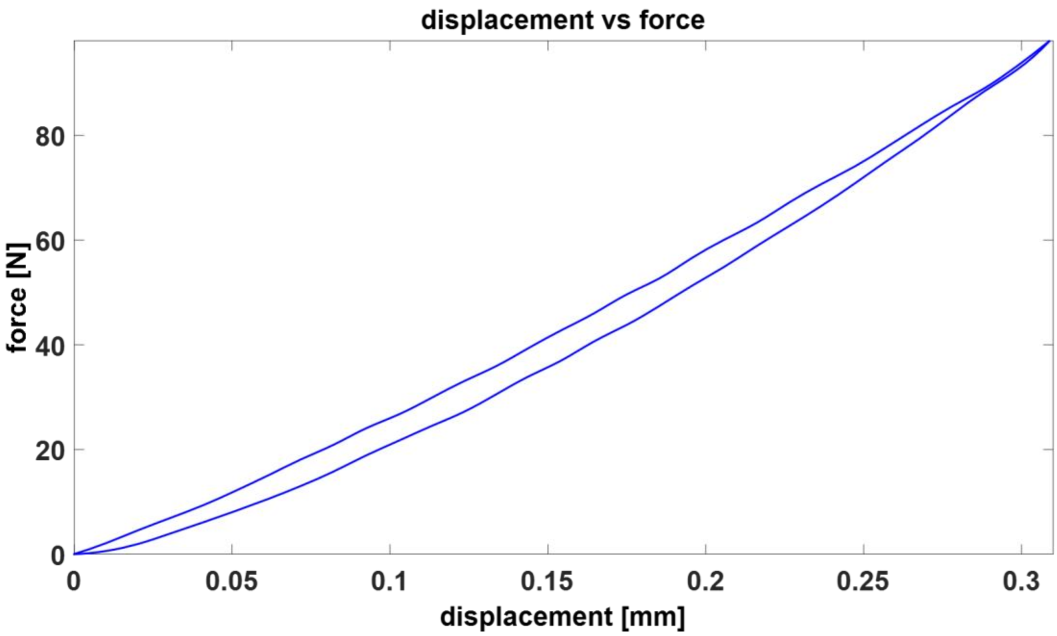


Figure 3-2 displacement vs load of the cable

The wire's stiffness can be found from the data acquired using the general formula or finding the plot's slope value. The stiffness value is found to be k = 3.082334 e5 [N/m] for a wire of length 200mm from the plot's slope value (Figure 3-2).

3.1.2 Damping of wire

Damping reduces the effect of oscillations in a system over time [17]. The value of damping was found by experiment, as it could improve the mathematical model's accuracy. Positioning the accelerometer on the wire was not done. Instead, the accelerometer was placed on the side of the pulley, as shown (Figure 3-3). The hanging mass provides tension on the wire. The wire position can be computed in all three directions from data provided by the accelerometer. Though the accelerometer can record in three directions, only one is prominent.

Figure 3-3 accelerometer positioned on pulley

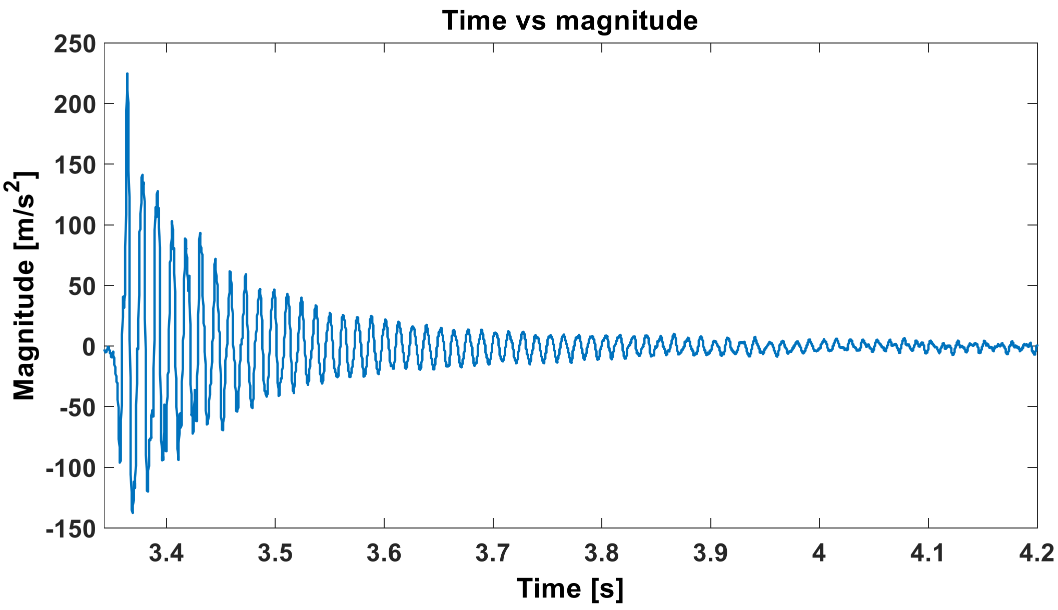


Figure 3-4 acceleration data from the accelerometer vs time after excitation

Then, the wire is pulled manually to induce vibration, and the position data is recorded. From the recorded data, the wire's damping value can be found by the logarithmic decremental method. A detailed explanation of the damping factor of wire is given in section [3.5]. The acceleration sample recorded for mass number 8 (Table 1) approximately equal to 10.7 kg is shown in Figure 3-4.

Other properties like the wire's diameter, length of the wire, and the wire's mass are measured using standard measuring devices. The standard measuring includes vernier calliper, measuring tape, weight scale.

3.2 Mathematical model

The vibration on the wire comprises of both lateral and longitudinal vibration. It is good to identify the vibration's natural frequencies on the wire in both lateral and longitudinal directions.

Here mathematical models are created for three different lumped models. The lumped models are

- longitudinal vibration on wire
- lateral vibration on wire
- combined lateral and longitudinal vibration on wire

Three different models are used because each has its advantages. Lateral and longitudinal models are created as they are simpler, faster, and easier for identifying the wire's natural frequencies in their respective directions.

In the real situation, when the wire is longitudinally excited, the wire excites in the lateral direction. Since there is a relationship between the lateral and longitudinal in a real situation, the third model is created. The third model is created as it combines both the previous models and describes the wire's real situation.

3.2.1 Lateral vibration on wire

Vibration in the direction perpendicular to the wire axis is known as Lateral vibration [9][11]. The wire's mass and the tension plays a vital role in this kind of vibration, resulting in the wire's elongation while during operation.

3.2.1.1 Assumption and simplification

Here for calculating the lateral vibration, the lumped mass approach is used [7][15]. Furthermore, this method has the following assumptions and simplifications:

- mass of the wire is equally divided into n number of mass points.
- The angle of deviation between the two consecutive mass points is considered to be small.
- Damping due to air resistance is neglected.
- Bending resistance of the wire is considered as zero.

3.2.1.2 Motion equations

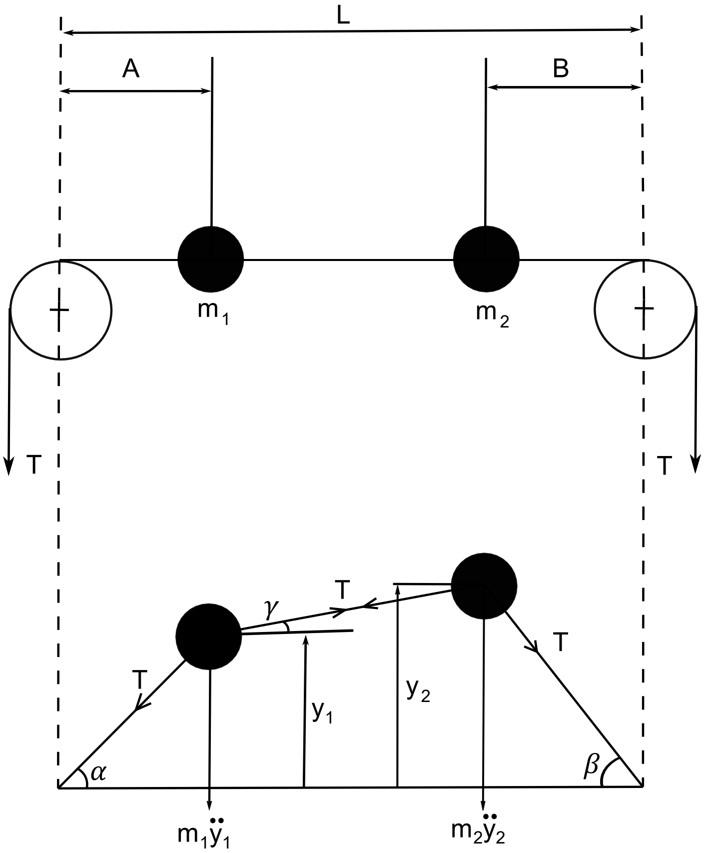


Figure 3-5 lumped mass model for lateral vibration with two mass points

The lumped mass model of the wire is drawn for only two mass points for illustration purposes. However, the number of mass points on the wire can be increased or decreased. Generally higher, the number of mass points greater the accuracy of the simulation. The free-body diagram is as follows (Figure 3-5),

The description of the free body diagram is as follows:

- m_1, m_2 --- mass at the mass points [kg/m]
- T --- tension on the wire [N]
- y_1, y_2--- displacement in the lateral direction [m]
- α, β, γ--- angle of deviation [degree]
- A, B --- distance between mass points [m]
- L --- total length of the wire [m]

From the above free body diagram, it is possible to derive these equations,

$$m_{\mathrm{i}}\ddot{y}_{\mathrm{i}} + T \sin \theta_i - T \sin \theta_{i+1} = 0$$

For small angles of deviation between two mass points,

$$\sin \theta_i \approx \tan \theta_i \approx \frac{y_{\mathrm{i}} - y_{\mathrm{i-1}}}{\frac{L}{n+1}}$$

3.2.1.3 Simulink scheme

The scheme (Figure 3-6) can be done using Simulink from the simplification of the above-mentioned equations. To run the Simulink model, input parameters are given in a MATLAB code (A7).

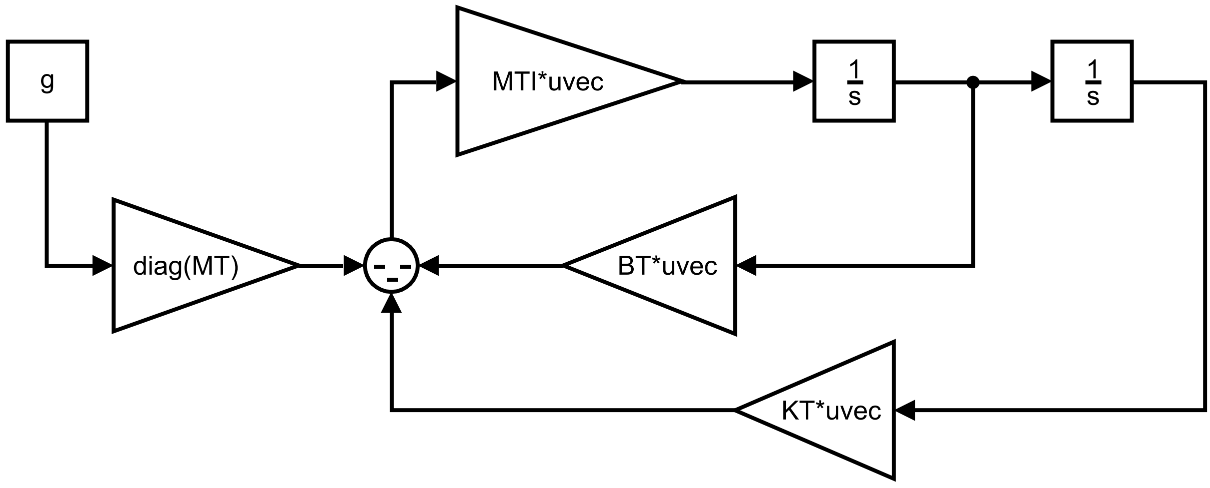


Figure 3-6 scheme for traverse vibration on the wire

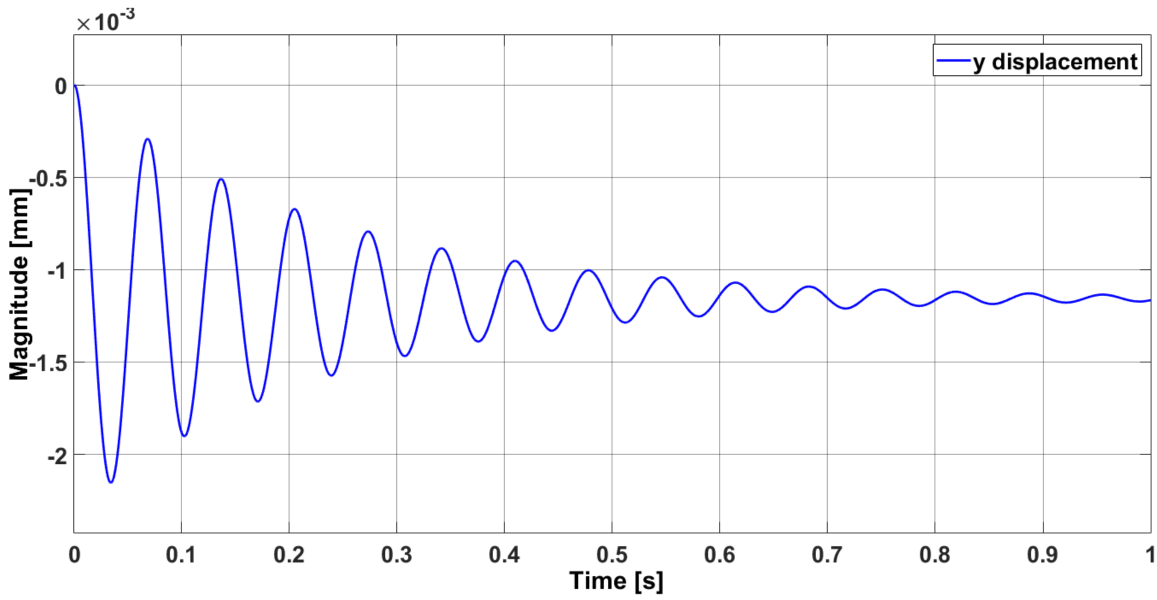


Figure 3-7 simulation result from the scheme

The simulation is done and the result of the is displayed in the Figure 3-7.

3.2.2 Longitudinal vibration on wire

Vibration along the wire's axial direction is known as longitudinal vibration [9][11]. This kind of vibrations is generally high in frequency.

3.2.2.1 Assumption and simplification

Here for calculating the longitudinal vibration, the lumped mass approach is used [15] [20]. Furthermore, this method has the following assumptions and simplifications:

- mass of the wire is equally divided into n number of mass points.
- displacement only in the axis of the wire was considered

3.2.2.2 Motion equations

The free-body diagram of the wire is drawn as follows,

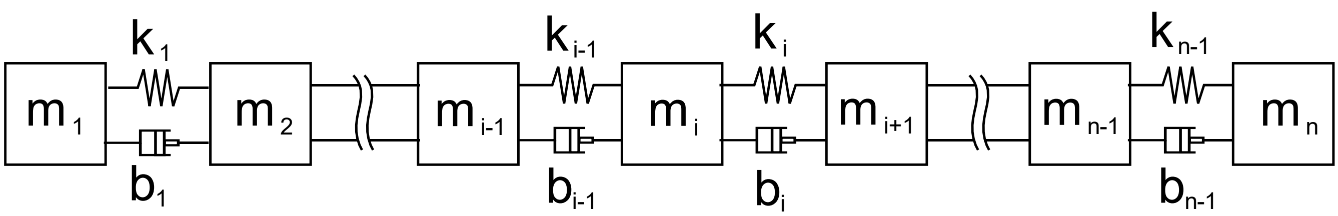


Figure 3-8 lumped model of wire

The description of the free body diagram is as follows:

- m_1, m_2 --- mass at the mass points [kg/m]
- T --- tension on the wire [N]
- x_1, x_2 --- displacement in the lateral direction [m]

- a --- distance between mass points [m]
- L --- total length of the wire [m]

From the above diagram, the equations of motion can be written. The general equation of motion is as follows,

$$M\ddot{x} + B\dot{x} + Kx = 0$$

where,

M --- mass matrix

B --- damping matrix

K --- stiffness matrix

x --- displacement vector

The matrix constructed by using the general motion equation of the i[th] node, which is described by using the following drawing,

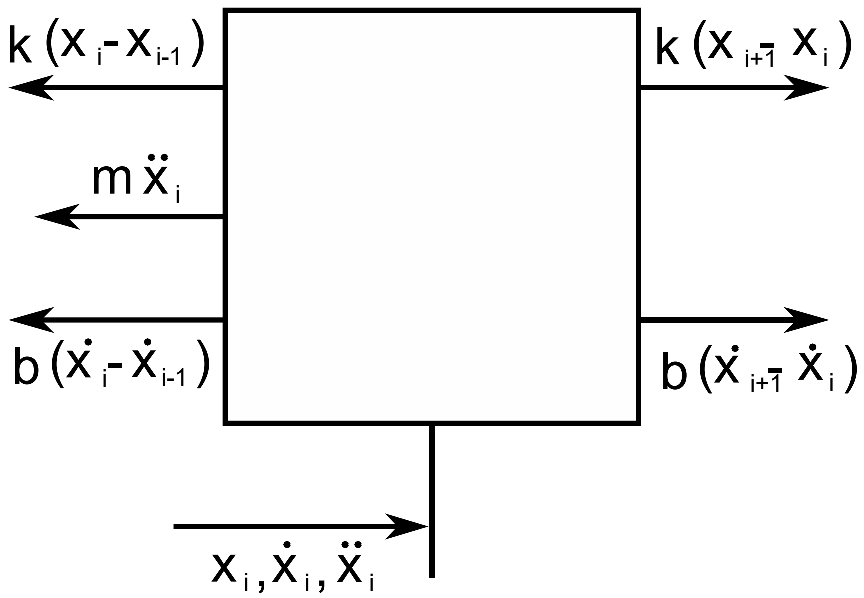


Figure 3-9 free body diagram of a mass point

$$M = m.\begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix} \qquad K = k.\begin{bmatrix} 1 & -1 & 0 \\ -1 & 2 & -1 \\ 0 & -1 & 1 \end{bmatrix} \qquad B = b.\begin{bmatrix} 1 & -1 & 0 \\ -1 & 2 & -1 \\ 0 & -1 & 1 \end{bmatrix}$$

where,

m --- mass of the wire [kg/m]

k --- stiffness of the wire[N/m]

b --- damping coefficient [Ns/m]

Here, the matrix is constructed for a three-mass system. Nevertheless, the size of the matrix varies depending on the number of mass points considered. It is important to note that the result's accuracy will increase with the rise in the number of mass points considered.

3.2.2.3 Simulink scheme

From the above-mentioned equations, the scheme (Figure 3-10) can be done using Simulink. To run the Simulink model, input parameters are given in the MATLAB code (A6).

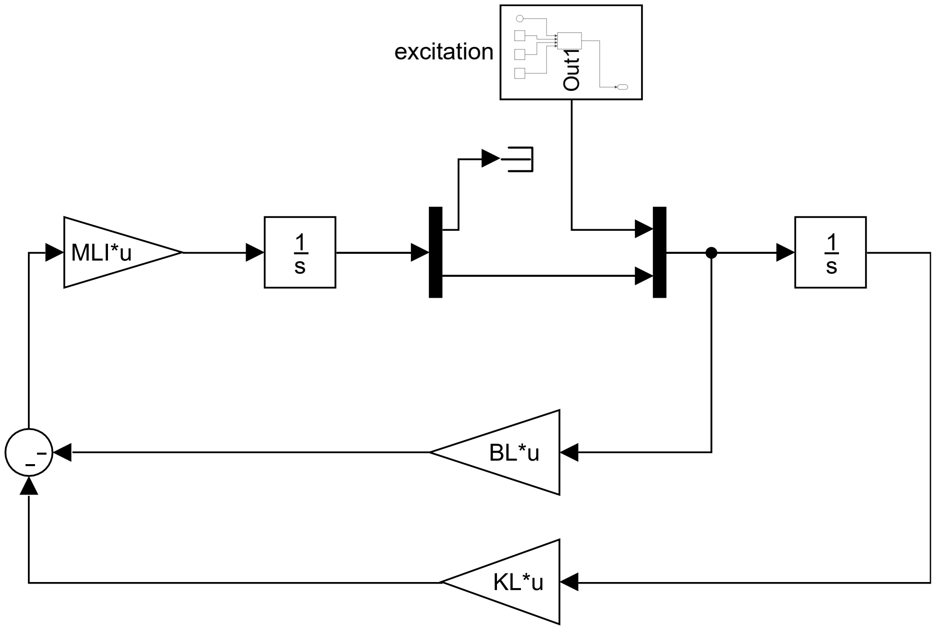


Figure 3-10 scheme of the longitudinal vibration

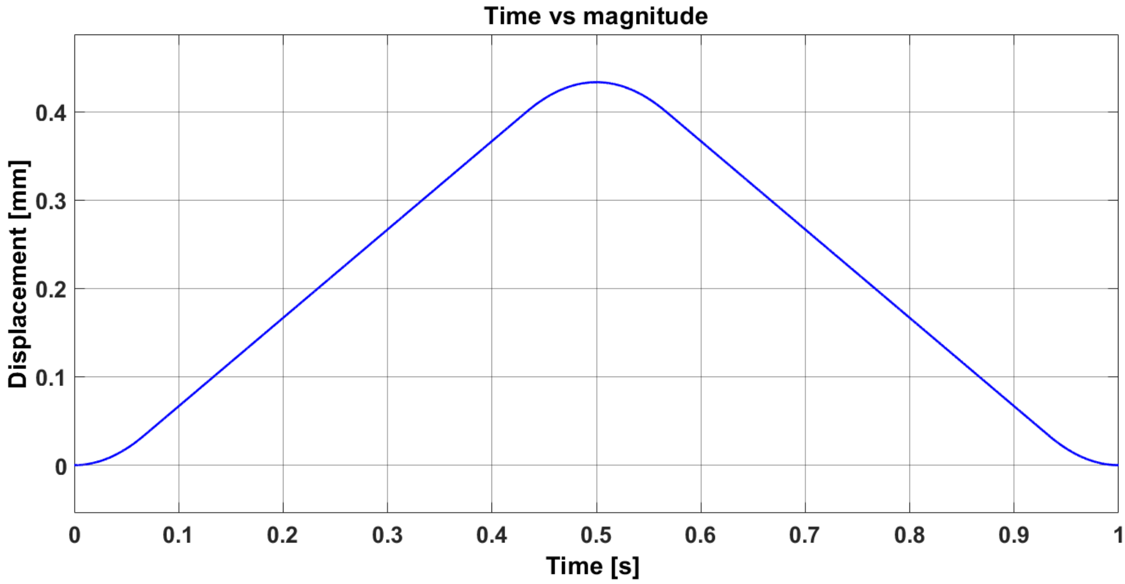


Figure 3-11 simulation result from the scheme

The simulation is done, and the result of the displacement of the mass points is displayed in Figure 3-11. The displacements of different mass points overlap each other, and it shows that the input parameters are equally divided between them.

3.2.3 Combining both lateral and longitudinal vibration

The vibration of both the lateral and longitudinal directions is combined to compare it to the real-world results. When the wire is excited in the longitudinal direction, it excites in the lateral direction, and it is necessary to combine the vibrations to compare it with the real world.

3.2.3.1 Assumption and simplification

Here for calculating the combination of lateral and longitudinal vibration, the lumped mass approach is used [12]. Furthermore, this method has the following assumptions and simplifications:

- mass of the wire is equally divided into n number of mass points.
- The angle of deviation between the two consecutive mass points is considered to be small.
- Damping due to air resistance is neglected.
- Bending resistance of the wire is considered as zero.

3.2.3.2 Motion equations

The free-body diagram is drawn with both lateral and longitudinal forces acting on it for a reason mentioned above. It is as follows (Figure 3-12),

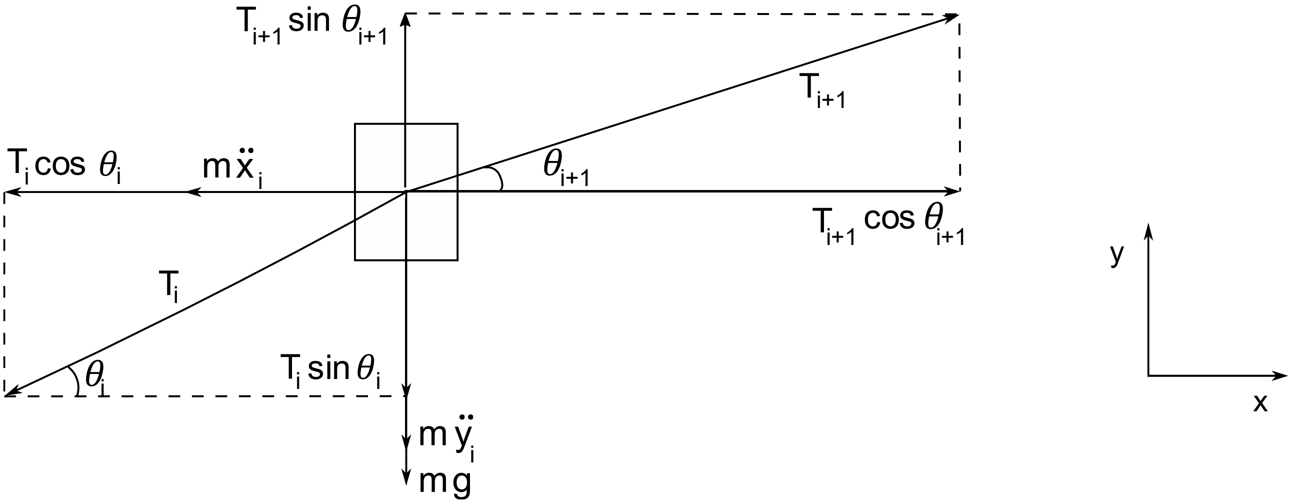


Figure 3-12 free body diagram of combined lateral and longitudinal forces

The description of the free body diagram is as follows:

- T_1, T_2, T_3 --- tension on the wire concerning the mass points [N]
- $\theta_1, \theta_2, \theta_3$ --- angle of displacement [degree]
- a --- distance between mass points [m]
- x_1, x_2, x_3 --- displacement along longitudinal direction [m]
- y_1, y_2, y_3 --- displacement along lateral direction [m]

From the above free body diagram, the following motion equations can be written,

$$T_{\mathrm{i}} \cos \theta_{\mathrm{i}} + m\ddot{x} - T_{\mathrm{i+1}} \cos \theta_{\mathrm{i+1}} = 0$$

$$T_{\mathrm{i}} \sin \theta_{\mathrm{i}} + m\ddot{y} + mg - T_{\mathrm{i+1}} \sin \theta_{\mathrm{i+1}} = 0$$

The values of T_1, T_2, T_3 and $\theta_1, \theta_2, \theta_3$ can be calculated by using the following derivations from the drawing,

$$T_i = k\,(\sqrt{(a + (x_i - x_{i-1}))^2 + (y_i - y_{i-1})^2} - a)$$

$$\theta_i = tan^{-1}\left(\frac{y_i - y_{i-1}}{a + (x_i - x_{i-1})}\right)$$

3.2.3.3 Simulink scheme

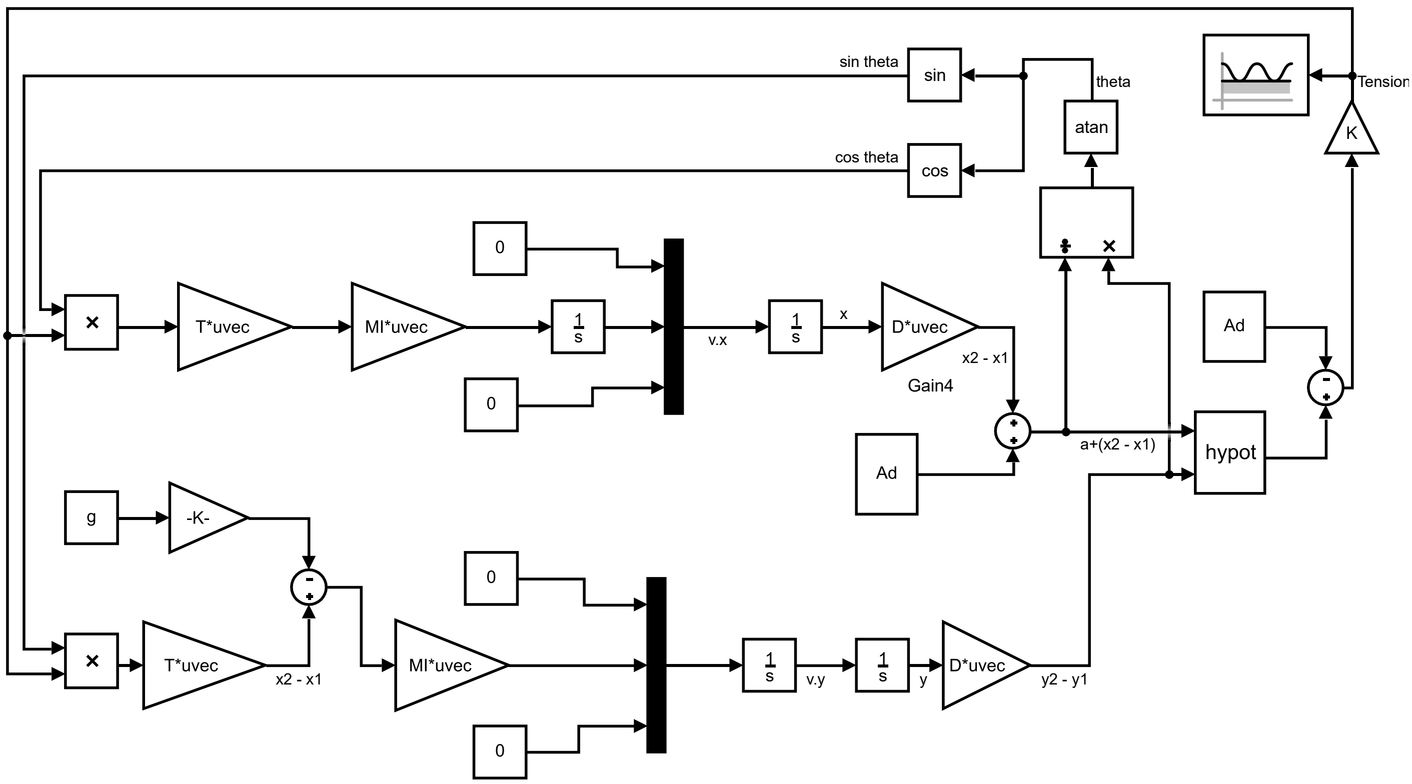


Figure 3-13 scheme for the combined vibration of longitudinal and lateral directions

The Simulink contains the combination of both lateral and longitudinal vibrations on the wire. The scheme above-mentioned gives the output similar to the results of lateral and longitudinal schemes when the input parameters and excitation are properly given.

3.3 Natural frequency

The frequency at which the system vibrates even without any external force is known as natural frequency. It is crucial to find the natural frequency of the system to avoid a possible failure [1][5][8].

In general, the vibration should be avoided to increase machine health and performance. In this specific application, the vibration on the wire must be kept as minimal as possible, or it should be avoided. Because in this specific application, a thin layer of specific fluid is coated on the wire while it is in operation. The thin layer of the fluid is coated on the wire to meet the needs of the machine. If the wire vibrates, the fluid coated on the wire starts to drip away from the wire, which is not desirable. So, it is vital to reduce the wire's vibration to reduce the wire's dripping of fluid.

In a wire, the natural frequency is dependent on tension on the wire. The formula [9] to calculate the natural frequency is as follows,

$$fn = \frac{n}{2L}\sqrt{\frac{F}{m}}$$

where

fn---natural frequency [Hz]

n---mode number

L---length of the wire [m]

F---tension on the wire [Nm]

m---mass/length of the wire [kg/m]

The above formula results will provide us with all the natural frequencies up to n number of modes. It gives the value of all lateral natural frequencies. Minimal vibration is achieved when the excitation frequency is not the same as the natural frequency. In this specific application, it is easy to vary the wire's tension. Thereby varying the natural frequency can be useful in avoiding the excitation frequency to prevent vibration. The parameters used for the plot (Figure 3-14) are length of the wire = 0.56 m, number of modes = 10, tension range = 0 N to 1 N

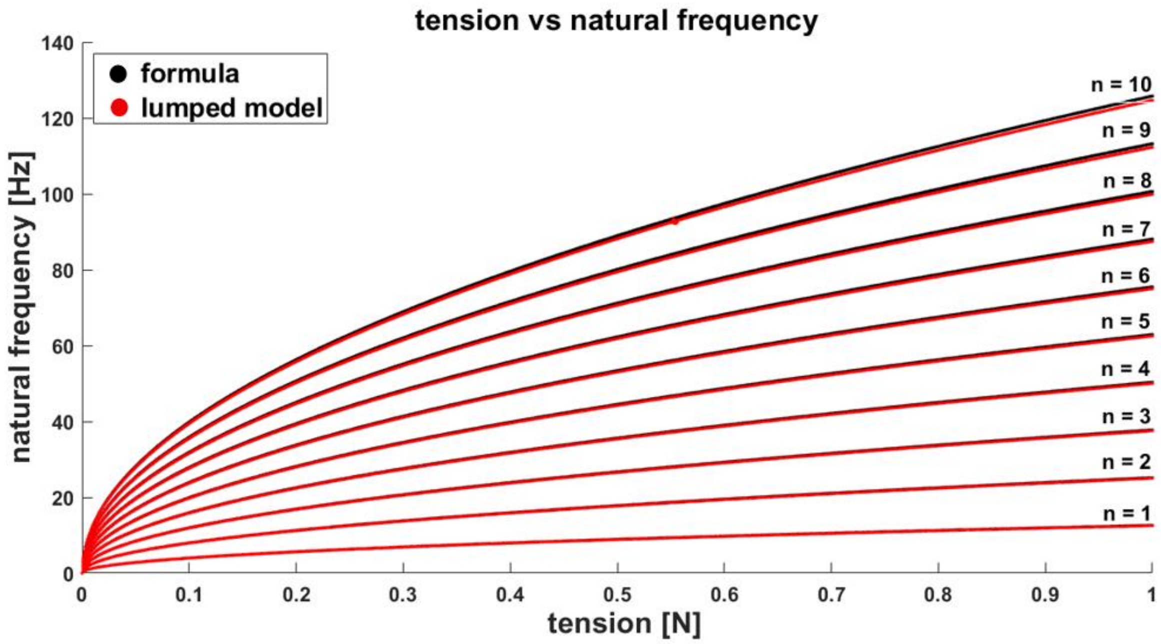


Figure 3-14 comparison of natural frequencies from formula and lumped model method

3.4 Measurement

The wire is set up between the pulleys. The motor holds one end of the wire, and the mass's tension holds the other end. An accelerometer is used to find the natural frequency and damping factor of the string experimentally. Positioning the accelerometer on the wire was done on the pulley (Figure 3-1). The accelerometer used can measure acceleration from three directions.

Furthermore, when the system is at equilibrium, the wire is plucked to create some vibration. The vibration created is recorded in terms of acceleration with respect to time by the accelerometer in all three directions. Then, the data is transferred via the data acquisition unit into the computer. The measurement is done multiple times to verify the data collected.

The data is collected for a different set of tensions on the string.

3.5 Calculation & Result

From the acceleration data collected, it is possible to compute the damping coefficient of the wire. Firstly, the acceleration recorded from the measurement and the simulation of lateral vibration were plotted. Input parameter on the lumped model is closely matched with that of the measurement. Then, the wire's damping coefficient value is manually altered until both the curves match each other (Figure 3-15). It was a trial-and-error process, and the chance of error is less in this approach. Since the accelerometer was not positioned on the wire, it is necessary to match the simulation peaks with every second peak of the measurement. The plot below shows that both the curves are almost matching and the damping value is approximated.

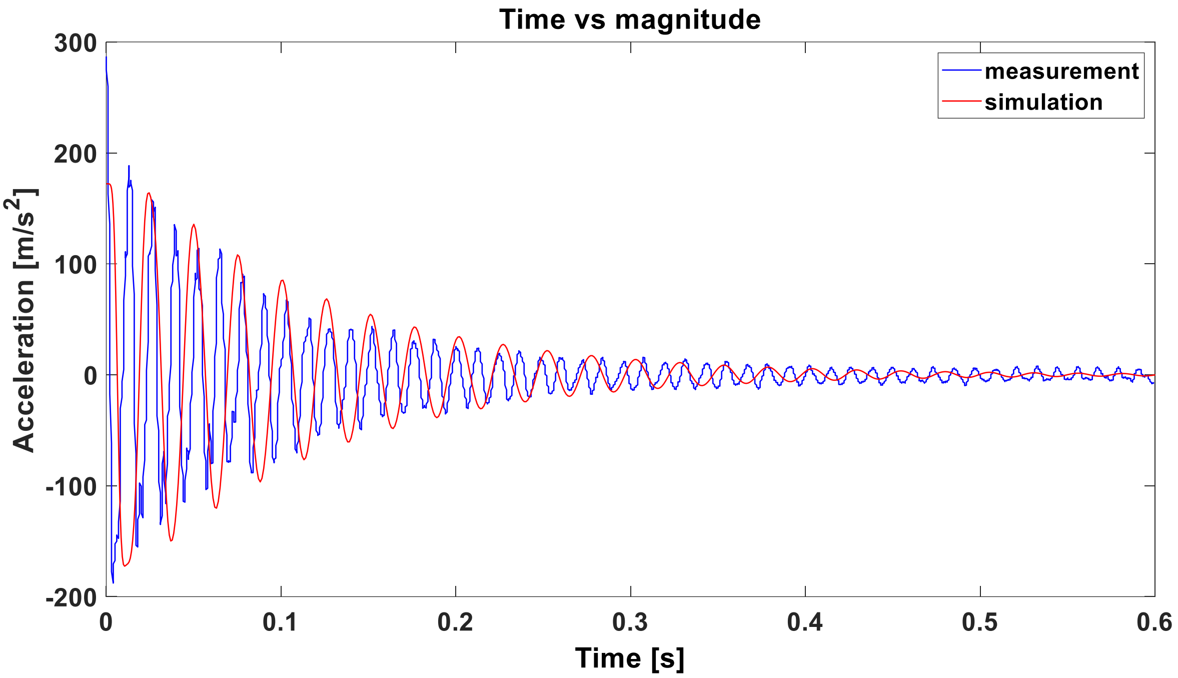


Figure 3-15 comparison between measurement and simulation

As a result of trial and error & comparison, the wire's actual damping coefficient was found to be 0.003 Ns/m.

Furthermore, the fast Fourier transform on the acceleration from the accelerometer is done to get the natural frequencies. Then, the natural frequencies result from the formula, lumped model, and measurements are compared. In comparison, it is observed that the results from the formula and the mathematical model match (Figure 3-14). Whereas the result from the measurement is found to be twice the value of the first natural frequency. The following theory can explain the difference in result:

- The accelerometer is placed on the pulley and not the wire while measuring (Figure 3-3).
- Since the pulleys hold the wire during vibration, the pulley is being pulled twice by the wire for one oscillation, explained in detail below (Figure 3-16).

Based on the theory mentioned above, we can say that the natural frequencies from all these sources are matching each other.

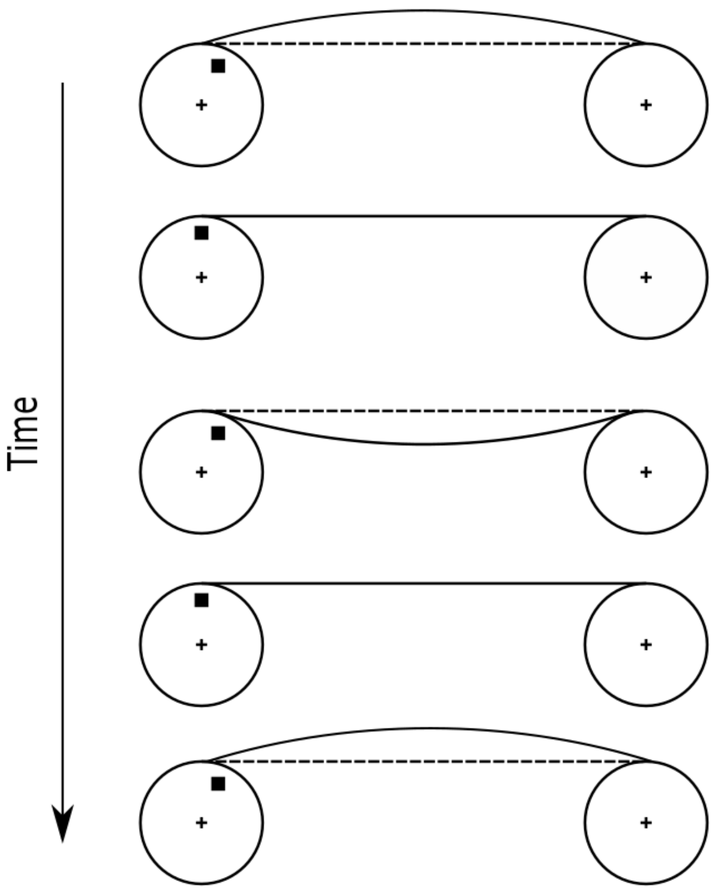


Figure 3-16 vibration of wire and position of the accelerometer (black square)

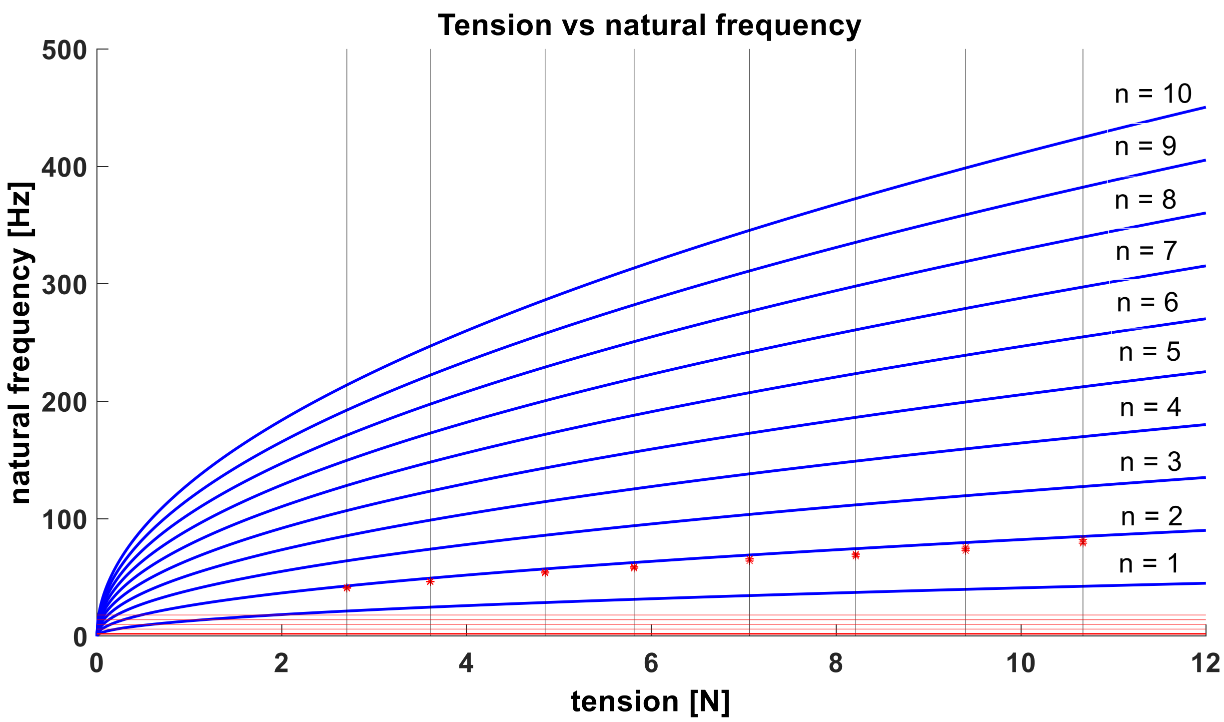


Figure 3-17 tension vs natural frequency compared with measurement values (red stars) and excitation frequency (red lines)

The parameter used for the plot (Figure 3-17) is the length of the wire = 0.56 m, the number of modes = 10, tension range = 0 N to 12 N. More details on plotting the above figure can be seen in supplements(A5). The red lines define the excitation frequencies calculated by performing FFT on the excitation functions (A4). The excitation frequencies are well below the wire's first natural frequency and can be seen in Figure 3-17. The FFT provides all the natural frequency of the excitation functions. However, the frequencies above the threshold (A2) is critical to be considered because the frequencies higher than the threshold are likely to induce dripping of fluid from the wire. Since the excitation frequencies are below the first natural frequency, the vibration can be eliminated if tension is approximately higher than 2 N for this specific excitation frequency used.

CHAPTER 4: Torsion spring system

4.1 Description

Spring that stores energy when twisted along an axis is known as torsional spring [2]. The torsion spring is modernized according to the needs, and some types are as follows:

- constant force spring
- constant torque spring
- power springs

Here for the design, the constant torque spring [3][4] (Figure 4-1) is used. The constant torque spring provides constant torque and a more significant number of better functional turns than other springs.

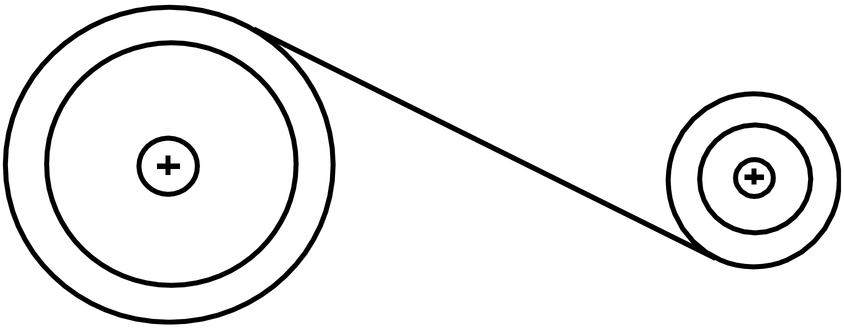


Figure 4-1 constant torque spring

This idea is selected mainly due to its simplicity, as it can be easily installed with minimal modifications. It also satisfies the torque required for the operation. The CAD image (Figure 4-2) below shows that the spring can be accommodated in a small area and requires limited and straightforward essential parts.

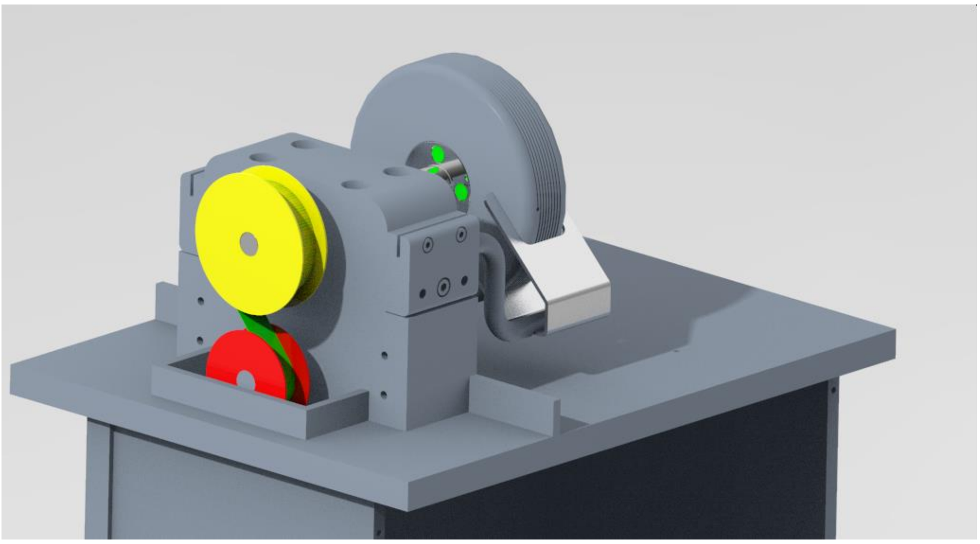


Figure 4-2 cad model of the torsion spring system

One end of the spring is attached to the primary pulley shaft, whereas the other is attached to another shaft below the main shaft. The rotation caused by the motor rotates the pulley through the wire. The pulley, in turn, rotates the spring as it is directly coupled to the pulley shaft. The spring winds on to the pulley shaft, creating tension in the other end of the spring. The tension created helps the system complete the other half of the cycle, thereby satisfying the machine's needs.

The torsion spring system should work well for similar applications. Nevertheless, it is not suitable for this specific application, as the number of cycles of this specific application is very high. The system will work great, but the lifetime of the spring will be significantly less. However, the mathematical model of the system is done to know how it behaves dynamically.

4.2 Mathematical model

A mathematical model is advantageous in understanding the dynamic behaviour of the system. The following are the steps taken for the creation of the mathematical model.

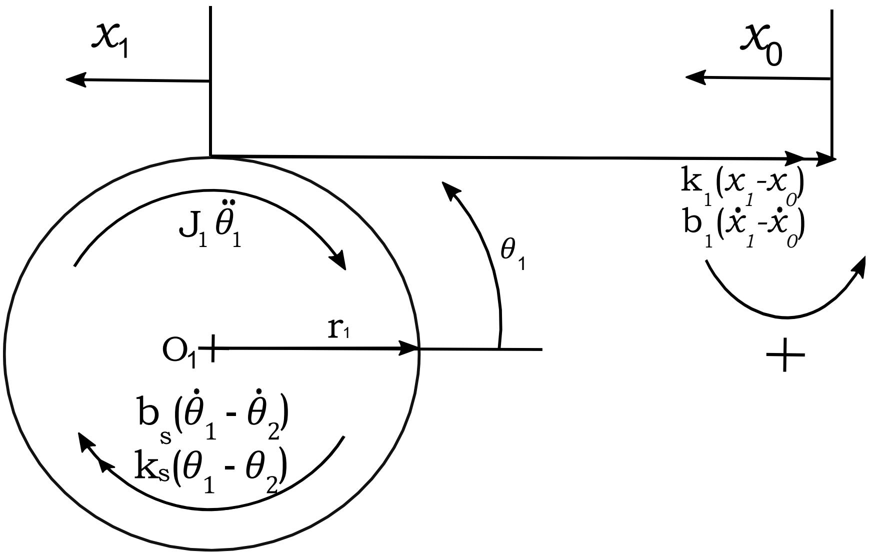
4.2.1 Assumption and simplification

Firstly, to create the mathematical model, the following are considered as assumptions and simplification:

- the pulley is considered rigid
- mass of the wire on the pulley is neglected
- centrifugal force is neglected due to the low speed of rotation

4.2.2 Motion equation of the system

Based on the assumptions mentioned above, the two bodies' free-body diagrams (pulleys) are drawn.



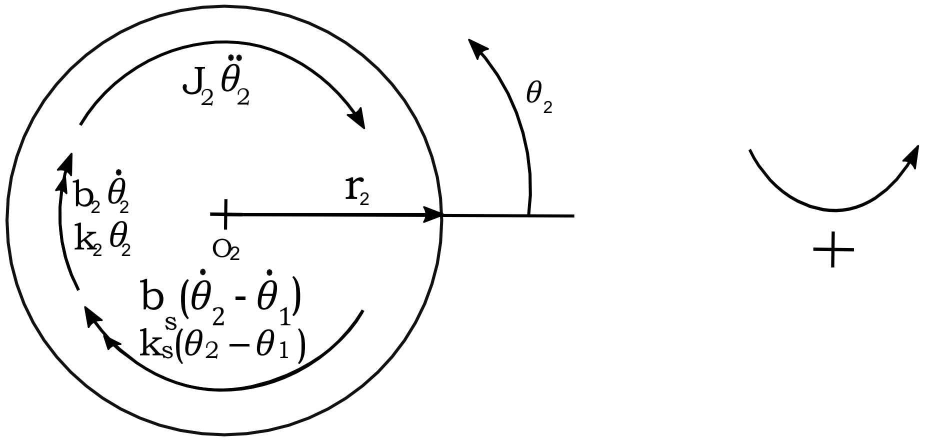


Figure 4-3 free body diagrams of the bodies with the forces acting on pulleys

From the above the free body schematics (Figure 4-3), the following equations are written and they are,

$$J_1\ddot{\theta}_1 + k_1.r_1.(x_1 - x_0) + b_1.r_1.(\dot{x}_1 - \dot{x}_0) + k_s(\theta_1 - \theta_2) + b_s(\dot{\theta}_1 - \dot{\theta}_2) = 0$$

$$J_2\ddot{\theta}_2 + k_2\theta_2 + b_s\dot{\theta}_2 + k_s(\theta_2 - \theta_1) + b_s(\dot{\theta}_2 - \dot{\theta}_1) = 0$$

The description of the free body diagram is as follows:

- J_1, J_2 --- moment of inertia of the drums respectively [kg m^2]
- θ_1, θ_2 --- displacement along the axis of rotation [degree]
- r_1, r_2 --- radius of the drums [m]
- k_1, k_2 --- stiffness of the wire and spring respectively [N/m]
- k_s --- stiffness of the shaft [N/m]
- b_1, b_2 --- damping coefficient of the wire and spring respectively [Ns/m]
- b_s --- damping coefficient of the shaft [Ns/m]
- x_0, x_1 --- displacement along x direction [m]

4.2.3 Simulink scheme

Based on the above equations, the Simulink scheme is created. It is needed to give input parameters of the system to make the scheme work. The input parameters, such as radius of the pulley, stiffness of pulley and shaft, damping coefficient and mass, are given in a MATLAB code [10]. The code offers excellent flexibility to find the proper input for the desired output. The dynamic behaviour of the system is found using the scheme. The Simulink scheme is as follows.

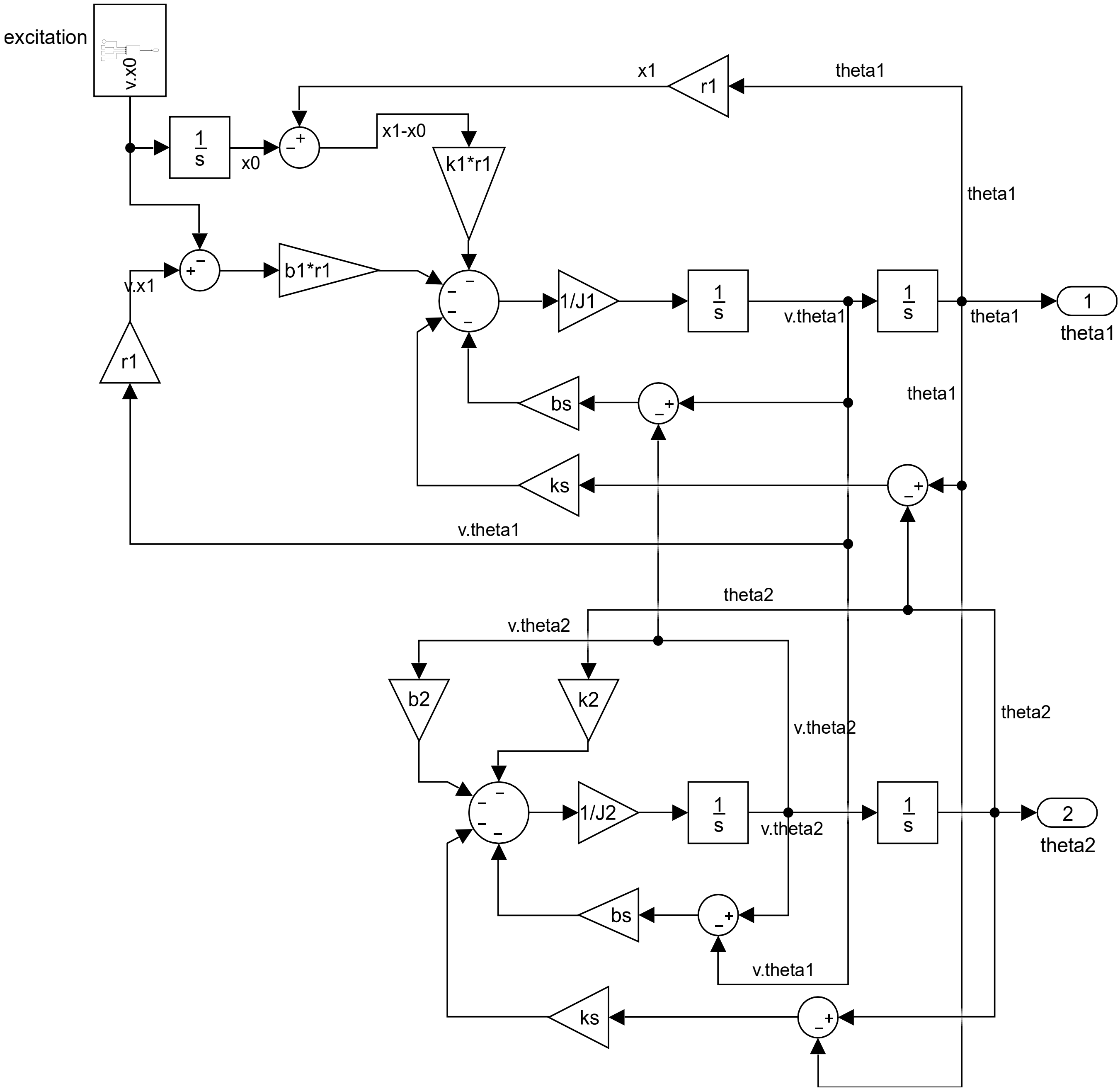


Figure 4-4 scheme of torsion spring system

4.3 Simulation

The simulation is done using the Simulink scheme (Figure 4-4) mentioned above scheme and input parameters. The simulation results will give us the dynamic behaviour of the system. From the plot below, we can visualize each of the parts' motion profile in the system. The MATLAB also gives us great flexibility for the analysis of dynamic behaviour.

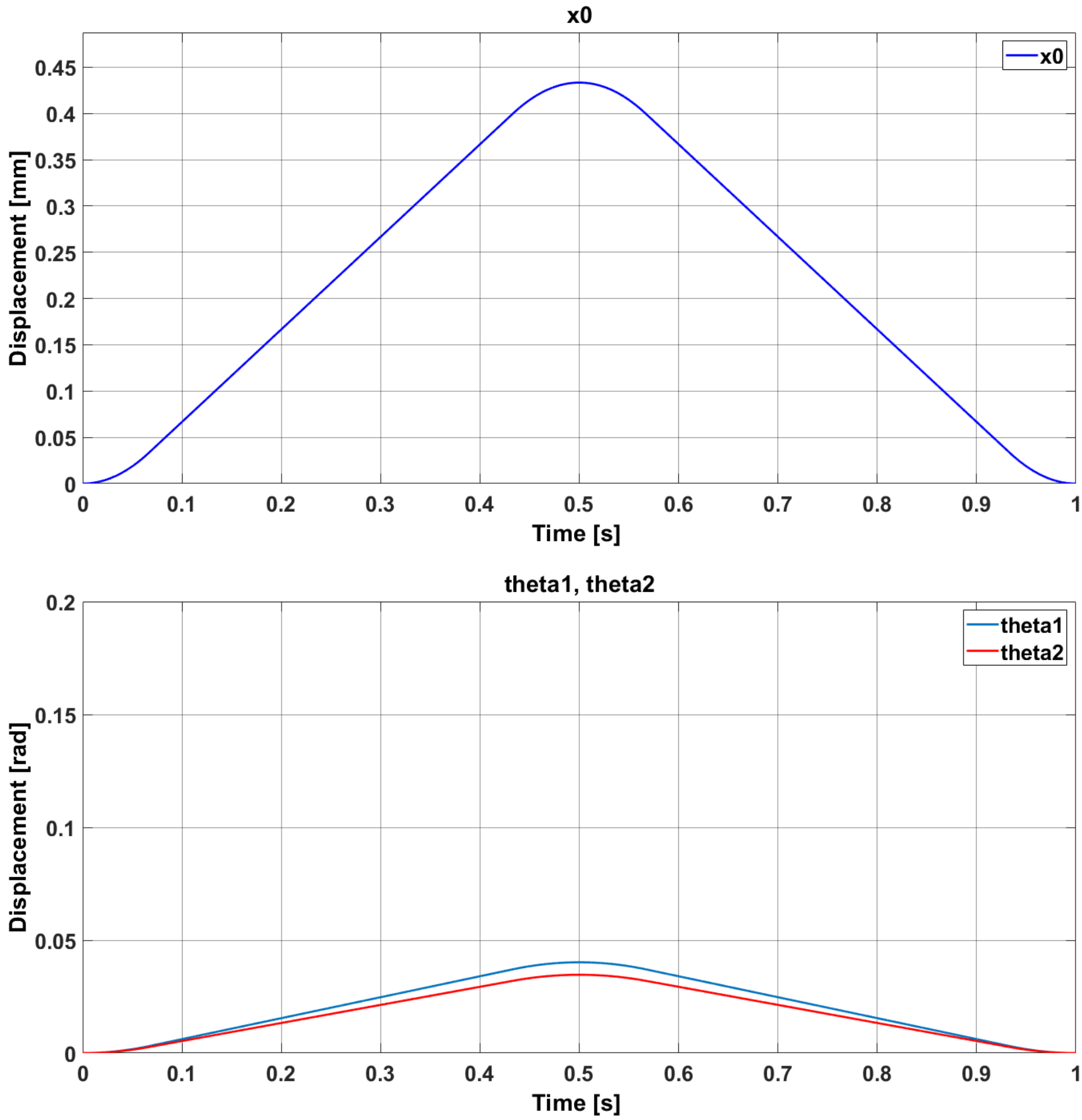


Figure 4-5 simulation result of the torsion spring system

4.4 Discussion

From the above results (Figure 4-5), it can be seen that the proposed system will satisfy the needs of the machine. Despite its various advantages, the idea was not tested. Because the spring used here has a life cycle of around 20,000 cycles, which might be useful in general applications, however, it is found that the spring is not suitable for this specific application as this specific application takes a more significant number of cycles quickly, which is not satisfied by the spring over time.

As a result, the torsion spring system is not suited for this specific application despite being simple, compact and efficient.

CHAPTER 5: Gravitational potential energy system

5.1 Description

The potential energy is the energy stored in an object when it is at rest. The system's potential energy can be changed by varying the height at which the object is placed because we know that the potential energy is directly proportional to the height.

$$PE = mgh$$

where,

PE --- potential energy [J]

m --- mass of the object [kg]

g --- acceleration due to gravity [m/s^2]

h --- height of the object from the ground [m]

Based on potential energy storage, the idea is being proposed. The proposed idea consists of a hanging mass connected to the pulley shaft with the ring's help. The ring is firmly attached to the pulley shaft. A wire connecting the mass is attached to the hole in the ring. It is in connected such a way that, when the shaft rotates, the ring also rotates. Then, the wire connecting the mass is wound over the shaft during the first half of the cycle driven by the servo motor. By winding the wire over the shaft, the height of the mass is increased. So, the potential energy is increased, stored, and used for the rest of the cycle. In the second half of the cycle, the mass's potential energy is used to drive the system where the servo motor provides tension. For the whole cycle, the motor acts only in position mode, and torque mode is disabled.

5.2 CAD model

The proposed idea is converted into a cad model (Figure 5-1). The primary constraint for the design is to make it compact. If the design is compact, the system can be replaced in the same location as the servo motor in the current situation without significant modifications.

Firstly, the designing started with the mass's conceptualisation, which is to be accommodated inside the box [6][13]. The potential energy required for the system is calculated. The required potential energy can be generated in many ways, but here two approaches are considered. They are low mass and high height, high mass and low height.

Since here, we have limited space high mass, and low height approach is used. The main reason to use this approach is to keep the mass's acceleration to a minimum compared to the acceleration due to gravity. If the displacement is low, the mass's acceleration will be less, which will keep the total downward force on the mass to fluctuate within minimal limits. Then, the stability of the system will increase.

Based on the calculation of stroke, mass, and available space, it is found that the system with pulleys will suit them the best. So, the system with pulley (2:1 ratio) is used. The ratio is selected because of the stroke available inside the box. However, the ratio used will also double the mass to be hanged. All these factors are taken into account, and the designing is done. The dimensions and calculation are made for a specific diameter of the ring. If the ring or the primary shaft diameter is changed, the dimensions and the pulley's ratio may require a change. The pulleys are located at two locations on the mass, which increases the stability of the system. The mass also can be hanged or moved without any guidance to any position due to this design.

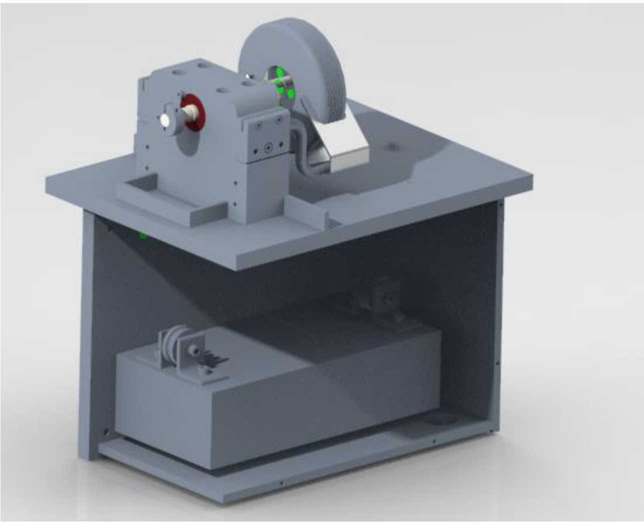


Figure 5-1 cad model of the gravitational potential system

5.3 Mathematical model of the system

The mathematical model helps in understanding the dynamic behaviour of the system. The following are the steps taken for the creation of the mathematical model.

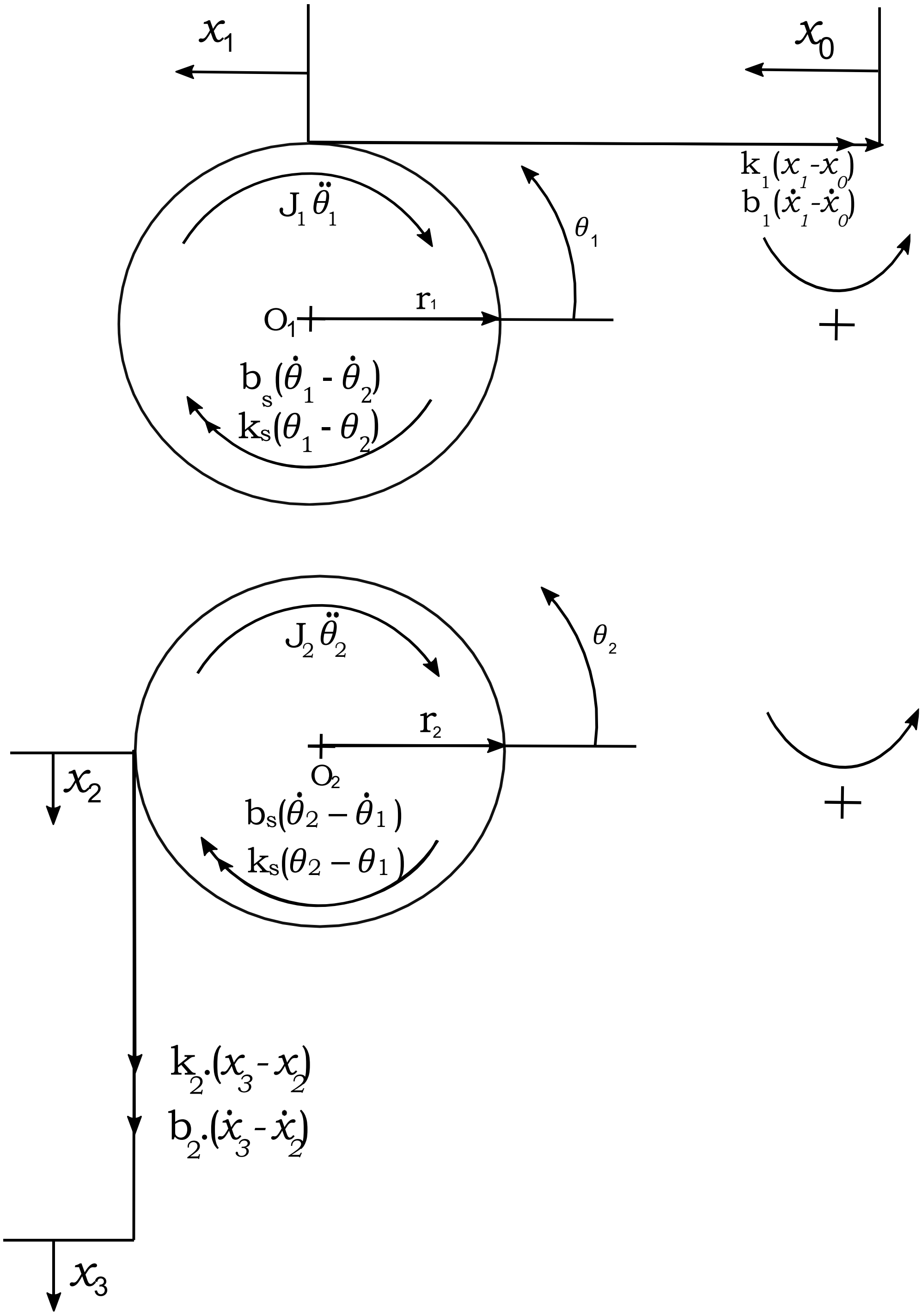
5.3.1 Assumption and simplification

Firstly, to create the mathematical model, the following are considered as assumptions and simplification,

- the pulley is considered rigid

- mass of the wire on the pulley is neglected
- centrifugal force is neglected due to the low speed of rotation

5.3.2 Motion equation of the system



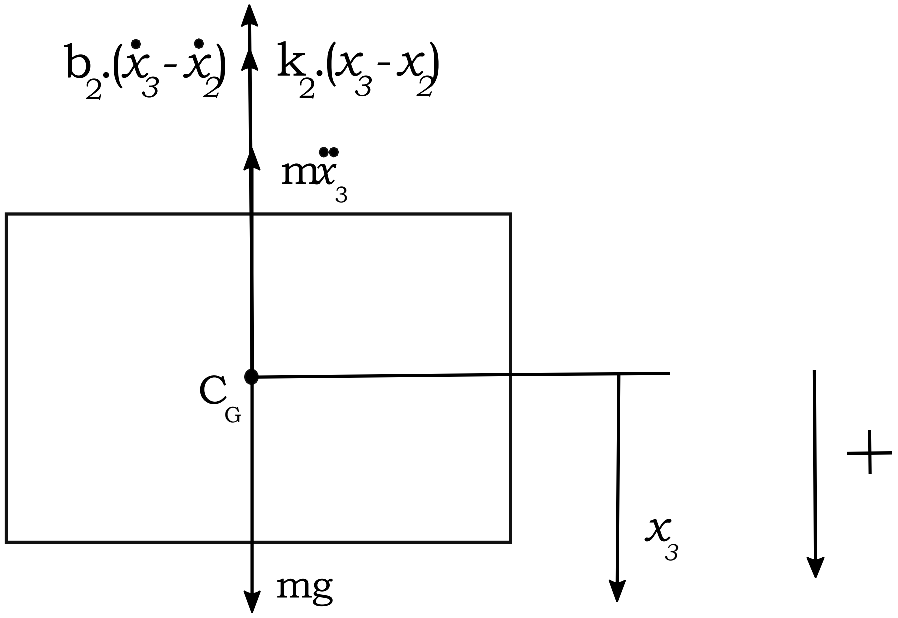


Figure 5-2 free body diagrams of the bodies with the forces acting on pulleys and mass

From the above the free body schematics (Figure 5-2), the following equations are written, and they are,

$$J_1\ddot{\theta}_1 + k_1.r_1.(x_1 - x_0) + b_1.r_1.(\dot{x}_1 - \dot{x}_0) + k_s(\theta_1 - \theta_2) + b_s(\dot{\theta}_1 - \dot{\theta}_2) = 0$$

$$J_2\ddot{\theta}_2 + k_2.r_2.(x_3 - x_2) + b_2.r_2.(\dot{x}_3 - \dot{x}_2) + k_s(\theta_2 - \theta_1) + b_s(\dot{\theta}_2 - \dot{\theta}_1) = 0$$

$$m\ddot{x}_3 + k_2(x_3 - x_2) + b_2(\dot{x}_3 - \dot{x}_2) - mg = 0$$

The description of the free body diagram is as follows:

- J_1, J_2 --- moment of inertia of the drums respectively [kg m^2]
- m --- hanging mass [kg]
- θ_1, θ_2 --- displacement along the axis of rotation [degree]
- r_1, r_2 --- radius of the drums [m]
- k_1, k_2 --- stiffness of the wire [N/m]
- k_s --- stiffness of the shaft [N/m]
- b_1, b_2 --- damping coefficient of the wire [Ns/m]
- b_s --- damping coefficient of the wires [Ns/m]
- x_0, x_1, x_2, x_3 --- displacement along x directions respectively [m]
- g --- acceleration due to gravity [m/s^2]

5.3.3 Simulink scheme

Based on the above equations, the Simulink scheme (Figure 5-3) is created. It is needed to give input parameters of the system to make the scheme work. The input parameters, such as radius of the pulley, stiffness of pulley and shaft, damping coefficient and mass, are given in a

MATLAB code [10]. The code offers excellent flexibility to find the proper input for the desired output. The dynamic behaviour of the system is found using the scheme. The Simulink scheme is as follows.

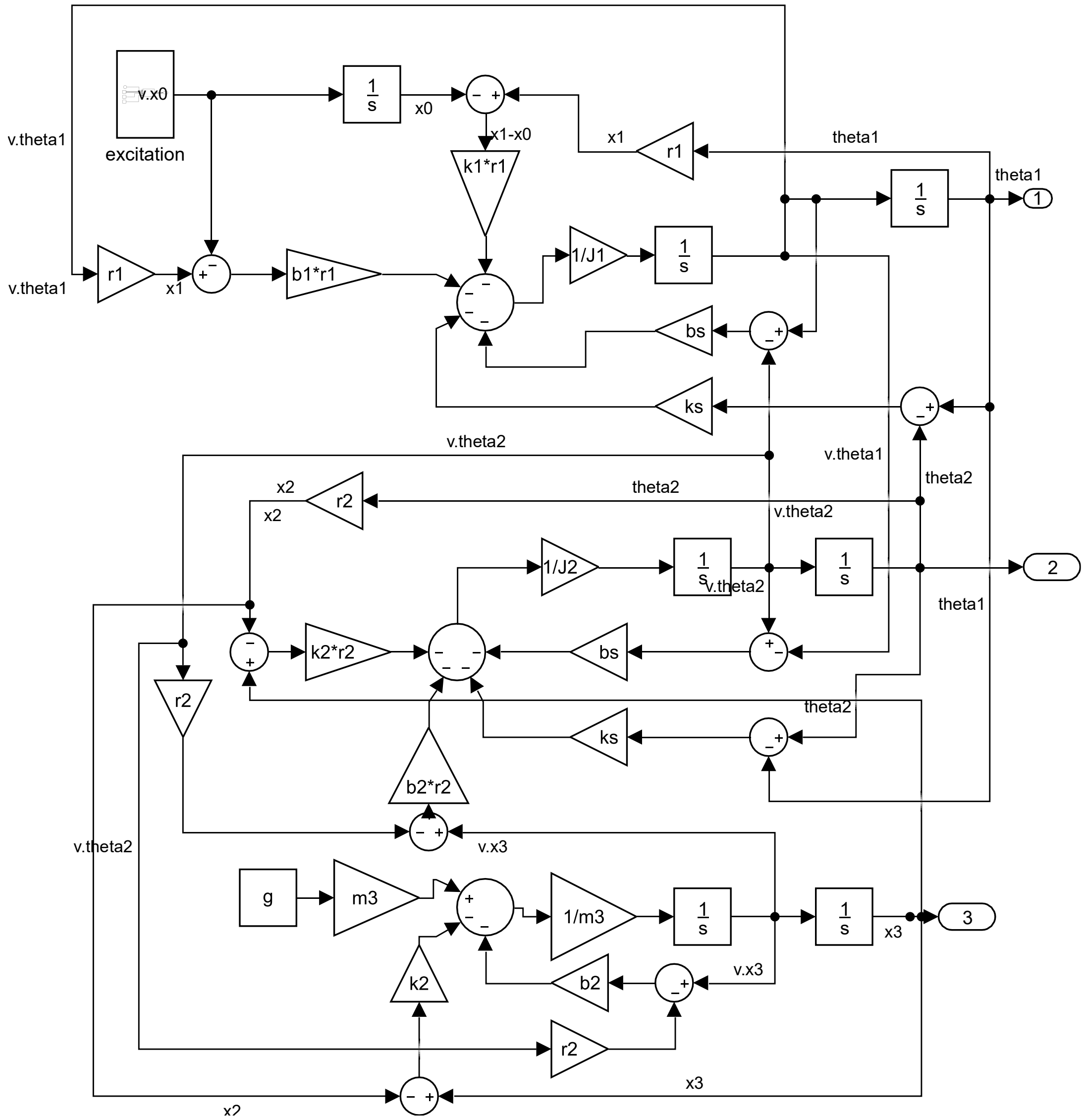


Figure 5-3 scheme of the equations of the gravitational potential system

5.4 Simulation

The input parameters are compiled in a MATLAB code and above-mentioned scheme (Figure 5-3) is done. The dynamic behaviour of the system is given by the simulation results. Motion profile of each part can be visualized in the plots below (Figure 5-4). The MATLAB also gives us great flexibility for the analysis of dynamic behaviour.

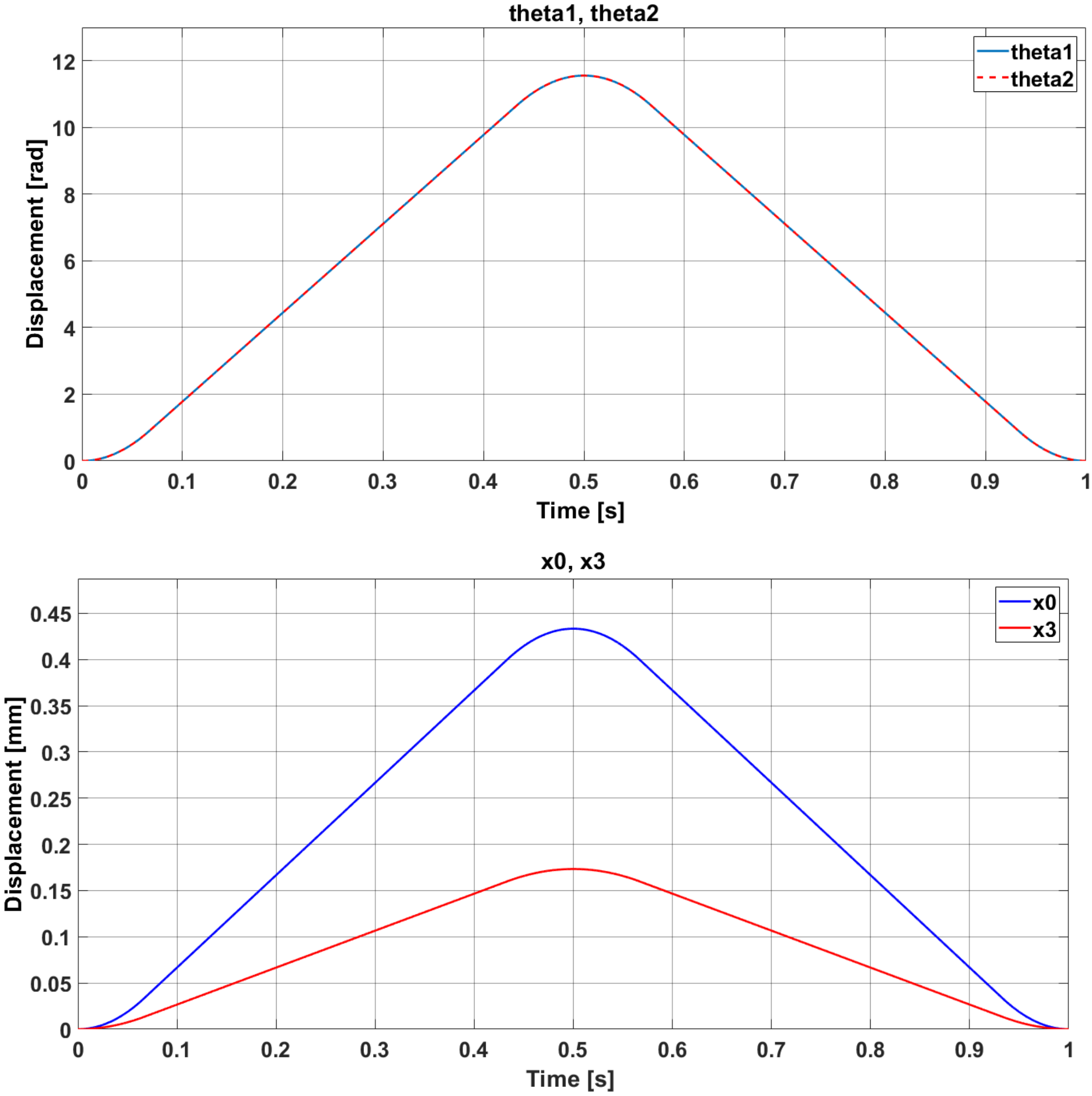


Figure 5-4 simulation result of the gravitational potential system

5.5 Measurement

The system is manufactured and tested to verify whether the system's dynamic behaviour satisfies the machine's needs. Though the simulation results (Figure 5-4) are acceptable, it is always better to test them in real-world conditions.

5.5.1 Methodology

This idea's manufacturing requires only minimal changes like removing the connections between the shaft and servo motor, adding some pulleys and mass. Though the changes were minimal, the mass to be hanged was kept outside the machine as it offers more flexibility for tuning the system while testing.

Figure 5-5 ring used to connect the shaft and the wire with hanging mass

The hanging mass is connected to the shaft using a cable and the ring (Figure 5-6 & Figure 5-7). The ring used is, as shown in Figure 5-5. When the motor rotates in one direction for the first cycle, the wire connecting the mass gets wound onto the shaft. Thereby raising the mass, and the potential energy is stored. The potential energy stored is then used when the next cycle starts. During the next cycle, the stored potential energy is used to move the pulleys, and the wire and the motor still operate in position mode to gain control over the position of the wire.

From this setup, the damping and the wire's natural frequency can be found from the data given by the accelerometer

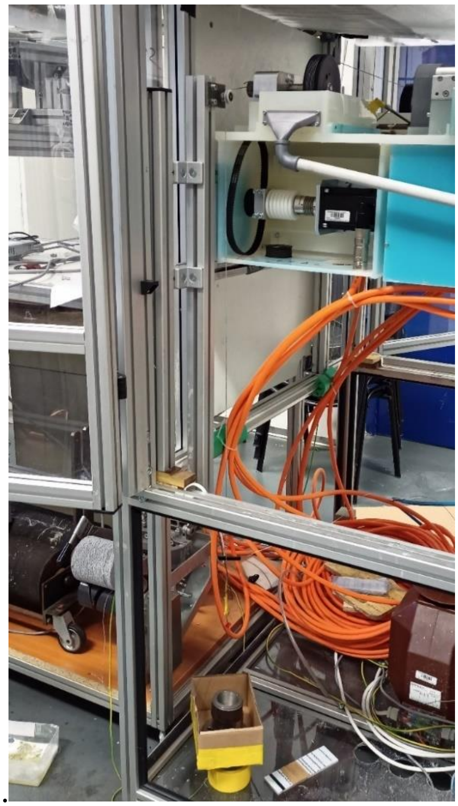


Figure 5-6 experimental setup of gravitational potential system

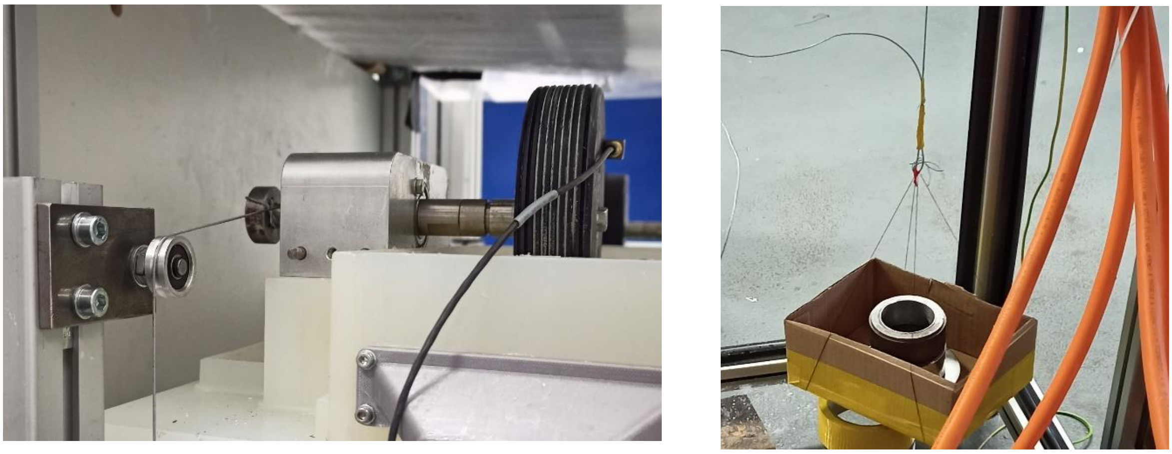


Figure 5-7 wire with hanging mass is connected to the shaft by using the ring (left), the hanging mass on the wire (right)

5.5.2 Data acquisition

The gravitational potential energy system uses mass for providing tension and the energy for the system to work. So, the idea is to vary the tension on the wire to prevent vibration on the wire. Small blocks of masses are used. Measuring the mass of the blocks is done for the calculation of tension provided by it. Totally eight sets of masses were used in the measurement. The masses used and the respective tension provided are as follows:

Number	Mass [kg]
m1	2.71
m2	3.62
m3	4.86
m4	5.83
m5	7.08
m6	8.23
m7	9.43
m8	10.70

Table 1 masses used

Then for each set of mass, readings of position, velocity, and torque are measured from the servo motor at the other end. For each set of mass, four different speeds were used, and the readings are taken. Other than the listed masses (Table 1), a measurement named ‘m0’ was made without any mass and wire. m0 reading was taken to know the actual values, so that the other readings can be compared.

5.6 Results

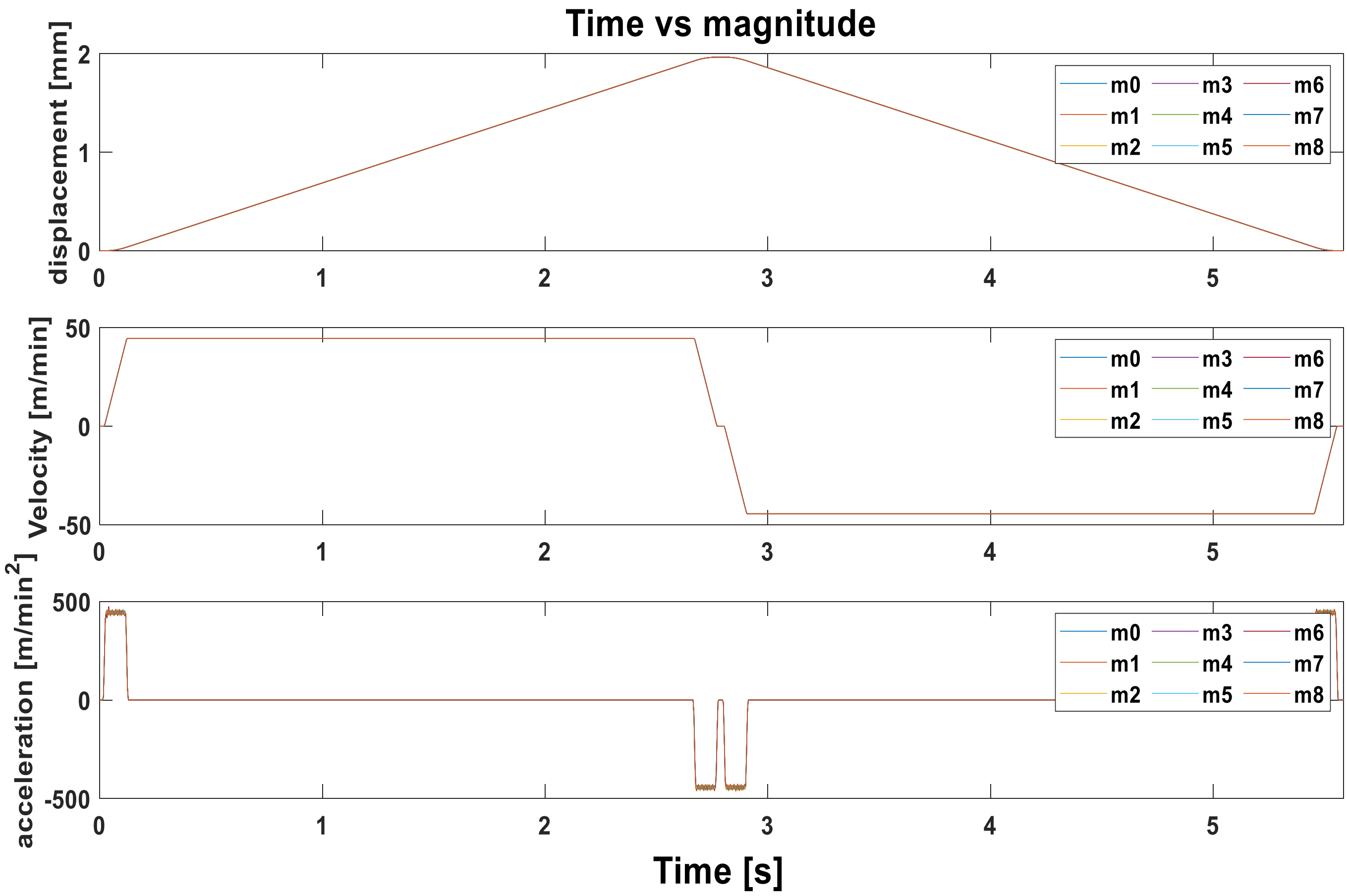


Figure 5-8 data from measurement

From the plot (Figure 5-8), the data from the servomotor can be visualized. It is observed that the position and velocity plots look similar despite varying speeds and tension compared with actual curves. The results look similar to the current situation, as the servo motor handles the situation well as it acts in position mode for the whole cycle. Actually, the different masses' results overlap each other, and therefore it seems like one curve.

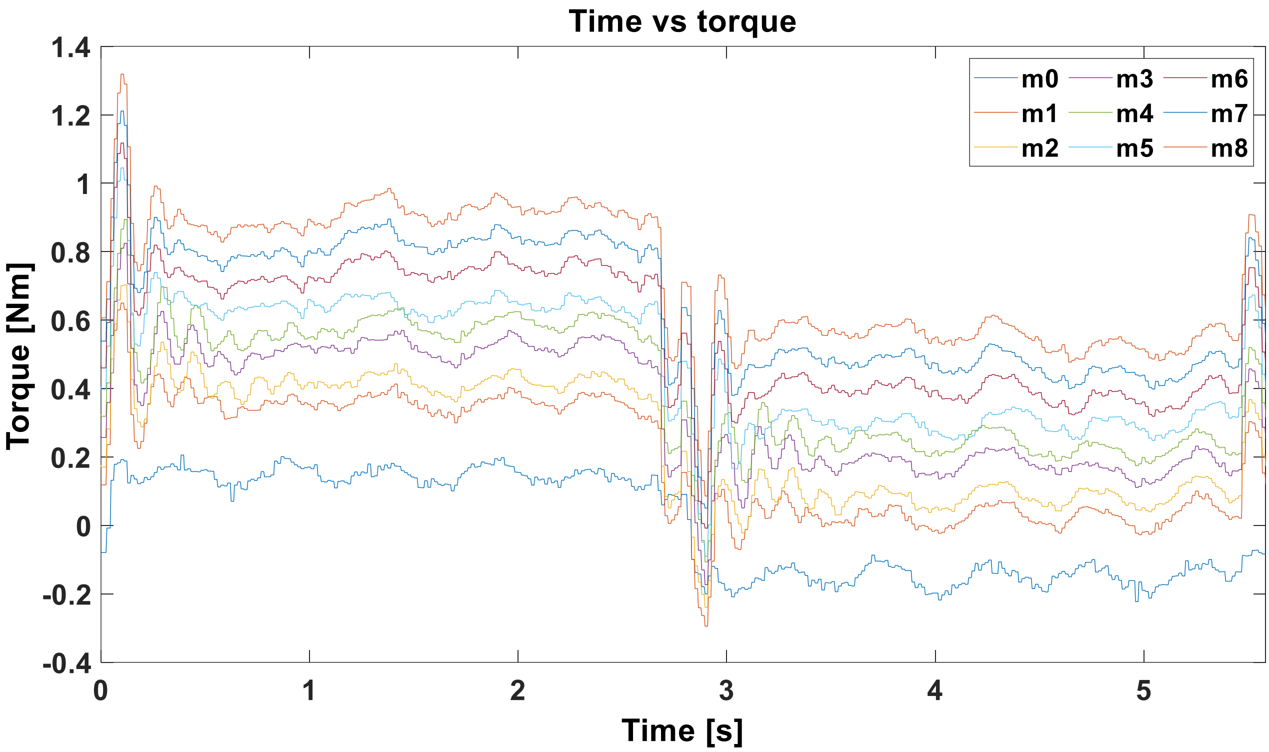


Figure 5-9 Time vs torque from measurement with various masses

Due to the motor, the lowering of the mass is controlled as required. There could be a slight variation in the motor's torque (Figure 5-9), as there is a change in tension with different masses. The experiment also showed that the vibrations are minimum with the increase in tension on the wire. The observation also corresponds to the result of the mathematical model and formulas.

CHAPTER 6: Conclusion

The thesis deals with designing a mechanical system that ensures defined cable tension under dynamic load conditions. The designing procedure includes several steps. Firstly, the research on automatic reel systems— followed by, proposal and selection of ideas. Then, the mathematical model of the proposed ideas was created. Following that, CAD modelling and drawings were done for manufacturing. Then, the parts are manufactured, and the experimental setup were made. Furthermore, the experimental results were collected and compared with the expected values. Based on the comparison results, the design's conclusion is made, and it is as the following.

From the thesis, it is concluded that the gravitational potential system works brilliantly, and it satisfies the requirements of the machine. The gravitation potential system’s performance was observed to be similar to that of the current situation with one less servo motor. By tuning the proper tension on the wire, the vibrations are reduced, and the chance of liquid dripping away from the wire is minimal. The proper tension on the wire for minimal vibration can be taken from the plot (Figure 3-17) corresponding to the excitation frequency. The tension can also be calculated for similar application by using the same plot. The experiment results were satisfying, as it was observed that the wire had almost no vibration with higher tensions. In order for the system to work, a few modifications to the current situation are required. Though the gravitational potential energy system works excellent, it requires proper cleaning after use of the machine. Due to the fluid sticky nature, the system needs to be adequately cleaned after every use for uninterrupted work.

The results achieved are satisfying, and the system will work exceptionally well with the variant discussed. The variant can also be used with other similar general applications.

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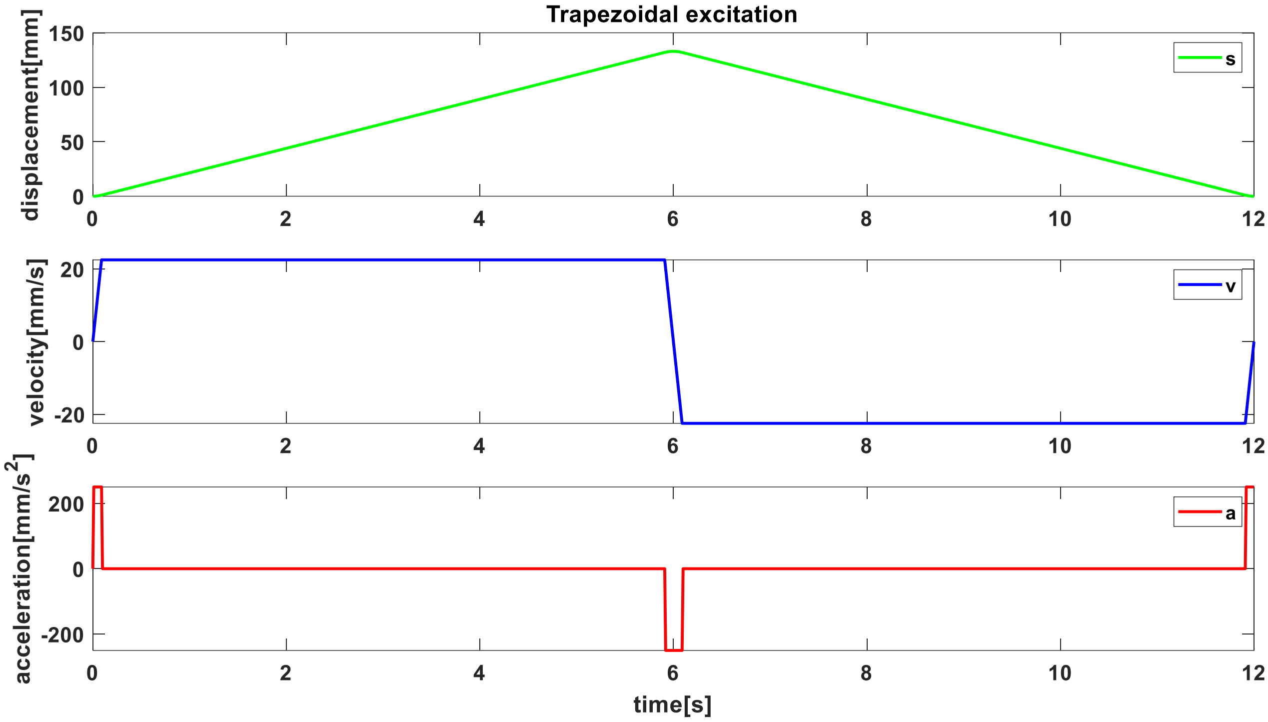
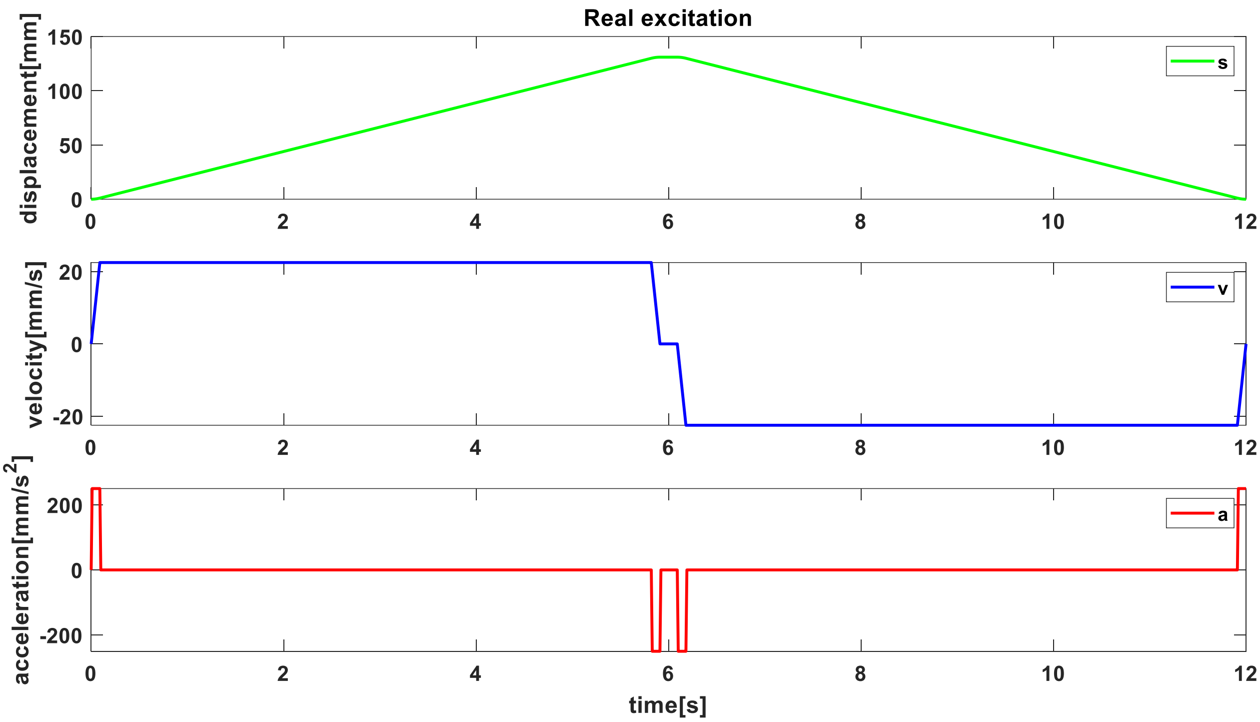
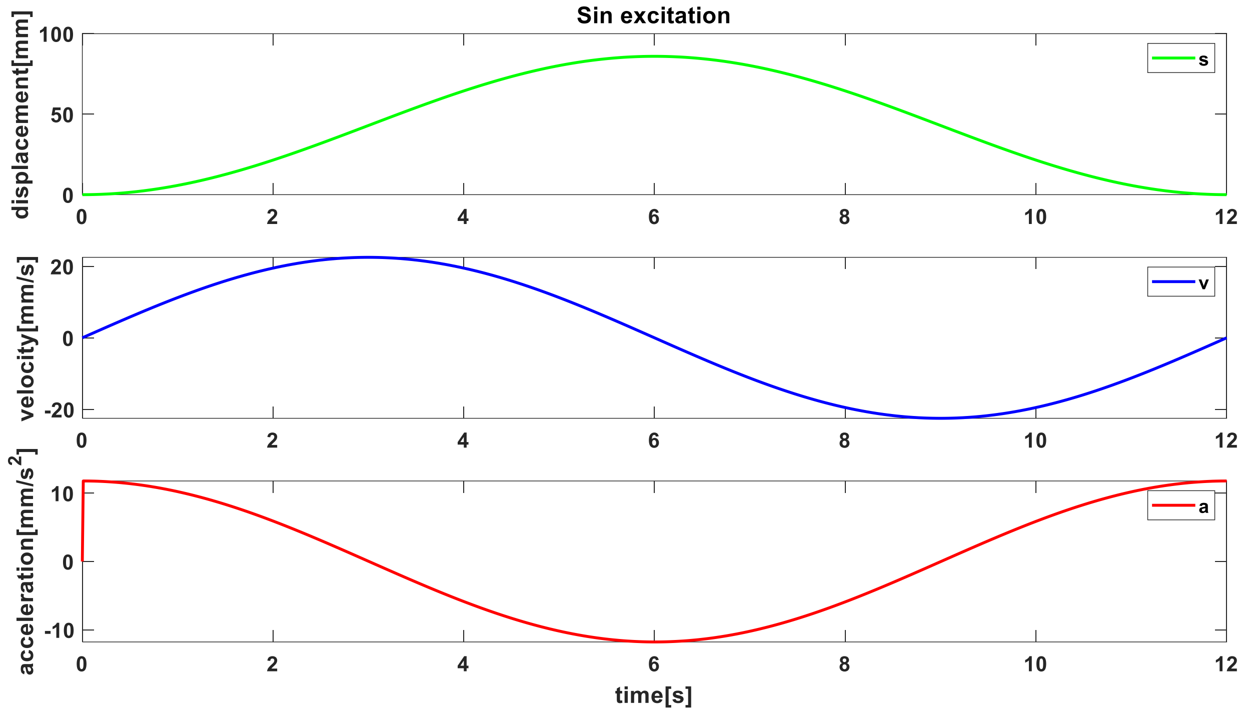
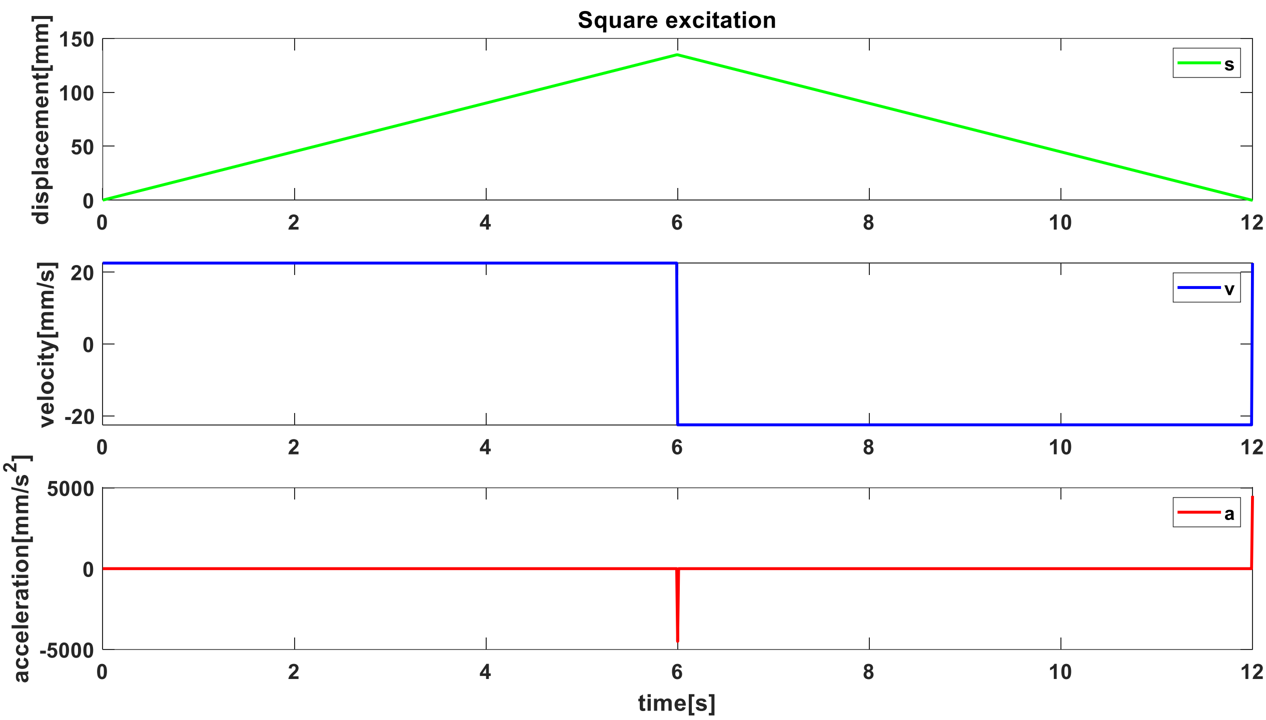
Supplements

A. 1 Excitation functions position, velocity and acceleration

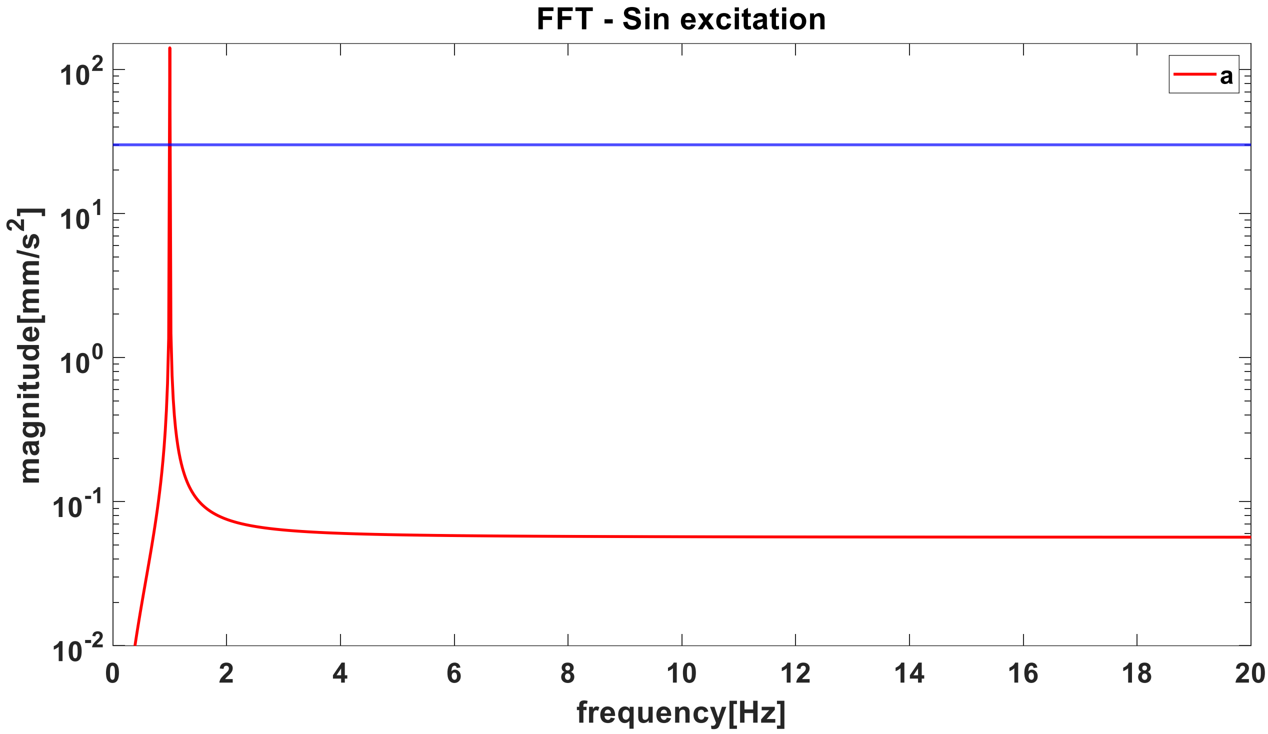


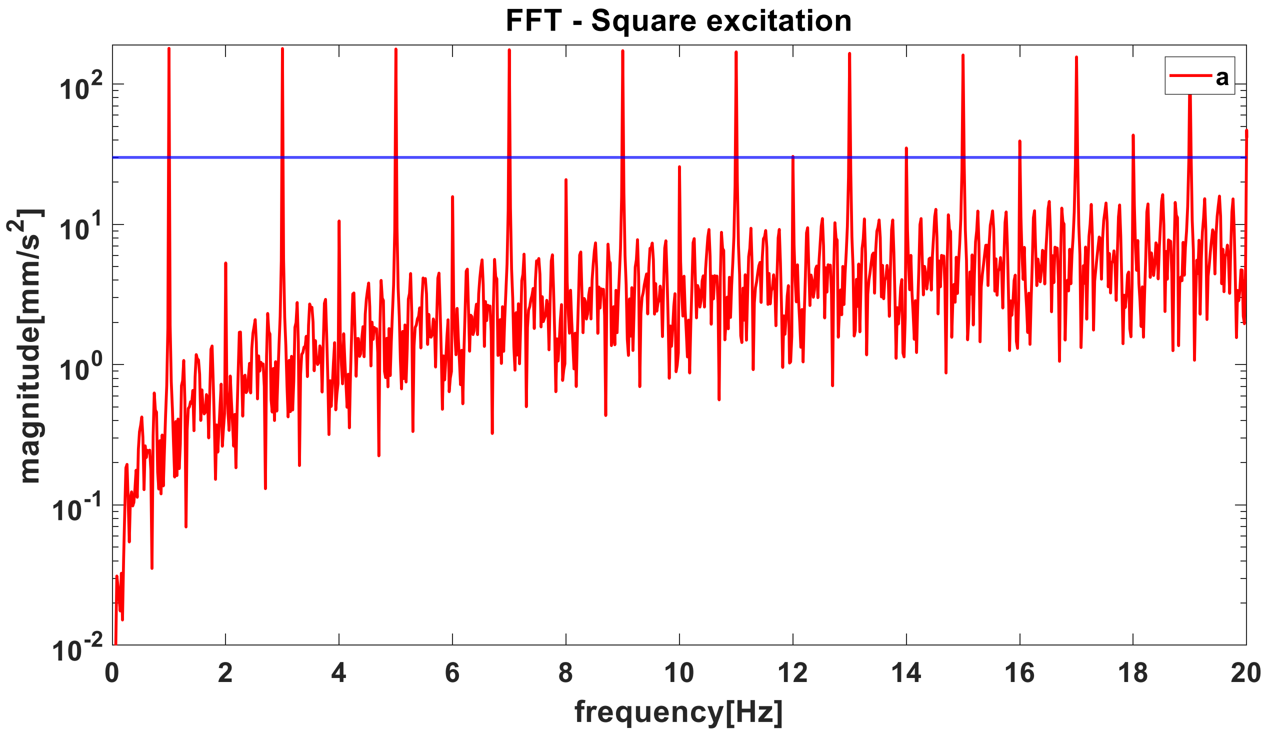


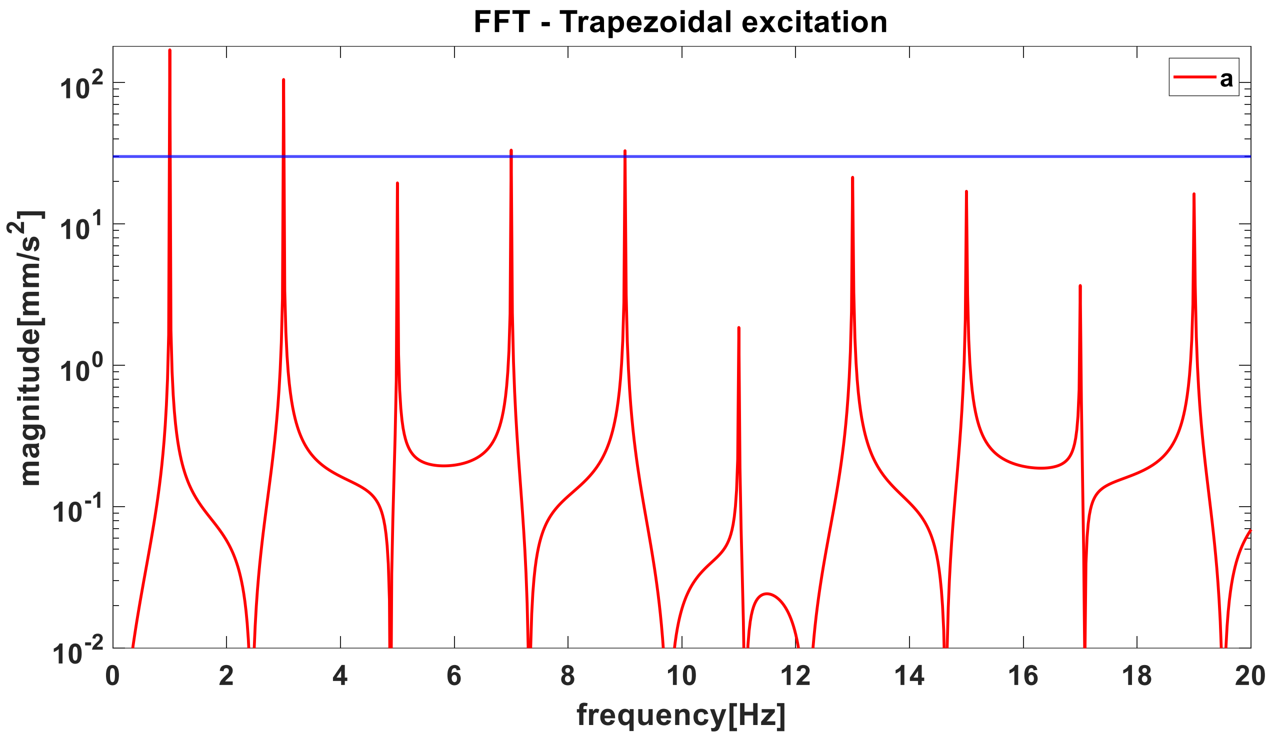

Sin excitation
displacement[mm]
velocity[mm/s]
acceleration[mm/s^2]
time[s]
s
v
a

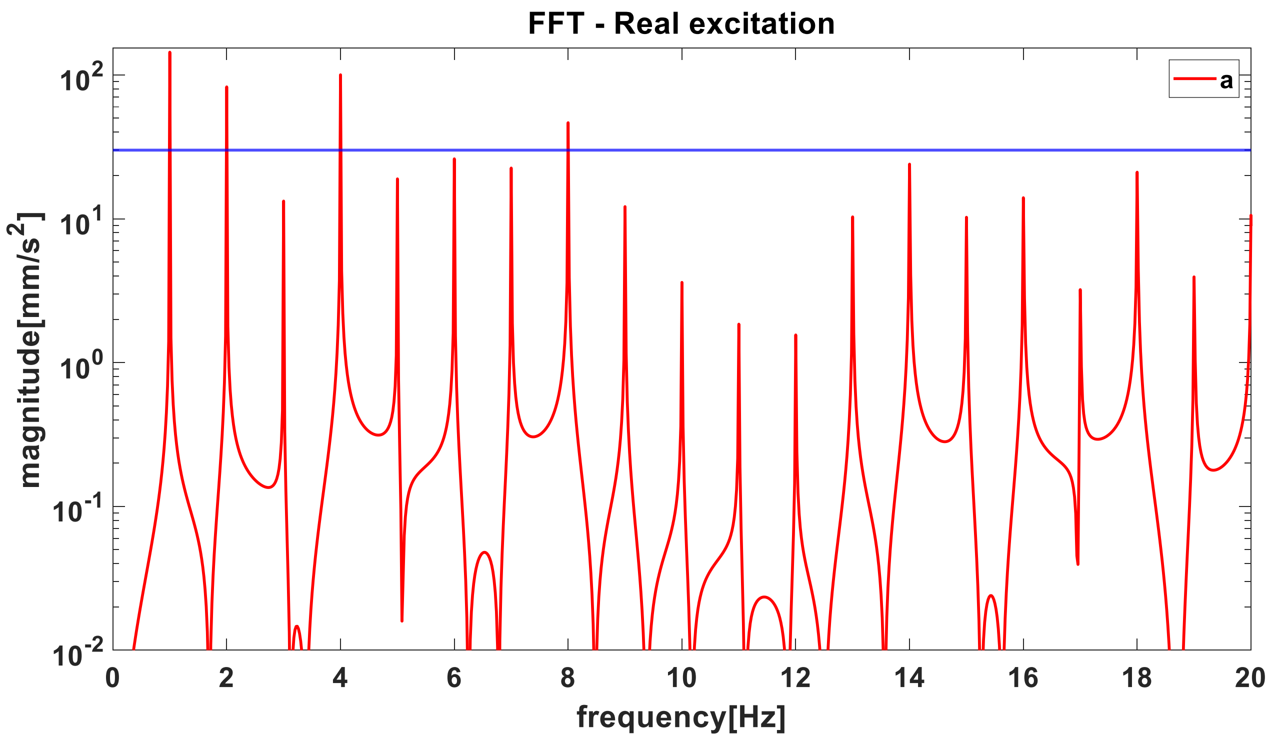

Square excitation
displacement[mm]
velocity[mm/s]
acceleration[mm/s^2]
time[s]
s
v
a

A 2 FFT of acceleration of excitation functions






FFT - Trapezoidal excitation
a
magnitude[mm/s^2]
frequency[Hz]


FFT - Real excitation
a
magnitude[mm/s^2]
frequency[Hz]

A 3 Functions for the creation of excitation functions

Real excitation

```
function [y] = ex_trap_real(t,f1,vel,acc)
%EX_TRAP_REAL real excitation function

T1= 1/f1; %(s) time period
ta=vel/acc; %acc. time
if ta>(T1/4)
    warning( 'desired velocity cannot be reached with current
acceleration');
    ta=T1/4;
end
t1=ta;
t2=T1/2-(2*ta);
t23=t2+ta;
t3=t23+(2*ta);
t4=t3+ta; %t3=T1/2+ta;
t5=T1-ta;
t6=T1;
k=(vel/ta);
tp=mod(t,T1);
y=zeros(length(t),1);

for i=1:length(t)
    if tp(i)<t1
        y(i)=k*tp(i);
    elseif tp(i)<t2
        y(i)=vel;
    elseif tp(i)<t23
        y(i)=-k*(tp(i)-(T1/2 - ta));
    elseif tp(i)<t3
        y(i)=0;
    elseif tp(i)<t4
        y(i)=-k*(tp(i)-(T1/2 + ta));
    elseif tp(i)<t5
        y(i)=-vel;
    elseif tp(i)<t6
        y(i)=k*(tp(i)-T1);
    end
end

end
```

Trapezoidal excitation

```
function [y] = ex_trap(t,f1,vel,acc)
%EX_TRAP trapezoidal excitation unction

T1= 1/f1; %(s) time period
ta=vel/acc; %acc. time
if ta>(T1/4)
    warning( 'desired velocity cannot be reached with current
acceleration');
    ta=T1/4;
end
t1=ta;
t2=T1/2-ta;
t3=T1/2+ta;
t4=T1-ta;
t5=T1;
k=(vel/ta);
tp=mod(t,T1);
y=zeros(length(t),1);

for i=1:length(t)
    if tp(i)<t1
        y(i)=k*tp(i);
    elseif tp(i)<t2
        y(i)=vel;
    elseif tp(i)<t3
        y(i)=-k*(tp(i)-(T1/2));%-k*tp(i)+(T1/2)*k
    elseif tp(i)<t4
        y(i)=-vel;
    elseif tp(i)<t5
        y(i)=k*(tp(i)-T1);%k*tp(i)-T1*k;
    end
end

end
```

Square excitation

```
function [y] = ex_square(t,f,A,~)
%EX_SQUARE square wave excitation
y=A*square(2*pi*f.*t);
end
```

Sin excitation

```
function [y] = ex_sin(t,f,A,~)
%EX_SIN Sine excitation fcn
y=A*sin(2*pi*t*f);
end
```

A 4 plotting the excitation functions, FFT

```
%%input parameters
nt = 10; %no of modes
n = (1:1:nt)'; %mode number
L = 0.5; %length of the wire [m]
r1 = 37.5/1000; %radius of the drum1 (m)
M = 0.5; %moment [Nm]
f1 = 1; %1:1:10 [Hz] excitation frequencies
A = 22.5; %amplitude of the excitation wave
vel = A; %[m/s] velocity
acc = 250; %[m/s^2] acceleration
at = 30; %amplitude threshold
ex_fcn = @ex_trap; % type of excitation fcn
T1 = 1/f1;% (s) total time for fft
np = 10;  %no of cycles in total time period
T = np*T1; %total time period
dt = 1e-2;% (s) time increment for fft

%% calculation
t = (0:dt:T)';
F = r1*M; %tension on the wire (Nm)
fm = 1/(2*dt); %max frequency of the plot (Hz)
y = ex_fcn(t,f1,A,acc);
s = cumtrapz(t,y);
a = [0;(diff(y)./diff(t))];
[fe,Py] = plot_fft(t,y);
[fe,Ay] = plot_fft(t,a);

%% plotting s,v,a
figure (1) ;
subplot(3,1,1)
plot(t,s,'g');
legend('s');
ylabel('displacement[mm]');
title('Sin excitation');

subplot(3,1,2)
plot(t,y,'b');
legend('v');
ylabel('velocity[mm/s]');

subplot(3,1,3)
plot(t,a,'r');
legend('a');
xlabel('time[s]');
ylabel('acceleration[mm/s^2]');
%% plotting fft of a
figure (2) ;
semilogy(fe,Ay,'r');
```

```
yline(at,'b');
ylim([1e-2 max(Ay)+10]);
xlim([0 20]);
legend('a');
xlabel('frequency[Hz]');
ylabel('magnitude[mm/s^2]');
title('FFT - Real excitation');
```

A 5 Comparing natural frequency, excitation frequency, measurement vs tension

```
n = 10; %no of DOF's
L = 0.55; % [m] length of the rod %0.5
d = 0.001; % [m] diameter of the wire
r=d/2; % [m] radius of the wire
% rho = 8042; % [kg m^-3] density
% % E=180e9; % [Pa] Young's modulus
% % n = 100; % number of nodes
% S = pi*(r^2); % [m^2] cross-sectional area
% m = (rho*S*L)/n; % [kg] node mass
m = (35.1e-3 / 7.18) * L; %mass per unit length of the wire
b = 0.1; % [N*s/m] damping
s = L/(n+1); % distance between mass points
g = 9.81 ; %[N/m^2] gravity
nv=1:1:10;% number of modal shapes
load mass_used.mat;

%% mass matrix
M = eye(n); %identity matrix
M = m*M; %mass matrix
q = zeros(n,2);
w = 1;

%% stiffness matrix
for a=1:length(nv)
for T=0:0.01:12
k = T/s;
K=zeros(n,n);
for i=1:(n-1)
K([i,i+1],[i,i+1])=K([i,i+1],[i,i+1])+[1 -1;-1 1];
end
K(1,1)=K(1,1)+1;
K(n,n)=K(n,n)+1;
K = k*K;

%%
[ve,ei] = eig(K,M);% eigenvalue calculation
vfrq= sqrt(diag(ei))./(2*pi);
```

```
fn=cable_natural_frequency(nv,L,T); %by formula
q(w,1) = T;
q(w,2) = fn(a);
w = w+1;
end
end
%%
figure (5);
u1 = 1;
u2 = 1201;
hold on;
for r = 1:10
    drawnow;
    plot(q(u1:u2,1),q(u1:u2,2),'-b');
    u1 = u1+1201;
    u2 = u2+1201;
end
hold off;
xlabel('tension [N]');
ylabel('natural frequency [Hz]');

%%
hold on;
tension = zeros(8,1);
for o = 1:8
    ma = mass(o);
    tension(o) = mass_tension_calc(ma);
    xline(tension(o),'k');
end
%%
load fe_max_wire.mat;
k = 24;
for tn = 1:size(tension)
    for p = 1:3
        plot(tension(tn),fe_max(k),'r*');
        k = k-1;
    end
end
hold off;
%%
for i = 1:nIea
       yline(fea(i),'-r');
end
```

A 6 Longitudinal vibration

```
n = 10; %no of DOF's
L = 0.5; % [m] length of the rod
d = 0.001; % [m] diameter of the wire
r=d/2; % [m] radius of the wire
rho = 8050; % [kg m^-3] density
A = pi*(r^2); % [m^2] cross-sectional area
m=(35.1e-3/7.18)/n; %mass of wire (kg /m)
k = (3.082334e+05*(L/0.2))*n;% [N/m] spring stiffness
b = 0.01 * n; % [N*s/m] damping
T = 0.5; % tension on the string [Nm]
E=k*L/A; % [Pa] Young's modulus
g = 9.81 ; %[N/m^2] gravity
s = 0; %displacement
%%
f1 = 1; %1:1:10 (Hz) exitation frequencies
A = 1; %amplitude of the excitation wave
vel = A; %(rev/sec) velocity
acc = 15; %(rev/s^2) acceleration
%% mass matrix
ML = eye(n); %identity matrix
ML = m*ML; %mass matrix
MLI = inv(ML);
%%
KL=zeros(n,n);
for i=1:(n-1)
KL([i,i+1],[i,i+1])=KL([i,i+1],[i,i+1])+[1 -1;-1 1];
end
%%
BL = b*KL; %damping matrix
KL = k*KL; %stiffness matrix
%%
SL = ones(n,1);
SL = s*SL;
SL(1,1) = 0;
SL(end,end) = 0;
%%
nv=1:1:10;% number of modal shapes
[ve,ei] = eig(KL,ML);% eigenvalue calculation
vfrq= sqrt(diag(ei))./(2*pi);
disp(vfrq(nv)');
```

A 7 Lateral vibration on the wire

```
%%input parameters
n = 1000; %no of DOF's (increase to improve accuracy)
L = 0.56; % [m] length of the rod %0.5
d = 0.001; % [m] diameter of the wire
r=d/2; % [m] radius of the wire
m = (rho*S*L)/n; % [kg] node mass
b = 0.001; % [N*s/m] damping
a = L/(n+1); % distance between mass points
load mass_used.mat;


%%
ma = mass(4); %mass to be used
T = mass_tension_calc(ma); %tension on the string
%% mass matrix
MT = eye(n); %identity matrix
MT = m*MT; %mass matrix
MTI = inv(MT);
%% stiffness matrix
k = T/a;
KT=zeros(n,n);
for i=1:(n-1)
KT([i,i+1],[i,i+1])=KT([i,i+1],[i,i+1])+[1 -1;-1 1];
end
KT(1,1)=KT(1,1)+1;
KT(n,n)=KT(n,n)+1;

KT = k*KT;
BT = b*KT;
%%
nv=1:1:10;% number of modal shapes
[ve,ei] = eig(KT,MT);% eigenvalue calculation
vfrq= sqrt(diag(ei))./(2*pi);
fn=cable_natural_frequency(nv,L,T); %by formula
disp(vfrq(nv)');
disp(fn(nv));
```

List of Drawings

Number	Name	Size
1	modul_lanovice	A3
2	kts1_00_00_mass_assembly	A3
3	kts1_00_01_mass_block	A4
4	kts1_00_02_l_profile	A4
5	kts1_00_03_shaft_h	A3
6	kts1_00_04_pulley	A4
7	kts1_00_11_set_collar	A4