Design and Analysis of Picking and Checking Mechanism of Shuttle Loom

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Design and Analysis of Picking and Checking Mechanism of Shuttle Loom

Major Project

Submitted in partial fulfillment of the requirements

For the degree of

Master of Technology in Mechanical Engineering

(Design Engineering)

By

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Guided by

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MAY 2015

Declaration

This is to certify that

I. The thesis comprises my original work towards the degree of Master of Technology in Mechanical Engineering (Design Engineering) at Nirma University and has not been submitted elsewhere for Degree.

II. Due Acknowledgment has been made in the text to all other material used.

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Undertaking for originality of work

I, **Vipal R. Panchal, Roll. No.13MMED06**, give undertaking that the Major Project entitled "**Design and Analysis of Picking and Checking Mechanism of Shuttle Loom**" submitted by me, towards the partial fulfillment of the requirements for the degree of **Master of Technology** in Mechanical Engineering **(Design Engineering)** of Nirma University, Ahmedabad, is the original work carried out by me and I give assurance that no attempt of plagiarism has been made. I understand that in the event of any similarity found subsequently with any published work or any dissertation work elsewhere; it will result in severe disciplinary action.

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This is to certify that the Major Project Report, entitled "**Design and Analysis of Picking and Checking Mechanism of Shuttle Loom**", submitted by **Mr. Vipal R. Panchal (Roll No. 13MMED06)**, towards the partial fulfillment of the requirements for the award of Degree of **Master of Technology** in Mechanical Engineering **(Design Engineering)** of Institute of Technology; Nirma University, is the record of work carried out by him under our supervision and guidance. The work submitted has, in our opinion reached a level, required for being accepted for examination. The results embodied in this major project work, to the best of our knowledge, have not been submitted to any other University or Institution for award of any degree or diploma.

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Abstract

Shuttle loom is one of the oldest machine for weaving of cloth, and is the vital machinery for power loom industry in India. With this basic loom it is possible to weave nearly all type of cloths very efficiently. This ancient machine is still used by many weavers in India, so they face many problems because of high power consumption and more losses of energy, than any other machinery for this industry.

To improve the production rate of this machine, many changes can be done with respect to mechanical or textile aspects to achieve minimum power consumption. To improve the production rate, it is mandatory to increase the speed of picking mechanism, which plays main role in weaving of the fabric. Various researchers have contribute for this work.

To obtain minimum power consumption, detailed analysis of another auxiliary mechanism called checking, which is also related to picking mechanism, is required. The mechanism operates by applying brakes to the shuttle, which is at very high speed, and prepare the shuttle for next cycle of picking mechanism. It uses spring loaded swells to retard the shuttle. Due to retardation, velocity of shuttle decrease to zero. If these brakes are not released then picking mechanism needs to generate more power to strike picker on the shuttle. Hence it is extremely necessary to design and develop a mechanism to compliment the checking mechanism and to move in the direction of achieving minimum power consumption.

Detailed study is done on the working of checking mechanism of the present loom. The aim of this project is to develop a checking mechanism which releases the spring loaded swell at the time of initialization of picking mechanism and load that spring to brake the shuttle when picking mechanism ends. The kinematic and dynamic analysis is performed for the newly developed mechanism. Dynamic simulations is carried out for the mechanism using commercial simulation package creo 2.0 and the results are validated by theoretical calculations.

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Chapter 1

Introduction

1.1 Introduction

From time immemorial clothing is one of basic need along with food and house for mankind. With the growth of mankind all over the world, there has been a growth of textile manufacturing processes, products and textile machinery simultaneously. The process of development in this area still continues. This industry contributes 4 percent to countries G.D.P and 14 percent to countries industrial production. India has very large source of installed looms for textile manufacturing products. Different types of fabrics are woven by loom to manufacture cloth. According to latest survey by textile ministry of India, India has 1.8 million shuttle looms which is 45 percent of world capacity. There are different types of loom available to wave cloth and it is classified in following paragraph. Then its necessary to understand the weaving process in this loom, basically we will see it according to shuttle loom.

1.2 Classification of Looms

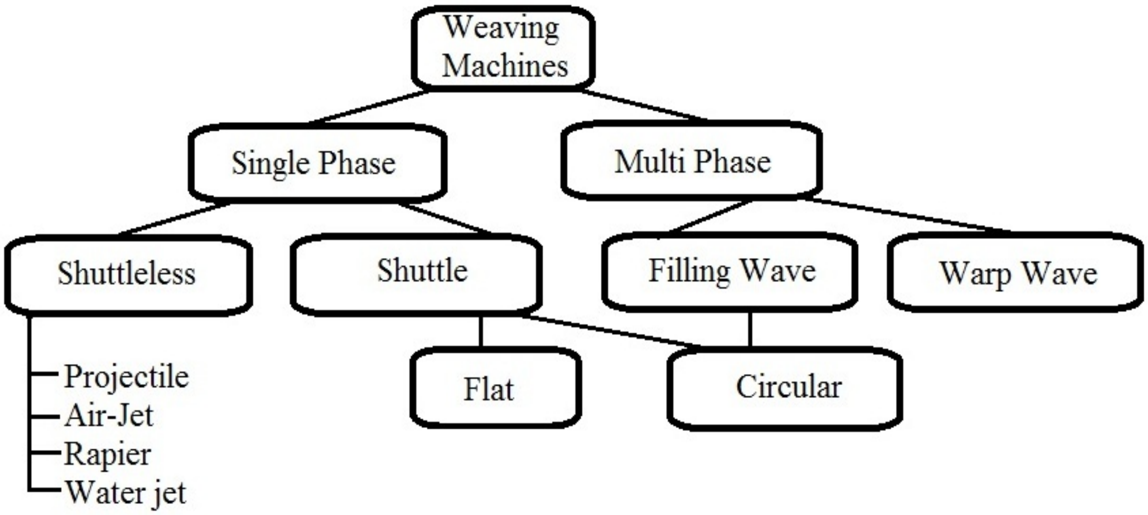


Figure 1.1: Classification of weaving machines

1.2.1 Shuttle Loom

For power loom machinery, shuttle loom is the basic and former weaving machine for all types of fabrics. In shuttle loom filling is inserted by a shuttle that traverses back and forth across the loom width. The shuttle can be made of compress wood, plastic or a combination of both. As the shuttle move across the loom the filling yarn is unwound from the pirn and laid in the shed. The shuttle moves continuously back and forth across the loom

1.2.2 Shuttle less Loom

As the latest technologies for weaving of cloth developed it uses different devices rather than the shuttle, which carries weft from one side to another side for this process. There are many types of shuttle less looms, which are used for weaving such as Projectile Looms; Rapier Looms; Water Jet Looms; and Air Jet Looms.

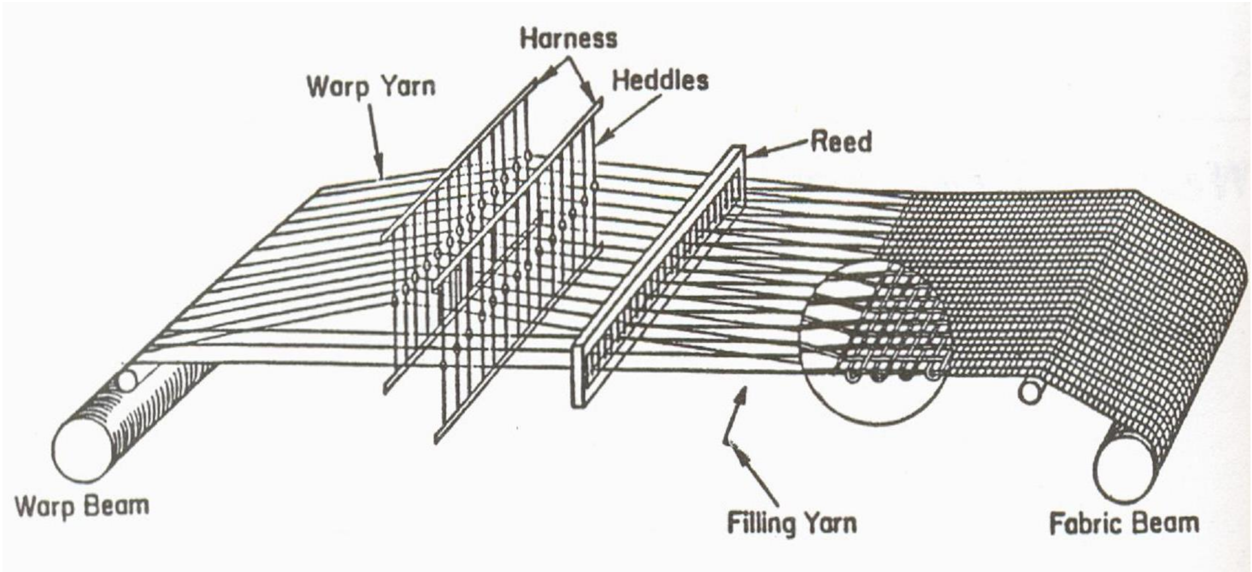


Figure 1.2: Power loom [1]

Projectile loom

The projectile weaving machine was introduced by Sulzer in 1952. And this was the first successful, shuttleless weaving machine. Projectile weaving machine use a projectile equipped with a gripper to insert the filling yarn across the machine. The unique principle of projectile filling insertion allows the insertion of practically any yarn. This loom works up to 300 ppm and is less noisy compared to shuttle loom.

Rapier loom

In rapier loom a 'rapier' is used to insert the filling yarn for weft insertion process. Rapier is a flexible or solid element. The rapier head picks up the filling yarn and carries it through the shed. After reaching the destination, the rapier head returns empty to pick up the next filling yarn, which completes a cycle. Rapier weaving machine can be of two types. It is Single rapier machine and Double rapier machine.

Air jet loom

In Air-Jet weaving compressed air is used to insert filling yarn into the warp shed.The filing is fed into the reed tunnel via tandem and main nozzles. The tandem and main nozzle combination provides the initial acceleration, where the relay nozzle provide the high air velocity across the weave shed.This type of loom is not suitable for heavy threads. But many varieties can produce by this loom.

Water jet loom

In Water-jet weaving machine highly pressurized water is used to inserts the filling yarn. The tractive force is stipulated by the relative velocity between the weft and the water jet. The traction force can be affected by the viscosity of water and the roughness and length of the filling yarn, higher viscosities cause higher tractive forces. The viscosity of water relies on the temperature. For this loom filament yarn of acetate, nylon polyester, and glass are more suitable because it is non-absorbent fabric to water.

Circular looms

These looms are particularly used for making tubular fabrics rather than at fabrics. A shuttle device in it circulates the weft in a shed formed around the machine. A circular loom is primarily used for bagging material.

1.3 Weaving operations

1.3.1 Shedding

Division of warp is called shedding. Shedding is done by heald frames mounted on the base frame of loom. Heald frames are moved with the help of dobby or cam or jacquard in order to create necessary pattern in cloth. Needles are connected with the heald frame and the warp coming from the let off passes through these needles as shown in figure 1.3.

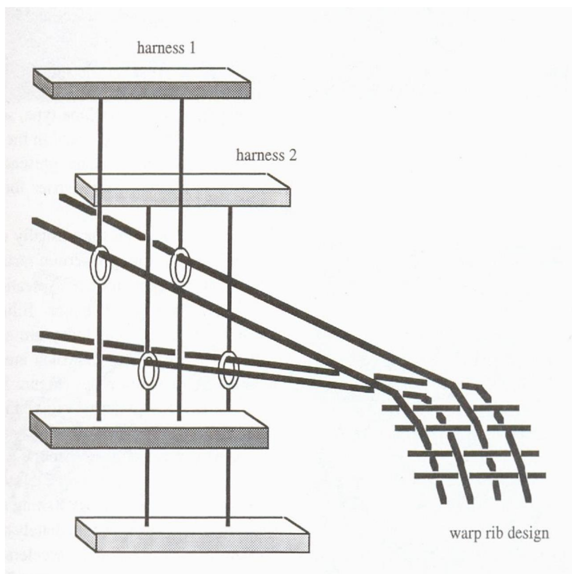


Figure 1.3: Shedding [1]

1.3.2 Picking:

As the harnesses raise the heddles or healds, which raise the warp yarns, the shed is created. The filling yarn in inserted through the shed by a small carrier device called a shuttle. The shuttle is normally pointed at each end to allow passage through the shed. In a traditional shuttle loom, the filling yarn is wound onto a quill, which in turn is mounted in the shuttle. The filling yarn emerges through a hole in the shuttle as it moves across the loom. A process of crossing the shuttle from one side to another side of the loom is known as a pick. In short, shuttle is used for the filling purpose of weft through the warp shed, and this cyclic operation is called picking operation. The inserted weft thread is known as pick as shown in figure1.4

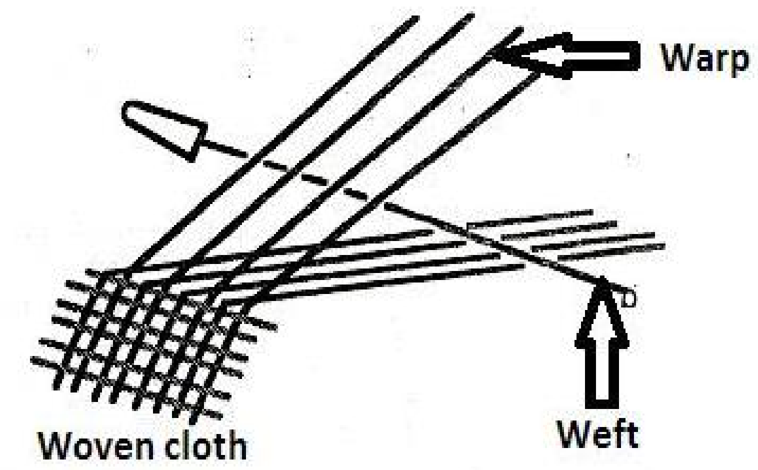


Figure 1.4: Picking [1]

1.3.3 Beat-up

When the weft is inserted from the shed the weft is lies very far from the last thread of woven cloth. To make compact weaving arrangement a reed is used to compact the weft. This operation is performed by beat-up mechanism. The imaginary line from where the fabric starts is called fell. Pushing of the last inserted filling yarn to the fell should be performed to weave cloth. This pushing is performed by beat-up mechanism. For all types of fabric and all type of weaving machine beat-up mechanism plays an important role. The motion of reed is shown in figure1.5.

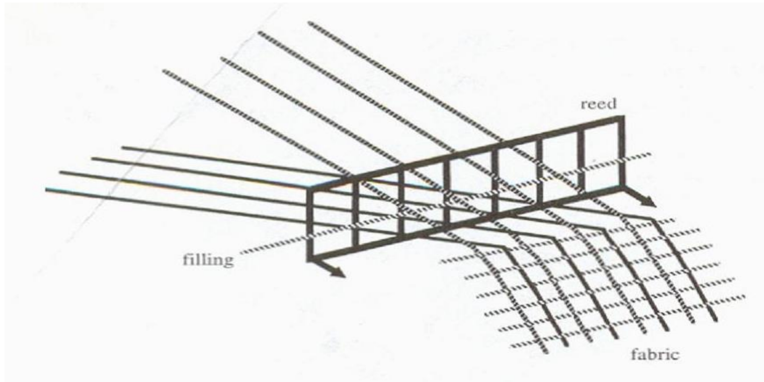


Figure 1.5: Beat-up [1]

1.3.4 Taking up and Let-off

Take-up and let-off are secondary motions for weaving of cloth. After each picks, newly inserted weft is compressed to the cloth and this cloth is wound by the take-up motion. Simultaneously the warp should be released from the beam to feed new yarn to cloth and this warp release mechanism is called let-off motion. Both the motions are operated by the gear trains to achieve desired output rotation of roller.

Chapter 2

Literature Review

Picking and checking mechanisms are one of the essential mechanism for shuttle loom. For study and development of these mechanisms following literature is referred. These literature helps a lot to understand shuttle loom and its mechanisms. It also helps to know developments of this mechanism and inspire to develop some noteful stuff for this mechanisms.

T. Ishida and K. Chikaoka,[3] The shuttle flight conditions are discussed in very basic manner with the relation to an increased speed for shuttle loom. All this conditions are explained by different types of experiments. Loom vibration and shuttle flight conditions are explained for different part relation. With all this experiment data for different condition of loom speed we can justify the parts according to its working criterion.

Zdenek Koloc and Miroslav Vaclavik [4], presented mathematical modelling of the picking mechanism for a rapier loom. The system consists of two subsystems. The first system is a discrete one and the second system is a continuous one. The interaction of these subsystems is concentrated at one point, this point moves relative to the second subsystem dependent on the position of the system. The iteration method is suggested for the solution of the equations of motion for the periodic excitation and for the steady-state motion.

M. Jederan,[5] Various Conditions and method of the theoretical investigations on shuttle movement are explained with detailed calculations. In the present investigations the laws of motion of the picking mechanism have been derived, taking into account its masses and the force of checking the shuttle. The effects of the various parameters of the given picking mechanism on the final velocity of the shuttle, respectively. Control experiments under practical conditions and analysis of the parameters for which the method did not give possibility are being followed up.

N. Gokarneshan, et al[6] This chapter contains analysis of shuttle movement in weaving machine in which the practical data are compared with the theoretical calculations and the process of picking and checking is explained with practical data given by calculations. Kinetics of shuttle picking and checking is also explained with practical data and compared with theoretical calculations. This gives a better view on actual and hypothetical aspects of shuttle loom.

S. Sen,[7] in this thesis a study of automatic shuttle loom dynamics and power consumption of its various mechanisms are explained. As per my project picking and checking mechanism are also explained for dynamics and power consumption calculations. Then various proposed methods for improving the efficiency of these looms have been introduced and their advantages and disadvantages discussed.

Glen E. Johnson [8] presented an analytical estimation of the octave band sound pressure levels at a reference point due to various vibrating fly shuttle loom components. This shows that major noise producing mechanism is picking and checking mechanism. This thesis gives some methods to reduce the noise.

Thomas W. Fox [9] has mentioned basic principles of weaving and its calculations, warp and weft motion, detail description on shedding, picking and beat up operation, Crank and reed motion for beat up mechanism.

M.K.Talukdar, et al[10] have described the various calculations of mechanisms involved in weaving and the basic equations for sley velocity, acceleration and acceleration force of beat up mechanism in their weaving-machine book.

Sabit Adanur [1] has explained the weaving fundamentals of polymers , fibres, yarns etc which are used in woven fabric. In this book, he has also described the classification of Looms with description and the fabric structure ,properties and its testing.

Joseph K. Davidson [11] presented simple slider-crank mechanism connected in series with a spring and mass, here the spring has certain elasticity between the mechanism and an inertial load. Analysis of the slider-crank mechanism is discussed and a method of approximate synthesis that utilizes precision conditions is developed. The driving crank rotates at constant speed. Friction is assumed small.

R. L. Norton.[12] The examples of kinematic and dynamic analysis of mechanism are outstanding, clear exposition of complex topics and use of realistic engineering examples succeeds in conveying design as well as the use of modern tools needed for analysis of kinematics and dynamics of machinery. It also gives vector mathematical and matrix solution methods for analysis of both kinematics and dynamics.

Alexander Liniecki [13] presented method of dynamic synthesis of a slider-crank mechanism to meet specified velocity conditions at three design points including the inertial effects of the masses of the main links is presented. Velocity specifications are developed assuming a constant velocity of the input crank. The dynamic velocity error due to inertial effects is calculated by numerical solution of the differential equations of motion including the torque-speed relation for a typical three-phase electrical motor drive.

V. S. Karelin [14] presented the spatial slider-crank mechanism formulas are obtained to allow the user determine the link sizes and slider offset for a predetermined stroke with an acceptable pressure angle at the extremes of slider travel as well as by a given angle between the cross head guide of the slider and a plane in which the crank revolves. Formulas for determining the sizes of links of the plane slider-crank mechanism as a special case of the spatial mechanism are also presented. The particular cases of mechanisms, having equal angles of pressure as well as at extreme values of an angle between cross head guide and the revolution plane of the crank are discussed. It is shown, that the plane mechanism case provides maximum slider travel when other things are equal.

I. H Thomas [15] The main part of this paper is concern with the mechanics of cam operated mechanism which is used to accelerate the shuttle. The second part of this paper describes a method of ensuring a constant rotational speed independently of the quality of the cutting and mounting of the gear wheels.

Shigley, Uicker, Pennock.[16] Provides the foundation for the study of displacements, velocities, accelerations, and static and dynamic forces required for the proper design of mechanism linkages, cams, and gear systems. Coverage of all analysis and development methods is balanced, with the use of both analytic and graphics tools. For the basic understanding of kinematic and dynamic of machinery, this book is very useful for designer.

A. Ghosh and A. Malik, [17] in this book synthesis of mechanisms are explained from basic. Many cases are explained for synthesis of mechanism with examples which is very useful for understand the concept of synthesis of linkages. Both analytical and graphical methods are explained for synthesis of mechanism.

Oleg Vinogradov [18] , in this book dynamic of mechanism is explained fundamentally. Here, the force analysis of mechanism is explained with different classifications and for different cases.

V.B Bhandari, [19] in this book many machine element are explained with detailed fundamentals . For designing of components all methods are explained, based on both theory and practical aspect. Which is useful for researcher and industrial person. Design procedure is based on Concurrent Engineering, Design for Manufacture and Assembly(DFMA).

Chapter 3

Kinematic analysis for Picking and Checking Mechanism

3.1 Preliminary Remarks

In this chapter kinematic analysis for picking and checking is performed by studying timing diagram for picking and checking. Kinematic factors affecting on this mechanism are explained and Velocity and acceleration for shuttle is find out. Checking mechanism is depends on picking mechanism, so finding all data for picking first.

3.2 Picking mechanism

Picking is a primary motion which intent to propel the weft carrying shuttle along the correct trajectory maintaining requisite velocity through the shed in order to provide lateral sets of threads filling the cloth.

3.2.1 Factors affecting initial shuttle speed

Several factors affects on the initial shuttle speed in an over pick machine which were thoroughly investigated by Thomas and Vincent. The primary factors which are responsible for controlling the shuttle speed are discussed below

Shape of picking tappet

Shape of that part of the picking tappet in contact with the picking bowl will depend on the length of the nose bit of the picking tappet. Depending on the size of the machine width, the nose bits are marked by punches to indicate the range of shuttle speed that would be required for a machine. Wider the machine greater the number of punches marked.

Machine speed

The average machine speed during the shuttle traverse does not matter materially. During picking the machine speed does not differ much. But there is evidence at present time that using sophisticated measuring instruments machine speed does fall during shuttle acceleration.

Time of picking

Variation in the time of picking has a tendency to give higher shuttle speed when picking was formed to be late with respect to the crankshaft position starting from 45^o. The changes in speed with timing were less with under pick when with over pick system, and were absent with over pick when the sley was fixed. Higher shuttle speed is necessary as it has to force through the shed as it progressively lowers as the shuttle emerges out of shed.

Mass of shuttle

The amount of checking is dependent on its mass and its speed. Reduction in momentum is the direct result in reduction of mass and not its speed. It is often viewed that a lighter shuttle moves more slowly than a heavier one but this is not true. From experimental observations, it was found that the shuttle speed is substantially independent of its mass. Timing was not affected though the heavier shuttle has a tendency to be picked later.

Initial gap between picker and shuttle

From time to time in course of routine, weaving the shuttle may be obstructed in its passage through the shed and causes a gap between the shuttle and the picker. This naturally leads to a reduction in shuttle speed for the next pick, causing loom bang-off or shuttle trap. With ordinary nose-bits the fall in speed becomes serious when the initial gap exceeds 25 mm, but with constant nominal acceleration the fall is not that serious.

3.3 Checking mechanism

The objective of the shuttle checking is to retard the shuttle nullifying its kinetic energy to zero. The shuttle checking mechanism is shown in the Figure. The incoming shuttle gets rubbed on the spring loaded swell and thereby the frictional force slows down the shuttle velocity. The velocity of the incoming shuttle is reduced around 30% by the action of swell. The shuttle is finally stopped as it collides with the picker, which is cushioned by a suitable buffer system. The arrangement is shown in figure 3.1

3.3.1 Shuttle mass and checking

The mass of the shuttle gradually decreases as the pirn weaves down. However, the checking force remains unaltered, and hence the effectiveness of checking the shuttle must have to be better. As the shuttle mass is gradually reduced the impact velocity of the shuttle will also be less. The checking system should be efficient to allow for variation in the shuttle speed due to reduction of its content. Furthermore, it

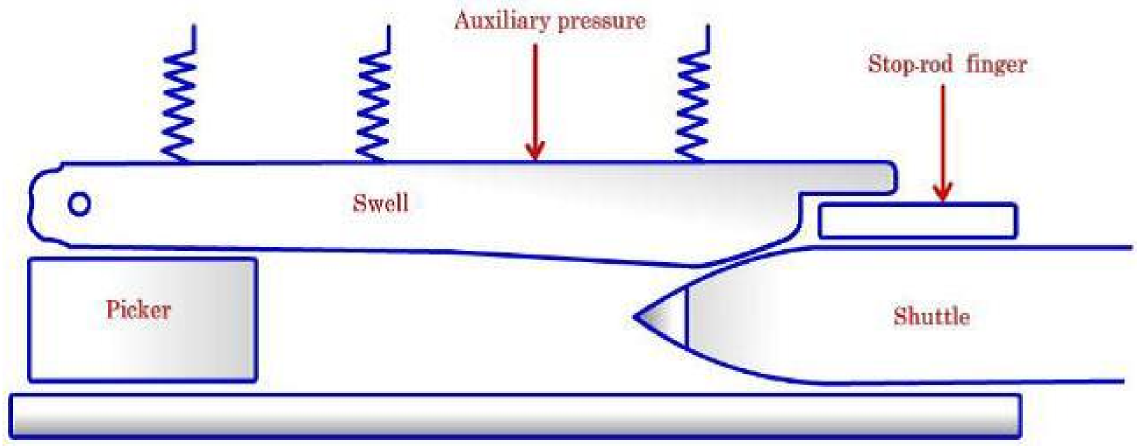


Figure 3.1: Checking Mechanism [2]

should be able to allow for any variation in the shuttle speed due to friction and other resistances during its trajectory through the shed.

3.3.2 Ideal checking conditions

For better operation of checking some usual conditions should be fulfilled.

They are

1. The shuttle should be retarded at the same position after every picks.

2. The retardation between shuttle and picker should be kept as small as possible.

3. The impact velocity of the shuttle with the picker need to keep smaller.

3.3.3 Checking limits

The checking limit is figured out by the capability of the checking apparatus, which can absorb the kinetic energy of the shuttle during the period of a cycle of checking in one box only. As the shuttle come to its stationary position, due to friction between shuttle and swell the kinetic energy is converted into heat. This heat in checking system is essential to control. Checking limit which is depended on heat-disposal capability is important.

1. The maximum temperature generated by the friction of swell and shuttle is about 65^0.
2. The swell temperature is affected by the kinetic energy energy of the shuttle, the maximum temperature is generated at the point, where higher retardation of the shuttle occurs.
3. The swell temperature becomes maximum after 20 min of machine started.
4. The machine speed is kept under 300 ppm, to maintain the quality and effectiveness of swell cover used.

3.3.4 Cause of rejection for checking mechanism parts

Shuttles

1. About 40% of the shuttles were rejected because of breakages of the shuttle wall. Most of the shuttles had become very thin due to wear. These shuttles can be said to have given a good service life.
2. Another 40% of the shuttles were rejected because of chipping, cracks in the wall or tips becoming loose. It was observed that the walls or shuttles made from coarse grain wood and from wooden blocks which were not cut parallel to the grain of the wood were prone to crack and chip
3. The remaining 20% of the shuttles were rejected because of damage due to faulty loom setting such as incorrect alignment of reed and box back plate, or due to defective shuttle flight, harsh picking or shuttle taps.

Pickers

1. About 20% of the pickers were rejected due to damaged or worn-out picker foot, 15% due to expanded spindle holes, and 10% due to loosening of rivets. Such defects have to be attributed to manufacturing or raw material deficiencies.

2. About 15% of the picker were rejected due to incorrect striking of the shuttle or excessive use of one side of the picker. These have to be attributed to incorrect loom settings and work practices.

3. The remaining 40% of the pickers were rejected after a gradual expansion of the striking point or of the spindle hole, or due to worn-out picker foot and could be said to have given a satisfactory service life.

3.4 Shuttle movement for picking mechanism

Following data contains details for proposed loom. This specifications are for 'Ganesha',a product of Dynamic Looms India PVT LTD., which is under shuttle loom category.

Weight nearly: 1000 kg

Speed range: 120 to 150 P.P.M

Material: Mostly casting

Capable of weaving nearly all fabrics.

Traverse = 1.523 meters

Shuttle Length= 0.4064 m

Shuttle weight = 0.450 kg

Loom Speed = 140 R.P.M

110 degrees allocated for the travel of shuttle

3.4.1 Time traverse for shuttle

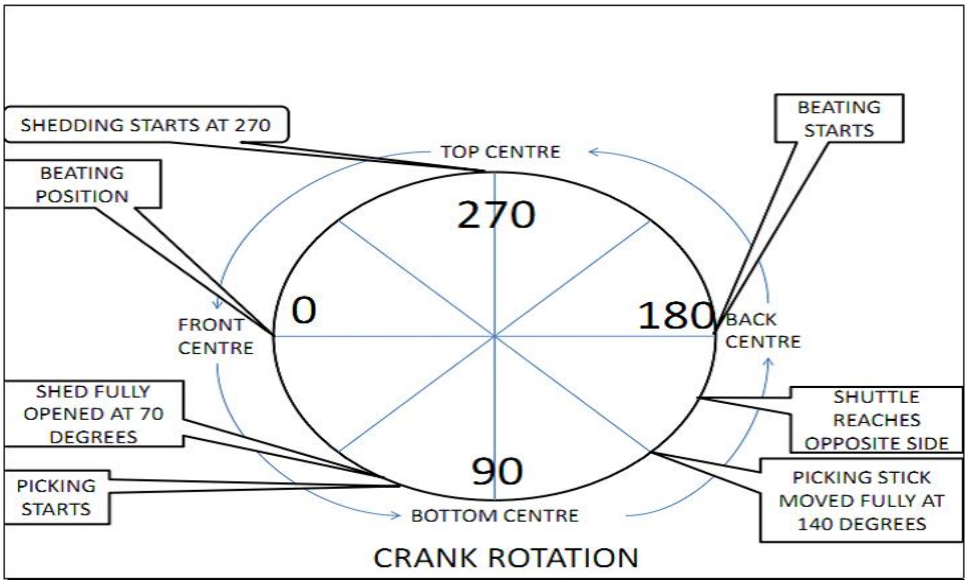


Figure 3.2: Operational layout with time based on crank rotation

All the operations are represented by crank rotation angle,which also gives the time for different operations.

Picking and Checking timing

80^0-110^0: Picking mechanism operates

110^0: Shuttle enters in the shed

250^0: Shuttle leaves the shed

270^0: Shuttle strikes the swell in the shuttle box

300^0: Shuttle comes to rest

The traverse time can be found by,

1 revolution of crankshaft is taking place in $\frac{60}{N}$ sec

360^0 of crankshaft is taking place in $\frac{60}{N}$ sec

θ^0 of crankshaft is rotating in

$$t = \frac{\theta}{6N}\ sec \tag{3.1}$$

$$t = \frac{110}{6 \times 140}\ sec$$

$$t = 0.13095\ sec$$

where,

t= time

θ= Degrees of crank shaft rotation

N= Loom Speed

3.4.2 Velocity of the shuttle

To find forces at each part it is necessary to calculate the velocity of the shuttle. Velocity is given by

$$v = \frac{6(R+L)N}{\theta} \tag{3.2}$$

$$v = \frac{1.9304 \times 6 \times 140}{110}$$

$$v = 14.74\ m/s$$

Where,

L= Effective length of shuttle

R= Width of the warp reed

3.4.3 Max.Velocity of the shuttle

As seen in above equation, the velocity found by the rotation of crank is the average velocity of the shuttle, but the velocity at which the shuttle leaves the picker is the maximum and is of importance. So, Assuming retardation equal to 9.8 m/s^2 from the equation of motion.

$$s = ut + \frac{1}{2}(at^2) \tag{3.3}$$

$$1.9304 = (u \times 0.13095) - \frac{1}{2}(9.8 \times 0.13095^2)$$
$$u = 15.3831\ m/s$$

3.4.4 Acceleration of shuttle

Acceleration of shuttle is affected only in the region, when the shuttle is in contact with the picker till the stroke of the stick and so it will be accelerated only during this span.

$$(v^2 - u^2) = 2as \tag{3.4}$$

$$a = \frac{v^2 - u^2}{2s}$$
$$a = \frac{(14.74)^2}{2 \times 0.254}$$

$$a = 427\ m/s^2$$

Where,

u=initial velocity of shuttle=0

v=final velocity of shuttle

s= distance of shuttle acceleration

3.4.5 Force to accelerate the shuttle

To achieve such a high velocity and acceleration, very high amount of force required. Mass of the shuttle and acceleration of shuttle are required get the force. The force is given by

$$F = ma \tag{3.5}$$

$$F = (0.45)(427)$$

$$F = 192.16\ N$$

3.4.6 Power required for picking

Power reduction is the general motive for any researchers and to compare power consumption for this loom, it is important to find out for this purpose. Power is denoted by P

$$P = \frac{3mN^3(R+L)^2}{\theta^2} \times 10^-4\ kW \tag{3.6}$$

$$P = \frac{3(0.45)(140)^3(1.9304)^2}{110^2} \times 10^-4$$

$$P = 0.1140\ kW$$

3.5 Kinetics of checking mechanism

Some of the important parameters are calculated here for checking mechanism. These calculations are based on input data of picking mechanism. Checking Mechanism is used to nullify the velocity of shuttle, so all forces are absorbed by checking mechanism components.

Maximum Shuttle velocity is 15.3831 m/s

Shuttle retardation is 14.74 m/s

Traverse Time for shuttle is 0.13095 sec

The terminal velocity at the end of its flight would be

$$V_t = 15.3831 - (14.74 \times 0.13095)$$

$$V_t = 13.4528 \ m/s$$

The actual retardation between the shuttle striking the swell first and its contact with the picker is found by using the relation:

$$V^2 = U^2 - 2as$$

$$10.46^2 = 15.3831^2 - (2 \times \times 0.127)$$

$$a = \frac{15.383^2 - 10.46^2}{2 \times 0.127}$$

$$a = 500.886 \ m/s^2$$

Where,

V= Velocity after collision with swell

U= Velocity before collision with swell

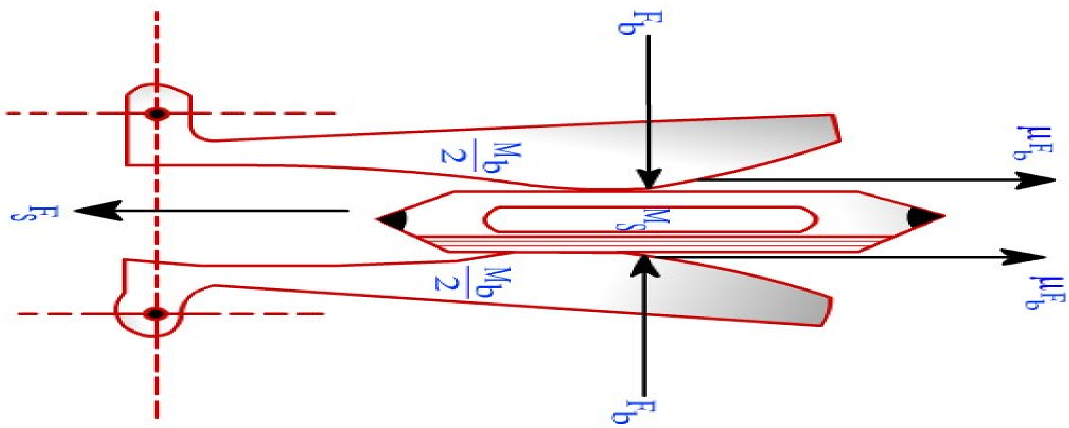


Figure 3.3: Forces acting during shuttle strikes the swell [2]

Retarding force acting on the shuttle during its collision with collision with the swell. The force excreted by swell on shuttle during the collision is reduced by about 32%.

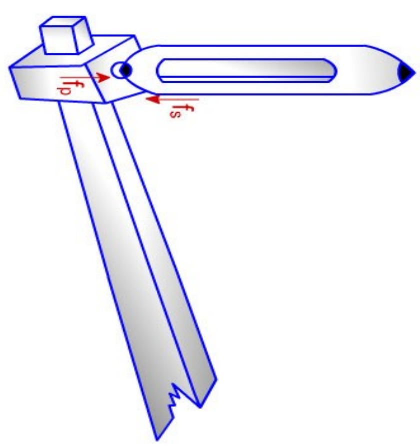


Figure 3.4: Forces acting during shuttle strikes the picker [2]

$$F_s = 2\mu F_b$$

$$F_s = 2 \times 0.25 \times 61.5$$

$$F_s = 30.75\ N$$

Force excreted by swell on shuttle during the collision.

$$F_b = \frac{1}{2} M_b a_b$$

$$F_b = 61.5$$

The force excreted by the picker against shuttle during the collision is given by,

$$F_p = F_s$$

$$F_p = 30.75\ N$$

3.6 Proposed mechanism for checking

Previous mechanism in this machine contains only spring loaded swells, which only brakes the shuttle. This brakes do not release at the time of picking, for picking operation a picker hit the shuttle to reach the next shuttle box. For this striking period picker need to develop more power to overcome friction between shuttle and swell. If this springs are unloaded at the time of stroke then the picker need to generate less power to throw the shuttle at next shuttle box. Proposed mechanism is suggested for this problem, which will release the spring for next cycle of checking mechanism. Previous mechanism is shown in following model.

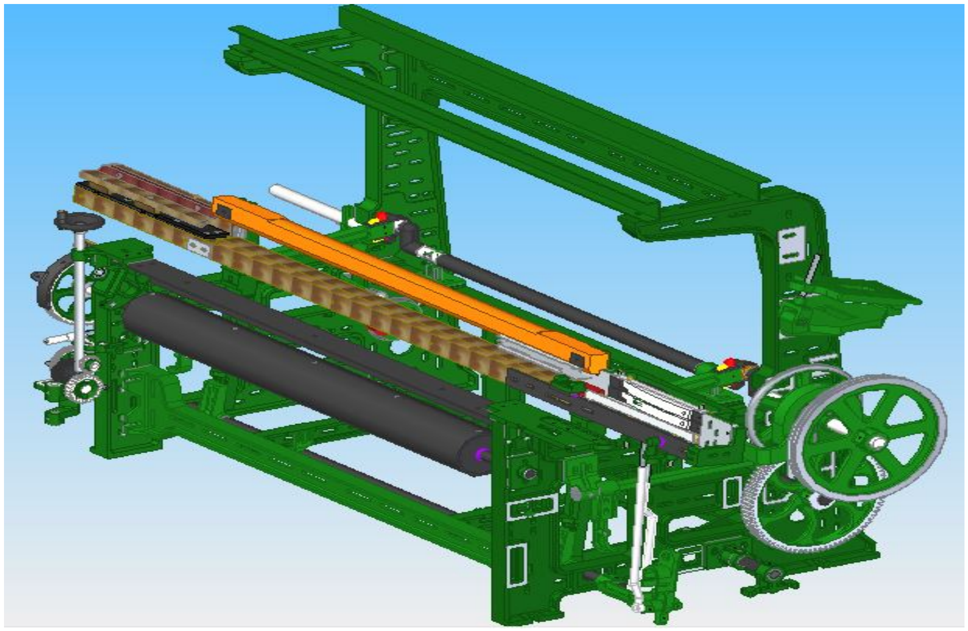


Figure 3.5: Previous Model of shuttle loom

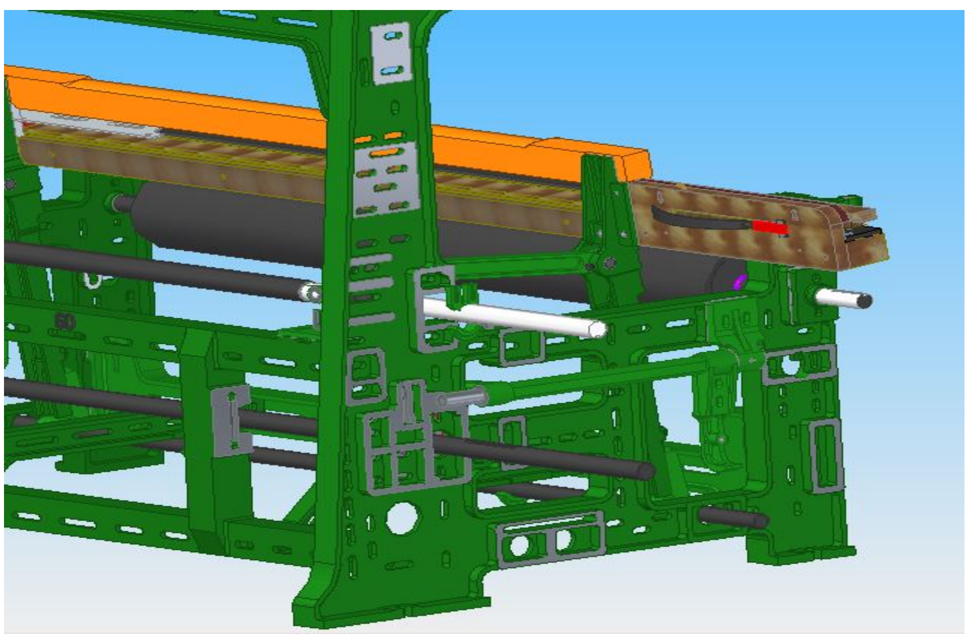


Figure 3.6: Previous Model of shuttle loom

3.6.1 Synthesis for proposed mechanism

Before developing the new mechanism for checking a synthesis for new mechanism has carried out to obtain the link length. This mechanism is a type of slider-crank mechanism. The synthesis is carried out for slider-crank mechanism.

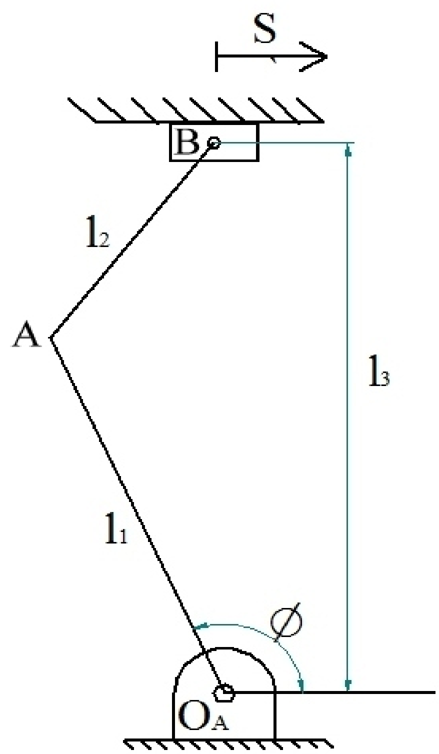


Figure 3.7: schematic diagram for linkages

Coordinates of A and B points,

$x_A = l_1 cos\phi$, $\qquad x_B = S$

$y_A = l_1 sin\phi$, $\qquad y_B = l_3$

Then expressing distance AB= l_2

$$AB^2 = (x_B + x_A)^2 + (y_B - y_A)^2$$
$$l_2^2 = (S + l_1 cos\phi)^2 + (l_3 - l_1 sin\phi)^2$$
$$l_2^2 = l_3^2 + l_1^2 + S^2 + 2l_1 Scos\phi - 2l_3 l_1 sin\phi$$
$$S^2 = l_2^2 - l_1^2 - l_3^2 + 2l_1 l_3 sin\phi - 2l_1 Scos\phi$$
$$S^2 = 2l_1 l_3 sin\phi - 2l_1 Scos\phi - (l_1^2 - l_2^2 + l_3^2)$$
$$S^2 = K_1 sin\phi - K_2 Scos\phi - K_3$$

Where, $K_1 = 2l_1 l_3$

$K_2 = 2l_1$

$K_3 = l_1^2 - l_2^2 + l_3^2$

Synthesis of this mechanism is calculated by Chebyshev spacing with three precision point method. Here for this mechanism the displacement of the slider is proportional to the crank rotation in the interval $90^o \leq \phi \leq 117^o$

The desired relationship between S and ϕ is

$$\frac{S - S_s}{S_f - S_s} = \frac{\phi - \phi_s}{\phi_f - \phi_s} \tag{3.7}$$

The values of S_s and S_f is assumed to be 2 and 42 respectively. And the values of ϕ_s and ϕ_f is assumed to be 117^o and 90^o respectively.

The accuracy points are determined by chebyshev spacing

$$\phi_1 = a + hcos\left(\frac{2l - 1}{2k}\right)\pi \tag{3.8}$$

Where, $a = \frac{\phi_s - \phi_f}{2}$ and $h = \frac{\phi_f - \phi_s}{2}$

The values of accuracy points are determined by means of equation3.8.

$$\phi_1 = 91.8,\ \phi_2 = 103.5,\ \phi_3 = 115.2$$

The corresponding values of S are obtained by means of equation 3.7

$$S_1 = 36.84,\ S_2 = 12,\ S_3 = 2.18$$

Putting all values in displacement equation

$$K_1 + 1.15K_2 - K_3 = 36.84^2$$
$$0.97K_1 + 2.8K_2 - K_3 = 12^2$$
$$0.91K_1 + 0.928K_2 - K_3 = 2.18^2$$

Solving for $K_1,\ K_2,\ K_3$ we have

$K_1 = -15151.11,\ K_2 = 459.78,\ K_3 = -13265.17$

Thus,

$l_1 = 91.5,\ l_2 = 43,\ l_3 = 88$

Here, all the dimensions of linkages are shown above.

3.6.2 Position analysis of proposed Mechanism

Position analysis of proposed mechanism is performed by simple algebraic Method. Link dimensions are mentioned

l_1	91.5
l_2	43
l_3	88
ϕ	117^o

For x-axis: $l_2 cos\phi_2 = S - cos\phi$

For y-axis: $l_3 = l_1 sin\phi + l_2 sin\phi_2$

$$cos\phi_2 = \frac{S - l_1 cos\phi}{l_2} \tag{3.9}$$
$$sin\phi_2 = \frac{l_3 - l_1 sin\phi}{l_2} \tag{3.10}$$

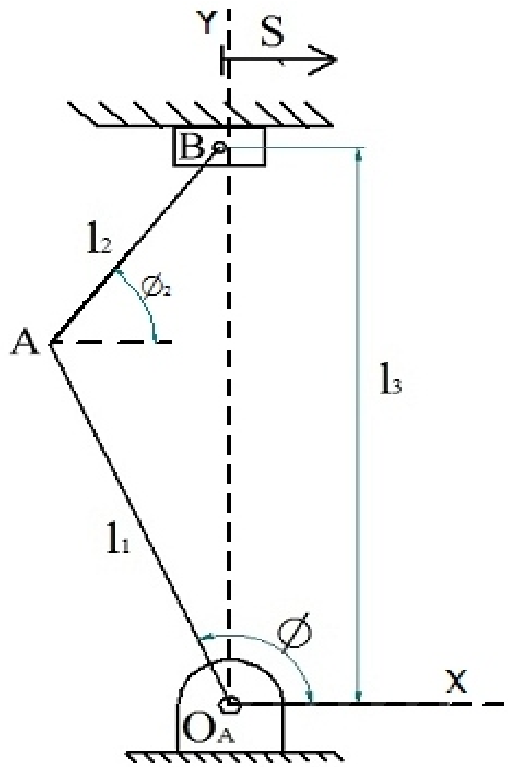


Figure 3.8: schematic diagram for linkages

$$\phi_2 = sin^{-1}\Big(\frac{l_3 - l_1 sin\phi}{l_2}\Big)$$

$$\phi_2 = sin^{-1}\Big(\frac{88 - 91.5 sin117}{43}\Big)$$

$$\phi_2 = 8.63^o$$

$$S = l_2 cos\phi_2 + l_1 cos\phi$$

$$S = 43cos(8.63) + 91.5cos(117)$$

$$S = 1.13\ mm$$

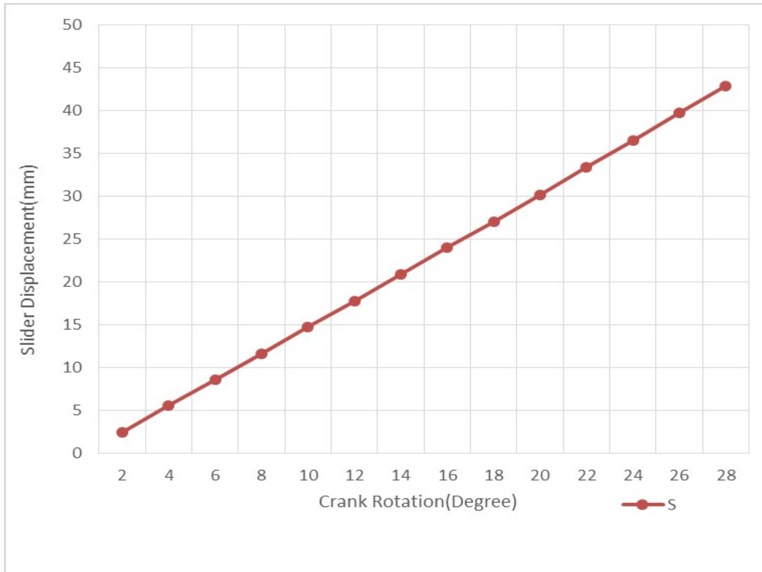


Figure 3.9: Plot of crank and slider position according to theoretical data

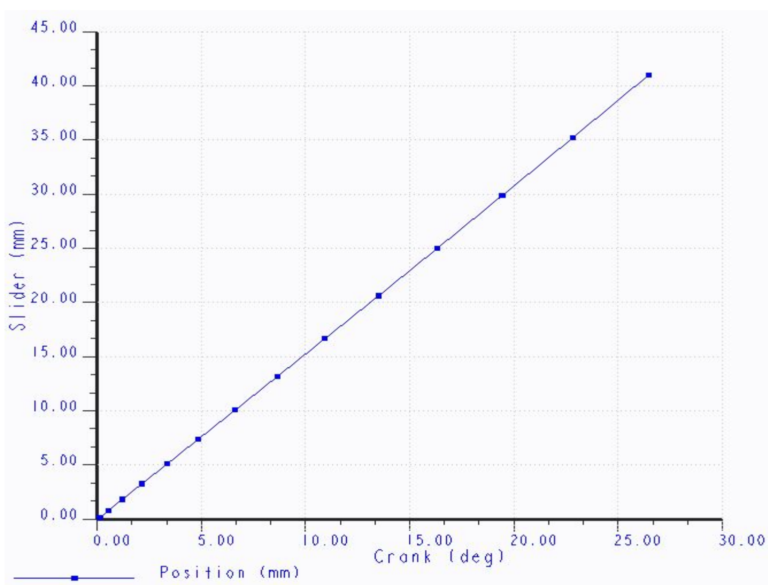


Figure 3.10: Plot of crank and slider position according to Mechanism simulation

3.6.3 Force Analysis for Negligibly small inertial forces

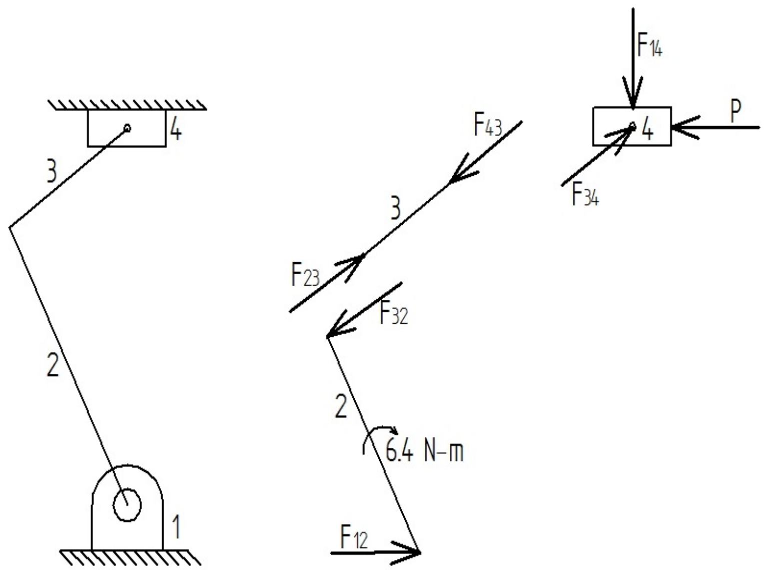


Figure 3.11: Free Body Diagram for linkages with negligible inertial forces.

The skeleton of the mechanism and the free-body diagrams of links are shown in figure 3.11. Here τ is the known external Torque i.e 6.4 N-m, 'P' is the unknown resistance spring force, and the inertial forces can be neglected because of very small values. All other forces shown in Figure 3.11 are internal (constraint) forces. Each internal force is identified by two indices, first one indicates the adjoining link and the second one the link to which the force is applied.

On link 3 external forces are not applied. so,It follows that the internal forces are directed along the link and they are equal in magnitude and opposite to each other. Thus, the forces in the joints of link 3 contain only the unknown magnitude and the directions of these forces are known.

Then all other unknown forces are like this.

$$F_{43} = F_{32} = -F_{23} = -F_{34}$$

For Link-2,

$$F_{32} = \frac{\tau}{l_2}$$

$$F_{32} = 70N$$

For link-4,

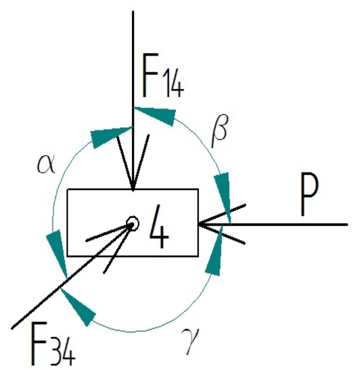


Figure 3.12: Free Body Diagram for link-4

For figure 3.12 values of angles are as follows,

$$\alpha = 98.63^o, \beta = 90^o, \gamma = 171.37^o$$

$$\frac{F_{34}}{sin\beta} = \frac{F_{14}}{sin\gamma} = \frac{P}{sin\alpha}$$

$$P = F_{34}\frac{sin\beta}{sin\alpha}$$

$$P = 69N$$

Spring design for checking mechanism

To achieve the checking mechanism need we have to put a spring which apply force on the swell. That force should be capable to slow down 30% of shuttle velocity . To reduce that much velocity spring should apply 63 N force on swell. so, design of spring should be done under that loading condition. The deflection of spring is considered by slider displacement of proposed mechanism. To design this spring, spring index is taken most common one. Spring is designed by trial and error method. All parameters are calculated as follows.

Input Parameters,

Deflection(δ)=40 mm

Axial Load(P)= 69 N

Stiffness(k)=1.575

Spring Index(C)= 8

Wahl's stress Factor,

$$K = \frac{4C-1}{4C-4} + \frac{0.615}{C}$$

$$K = 1.1840$$

$$\begin{aligned} Induced\ stress(\tau) &= \frac{8KPC}{\pi d^2} \\ &= \frac{1520}{d^2} \end{aligned}$$

Itr-1: Assuming Diameter of wire as 1 mm

$$S_{ut} = 1570 \frac{N}{mm^2}$$

$$\begin{aligned} permissible\ stress(\tau_d) &= 0.3 S_{ut} \\ &= 471\ N/mm^2 \end{aligned}$$

$$Induced\ stress(\tau) = \frac{1520}{1^2}$$
$$= 1520 N/mm^2$$
$$\tau > \tau_d$$
$$Hence, Design\ is\ not\ safe$$

Itr-2: Assuming wire diameter as 2 mm

$$S_{ut} = 1420 \frac{N}{mm^2}$$

$$permissible\ stress(\tau_d) = 0.3 S_{ut}$$
$$= 426\ N/mm^2$$

$$Induced\ stress(\tau) = \frac{1520}{2^2}$$
$$= 380 N/mm^2$$
$$\tau < \tau_d$$
$$Hence, Design\ is\ safe$$

Now, calculating spring parameters for 2 mm wire diameter

Mean Diameter of Spring(D)

$$C = \frac{D}{d}$$
$$D = (8)(2)$$
$$D = 16mm$$

Total No. of Turns(N_t)

$$
\begin{aligned}
No.\ of\ active\ turns(N) &= \frac{Gd^4}{8D^3K} \\
&= 25.22mm \\
&\approx 26mm
\end{aligned}
$$

Taking square and ground ends,

$$
\begin{aligned}
Total\ no.\ turns N_t &= N + 2 \\
&= 28\ turns
\end{aligned}
$$

$$
\begin{aligned}
Solid\ Length &= N_t \times d \\
&= 56mm
\end{aligned}
$$

$$
\begin{aligned}
Toatal\ axial\ gap\ between\ coils &= (28 - 1) \times 0.6 \\
&= 16.2mm
\end{aligned}
$$

$$
\begin{aligned}
Free\ length &= Solid\ Lengh + Toatal\ axial\ gap + \delta \\
&= 56 + 16.2 + 40 \\
&= 112.2mm
\end{aligned}
$$

$$
\begin{aligned}
pitch\ of\ length &= \frac{Free\ Length}{N_t - 1} \\
&= 4.15mm
\end{aligned}
$$

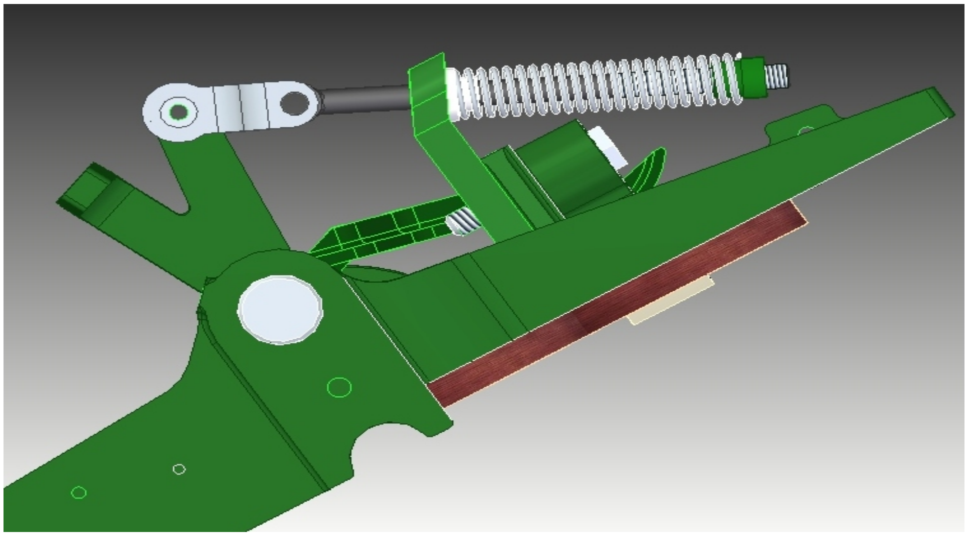


Figure 3.13: Model of proposed Mechanism

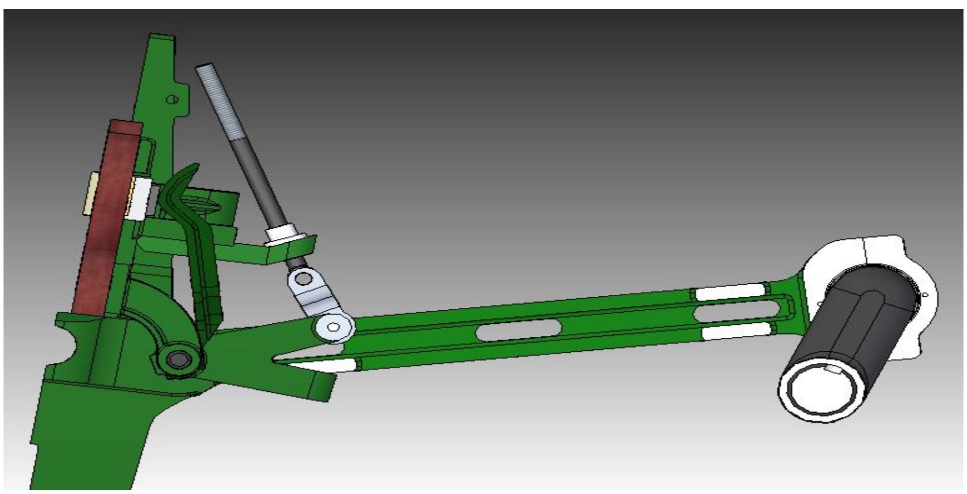


Figure 3.14: Model of proposed Mechanism

The above mechanism 3.13 is proposed to release the brake when the picker is about to strike the shuttle.

The figure3.14 shows the application of proposed mechanism to a parametric model for the previous machine

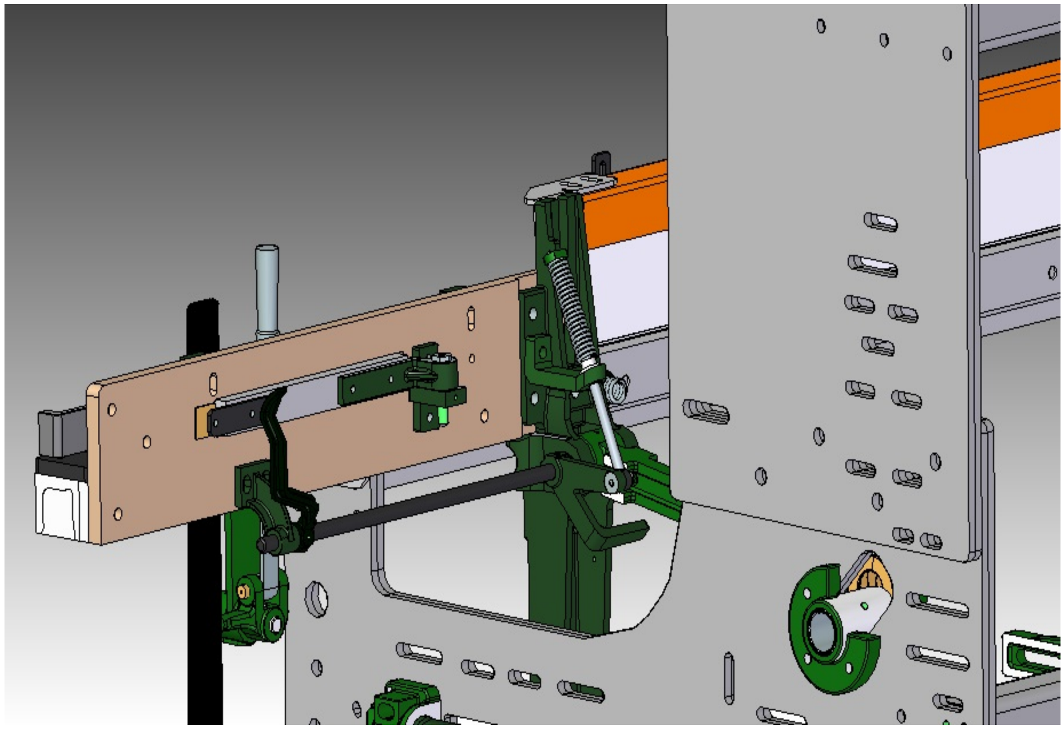


Figure 3.15: Model of proposed Mechanism

Chapter 4

Proposed design for Smooth Picking

For picking mechanism, the operation should smooth and jerk free in running condition. Picking is classified as smooth picking and harsh picking according to its performance while running. When the fabric is very much thin then picking must be smooth. Smooth picking is achieved by designing the component under high factor of safety so that it can sustain extra forces. This extra forces are generated when shuttle is misplaced by its path of free flight. If shuttle is not placed exactly at its prescribed starting position then it disturbed the static equilibrium forces in shaft. Picking Shaft is under twisting moment so its better to design circular shaft earlier, its cross section was hexagonal.

Shuttle is flying on its path which is predefined on sley. In case of shuttle disturbed from its way then whole sley assembly, which is oscillating, will have unbalanced forces which disturbs force equilibrium at sley assembly support bearings mounted on tappet shaft. Single groove ball bearing is preferred for this support so here choose bearing unit to simplify assembly of sley on tappet shaft and make picking smooth. For this, bearing selection is done from bearing manufacturer's catalogue.

4.1 Design of shaft for picking mechanism

To modify shaft for better performance in twisting moment its cross section is changed to circle from hexagon. Present shaft is over weighted to sustain twisting moment with hexagonal shape. Its inscribed circle diameter is 39.26 mm. Present shaft is shown in figure 4.1

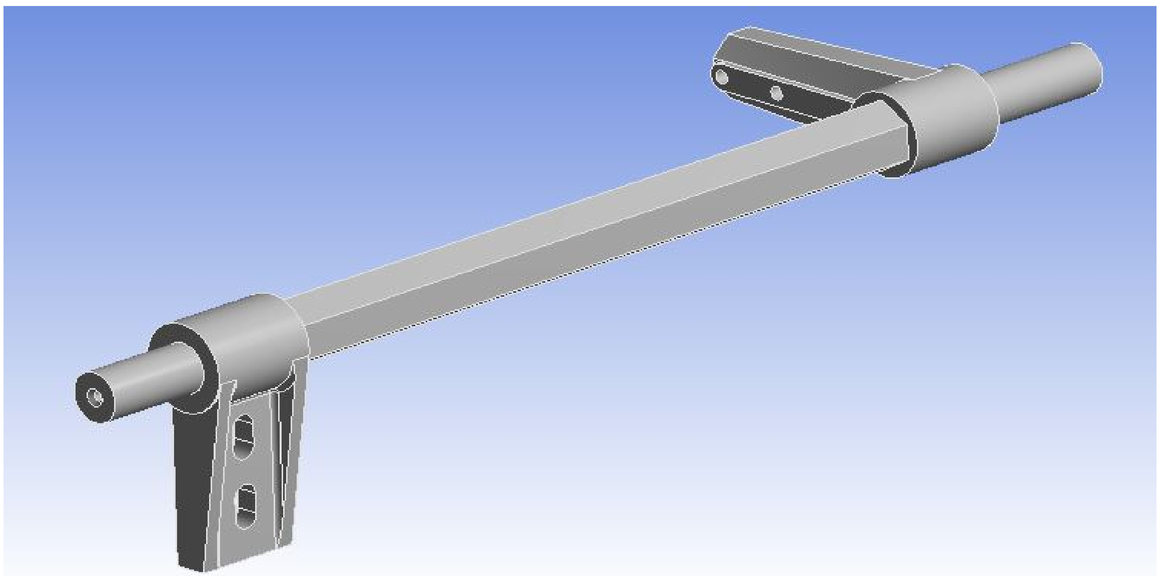


Figure 4.1: Model of present shaft

To perform Finite Element Analysis for stress validation in Ansys, model is meshed under local meshing conditions. Hex-dominant type mesh with element size 10 mm. Meshed Model is shown in figure 4.2

For evaluating the stresses due to structural loads, one end of the shaft is kept fix and another end loaded with roller, pushed by cam is applied linear force at the roller pin hole. Load exerted by the cam on roller is 200 N. Figure4.3 shows the stress distribution due to torsion.

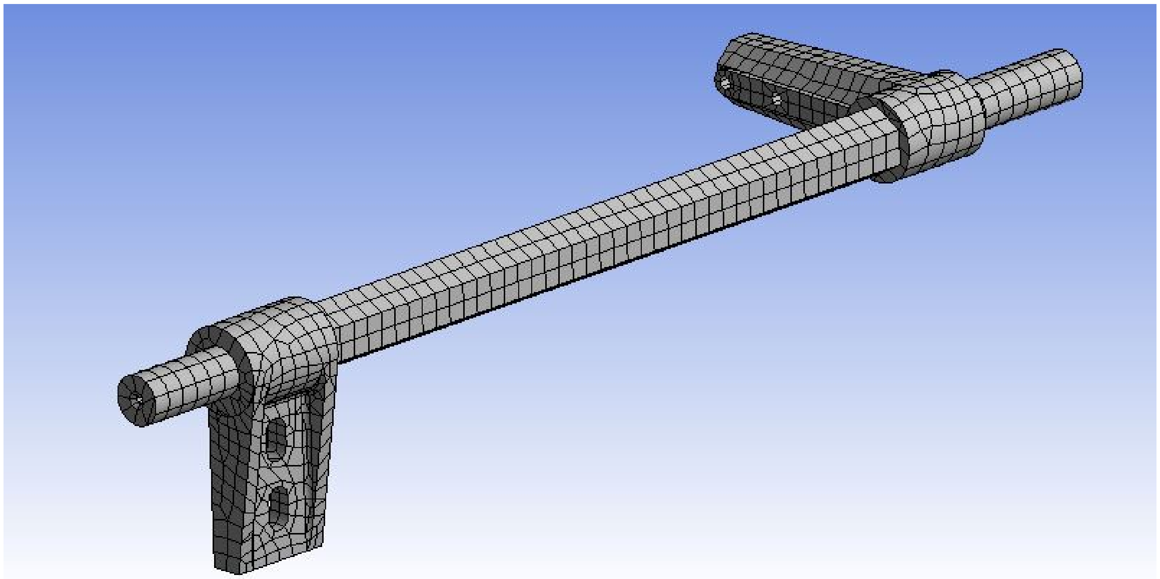


Figure 4.2: Mesh model of present shaft

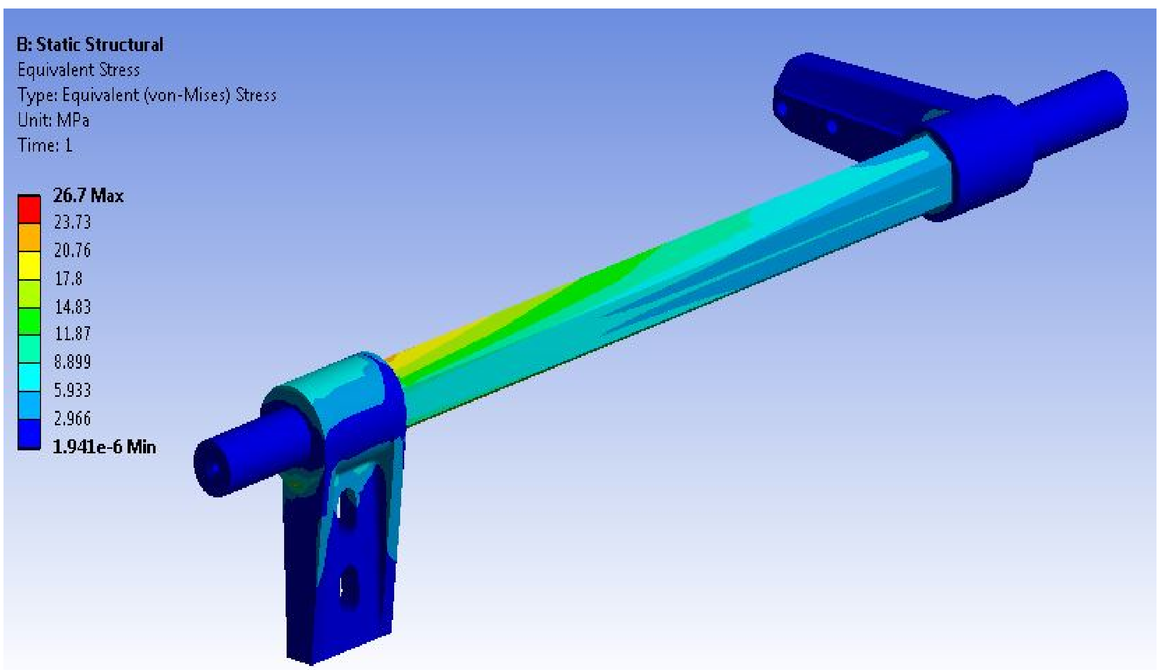


Figure 4.3: Stress distribution in present shaft

Design For Proposed shaft

Material property for Grey cast iron grade 25

$Ultimate\ tensile\ strength(S_{ut}) = 170\ to\ 180\ Mpa$

$Ultimate\ yield\ strength(S_{yt}) = 110\ Mpa$

According to ASME code for shaft design the permissible shear stress for shaft is taken 30% of yield strength or 18 % of the ultimate tensile strength of the material, whichever is minimum.

$\tau = 0.30S_{yt} = 0.30(110) = 33Mpa$

$\tau = 0.18S_{ut} = 0.18(175) = 31.5Mpa$

Take minimum value of τ, which is 31.5 Mpa

For suddenly applied load (Heavy Shock), the values for combined shock and fatigue factor for bending moment(K_b) and combined shock and fatigue factor for torsional moment(K_t) are 2.5 and 2.2 respectively.

For shaft support with bearing at both ends having length 562 mm. Bending Moment(M_b) for this shaft is 13783 N-mm and Torsional Moment(M_t) for this shaft is 22625 N-mm.

To design shaft under torsional moment subjected to fluctuating loads using ASME code for shaft design.

$$\tau = \frac{16}{\pi d^3}\sqrt{(K_b M_b)^2 + (K_t M_t)^2}$$

$$d^3 = \frac{16}{\pi 31.5}\sqrt{(2.5 \times 13783)^2 + (2.2 \times 22625)^2}$$

$$d = 22\ mm$$

$$d \approx 25\ mm$$

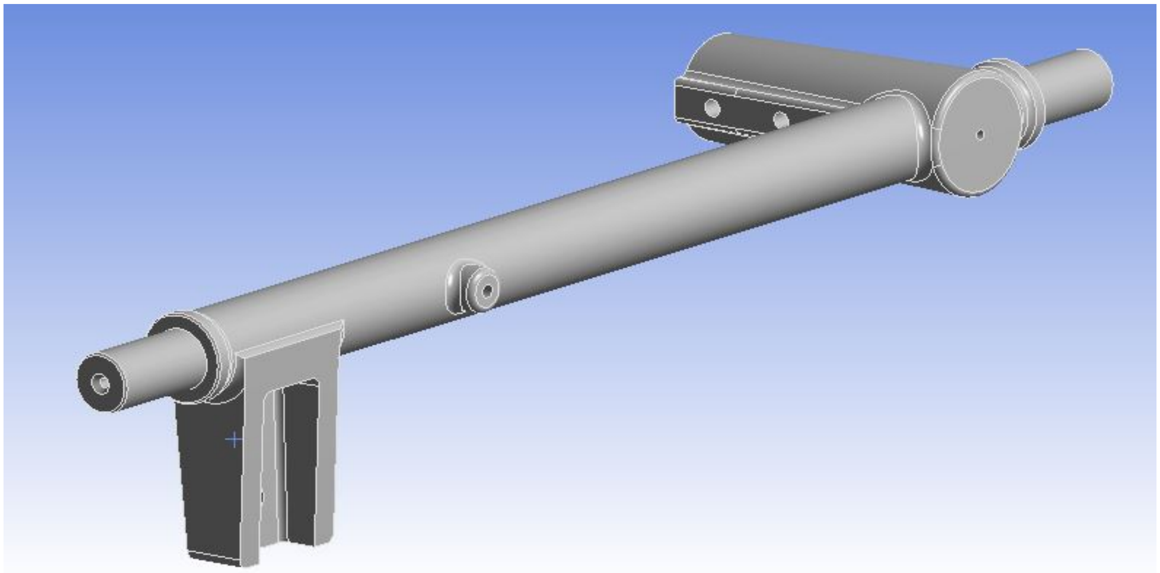


Figure 4.4: Model of proposed shaft

For the same length and same loading condition the diameter and the X-section of shaft is changed. The shape is changed from hexagonal to circle and diameter is changed from 39 mm to 25 mm. The model for proposed mechanism is shown in figure 4.4.

To perform Finite Element Analysis for stress validation in Ansys, model is meshed under local meshing conditions. Hex-dominant type mesh with element size 10 mm. Meshed Model is shown in figure 4.5

For evaluating the stresses due to structural loads, one end of the shaft is kept fix and another end loaded with roller, pushed by cam is applied linear force at the roller pin hole. Load exerted by the cam on roller is 200 N same as present condition. Figure4.6 shows the stress distribution due to torsion.

Hence, from FEA analysis for present and proposed shaft, it is concluded that present shaft is over safe with improper shape. Proposed shaft is designed circular to sustain torsional moments. With theoretical calculation and FEA of shaft it is concluded that with reduced dimensions, shaft is safe under same loading condition.

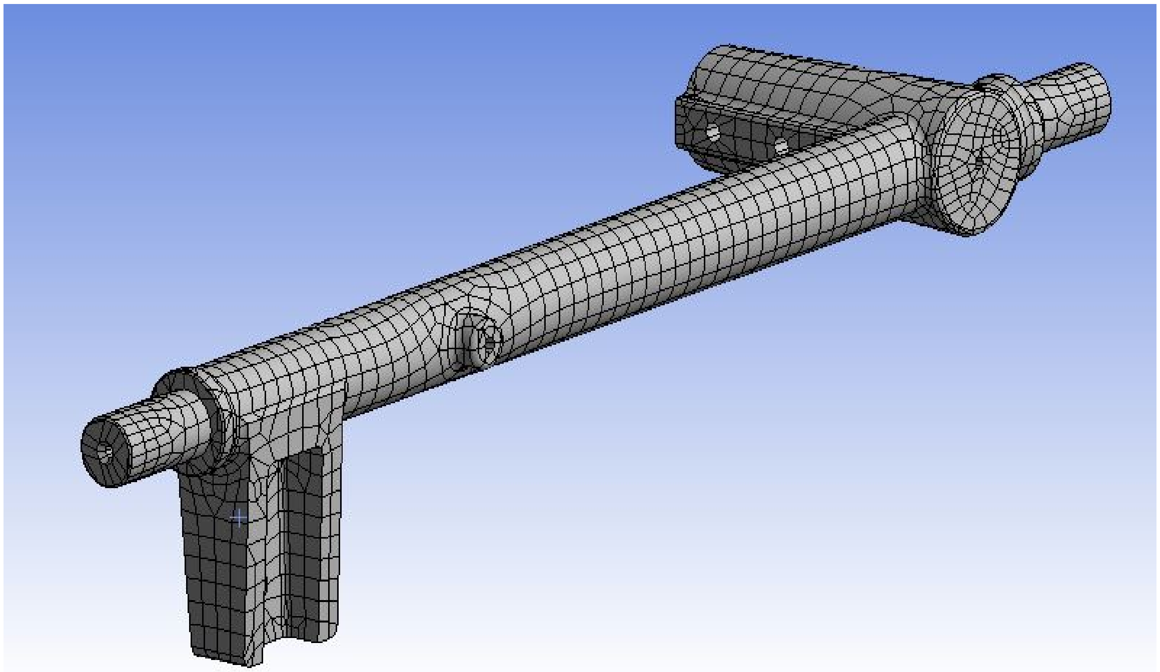


Figure 4.5: Mesh model of proposed shaft

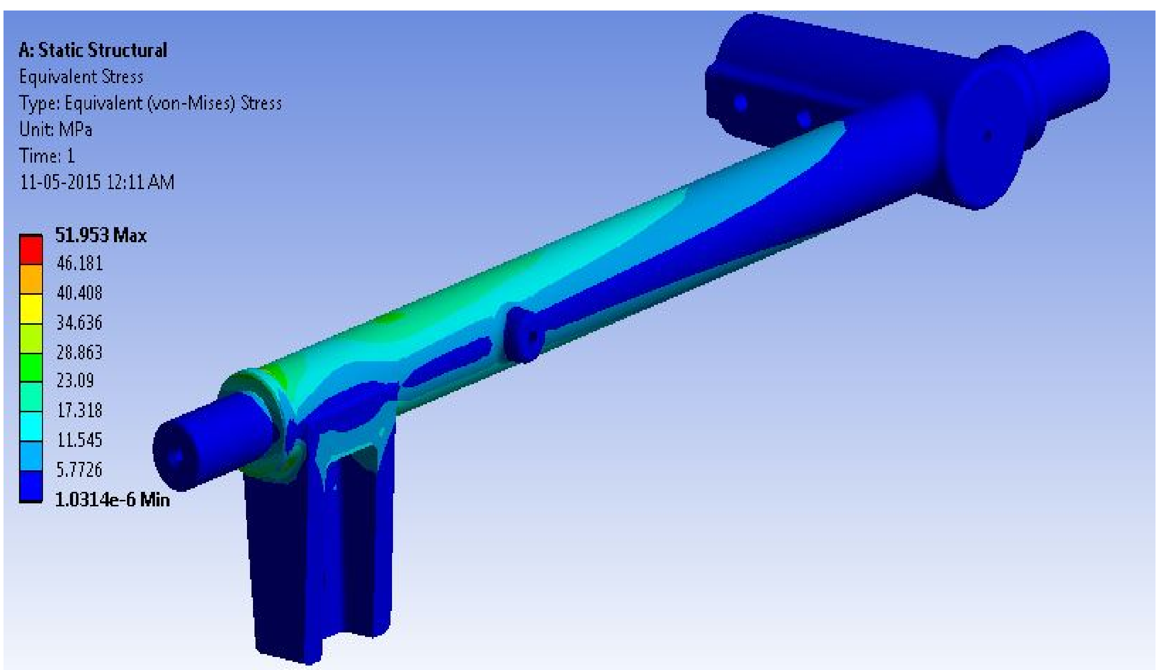


Figure 4.6: Stress distribution in proposed shaft

4.2 Bearing Selection for Tappet Shaft

Tappet shaft and sley assembly are assembled rigidly and this tappet shaft is supported by bearings. These bearings are subjected to a pure radial load. They are not subjected to any axial force. Bearing units are suitable for this arrangement so, select bearing unit by manufacturer catalogue.

Input data

Shaft rotation(n)=150 rpm

Life of bearing(L_{10h}) = 30000

Radial Force(P)= 1906 N

Diameter of Shaft(D)= 30 mm

Life in million revolution

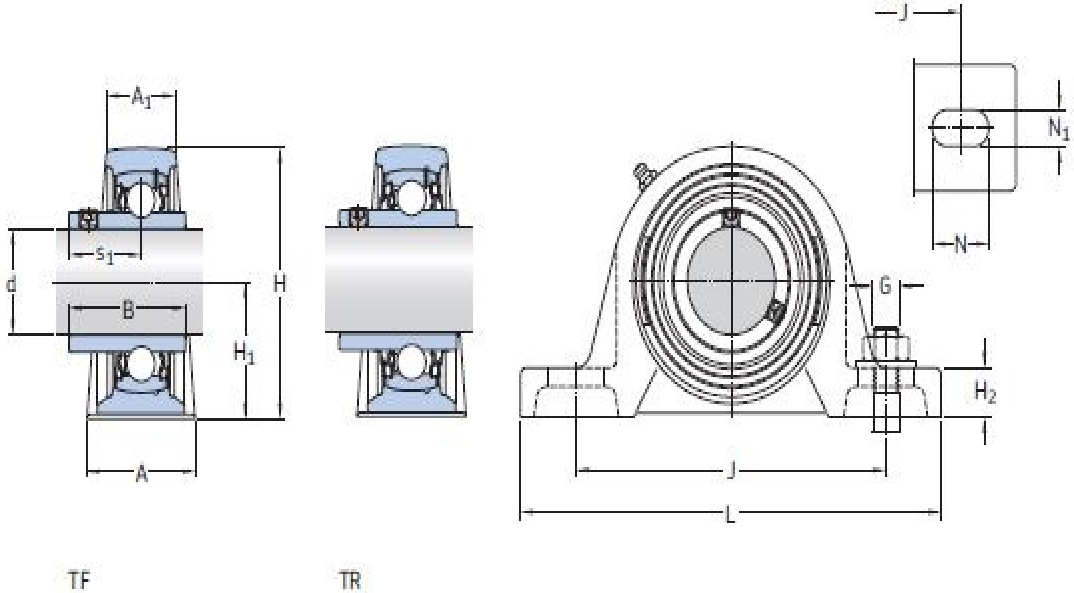
$$\begin{aligned} L_{10} &= \frac{60nL_{10h}}{10^6} \\ &= \frac{60 \times 150 \times 30000}{10^6} \\ &= 270 \ million \ rev. \end{aligned}$$

Basic Dynamic Load Rating

$$\begin{aligned} C &= P(L_{10})^{1/3} \\ &= 1906(270)^{1/3} \\ &= 12.32 \ kN \end{aligned}$$

From SKF Y-bearings and Y-bearing Units Catalogue, SY30TF bearing unit is selected. It has Basic dynamic load rating 19.5 which is greater than 12.35. Hence selection is safe.

Y-bearing plummer block units with a cast housing and grub screws, metric shafts
d **12 – 60** mm



Dimensions													Basic load ratings		Fatigue load limit	Limiting speed	Designation
d	A	A_1	B	H	H_1	H_2	J	L	N	N_1	G	s_1	dynamic C	static C_0	P_u	with shaft tolerance h6	Bearing unit
mm													kN		kN	r/min	–
12	32	18	27,4	57	30,2	14	97	127	20,5	11,5	10	15,9	9,56	4,75	0,2	9 500	**SY 12 TF**
15	32	18	27,4	57	30,2	14	97	127	20,5	11,5	10	15,9	9,56	4,75	0,2	9 500	**SY 15 TF**
17	32	18	27,4	57	30,2	14	97	127	20,5	11,5	10	15,9	9,56	4,75	0,2	9 500	**SY 17 TF**
20	32	21	31	65	33,3	14	97	127	20,5	11,5	10	18,3	12,7	6,55	0,28	8 500	**SY 20 TF**
	34	23	31	65	33,3	14	97	127	20,7	13	10	18,3	12,7	6,55	0,28	8 500	**SYJ 20 TF**
	32	21	31	65	33,3	14	97	127	20,5	11,5	10	18,3	12,7	6,55	0,28	5 000	**SY 20 TR**
25	36	22	34,1	70,5	36,5	16	102	130	19,5	11,5	10	19,8	14	7,8	0,335	7 000	**SY 25 TF**
	38	24	34,1	70,5	36,5	16	102,5	140	21,5	13	10	19,8	14	7,8	0,335	7 000	**SYJ 25 TF**
	36	22	34,1	70,5	36,5	16	102	130	19,5	11,5	10	19,8	14	7,8	0,335	4 300	**SY 25 TR**
30	40	25	38,1	82,5	42,9	17	117,5	152	23,5	14	12	22,2	19,5	11,2	0,475	6 300	**SY 30 TF**
	42	27	38,1	82,5	42,9	16	118	165	24	17	14	22,2	19,5	11,2	0,475	6 300	**SYJ 30 TF**
	40	25	38,1	82,5	42,9	17	117,5	152	23,5	14	12	22,2	19,5	11,2	0,475	3 800	**SY 30 TR**
35	45	27	42,9	93	47,6	19	126	160	21	14	12	25,4	25,5	15,3	0,655	5 300	**SY 35 TF**
	46	28	42,9	93	47,6	17	129	167	24	17	14	25,4	25,5	15,3	0,655	5 300	**SYJ 35 TF**
	45	27	42,9	93	47,6	19	126	160	21	14	12	25,4	25,5	15,3	0,655	3 200	**SY 35 TR**

Figure 4.7: Bearing Unit selection table

Y-bearing unit designation system

Examples							
	FYTBK 30 TR	FY	TB	K	30	TR	
	SY 1.1/2 TF	SY			1.1/2	TF	
	TUJ 50 TF	TUJ			50	TF	
	PFD 40	PF	D		40		

Identification of housing type

	J stands for dimensions to standard JIS 1559-1995
FY(J)	Flanged housing
P	Plummer block housing, pressed steel
PF	Flanged housing, pressed steel
SY(J)	Plummer block housing
TU(J)	Take-up housing

Identification of unit design

–	Base version; when flanged: square flange
C	Flanged unit, round flange
D	Flanged unit, triangular flange
F	Plummer block unit, short base
H	Plummer block unit, lower centre height
M	Plummer block unit, higher load carrying capacity
T	Flanged unit, oval flange
TB	Flanged unit, oval flange
TF	Flanged unit, oval flange, no relubrication facility

Identification of housing material

–	Grey cast iron
K	Composite
Z	Grey cast iron zinc coated
R	Stainless steel

Identification of size

	Bearing units for metric shafts: in millimetres uncoded
12	12 mm bore diameter
	to
100	100 mm bore diameter
	Bearing units for inch shafts: in inches uncoded
3/4	3/4 in. = 19,05 mm bore diameter
	to
2 1/2	2 1/2 in. = 63,5 mm bore diameter
	Housings from grey cast iron or composite material
503	for Y-bearings of sizes 203, 203/12 and 203/15
	to
520	for Y-bearings of size 220
	Housings from sheet steel (not supplied as bearing unit)
40	40 mm housing bore diameter
	to
90	90 mm housing bore diameter

Identification of inserted Y-bearing

FM	Y-bearing with an eccentric locking collar, YET 2 series
KF	Y-bearing with a tapered bore, YSA 2-2FK series
LF	Y-bearing with SKF ConCentra locking
RM	Y-bearing with grub screws, YAT 2 series
TF	Y-bearing with grub screws, YAR 2-2F series
TR	Y-bearing with grub screws, YAR 2-2RF series
WF	Y-bearing with an eccentric locking collar, YEL 2-2F series

Figure 4.8: Bearing Unit designation table

Chapter 5

Conclusion and Future Scope

5.1 Conclusion

To improve functionality of shuttle loom various parameters are identified for picking and checking mechanism. Both picking and checking are interrelated with each other. Here, checking mechanism is developed from scratch to reduce picking force transmitted by picker to shuttle.

Kinematic Analysis for picking mechanism and checking mechanism has been carried out to develop new checking mechanism. Synthesis for proposed mechanism has been performed to obtain suitable dimensions of linkages. By following the methods of kinematic analysis of mechanism, the position analysis of checking mechanism has been carried out and the results have been plotted for slider displacement to crank rotation. The same mechanism was replicated in cad package, creo, where the crank rotation is supplied as an input parameter and the values of slider displacement vs crank rotation are plotted. The values obtained from both graphs complement each other and hence the results are verified.

In order to get smooth and less expensive picking mechanism, checking mechanism is developed here to brake and release the swell to stop shuttle. For present mechanism picker need to generate 260 N force to accelerate shuttle. With this proposed mechanism picker need to develop 190 N force to accelerate shuttle. Which is to improve life of shuttle and picking stick, which are most commonly failed parts.

To obtain smooth picking mechanism, picking shaft has been replaced to work under torsional loading condition. For the picking shaft present and proposed models are verified by FEA Tool, Ansys, to check stress concentration under twisting.

5.2 Future Scope

Due to fatigue loading in picking Mechanism Structure of loom is under vibration. Study on vibration can be performed for structure of loom.

For Picking mechanism power transmission from cam to picking stick is done by flexible link. Transmission can be modified through rigid links for better power transmission.

Optimization of side structure wall can be performed under different conditions to reduce material.

Development of Beat-up,Shedding,Take-up and Let-off mechanism can be performed to operate at higher speed.

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