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DESIGN OF ALTERNATIVE MAIN WEAVING MECHANISMS FOR HANDMADE CARPET LOOMS

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H.İBRAHİM ÇELİK
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Design of Alternative Main Weaving Mechanisms for Handmade Carpet Looms

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Supervisor
Assist.Prof.Dr. Mehmet TOPALBEKİROĞLU

by

H. İbrahim ÇELİK

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Prof. Dr. Sadettin ÖZYAZICI

I certify that this thesis satisfies all the requirements as a thesis for the degree of Master of Science/Doctor of Philosophy.

Prof. Dr. Ali KİREÇCİ
Head of Department

This is to certify that we have read this thesis and that in our opinion it is fully adequate, in scope and quality, as a thesis for the degree of Master of Science/Doctor of Philosophy.

Assist. Prof. Dr. Mehmet TOPALBEKİROĞLU
Supervisor

Examining Committee Members

Prof. Dr. Erdem KOÇ

Prof. Dr. Ali KİREÇCİ

Prof. Dr. L. Canan DÜLGER

Assist. Prof. Dr. Nihat ÇELİK

Assist. Prof. Dr. Mehmet TOPALBEKİROĞLU

ABSTRACT

DESIGN OF ALTERNATIVE MAIN WEAVING MECHANISMS FOR HANDMADE CARPET LOOMS

ÇELİK, H. İbrahim
M.Sc. in Textile Engineering
Supervisor: Assist. Prof. Dr. Mehmet TOPALBEKİROĞLU
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The most important production steps of the handmade carpets produced still by human labour are; knotting (Turkish knot or Persian knot), shedding, picking and tightening the carpet with the beater (beat-up operation) operations. These operations tire weaver and take long time during the carpet production. The aim of this thesis is to design a suitable mechanism for beat-up operation and develop different alternative mechanisms for picking and shedding operations in the handmade carpet looms. Thus, the weaver could weave more carpet by consuming less labor. Furthermore, the weaving faults caused by beat-up and picking operations could be decreased.

In the view of this aim, the alternative mechanisms with different structures which can mechanically perform the picking, shedding and beat-up operations in the handmade carpet looms were developed. The suitable mechanisms for each operation were determined by comparing these developed mechanisms in terms of design requirements. Moreover, the dimensions of the mechanisms selected for beat-up operation were determined by using two different dimensional synthesis methods. The beat-up mechanism providing the best solution was selected by evaluating the obtained mechanisms in terms of mechanical criterias. The dynamic analysis of the selected beat-up mechanism was then performed by using a package program and the results were examined.

Key Words: Weaving, Handmade woven carpets, Beat-up mechanism, Picking mechanism, Shedding mechanism.

ÖZET

EL DOKUMA HALI TEZGAHLARI İÇİN ALTERNATİF TEMEL DOKUMA MEKANİZMALARININ TASARIMI

ÇELİK, H. İbrahim
Yüksek Lisans Tezi, Tekstil Mühendisliği
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Günümüzde halen insan gücüyle üretilmekte olan el dokuma halılarının en önemli üretim aşamaları düğüm oluşturma (Türk düğümü veya İran düğümü), ağızlık açma, atkı atma ve kirkitle halıyı sıkıştırma (tefe işlemi) işlemleridir. Bu işlemler halının üretimi esnasında dokumacıyı çok yormakta ve uzun zamanını almaktadır. Bu çalışmada, el dokuma halı tezgahlarında kullanılmak üzere tefe işlemi için uygun bir mekanizma tasarlamak, atkı atma ve ağızlık açma işlemleri için de farklı alternatif mekanizmalar geliştirmek amaçlanmıştır. Böylece, dokumacı daha az emek harcayarak daha fazla halı dokuyabilecektir. Ayrıca tefe ve atkı atma işlemlerinden kaynaklanan hatalar da azaltılabilecektir.

Bu amaç doğrultusunda atkı atma, ağızlık açma ve tefe işlemlerini el dokuma tezgahlarında mekanik olarak yapabilecek farklı yapılara sahip alternatif mekanizmalar geliştirilmiştir. Geliştirilen bu mekanizmalar tasarım gereksinimleri açısından karşılaştırılarak her bir işlem için uygun olabilecekler belirlenmiştir. Ayrıca tefe işlemi için seçilen mekanizmaların boyutları iki farklı boyutsal sentez metodu kullanarak tespit edilmiştir. Elde edilen mekanizmalar mekanik kriterler açısından değerlendirilerek en iyi çözümü sağlayan tefe mekanizması seçilmiştir. Seçilen mekanizmanın dinamik analizi bir paket program kullanarak yapılmış ve sonuçlar irdelenmiştir.

Anahtar Kelimeler: Dokuma, El dokuma halıları, Tefe mekanizması, Atkı atma mekanizması, Ağızlık açma mekanizması.

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CHAPTER 1

INTRODUCTION

1.1 INTRODUCTION

The history of the carpet, which is evaluated in the pile weavings group, is very ancient. It has been woven in different regions of Anatolia for thousands of years. It is still an important decoration product and also an important industry branch. People produced carpet on looms made of wood for a very long time. By means of the technological developments, some of the motions in weaving process were first transferred to simple mechanisms, and then the weaving processes were completely performed by machines. As the machine use for carpet manufacturing became widespread, carpet weaving on handlooms decreased. The weaving machines are then succeeded the hand looms. Now, great proportion of the carpet manufacturing is performed on weaving machines and hand weaving carpets have a little proportion on whole. Nevertheless, the handmade carpet production kept alive, since they have much superiority to machine carpets. Today, the handmade carpets are still woven at all regions of the Turkey except Black Sea region because of climate conditions. Its weaving is still an important mainstay for many regions of Anatolia [1- 3].

The carpet can be divided into two main groups according to structure and production method; the handmade carpets and the machine carpets [4-6]. The structure and the raw material of the carpets that are produced on the hand looms are different from that of the carpets produced on machines. These differences, such as knot shape, type of yarn, and design provide more advantages to the handmade carpets over the machine carpets. The main difference between them is the pile surface forming type. In the machine carpet production, the surface is formed by intersection of the pile yarn in the warp direction with weft yarns. As shown in the Figure 1.1, the pile yarns are bound into carpet structure as the shape of “u”. Because

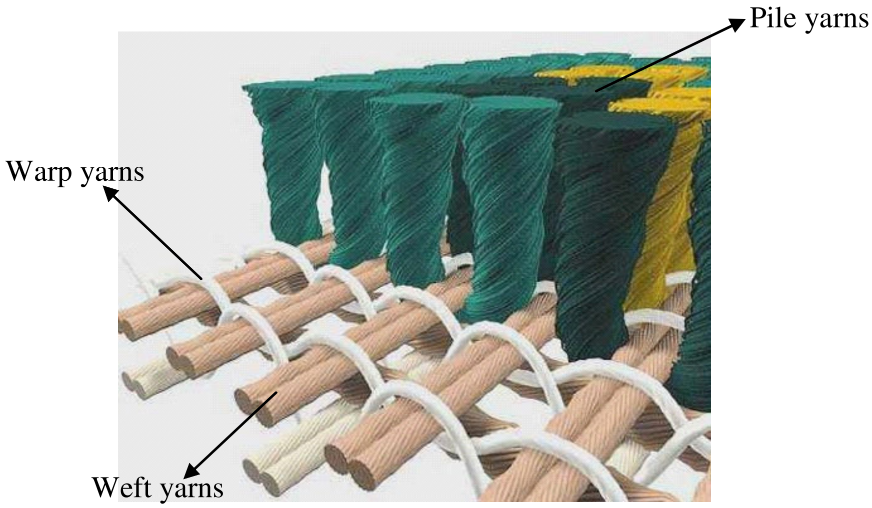
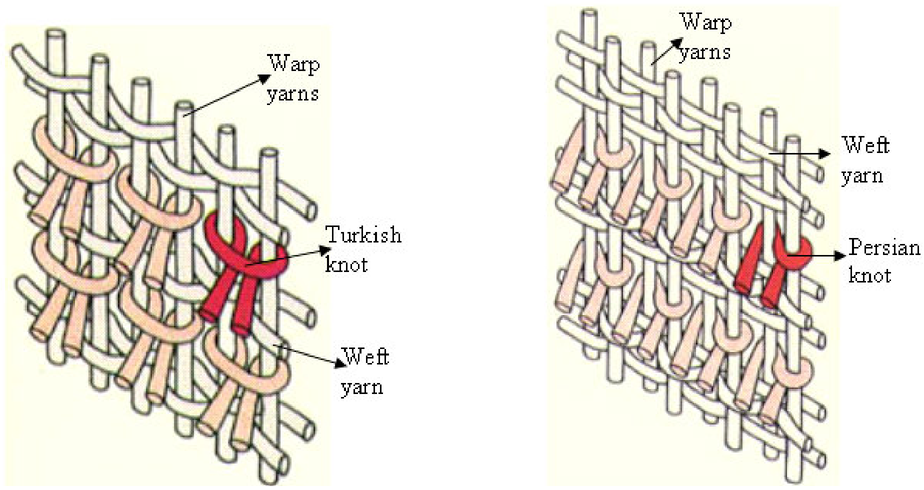


Figure 1.1 Structures used in the machine carpets

of this structure, the pile yarns can leave the carpet structure easily. So the back of the machine carpets are covered by a binding material. They have some advantages such as low cost and rapid production speed. The pile surface of the handmade carpets is formed by knotting each pile yarn on two warp yarns. The surface of the handmade carpet consists of individual knots. There are two main knot types; the Turkish knot and the Persian knot (Figure 1.2). In general, the Turkish knot is stronger than that of the Persian [6]. It is impossible to pull apart the knots from the carpet surface. These real knots make the handmade carpets stronger and more durable. Since the each pile thread knotted individually, unlimited kinds of colour can be used for handmade carpet designs. But the machine carpet designs have limited number of colours depending on the creel capacity and width of the machine. Moreover, processing natural fibers such as silk and wool on weaving machines have many difficulties and problems, but they can be woven on handmade looms. Thus the handmade carpets provide more healthy usage. It has better resilience and higher knot density per square centimeter than machine carpet. These carpets are used for prestige and decorative aims. So they are rendered as expensive products. When the quality of a handmade carpet is being judged, a consideration should be given to coloring, design, depth of pile, quality of yarn, age, condition, and fineness of texture [7]. Detailed information on handmade carpets can be found in Chapter 2.



a) The Turkish knots b) The Persian knots

Figure 1.2 Types of knot used in the handmade carpets

In order to interlace warp, pile and weft threads to produce carpet, three operations are necessary such as shedding, picking and beat-up. These operations are called primary motions. Two additional motions such as warp let-off and cloth take-up are performed for a continuous weaving. These two operations are called as secondary motions. In weaving machines all of these motions are performed by different mechanisms in a time order. In order to obtain different fabric designs and increase production speed many different mechanisms are developed for primary operations (shedding, picking and beat-up) of the weaving machine. But in the handmade carpet looms, all weaving operations are performed by weaver. Expect a simple shedding mechanism, any kind of mechanism has not been developed for handlooms. The production of the handmade carpets lasts for very long time and too much labor is consumed during its production. The most weaver-exhausting and time-consuming operations are picking the weft yarn through the warp yarns and beating-up the knots and weft yarn into the carpet structure via a comb in order to obtain required tightness. These operations have great influence on many quality factors such as; the appearance, strength, tightness, compactness and surface smoothness. During the beat-up process, the comb must beat every point at same intensity along the carpet width and the beating force must be at required level. Otherwise, the beat-up process made at higher intensity or lower intensity not only deforms the carpet structure but also decreases the quality of the carpet. In the picking operation, the tension of the weft yarn must be at an optimum value. If the weft yarn is laid into the shed tauter than the required value, the structure of the carpet is deformed. The knots can not be tightened into carpet structure thoroughly. If the weft yarn is passed through the shed loosely, the weft yarn spurts to the back of the carpet at somewhere. So the smoothness of the carpet back is deformed and the carpet loses its quality. Since the tension of the weft yarn during picking and the force exerted on the weft thread during beat-up are adjusted by weaver sensitivity, these process requires experience and attention.

The aim of this thesis ***is to design a suitable mechanism for beat up operation and develop different alternative mechanisms for picking and shedding operations in the handmade carpet looms***. Thus, the handmade carpet manufacturing rate will be increased and the faults caused by these operations will be overcome. Also, the weaver will be able to produce more carpets by consuming less labor and time by using these mechanisms. These developments will certainly provide considerable contribution in economical point of view.

1.2 LITERATURE SURVEY

A few studies were found on the development of looms and improving production of handmade carpets in the available literature. Topalbekiroğlu [7] has performed a study of design, construction and control of an electromechanical knotting system. An experimental set-up was designed and manufactured to realize the practical production of the Turkish knots by an electromechanical system. Topalbekiroğlu [8] has studied the conceptual design and dimensional synthesis for mechanisms used to produce knots of handmade carpets. Kireçci, Doğan and Topalbekiroğlu [9] have offered an alternative solution for producing Turkish knots in a handmade carpet. Kireçci, Dülger and Topalbekiroğlu [10] have then analysed the design of an electromechanical loom to produce carpets with Gördes knots utilizing the advantages of technological developments. Topalbekiroğlu, Kireçci and Dülger [11] have recently worked on a pattern study applied on handmade carpet and designed a computer controlled mechanism to prepare the pile yarn according to the color code to a knotting mechanism. Chaudhary [12] has carried out a metallic loom in order to determine the critical stresses and deflection in its components so that optimum sizes and shapes of the structural members can be selected. Chaudhary [13] has improved metallic loom was developed at IIT Delhi in 2001 and made optimisation of this loom which was carried out resulting in relatively lightweight and reduced cost. Only many documents were found on the handmade carpets. These studies are generally in the view of culture, design, history and economy [14-16].

Some studies were related to about beat-up and picking mechanisms of power looms. Dawson [17, 18] has analyzed the effect of reed motion geometrically and determined factors affecting the dimensions of a beat-up mechanism. Eren [19] have study about the beat up mechanisms of the weaving machines. Dekun Dao [20] has performed a study about dynamic analysis of beat-up process. In this study, theoretical prediction of beat-up force and warp tension response during beat-up are made by setting up a weaving model under dynamic conditions. Sihih, Mohamed, Burllerwll and Dao [21] have measured the beat-up force of a weaving machine. Katunskis [22] has made an investigation in order to determine experimental and theoretical methods of beat-up force measurement. Kumpikaite [23] made a study about relation of beat-up process parameters and fabric's structure factors. Eren [24] has made an investigation about cam design for beat-up mechanism of weaving machines by using analytical method. Mrazek J. [25] had a study of dynamic analysis of a beat-up mechanism of a loom. Vaclavik and Koloc [26] made a simulation study of some mechanism on weaving machines such as picking and beat-up mechanisms. Sternheim and Grosberg [27] have done measurements on a computer controlled hydraulic beat-up mechanism. Zhand and Mohamed [28] have study about the behavior of the warp yarn during a beta-up operation of weaving machine. Eren [29] have study of designing a four-bar linkage sley drives mechanism for a desired sley motion curve. In this study, kinematic design equations are given and the link lengths of the mechanism are determined. Adanur [30] has built an air-jet weaving simulator and determined the characteristic of yarn in air-jet. Turel [31] have investigated the air flow and effect of several parameters on weft insertion, and determined air-yarn interactions in air-jet weaving machines. Vangheluwe [32] had a study providing an analysis of weft insertion elements, including a comparison between water-jet and air-jet weaving. There are many other studies dealing with the machine carpet production and weaving machines [4, 33-38].

Any mechanical systems to perform the beat-up and picking operations of handmade carpet have not been encountered in the literature.

1.3 LAYOUT OF THESIS

Chapter 2 includes the detailed information about the handmade carpets. The knot types used in the handmade carpet production are presented. The handmade carpet looms and the parts of the carpet are explained. The weaving steps of the handmade carpet are also explained by figures.

Chapter 3 contains two sections. In the first section, the general information about the shedding process of the handmade carpet weaving and the shedding mechanisms used in the weaving machines are presented. Some design requirements on shedding mechanism used in the handmade carpet loom are determined. Five alternative shedding mechanism models are developed and evaluated. In the second section, alternative models are generated for picking operation of handmade carpet weaving. Firstly, picking operation of the weaving machines and the handmade carpet weaving are explained. Then, the design requirements of the picking mechanism used in the handmade carpet looms are determined. Finally, four different alternative solutions are generated and evaluated.

The design of beat-up mechanism for handmade carpet looms is presented in Chapter 4. Firstly, the function of the beat-up process in the handmade carpet weaving and effect of this process to the quality of the handmade carpet are explained. Then, the functional requirements and the design specifications of the beat-up mechanism are determined. Eight alternative models are then generated. All alternative models are evaluated and compared according to functional requirements and design specifications. Two of the four-link mechanisms are selected for the dimensional synthesis. The dimensional syntheses of these mechanisms are made by using graph-analytical and analytical methods. The results are evaluated depending upon some mechanism design criterias such as transmission angle, Grashof’s condition and optimum crank/coupler link ratio. Then, the best solution satisfying design criterias is selected for each beat-up mechanism. The optimum dimensions of beat-up mechanism are resolved by using the geometrical method. The same results are obtained with the analytical solution.

Dynamic analysis of the selected beat-up mechanism is presented in Chapter 5. The force on the beater and the loads on the joints are determined and analyzed by using a mechanism simulation program. The joint forces are also determined to check there is overloading on the joints or not. A beater model is designed and developed for the beat-up mechanism. A prototype model of the mechanism is presented.

Finally, the conclusions on thesis and recommendation for further studies are included in Chapter 6.

CHAPTER 2

HANDMADE CARPETS

2.1 INTRODUCTION

Handmade carpets are textile floor covering that are woven by forming knot rows of wool, silk or rayon yarns on each pair of warp yarns. They are composed of abreast, parallel and vertical arranged yarns and picking at least one row of weft yarn, beating up of this row and forming knot rows [39]. The weaving of the pile carpet is a difficult and tedious process which may take from a few months to several years depending on the quality and size of the carpet. The main technical structures of the carpet can be explained as:

- The handmade carpets are woven on the fixed village loom, the Tabriz or Bunyan loom, and the roller beam loom which are made of wood or metal.
- The knot can be formed into two ways; the Turkish (close) knots and the Persian (open) knots.
- The height of the pile yarns are in the rage of 3.5-4.5 mm. The height of the pile yarns are adjusted by adjustable scissors.
- The completely woven carpets are laundered, dried and after the cutting process, it is subjected to quality control and finally presented to the market.
- In order to obtain a high quality carpet, high quality and suitable yarn must be selected. The yarns must be dyed by natural dyes.
- The number knots, evenness of the knots, pattern and congruity of the colors affect quality of the carpet substantially [40].

In order to perform the weaving process, the weaver needs a number of essential tools; a knife, comb (beater) and a scissors shown in Figure 2.1. These tools are simple and have been used without any change for a very long time [41, 42]. The knife shown in Figure2.1 (a) is used for cutting the yarn after each knot is tied. In some areas, a hook is used instead of a knife to perform this function. The comb or

(a) Knife (b) Comb or Beater (c) Scissors

Figure2.1 Carpet weaver's tools

beater is a instrument made of iron or wood that is used for beating the weft yarn and knots and securing them in carpet structure (Figure 2.1 (b)). Its size is various in different places. It has a mass of about 2 kg. Comb has various kinds of thickness that varies according to the weaver and carpet specifications, such as the wales number chosen for carpet or kilim. The beater has a handle in order to keep and move it easily. Since the distance between the warp threads is different dependent upon the quality of the carpet, the closeness of the beater teeth changes according to the carpet quality. The beater with thin teeth is used for fine quality carpets and the beater with thick teeth is used for coarse quality carpets. A special scissors shown in Figure2.1 (c) is used for cutting long and uneven pile after one or few rows of knots have been woven. The scissors has handles bent so that the blades can cut flush with the face of the rug [41-43].

Carpet weaving which reflect on the custom religions and aspirations of more than one civilization is a very ancient tradition of many cultures [4]. The carpet is an important decorative object and its market is an important branch of industry. The machine carpets are used for both decorative and floor covering but the handmade carpets are used for ornamental and decorative aims. The handmade carpets have three primary functions; religious, artistic and home furnishing. Although the history of the hand made carpet production is very old, the production techniques and its technology has not changed. The main reason is that; it is difficult to form the knot structure mechanically. Today the handmade carpets are still manufactured completely by human labor. Hand knotting gives the carpet its essential quality and its uniqueness. The fineness of the weave and the weaving qualities are determined by the number of knots. In the major production centers the number of knots could often reach 1,000 or more knots per square inch [44, 45].

In the following sections; the types of knots, the weaving looms, the structure of carpet, the carpet weaving processes are presented.

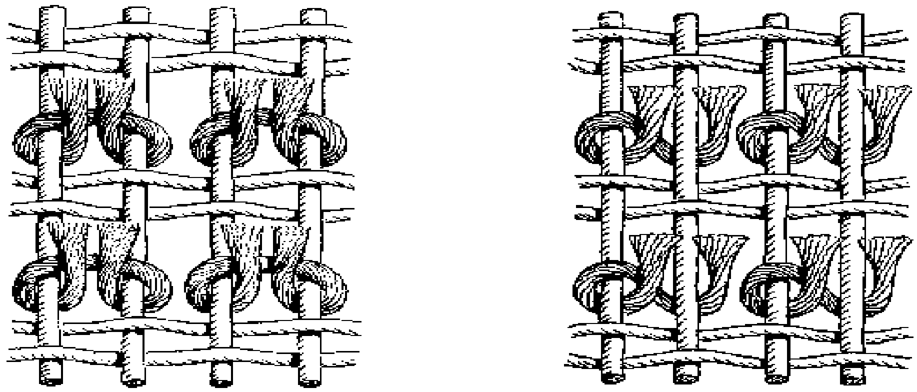
2.2 TYPES OF CARPET KNOTS

The most important feature of the handmade carpet is knots tied on the warp yarn pairs. Usually a handmade carpet quality is judged by the knots per square inch. The more the density of the knots is, the better the quality of the carpet is. The number of knots depends on the design of the carpet. The higher the knot density, the more detailed the design can be made. The carpets are classified (see Table 2.1) regard to the number of knots per square decimeters as the following [39].

There are many ways of knotting the pile yarn around the warp. The two most common types of knots used in a handmade carpet are the Turkish knot (Figure 2.2 (a)), and the Persian knot (Figure 2.2 (b)). Both are double warp knots, tied around adjacent warp ends. All the carpets made of these knotting systems have longer life. The Turkish knot is generally stronger than the Persian knot. Since the Turkish knot is wrapped around two warps and the Persian knot is wrapped around a single warp, the carpet with the Turkish knot has higher resilience. The Turkish knots (Figure 2.2(a)) symmetrically match around of pair a line of the placed threads of a basis; both threads of the basis are completely twisted with a pile yarn. Here, the pile thread forms a loop around two warps. Both ends of the pile thread come out between both warps. The Persian knots, as shown in Figure2.2 (b), matches asymmetrically around of a pair placed adjacent with each other threads of a basis; the pile yarn twists one thread of a basis entirely, and the second only half [7, 46-50].

Table 2.1 Carpet classification (number of knot per dm^2)

Carpet Classes	Number of knots per dm^2
Extra extra fine	$10000\text{-}2401/dm^2$
Extra fine	$2400\text{-}1851/dm^2$
Fine	$1850\text{-}1401/dm^2$
Medium	$1400\text{-}701/dm^2$
Coarse	$700\text{-}215/dm^2$



(a) Turkish knot (b) Persian knot

Figure 2.2 Turkish and Persian knots

There are different knotting systems such as Jufti and Tibetan knots. However they have not a wide usage [50-52].

2.3 WEAVING LOOM

Loom is a wooden or metal rectangular frame which holds the carpet together while it is being woven. There are two types of handmade carpet weaving looms. These are the horizontal loom, the vertical loom. The vertical loom is the most widely used one. The city and village weavers use the vertical loom. The assembly of the vertical loom is more complicated than the horizontal looms. A vertical loom seen in Figure 2.3 consists of four bars, two side bars that go from ground up and two horizontal bars.

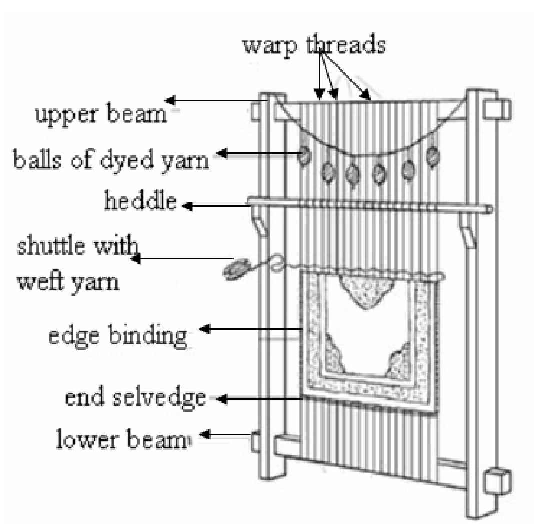


Figure 2.3 Vertical loom

One of the horizontal bars is at the bottom, close to the ground, and the other one is at the top. The warp threads are secured between these top and bottom bars. The top bar is called as ‘upper beam’ and the bottom beam is called as ‘lower beam’. The other parts of the loom are balls of pile yarn, heddle and shuttle with weft string. The heddle is used to separate alternate warps so that the shuttle carrying the weft thread can be passed between warps from one side of the rug to the other. More sophisticated vertical looms have their upper and lower beams constructed as rollers [7, 41, 42, 53, 54].

2.4 THE STRUCTURE OF THE CARPET

Carpets are made of essentially three types of threads; the warp, the weft and the pile. The carpet structure is formed by vertical and horizontal yarns known as “warps” and “wefts” shown in Figure 2.4 respectively. These yarns form the skeleton of the carpet. The pile of the carpet is made by knotting selected color of yarn according to the design. The pile is formed in several rows along the wide of carpet. The pile yarn is wrapped around a pair of warps and then it is cut to a uniform height after weaving. Edge bindings are made by wrapping several warps at the edge of the rug with yarn to reinforce this part of the rug. End finishes hold knots and wefts from working off the carpet’s warp thread. Fringes are formed by gathering and knotting together bundles of warp threads at both ends of the carpet after the carpet has been cut from the loom. The knots in these bundles of warp thread keep pile knots and end finishes tight at the carpet’s ends. The physical properties of the carpet depend on the

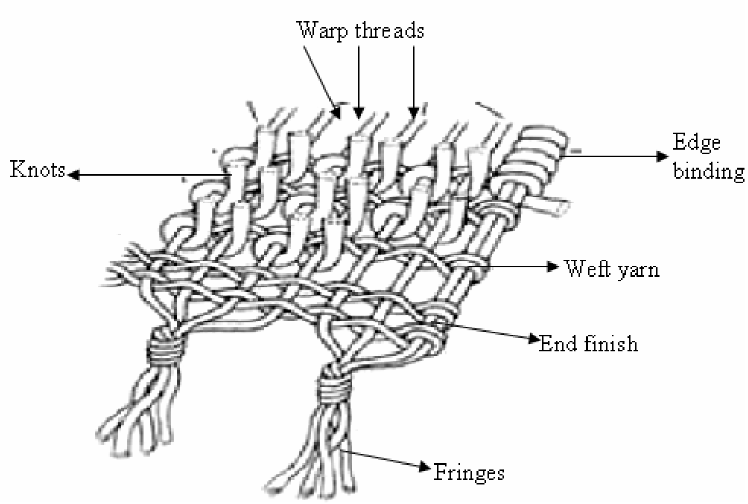


Figure2.4 Structure of the handmade carpet

quality and kind of the yarn that get into construction. Most weavers use cotton warp thread if it is available because it is easier to weave a flat, straight carpet on cotton warps than on wool warps. Wefts and piles are generally made of cotton, wool, or silk. Wefts hold rows of knots in place and strengthen the structure of the rug [7, 42, 54, 55].

2.5 CARPET WEAVING

During centuries the technology, of weaving of carpets has not changed. The weaving process is performed in several steps. In this section, the weaving operation will be presented step by step [56-58].

i. Warp threads are vertically wound around the loom parallel to each other, depending on the type and size of the carpet.
ii. After preparing the warp, a chain like plait called "chiti" is woven, leaving a margin for fringes, and then a 2-4 cm wide kilim weaving is done so as to prevent the pile knots from shifting and dropping out. Upon completion of this procedure, the carpet is ready for weaving.
iii. A weaver uses a finger to push the yarn through the warps, then uses the hook on the knife to catch the yarn behind the warps and pull it to the face of the rug (Figure2.13a-b)
iv. After the knot is tied, the weaver cuts the yarn with a flick of the blade (Figure2.13c).
v. After a row of knots is completed, one or several weft yarns are passed through the warp yarns (Figure2.13d).
vi. Then a special comb is beaten down the knots and weft yarn to obtain the desired tightness in carpet structure (Figure2.13e).
vii. The weaver cuts the surplus colored threads with a pair of adjustable scissors. A uniform level pile thickness is obtained (Figure2.13f).

(a) (b)

(c) (d)

(e) (f)

Figure2.5 Weaving process steps

CHAPTER 3

DEVELOPING SHEDDING and PICKING MECHANISMS

3.1 INTRODUCTION

Weaving is the process of making cloth, rugs, blankets, and other products by interlacing two sets of threads over and under each other. The thread sets which consist of the fabric are warp and weft (filling) yarns. During the weaving operation, the warp yarns are separated in groups and moved up and down to form an opening. This opening is called ‘shed’. After each shed change, a weft yarn is inserted through the shed. In all forms of the machine weaving, both the control of the warp yarns to form a shed and the insertion of the weft thread through this shed operations are carried out by different types of mechanisms. However, these operations are performed manually by weaver in the handmade carpet production.

In this chapter, different alternative mechanism models are presented for picking and shedding operations of the handmade carpet looms and these mechanisms are discussed in terms of design requirements.

3.2 DEVELOPING SHEDDING MECHANISMS

The shedding mechanism moves the wrap yarns up and down according to the required pattern and makes an angled opening for the filling yarn. Every weaving machine provides a control device for the warp yarns. There are four systems; crank shedding, cam shedding, dobby shedding, and jacquard shedding to provide manipulation of warp yarns. The crank, cam and dobby shedding systems control the heald frames; jacquard shedding system provides the control of individual warp yarns. The crank and cam systems are simple and inexpensive. The crank system is used for plain and its derivatives; the cam system has more alternatives than the crank system. The cam system can weave patterns with up to 14 different heald frames. The dobby system is more complicated than the cam system. It can weave patterns with up to 30 heald frames. It is also more expensive than the cam shedding system. The jacquard system provides control of each warp yarns individually. So it can weave very complex patterns. It is the most expensive and the slowest shedding system [59-61].

In the handmade carpet looms, the warp yarns are arranged vertically in contrast to horizontal arrangement of weaving machines. So, the shedding mechanism used for weaving machines may not be suitable for the handmade carpet looms. The shedding operation of the handmade carpet looms is performed by two bars. One of them is stationary and the other one is moved up and down. The stationary one is called as 'the shedding bar' and the moving bar is called as 'the heddle bar' (Figure 3.1).

The warp yarns are separated into two groups by knotting one warp thread on the shedding bar and leaving the adjacent one free. The heddle bar is passed through the warp yarns by passing over knotted thread and under adjacent free one. When the heddle bar is up, one shed is already opened. Then the heddle bar is moved down by the weaver and the next shed is formed. The preparation of this shedding system is

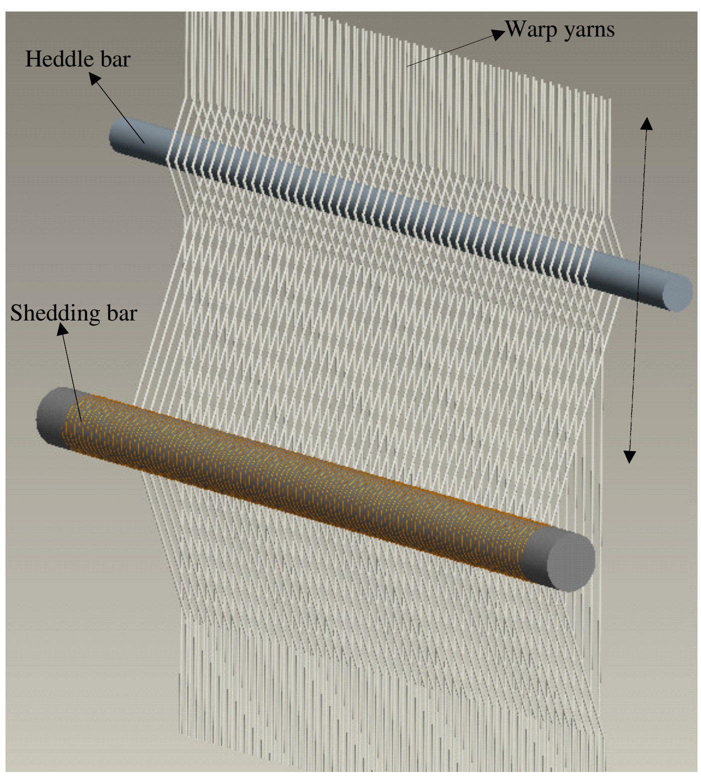


Figure 3.1 Shedding system of the handmade carpet looms

very difficult and it takes approximately one hour. Also, the system must be formed by an experienced person in order to provide a uniform warp tension along the loom and proper arrangement of the warp threads. If the warp yarns are not arranged with a uniform tension, the weft yarn and the knots can not be inserted into carpet at required density and the structure of the carpet is deformed. If the arrangement of the warp yarns are not made as required, a clear shed can not be formed and the weft yarn can not be inserted. So the shedding operation has great influence on performance of the picking operation.

3.2.1 Design Requirements

The ground structure of the handmade carpet is plain. In plain weave structure, each weft thread crosses the warp threads by going over one, then under the next, and so on. Two alternative sheds are formed for plain configuration. Hence, in the handmade carpet shedding mechanism, two heald frames are enough for shed formation. Figure 3.2 shows the position of the shedding mechanism, beat-up mechanism and the trajectory of the wraps with dashed lines. The warp yarns are threaded through the heddle eyes by passing the adjacent ones through different heald frames. Thus the warp threads are separated into two groups. The trajectories of the heald frames are shown by dashed lines. The heald frames run in sequence, when the heald frame-1 moves one groups of warp yarns backward, the heald frame-2 moves the other warp yarn group forward or vice versa.

The mechanism generated for shedding operation of the handmade carpet looms must satisfy the following design requirements:

i. The shedding mechanism must move the warp yarns into two separate groups and form a clear shed.

ii. The heald frames of the shedding mechanism must work in a sequence. When one of the heald frames moves forward, another one moves backward.

iii. The mechanism must form optimum shed geometry to allow weft insertion and must not cause high tension on the warp yarns.

iv. Shedding mechanism must be simple and has a compact structure. The mechanism must not need large space for working.

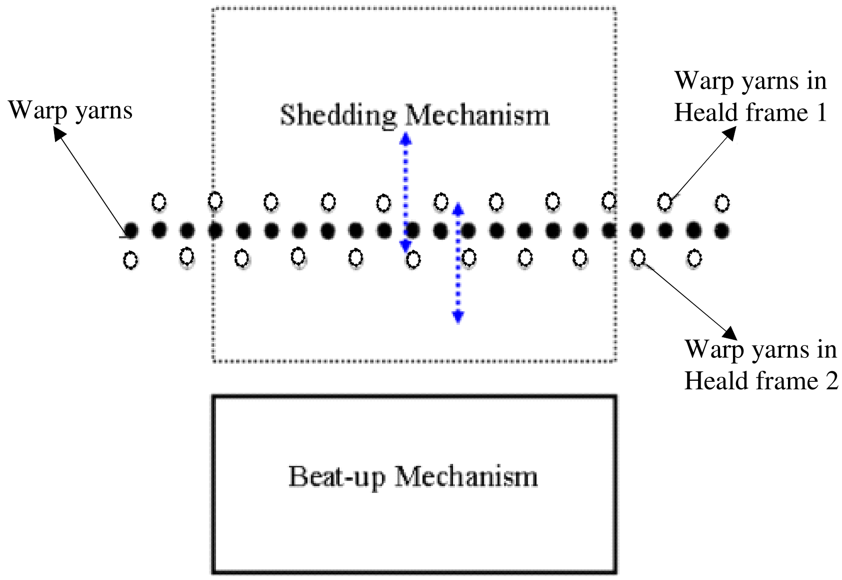


Figure 3.2 A representative figure of the shedding mechanism

v. The mechanism must have simple control and must be manufactured easily.

3.2.2 Generating Alternative Models for the Shedding Mechanism

Five different alternative solutions are generated for the shedding mechanism. The first model is an eight-link mechanism with two DOF (Degree of freedom). The second model is an inverted slider crank mechanism with two DOF. The third model is a geared six link mechanism with one DOF. In the fourth model, the heald frames are controlled by two pistons. The fifth model is a geared four link mechanism with one DOF. In the following section, the working principles of these five mechanisms are explained, and these mechanisms are discussed and compared in term of design requirements.

a. First alternative model

As shown in Figure 3.3, there are two heald frames for the shed formation. The warp yarns are threaded through the heald frames according to plain weaving motion. Thus the warp yarns are separated into two groups. The motions of the heald frames are carried out by a mechanism consist of five- link and two circular discs. The heald frames are connected to the tip of output links (CD, FG) with rigid connections.

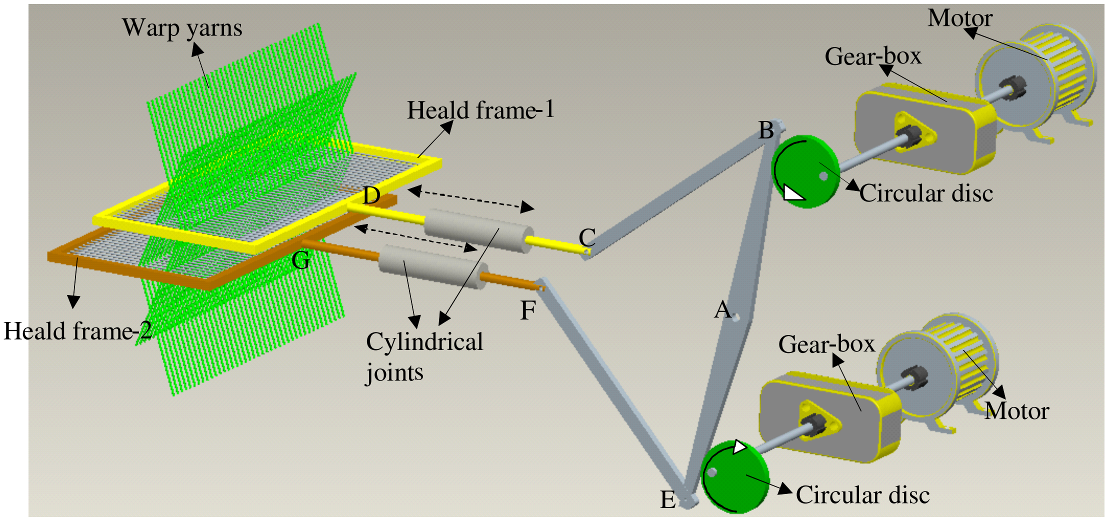


Figure 3.3 First alternative for shedding mechanism

The links consisting of the mechanism are joined to each other with the revolute joints. The output links CD and FG are passed through cylindrical connections. The mechanism is driven by two motors. The circular discs rotated by the motors force the link BE forward and backward in a sequence. The link BE is pivoted at the midpoint A with a pin connection. So, the link BE makes an oscillation motion at pivoted point (A). This oscillation motion is converted linear motions of output links (CD and FG) via the links BC and EF. The motors have reverse rotation sides. So, the heald frames move to opposite directions and two alternative sheds are formed.

This mechanism model satisfies the design requirements except the design requirements (iv) and (v). Here, the lengths of the links and diameter of the circular discs determine the amount of shed. Dimensions of the mechanism must be analyzed and determined according to the required shed width.

b. Second alternative model

This alternative model shown in Figure 3.4 is an inverted slider crank mechanism. It satisfies the design requirements (i), (ii), and (iii). It consists of two identical heald frames, two identical circular discs, and four links (AC, CD, HF, FE). Each heald frame has two identical rollers both sides. Heald frames are placed on roller ways parallel to each other. Rollers provide an easy motion to heald frames by rotating in the roller ways. The link CD is a coupler link between heald frame 2 and the link AC

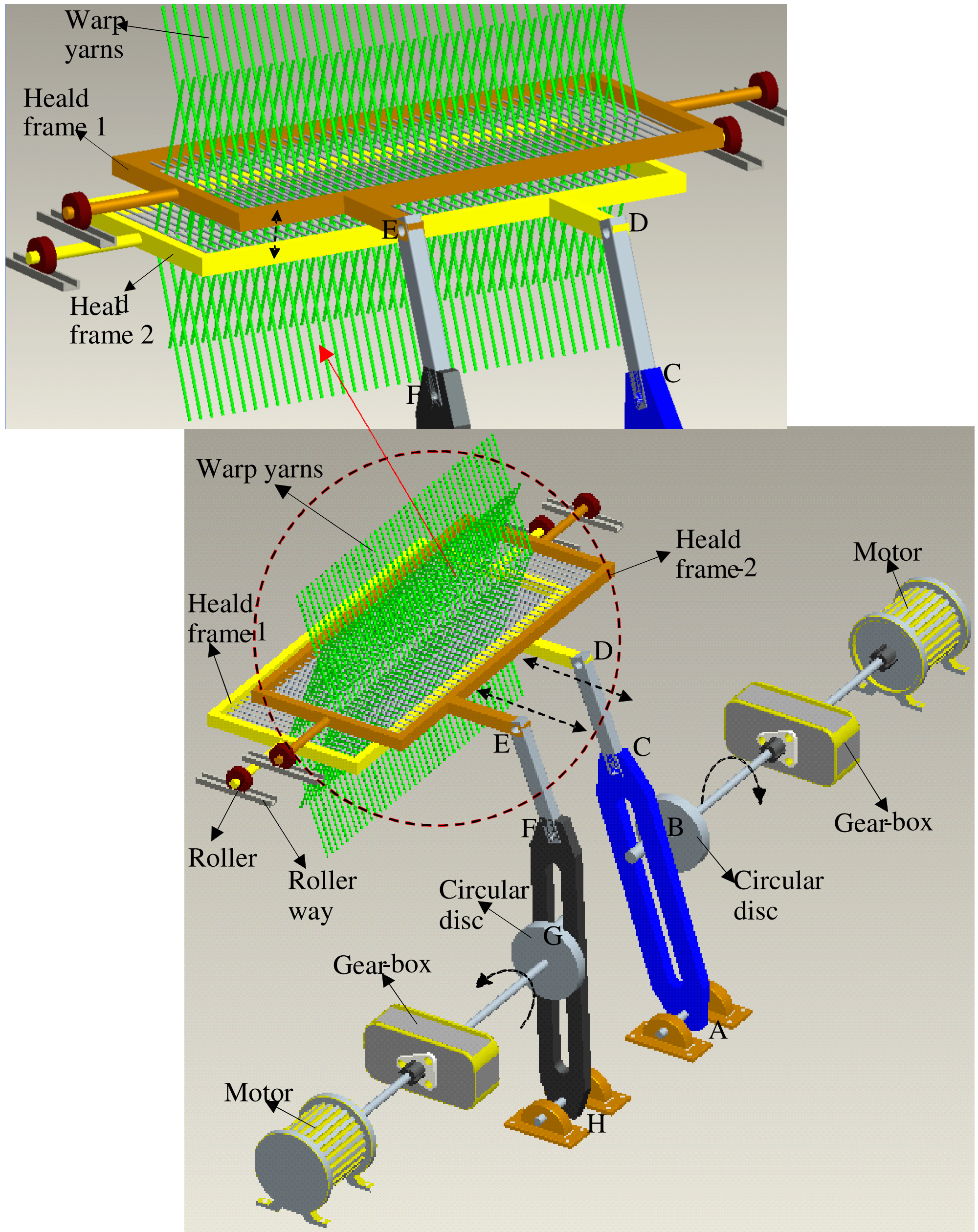


Figure 3.4 Second alternative for shedding mechanism

The link FE is also a coupler link between heald frame 1 and link HF. The circular discs are joined to links AC and HF with slotted cylinder connections at points B and G respectively. The mechanism is driven by two motors giving rotation motion to circular discs at same speed and to reverse sides. As the circular discs are rotating, they force the AC and HF links to move forward and backward via the slotted cylinder connections. The oscillation motion of the links AC and HF are converted to linear motion of heald frames through the coupler links CD and FE. Since the circular discs have reverse rotation directions, the heald frames move opposite to each other. Thus two alternative sheds are formed.

Although some of the members are identical (circular discs, the links AC and HF), the coupler links are different in length because of the position of the heald frames. The width of the shed is determined by the oscillation stroke of AC and HF links. The stroke of these links is determined by distance of the wedges on the points B and C from center of circular discs. In order to form a clear shed and not to cause high tension on warp yarns, the stroke of the links AC and HF must be well adjusted and analyzed. And then, the distance of the wedges at points B and G from center of the circular discs must be determined according to the required stroke.

d. Third alternative model

Third alternative model illustrated in Figure 3.5 consists of two identical heald frames and three spur gears. The heald frames are placed one above other as parallel. They have two identical rollers on both side and the rollers are placed on roller ways. Thus the heald frames moves easily on their axis. A link is fixed to each heald frame. Each link has teeth along its longitudinal axis to be driven by gears. The warp yarns are passed through the heddle eyes of heald frames in the form of plain fabric configuration. Thus the warp yarns are separated into two groups to form two alternative sheds. The mechanism is driven by a motor form the center gear. The motor has bi-directional control. It rotates the center gear clockwise and counterclockwise with the required angle. The gear 2 at the center drives the other gears 1 and 3. When the gear 2 is driven by the motor, the motion is transmitted to the heald frames via gear 1 and gear 3. As one heald frame moves forward, the other moves backward vice versa. Thus two alternative shed are formed. The distance between the heald frames must be at an optimum value, so the diameter of the gears must be determined according to required distance. The rotation angle of the motor must have an optimum value to form a clear shed without causing over tension on the warp yarns. The teeth on the links and gears must be meshed properly to transmit the motion uniformly. The mechanism satisfies the design requirements except the design requirement (v). However, the manufacturing of the mechanism may be difficult because of teethed links and properly meshed gears.

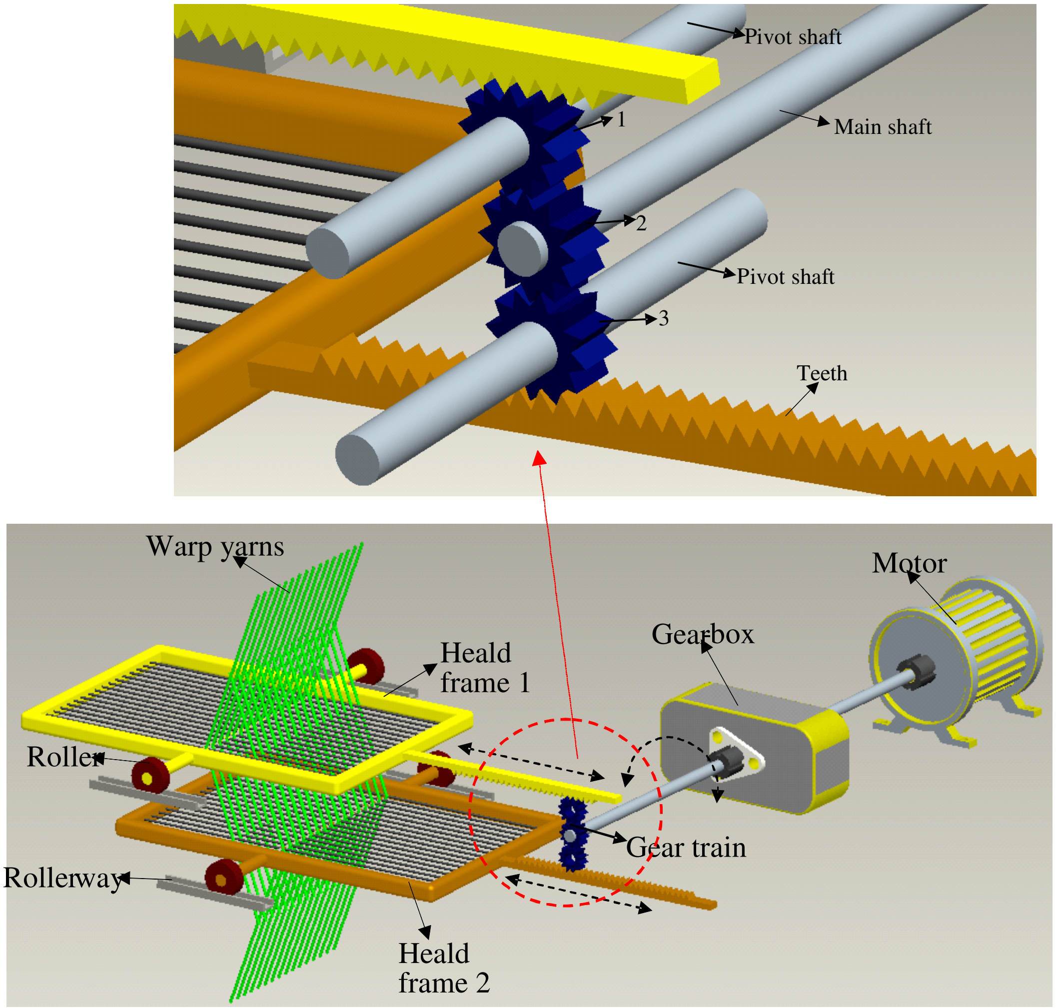


Figure 3.5 Third alternative for shedding mechanism

e. Fourth alternative model

This alternative consists of two identical heald frames and two identical pistons. As shown in Figure 3.6, the warp yarns are threaded through heddle eyes of the heald frames according to plain weaving configuration. The heald frames have rollers on both sides as the second and the third alternatives. Two pistons actuate the heald frames. The pistons work in a sequence. When one piston pushes the one heald frame, the other piston pulls the other one. Thus two alternative sheds are formed. The pistons are placed at a suitable position. The stroke of the pistons determines the shed geometry. So, suitable pistons with the required stroke must be used for the optimum shed width. The mechanism satisfies the design requirements of the shedding mechanism.

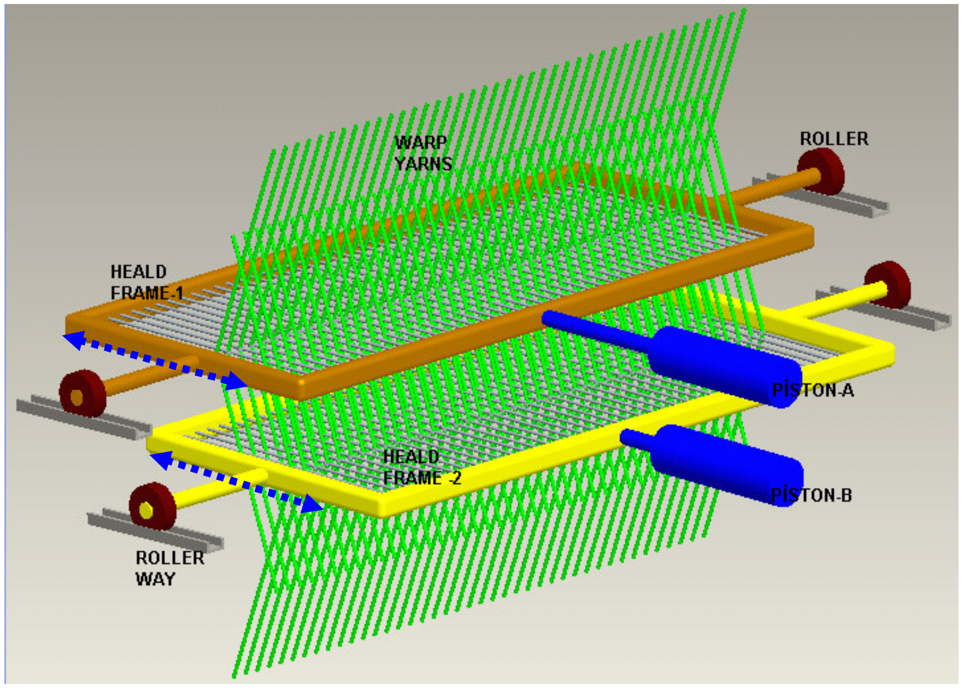


Figure 3.6 Fourth alternative for shedding mechanism

f. Fifth alternative model

As shown in Figure 3.7, the mechanism consists of two bars and two pistons. The warp yarns and shedding bars are arranged as in Section 3.2. This shedding operation arrangement is now used by handmade carpet weavers. The weaver forms the alternative sheds by moving the heddle bar up and down manually. During the motion of heddle bar, the warp yarns change place in groups and required sheds are formed. In this mechanism model, the heddle bar is acted by two pistons. The pistons are placed on two sides of the heddle bar. The tip of the pistons are connected to the heddle bar, thus they can push it down and pull up. The pistons are actuated together. The stroke of the pistons must be proper to form the required shed width, it must not cause over tension on the warp yarns. The material of the heddle bar must be suitable to move easily between warp yarns. If the friction between heddle bar and warp yarns is high, the warp yarns are damaged and more energy is required to move the heddle bar. The mechanism satisfies the design requirements, but the arrangement of the warp yarns for this model takes long time and necessitates the experienced weaver.

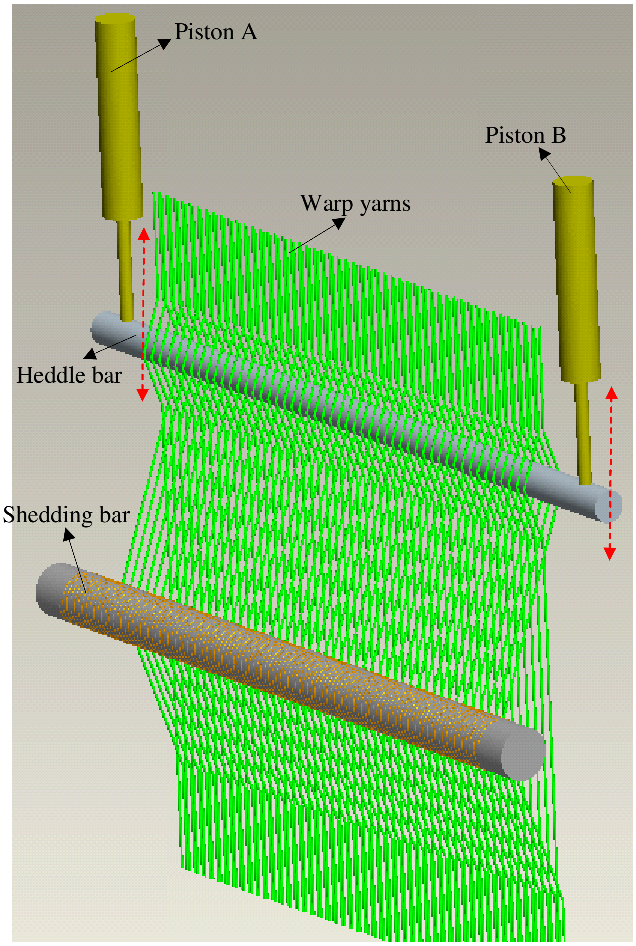


Figure 3.7 Fifth alternative for shedding mechanism

Evaluation of Alternative Models

Five alternative models are presented for shedding operation of the handmade carpet weaving. All of these models can achieve shedding operation on the handmade carpet loom. The alternative models except the fifth model use two heald frames to control warp yarns.

In the first model, the motions of the heald frames are performed by a link mechanism. The position of the circular discs must be properly adjusted to obtain uniform oscillation of the mechanism. The length of the links in the mechanism and the diameter of the circular discs must be analyzed with regard to required shed width. This model requires two motor drivers. So, the control of the model may be difficult. This alternative model requires much space operate to and construction of the mechanism may be difficult. The friction force in the cylindrical joint may be a disadvantage for this model.

The working principle and the control of the second model are same as the first alternative model, but both design and the structure of the links are different. In the second alternative model, the manufacturing of the links which are joined to circular discs with slotted cylinder connections may be difficult. The friction force in the slotted cylinder connection may be a disadvantage for the second model also.

The third alternative provides the design requirements. This model needs less space than the first and second models. The mechanism is driven by a bi-directional motor. The control of the third alternative model is simple and it has compact structure. Design and manufacturing of the gears and teeth on the heald frame links may be difficult. The gear train and teeth of the heald frame links must be properly meshed, so the construction of the mechanism necessitates attention.

The fourth and fifth alternative mechanisms have the same working principle, but the fifth model uses two bars to control the warp yarns instead of heald frames. Both of the mechanisms have simple control and compact structure. These mechanisms don't need large space for the shedding operation. The construction of the fourth model may easier than the fifth model. Although the heald frames used in the fourth model increase the manufacturing cost, they make the arrangement of the warp yarn easier during the weaving preparation and provide more uniform warp control. The friction between heddle bar and warp yarns may causes a disadvantage for the fifth alternative model.

3.3 DEVELOPING PICKING MECHANISMS

The picking system is the most important factor that influences not only the fabric production speed but also quality of the fabric. So, it is the most studied subject of weaving machines. Many picking systems have been designed for weaving machines and many developments have been made to increase the picking speed and quality. The weaving machines are usually classified according to the filling insertion mechanism type. There are four types of picking mechanism widely used for weaving machines. These mechanisms are the rapier system, the air-jet system, the projectile system and the shuttle system. The shuttle picking systems are firstly used for fabric production. By the technological developments, some other more sophisticated systems are developed such as the rapier, the projectile, the air-jet and the water-jet. These developments performed on new picking systems improved the production speed and lessened the weaving faults of a weaving machine. Now the shuttle picking system is rarely used, the air-jet and rapier picking systems are the most widely used ones [59-64].

Although the picking process has also great influence on the production of the handmade carpets, it is still performed by weaver manually and any mechanism is not used for this process. It takes much time and it is one of the most laboring processes of the carpet weaving. Because the weft insertion affects structure and quality of the handmade carpets, it requires experience and attention. The main functions of the weft yarn in the handmade carpet structure are;

- to give strength to the carpet,
- to form the carpet base structure
- to hold the knots in the carpet structure

The knots are not directly inserted into the carpet by beating. The weft yarn is inserted above the knots and they are beaten together in order to distribute beating force equally and achieve a uniform density in the carpet structure.

Before beginning of handmade carpet weaving, the proper weft yarn must be selected in order to produce the required carpet size. If the selected picking yarn is thinner than the determined size, the carpet will be shorter and poky. Similarly, if the selected picking yarn is thicker than the determined yarn size, the carpet will be rough and longer than required size. The weft yarn used in the handmade carpets is generally 2-3 Ne. During picking process of the handloom carpets, the weft yarn is passed through the shed by manually. When one filling is completed, the shed is changed and the same weft skein is passed through the new shed for the next picking. The weft yarn is not severed from the carpet at the end of the picking operation. Thus the selvage is formed on both side of the carpet.

The most important point of the picking process in the handmade carpet is the tension of the weft yarn laid in the shed. Just as the picking yarn is being inserted through the shed, the tension of the weft yarn must be adjusted at an optimum value. If the weft yarn is inserted into the shed loosely, it causes unevenness at the back of the carpet and breakage of threads during the beat-up operation. If the weft yarn is inserted into the shed at a higher tension than required, this will cause high tension on warp yarns and the knots cannot be inserted into carpet structure tightly. The tension of weft yarn during the picking process is judged by the weaver. So, in order to decrease the faults and obtain a high quality carpet, this process must be performed by an experienced weaver by attention.

In this section, some mechanisms which can perform the picking process for the handmade carpet looms will be generated. By using these mechanisms, it is aimed that the weaver produce more carpet with less labor and the faults caused by picking process are decreased.

3.3.1 Design Requirements

In Figure 3.8, the dashed line shows the trajectory of the weft yarn during a picking operation. The weft yarn forms selvages on both sides of the carpet. These selvages prevent the carpet structure and keep the carpet forming yarns compact. The weft yarn is carried in the form of skein by the weaver. However, this weft carriage form is not suitable for a picking mechanism. Shuttle is a suitable weft yarn carrier to form selvage on the carpet sides. The shuttle can be carried easily by a suitable mechanism and the weft yarn inserted into the shed. The picking mechanism which is showed by the dashed rectangular is placed in front of the warp yarns. As shown in Figure 3.9, the warp yarns are separated by a shedding mechanism and a shed is formed. Then, the weft yarn is inserted through the shed by the picking mechanism. The picking mechanism moves along the loom width and it carries the weft shuttle to other side after each shed change.

The design requirements of the picking mechanism may be summarized as follows;

i. The picking mechanism must carry the shuttle securely along the width of the loom between warp yarns.
ii. The mechanism should form a uniform selvage at two sides of the carpet.

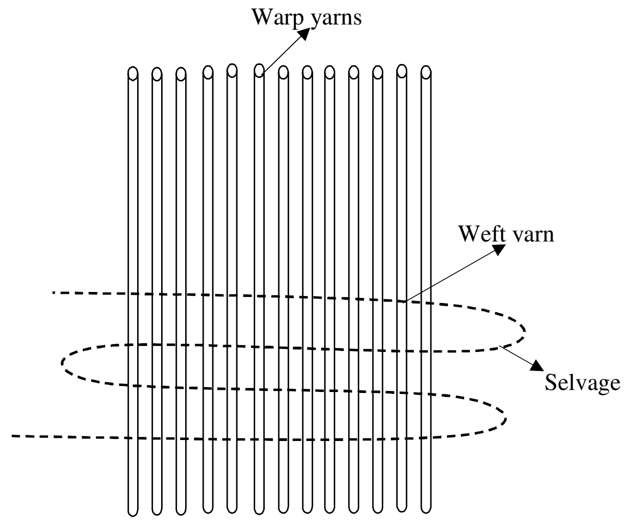


Figure 3.8 Trajectory of the weft yarn

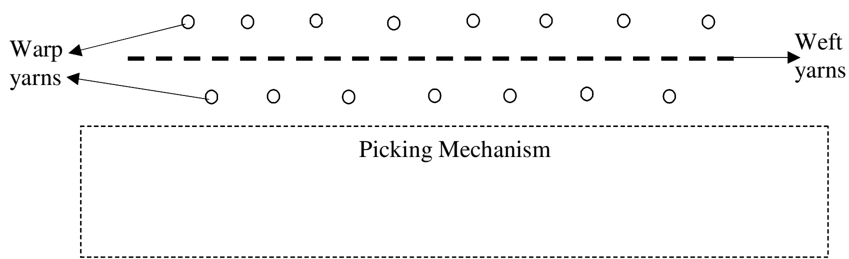


Figure 3.9 A Schematic representation of the picking mechanism

iii. The mechanism should not cause high tension and deformation on warp yarns and weft yarn during the picking operation.

iv. The mechanism should be compact in size, easy to control and manufacture, it should not need large space.

v. The picking mechanism should be able to move forward and backward and so the driver must have a feature of bi-directional control.

3.3.2 Generating Alternative Models for Picking Mechanism

According to the design requirements four different alternative models are developed for the picking process used in the handmade loom. All alternatives are based on the shuttle picking. Working principles of them are different but all the mechanisms are developed in order to carry the shuttle from one side of the loom to other side through the shed.

a. First alternative model

The first alternative model shown in Figure 3.10 consists of three pistons and a sley. Two pistons are for shuttle driving and one piston is for sley motion. The sley is pivoted near to warp yarns at a suitable distance. It makes an oscillatory motion as much as the stroke of the sley piston. The sley has two shuttle magazines at left and right sides. The sley is teethed along the wide of it, so a reed is formed between shuttle magazines. The sley piston is placed at the back of sley and it is fixed to the midpoint of the sley. Before the picking process is commenced, the sley piston is activated and it pushes the sley forward. When the new shed is opened, the sley reaches to its forward position and the teeth of the sley reed pass through the warp yarns. Thus a sliding way is formed in the shed for the shuttle. The pistons work in a

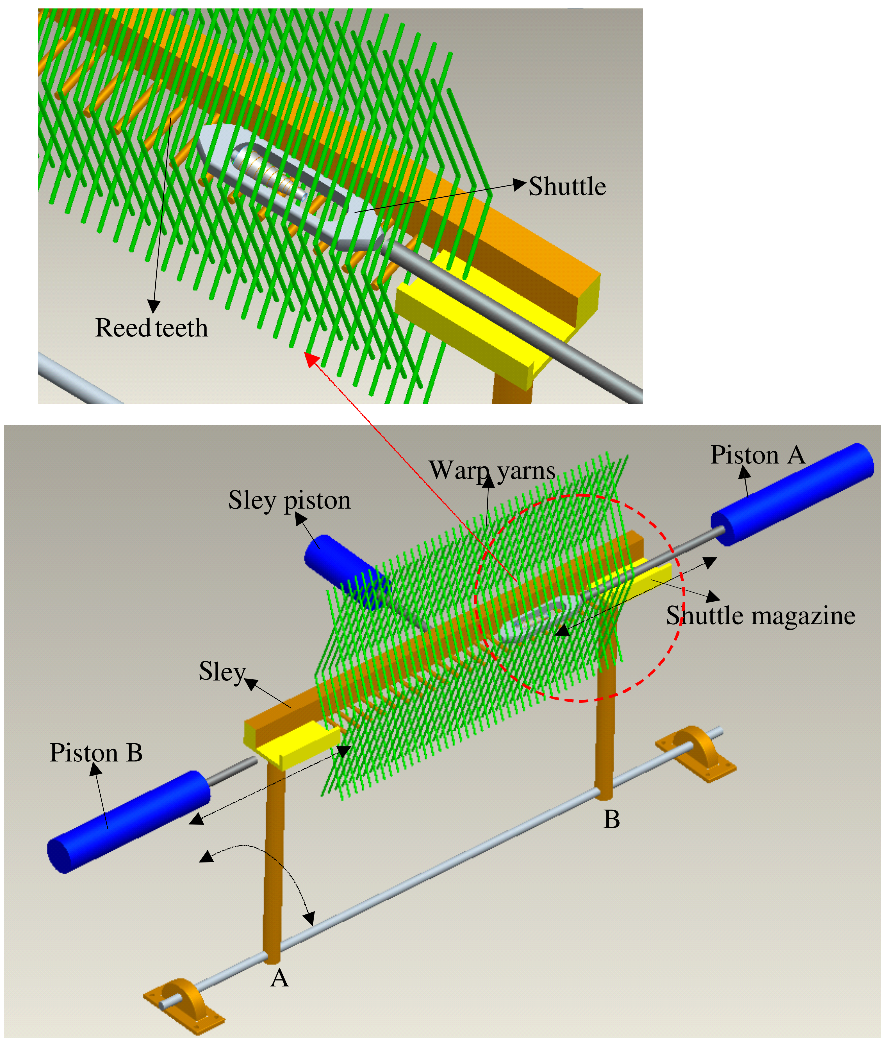
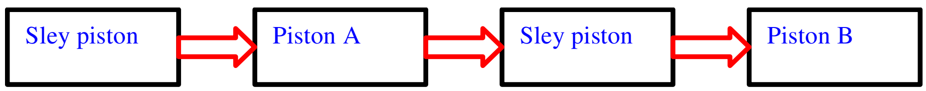


Figure 3.10 First alternative for the picking mechanism

sequence. Piston A or the piston B is activated depending on the place of the shuttle. As shown in Figure 3.10, the piston A pushes the shuttle toward the shuttle magazine at opposite side. When the shuttle is reached to the cross shuttle magazine, the sley piston pulls the sley backward. Thus it allows the beat-up mechanism to insert weft yarn and knots into carpet structure by beating on them. When the new shed is formed, the sley pushes the sley forward again. At this time the piston B is activated and it pushes the shuttle from its magazine to the opposite side. The process continues in this sequence. Before each picking operation, the sley makes one oscillation. The working sequence of the pistons is as given below;



The teeth of the reed are arranged in a suitable form. The teeth of the reed must be dense enough to provide a way for easy motion of the shuttle. On the other hand, the density of the teeth must not be over an optimum level. Since high teeth density causes higher tension on the warp yarns. In order to achieve the required picking process, the distance of the sley from warp yarns and the sequence of the pistons must be well adjusted. The stroke of the piston A and B is equal but the stroke of the sley piston is different. The strokes of the pistons are selected according to the width of the loom.

b. Second alternative model

The second alternative model shown in Figure 3.11 is a rack and pinion mechanism. It consists of a teethed shuttle carrier rod (the rack) and a gear (the pinion gear). The pinion gear forces the shuttle carrier rod forward and backward. It is driven clockwise (CW) and counter clockwise (CCW) at required rotation angle by the motor. As it rotates CW, it forces the shuttle carrier forward. When the shuttle reaches the other side of the loom and leaves the warp yarns exactly, it is driven CCW and the shuttle carrier is forced backward again. The shuttle carrier rod grooved from the bottom moves over a carrier guide placed on the loom with a distance from the wrap yarns. The carrier guide provides not only easer reciprocating to the carrier rod but also support it during the motion. As shown in Figure 3.11, the shuttle has a magnet and the carrier rod has a metal part. There is a magnetic

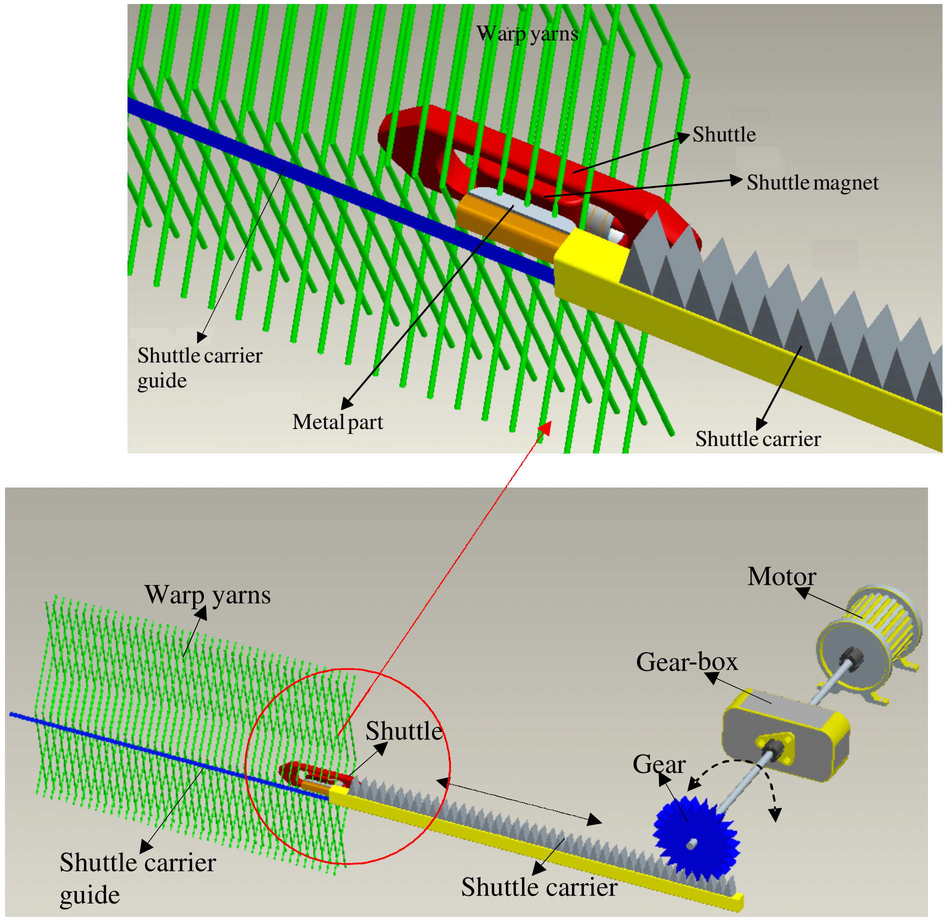


Figure 3.11 Second alternative for the picking mechanism

attraction between the shuttle and the metal part of the carrier and so, the shuttle stays on the metal part during the picking. When the shuttle inserted into shed and passes through the warp yarns, the warp yarns pass between shuttle magnet and the carrier metal part.

c. Third alternative model

The third alternative model represented in Figure 3.12 is based on carrying shuttle with magnetic attraction. It is driven by a helix shaft and the shuttle carrier is pivoted on this shaft. The pivoting point of the shuttle carrier is grooved in helix shape. When the helix shaft is rotating, the tooth of the shaft forces the shuttle carrier forward from the grooved areas. The shuttle carrier is also pivoted from the point B in order to provide a stationary vertical position during the horizontal motion. The driver motor of the mechanism is bi-directional. Thus the shuttle carrier unit can be moved forward and backward along the loom width. The helix shaft rotates

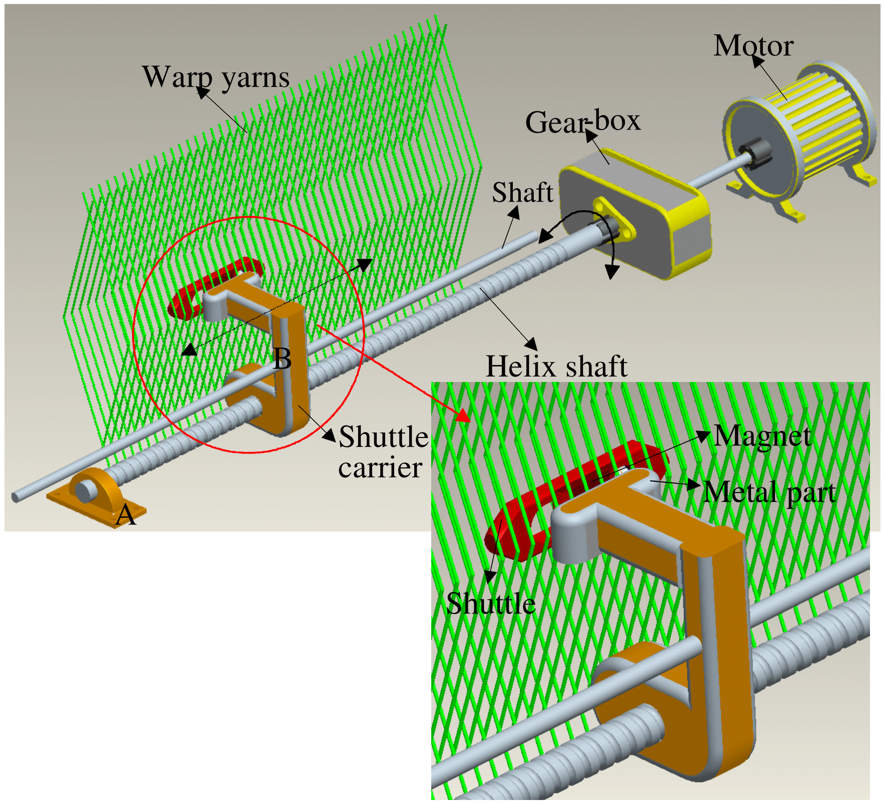


Figure 3.12 Third alternative for the picking mechanism

clockwise (CW) and counterclockwise (CCW) with a suitable rotation angle. The shuttle begins its action from one side of the loom and continues until it reaches the other side of the loom. When the shuttle carrier reaches the opposite side of the loom, the helix shaft stops its rotation in order to allow the new shed formation. When the new shed is opened, the helix shaft begins rotating opposite direction and the shuttle carrier moves to starting point. Suitable number of turns to reach the shuttle carrier from one side to opposite side will be calculated according to loom wide and helix angle of the shaft. During the picking process, the warp yarns passes between shuttle and the carrier, the shuttle doesn't leave the carrier and moves together because of the magnetic attraction.

d. Forth alternative model

This picking mechanism is used for picking operation of a shuttle weaving machine. The shuttle picking was used for old weaving machines but it is used rarely now. The rapier and the air jet picking system are widely used. Because a shuttle picking mechanism is needed, this system can be used although it is an old system.

As shown in Figure 3.13, the mechanism consists of four links, four springs, two cams, one reed and a gear train. The links A and B are joined with pin connections at points A and B respectively. These links are driven by link D and link C respectively. The link D is pivoted at the point D with a pin connection and the link C is pivoted at the point C with a pin connection. The link D and the link C are driven by cam 1 and cam 2 respectively. The directions of the cams are opposite to each other. So, when the link C is forced down by the cam 2, cam 1 and the link D are up. The return motions of the links (A, B, C and D) are provided by the springs. The reed is used to form a way in the shed for shuttle carriage. When the main shaft rotates, the motion is transmitted to the cam-shaft via the gear train. The cam-shaft has the reed makes oscillatory motion. The teeth of the reed pass through the warp yarns at the forward motion and form a way for the shuttle. The cams force the links D and C down sequentially. The links D and C transmit motion to the links A and B. The links A and B hit the shuttle in magazine and propel it forward. Thus the shuttle is carried from one magazine to other on the teeth of the reed. Because of the cams positions, when the main shaft completes a full rotation, two picking operation will be achieved. So the reed must perform two oscillations during a full cycle of the main shaft. In order to provide this situation, the diameter ratio between the main shaft gear and the cam-shaft gear must be two.

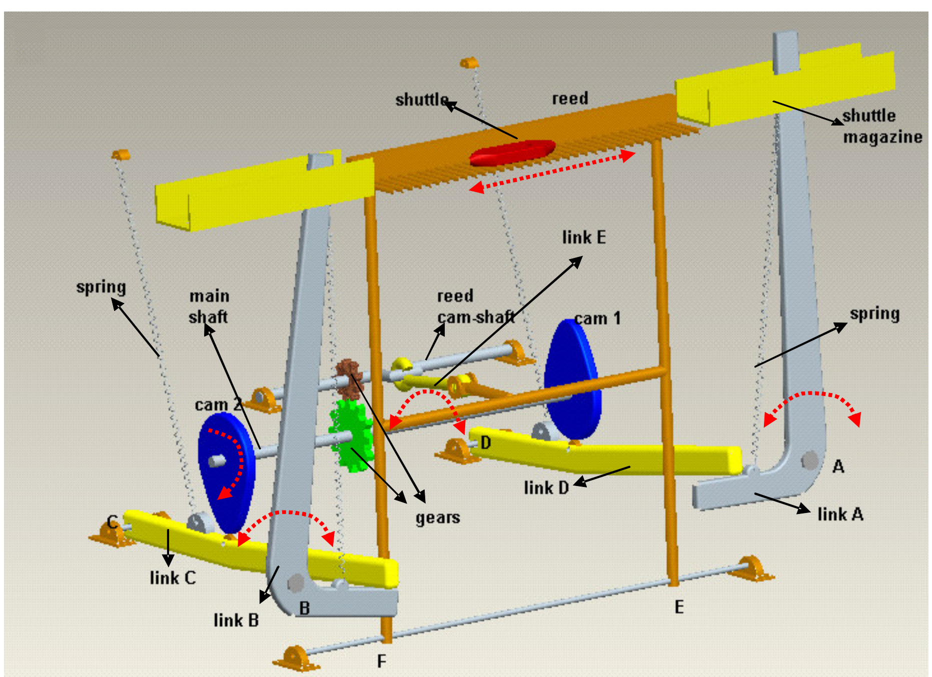


Figure 3.13 Fourth alternative for picking mechanism

Evaluation of the alternative models

There are four different alternative models for the picking mechanism. Since the selvage formation is required in the design criterias, the mechanisms are generated in order to carry the shuttle along the width of loom. All of the mechanisms can carry out the transport of the shuttle and manage the required picking process.

The first model consists of three pistons and these pistons must run in a sequence. So, the control of the first model may be difficult. The longer the width of the loom is, the higher the stroke is needed. The more energy is consumed for the shuttle carriage. This may increase the cost of this model. This model needs a reed for the shuttle carriage. The design and construction of this reed may be difficult and cause high manufacturing costs.

The second and the third models carry the shuttle with a magnetic action. These models need smaller shed than the first and the fourth models need. Also, these models have fewer members than the other two ones. So, the control of these models may be easier than the others. Both second and the third model need a bi-directional control. The second model needs more working area than the third model needs. The third model has a compact structure. The carrier link of the second model must have teeth in order to be forced forward and backward by gear and the carrier of the third model must be grooved with regard to shape of helix shaft. These parts must be properly designed and meshed. So, the design and manufacturing of these models may be difficult.

The fourth model also provides the design requirements of picking operation. However, it is the most difficult one to control within four alternative models. Since there are many links that must be adjusted according to each other motions, the construction of this model may be difficult and high cost. The design of the cam profile and synthesis of the links may be difficult than the other models.

CHAPTER 4

DESIGN OF BEAT-UP MECHANISM

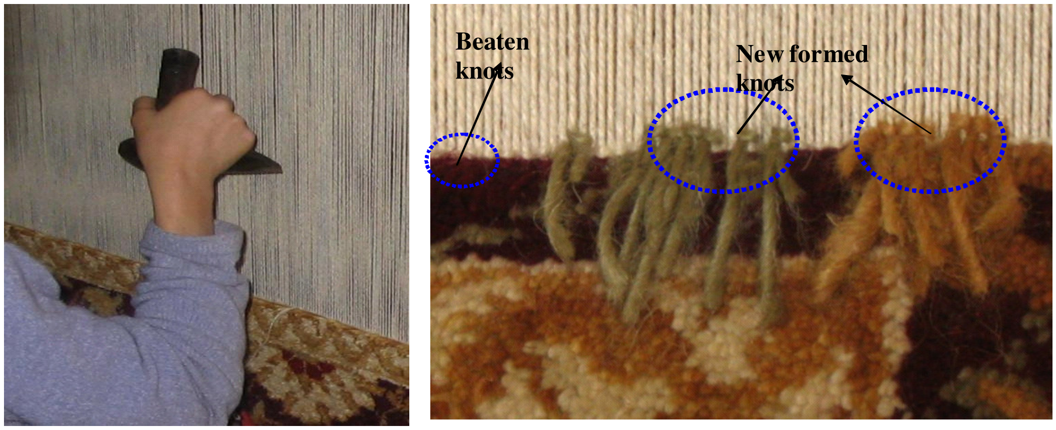
4.1 INTRODUCTION

In the handmade carpet production, the beat-up operation is one of the most important processes after the knotting. This operation has a great influence on the quality of the handmade carpet. It provides tightness and compactness to the carpet structure. A smooth pile surface is obtained as a result of this process. The beat-up process is also one of the most time-consuming and weaver-exhausting process. It requires special attention and experience. This process is now performed by a weaver via the comb instrument. If it is performed by a suitable mechanism, the production speed will certainly be increased. The weaver will be less tired and the faults will be decreased.

In this chapter, the beat-up process in hand made carpet production and its influence on the carpet structure and the quality will be explained. The alternative mechanisms that can be used for beat-up process are then generated. These mechanisms are evaluated, and the best solutions satisfying the functional requirements and design criterias are selected. Finally, dimensions of the mechanisms are determined by analytical, graphical and graph-analytical methods.

4.2 BEAT-UP PROCESS

After one row of the knots is completed, a weft yarn is passed through the warp yarns. Then, the weaver beats the knots and weft yarn into the carpet structure by using an instrument called as comb (Figure 4.1a). The new formed knots are loose away from the cloth fell as shown in Figure 4.1b. They are not in the carpet structure. The knots are inserted into the carpet structure and they are tightened by means of the beat-up process.



(a) Beat-up operation (b) New formed and beaten knots on the carpet

Figure 4.1 The beat-up process

The force exerted on the knots and weft yarns in the width of comb must have same intensity at every point of the carpet. This is the most important point for this process. The amount of the force must be adjusted to an optimum value. If this force is higher than the optimum value, the beat-up process causes high tension on the warp yarns and damages the warp yarns and the knots. So, the strength of the carpet decreases and the life of the carpet gets shorten. When the force is lower than the optimum value, the required carpet density and tightness cannot be obtained. This condition decreases the carpet quality. If the force is exerted on the knots and the weft yarn at the optimum value, the required carpet density and quality will be obtained. The back of the carpet will also be uniform, and so it is seen beautiful. In the literature, there isn't any calculated value on the amount of the force applied by the beater during the carpet weaving. It is difficult to determine the amount of the force, since it is completely adjusted by human sensitivity. According to the experienced weavers, the optimum force value for beating process is about 8-10 kg-force in about 5 cm width of the comb.

4.3 BEAT-UP MECHANISM

In this study, the movement of the experienced weaver hand is taken as a model for designing the beat-up mechanism. Therefore, the weaver hand motion is analyzed and investigated before developing the alternative trajectories for the mechanism. An experienced weaver is observed to determine the path of the weaver hand during the beat-up process as shown in Figure 4.2. The initial (P_1), the second (P_2) and the final

a) Initial position of the weaver hand b) Midpoint of the weaver hand c)End point of the weaver hand

Figure 4.2 The beat-up process performed by the weaver

(P_3) positions of the weaver hand are established on the trajectory shown in Figures 4.2(a), (b), (c) respectively. When these points are represented in a coordinate system, the trajectory followed by the weaver is made firm approximately as in Figure 4.3.

Here, the problem is to design a beat-up mechanism that will tighten the knots and weft yarn into carpet structure by exerting an enough force on them (about 8-10 kg-force). The mechanism has at least one degree of freedom and a beater, like in Figure 4.1(a), must be designed for the output link. The warp yarns, weft yarn, knots, the position of the designed beat-up mechanism and the trajectory of the mechanism are

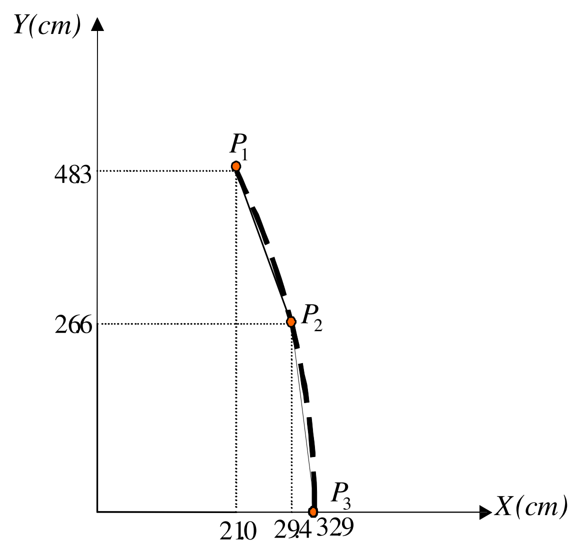


Figure 4.3 Path of the weaver hand during beat-up process

presented in Figure 4.4. The mechanism is capable of producing an oscillatory motion by moving down and up. In the down motion, the mechanism traces a trajectory (represented by dashed line), passing through three precision points (P_1, P_2, and P_3). It starts its motion from the point P_1, then the teeth of the beater pass through the warp yarns at point P_2. Finally it pushes the weft yarn and corresponding knots to point P_3 (cloth fell). Similarly, it traces the same trajectory in the up motion and returns to its initial position for the next beat-up operation. After beat-up operation is completed, the mechanism is moved to a certain distance for the next operation. In order to move the beat-up mechanism along the loom width after each beat-up operation, it is placed on a slider mechanism.

The degree of freedom for the beat-up mechanism can be selected as one or two. In general, when designing a mechanism having one DOF, the point synthesis is suitable. In this method, a specific point on the mechanism follows a trajectory which passes through a set of predefined positions. However, if the control of the full trajectory is important for the mechanism design, a two-DOF mechanism or a cam mechanism can be recommended. Three alternative trajectories are given in Figure 4.5 (a), (b) and (c), respectively. The first alternative is similar to the path of the weaver hand during beat-up process. It requires a single DOF mechanism and follows the trajectory by passing through the three design point. It is also easier than others to analysis. The second alternative trajectory may require two DOF

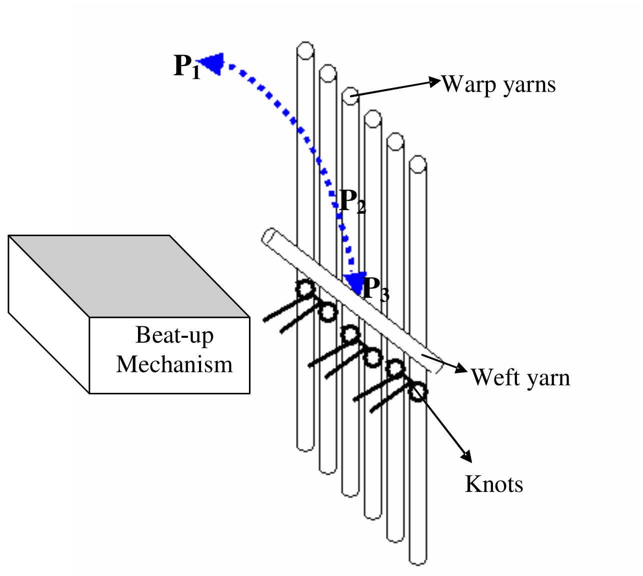
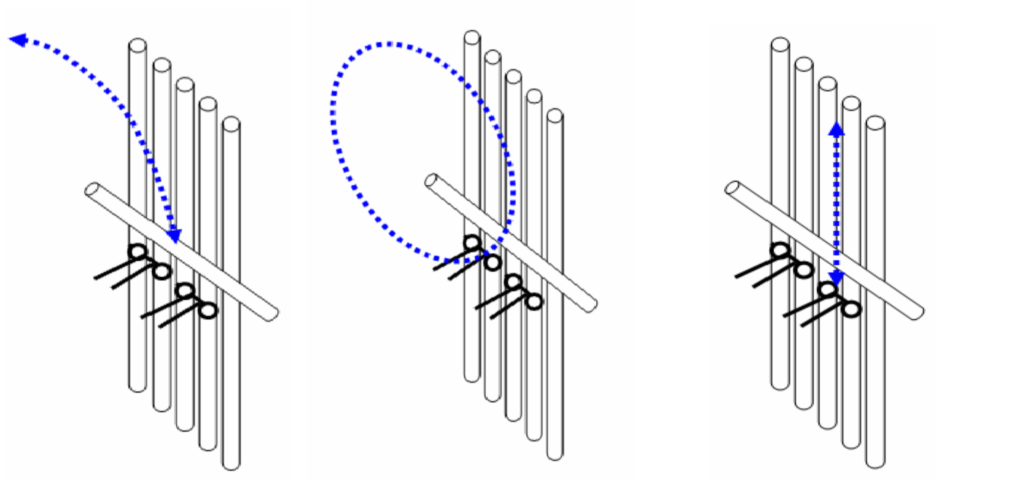


Figure 4.4 Representation of the beat-up mechanism



a)The first alternative trajectory b)The second alternative trajectory c)The third alternative trajectory

Figure 4.5 Alternative trajectories for beat-up mechanism

mechanism so it is more difficult to control and analysis. The third alternative trajectory is an up-down motion. It is suitable to use a reed along the loom width for beat-up process. When the beater is used in the third alternative, the mechanism can not move along width of the loom because the beater doesn't leave the warp yarns. The second trajectory could require more time than the first and the third alternatives to complete a full cycle during a beat-up operation. As a result, the first alternative can be selected as the best trajectory. The functional requirements of the mechanism can be summarized as the following:

1. If the control of a specific point on the beat-up mechanism draws a trajectory which passes through three precision points, the point synthesis will be used. One-DOF mechanism will be selected.
2. If the control of specific points on the beat-up mechanism following a full trajectory is required, a two-DOF mechanism or a cam mechanism will be used.
3. The beat-up mechanism should be able to make an oscillatory motion by down-up motions and the driver system may have a feature of bi-directional control.

4.3.1 Design Specifications

The functional requirements of the mechanism containing the general information are translated into the design specifications given. These specifications are used for determining the factors before designing the beat-up mechanism.

i. The mechanism should be able to run at high speeds to tighten the knots and weft yarn into carpet structure by exerting an enough force (8 or 10 kg-force) on them.

ii. The mechanism should be capable of producing an oscillatory motion by moving down-up. It should trace a trajectory represented in Figure 4.4 by passing through precision points.

iii. The mechanism should have possible one degree of freedom at least and should be designed with a beater shown in Figure 4.1(a) for its output link.

iv. The mechanism should ensure that the amount of force (8 or 10 kg-force) applied by the beater is equal at every point of the carpet along the width.

v. The mechanism should have a compact structure and shouldn't need much space to operate.

vi. The mechanism should be simple, easy to control, and with low cost.

4.4 ALTERNATIVE SOLUTIONS

Eight different alternative models are developed for beat-up mechanism. These mechanisms are two-link mechanism (1), four-link mechanism (5), cam-link mechanism (1), and inverted slider crank mechanism (1). In following sections; the working principle of the mechanisms will be explained and compared in terms of the functional requirements and design specifications.

4.4.1 Two Link Mechanism

The mechanism shown in Figure 4.6 consists of one link, a gear train and a driving unit (motor). The beater is placed on the tip of the link. The torque generated by a motor is transmitted to the link via a gear train. The motor is bi-directional. So, it can drive the link clockwise (CW) and counterclockwise (CCW). The ground of the

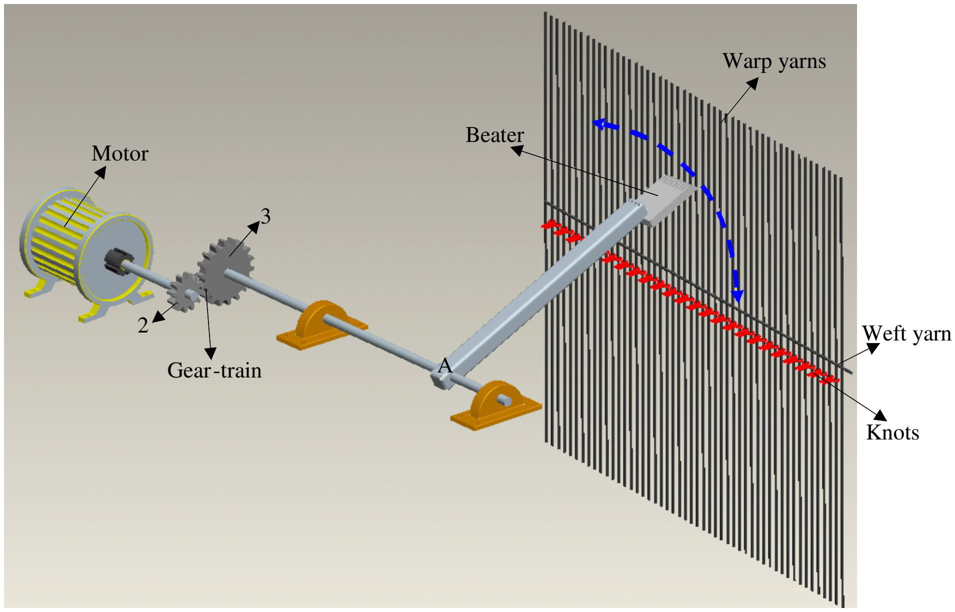


Figure 4.6 Two link mechanism

mechanism must be placed on a suitable position. The rotation angle of the motor shaft should be adjusted according to the required beat-up force and stroke.

Evaluation

The mechanism shown in Figure 4.6 is evaluated. The mechanism satisfies the functions of the beat-up process theoretically. Since the mechanism has a single link, the control of the mechanism is quite easy. The more links in the mechanism, the more difficult to control it. This mechanism doesn't need much space on the loom frame. Required force for beat-up process can be adjusted with gear ratio of the gear train and the power of the motor. As a result, this mechanism can be used in a beat-up process.

4.4.2 Four-Link Mechanisms

a. First alternative model

The first alternative model is a crank-rocker mechanism with four links & ground in Figure 4.7. The links are jointed to each other with revolute connections. The beater is placed on the tip of link DC by a rigid joint. The ground pivots must be located at reasonable locations, so that the beater encounters with the warp threads at a suitable point. It then exerts the required force without causing over tension on warp threads.

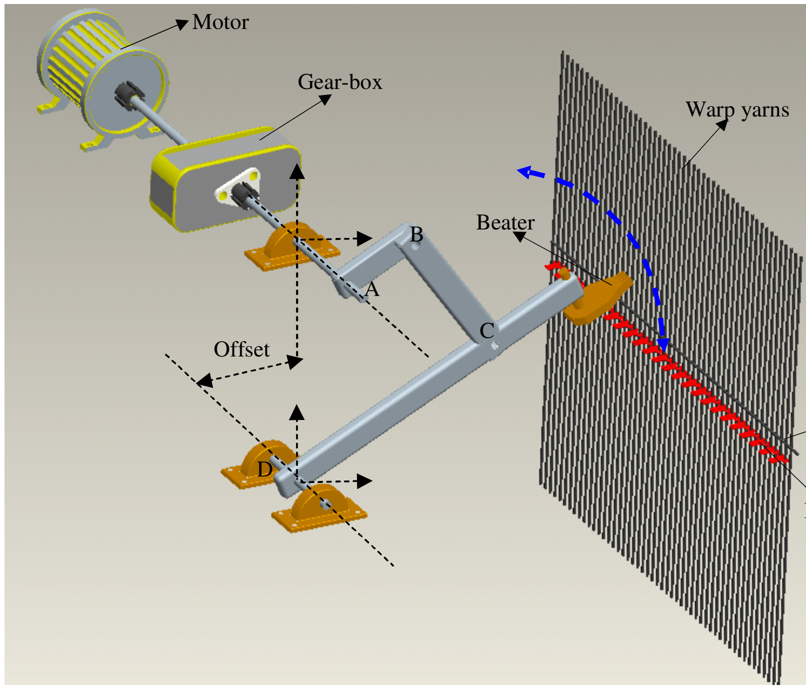


Figure 4.7 First alternative for four-link mechanism

As shown in Figure 4.7, the mechanism is driven by a motor through the crank AB. There is an offset between the point A and point D, these points are not collinear. As the link AB is driven, its motion is transmitted to the output link DC (rocker) by coupler link BC. During a beat-up operation, as the link AB makes a full rotation, the output link DC oscillates between two limit points as shown with dashed line in Figure 4.7. The trajectory followed by the link DC is similar to the path of the weaver hand during beat-up process (Figure 4.3). When the link DC is moving down from its top limit point, the teeth of the beater inserts through the warp yarns and comes in contact with the weft yarn. The beater pushes the weft thread and the corresponding knots to the cloth fell until the link DC reaches to its bottom limit point. Thus the weft yarn and knots at the processed area are inserted into the carpet structure. After the link DC has reached to its bottom limit point, it moves to its initial position from the same path. During the full rotation of the link AB, the beater makes one beating and returns to its initial position. The stroke of the output link DC determines the link length ratios of the mechanism and also force exerted by the beater. So, the required density is obtained in the carpet structure. Since the beater will beat the weft yarn and knots with small areas, the mechanism must be moved along the loom width by a slide mechanism.

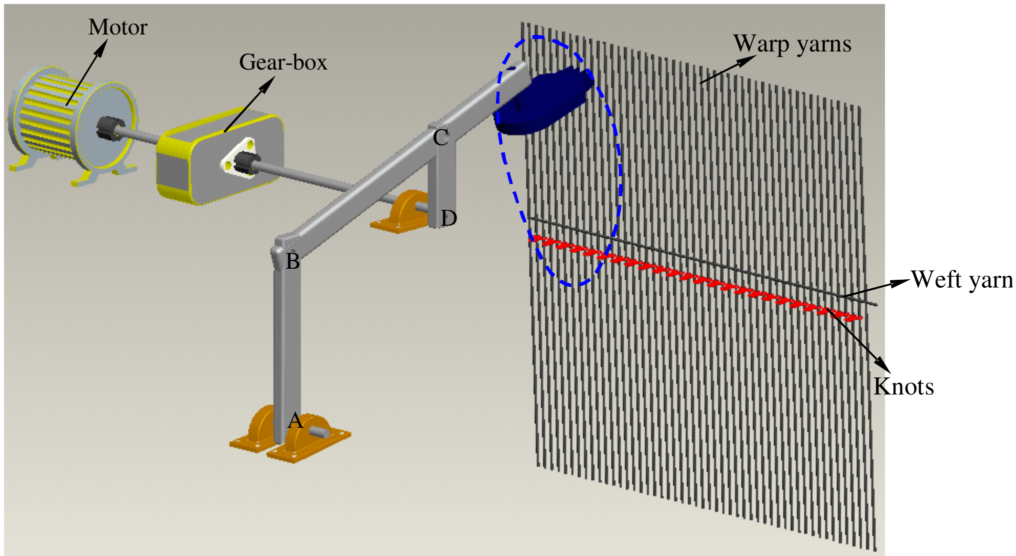


Figure 4.8 Second alternative for four-link mechanism

b. Second alternative model

The second alternative model represented in Figure 4.8 is another type of crank-rocker mechanism. The links of the mechanism are joined to each other with pin connections. The beater is joined to the coupler link BC with a rigid connection. The trace of the mechanism showed with dashed line is followed by the coupler link BC. As mentioned in the first model, the mechanism is placed in a proper place to obtain required beating process. When the crank DC is given a rotation motion by the motor, the coupler BC moves down and the teeth of beater inserts through the warp threads. As the link BC continues moving down to the cloth fell, the beater pushes the weft yarn and knots into carpet structure. After the link BC has reached to the cloth fell, it moves a distance backward and the teeth of the beater leaves exactly from the warp threads. Then, the link BC rises to its initial position. During a full rotation of the link DC, the link BC with the beater makes one beating and returns to its initial position by following trajectory shown with dashed line. The link lengths can be determined by selecting three points on the mechanism path and making synthesis over these points.

c. Third alternative model

This model shown in Figure 4.9 is a movie camera film advance mechanism. It is used for pushing forward the film into the camera. The profile of the mechanism is

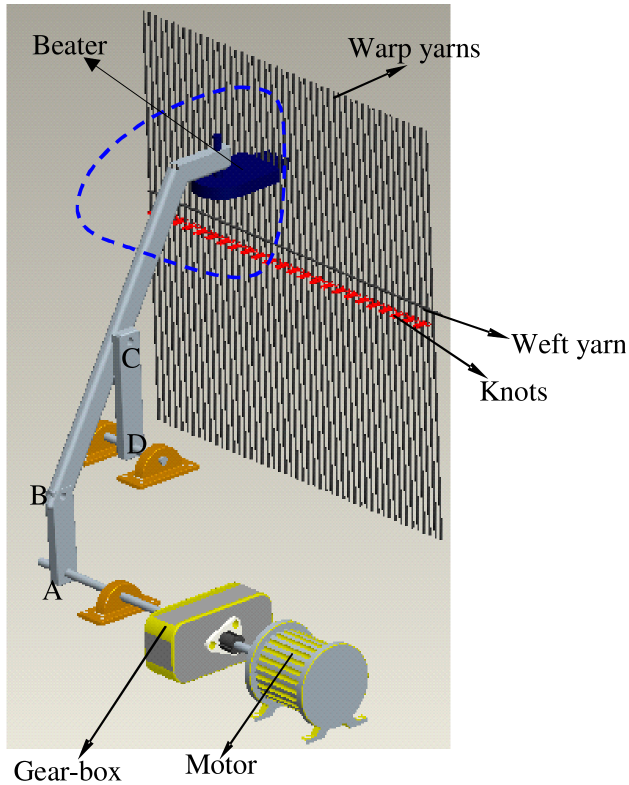


Figure 4.9 Third alternative for four-link mechanism

also suitable for beat-up operation of a hand loom. Working principle and structure of this model is same as that of second alternative model. The trajectory of the mechanism shown with dashed line in Figure 4.9 is followed by the coupler link BC during a beat-up operation. Their trajectories are different. This causes kinematic and kinetic differences between them. During every cycle of the link AB, the link BC performs one beating operation and returns to its top position.

d. Fourth alternative model

This alternative model (Figure 4.10) is a crank-rocker type four-link mechanism. The points A and D are collinear. The link DC is crank and the link AB is the rocker of the mechanism. As the crank DC performs a full rotation, the rocker AB of the mechanism oscillates between two dead positions. The beater is joined to the rocker AB with a rigid connection. The profile and the working principle of the mechanism are similar to that of the first alternative model. When the crank DC is driven by the motor, the rocker AB begins to move down from its top position $P_{1.}$ The teeth of the beater inserts through the warp yarns at point P_2. As the rocker AB moves down to

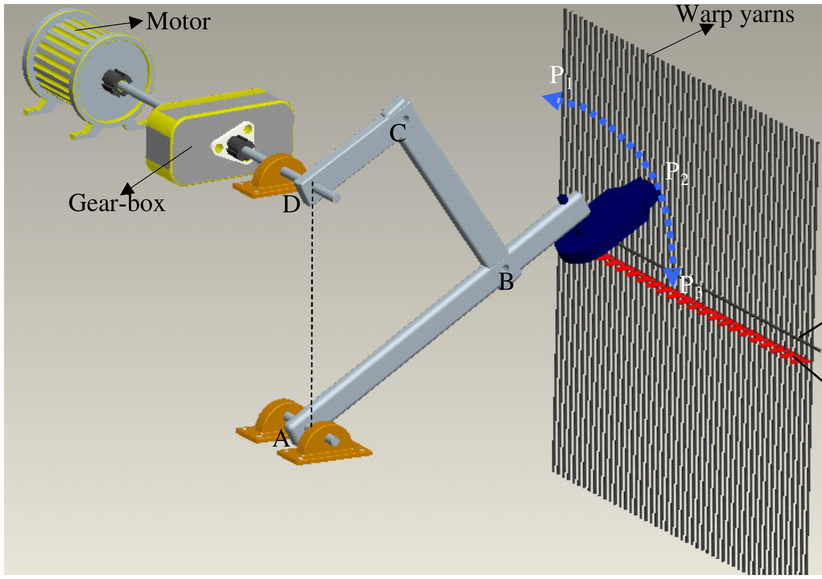


Figure 4.10 Fourth alternative for four-link mechanism

its bottom position, the beater pushes the weft yarn and corresponding knots to the cloth fell point P_3. The beater inserts the knots and the weft yarn at processed area into the structure of the carpet at point P_3. After the rocker AB has reached to bottom point P_3, it begins to move up and reaches to the initial point following the same path. The mechanism is placed on a slide mechanism and moved along the loom width.

e. Fifth alternative model

A reed is used in order to perform the beat-up process in this model (Figure 4.11). In the previous models, the knots and the weft yarn are inserted into the carpet structure via the beater with many beating process along the loom width. But in this model, the weft yarn and knots are inserted into carpet with a single beating operation. Thus, this mechanism provides a quick beat-up process. Since the input links (AB and DE) and the output links (GC and HF) make oscillatory motions, this model is a rocker-rocker type mechanism. The reed is joined to link GC and link HF with rigid connections. The mechanism is driven from the link AB and the link DE as shown in Figure 4.11. These links are driven from a single driver via a shaft and perform the same motion. The driver must have a bi-directional control, since the links BC and EF will coincide with the driving shaft the when link AB and the link DE perform full rotation. The oscillatory motion of the link AB and the link DE is transmitted to

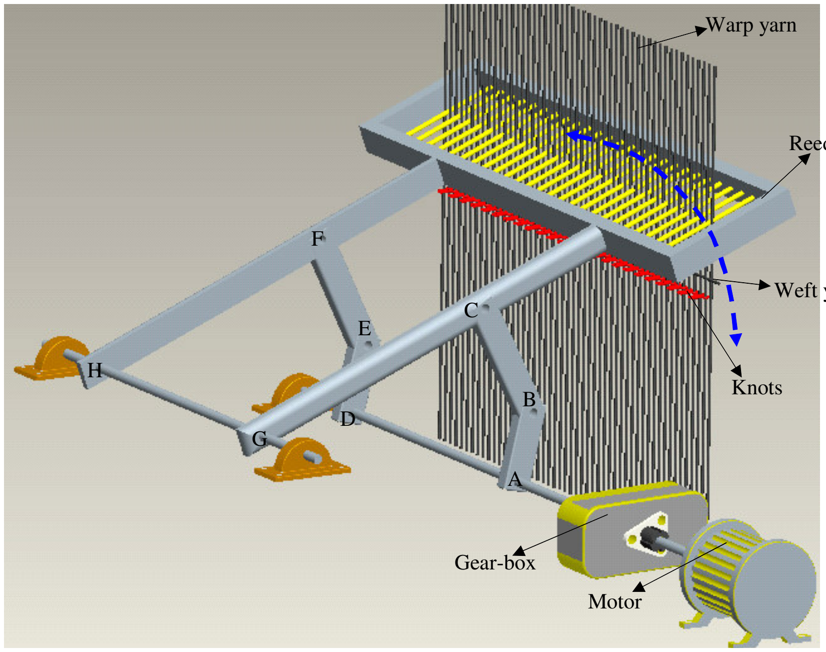


Figure 4.11 Fifth alternative for four-link mechanism

the link GC and the link HF at the same time via the links BC and EF respectively. There is an offset between the line HG and AD. If the wires of the reed have better contact with the weft yarn and knots, the beat-up operation becomes more effective. In order to provide this, the reed comes to horizontal position around the beat-up point. So the links GC and HF carrying the reed are pivoted at a higher position than the line AD. In order to provide a uniform beating along the loom wide, the reed must be rigid and compact. The dents (wire) of the reed must have enough strength to provide required beat-up force.

<u>Evaluation of the mechanisms</u>

Here a comparison between the alternative models is performed. The mechanisms are compared in terms of various specifications; functional requirements and some design specifications (D.S) like in Table 4.1 the first row shows the alternative models and the first column shows the specifications. "+" sign means that the mechanism may satisfy the specification and "-"sign means that it may not. All models can certainly satisfy the functional requirements for the beat-up process. The first and the fourth mechanisms have the same trajectory resembling that of handmade carpet weaver. These mechanisms have similar working principle and

Table 4.1 Evaluation of four-link mechanisms

Specifications	**The First Model**	**The Second Model**	**The Third Model**	**The Fourth Model**	**The Fifth Model**
Functional Requirements	+	+	+	+	+
Design Spec. (ii) – Oscillatory motion	+	+	+	+	+
Design Spec. (iii) –one DOF	+	+	+	+	+
Design Spec. (v) -Compact Structure	+	+	+	+	-
Design Spec. (v) -Less Spacing	+	-	-	+	-
Design Spec. (vi) -Better Controllability	+	-	-	+	+
Design Spec. (vi) -Easy Manufacturing	+	+	+	+	-
Target	**+7**	+5	+5	**+7**	+4

structure. The second and the third mechanisms have similar working principle. The trajectory is followed by the coupler link in both of these mechanisms. Because of their moving trajectory, the beat-up process may take more time then that of the first and the fourth mechanisms. Moreover the second and the third mechanisms need more space to complete their process cycle. All of the mechanisms have one DOF. The fifth mechanism is different from other mechanisms in terms of construction. Reed is used for beat-up operation instead of beater. The beat-up operation takes less time with the fifth model, because the operation is made along the width of the loom with single rotation of the shaft. But it may have some disadvantages in terms of the proper reed manufacturing and mechanism construction. The fifth mechanism needs more space. The reed must be rigid and compact enough to provide a uniform beating along the loom width and to obtain uniform density in carpet structure. The dent (wire) of the reed must be able to withstand the beat-up force.

The first mechanism in Figure 4.7 and the fourth mechanism in Figure 4.10 are seen to be the best alternatives between other models. Both offers compact structure and higher processing speed.

4.4.3 Cam-Link Mechanism

In this alternative model (Figure 4.12), the beat-up operation is carried out by a reed as the fifth alternative model. The mechanism consists of a reed, two identical cams having the same profile and shape and two follower links. The reed is connected to two follower links on both sides. The followers are jointed with cylindrical connections in order to provide a linear motion. As the cams are rotating, the rollers at the bottom of the follower links move on the cams and trace the cam profile. During a full cycle of the cams, the reed is moved up and down via the follower links. During the down motion of the reed, it pushes the weft yarn and knots into the carpet fell. After the reed has reached its bottom position, it begins rising again and reaches to peak point for the next beat-up operation. The rigidity of the reed must be sufficient to exert the required force on the weft yarn and knots along the loom width and provide a uniform force distribution.

<u>Evaluation</u>

This mechanism satisfies the beat-up process. The most important factor in this mechanism model is to design of cam profile acting the reed. The stroke of the reed must be enough to exert required beat-up force (8-10 kg.force) on the weft yarn and

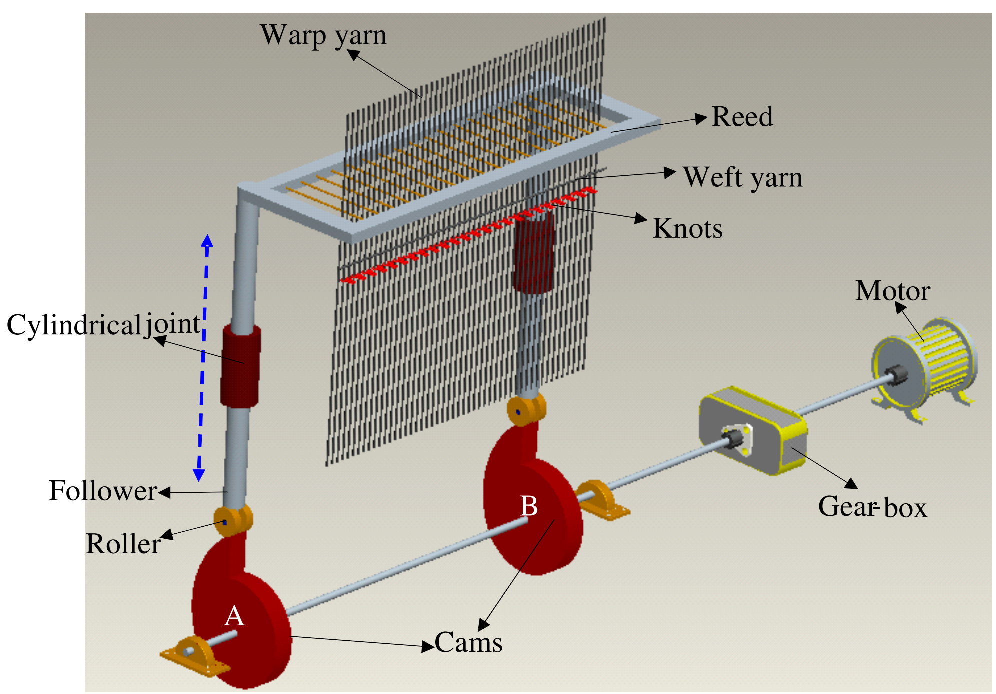


Figure 4.12 Cam-link mechanism

knots. If the reed exerts the excess force on the warp yarns, it may cause the warp yarn breakages because of high tension. If the reed does not provide the required force on the weft yarn and knots, the proper carpet tightness can not be obtained. The mechanism needs much space to run. The reed must be rigid, compact and strong enough to obtain a uniform force distribution. The wires of the reed must have enough strength to provide the required beat-up force. So, this mechanism is complex and difficult to analyze and manufacture.

4.4.4 Inverted Slider-Crank Mechanism

This model is an inverted slider crank mechanism (Figure 4. 13) which consists of a circular disc and a slotted link. The circular disc 2 is joined to the link 1 with a slotted cylinder connection. The beater is placed on the link 1 and follows the trajectory shown with dashed line in Figure 4.13. The working principle and the trajectory of the mechanism are similar to that of the first and fourth alternative models of the four-link mechanisms. However, the link structure and the mechanism designs are different. The mechanism is driven through the circular disc 2. The wedge placed on the circular disc at point B transmits the rotation motion to the slotted link 1. As the circular disc 2 rotates, the link 1 oscillates between two limit points and pushes the weft yarn and knots to the cloth fell. The stroke of the link 1 is determined according to the required beat-up force and specified points that must be

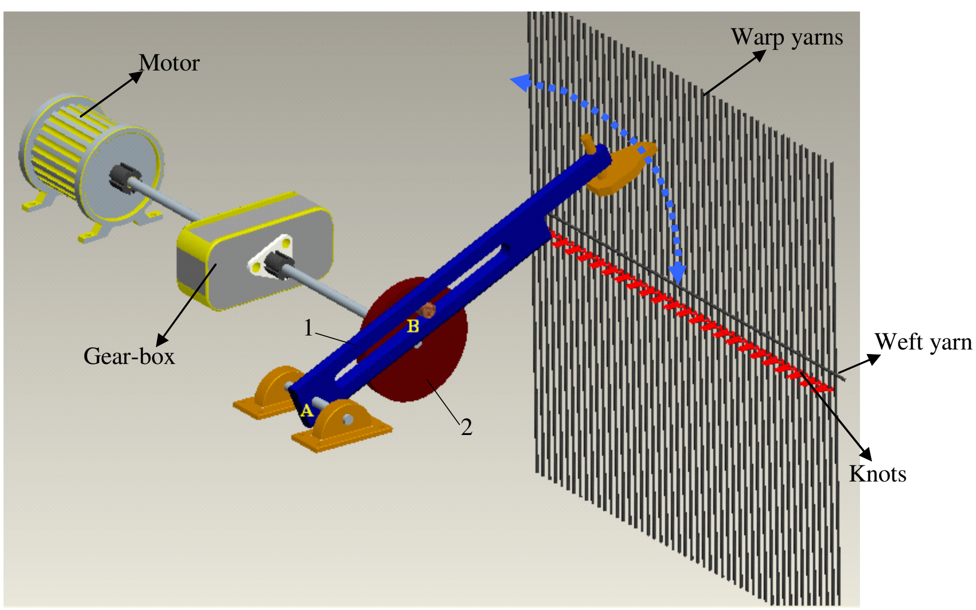


Figure 4.13 Inverted slider-crank mechanism

followed on the trajectory. The distance of the wedge at the point B from the centre of circular link 2 determines the stroke of the link 1. During a full rotation of the circular disc, the beater performs one beat-up process.

Evaluation

This mechanism satisfies the functional requirements of the beat-up process, and it has less number of links than all other alternative models. So the control of the mechanism is easier than the others. The trajectory followed by the mechanism is similar to that of a weaver. But it is quite difficult to manufacture and analyze. The friction force at the slotted cylinder kinematic pair (Figure 4.13) is also a disadvantage for this mechanism model.

4.5 DIMENSIONAL SYNTHESIS

In concept design studies, the solutions to kinematic synthesis problems, related to the whole process of design of a new mechanical device are grouped into two different categories [65-68]. The first category includes the determination of the topological arrangement of links and the nature of the kinematics connections between links, and is called 'type and number synthesis' (conceptual design). The second category concerns with the determination of the suitable dimensions of the mechanism; necessary to satisfy the desired motions, and is called as 'dimensional synthesis' (size or geometric synthesis). That is, the requirement is to determine all necessary design variables by defining the geometry of a mechanism while satisfying certain motion requirements. Many dimensional synthesis problems are included in one of the three following categories depending on the task that the linkage performs [65-68]; function generation synthesis, path generation synthesis and motion generation synthesis (rigid body guidance).

A four link mechanism in Figure 4.14 is used to illustrate each synthesis category. An example of function generation synthesis is shown in Figure 4.15 (a). The objective of this synthesis is to obtain an output angle σ as close as possible to a given function of the input angle θ . Typically a double-rocker or crank rocker results with rotational input and pure rotational output. A slider-crank linkage can be a function generator as well rotation in and translation out or vice versa. In path generation synthesis, (Figure 4.15 (b)), the main concern is only with the path of a tracer point and not with the rotation of the coupler link. This is typically accomplished by a four link crank-rocker or double-rocker wherein a point on the coupler traces the desired output path. No attempt is made in path generation to control the orientation of the link containing the point of interest. The coupler curve is made to pass through a set of desired output points. In motion generation synthesis (Figure 4.15 (c)), the entire motion of the coupler link is concerned: the path tracer point x-y co-ordinates and the angular orientation α of the coupler link. Here orientation of the link containing the line is important. This is a more general problem than the path generation, and in fact, it is a subset of motion generation [65].

In this section the dimensional synthesis of two mechanisms are given. The selected mechanisms for dimensional synthesis are the first and fourth alternative models (Figure 4.7 and Figure 4.10 respectively). The function generation method is suitable for the dimensional synthesis of both mechanisms.

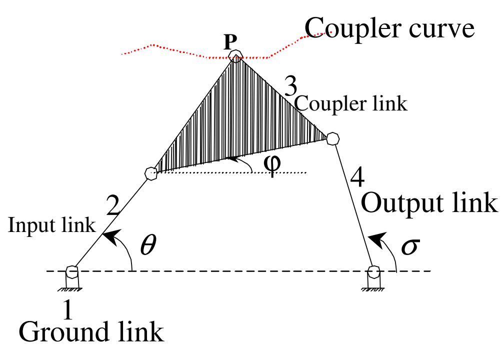


Figure 4.14 Four-link mechanism

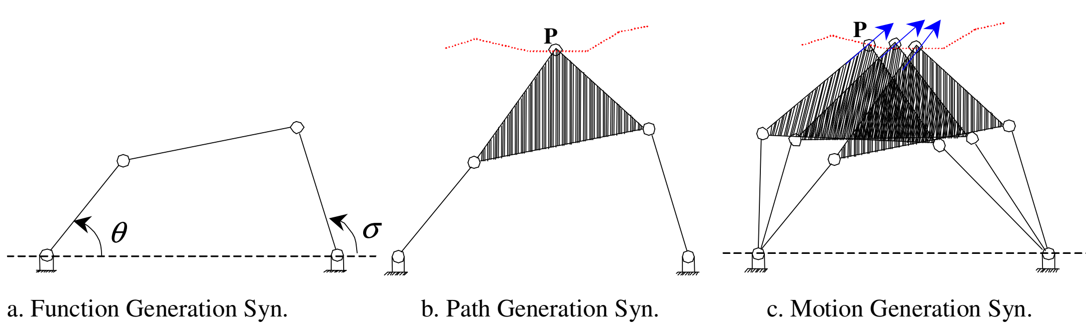


Figure 4.15 Four-link mechanism with function, path and motion generation tasks

In the synthesis of first model of the four-link mechanisms, a crank-rocker type mechanism, the calculations are carried out for different crank rotation angles. An appropriate crank rotation angle satisfying the design requirements is selected from the results. The analytical solution represented by Söylemez [69] is used for synthesis with optimum transmission angle. The link length ratios are found for the selected optimum crank rotation angle from the Chart-1 [70] and Chart-2 [71]. The results obtained both from the analytical method and found on the Charts (Chart 1 and Chart 2) are then compared.

In dimensional synthesis of the fourth model of four-link mechanisms, a crank rocker type mechanism, the swing angle of the output link between its dead centre positions and the corresponding crank rotation determines length of each link of the mechanism. The synthesis of the fourth model, carried out with both the graph-analytical and the graphical methods. The results are then compared. The graphical solution is performed by using Alt charts (Chart-3) [70]. From this chart, the best initial crank angle is selected that satisfies the condition of least deviation of transmission angle from 90^o. Then dimensions of the links in the mechanism are determined by applying the graphical solution. The graph-analytical solution of the crank-rocker mechanism is made according to the theory represented by Khare and Dave [72] for the optimum transmission angle. Then calculations are done for the different crank rotation angles according to given theory and the best solution is selected.

The selected mechanisms can perform an oscillatory motion by the down and up forward motions in the output links. By applying the three point synthesis on the trajectory given in Figure 4.16, the oscillation angle (ψ) of the output link is determined. For the beater on the output link, P_1 is initial point, P_2 is the middle point and P_3 is beating point. D is the pivoting point of the output link (rocker). When a line is drawn from this point to P_1 and another line is drawn to point P_3 (beating point), the angle between these lines give the oscillation angle, which is the angle between extended death position and folded death position of the mechanism. This angle is measured as $\psi = 30^\circ$. On the other hand, there are some structural properties that the mechanism must have. These are transfer of the force exerted by a

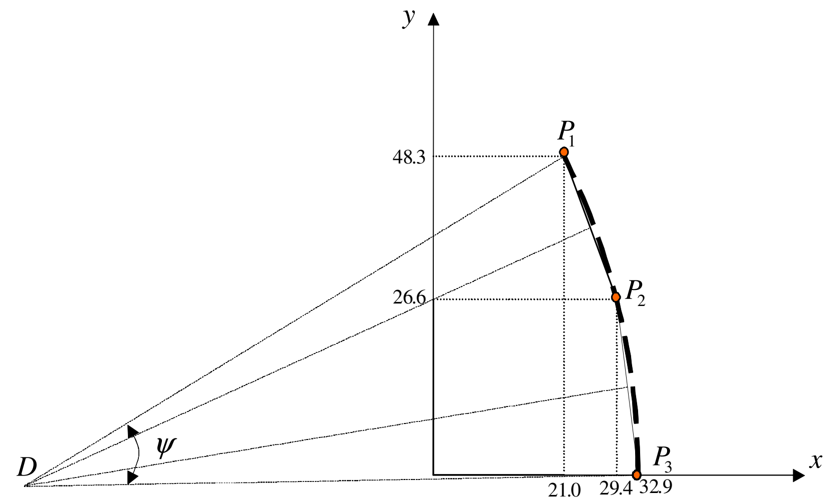


Figure 4.16 The oscillation angle (ψ)

driver to the rocker without losing it. Furthermore, the excess load must not be exerted to joints while the mechanism is running and the mechanism must not have toggle. As a result, the mechanism to be designed must satisfy the following criterias besides the oscillation angle.

1. Optimum transmission angle
2. Grashof's condition
3. Optimum crank\coupler ratio

1. Transmission angle

It is rather important to understand how the mechanism will function under loaded conditions in practice. The transmission angle determines the performance of the mechanism which is meant as the effective transmission of motion from the input link to the output link. This also means that for a constant torque input, in a well performing mechanism, the maximum torque output must be obtained and the bearing forces must be in minimum value. For a given force in the coupler link, the torque transmitted to the output bar is maximum when the transmission angle approaches to 90°.

In Figure 4.17 and Figure 4.18, the transmission angle for a four-bar mechanism and a slider-crank mechanism are shown respectively. When the transmission angle deviates significantly from 90°, the torque on the output bar decreases and may not

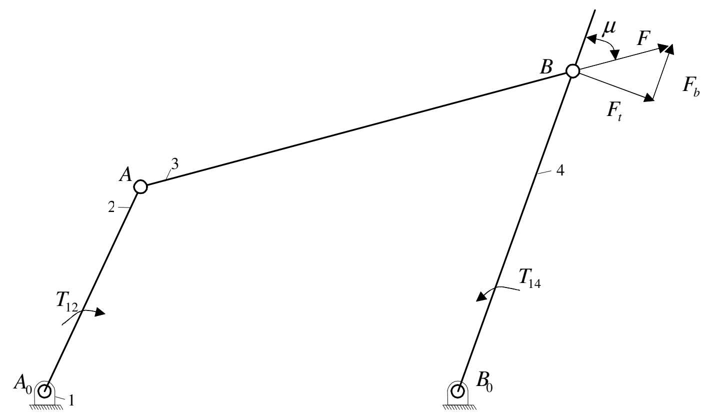


Figure 4.17 Transmission angle of four-bar mechanism

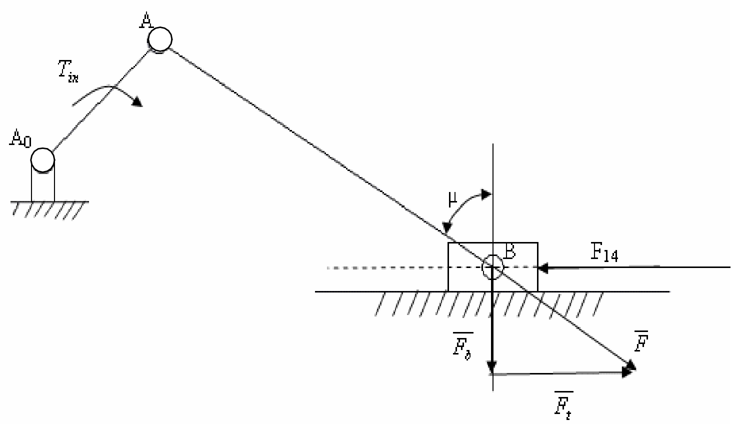


Figure 4.18 Transmission angle of slider-crank mechanism

be sufficient to overcome the system friction. For this reason, the deviation angle $\alpha = |90^\circ - \mu|$ should not be too great. In practice, there is no definite upper limit for α, because the existence of the inertia forces may eliminate the undesirable force relationship that is present under static conditions. In the practical application of a mechanism in order to give a limit to this deviation the following criterion can be given [65, 72, 70, 71];

$$\alpha_{\max} = |90^\circ - \mu|_{\min} \langle 50^\circ \text{ or,} \quad \alpha_{\max} = |90^\circ - \mu|_{\min} < 40^\circ$$

2. Grashof's condition

The motion characteristics of a four-bar mechanism will depend on the ratio of the link length dimensions. Grashof's condition is a very simple relationship which predicts the rotation behavior of a four-bar linkage's inversion based only on the link lengths [65]. The links that are connected to the fixed link can possibly have two different types of motion:

i) The link has a full rotation about the fixed axis, is called the ‘crank’.

ii) The link may oscillate (swing) between two limiting angles, is called the ‘rocker’.

In a four-bar mechanism, three different types of motion are obtained:

i) Both of the links connected to the fixed link can have a full rotation. This type is called as ‘double-crank’ or ‘drag-link’.

ii) Both of the links connected to the fixed link can only oscillate. This type is called as ‘double-rocker’.

iii) One of the links connected to the fixed link can only oscillates while the other has a full rotation. This type is called as ‘crank-rocker’.

The type of motion is a function of the link lengths. Grashof’s theorem gives the criteria for these various conditions as follows:

Let: s - length of shortest link p - length of one remaining link

l - length of longest link q - length of other remaining link

Then:

1. If $l+s<p+q$, the linkage is Grashof and at least one link will be capable of making a full revolution with respect to the ground plane.
2. If $l+s>p+q$, only double-rocker mechanisms are possible.
3. If $l+s=p+q$, the four possible mechanisms will result. However these mechanisms will suffer from a condition known as the change point. The center lines of all the links are collinear at this position.

If all the link lengths are multiply or divided by a constant, the type of four-bar or the angular rotations of the links will not be affected. Therefore the mechanism with same link length ratios will have the same motion characteristic no matter how big or small the mechanism is constructed [65, 70].

3. Optimum value for r/l ratio

In most looms, during the beat-up operation, the sley is operated by the crank and the crank-arm and its motion approximates to a simple harmonic (Figure 4.19). The extent to which it deviates from the simple harmonic motion has practical significance and is governed by the following factors:

(a) the radius of the arc along which the axis of the swordpin reciprocates,

(b) the relative heights of the swordpin and the crankshaft

(c) the length of the crank in relation to that of the crank-arm (r/l)

For a weaving loom the swordpin travels along an arc of the circle centered upon the rocking shaft. Since the radius of the arc is large, the effect is small enough to be neglected here. In practice the relative height of the swordpin and the crankshaft has no practical significance, since the increase could easily be obtained by slightly lengthening the crank. The ratio r/l, where r is the radius of the crank circle and l is the length of the crank arm is called the 'sley-eccentricity ratio'. The larger it is, the greater is the deviation from simple harmonic motion. High sley eccentricity ratio facilitates the passage of weft yarn through shed and increases the effectiveness of the beat-up.

The disadvantages can also be associated with a high sley-eccentricity. A high value implies rapid acceleration and deceleration of the sley around beat-up. Thus it increases the force acting on the swordpins, the crankpins, the cranks, the crankarms,

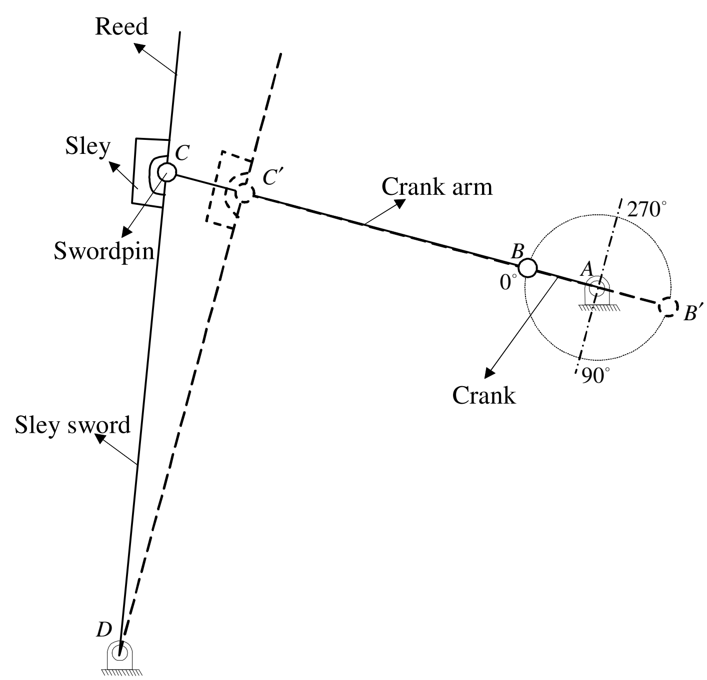


Figure 4.19 Loom sley motion

the crankshaft and their bearing and indirectly on the loom frame. A high sley-eccentricity ratio will therefore demand more robust loom parts and more rigid loom frame in order to prevent excessive vibration and wear. So for a given standard the loom will cost more. For this reason most loom makers tend to avoid eccentricity ratios greater than about 0.3. When the connecting rod is made too short, the system will jam and it is normal for $l > 2r$. Thus according to the given information about the ratio of length of crank to the length of the crankarm (connecting rod), in the practical application of a mechanism following criterion can be followed [60, 73, 74].

$$\frac{crank\ length(a)}{crank\ arm\ length(b)} \leq 0.3, \text{ and } \quad \frac{crank\ arm\ length(b)}{crank\ lenght(a)} > 2$$

4.5.1. Dimensional Synthesis of First Model of Four-link Mechanisms

In this model, the beater is jointed on the tip of the output link of the mechanism. The reciprocating movement of this link B_0C is responsible for the generation of dynamic forces (Figure 4.20). To simplify the motion, it is possible to assume that the point B moves along a straight line rather than an arc. Thus the dimensional synthesis of the mechanism can be made as a slider-crank mechanism.

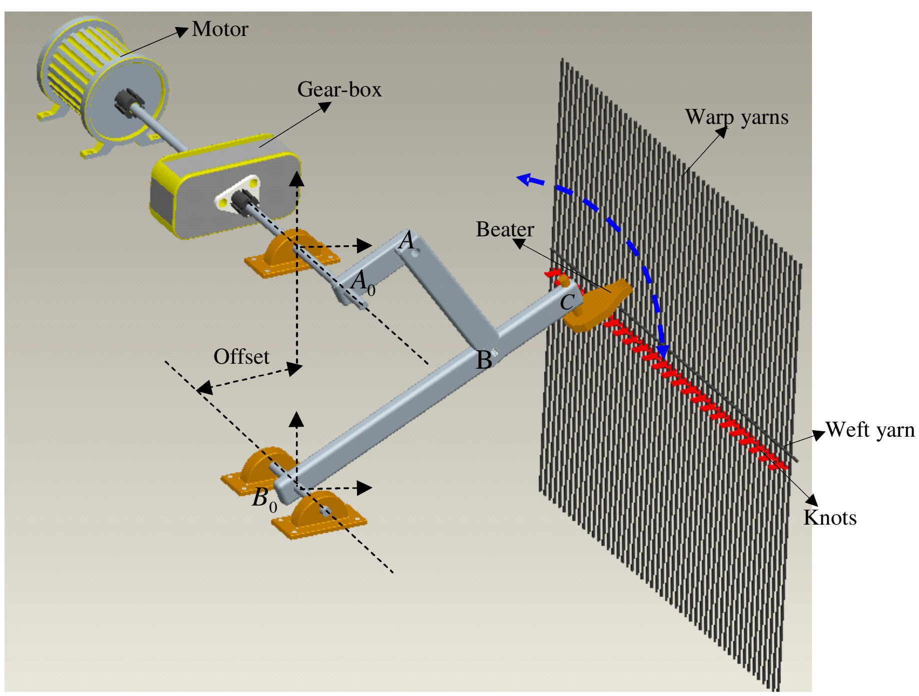


Figure 4.20 First model of four-link mechanisms for slider-crank type solution

a. Analytical method

The length of each link is calculated analytically according to the method that is presented by Söylemez [69].

Dead centers of the slider-crank mechanism

In Figure 4.21 a planar slider-crank mechanism is shown. The link lengths are $a = A_0A_e$ (crank); $b = A_eB_e$ (rocker) and c is the eccentricity (c>0). The crank and the coupler links are collinear as extended $(A_0A_eB_e)$ or folded $(A_0A_fB_f)$ forms (Figure 4.21). The stroke $s = B_eB_f$ is the total displacement of generality, stroke is taken as unity (s=1). Any given stroke can be obtained by the appropriate scaling of the mechanism.

The vector loop (Figure 4.24) equations at the dead centers are

$$\overrightarrow{A_0B_e} + \overrightarrow{B_eA_e} + \overrightarrow{A_eA_0} = 0 \tag{4.1}$$

$$\overrightarrow{A_0B_f} + \overrightarrow{B_fA_f} + \overrightarrow{A_fA_0} = 0 \tag{4.2}$$

Or in complex numbers:

$$-ic + s_e + (b+a)e^{i\phi_1} = 0\ , \tag{4.3}$$

$$-ic + s_f + (b-a)e^{i(\phi_1+\phi-\pi)} = 0\ , \tag{4.4}$$

where $i = \sqrt{-1}$.

Subtracting Eqn. (4.4) from Eqn. (4.3) and noting that $s_e - s_f = s = 1$:

$$1 + (b+a)e^{i\phi_1} + (b-a)e^{i(\phi_1+\phi)} = 0 \tag{4.5}$$

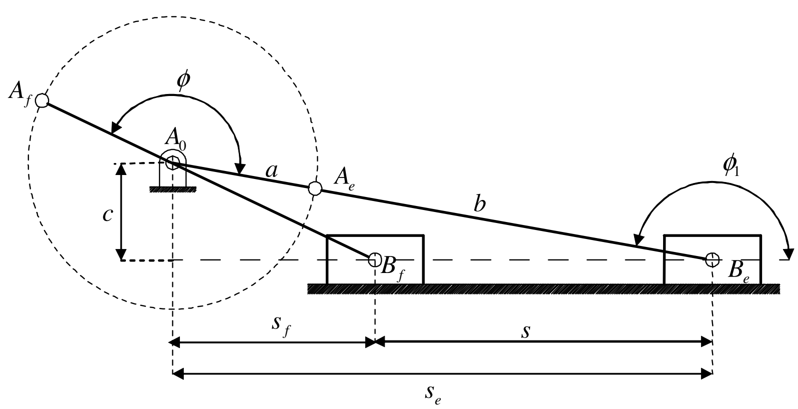


Figure 4.21 Dead center position of slider-crank mechanism

If we let $Z = be^{i\phi_1}$ and $\lambda = a/b$, Eqn.(4.5) can be rewritten in the form

$$Z(1+\lambda)+Z(1-\lambda)e^{i\phi}+1=0 \tag{4.6}$$

Eqn. (4.6) can be solved for Z to yield:

$$Z = \frac{-1}{\lambda(1-e^{i\phi})+(1+e^{i\phi})} \tag{4.7}$$

Where, Z is a circle locus of the crank moving pivot in the extended position (k_a) in the terms of a single parameter λ. The fixed pivot locus is another circle (k_o) which is $Z(1+\lambda)$. These circles are the well known degenerate form of Burmester circle and center points for the relative motion considered. Any line drawn from B_e intersects these circles at A_e and A_0, respectively, yielding the slider-crank mechanism in an extended dead center position. These circles are shown for $\phi = 160^\circ$ in Figure 4.22 $(-\infty < \lambda < +\infty)$.

The eccentricity can be obtained as the imaginary component of the vector $B_eA_0 = B_eA_e + A_eA_0$ which can be written as:

$$2ic = (b+a)e^{i\phi_1} - (b+a)e^{-i\phi_1} \tag{4.8}$$

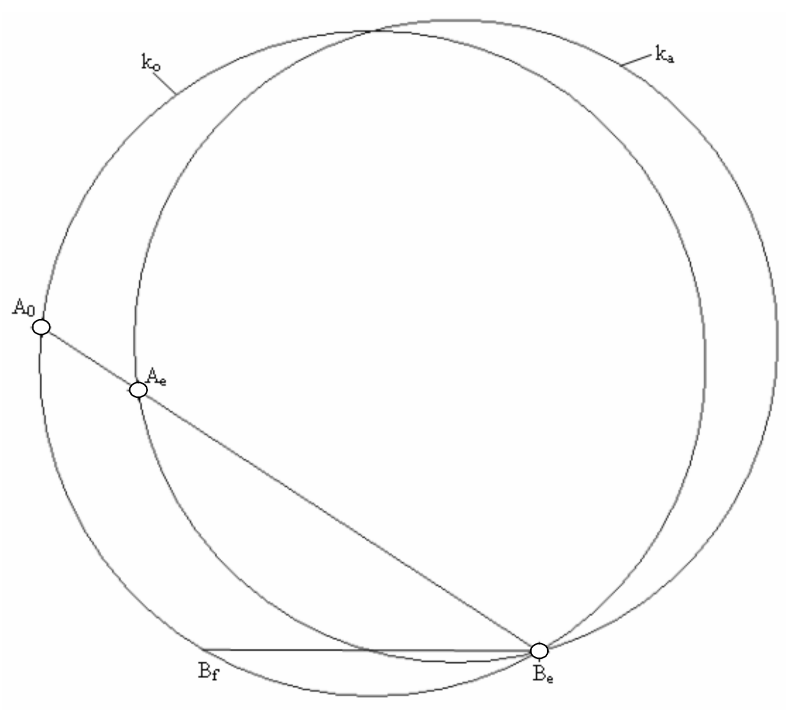


Figure 4.22 Circlepoint and centrepoint locus

or using Z and λ:

$$2ic = Z(1+\lambda) - \bar{Z}(1+\lambda),\tag{4.9}$$

where $\bar{Z}$ is the complex conjugate of Z. substituting the value of Z,

$$c = \frac{1}{2}\frac{(1-\lambda^2)\sin\phi}{[(1+\lambda^2)+(1-\lambda^2)\cos\phi]}\tag{4.10}$$

Using Eqn. (4.7) and noting $b^2 = Z\bar{Z}$, the link lengths can now be expressed as:

$$b^2 = \frac{1}{2}\frac{1}{[(1+\lambda^2)+(1-\lambda^2)\cos(\phi)]} = \frac{1}{4}\frac{1}{\left[\cos^2\frac{\phi}{2}+\lambda^2\sin^2\frac{\phi}{2}\right]},\tag{4.11}$$

$$a^2 = b^2\lambda^2 = \frac{1}{2}\frac{\lambda^2}{[(1+\lambda^2)+(1-\lambda^2)\cos(\phi)]} = \frac{1}{4}\frac{\lambda^2}{\left[\cos^2\frac{\phi}{2}+\lambda^2\sin^2\frac{\phi}{2}\right]}\tag{4.12}$$

Eqns. (4.10)-(4.12) yield a singly infinite set of solutions for the slider-crank mechanisms satisfying a given crank rotation (stroke=1 unit). One can also use the eccentricity, crank or coupler link length as the free parameter to determine the other link lengths.

<u>Ranges of ϕ and λ</u>

According to Grashof's rule, a slider-crank mechanism with a full rotatable crank must satisfy the following two inequalities:

$$b \geq a\tag{4.14}$$

and

$$b - a \geq c.\tag{4.15}$$

Using eqns. (4.10)-(4.12) these conditions yield the ranges for ϕ and λ as:

$$\frac{\pi}{2} \leq \phi \leq \tan^{-1}(\frac{-1}{c}),\tag{4.16}$$

$$\frac{1}{\tan^2\phi/2} \leq \lambda \leq 1\tag{4.17}$$

<u>Transmission angle optimization of the slider-crank mechanism:</u>

The transmission angle μ at any input crank angle ϕ is given by:

$$\mu = \cos^{-1}\left[\frac{c + a\sin\phi}{b}\right]\tag{4.18}$$

The minimum transmission angle is when $\phi = \pi / 2$:

$$\mu_{\min} = \cos^{-1}\left[\frac{c+a}{b}\right] \tag{4.19}$$

Expressing $\mu_{\min}$ in terms of λ and ϕ:

$$\cos\mu_{\min} = \lambda + \frac{1}{\sqrt{2}} \frac{(1-\lambda^2)\sin(\phi)}{\left[(1+\lambda^2)+(1-\lambda^2)\cos(\phi)\right]^{1/2}} \tag{4.20}$$

since λ is a free design parameter, the necessary condition for the minimum transmission angle to be maximum is $d\mu_{\min} / d\lambda = 0$. If the value of λ which makes the derivative equal to zero is $\lambda = \lambda_{opt}$, differentiating Eqn.(4.20) and setting $d\mu_{\min} / d\lambda = 0$ yields:

$$2[\lambda^2{}_{opt}(1-\cos\phi)+(1+\cos\phi)]^3 = \lambda_{opt}^2 \sin^2\phi[\lambda_{opt}^2(1-\cos\phi)+(3+\cos\phi)]^2 \tag{4.21}$$

Eqn. (4.21) is a cubic in terms of λ_{opt}^2. Setting $Q = \lambda_{opt}^2 t^2$, where $t = \tan(\frac{1}{2}\phi)$, the cubic equation in terms of the new parameter Q is:

$$t^2 Q^3 - (1-t^2)Q^2 - (t^4 + t^2 + 1)Q + (1+t^2) = 0 \tag{4.22}$$

The roots of Eq. (4.22) are:

$$Q_1 = -\frac{1}{2} + \frac{1}{2}\sqrt{(5+4t^2)},$$

$$Q_2 = -\frac{1}{2} - \frac{1}{2}\sqrt{(5+4t^2)},$$

$$Q_3 = \frac{1}{t^2}. \tag{4.23}$$

The root, Q_1 is within the range $(1/t^2, t^2)$, satisfies the necessary and sufficient condition for a slider-crank mechanism with optimum transmission angle characteristics. Since Q must be positive, Q_2 is not solution. Corresponding to Q_3, $\lambda = 1/t^2$ the deviation of the minimum transmission angle from 90° is maximized. Therefore:

$$\lambda_{opt}^2 = \frac{1}{2t^2}\left[\sqrt{(5+4t^2)} - 1\right] \tag{4.24}$$

is the unique optimum solution.

Case Study:

The stroke of the output link is taken as unity. Corresponding crank rotation (ϕ) is selected in the range of $90^\circ \le \phi \le 180^\circ$ and dimensions of the mechanism are determined for each ϕ value. The crank rotation angle for optimum transmission angle value is determined. The results of the calculations for each angle in the range of $90^\circ \le \phi \le 180^\circ$ are given in the Table 4.2. The λ (a/b) value must be near to 0.3 and it must not exceed 0.5 for the optimum mechanism, the maximum deviation of the $\mu_{\min}$ from 90^0 must not exceed 50^0. According to these criterias the optimum transmission angle is obtained at $\phi = 169^\circ (\lambda = 0.3)$ and $\phi = 175^\circ (\lambda = 0.2)$, the worst result is obtained at $\phi = 90^\circ$.

Table 4.2 Results of the slider-crank mechanism

ϕ	a	b	c	μmin(degree)	deviation	λ
90	0.500000	0.500000	0.000000	0.00	90.00	1.0000
91	0.497144	0.502940	0.005796	0.07	89.93	0.9885
92	0.494393	0.505944	0.011548	0.20	89.80	0.9772
93	0.491742	0.509013	0.017260	0.37	89.63	0.9661
94	0.489190	0.512149	0.022933	0.57	89.43	0.9552
95	0.486735	0.515353	0.028569	0.79	89.21	0.9445
96	0.484372	0.518628	0.034170	1.04	88.96	0.9339
97	0.482101	0.521975	0.039738	1.31	88.69	0.9236
98	0.479918	0.525396	0.045275	1.59	88.41	0.9134
99	0.477822	0.528894	0.050782	1.90	88.10	0.9034
100	0.475811	0.532470	0.056261	2.22	87.78	0.8936
101	0.473882	0.536127	0.061713	2.55	87.45	0.8839
102	0.472033	0.539867	0.067141	2.90	87.10	0.8744
103	0.470263	0.543692	0.072545	3.27	86.73	0.8649
104	0.468570	0.547605	0.077928	3.64	86.36	0.8557
105	0.466953	0.551609	0.083289	4.03	85.97	0.8465
106	0.465409	0.555706	0.088632	4.44	85.56	0.8375
107	0.463937	0.559900	0.093957	4.85	85.15	0.8286
108	0.462536	0.564193	0.099265	5.28	84.72	0.8198
109	0.461205	0.568588	0.104558	5.71	84.29	0.8111
110	0.459941	0.573090	0.109837	6.16	83.84	0.8026
111	0.458744	0.577701	0.115103	6.62	83.38	0.7941
112	0.457612	0.582425	0.120358	7.09	82.91	0.7857
113	0.456545	0.587266	0.125602	7.57	82.43	0.7774
114	0.455540	0.592229	0.130836	8.06	81.94	0.7692
115	0.454598	0.597317	0.136063	8.56	81.44	0.7611
116	0.453716	0.602536	0.141282	9.07	80.93	0.7530
117	0.452895	0.607889	0.146495	9.59	80.41	0.7450

Table 4.2 (continue) Results of the slider-crank mechanism

ϕ	a	b	c	μmin(degree)	deviation	λ
118	0.452133	0.613382	0.151702	10.12	79.88	0.7371
119	0.451428	0.619021	0.156906	10.66	79.34	0.7293
120	0.450781	0.624811	0.162106	11.21	78.79	0.7215
121	0.450191	0.630757	0.167304	11.77	78.23	0.7137
122	0.449656	0.636868	0.172501	12.34	77.66	0.7060
123	0.449176	0.643148	0.177698	12.92	77.08	0.6984
124	0.448750	0.649606	0.182895	13.50	76.50	0.6908
125	0.448377	0.656249	0.188094	14.10	75.90	0.6832
126	0.448058	0.663085	0.193295	14.71	75.29	0.6757
127	0.447790	0.670122	0.198499	15.33	74.67	0.6682
128	0.447574	0.677371	0.203708	15.95	74.05	0.6608
129	0.447409	0.684840	0.208921	16.59	73.41	0.6533
130	0.447294	0.692541	0.214141	17.24	72.76	0.6459
131	0.447230	0.700484	0.219367	17.89	72.11	0.6385
132	0.447214	0.708682	0.224600	18.56	71.44	0.6311
133	0.447248	0.717147	0.229842	19.24	70.76	0.6236
134	0.447330	0.725894	0.235093	19.93	70.07	0.6162
135	0.447461	0.734937	0.240354	20.63	69.37	0.6088
136	0.447639	0.744293	0.245626	21.34	68.66	0.6014
137	0.447864	0.753979	0.250909	22.06	67.94	0.5940
138	0.448136	0.764014	0.256204	22.80	67.20	0.5866
139	0.448455	0.774419	0.261513	23.54	66.46	0.5791
140	0.448821	0.785215	0.266835	24.30	65.70	0.5716
141	0.449232	0.796427	0.272173	25.07	64.93	0.5641
142	0.449689	0.808082	0.277525	25.85	64.15	0.5565
143	0.450192	0.820208	0.282894	26.65	63.35	0.5489
144	0.450740	0.832837	0.288280	27.46	62.54	0.5412
145	0.451333	0.846003	0.293683	28.28	61.72	0.5335
146	0.451970	0.859747	0.299105	29.12	60.88	0.5257
147	0.452653	0.874108	0.304546	29.97	60.03	0.5178
148	0.453380	0.889136	0.310007	30.84	59.16	0.5099
149	0.454151	0.904880	0.315489	31.73	58.27	0.5019
150	0.454967	0.921401	0.320993	32.63	57.37	0.4938
151	0.455827	0.938763	0.326518	33.55	56.45	0.4856
152	0.456731	0.957039	0.332067	34.49	55.51	0.4772
153	0.457679	0.976312	0.337640	35.45	54.55	0.4688
154	0.458671	0.996675	0.343237	36.43	53.57	0.4602
155	0.459706	1.018234	0.348859	37.43	52.57	0.4515
156	0.460786	1.041112	0.354508	38.45	51.55	0.4426
157	0.461909	1.065447	0.360183	39.50	50.50	0.4335
158	0.463076	1.091403	0.365886	40.58	49.42	0.4243
159	0.464287	1.119167	0.371617	41.68	48.32	0.4149
160	0.465542	1.148960	0.377378	42.81	47.19	0.4052
161	0.466840	1.181043	0.383169	43.97	46.03	0.3953
162	0.468183	1.215728	0.388990	45.16	44.84	0.3851
163	0.469569	1.253387	0.394844	46.40	43.60	0.3746
164	0.471000	1.294476	0.400729	47.67	42.33	0.3639
165	0.472474	1.339552	0.406648	48.98	41.02	0.3527

Table 4.2 (continue) Results of the slider-crank mechanism

ϕ	a	b	c	μmin(degree)	deviation	λ
166	0.473993	1.389310	0.412601	50.35	39.65	0.3412
167	0.475556	1.444629	0.418589	51.76	38.24	0.3292
168	0.477164	1.506638	0.424612	53.23	36.77	0.3167
169	**0.478817**	**1.576817**	**0.430672**	**54.78**	**35.22**	**0.3037**
170	0.480514	1.657151	0.436770	56.39	33.61	0.2900
171	0.482256	1.750378	0.442906	58.09	31.91	0.2755
172	0.484044	1.860397	0.449081	59.90	30.10	0.2602
173	0.485877	1.992990	0.455296	61.82	28.18	0.2438
174	0.487756	2.157189	0.461552	63.89	26.11	0.2261
175	0.489680	2.368071	0.467850	66.15	23.85	0.2068
176	0.491651	2.653209	0.474190	68.65	21.35	0.1853
177	0.493668	3.070213	0.480575	71.50	18.50	0.1608
178	0.495731	3.768318	0.487004	74.88	15.12	0.1316
179	0.497842	5.340742	0.493479	79.30	10.70	0.0932
180	0.500000	34912.708656	-0.500000	90.00	0.00	0.0000

In order to obtain optimum link lengths that are proper for the hand loom, the trajectory given in Figure 4.16 is scaled by an assumed value of 1/3.5. The length of the output link of the mechanism (line DP_1 in the Figure 4.16) is measured as 30.1 cm. The stroke of the point B (Figure 4.20) during a beat-up operation is assumed as 12 cm. So the link lengths of the solution mechanism are given in Table 4.3 which are calculated for the stroke is 12 cm.

In Figure 4.23 the motion of the each link of the solution mechanism during a beat-up operation is represented. The crank link (a) makes a full rotation around the fixed point A_0. The point B moves from point B_f to point B_e between two dead positions of the mechanism. During the motion of the B_0B link from position 1 to position 2, the point B moves approximately on a straight line from B_f to B_e. The distance between

Table 4.3 Length of each link of the slider-crank mechanism

s (stroke) = 12 cm			
	$\lambda = 0.3$	$\lambda = 0.2$	$\lambda = 1$
ϕ (degree)	169	175	90
$\mu_{\min}$ (degree)	54.78	66.15	0
a (cm)	5.745804	5.87616	6
b (cm)	18.921804	28.416852	6
c (cm)	5.168064	5.6142	0.000000

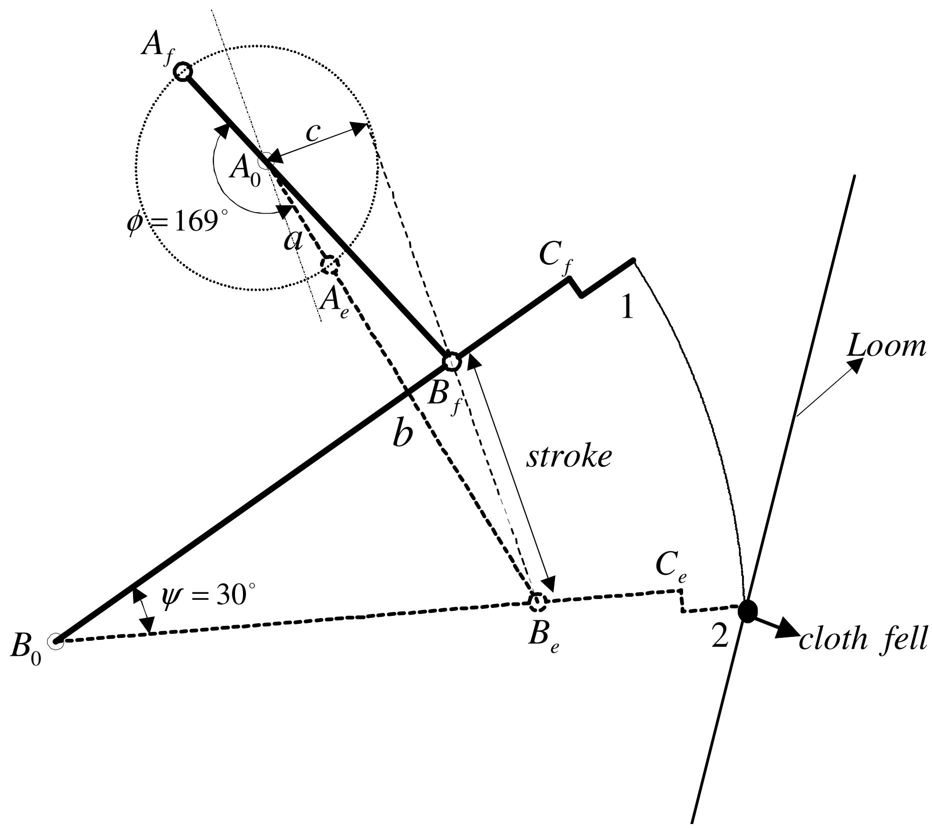


Figure 4.23 Dimensions and motion of first model of four-link mechanisms

point B_f and point B_e is called as the stroke of the mechanism. The angle between A_f and A_e is corresponding rotation angle (ϕ) of the crank between dead centers of the mechanism. The angle between two dead points of the B_0B link is its swing angle (ψ). The eccentricity (c) of the mechanism is the distance between the crank rotation axis and the sliding axis of the point *B*.

b) Graphical method

The graphical solution of a slider crank mechanism is made according to following procedure;

1. Locate B_1 and B_2 on a straight line. B_1B_2 is the stroke of the mechanism.
2. Draw a line L that makes an angle $\phi - 90^\circ$ with respect to B_1B_2.
3. Draw a line perpendicular to bisector of B_1B_2.
4. Point m is the intersection of line L with perpendicular bisector.
5. A circle k_o that passes through B_1 and B_2 with the centre m is drawn. k_0 is the locus of A_0, the fixed pivot of the crank.

6. The perpendicular bisector to B_1B_2 will intersect k_0 at point N. Draw circle k_a with NB_2 as its diameter. k_a is the locus of A_2, the location of moving pivot of the crank as its extended dead centre.
7. Draw a line from B_2 that intersect k_a at A_2 and k_0 at A_0.

This will give the proportions of a slider crank mechanism that has a given stroke (s) and corresponding crank rotation (ϕ). As shown in Figure 4.24, many lines that intersect k_a and k_b at different angle with respect to B_1B_2 can be drawn and so many solutions that satisfy the required stroke and corresponding crank rotation can be obtained. In order to find the slider-crank mechanism that has the least deviation of transmission angle for the required crank rotation, Chart-1 and Chart-2 can be used. In Chart-1[70], the slider-crank link lengths (a, b, c), λ_{opt} and optimum minimum transmissions angle ($\max \mu_{\min}$) values as function of crank rotation in between dead centers are given. Since the result of b/c value for the selected crank rotation can not be found on Chart-1 due to the dimensions of the links, Chart-2 is used. In Chart-2

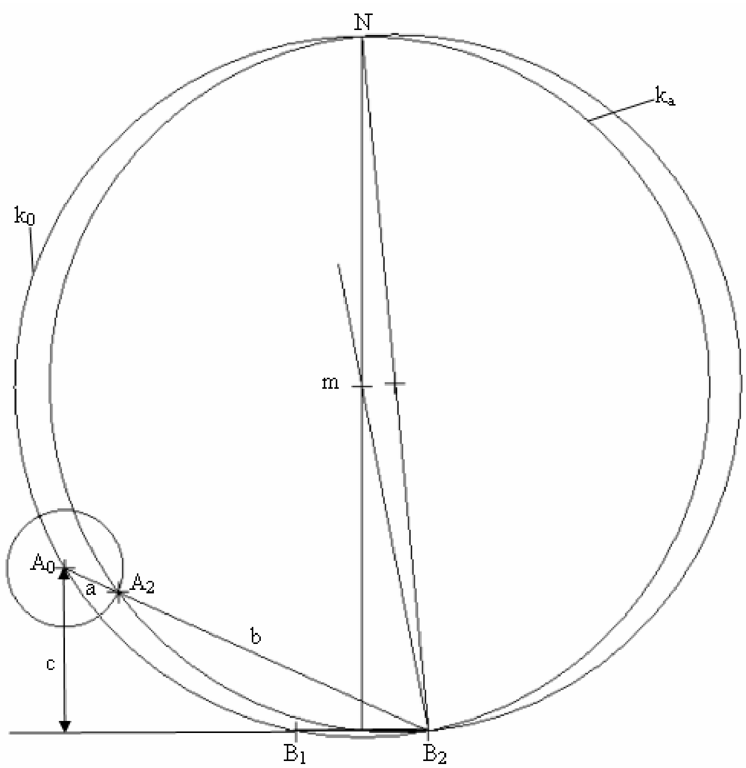


Figure 4.24 Graphical solution of slider-crank mechanism for $\phi = 169^\circ$

Table 4.4 Length ratios for $\phi = 169^\circ$

Length ratios for $\phi = 169^\circ$
$\lambda = 0.29$
$a/s = 0.48$
$b/s = 1.58$
$c/s = 0.43$
$\max \mu_{\min} = 55^o$

Table 4.5 Link lengths for s=12 cm

Link lengths of the mechanism for s=12 cm
$a = 5.76$
$b = 18.96$
$c = 5.16$

[71], connecting rod to stroke ratio, b/c, as a function of corresponding eccentricity (c) and crank rotation between dead centers, ϕ. The link lengths and minimum transmission angle value for $\phi = 169^\circ$ are found from Chart-1 and Chart-2 as given in Table 4.4. When the link lengths of the mechanism are calculated for s= 12 cm, the following results given Table 4.5 are obtained.

If these results in Table 4.5 are compared with the results given in Table 4.3, it is seen that they coincides. Thus link lengths of the first model of the four-link mechanisms that satisfies the functional requirements are determined using graphical and analytical methods.

4.5.2 Dimensional Synthesis of Fourth Model of Four-link Mechanisms

In crank-rocker mechanism, the rocker oscillates between two limiting angles. In general situation the crank is input and the rocker is output links. The position of the mechanism when the rocker is at a limit position is called dead center position of four-bar. The oscillation of rocker between the dead center positions and measured from the extended dead centre to the folded dead centre position is called 'swing angle ψ'. There is also a corresponding crank rotation ϕ for this swing angle (Figure 4.25). As the input link (crank) makes a full rotation, the output link (rocker) performs a swing motion. The synthesis of this mechanism is made according to the input angle of the crank and corresponding output angle of the rocker [65, 66, 70,71].

The fourth model of four-link mechanisms is a crank-rocker type mechanism (Figure 4.26). The synthesis of the mechanism is made with both graph-analytical method and graphical method.

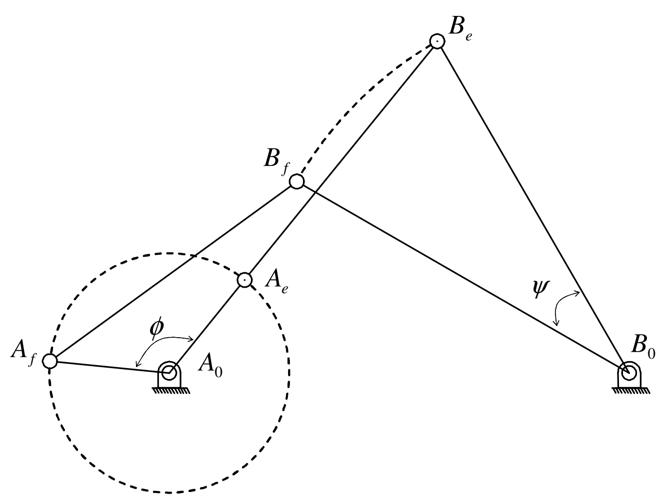


Figure 4.25 Dead centers of the crank-rocker mechanism

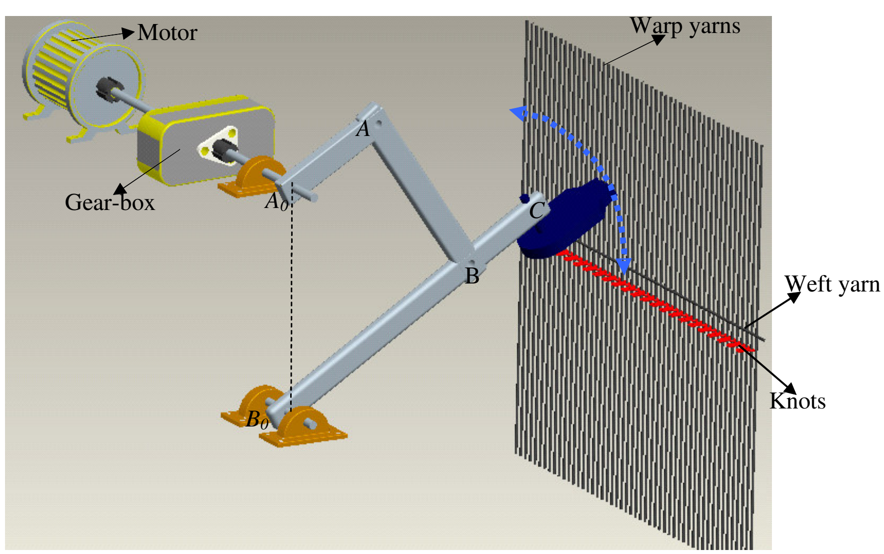


Figure 4.26 Fourth model of four-link mechanisms for crank-rocker type solution

a. Graph-Analytical Method

Figures 4.27 and 4.28 represent the graphical method of synthesizing the crank-rocker mechanisms for different ϕ values, where a, b, c and d are the link lengths. Usually d, the length of the fixed link, is taken equal to unity for calculation of the other links easily. Then a, b, and c will represent the length ratios.

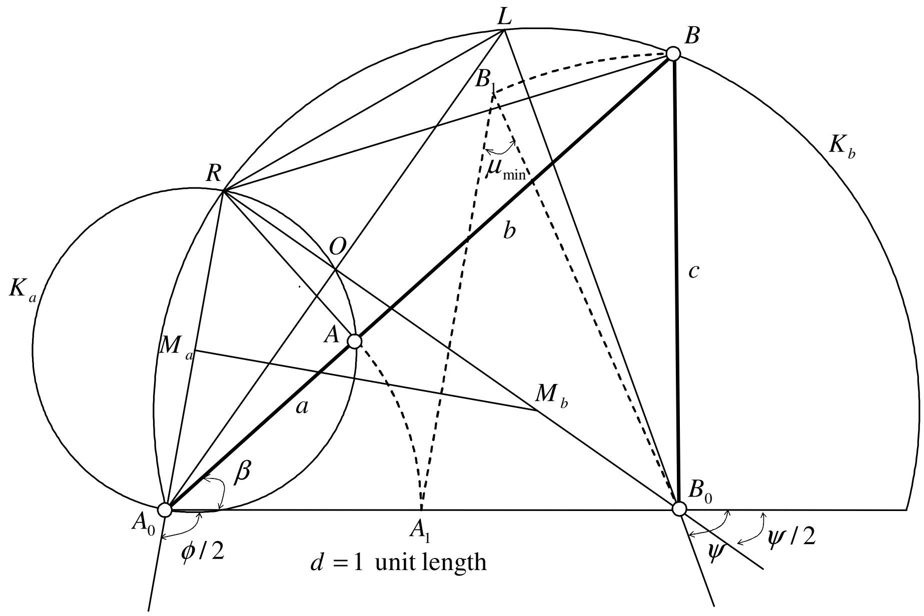


Figure 4.27 Graphical method of synthesizing the crank-rocker mechanism

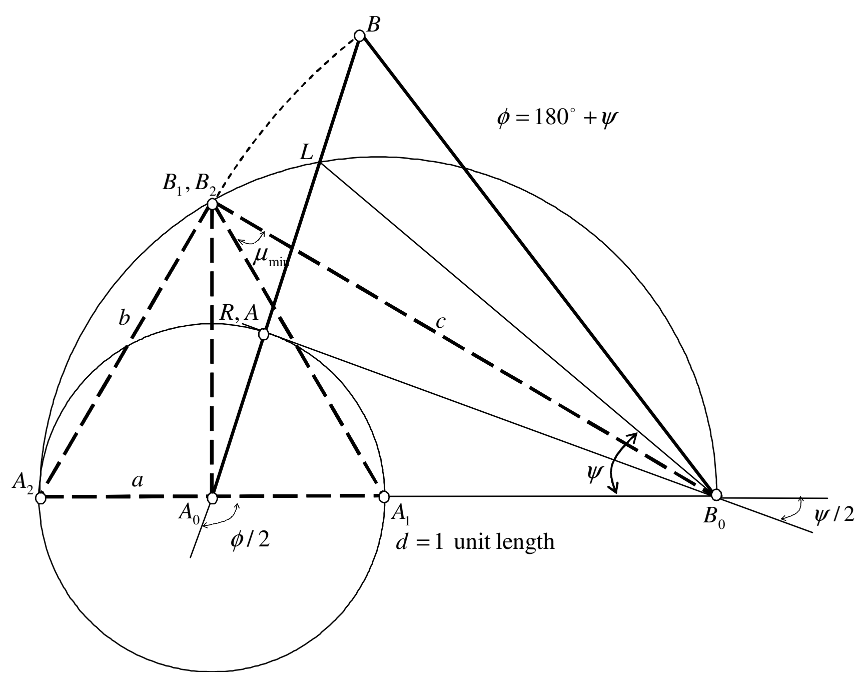


Figure 4.28 Graphical method of synthesizing of the crank-rocker mechanism for $\phi = 180^\circ + \psi$

Refering to Figure 4.27;

$$\angle A_0RB_0 = \angle ARB = \frac{\phi}{2} - \frac{\psi}{2}$$

$$\angle ROA_0 = \angle RAA_0 = \frac{\pi}{2}$$

Therefore;

$$A_0O = \sin\frac{\psi}{2}$$

$$\frac{b}{RB} = \sin\left(\frac{\phi}{2} - \frac{\psi}{2}\right)$$

$$\frac{a}{A_0R} = -\cos\left(\frac{\phi}{2} + \beta\right)$$

$$\frac{AR}{A_0R} = \sin\left(\frac{\phi}{2} + \beta\right)$$

$$\frac{AR}{RB} = \cos\left(\frac{\phi}{2} - \frac{\psi}{2}\right) \tag{4.25}$$

From similar triangles A_0RO and RAB

$$\frac{A_0O}{A_0R} = \frac{b}{RB} \tag{4.26}$$

Simplifying equation (4.25) and (4.26)

$$a = -\frac{\sin\alpha_1 \cos\beta_1}{\sin\alpha_2}$$

$$b = \frac{\sin\alpha_1 \sin\beta_1}{\cos\alpha_2} \tag{4.27}$$

Where;

$$\alpha_1 = \frac{\psi}{2},\ \alpha_2 = \frac{\phi}{2} - \frac{\psi}{2},\ \beta_1 = \frac{\phi}{2} + \beta \tag{4.28}$$

Applying the cosine law to the triangle A_0BB_0

$$c^2 = (a+b)^2 + 1 - 2(a+b)\cos\beta \tag{4.29}$$

Substitution for a and b in to equation (4.29) and simplification results in the following expression

$$c^2 = K_1 + K_2\cos(\alpha_3 + 2\beta) + \frac{K_2^2}{2}\cos(2\alpha_3 + 2\beta) \tag{4.30}$$

Where;

$$K_1 = 1 + K_2 \cos\alpha_3 + \frac{K_2^2}{2}$$

$$K_2 = \frac{2\sin\alpha_1}{\sin 2\alpha_2}$$

$$\alpha_3 = \phi - \frac{\psi}{2} \tag{4.31}$$

<u>*Case* I. $\phi > 180^\circ$</u>

Grashof's criterion for the crank-rocker mechanism is

$$(l + s) \le (p + q)$$

Where l is the length of the longest link, s is the length of the shortest link –the crank-. p and q represent the lengths of the reaming two links.

$< LA_0B_0 \ge \beta \ge 0$ or,

$$\left(\frac{\pi}{2} - \frac{\psi}{2}\right) \ge \beta \ge 0 \tag{4.32}$$

Eqn. (4.32) gives the upper and lower bounds on β

If $\phi > 180^\circ$, the transmission angle is a minimum when the crank superimposes the fixed link as shown in Figure 4.27.

$$\cos\mu_{\min} = \frac{b^2 + c^2 - (1-a)^2}{2bc} \tag{4.33}$$

The desired linkage is one for which $(90 - \mu_{\min})$ is a minimum or $\cos\mu_{\min}$ has the smallest positive value. This can be achieved by minimizing $\cos\mu_{\min}$ with respect to β. The link length ratios a, b and c may then be obtained by using equation (4.27) and (4.30) and the mechanism can be synthesized by suitably assuming the length of the fixed link.

<u>*Case* II. $\phi < 180^\circ$</u>

The Grashof's condition for the crank-rocker mechanism is satisfied if

$$\left(\frac{\pi}{2} - \frac{\psi}{2}\right) \ge \beta \ge (\pi - \phi). \tag{4.34}$$

When $\phi < 180^\circ$, the transmission angle during the complete rotation of the crank is minimum when the crank is in line with the fixed link and away from it. Then,

$$\cos\mu_{\min} = \frac{b^2 + c^2 - (1+a)^2}{2bc} \tag{4.35}$$

The desired value of β may be obtained minimizing $\cos\mu_{\min}$ with respect to β subjected to condition given by equation (4.34).

<u>*Case* III. $\phi = 180^\circ + \psi$</u>

This situation deserves special attention, because equation (4.27) does not hold good in this case. Referring to Figure 4.28 of all possible linkages, the link length ratios in the mechanism, in which the transmission angle $A_2B_2B_0$ is 90°, are such that the minimum transmission angle $A_1B_1B_0$ is nearest to 90°.

In this case;

$$\beta = \pi - \frac{\phi}{2} \tag{4.36}$$

$$a = \cos\beta = -\cos\frac{\phi}{2} \tag{4.37}$$

From similar triangles $A_2B_2B_0$ and $A_1B_1A_0$

$$\frac{a}{b} = \frac{b}{(1+a)}$$

Therefore,

$$b = \sqrt{[a(1+a)]} \tag{4.38}$$

Also

$$(1+a)^2 = b^2 + c^2 \text{ or,}$$

$$c = \sqrt{(1+a)}\text{, and } \cos\mu_{\min} = \frac{2\sqrt{a}}{(1+a)} \tag{4.39}$$

<u>*Case* IV. $\phi = 180^\circ$</u>

The centric crank-rocker mechanism, defined by $\phi = 180^\circ$, has unity ratio. It also has equal transmission angle $\mu_{\min}$ at both the position when the crank is line with the fixed link.

Therefore,

$$(1-a)^2 = b^2 + c^2 - 2bc\cos\mu_{\min}\text{, and } (1+a)^2 = b^2 + c^2 + 2bc\cos\mu_{\min}$$

Elimination of $\cos\mu_{\min}$ results in

$$c^2 = 1 + a^2 - b^2 \tag{4.40}$$

Since $\phi = 180^\circ$, equation (4.27) gives

$a = \tan\dfrac{\psi}{2}\sin\beta$ and,

$$b = \cos\beta \tag{4.41}$$

Simplifying equation (4.40) by using equation (4.41)

$$c = \sec\frac{\psi}{2}\sin\beta \tag{4.42}$$

Substituting for a, b and c in equation (4.33) or (4.35) and simplifying them

$$\cos\mu_{\min} = \frac{a}{bc} = \frac{\sin(\psi/2)}{\cos\beta} \tag{4.43}$$

Equation (4.43) gives a direct relation between $\mu_{\min}$ and β. Any value of angle β (subjected to the condition given by equation (4.32) or (4.34)) corresponds to a mechanism for which $\mu_{\min}$ may be calculated from equation (4.43). Also for any desired $\mu_{\min}$ the β may be obtained. The link length ratios a, b, c may then be obtained by using equation (4.41) and (4.42).

Case study 1:

$\phi = 120^\circ$, $\psi = 30^\circ$

$75^\circ \geq \beta \geq 60^\circ$. The upper and lower bounds on the β may be obtained by using equation (4.34). The calculation result for $\phi = 120^\circ$ is given Table 4.6.

Table 4.6 Results for $\phi = 120^\circ$

β	a	b	c	μ_{min}(degree)
75	0.25881905	0.258819	1	0
74	0.25426261	0.263297	0.991237	2.92
73	0.24962873	0.267694	0.982406	3.84
72	0.2449188	0.27201	0.973517	4.33
71	0.24013427	0.276243	0.964579	4.59
70	0.23527659	0.280392	0.955602	4.66
69	0.23034725	0.284455	0.946596	4.60
68	0.22534774	0.288432	0.937571	4.42
67	0.22027959	0.292321	0.928537	4.13
66	0.21514433	0.296121	0.919506	3.76
65	0.20994355	0.29983	0.910489	3.31
64	0.20467881	0.303449	0.901497	2.78
63	0.19935172	0.306975	0.892542	2.18
62	0.19396391	0.310407	0.883637	1.51
61	0.18851702	0.313745	0.874794	0.79
60	0.1830127	0.316987	0.866025	0

Case study 2:

$\phi = 160^\circ$, $\psi = 30^\circ$

$75^\circ \geq \beta \geq 60^\circ$. The upper and lower bounds on the β may be obtained by using equation (4.34).The calculation results for $\phi = 160^\circ$ are given Table 4.7.

Table 4.7 Results for $\phi = 160^\circ$

β	a	b	c	μ_{min}(degree)
75	0.25881905	0.258819	1	0
74	0.25667331	0.268466	0.993114	12.36
73	0.25444938	0.278032	0.985988	17.05
72	0.25214795	0.287513	0.978624	20.38
71	0.24976971	0.296906	0.971027	22.97
70	0.24731539	0.306209	0.963201	25.08
69	0.24478574	0.315419	0.955149	26.83
68	0.24218152	0.324532	0.946876	28.30
67	0.23950353	0.333547	0.938387	29.55
66	0.23675259	0.34246	0.929684	30.62
65	0.23392952	0.351269	0.920775	31.53
64	0.23103521	0.35997	0.911662	32.31
63	0.22807051	0.368562	0.902352	32.96
62	0.22503634	0.377042	0.89285	33.51
61	0.22193363	0.385407	0.883161	33.96
60	0.21876331	0.393655	0.873291	34.33
59	0.21552635	0.401782	0.863247	34.62
58	0.21222375	0.409788	0.853034	34.83
57	0.20885649	0.417668	0.842659	34.97
56	0.20542562	0.425421	0.832128	35.05
55	0.20193217	0.433045	0.82145	35.06
54	0.19837722	0.440537	0.810631	35.02
53	0.19476183	0.447894	0.799679	34.92
52	0.19108712	0.455115	0.788602	34.77
51	0.1873542	0.462198	0.777409	34.56
50	0.18356421	0.469139	0.766109	34.30
49	0.17971831	0.475938	0.75471	33.99
48	0.17581766	0.482592	0.743222	33.63
47	0.17186346	0.489099	0.731656	33.22
46	0.1678569	0.495457	0.720022	32.76
45	0.16379922	0.501663	0.708332	32.25
44	0.15969164	0.507718	0.696597	31.69
43	0.15553541	0.513617	0.684829	31.08
42	0.15133181	0.51936	0.673042	30.42
41	0.14708211	0.524945	0.661249	29.71
40	0.14278761	0.53037	0.649464	28.94
39	0.13844961	0.535633	0.637704	28.12
38	0.13406944	0.540733	0.625983	27.25
37	0.12964844	0.545668	0.61432	26.31
36	0.12518794	0.550438	0.602731	25.32
35	0.1206893	0.555039	0.591236	24.27
34	0.11615391	0.559472	0.579855	23.15
33	0.11158313	0.563734	0.568609	21.98
32	0.10697836	0.567824	0.55752	20.73
31	0.10234101	0.571741	0.546612	19.42

Table 4.7 (continue) Results for $\phi = 160^{\circ}$

β	a	b	c	μ_{min}(degree)
30	0.09767248	0.575485	0.535909	18.04
29	0.0929742	0.579053	0.525437	16.58
28	0.0882476	0.582444	0.515224	15.05
27	0.08349411	0.585658	0.505299	13.45
26	0.0787152	0.588694	0.49569	11.77
25	0.07391231	0.59155	0.486428	10.00
24	0.0690869	0.594227	0.477547	8.16
23	0.06424045	0.596722	0.469077	6.24
22	0.05937443	0.599035	0.461053	4.24
21	0.05449032	0.601166	0.453508	2.16
20	0.04958962	0.603114	0.446476	0.00

Case study 3:

$\phi = 180^{\circ}, \psi = 30^{\circ}$

$75^{\circ} \geq \beta \geq 0^{\circ}$. The upper and lower bounds on β may be obtained by using equation (4.34). The calculation results for $\phi = 180^{\circ}$ are given Table 4.8.

Table 4.8 Results for $\phi = 180^{\circ}$

β	a	b	c	μ_{min}(degree)
75	0.258819	0.258819	1	0
74	0.257569	0.275637	0.995171	20.12
73	0.256241	0.292372	0.99004	27.72
72	0.254835	0.309017	0.984606	33.12
71	0.253351	0.325568	0.978873	37.35
70	0.25179	0.34202	0.972841	40.82
69	0.250152	0.358368	0.966514	43.76
68	0.248438	0.374607	0.959891	46.30
67	0.246649	0.390731	0.952977	48.52
66	0.244784	0.406737	0.945772	50.48
65	0.242844	0.422618	0.938279	52.24
64	0.240831	0.438371	0.9305	53.81
63	0.238744	0.45399	0.922438	55.24
62	0.236585	0.469472	0.914095	56.54
61	0.234354	0.48481	0.905473	57.73
60	0.232051	0.5	0.896575	58.83
59	0.229677	0.515038	0.887405	59.83
58	0.227234	0.529919	0.877964	60.76
57	0.224721	0.544639	0.868256	61.63
56	0.22214	0.559193	0.858283	62.43
55	0.219491	0.573576	0.848049	63.18
54	0.216775	0.587785	0.837556	63.88
53	0.213994	0.601815	0.826808	64.53
52	0.211147	0.615661	0.815809	65.14
51	0.208236	0.62932	0.804561	65.72
50	0.205261	0.642788	0.793068	66.26
49	0.202224	0.656059	0.781333	66.76
48	**0.199125**	**0.669131**	**0.76936**	**67.24**

Table 4.8 (continue) Results for $\phi = 180^{\circ}$

β	a	b	c	μ_{min}(degree)
47	0.195966	0.681998	0.757153	67.70
46	0.192747	0.694658	0.744715	68.12
45	0.189469	0.707107	0.732051	68.53
44	0.186133	0.71934	0.719163	68.91
43	0.182741	0.731354	0.706057	69.27
42	0.179293	0.743145	0.692735	69.62
41	0.17579	0.75471	0.679202	69.94
40	0.172234	0.766044	0.665463	70.25
39	0.168626	0.777146	0.65152	70.55
38	0.164966	0.788011	0.63738	70.83
37	0.161256	0.798636	0.623045	71.09
36	0.157497	0.809017	0.60852	71.34
35	0.153689	0.819152	0.59381	71.58
34	0.149835	0.829038	0.578919	71.81
33	0.145936	0.838671	0.563852	72.02
32	0.141991	0.848048	0.548613	72.23
31	0.138004	0.857167	0.533207	72.43
30	0.133975	0.866025	0.517638	72.61
29	0.129904	0.87462	0.501912	72.79
28	0.125795	0.882948	0.486033	72.95
27	0.121646	0.891007	0.470006	73.11
26	0.117461	0.898794	0.453835	73.26
25	0.11324	0.906308	0.437527	73.41
24	0.108985	0.913545	0.421085	73.54
23	0.104696	0.920505	0.404515	73.67
22	0.100376	0.927184	0.387821	73.79
21	0.096024	0.93358	0.37101	73.90
20	0.091644	0.939693	0.354085	74.01
19	0.087236	0.945519	0.337053	74.11
18	0.082801	0.951057	0.319918	74.21
17	0.078341	0.956305	0.302685	74.30
16	0.073857	0.961262	0.285361	74.38
15	0.06935	0.965926	0.267949	74.46
14	0.064823	0.970296	0.250456	74.53
13	0.060275	0.97437	0.232886	74.60
12	0.05571	0.978148	0.215246	74.66
11	0.051127	0.981627	0.19754	74.71
10	0.046529	0.984808	0.179774	74.76
9	0.041916	0.987688	0.161953	74.81
8	0.037291	0.990268	0.144083	74.85
7	0.032655	0.992546	0.126168	74.88
6	0.028008	0.994522	0.108216	74.92
5	0.023353	0.996195	0.09023	74.94
4	0.018691	0.997564	0.072217	74.96
3	0.014023	0.99863	0.054182	74.98
2	0.009351	0.999391	0.036131	74.99
1	0.004676	0.999848	0.018068	75.00
0	0	1	0	∞

Case study 4:

$\phi = 200^\circ, \psi = 30^\circ$

$75^\circ \geq \beta \geq 0^\circ$. The upper and lower bounds on the β may be obtained by using equation (4.32). The calculation results for $\phi = 200^\circ$ are given Table 4.9.

Table 4.9 Results for $\phi = 200^\circ$

β	a	b	c	μ_{min}(degree)
75	0.258819	0.258819	1	0
74	0.258384	0.310409	1.00497	26.91
73	0.257871	0.361905	1.010798	34.64
72	0.257279	0.413291	1.017462	38.91
71	0.256609	0.46455	1.024938	41.46
70	0.255861	0.515668	1.033199	42.97
69	0.255034	0.566629	1.042217	43.81
68	0.25413	0.617418	1.05196	44.18
67	0.253149	0.668018	1.062399	44.23
66	0.25209	0.718415	1.073499	44.02
65	0.250955	0.768593	1.085228	43.63
64	0.249743	0.818537	1.09755	43.11
63	0.248455	0.868232	1.110432	42.48
62	0.247092	0.917662	1.123839	41.77
61	0.245653	0.966812	1.137736	41.01
60	0.244139	1.015668	1.152088	40.20
59	0.242551	1.064215	1.166863	39.35
58	0.240889	1.112438	1.182026	38.49
57	0.239154	1.160321	1.197545	37.61
56	0.237346	1.207851	1.213387	36.72
55	0.235466	1.255014	1.229521	35.82
54	0.233514	1.301794	1.245916	34.93
53	0.23149	1.348177	1.262543	34.04
52	0.229397	1.39415	1.279372	33.15
51	0.227233	1.439698	1.296376	32.27
50	0.225	1.484808	1.313528	31.40
49	0.222699	1.529465	1.330801	30.54
48	0.220329	1.573656	1.34817	29.69
47	0.217893	1.617369	1.365611	28.85
46	0.21539	1.660588	1.3831	28.02
45	0.212822	1.703301	1.400615	27.21
44	0.210189	1.745496	1.418133	26.41
43	0.207492	1.787159	1.435633	25.62
42	0.204731	1.828278	1.453095	24.84
41	0.201908	1.86884	1.4705	24.08
40	0.199024	1.908832	1.487829	23.33
39	0.196079	1.948243	1.505063	22.59
38	0.193075	1.987061	1.522186	21.86
37	0.190011	2.025273	1.53918	21.15
36	0.18689	2.062868	1.556029	20.44
35	0.183712	2.099835	1.572718	19.75
34	0.180478	2.136163	1.589232	19.07
33	0.177188	2.171839	1.605556	18.39
32	0.173845	2.206854	1.621677	17.73
31	0.170449	2.241197	1.63758	17.08

Table 4.9 (continue) Results for $\phi = 200^\circ$

β	a	b	c	μ_{min}(degree)
30	0.167001	2.274857	1.653254	16.44
29	0.163502	2.307825	1.668686	15.81
28	0.159954	2.340089	1.683864	15.18
27	0.156356	2.37164	1.698777	14.57
26	0.152711	2.402469	1.713413	13.96
25	0.14902	2.432567	1.727763	13.36
24	0.145283	2.461923	1.741815	12.77
23	0.141501	2.490529	1.75556	12.18
22	0.137677	2.518377	1.768989	11.60
21	0.133811	2.545457	1.782092	11.03
20	0.129904	2.571762	1.794862	10.46
19	0.125957	2.597284	1.807289	9.90
18	0.121972	2.622015	1.819366	9.35
17	0.11795	2.645947	1.831085	8.80
16	0.113892	2.669073	1.842438	8.25
15	0.109799	2.691386	1.85342	7.71
14	0.105673	2.712879	1.864022	7.18
13	0.101515	2.733545	1.874239	6.65
12	0.097326	2.75338	1.884065	6.12
11	0.093107	2.772375	1.893493	5.60
10	0.088859	2.790526	1.902519	5.07
9	0.084585	2.807827	1.911137	4.56
8	0.080285	2.824272	1.919342	4.04
7	0.07596	2.839857	1.927129	3.53
6	0.071613	2.854578	1.934495	3.02
5	0.067243	2.868428	1.941434	2.51
4	0.062853	2.881405	1.947944	2.01
3	0.058444	2.893504	1.95402	1.50
2	0.054017	2.904722	1.959659	1.00
1	0.049574	2.915055	1.964859	0.50
0	0.045115	2.9245	1.969616	0.00

Case study 5:

$\phi = 210^\circ$, $\psi = 30^\circ$.

$\beta = \pi - \dfrac{\phi}{2} = 75^\circ$ from the equation 4.36.

Ground link (A_0B_0): 1 unit

Crank link (A_0A): 0.258819 (Eqn. 4.37)

Rocker link (B_0B): 1.121971 (Eqn. 4.38)

Coupler link (AB): 0.570794 (Eqn. 4.39)

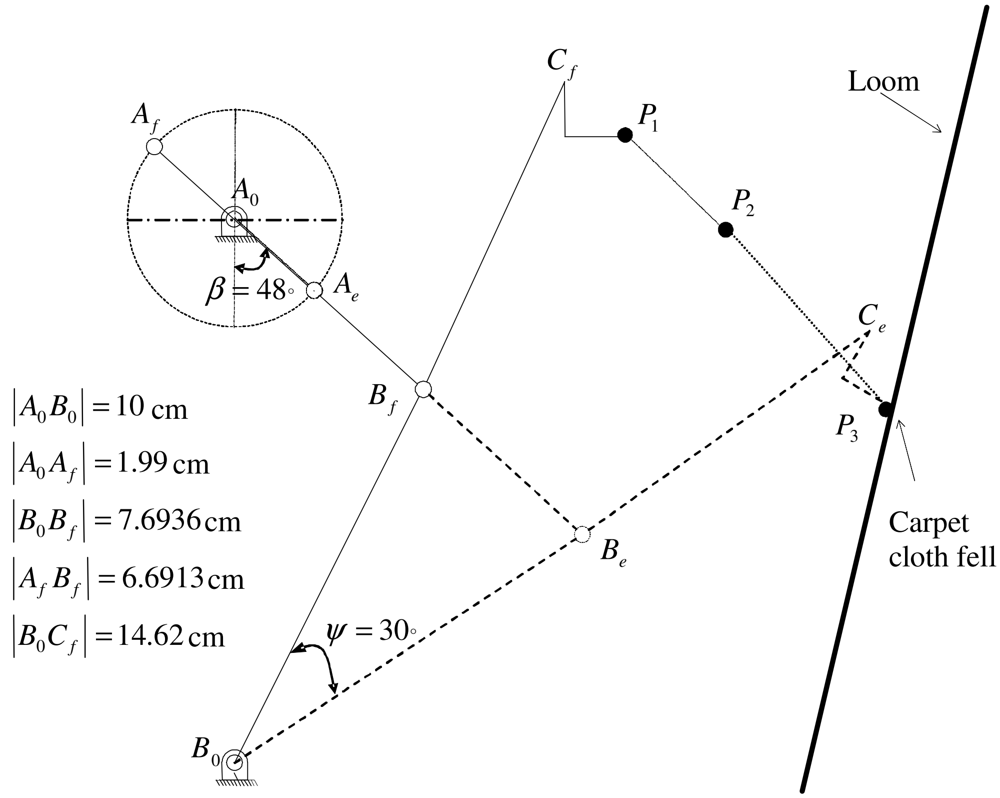


Figure 4.29 Dimensions and motion of fourth model of four-link mechanisms

As result of graph-analytical calculations, the range of the transmission angle that has minimum deviation from 90^o is obtained at $\phi = 180^\circ$. According to the required design criterias, the optimum transmission angle is obtained at $\beta = 48^\circ$.

Consequently the best solution for crank-rocker type beat-up mechanism is given in Figure 4.29.

b. Graphical Method

The graphical solution of the kinematic synthesis of the crank-rocker mechanism is performed by applying the following procedure (Figure 4.30) [71].

1. An arbitrary fixed link length A_0B_0= d is selected.
2. A line that makes an angle $-\phi/2$ with respect to A_0 is drawn. The negative sign means that this angle is in opposite direction to the angle ϕ.
3. Another line that makes an angle $-\psi/2$ with respect to $B_{0.}$
4. These lines makes will intersect at a point R. A circle, k_a, which passes through R and A_0 is drawn. The centre of the circle is on RA_0.

5. Another circle, kb, which also passes through R and A_0 is drawn. The center of the circle is on the line RB_0.
6. A line that makes an angle β with respect to A_0 is drawn from A_0. The line intersects the two circles at points A and B. The points of intersection are the locations of the moving pivot points of four-bar mechanism at the extended dead centre position. Angle β is called the "initial crank angle". There is a certain region for the angle β in which crank-rocker proportions are possible. If β is greater than β_2, the coupler link will be the shortest link and if β is less than β_1 the crank-rocker proportion will not be satisfied. The only possible solution is when $\beta_2 > \beta > \beta_1$.

In order to apply the above graphical solution procedure the swing angle of the rocker (ψ), corresponding rotation angle of the crank between two dead points of the rocker (ϕ) and appropriate ground link length (g) must be determined. The swing angle of the rocker is determined according to the motion of the weaver hand during the beat-up operation. The angle between the top position and the bottom position of the output link to obtain the same path with weaver hand is calculated as 30^o (Figure 4.16). So the swing angle of the rocker link can be selected as 30^o. The length of the ground link is selected as 10 cm for calculation of the link ratios easily. Four different crank rotation angles between two dead points of the rocker link are selected for the graphical solution. Four case studies are done for the selected crank

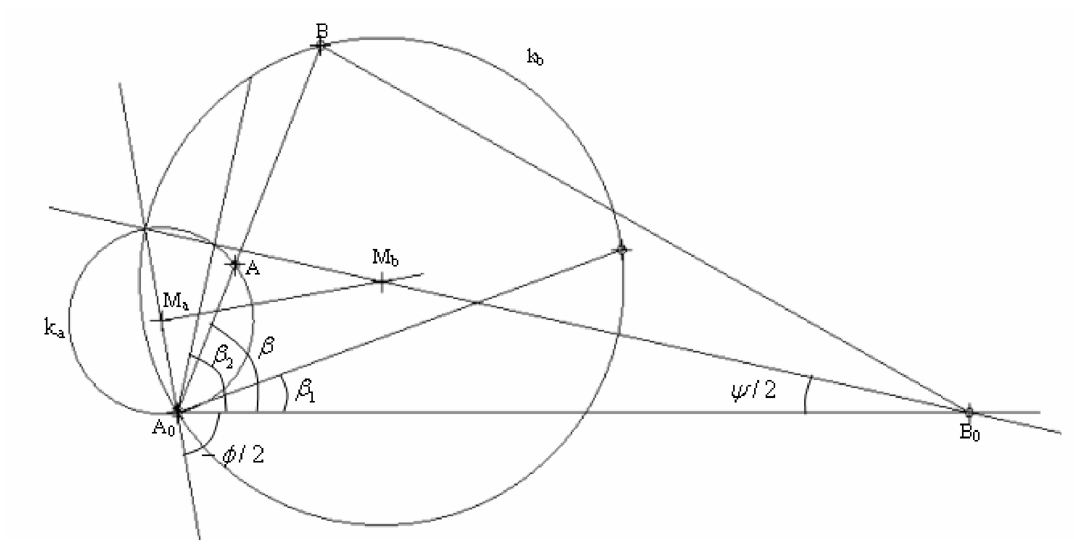


Figure 4.30 Graphical solution of crank-rocker mechanism

rotation angles. The second part of the solution is the selection of a particular crank-rocker proportion in which the transmission angle deviation from 90^o is a minimum. The solution of the problem has been put in a chart form as an easy reference for practical solution. These charts are commonly known as 'Alt Charts' in memory of the first person who recognized the importance of the problem. As mentioned above the last step of the solution procedure is drawing a line from A_0 that makes an angle β with respect to the A_0. The suitable angle for β is selected from Alt Charts (Chart 3) with respect to given the swing angle (ψ) and the crank rotation (ϕ).

Graphical synthesis of the mechanism

Case Study 1:
In this case, the graphical solution procedure is applied for following values.

A_0B_0 = 10 cm, $\psi = 30^\circ$, $\phi = 120^\circ$

According to given ψ and ϕ values β for the optimum transmission angle is found as 70° from Alt Charts. As a result of the graphical solution, the length of the each link of the mechanism for given values are as below (Figure 4.31).

Ground link (A_0B_0): 10 cm
Crank link (A_0A): 2.3528 cm
Rocker link (B_0B): 9.5560 cm
Coupler link (AB): 2.8039cm

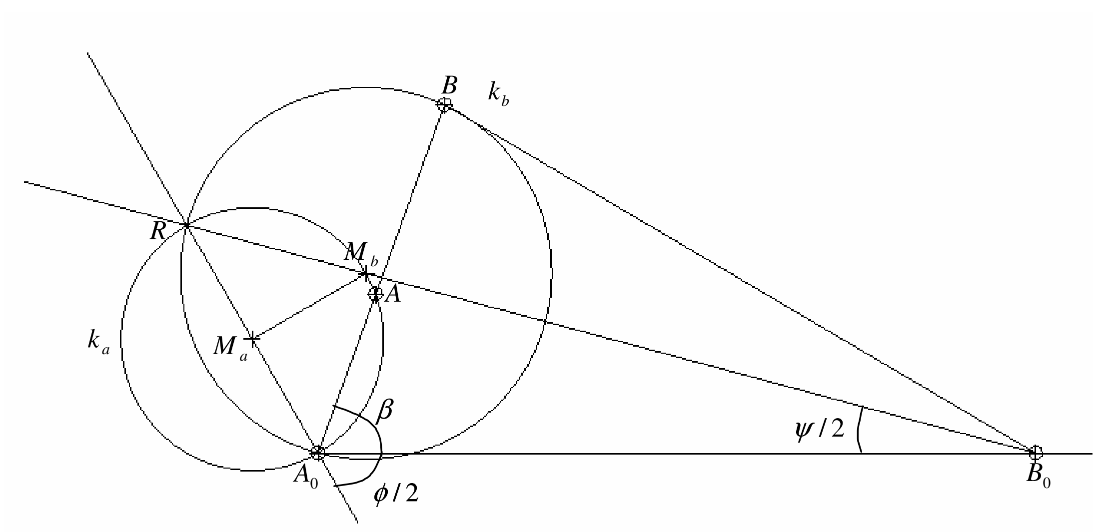


Figure 4.31 Graphical solution for $\phi = 120^\circ$

Case Study 2:
In this case the graphical solution procedure is applied for the following values.

A_0B_0 = 10 cm, $\psi = 30^\circ$, $\phi = 160^\circ$

According to given ψ and ϕ values β for the optimum transmission angle is found as 55° from Alt Charts. From the graphical solution, lengths of the each link of the mechanism are calculated as below (Figure 4.32).

Ground link (A_0B_0): 10 cm

Crank link (A_0A): 2.0193 cm

Rocker link (B_0B): 8.2145 cm

Coupler link (AB): 4.3304 cm

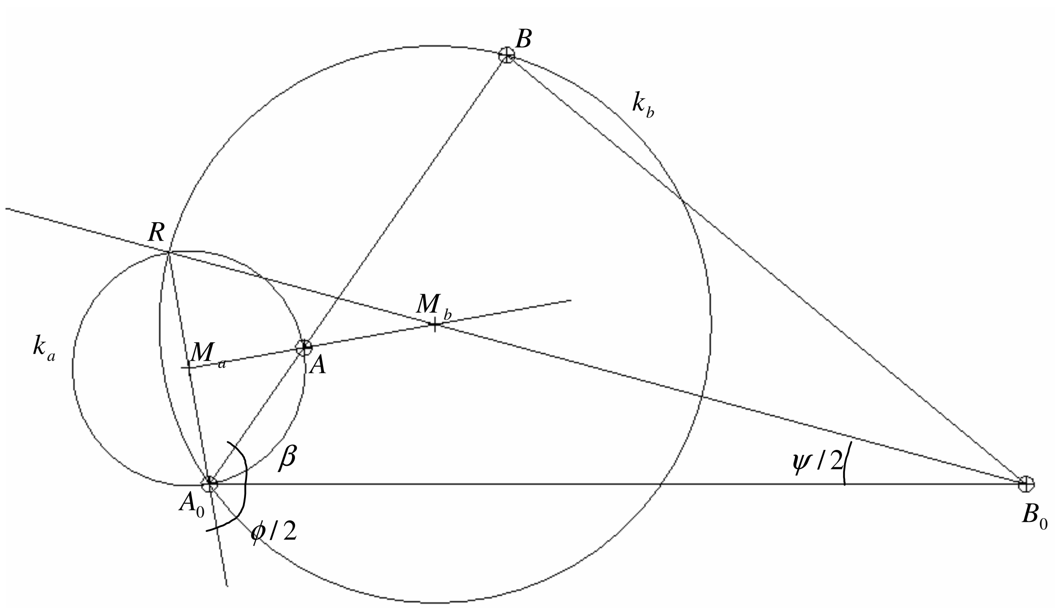


Figure 4.32 Graphical solution for $\phi = 160^\circ$

Case Study 3:

For this case study, following values are used for the graphical solution procedure.

A_0B_0 = 10 cm, $\psi = 30^\circ$, $\phi = 180^\circ$

The initial angle of the crank (β) is determined from the charts with respect to given crank rotation angle (ϕ) and rocker swing angle (β). But the chart which is used for this case study is different than others. Because this case is a special situation, a chart that is prepared for optimum β value with respect to swing angle is used. Then a line from the A_0 that makes an angle $\beta = 48^o$ is drawn.

As result of the solution the length of the each link of the mechanism is calculated as below (Figure 4.33).

Ground link (A_0B_0): 10 cm

Crank link (A_0A): 1.9913 cm

Rocker link (B_0B): 7.6936 cm

Coupler link (AB): 6.6913 cm

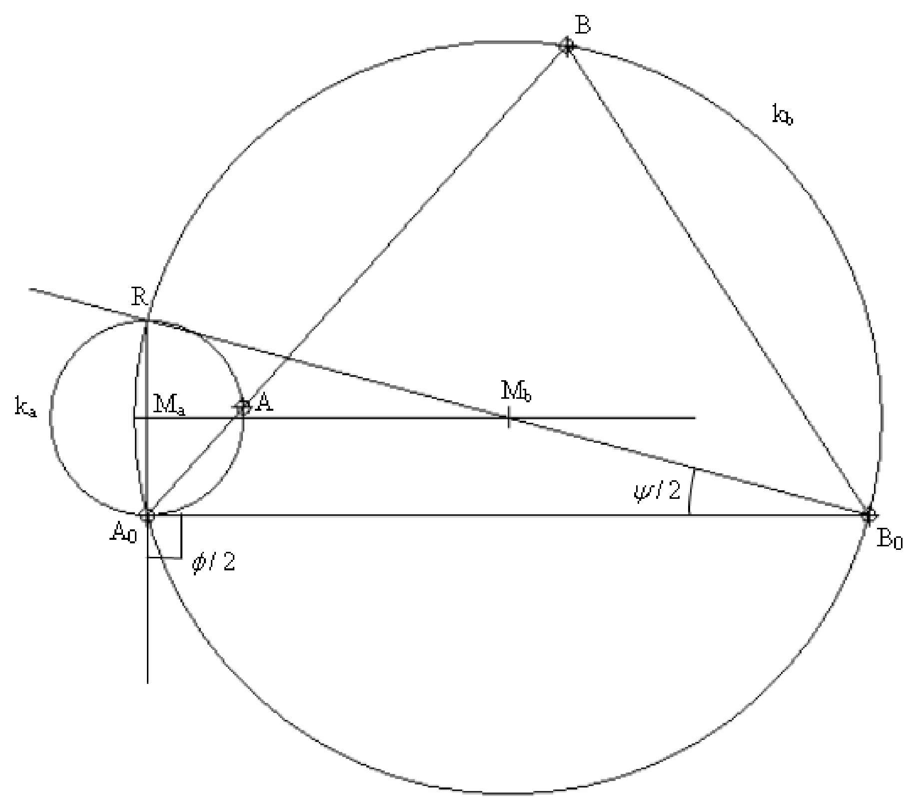


Figure 4.33 Graphical solution for $\phi = 180^\circ$

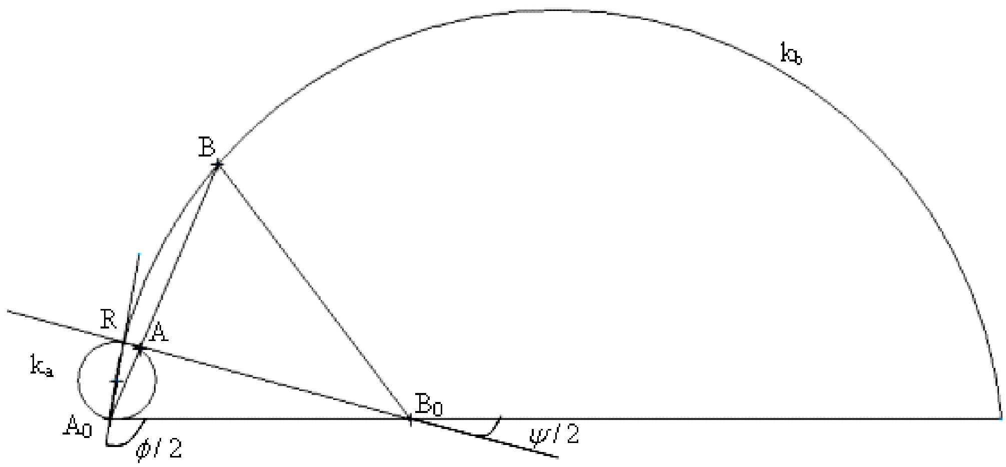


Figure 4.34 Graphical solution for $\phi = 200^\circ$

Case Study 4:
The solution is done according to given angle values. The graphical solution procedure is applied for the following values.

A_0B_0 = 10 cm, $\psi = 30^\circ$, $\phi = 200^\circ$

The angle β is determined from the charts according to given crank angle (ϕ) and rocker angle (ψ). The angle β is found as 67° from the chart.

As a result of the graphical solution, the length of the each link of the mechanism for given values are as below (Figure 4.34).

Ground link: 10 cm

Crank link (A_0A): 2.5315 cm

Rocker link (B_0B): 10.6239cm

Coupler link (AB): 6.6802 cm

Case Study 5:
The graphical solution procedure given above is applied for the following values.

A_0B_0 = 10 cm, $\psi = 30^\circ$, $\phi = 210^\circ$.

When $\phi = 180^\circ + \psi$, this is a special situation. If the locus of all possible solutions are constructed to satisfy the given swing angle and corresponding crank rotation, the locus of the moving pivot of the rocker becomes a straight line rather than a circle.

Initial crank angle β is equal to $180-\frac{1}{2}\phi$. A circle with diameter $a+d$ with center on the fixed link and passing through B_0 is drawn. B' is at the intersection of the perpendicular to the fixed link from A_0 and this circle. Point A is on the diameter passing through A_0. As a result of the graphical solution, the lengths of the each link of the mechanism for are given in Figure 4.35.

Ground link: 10 cm
Crank link (A_0A): 2.58819 cm
Rocker link (B_0B): 11.21971 cm
Coupler link (AB): 5.70794cm

The best transmission angle is obtained at $\phi = 180^\circ$. The same results which are obtained from the analytical solution (the case study 3) are obtained again in the graphical solution for $\beta = 48^\circ$.

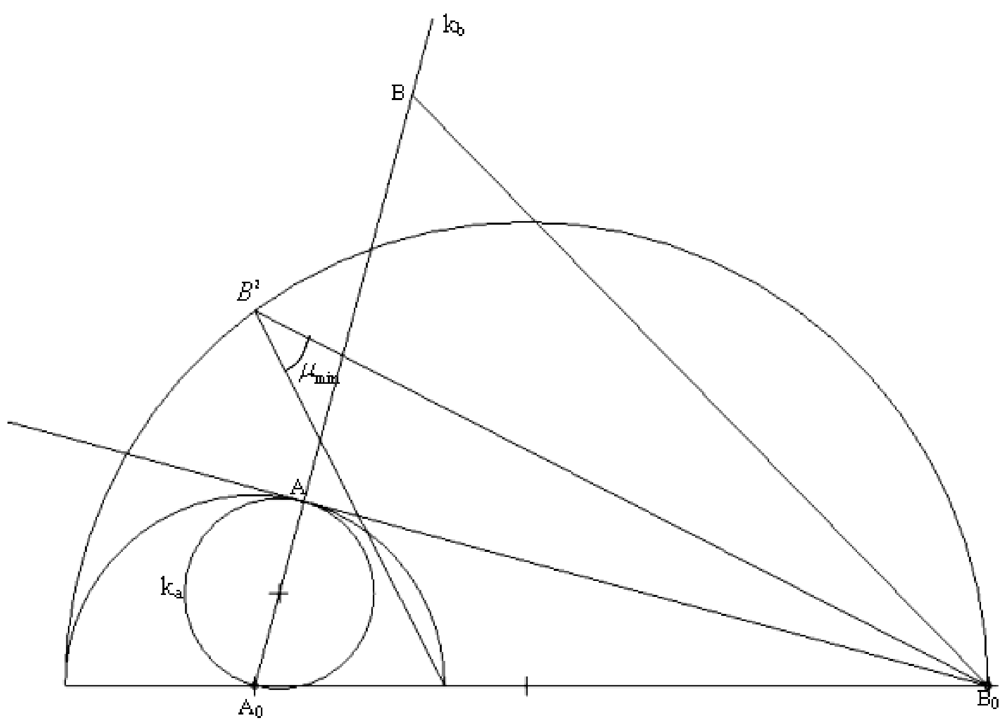


Figure 4.35 Graphical solution for $\phi = 210^\circ$

CHAPTER 5

DYNAMIC ANALYSIS OF BEAT-UP MECHANISM

5.1 INTRODUCTION

Mechanism is specifically designed to transmit force and motion. After determination of the motion in a mechanism, the shapes of each link and each joint are determined. The designs of these elements are mainly governed by the forces acting on them. The forces acting on the links and on the joints must be determined for such a design [75]. In the previous chapter, only the motion characteristic of the mechanism is considered and the dimensions of the mechanism are determined. In this chapter, the dynamic analysis of the selected beat-up mechanism shown in Figure 4.23 (four-link) is performed and the dynamic characteristic of the two link (Figure 4.6) beat-up mechanism is specified. A beater model which is proper for joining to the mechanism is generated. Also, a prototype model of the four-link beat-up mechanism shown in Figure 5.13 is presented.

The forces acting on the joints must be low and the force acting by the beater must satisfy needed force of the beat-up operation. Dynamic analysis of a four-link mechanism can be performed in two ways. The dynamic equation of the mechanism can obtained by using Newton method, Lagrange' formulation or Hamilton method. The equations can be solved using programs such as Turbo Pascal, Fortran, C etc. There are some studies about dynamic analysis of four-bar mechanism by using these methods [76-81]. In other way, some package programs such as Working Model, Pro-Engineer, Ch Mechanism Toolkit 2.0 (Softintegration), Solid Works can be used. Here, the aim of the dynamic analysis is to satisfy the design specification (i) of the beat-up mechanism.

5.2 DYNAMIC ANALYSIS OF FOUR-LINK MECHANISM

The mechanism represented in Figure 4.23 is assumed as made of cast iron. The density of the cast iron is 7210 kg/m^3. Except the length of the each link, the other dimensions of the links such as width and thickness are considered as the same. These dimensions of each link are given in Figure 5.1. The width of the links is 0.03 m and the thickness of the links is 0.01 m.

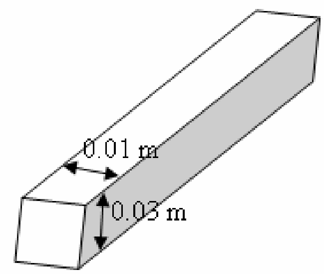


Figure 5.1 Width and thickness of links

Mechanical properties of the beat-up mechanism are given in Table 5.1. The configuration of the beat-up mechanism is like given in Figure 5.2. The weight of the beater connected to the tip of the B_0C_f link (Figure 4.23) with a rigid joint is assumed as 2.5 kg according to the actual beater used in the handmade looms.

Table 5.1 Mechanical properties of beat-up mechanism

a_1 = 0.213 m	a_2 = 0.58 m	a_3 = 0.189 m	a_4 =0. 232 m
m_2 = 0.125454 kg	m_3 = 0.408807 kg	m_4 = 0.651063 kg	
j_2 = 3.75212x10^{-5} kg.m^2	j_3 = 1.224581x10^{-3} kg.m^2	j_4 = 4.927787x10^{-3} kg.m^2	

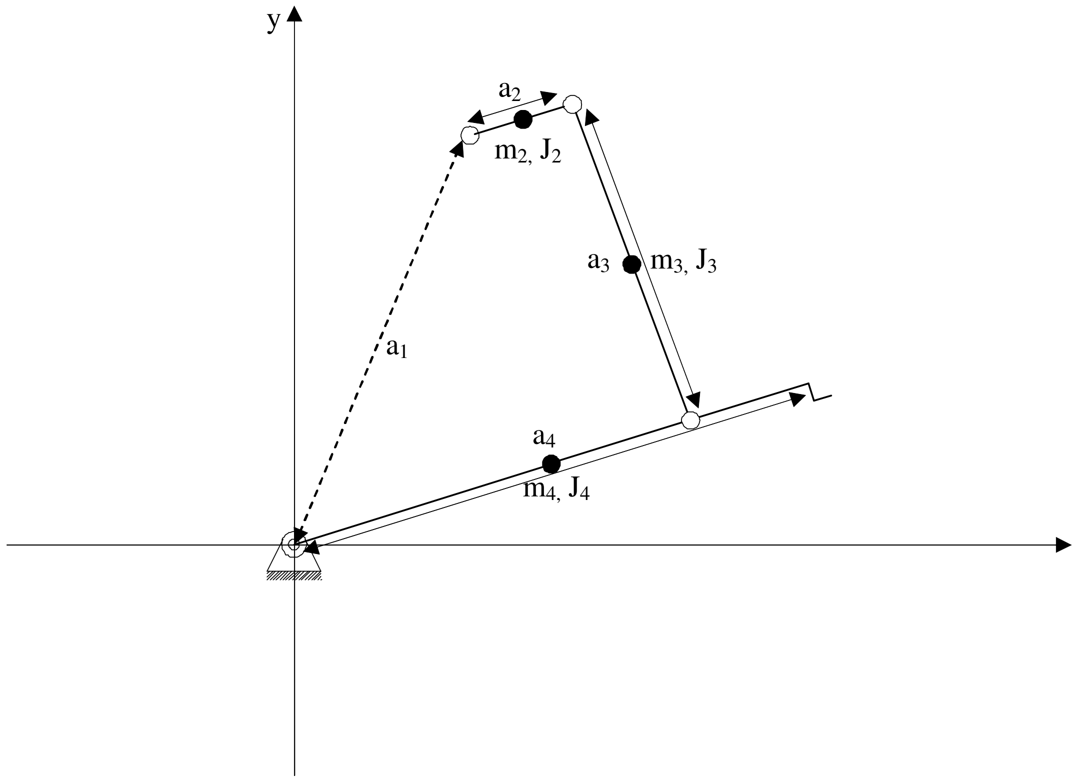


Figure 5.2 Configuration of beat-up mechanism

In order to achieve a proper beat-up process and obtain a tight, smooth carpet surface due to the design specifications (i) and (iv), the force applied by the beater must be about 8-10 kg (80-100 N). In the dynamic analysis of the beat-up mechanism, the force applied by the beater is analyzed in detail. The input value which generates required force on the output link is calculated. The loads on the joints are also analyzed to see whether the excess load is applied on the joints or not.

Dynamic analysis of the mechanism is made by using “Working Model 2D” program. The program provides understanding of how mechanical systems work and perform without building physical models. Working Model has some advantages such as quickly built, run, and refine simulations with pre-defined objects and constraints. And the required outputs can be seen as vectors or in numbers and graphs in English or metric units [82-84].

5.2.1 Working Model Study

The mechanism given in Figure 5.3 is constructed in the Working Model program. The coordinates of point A_0, the crank placed, is determined graphically. The mechanical properties of each link are defined to the program as in Table 5.1. Since the mechanism is actuated by the crank link (A_0A), driver of the mechanism is placed at point A_0. In order to obtain sufficient beater force at the point P, an input velocity of 15.5m/s is selected for link A_0A (crank). Position, velocity, acceleration and force graphics are obtained for three rotation of crank link (A_0A).

When the program is run, the position, velocity, acceleration of the beater and total force on the beater are obtained as in Figures 5.4 and 5.5 respectively. The beater reaches its maximum velocity value about beating point. The peaks shown in Figure 5.4 (b) are obtained when the mechanism is about the beat-up position. After the beat-up position, the velocity of the beater decreases sharply. The beater has approximately zero velocity, when the output link (B_0B) is at the highest position or at the lowest position. These positions are folded and extended dead centers of the beat-up mechanism respectively. The zero values just before the maximum values represent the highest positions (folded dead center) and the zero values just after the maximum points represent the lowest positions (extended dead center). The beater reaches its maximum acceleration value at the beating point given in Figure 5.4 (c). There are three peaks in the graphic for three beat-up processes. The maximum acceleration value is 34.819 m/s^2. The beater acceleration decreases after the beat-up operation.

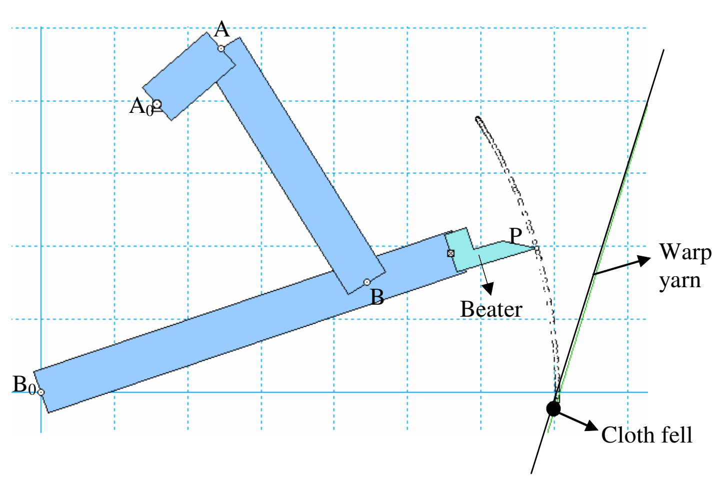
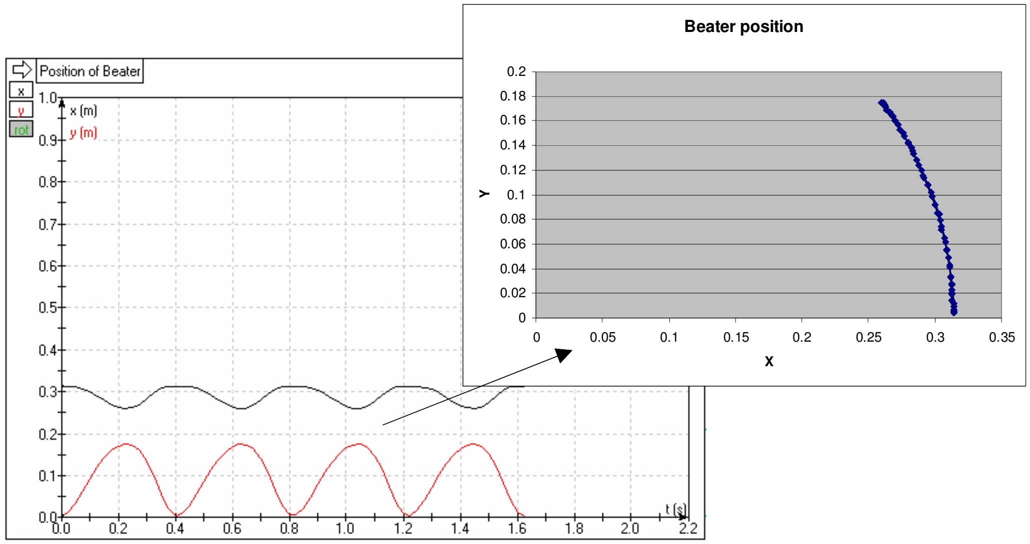


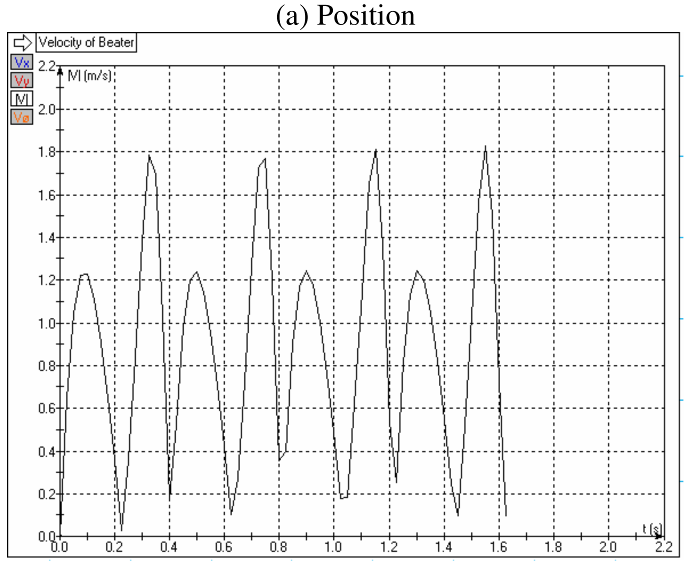
Figure 5.3 Four-link beat-up mechanism

Figure 5.5 shows the beater force change. Force applied at the beater point reaches a value over 80 N, so this force applied by the beater satisfies the design specification (i).for beat-up process. There are sharp changes in Figure 5.5. This is because; the force sharply increases about the beat-up point and it reaches the maximum value at the beating point, then the force sharply decreases after performing beat-up process. The peak points in the graphic represent the beat-up positions of the mechanism. Since the force exist at the beater point is caused by acceleration and mass of the beater, force and acceleration of the beater have similar characteristic.

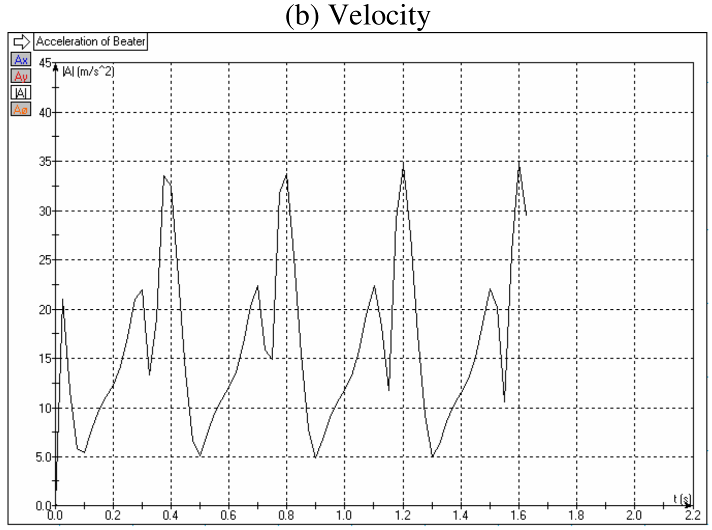
Figure 5.6 shows the position of the mechanism and force on the beater at the beating position. At this position, the force applied by the beater on the weft yarn reaches the maximum value. This value is about 87.047 N. The mechanism pushes the weft yarn and the knots until it reaches its bottom position. As shown in Figure 5.7, the force applied by the beater decreases to a small value when it reaches to the bottom position. The mechanism applies a force about 73.965 N at this position.



(a) Position



(b) Velocity



(c) Acceleration

Figure 5.4 Kinematics analysis for the beater in the four-link mechanism

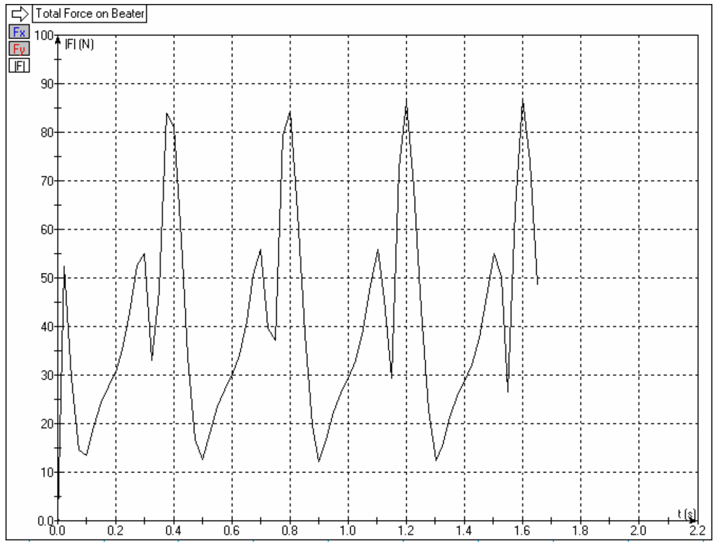


Figure 5.5 The force on the beater

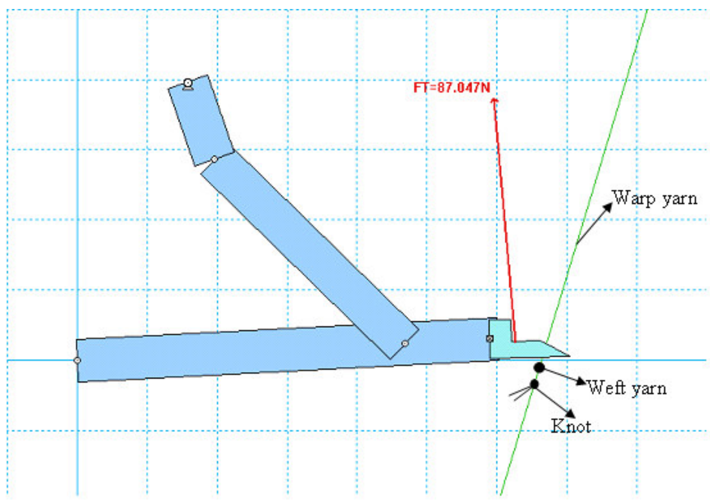


Figure 5.6 Mechanism at beat-up position

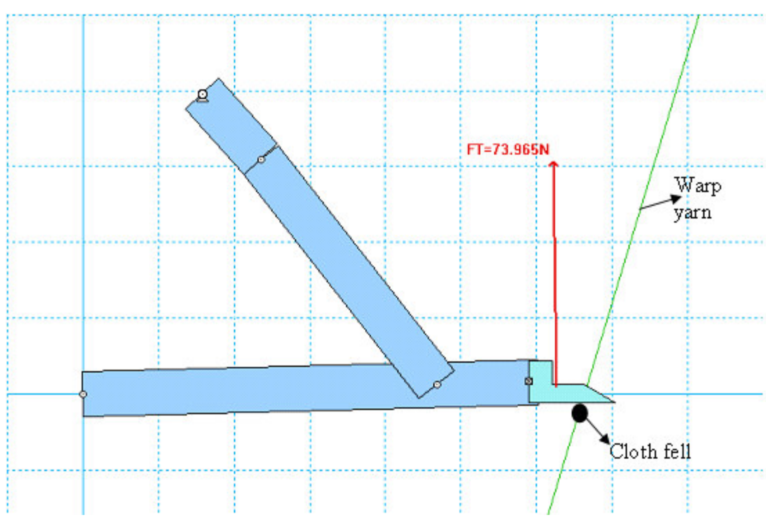
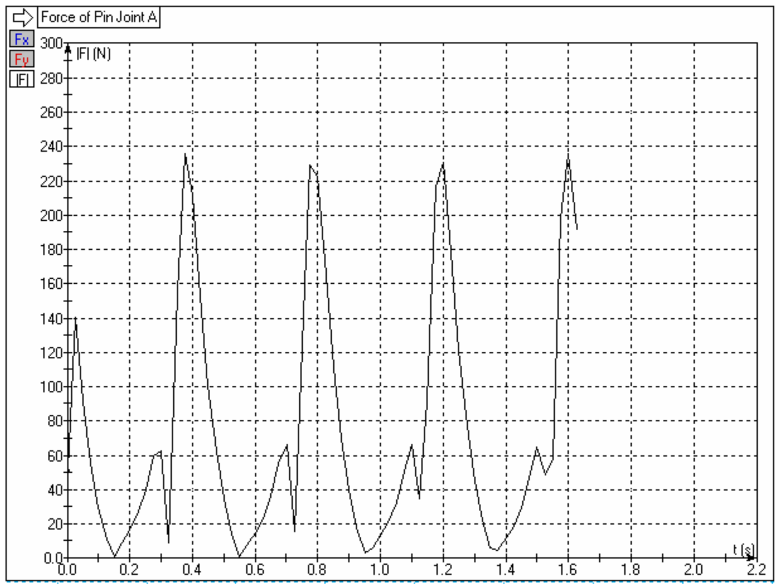
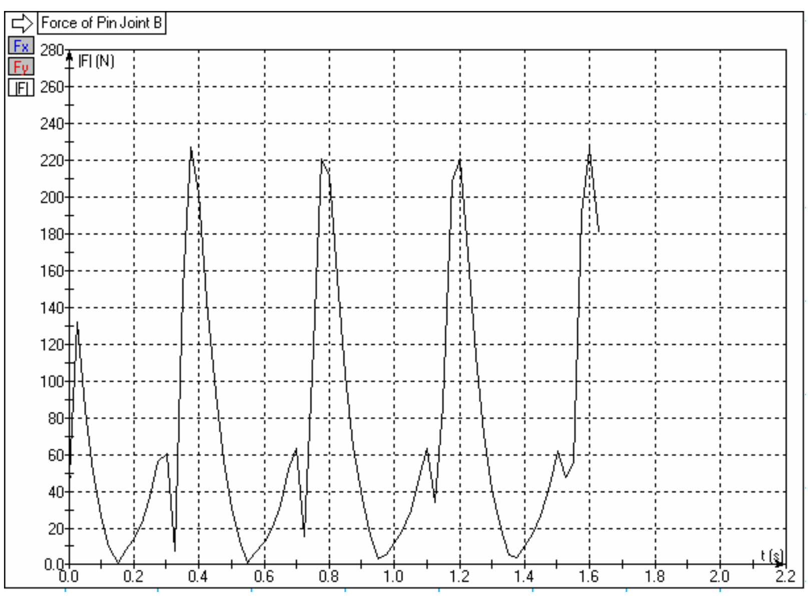


Figure 5.7 Mechanism at bottom position

Load on the joints during operation of the mechanism is an important factor. High loads on the joints cause can deformation of the mechanism and high vibration. So the strong materials are to be used with increased cost. Optimum length ratio is selected for mechanism links in order not to create high loads on joints. Figure 5.8 (a) and (b) show the force on joint A and B respectively. The force applied to the joints reaches the maximum value at the beat-up position of the mechanism.



(a) Load on the pin joint A



(b) Load on the pin joint B

Figure 5.8 Loads on the pin joints A and B

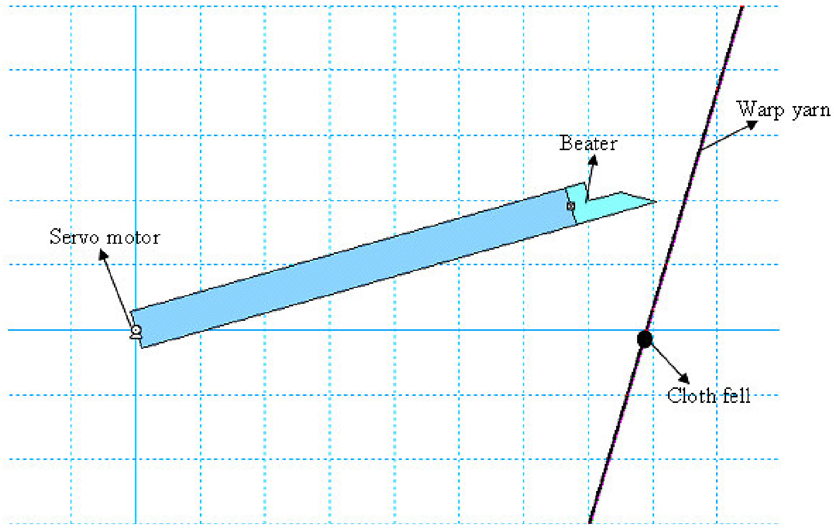


Figure 5.9 Two link beat-up mechanism

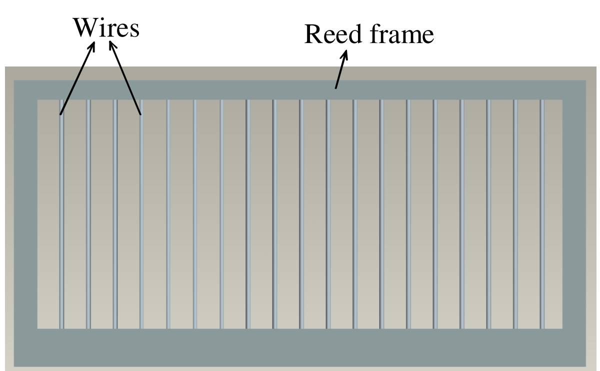
So, the peak points in Figure 5.8(a) and Figure 5.8(b) show the beat-up points. The force on the joints decreases sharply after beat-up process. The loads on the joints have the lowest value, when the output link of the mechanism is at folded dead center.

5.3 DYNAMIC ANALYSIS OF TWO-LINK MECHANISM

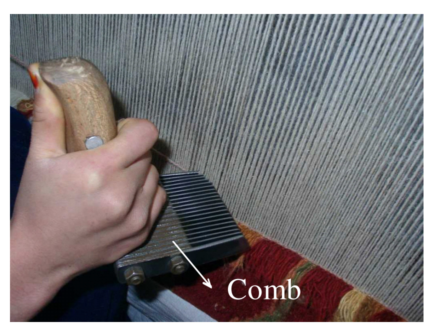
The two link mechanism represented in Figure 4.6 can be actuated by a servo motor. The servo motor feature a motion profile, which is a set of instructions programmed into the controller that defines the servo motor operation in terms of time, position, and velocity. The ability of the servo motor to adjust to differences between the motion profile and feedback signals depends greatly upon the type of controls and servo motors used [65, 85, 86]. Thus the required velocity and acceleration of the beater can be obtained by adjusting the servo motor. The link (Figure 5.9) oscillates at required angle. Adequate force required for a beat-up operation can be obtained on the beater by adjusting a suitable gear train and torque input.

5.4 A BEATER MODEL FOR THE BEAT-UP MECHANISM

In the weaving machines, the beat-up process which is pushing the last inserted weft yarn to the cloth fell is performed by using a device called reed [59]. Reed shown in Figure 5.10 (a) is closed comb of flat metal wires. The warp yarns are passed through the spaces between metal wires. The wires are uniformly spaced at intervals that correspond to spacing of warp ends in the fabric. The closeness of the wires to each other determines the warp yarn density. In the handmade carpet production, the beat-up process is performed by using an instrument called as ‘comb’ or ‘beater’ (Figure 5.10(b)). The comb or beater made of iron or wood weighs about 2 kg or 2.5 kg and has a handle in order to keep and move it easily. The width of the beater is changed between 5 cm and 10 cm. It has uniformly spaced teeth. During the beat-up process, these teeth inserts through the wraps and push the weft yarn and knots into carpet cloth fell. Because the knots and the weft yarn are pushed together to the cloth fell, a high beat-up force greater than tensile warp force plus the friction force between yarn and metal is required. The kind of the comb varies thick or thin ones according to the weaver and carpet specifications such as required the knot density and the pile thread length. The teeth construction and the size of the beater which will be joined to design beat-up mechanism will be the same as that of the beaters used by the handmade carpet weavers. As shown in Figure 5.11, the beater shown in Figure 5.10(b) is designed again in order be joined to the beat-up mechanism. The handle of the beater is removed and a groove is formed in order to be connected to the output link of the beat-up mechanism. The beater has the same teeth construction with the beater in Figure 5.10(b). The thickness of the beater (Figure 5.10(b)) is increased in order to provide required beat-up force.



(a) The reed used in woven fabric



(b) The beater (comb)

Figure 5.10 Photograph of instruments used in beat-up process

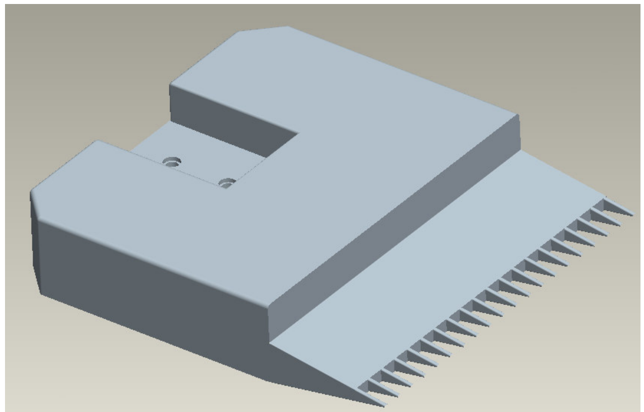


Figure 5.11 Beater designed for beat-up mechanism

5.5 PROTOTYPE MODEL OF THE FOUR-LINK BEAT-UP MECHANISM

The mechanism run in Working Model program (Figure 5.3) satisfies the required beat-up force. A prototype model is constructed (Figure 5.12), it has the same mechanical and dimensional properties with the model formed in Working Model program. The mechanism is driven by a motor via a gear train as in Figure 5.12. The beater designed for the mechanism is joined on the output link with a rigid connection. The bearings at the joints are selected according to the dynamic analysis results. This prototype model operates without any problem.

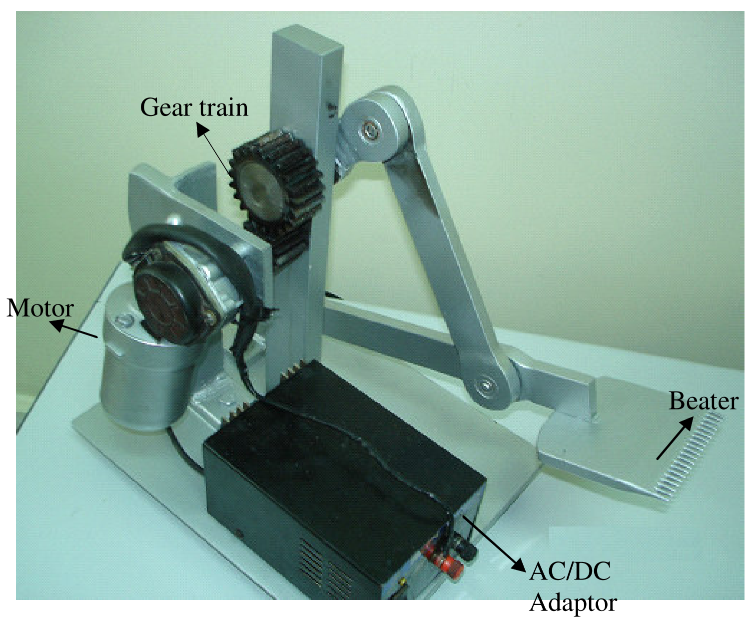


Figure 5.12 Prototype model of the four-link mechanism

CHAPTER 6

DISCUSSION AND CONCLUSION

6.1 DISCUSSION AND CONCLUSION

Handmade carpet weaving processes take long time and consist of many exhausting operations. Every process requires specific attention and experience. Weaving one m^2 high quality Hereke carpet takes approximately four days. Performance and quality of the knotting, beat-up and picking operations determine the quality of the carpet. The weaver spends most of its time for these processes. During weaving of a row of carpet, an experienced weaver expends approximately five minutes for knotting a row of pile yarn, two minutes for shedding and picking operation and three minutes for beat-up operation. The weaver must beat every point of knot row with same intensity and the tension of the picking yarn must be at optimum level. Otherwise the carpet structure is deformed and carpet loses its quality.

This work presents a study for developing shedding and picking mechanisms and designing beat-up mechanisms for handmade carpet looms. The systematic development of the work has been started with a brief description of the handmade carpet. The work has continued on the developing alternative models for shedding and picking mechanisms and designing of beat-up mechanism. Finally, dynamic analysis and kinematic synthesis of the beat-up mechanism are performed. A prototype beat-up mechanism is constructed.

a) Discussion on developing shedding mechanisms

- During development of alternative shedding mechanisms, initially shedding mechanisms used for weaving machines were examined. Then the shedding operation made on the hand looms was investigated. The important parameters of this operation and design requirements of the mechanism were determined.

- Depending upon the design requirements, five different models were generated. Since the fabric structure of the carpet has a plain configuration, two heald frames are enough to form the required sheds. So the shedding mechanism models were generated to control two heald frames.

- In the five alternative shedding models the gear train system, the link mechanism, the cam-link mechanism and the pistons are used. All shedding mechanism models were evaluated in terms of design requirements and they were compared.

- Between these alternative models, the fourth model (Figure 3.6) would be the most suitable one depending upon the evaluation criterias. In this model, the heald frames were controlled by two pistons. It has simple to control and in compact structure. This mechanism doesn't need large space for shedding operation. Also, the construction of this model would be easier than others.

b) Discussion on developing picking mechanisms

- While developing alternative picking mechanisms for handmade carpet looms, first of all the picking operation made on a hand loom was investigated and design requirements of the mechanism were determined.

- Four different alternative models were generated depending on the design requirements. In the structure of the handmade carpet, the selvages are formed on both sides.

- Since the selvage can be formed by using shuttle in the picking operation, the models were designed to carry a shuttle along the loom width.

- Four alternative models generated for picking mechanism were evaluated in terms of design requirements and they are compared.

- Because the third alternative model (Figure 3.12) is simple and easy to control, it is taken as the most suitable solution. This model has a compact structure, but the manufacturing of the members such as helix shaft and grooved carrier could be difficult. The speed of the mechanism can be adjusted by the motor speed. Since the shuttle is carried between both sides of the loom, this mechanism needs a bi-directional control.

c) Discussion on designing of beat-up mechanism

- During design of the beat-up mechanism, first of all, the function of the beat-up operation in a hand made carpet weaving was investigated and important parameters affecting the performance of the operation and the carpet quality were determined.

- Initially, the most suitable trajectory that the mechanism must follow was determined. Since the weavers achieve a high quality beat-up process, mechanism following similar trajectory with the weaver hand is considered to be the best solution. So, the motion trajectory of hand of an experienced weaver during a beat-up process was determined.

- The functional requirements of the beat-up mechanism were determined and they are translated into design specifications.

- Eight alternative models were generated in the view of the design specifications. All alternative models are evaluated in their groups.

- Two four-link mechanisms (Figure 4.7 and Figure 4.10) were selected as the most suitable ones. Both of the mechanisms were the crank-rocker type. These mechanisms have the same working principle, and with similar construction. Both of them have similar trajectory to the weaver hand motion.

d) Discussion on dimensional synthesis of the beat-up mechanism

- In the dimensional synthesis of the selected beat-up mechanisms, at first, the dimensional synthesis problem categories were investigated. The synthesis method suitable for the selected beat-up mechanisms was determined.

- Since there is relation between the crank rotation angle and the oscillation angle of the output link, the function generation synthesis method is suitable for both mechanisms.

- The trajectory of the weaver hand motion is scaled in the ratio of 1/3.5. Three precision point motion generation synthesis was performed on this trajectory. Thus length and oscillation angle between two dead centers of the output link of the beat-up mechanisms was determined.

- The dimensional solution of the first model of the four-link mechanisms (Figure 4.7) was performed as slider-crank type mechanism by using both analytical and graphical methods. For this beat-up mechanism, 91 alternative dimension solutions were found in analytical method.

- The dimensional solution of the fourth model of the four-link mechanisms (Figure 4.10) was performed as the crank-rocker type mechanism by using both graph-analytical and graphical methods and 225 alternative dimension solutions were found for this mechanism.

- Depending upon the some mechanism design criterias such as the transmission angle, Grashof's condition and the crank/coupler ratio, the best solutions for both beat-up mechanisms were selected. The solution for the first model of four-link mechanisms was more suitable than that of the fourth model in terms of easy manufacturing and compact structure.

e) Discussion on dynamic analysis of the beat-up mechanism

- The dynamic analysis was made on the first model of four-link mechanisms. It was assumed that the mechanism was made of cast iron and mechanical properties of the mechanisms; masses and mass moment of inertials of each link were determined.

- The mechanism was constructed in a simulation program (Working Model 2D) with its real size and mechanical properties and the beater placed at the tip of output link. The mechanism was run in the program.

- The position, velocity, acceleration and force of the beater were determined and their graphics were obtained.

- The most important parameters of the beat-up mechanism are force applied by the beater and load on the joints. The force applied by the beater must be in the range of 80-100 N. The input value which generates the required force on the beater was determined on the program. The loads on the joints were also determined to select a suitable bearing in the construction of the mechanism.

- A beater (comb) model was designed (Figure 5.11) for the mechanism. The dimensions, size and teeth construction of the model is same as beaters that are now used by handmade carpet weavers. This model is adapted for joining the beat-up mechanism.

- Then a prototype model which has the same mechanical properties and dimensions with the mechanism formed in the Working Model program is constructed. The bearings at the joints were selected by looking at the dynamic analysis. The prototype model is operating without any problem at present.

f) Conclusion

- In this study, picking and shedding mechanisms were developed and a beat-up mechanism was designed for handmade carpet looms.

- Since the handmade carpet weaving operations are completely performed by weaver, the production of a handmade carpet takes long time and requires experience. By using these mechanisms, the production speed of the handmade carpet would be increased and the performance of the weaver could be improved.

- The important parameters of these operations such as tension of the weft yarn and beat-up force are judged by the weaver. Hence, many weaving faults may potentially occur, when these operations are performed without attention or they are carried out by inexperienced weavers. By using these designed mechanisms, weaving faults would certainly be decreased, the carpet quality would be kept and the repair cost would be decreased.

6.2 RECOMMENDATIONS FOR FUTURE WORK

The following studies may be considered as a future work.

i. The beat-up mechanism is designed theoretically in this study. In a further study the construction of this mechanism may be made and it may be placed on a hand loom.

ii. A suitable mechanism which moves the beat-up mechanism along the loom width may be designed. A suitable actuator for this mechanism may be designed and the beat-up mechanism may be synchronized with it.

iii. Alternative shedding and picking mechanisms are generated in this study. In the further study design and construction of these mechanisms may be done.

iv. A computer control system may be suggested for the picking and beat-up mechanisms. Thus, these mechanisms may be synchronized and they work in a required sequence.

v. In order to generate required motions of the mechanisms and produce required force on the beat-up mechanism, suitable actuators such as servo motors, gear boxes and piston unit may be suggested.

vi. As mentioned in literature survey knotting mechanism is designed by Topalbekiroğlu [7]. As a further study, the picking and beat-up mechanisms may be combined and synchronized with this system.

vii. Letting-off and taking-up operation may be designed for handmade carpet looms. By combining these parts with beat-up, picking and knotting operation designs, the handmade carpets may be completely produced mechanically on a loom.

REFERENCES

1. Topalbekiroğlu M., Kireççi A., Dülger L. C (2001). *El Halıcılığında Bilgisayar Destekli Desen Çalışması ve Deseni Oluşturan İplerin Denetimi.* 10. Ulusal Mak. Teo. Sempozyumu, Konya, Bil. Kit. 2, Türkiye.767-775.

2. Geosign Technologies Inc., (January 2007), http://www.thecarpetsite.com/carpet-history.aspx

3. Wikipedia, The free Encyclopedia, (January2007), http://en.wikipedia.org /wiki/Carpet

4. Crossland A. (1958). *Modern Carpet Manufacturing*, Columbine Pres Manchester & London.

5. Corbman B.P (1938), *Textiles: Fiber to Fabric*. The Textile Institute, Manchester.

6. Aytaç Ç. (1997), *El Dokumacılığı*, 3.Baskı, Milli Eğitim Basımevi.

7. Topalbekiroğlu M. (2002), *Design, Construction and Control of Computer Controlled Knotting System*, A Ph. D. Thesis in Mechanical Engineering Department, University of Gaziantep.

8. Topalbekiroğlu M. (2005), Kinematic Analysis and Synthesis of The Knotting Mechanisms Can Be Used in The Production of Handmade Carpet: A Case Study. *J.Mechanical Engineering Science,* **Vol.219** Part. C. p 987-1005

9. Kireçci A, Doğan C, Topalbekiroğlu M (1995), *El Dokuma Halıların Mekanik Olarak Dokunması.* 7. Ulusal Makine Teorisi Sempozyumu,Yıldız Teknik Üniversitesi, İstanbul, Bil. Kit.19-26.

10. Kireçci A, Dülger C, Topalbekiroğlu M. (2001). *Gördes Düğümlü Elektromekanik Halı Üretiminin Analizi.* 10. Ulusal Makine Teorisi Sempozyumu, Selçuk Üniversitesi, Konya, Bil. Kit. sayfa 68-76.

11. Topalbekiroğlu M, Kireçci A, Dülger C (2001). 10. *El Halıcılığında Bilgisayar Destekli Desen Çalışması ve Deseni Oluşturan İplerin Denetimi.* 10. Ulusal Makine Teorisi Sempozyumu, Selçuk Üniversitesi, Konya, Bil. Kit. sayfa 767-775.

12. Chaudrary H. and Saha S.K. (2005 *Feb.24-25).* Finite Element Modeling of Carpet Weaving Loom Structure, *Published in the Proc. of the Nat. Conf. on Industrial Problems on Machines and Mechanisms, IIT Kharagpur*, pp. 197-203.

13. Chaudrary H. and Saha S.K. (May 2006). Optimal Design of an Indian Carpet Weaving Loom Structure, *Journal of Scientific and Industrial Research, CSIR, India,* **V. 65**, pp. 410-415.

14. Gürtanın N., Yazıcıoğlu Y. ve Canikli N. (1988). *Değişik Düğümleme Sistemleri Kullanarak Farklı Hav Yüksekliklerinde Üretimiş El dokusu*

Halıların Resilyans Özellikleri. IV. Tekstil Sempozyumu. Bursa, sayfa. 260-265.

15. Küçükerman Ö. (1997). Türk Halıcılığının Tarihten Gelen Sorunları. *Arış Dergisi, sayfa. 88-93, Ankara*

16. Bayraktaroğlu S. (1997). Türk Halılarında Batı Literatürü Konusu. *Arış Dergisi, sayfa. 88-93, Ankara.*

17. Dawson, R.R. (1983). Analysis of Geometrical Constrains on Reed Motion. Preliminaries. *Journal of Textile Institute, 350 p.*

18. Dawson, R.M. (1985). The Orientation and Motion of The Weaving Reed. *Journal of Textile Institute, 242 p.*

19. Eren R. (2000). Dokuma Makinelerinde Tefe Mekanizmaları Üzerine Bir İnceleme, *Tekstil-Teknik, Ocak 2001, sayfa 174-185.*

20. Dekun D. Bullerwell A. Mohamed M. (1991). Dynamic Analysis of Beat-up Process. *Textile Research Journal, **61 (12)**,* 760-773.

21. Shih, Y, Momamed M.H. ,Bullerwell A.C., Doa D. (1995). Analysis of Beat-up Force During Weaving, *Textile Research Journal,* **Vol. 65**, No. 12, 747-754.

22. Katunskis J. (July / October 2004). Theoretical and Experimental Beat-up Investigation. *FIBRES & TEXTILES in Eastern Europe,* **Vol. 12**, No. 3 (47), p 24-28.

23. Kumpikaite E. Milasius V. (2003). Analysis of Interrelation Between Fabric Structure Factors and Beat-up Parameters. *Material Science.***Vol.9**, No.2, p 228-232.

24. Eren R. Tarhan M. (2002). Dokuma Makinelerinde Tefe Tahrik Kamlarının Tasarımı. *Tekstil Maraton,* Temmuz-Ağustos 4/2002, sayfa 61-67.

25. Mrazek J. (1992). Theoretical Analysis of Dynamic of Four-bar Beat-up Mechanism of A Loom. *Mechanism and Machine Theory,* **Vol.27**, Issue 3, May, Pages 331-341.

26. Vaclavik M. Koloc Z. (1997). The Simulation of Some Mechanisms on Weaving Machines. *Mechanism and Machine Theory,* **Vol.12**, Issue 6, Pages 605-611.

27. Sternheim, Grossberg A. P. (1991). The Effect of Sley Motion The Beat-up Force. *Journal of Textile Institute,* p 331.

28. Zhang Z. Mohamed M.H. (July/October 2004). Theoretical Investigation of Beat-up. *FIBRES & TEXTILES in Eastern Europe,* **Vol. 12**, No. 3 (47), pg 24-28

29. Eren R. Aydemir A. (2004). An Approach to Kinematic Design of Four-bar Sley Drive Mechanisms in Weaving. *Journal of Textile Institute,* **Vol. 95**, Issue. 1-6, p 193-205.

30. Adanur S. Bakhtıyarov S. (June 2002). Characterization of Air-yarn Interface in Air-jet Weaving. *National Textile Center Research Briefs – Fabrication*

Competency: (*http://www.ntcresearch.org/pdf-rpts/Bref0602/F99-AE10-02.pdf*)

31. Turelt T. Bakhtıyarov S. Adanur S. (2004). Effects of Air and Yarn Characteristics in Air-jet Filling Insertion. Part I: Air velocity and Air Pressure Measurements. *Textile research journal,* **Vol. 74**, no.7, pp. 592-597.
32. Vangheluwe L. (January 1999). Air-jet Weft Insertion: *Textile Progress,* **Vol. 29** No. 4
33. Mangold, Weinsdörfer S., H. (2003). Problem of Incomplete Weft Insertion in Air-jet Weaving. *Melliand International,* **No. 9**, p. 220-223.
34. Osthus T., Weldige E. D. and Wulfhorst B. (March-April 1995). Reducing Set-up Times and Optimizing Processes by The Automation of Setting Procedures on Looms. *Mechatronics,* **Vol. 5**, Issues 2-3, Pages 147-163.
35. Vangheluwe L., Sleeckx B. and Kiekens P. (March-April 1995). Numerical Simulation Model for Optimisation of Weft Insertion on Projectile and Rapier Looms**.** *Mechatronics,* **Vol. 5**, Issues 2-3, Pages 183-195**.**
36. Koloc Z. and Vaclavik M. (July 1992). Dynamics of the Picking Mechanism of a Rapier loom**.** *Mechanism and Machine Theory,* **Vol. 27**, Issue 4, Pages 435-441.
37. Robinson G. (1972). *Carpets and Other Textile Floor coverings.* Division of Bonn Industries Inc. (2th ed.)
38. Grosicki Z. J. (1913). *Watson's Advanced Textile Design*. Newnes Butterworths.
39. TS 43/ Nisan 1992, *Tekstil Yer Döşemeleri – El Dokuması Halılar Türk Halıları,* Birinci Baskı, UDK 645.12
40. Ayyıldız R. (Şubat 1997). *El Halıcılığı Sanayi Profili*, Sanayi ve Ticaret Bakanlığı Sanayi Araştırma Geliştirme Genel Müdürlüğü, ANKARA
41. E-Rug Gallery. (October 2006). http://www.eruggallery.com/learnrugs/learn_tools/tools_hloom.htm
42. Ali Baba's Hand Made Carpet and Kilim Ware House. (October 2006). http://www.handmadecarpets.net/carpet_store_kilims.htm
43. *El Halıcılığı.* M.E.B çıraklık ve yaygın eğitim genel müdürlüğü
44. www.mteflooring .com\carpet_infrm1.htm
45. Persian Carpet. Inc. (June 2006). www.persiancarpet .com.za\how.htlm
46. Bukhara Carpet, Kazakhstan. (2006). http://www.bukharacarpets.com /making/making.html
47. Jacobsen Oriental Rugs, U.S.A. (June 2006). http://www.jacobsenrugs.com /knots.htm
48. Oriental Rug Repair Co., U.S.A. (July 2006). http://www.navajorugrepair.com/knots.htm
49. Islamic Arts and Architecture Organization. (June2006). http://www.islamicart.com/main/rugs/weave.html

50. Oriental Rugs and Kilim Co. Turkey. (June2006). http://www.orientalrugsandkilims. com/rugguide/knots.htm

51. Jacobsen Oriental Rugs, U.S.A. (June 2006). http://www.jacobsenrugs.com /countknots.htm

52. E-Rug Gallery. (October2006). http://www.eruggallery.com/learnrugs/ learn_detail/knot/lrn_knot.htm

53. Hassan's Carpets Pte. Ltd. (September 2006). http://www.hassanscarpets.com /index.htm

54. The Bagheri Group, Bagheri's. (September 2006). http://persianoutpost.com/htdocs/po_displaypage.php?pageid=rugtips

55. Bazaar Turkey Co. Ugur Senguler. (September 2006). http://www.bazaarturkey.com/parts_of_an.htm

56. Bukhara Carpet, Kazakhstan. (September 2006). http://www.bukharacarpets.com/making /knots.html

57. China Travel Service-Travel China Guide. (September 2006). http://www.travelchinaguide.com/shopping /silk_carpet/weave.htm

58. Halı Usta Öğretmeni ve Dokuyucusunun El Kitabı, Erdoğan ILDIZ. (June 2006). http://www.eracarpets.com/yayinlar/hali-dokuyucusu/sayfa2.htm

59. Adanur S. (2001). *Handbook of Weaving,* CRC PRESS.

60. Talavasek O., Svaty V. (1981). *Shuttleless Weaving Machines*, Textile Science and Technology Volume 3, Elsevier Scientific Publishing Company, New York,

61. Marks R., Robinson A.T.C. (1976). *Principle of Weaving*, The Textile Institute Manchester.

62. Ormerod A., Sondhelm W.S. (1998). *Weaving Technology and Operations*, The Textile Institute.

63. Free patents online, United States Patent 4223703. (May 2007). http://www.freepatentsonline.com/4223703.html

64. North Carolina Division of Pollution Prevention and Environmental Assistance. (May 2007). http://www.p2pays.org/ref/11/10023/AirJetLoom.asp

65. Norton R. L. (1992). *Design of Machinery*. Mcgraw-Hill Inc.

66. Erdman and Sandor. (1997). *Mechanism Design Analysis and Synthesis*. Prentice Hall.

67. Sandgren E. (1985). Design of Single and Multiple Dwell Six-Link Mechanism Through Design Optimization. *Mechanism and Machine Theory,* **Vol. 20**, No.6, pp. 443-490.

68. Jimenez J.M., Alvarezz G., Cardenal J. and Cuadrado J. (1997). A Simple and General Method for Kinematic Synthesis of Spatial Mechanisms. *Mechanism and Machine Theory*. **Vol. 32**, No.3, pp. 323-341.

69. Söylemez E. (2002). Classical Transmission-angle Problem for Slider-crank Mechanism, *Mechanism and Machine Theory,* **Vol. 37,** pg 419-425.

70. Söylemez E. (1999). *Mechanisms*. Middle East Technical University, Publication number: 64.

71. Söylemez E. (1979). *Mechanisms*. Middle East Technical University.

72. Khare A.K. and Dave R.K. (1979). Optimizing 4-bar Crank-Rocker Mechanism. *Mechanism and Machine Theory*, **14**, pg 319-322.

73. Önder E., (1999-2000). *Weaving Technology II Course Notes*, İstanbul Teknik Üniversitesi.

74. Alpay H.R. (1985). *Dokuma Makinaları*. Uludağ Üniversitesi

75. Söylemez E., Tümer S.T., Özgüven H.N., Özgören K. (1984). *Nots on Dynamics of Machinery*. Middle East Technical University.

76. Yu S.D., Cleghorn W.L. (2002). Dynamic Instability Analysis of High-Speed Flexible Four-Bar Mechanisms. *Mechanism and Machine Theory,* **37**, 1261–1285.

77. Yang K., Youn-Sık P. (1998). Dynamic Stability Analysis of A Flexible Four-Bar Mechanism and Its Experimental Investigation. *Mech. Mach. Theory,* **Vol. 33**, No. 3, pp. 307-320.

78. Tokuz Dulger L. C., Uyan S. (1997). Modelling, Simulation and Control of A Four-Bar Mechanism with A Brushless Servo Motor. *Mechatronics,* **Vol. 7**, No. 4, pp, 369-383.

79. Rittel D., Levin R., Maigre H., Fengfeng X., Sinatra. (1997). Effect of Dynamic Balancing on Four-Bar Linkage Vibrations. *Mech. Mach. Theory,* **Vol. 32**, No. 6, pp. 715-728.

80. Rezaei M., Tayefeh M., Bahrami M. (2006). Dynamic Behavior Analysis of Compliant Micromechanisms*. Journal of Physics: Conference Series 34, 583–588.*

81. Rundgren B. T. (2001). *Optimized Synthesis of A Dynamically Based Force Generating Planar Four-Bar Mechanism*. MS Thesis, Virginia Polytechnic Institute and State University, Blacksburg, Virginia, November 28, 2001

82. Design Simulation Technologies, Inc. (May 2007). http://www.design-simulation.com/WM2D/index.php

83. Design Simulation Technologies, Inc. (May 2007). http://www.design-simulation.com/

84. College of Engineering, University of Michigan. (May 2007). http://www.engin.umich.edu/caen/wls/software/listing/WorkingModel2D/

85. Seattle Robotics Society. (May2007). http://www.seattlerobotics.org/guide /servos.html

86. Penton Media, Inc. & Machine Design magazine. (May 2007). http://www.electricmotors.machinedesign.com/guiEdits/Content/bdeee4a/bdeee4a_1.aspx

APPENDICES

A1)

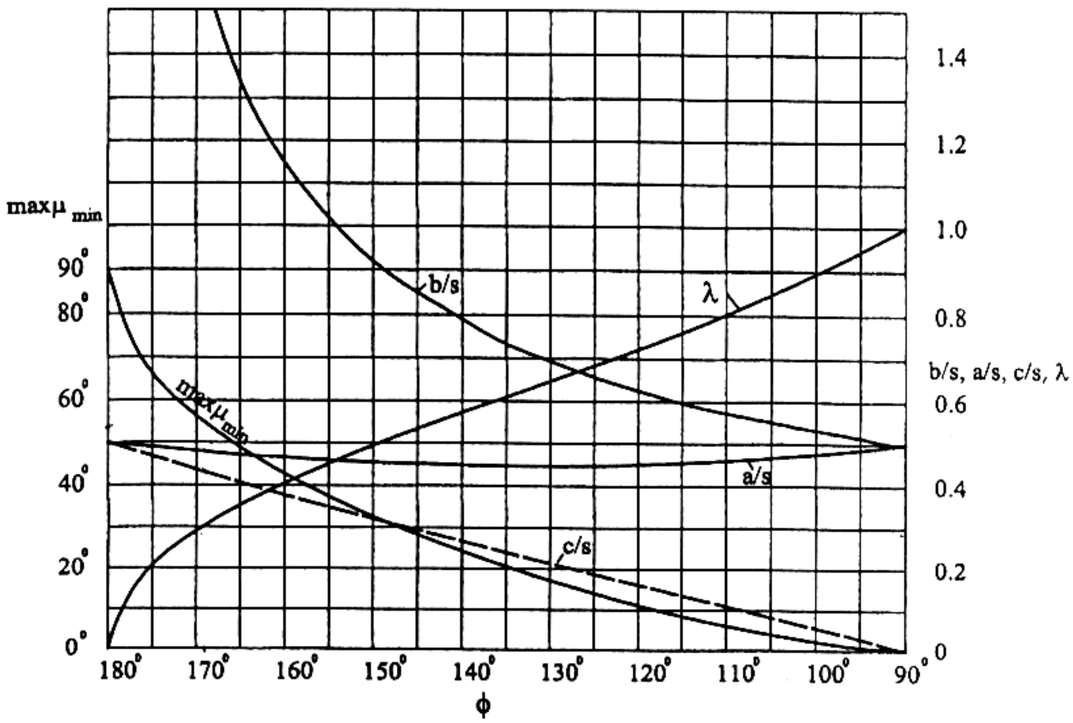


Chart 1. Slider-crank proportions with optimum transmission angle variation for a given crank rotation between dead centers, ϕ [70].

A2)

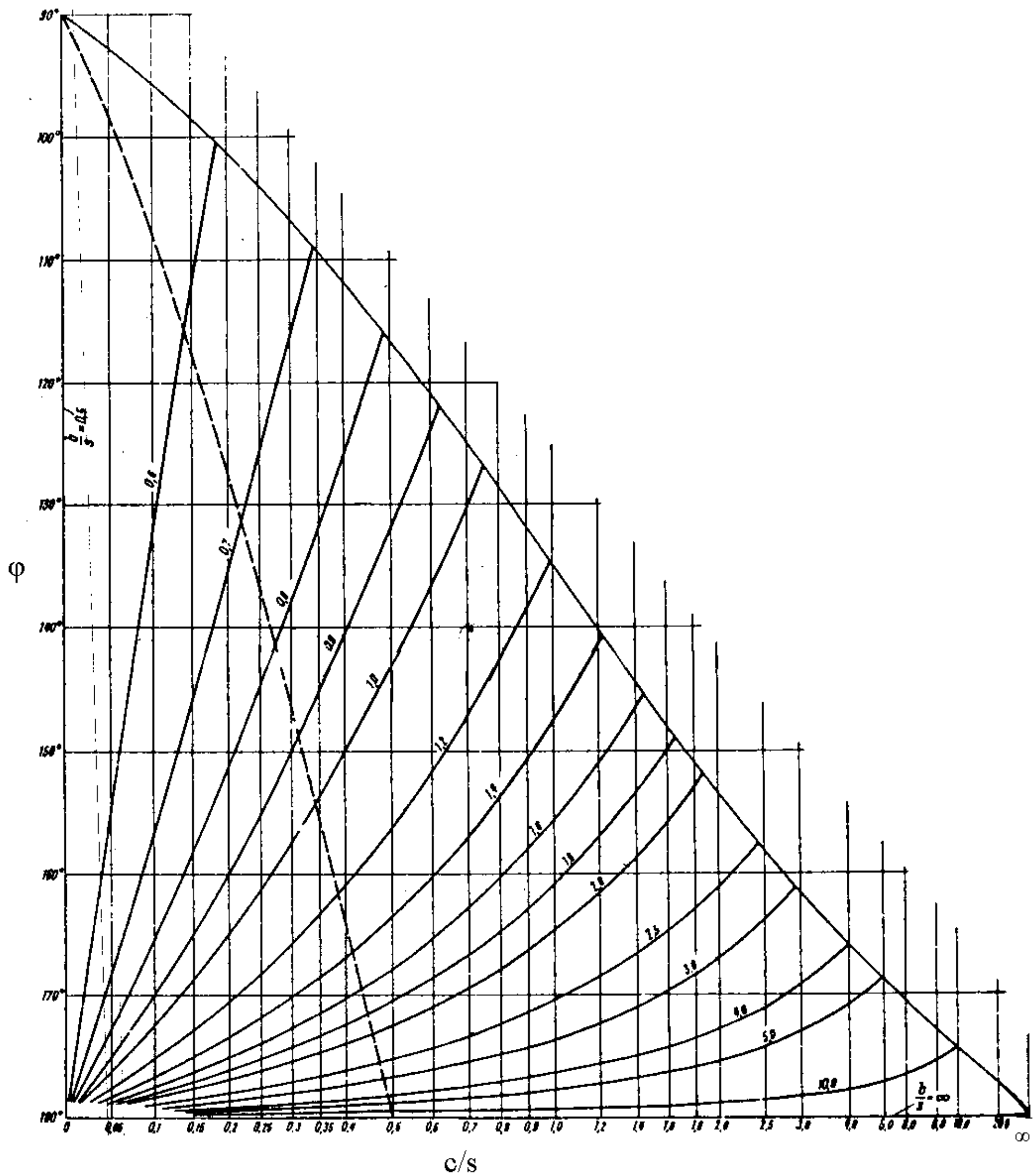


Chart 2. Connecting rod to stroke ratio b/s, corresponding to eccentricity ,c, and corresponding crank rotation between dead centers, ϕ [71].

A3)

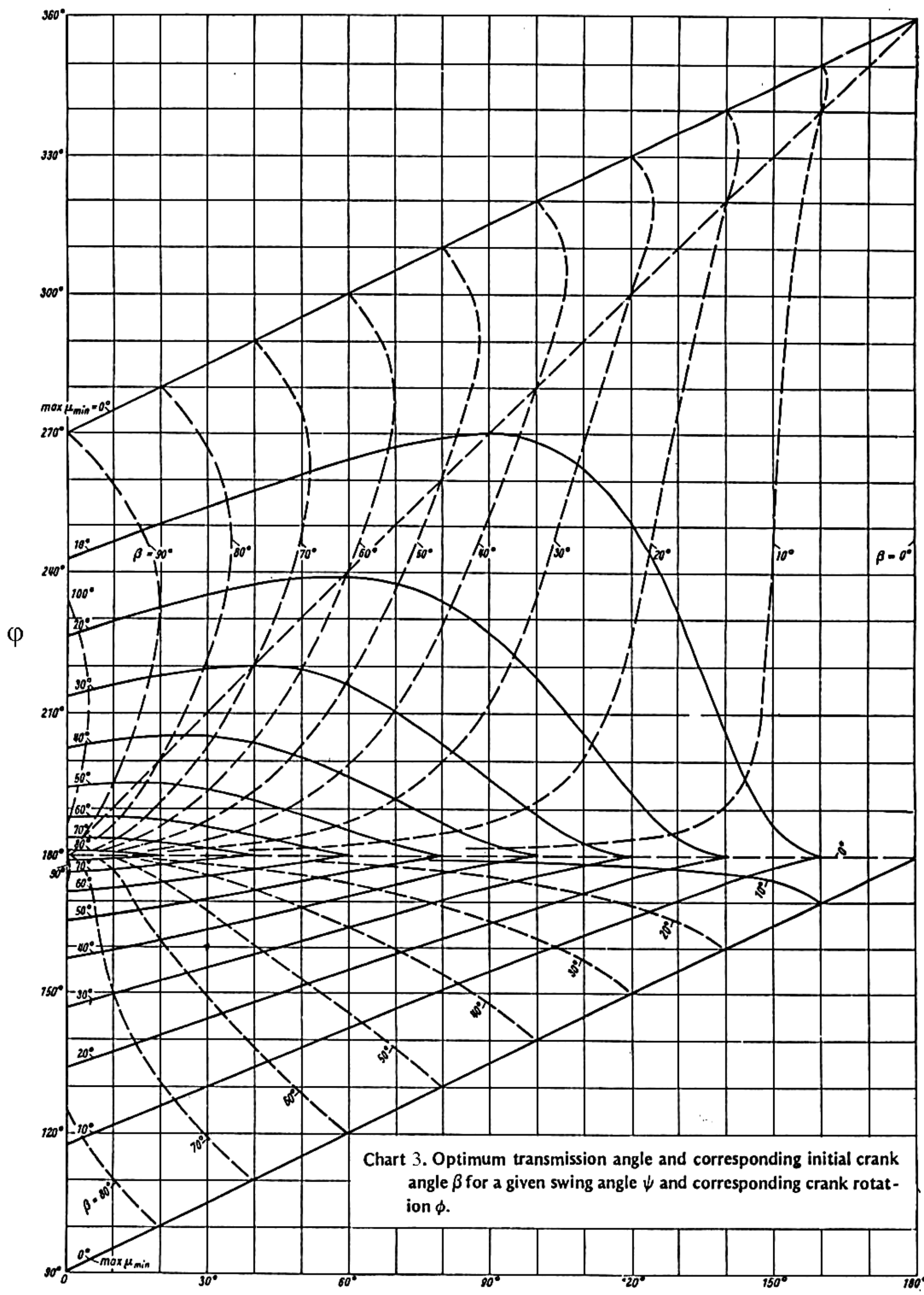


Chart 3. Optimum transmission angle and corresponding initial crank angle β for a given swing angle and corresponding crank rotation [70].