VELOCITY CONTROL USING TORQUE INPUTS TO THE MOTOR OF A TEXTILE CROSS-LAPPER MACHINE

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Received: 16-April-2022 / Revised: 15-July-2022 / Accepted: 15-July-2022
Karachi Institute of Economics and Technology || Technology Forces Journal, Volume 4, Issue 1, 2022

ABSTRACT

The main purpose of our research is to present modeling and control of a nonwoven textile cross lapper machine. The cross lapper machine is primarily based on pulley block structure mechanism order to make consistent web having suitable width and thickness. This is achieved mainly by pulleys' reciprocating motion for moveable rack system. The mechanism is based on attaining stable speed for carriages while moving back and forth. In this way the machine has ability of stopping at precise re-entry position during reciprocating process. Traditionally, systems inherit a large attenuation of the input; thus, systems tend to be sluggish. However, the system possesses stability. The significance of this study is to present a realizable controller and to discuss the corresponding properties of a dynamic system for controlling horizontal type cross-lapper machines. This not only deals with the achievement of meaningful design objectives but also makes the closed-loop system efficient for good tracking properties. This study is primarily focused on theoretical analysis, aiming to dynamically arrive at a model which can mathematically express the functioning of a cross- lapper machine. The nonlinear dynamic equation for the system is driven by applying the Lagrange equation. During the controller design phase, first the high order transfer function of the linearization system is obtained. Then, a corresponding system having a transfer function of lower order is reached by the application of an approximation criterion with the primary goal of having the analysis; and thus the effort of design is minimized. The control system design involves the application of pole-placement design, in addition to realizable proportional and integral control through state feedback. Application of this model is an initial step towards designing a controller working on a closed loop principle, a moveable rack determining speed and position simultaneously. The importance of such work is that, with the help of a vigilant input system, steady state errors will come to an end and outclass oriented tracking properties will be obtained. The final result of our research is to achieve a dynamic mathematical model through which overall stability of the system is ensured. This control scheme would also meet the needs for the engineering process of the card web, would reduce the vibration of the cross lapper machine, and in turn, the dynamic overloading is also reduced.

Keywords: Cross - Lapper Machine, Torque.

1. INTRODUCTION

There has been very rapid growth in non-woven products in the polymer industry during recent years. This is due to their wide application and as non-woven goods are being widely consumed in a variety of applications, including but not limited to textile, biomedical, and geo-technical engineering. The production process for non-woven products can be further subdivided as web finishing, web reinforcing, and web manufacturing. In view of this importance, in recent years, a number of studies have begun to examine the effect that web needle piercing and the subsequent completion process have had on the material of an uncircumcised product.

As illustrated earlier, non-woven fabrics have wide application areas as their goods are being used in numerous fields, not limited to medical treatment industries and daily necessities. The nonwoven fabrics are manufactured through a continuous process involving passing cotton fibers through a cross lapper, a card, and a needle punch in order to be processed into nonwoven fabrics. The nonwoven cross lapper machine is used in to-and-fro motion after carding. This process necessitates the consistency of reciprocal motion as it is necessary for the overall physical properties of nonwoven fabrics. Usually, cross lapper machines are mainly of two different categories, i.e., vertical and horizontal types. The vertical cross-lapper machine type uses gravity for the falling of the card web, and then layers of the web are made by a swinging arm. In the process of the to-and-fro motion of the vertical cross lapper machine, air resistance makes it become a non-uniformed broken web. From the complete mechanical structure of the vertical cross lapper type machine, its height arises as a problem due to the increased width of card web, which results in a longer swing arm. Therefore, the vertical type machines have already been replaced by the cross- lapper machines of the horizontal type as the preferred nonwoven cross-lapper machines for mainstream production, which have a to- and-fro motion on a movable rack. Stadnik applied the Lagrange's equation in order to derive the mathematical model of the traditional horizontal type cross-lapper machine and applied the optimization of driving torque for simulation of the reversal motion. Moreover, Kou implemented the Hamilton's Principle, also known as Newton's second law, for deriving the cross-lapper machine's mathematical model.

Due to the need for the conveyor belt high speed motions of card web and in order to overcome the classical problems of broken up production, mainly caused by web breakage, for the cross lapper machine's new horizontal type, a double belt layering system has already been designed. Meanwhile, the control of the speed of the machine as well as its position for movable racks is being carried out by a controller having an open-loop and through the use of limit switches, indirectly. However, the resulting web uniformity is inferior, mainly due to the lack of reverse feed info by movable racks. This is a property which is normally available, particularly at the finish points of the movable racks in to and fro motion. The movable racks vibrate under the influence of the inertia force and control the overall lapping of the web.

For a simple model of the control system, the input to the controller is the spinning torque. This shows the amount of power needed to overcome the conflict to move a piece of work through the arc you want. The output of this controller is an angular change in position, which also provides a change in the velocity of the controlled work part. Normally, the controller will compare the control inputs, in this case the spinning torque, to the desired value. This could be a continuous speed or a reference signal generated elsewhere. Converting the spinning torque into an angular area of velocity control is a mechanical or electrical device called an integrator. The simple compiler method takes the maximum amount of input and uses it as a subtraction with minimal delay. This focuses most of its errors on faster exits than accumulated errors over time. In many cases, this works well, especially for applications with low-velocity inputs and slow responses to machine tools such as screw jacks and cross-lapper equipment.

1.1 What does Velocity Mean?

Your idea of speed may be similar to its scientific definition. You know that a large migration in a short period of time means a great deal of speed and that the velocity has sub divisional units, such as miles per hour or KM Per hour. Medium speed is defined as a change in the state separated by travel time.

$$V_{avg} = \frac{d}{dx}\left(\frac{\partial x}{\partial t}\right) \qquad (1)$$

1.2 What is Torque

Torque is a quantification of the force which causes an object to rotate about an axis. It is similar to a force that causes an object to accelerate in terms of kinematics, torque is the property that causes an object to attain an angular acceleration. Torque is also known as moment and it is a vector value. The torque vector's direction is mainly dependent on the direction of force about the axis. In daily life the example of opening / closing a door can be utilized for accurate understanding of torque

1.3 Torque can be Static or Dynamic Torque

Vertical torque is one thing that does not result in the production of any angular acceleration. A person who presses a closed door puts static torque on the door as it does not rotate with its hinges, without power input. A person who pedals a bicycle at a constant speed, and uses static torque because it is not fast. The drive shaft in a fast race car from the first lane has a strong torque since it is necessary for production of angular acceleration of its wheels because the car is fast following the track. This is the same thing used in Cross Lapper machines

Keeping in view the theoretical analysis, the research primarily focus on the dynamic derivation of a mathematical model for the latest horizontal cross lapper machine type, and in turn to apply subject model as the initial stage towards design of a closed-loop controller intended to monitor the position along with the speed of movable racks. This would ultimately result in reduction of vibrations, thus, overall consistency for the lapping of web could be enhanced.

2. LITERATURE REVIEW

The invention of the Cross-lapper machine proved to be an absolute blessing for industrialists because it solved the problem of making quilts, pillows, and so on from the time it was invented. The machine has changed many-fold and enormous changes in using basic problem-solving skills also proved useful for the modification process of this machine. Plenty of research papers have been published illustrating the dynamic modeling and control of the subject cross lapper machine, horizontal type, and control of the carriage speed in the cross lapper by adapting various methods. Some examples of such academic and research work independently carried out by prestigious academic institutes are described below:-

2.1 Research carried out by Department of Polymer Engineering, NTUST

During research conducted by graduates of the National Taiwan University of Science and Technology, the problem of input optimization in order to steer an open loop cross lapper was observed. This not only included modeling and cross-lapper machine control, but it also derived the controlling mechanism for the system's inputs. As a result, the control criteria for an accurate time optimization control for the cross lapper system were proposed in order to follow the required cross rack velocity profile with a preset damping ratio and the un-damped natural frequencies. The derived controller was aimed so that it could minimize time for settling of cross-lapper moveable rack velocity by lowering power consumption while moving. Such a control approach is achieved by manipulation of input to the servomotor, which drives the lower pulley, and therefore, driving of the carriage by utilization of the driving belt can therefore be implemented with ease. This paper achieved the goal of deriving a mathematical formula as well as the development of a tool for computation that could lead to the design of a controller for real time.

2.2 Research carried out by Intelligence Control and Simulation Laboratory

Another study conducted by Taiwan's National University of Science and Technology on the dynamic modeling and control of the current horizontal cross lapper machine revealed that the width and web thickness required for the finished product are dependent on the control of constant movable racks steady velocity. Deliberations for the latest cross-lapper horizontal type machine were infrequent in the studies.

Therefore, this paper introduces the model and deliberates the corresponding elements of the cross-lapper machine's new horizontal type system. The compact order system of the controller design was formed with a rating scale, and its understanding was confirmed. Both parallel and inaccessible controls, as well as the design of the pole-bearing in response to the state, have been used to build a control mechanism. The controls can result in a solid speed of moving rack having high-quality tracking positions and can abolish fixed position errors. The consistent velocity of a moving rack may ensure web consistency and maintain machine performance. Similarly, the cross-lapper is usually operated by microprocessors. The control system will fulfill the requirements of the process for web card engineering, reduce machine oscillations, and reduce power overload.

3. MATHEMATICAL MODELING

The mechanism of a cross-lapper machine of the horizontal type is illustrated by Fig. 1. A movable rack with a motor drive system is its major component. The movable rack system consists of a conveyor belt supported by a pulley, which is present for transferring the web to the roller module of the movable rack. A positive torque is exerted by the motor drive system on the transmission pulley in order to tug the rack along with the conveyer belt. Through rollers' rotation along with racks' movement, it reaches the re-entry point in order to attain the desired width.

The servo motor is the main source of torque for the rotation of the transmission pulley in order to reach the re-entry position with the assistance of the conveyor belt. The horizontal type of cross lapper machine is initiated-continual movement at constant speed, with reversal of re-entry point, positioning at another turn around re-entry point, to and fro motion to keep the racks at a stable speed, allowing line to travel by the roller module of the movable rack in order to fold up in a Z–shaped pattern and layer the final web into the specified thickness and width.

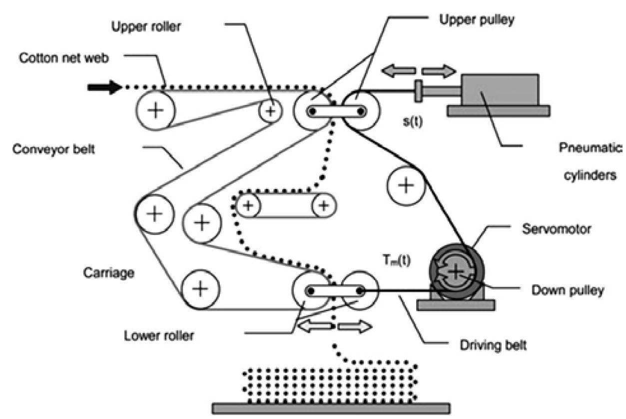

Figure 1: Working mechanism for the new cross-lappe r machine of horizontal type [21]

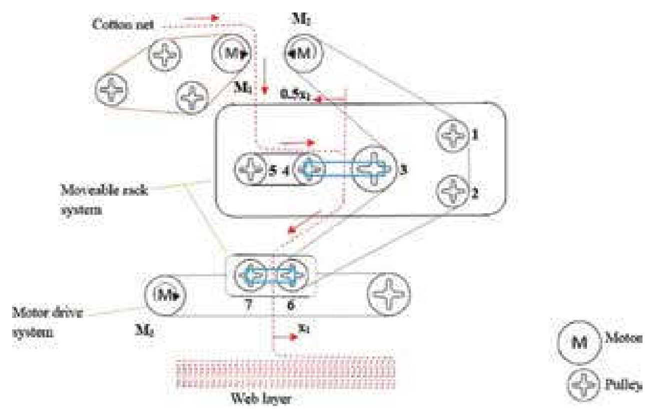

Figure 2: Current new horizontal type cross lapper [22]

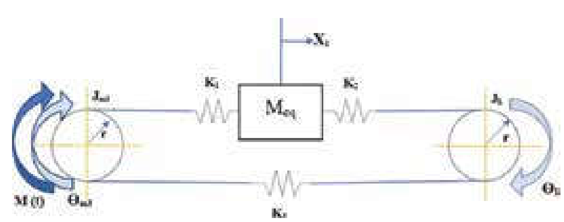

Figure 3: Equivalent design for new cross lapper of horizontal types [2]

3.1 System of Movable Racks

Movable racks, conveyor belt and the pulley get their driving power from motor for taking the web towards the mobile rack through roller module for the complete process of the web. The kinetic energy for this system is expressed as T1 and is shown as [23]

$$T_1 = \frac{1}{2}J_{M_2}\dot{\theta}_{M_2}^2 + \frac{1}{2}J_1\dot{\theta}_1^2 + \frac{1}{2}J_2\dot{\theta}_2^2 + \frac{1}{2}J_3\dot{\theta}_3^2 + \frac{1}{2}J_4\dot{\theta}_4^2 + \frac{1}{2}J_5\dot{\theta}_5^2 + \frac{1}{2}J_6\dot{\theta}_6^2 + \frac{1}{2}J_7\dot{\theta}_7^2 \quad (2)$$

and

$$x_1 = 2r_1\theta_1 = 2r_2\theta_2 = 2r_3\theta_3 = 2r_4\theta_4 = 2r_5\theta_5 = r_{M_2}\theta_{M_2} = r_6\theta_6 = r_7\theta_7$$

where

r_i: the pulley radius i = 1, 2, 3, 4, 5, 6, 7.

θ_i: displacement of the pulley i = 1, 2, 3, 4, 5, 6, 7.

J_{M2}: the inertia of the motor equivalent moment M_2. □$_{M2}$: the motor shaft angular velocity M2.

J_i: the inertia pulley moment i = 1, 2, 3, 4, 5, 6, 7.

□$_i$: the angular velocity for the pulley i = 1, 2, 3, 4, 5, 6, 7.

x_i: the movable rack displacement.

r_{M2}: the motor shaft radius M2.

θ_{M2}: the motor shaft angular displacement M2.

Substituting the values from equation of x_1 into the kinetic energy T_1 comes out to be [23]

$$T_1 = \frac{1}{2}J_{M_2}\left(\frac{\dot{x}_1}{r_{M_2}}\right)^2 + \frac{1}{2}J_1\left(\frac{0.5\dot{x}_1}{r_1}\right)^2 + \frac{1}{2}J_2\left(\frac{0.5\dot{x}_1}{r_2}\right)^2 + \frac{1}{2}J_3\left(\frac{0.5\dot{x}_1}{r_3}\right)^2 + \frac{1}{2}J_4\left(\frac{0.5\dot{x}_1}{r_4}\right)^2 + \frac{1}{2}J_5\left(\frac{0.5\dot{x}_1}{r_5}\right)^2 + \frac{1}{2}J_6\left(\frac{\dot{x}_1}{r_6}\right)^2 + \frac{1}{2}J_7\left(\frac{\dot{x}_1}{r_7}\right)^2 \quad (3)$$

3.2 System for Driving the Motor

For system for driving the motor, the source of both forward and reverse torques is servo motor responsible for the motion of driving belt for pulling the racks that provides velocity and displacement. The corresponding cross lapper machine free-body diagram of horizontal type is illustrated by the Fig 2 above. Three degrees of freedom are possessed by this system, for which the coordinates which are independent are x 1, θ m3, and θ L. This mechanism, has rigid belts for conveying that are provided in static stretch operation. On the other hand, the flexible driving belts having much more elasticity than that of the conveyor belts are used [2].

$$K_1 = \frac{2EA}{L_1 - x_1} K_2 = \frac{2EA}{L_2 - x_1} K_3 = \frac{2EA}{L_3 - x_1} \quad (4)$$

Where the Young's modulus is **E**, the cross-sectional area of the driving belt is A is, the spring initial length is L_i is, i = 1, 2 and 3. The kinetic energy for the springs can be expressed as follows [24]:-

$$T_{k_1} = \frac{1}{2}\left(\frac{1}{3}m_{k_1}\dot{x}_1^2\right) T_{k_2} = \frac{1}{2}\left(\frac{7}{12}m_{k_2}\dot{x}_1^2\right) T_{k_3} = \frac{1}{2}\left(\frac{1}{12}m_{k_3}\dot{x}_1^2\right) \quad (5)$$

4. SYSTEM LINEARIZATION

4.1 System Equilibrium Positions

In order to have the horizontal cross lapper machine instate of steady equilibrium, the following conditions need to be fulfilled:-

1. Movable rack velocity $\dot{x}_1 = 0$
2. Linear acceleration $\ddot{x}_2 = 0$
3. Motor Drive System motor axle equivalent angular velocity $\theta_{m3} = 0$
4. Equivalent angular acceleration of motor axle $\theta\theta_{m\,3} = 00$
5. The load axle Angular velocity is $\theta_L = 0$
6. The load axle angular acceleration is $\theta\theta_L = 0$,
7. Variable input of the angular movement is **M(t)** = 0.

When all the above conditions are attained at the state of stable equilibrium, following equation set is achieved:

$$-\left(\frac{2EA}{L_1 - x_1} + \frac{2EA}{L_3 - x_1}\right)r^2\theta_{m_3} + \frac{2EA}{L_1 - x_1}rx_1 + \frac{2EA}{L_3 - x_1}r^2\theta_L = 0$$

and (6)

$$\left(\frac{2EA}{L_1 - x_1} + \frac{2EA}{L_2 - x_1}\right)x_1 - \frac{2EA}{L_1 - x_1}r\theta_{m_3} - \frac{2EA}{L_2 - x_1}r\theta_L + EA\left[\frac{1}{(L_1 - x_1)^2}(x_1 - r\theta_{m_3})^2 + \frac{1}{(L_2 - x_1)^2}(r\theta_L - x_1)^2 + \frac{1}{(L_3 - x_1)^2}(r\theta_{m_3} - r\theta_L)^2\right] = 0$$

4.2 The Linear Equation of the System

For the new cross lapper machine of horizontal type, the dynamically liberalized equation for system may be written as follow:-

$$\dot{X}_1 = \frac{-1}{M_{eq}}\left\{\left(\frac{2EA}{L_1}+\frac{2EA}{L_2}\right)X_2 - \frac{2EA}{L_1}rX_5 - \frac{2EA}{L_2}rX_3\right\}$$

$$\dot{X}_2 = X_3$$

$$\dot{X}_3 = X_4$$

$$\dot{X}_4 = \frac{-1}{J_L}\left[B_L X_4 + \left(\frac{2EA}{L_2}+\frac{2EA}{L_3}\right)r^2 X_3 - \frac{2EA}{L_2}rX_2 - \frac{2EA}{L_3}r^2 X_5\right] \quad (7)$$

$$\dot{X}_5 = X_6$$

$$\dot{X}_6 = \frac{-1}{J_{m_3}}\left[M(t) - B_{m_3}X_6 - \left(\frac{2EA}{L_1}+\frac{2EA}{L_3}\right)r^2 X_5 + \frac{2EA}{L_2}rX_2 + \frac{2EA}{L_3}r^2 X_3\right]$$

$$Y = X_1$$

4.3 System Controllability

The current cross lapper machine of horizontal type is under control, at a first stage, if it is likely through the utilization of an unrestrained control in order to move the system from any original condition to an alternate estate during a limited time gap. If the system is entirely controllable, then the rank of the controllable matrix at any original given state is equal to **n** i.e.

$$Q_c = [B\,AB\,A^2B \ldots A^nB] = n$$

4.4 Approximation of High-order System through Low-order System

Assuming closed loop transfer function for higher order system is given by

$$M_{high}(s) = k\,\frac{1 + b_1 s + b_2 s^2 + \cdots + b_m s^m}{1 + a_1 s + a_2 s^2 + \cdots + a_n s^n},\; n \geq m \quad (8)$$

The gain k is to remain the same when subjected to zero frequency (direct current) for both transfer functions, to maintain the steady performance of the higher order system into the corresponding lower order system. The state ofdiscovering the low-order Mlow(s) requires under mentioned connection to be fulfilled as straight as possible:

$$M_{low}(s) = k\,\frac{1 + c_1 s + c_2 s^2 + \cdots + c_q s^q}{1 + d_1 s + d_2 s^2 + \cdots + d_p s^p},\; n \geq p \geq q \quad (9)$$

By application of this approximation standard, the transfer function of high order systems may be transformed to corresponding lower order system as expressed in following equation:

$$\left(\frac{2EA}{L_2 - x_1} + \frac{2EA}{L_3 - x_1}\right)r^2\theta_L - \frac{2EA}{L_2 - x_1}rx_1 - \frac{2EA}{L_3 - x_1}r^2\theta_{m_3} = 0 \quad (10)$$

Therefore, open-loop transfer function of the first low-order system is:

$$G_{p\,low} = \frac{z_0}{s + p_0}$$

5. CONTROL SYSTEM DESIGN

Fig. 4 illustrates the realizable design of a proportional and Integral (PI) controller which can mathematically also shown as

$$G_{cl}(s) = K_{P_1} + \frac{K_{I_1}}{s}$$

Then, after adding the controller the closed-loop transfer function is shown in fig3:

$$\frac{Y(s)}{R(s)} = \frac{G_{cl}(s)G_{p\,low}(s)}{1 + G_{cl}(s)G_{p\,low}(s)}$$

$$\frac{Y(s)}{R(s)} = \frac{z_0K_{P_1}(s) + z_0K_{I_1}}{s^2 + (p_0 + z_0K_{P_1})s + z_0K_{I_1}} \quad (11)$$

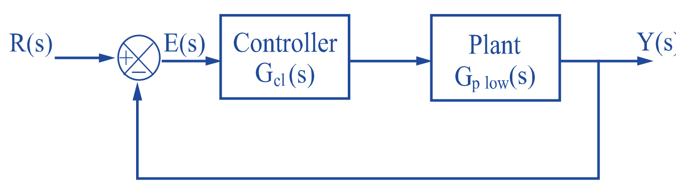

Figure 4: Horizontal cross lapper machine feedback control system [25-26]

The state feedback designs are used for the determination of the necessities with respect to each condition. The feed-back post of each constant gain, variables conditions then additionally change positions of their respective poles for the closed-loop. This methodology of design is employed during the feedback design of various control system operations and is popularly known as the pole placement design. It is pertinent to know that the system condition variables are fully and completely controllable. This means that all pole positions of may be arbitrarily arranged within system.

For the latest cross lapper machine of horizontal type, in the state equation the closed-loop system is determined by reverse feeding condition variables by means of a fixed feedback gain K matrix, as shown in Fig. 4, and is

$$\mathbf{u}(t)=\mathbf{r}(t)-\mathbf{K}\underline{\mathbf{X}}(t)$$

and then (12)

$$\dot{\underline{X}}(t)=(\mathbf{A}-\mathbf{BK})\underline{\mathbf{X}}(t)+\mathbf{Br}(t)$$

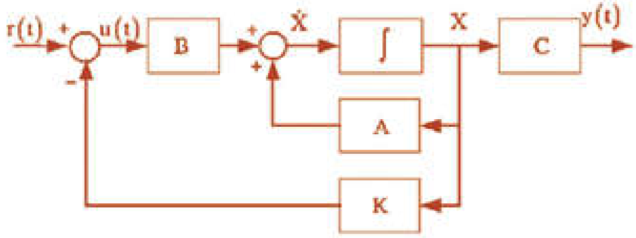

Figure 5: New current lapper machine horizontal type having state feedback

The placement of the poles is designed utilizing concepts of the state feedback. Main design aims determining the gain K matrix feedback which would give the certain prescribed values for all the characteristic values of the closed-loop system. As mentioned earlier, since the states of the system can be fully controlled, therefore, the condition of closed loop system, after designing of the state feed-back, also possess the property of complete controllability. The respective values within BK state matrix defined above may therefore be employed for the gain matrix K. This enables us to determine the pole positions for the controlling the cross lapper machine of horizontal type.

6. SIMULATION RESULTS AND DISCUSSIONS

6.1 Simulation without Controller for Linear and nonlinear System

Fig. 6 and 7 illustrated the linear and nonlinear system models for new cross lapper machine of horizontal type when it is fed with motor angular moment in the form of a unit step and with a sine function as the input. The figures depict the transient responses for the corresponding movable rack velocity of linear as well as the non-linear systems. As illustrated, both responses tend to be sluggish having large errors in their steady state. The achieved results are in cognition with anticipated results there for ere valid ate accuracy of the concept of equilibriumposition.

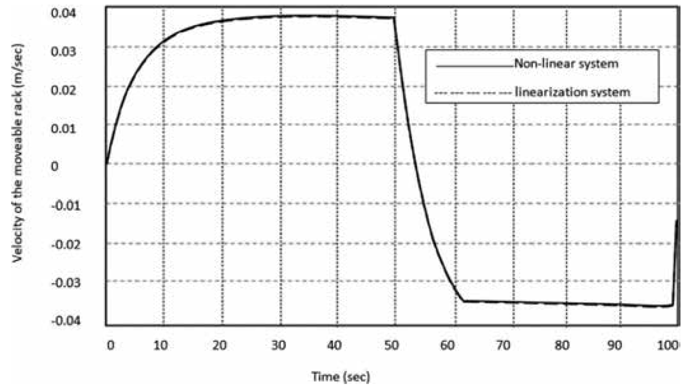

Figure 6: Response of velocity the moving rack with motor torque Inputs

$$\textit{unit step}\ u(t)=\begin{cases}1 & 0\le t<50\\-1 & 50\le t<100\end{cases}$$

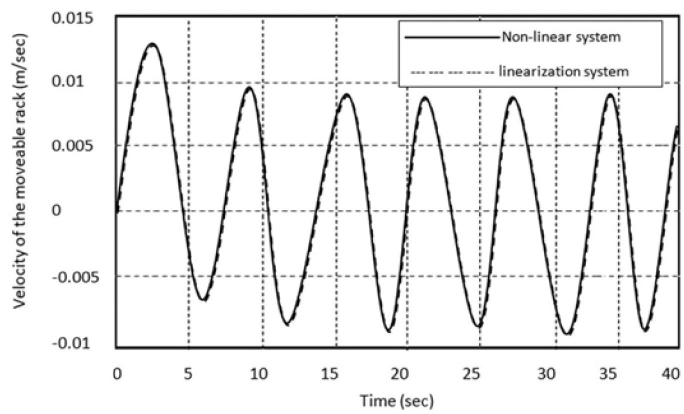

Figure 7: The response of velocity for the moving rack with motor torque input as sine function M (t) = sin t

6.2 Discussion on the open loop system types

An open loop system, commonly known as a non-response system, is a category of continuous control system where the output has no effect or does not affect the control action of the input signal. In other words, in the open source control system, the output is not measured or reversed compared to the input. Therefore, an open loop system is expected to faithfully follow the installation command or set point regardless of the end result. Also, the open system has no knowledge of the release status and therefore cannot correct itself for any errors that may occur when the present value goes up, even if this causes a significant deviation from the pre-set value. Another disadvantage of open-loop systems is that they are well- designed to manage disruptions or changes in conditions that may reduce their ability to complete the task. Such a type of "open-loop motor control" has the inherent advantage of being potentially cheap and simple in its implementation. As a result, such systems are ideal for use in well-defined systems where the input-output relationship is direct and unaffected by external disturbances.

Fig. 5 depicts the velocity response of a moving rack. Input torque is applied to the motor of one of the movable racks and embedded in a feedback loop. It is observed from above Fig. 5 Because the system output displays a nonzero steady state error for the given step input u(t), the open loop system type is Type 0. Therefore, we can say that for the above response of the velocity for the moving rack with motor torque input applied as a unit step, the open loop system has a type of Type 0 system.

6.3 Error in Steady State

Error in the Steady state may be expressed in terms of measure of difference of the desired value from that of actual values. Error in steady state is an inherited characteristic of the ratio of input to output for all linear system. Generally, control system with a low errors of steady state regarded as a good control system. The response of such a transfer function at steady state is shown in Fig . 8. The figure depicts that the output of the system exactly equals the input, when the system is in the steady-state. Hence, the steady-state error for the system is zero.

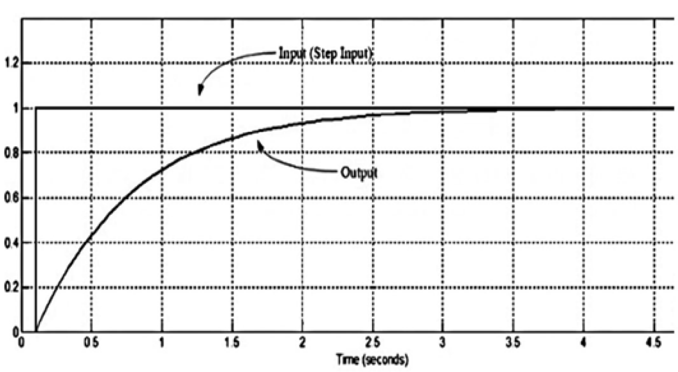

Figure 8: First order transfer function Time response for

$$\boldsymbol{step\ input\ M(t)} = \begin{cases} \mathbf{1} & \mathbf{0 \le t < 40} \\ \mathbf{-1} & \mathbf{40 \le t < 80} \end{cases}$$

From Fig. 5 for velocity response for the moving rack, while input torque is applied to the motor of one of the movable racks and embedded in a feedback loop. The steady state error can be calculated by expression from the basic definition

e(infinity) =input response–output response

Here, the input response is the unit step input and output response is the velocity output response from the Fig. 5. It can be observed from the Fig. 5 that the maximum value of the velocity of the output response for this case obtained must be approximately equal to 0.037 meter per second. By substituting 1 as the input response and 0.037 meter per second for the output response in the basic equation above we can find

e (infinity)=1-0.037=0.963

Therefore, the steady state error obtained is 0.963

6.4 What would be the Steady State Error for the Ramp Input

From Fig. 5 for velocity response for the moving rack, while Torque input is applied to the motor of one of the movable racks and embedded in a feedback loop. The steady state error when the input of the ramp is tu(t) can be expressed as

$$\boldsymbol{e}(\infty) = \frac{\mathbf{1}}{\boldsymbol{K_v}}$$

Where $\mathbf{K_v}$ is the velocity constant,

We know the velocity constant is zero for the ramp input when the Open loop system is Type zero. Therefore $\mathbf{K_v = 0}$ Substituting $\mathbf{K_v = 0}$ in the above relationship the steady state error for a ramp input would be "∞"

6.5 Simulation without Controller for the initial Higher-order Linear System along with Reduced System having First-order

The initial high-order equations are reduced to linear system of the first order for the new cross lapper machine of horizontal type. If unit step and sine functions are made as the inputs of a motor torque, transient response for the corresponding moving rack velocity for the lower order as labeled in Fig. 7 and Fig. 8, is quite similar to that of the original system of higher order therefore it can be verified that the reduced first-order model is accurate. The positive and reverse characteristics for the control motor through torque are responsible for reciprocating the moving rack to its position of re-entry. Hence validity of accuracy of the simplification for the complexity of system is prove.

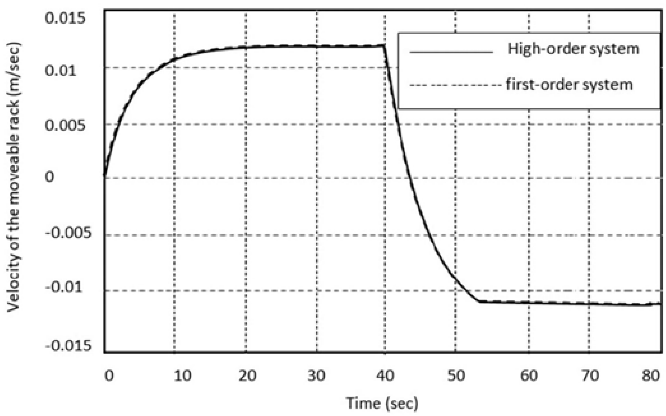

Figure 9: The moving rack's velocity response of first and higher order system with input of motor torque as unit step

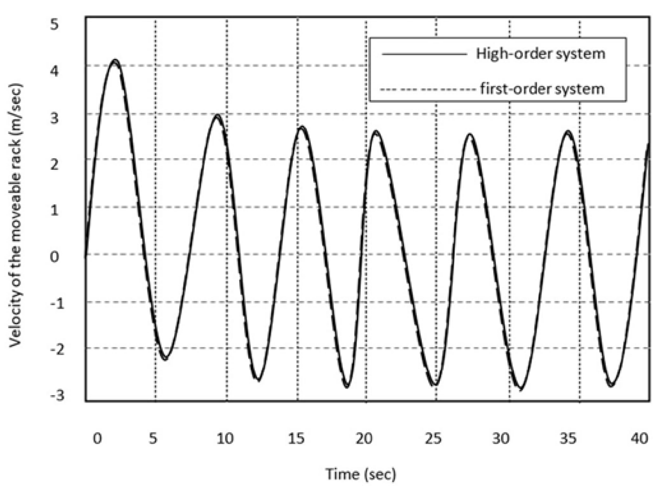

Figure 10: The movable rack's velocity response of first and higher order system with input of motor torque as unit step M(t) = sin t

6.6 Simulation with Controller for Original Linear System having Higher-order along with First-order System

Response of the moving rack velocity of the system is controlled through PI controller. To prevent error in the steady state caused by the transient reaction of the system and to reach to the steady state, design should conform to requirements major poles of closed-loop and the over damping ratio must be kept constant for ζ = 1. 36 with natural frequency ω n = 0.6 radian per second.

The major closed loop poles of the reduced first-order system are −0.816 ± i 0.553. These conditions of the poles are substituted for obtaining the gain parameters for the PI controller which come out to be of KP and KI = 496. 95. Same is depicted by Fig. 11, stable velocity response of the moveable rack can be kept stable, if the input is controlled at the moment of unit step. The significance of this response is that it is over damped having no steady state error, conforming to the expectations. With the application of the approximation criterion for reduction of a higher order system, resulting first order system which can be used for designing of controller, the response of the output is same as that of the original system of higher order. Hence, accuracy and ease of the designed control scheme is proven. In practical situations, such characteristics of input velocity for moving rack are typically applicable, as in case of the movement of moving rack from any position to another position throughout operating cycle. The three stages of cross-lapper system motion is shown by the diagrams i.e. start up, steady state motion and the reverse stage.

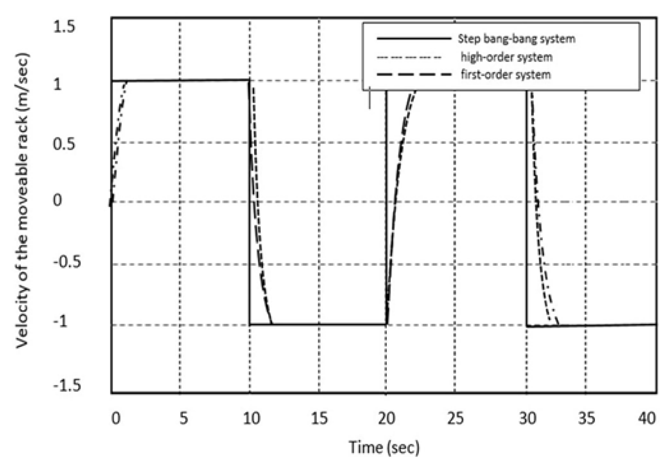

Figure 11: The response of velocity for moving rack for first and higher order system for unit step input as motor torque with PI controller

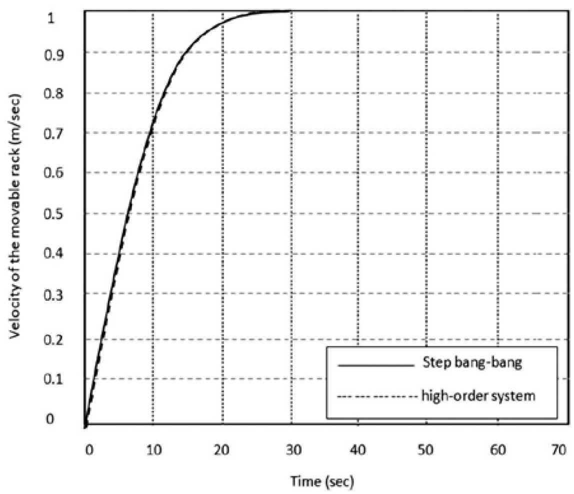

Figure 12: The movable rack's velocity response for higher order system for unit step input as motor torque with state feedback control

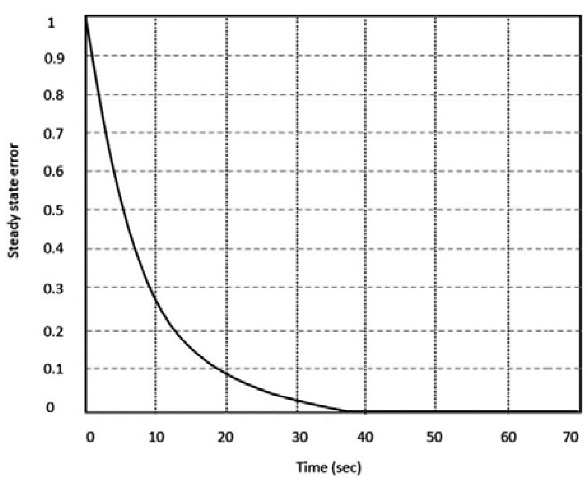

Figure 13: The movable rack's velocity response for high-order system model for unit step input as motor torque with state feedback control

During the start-up stage, the velocities of movement of movable rack is constant. While direction of the drive changes at the commencement of the reverse stage. In the reciprocating motion, it is proved that a good trajectory tracking property as well as the elimination of steady-state errors are achieved by the designed controller. The movable rack possess a smooth steady velocity, therefore it has the major contribution towards achieving the web uniformity and for sustaining the machine's operational life.

7. CONCLUSIONS

The new horizontal cross lapper machine plays a very important function in the unconventional production procedure. The density and the width of the web, as needed for the product, depend on the fast and stable speed control of the mobile storage. Conversations related to the current horizontal cross lapper are not uncommon in the study. Therefore, this paper offers a model and deliberates on the optimal alignment of the current horizontal machine system of the cross-lapper machine type. The lower order system of control design is achieved on the condition of authorization, and the accuracy of the system is guaranteed. Both coordinated as well as co-operative governance, along with the formulation of political inclusion in response to the state,are used in the construction of the system for effective control. The controls can result in a solid speed for the moving rack with precise tracking points and can remove fixed position errors. This constant velocity of the moving rack would therefore create some web similarity and support the machine's life. Meanwhile, the cross lapper may also be operated by using microprocessors. Such a control mechanism may be utilized to meet the requirements of web card engineering, reduce the vibrations of the machine, and reduce power overload.

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