(REFEREED RESEARCH)

SYNTHESIS WORK ABOUT DRIVING MECHANISM OF A NOVEL ROTARY DOBBY MECHANISM

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ABSTRACT

In this paper synthesis works of a dobby driving mechanism had been explained. Dobby experimental set has been constructed sum of three main parts; driving mechanism, program reading and locking unit; and mechanism for elevating motion to the frames. Sythesis work of driving mechanism is completed using analytical and mathematical approaches. Studied dobby construction is a unique fourteen framed rotary dobby mechanism, design works and experimental set building processes are carried out as part of a research project in a collaborative work of Suleyman Demirel University and Pamukkale University. The most important innovation is in the motion transmission mechanism on the main shaft of the rotational dobby. Main shaft is driven by oscilational motion with dwell period. Constructional analysis and synthesis works carried out during the mechanical designing period of the dobby driving mechanism is given in detail. Proposed dobby design can be considered to be manufactured and advised to be used on most of the standart weaving machines.

Key Words: Dobby, Driving mechanism, Rotary dobby, Weaving machinery.

ÖZET

Bu çalışmada armür tahrik mekanizmasının sentez çalışmaları matematiksel ve analitik yaklaşımlarla açıklanmaktadır. Armür mekanizması tahrik mekanizması, program okuma ve kilit ünitesi ve çerçeve kaldırma mekanizmasından oluşmaktadır. Bu mekanizma tasarım ve deneysel model üretimi Süleyman Demirel ve Pamukkale Üniversitelerinin ortak çalışmaları sonucunda gerçekleştirilmiş olan 14 çerçeveli bir rotatif armür mekanizmasıdır. Bu mekanizmada en önemli yenilik rotatif armür ana miline hareket iletiminin yapıldığı mekanizmadadır. Ana mil beklemeli salınım hareketi ile tahrik edilmektedir. Tahrik mekanizmasının mekanik tasarımı sırasında yapılan konstrüktif analiz ve sentez çalışmaları bu yazıda detaylı olarak verilmektedir. Geliştirilmiş olan armür mekanizmasının üretimi ve standart dokuma makinalarında kullanımı mümkün olabilecektir.

Anahtar Kelimeler: Armür, Tahrik mekanizması, Rotatif armür, Dokuma makinası.

Received: 10.11.2009 Accepted: 07.07.2010

1. INTRODUCTION

Shedding motion is one of the basic motions on weaving machine. There are three different principles to perform shedding motion which are cam controlled mechanism, dobby controlled mechanism, and jaquard controlled mechanism. Dobby machine constructions are classified according to their shedding principles, number of stroke, program controlling principle, construction, and motion transmission mechanism to the frames (1). Rotary dobby mechanism, as one of the commonly used dobby types, is known as capable to run up to 1000rpm speeds and capable to be used on all types of weaving machines (2). Driving system of the rotational dobbies are mostly plain and rigid structures which maintains less vibration and high endurance. Physical occupation area of the rotational dobbies is relatively lower than other dobby volumetric bodies. Another important point about rotational dobbies is their high technology required manufacturing processes and high level of manufacturing costs (3,4).

Dobby mechanisms are not one of the mostly studied mechanisms in the literature, there are only a few scientific papers and several patents available for reviewing. Abdulla et al.(5) are introduced an electronically controlled, sixteen harnesses, middle closed, and positive driven dobby construction, which reached upto speed of 604 rpm, with convenient manufacturing cost and acceptable level of working noise. Main innovative value of the proposed dobby is reported as absence of any reciprocating knife-hook pair in the mechanism. Eren et al. (2005) (6) are introduced a comparison study of heald frame motion characteristics generated by rotary dobby and crank and cam shedding motions. They found that higher heald frame maximum velocity and maximum acceleration, as well as a longer approximate heald frame dwell, are generated by the rotary dobby rather than the crank or cam shedding motions, due to the intermittent nature of the rotary dobby shaft's motion. Eren et al.(2009) (2) are introduced for a rotary dobby by which heald frame motion equations and heald frame motion curves are obtained. It is shown that although heald frame motion is mainly determined by the design of the modulator mechanism, the eccentric mechanism has a significant influence on it. The design of the modulator mechanism of a rotary dobby is crucial for controlling the heald frame motion characteristics. Yueyang and Ruiqi (7) are introduced a new type of microprocessor controlled dobby which is converted from dobby of Staubli 2521 model. The new model has been proved to be practicable, simple, and easy to be manufacturing than present positive dobbies.

Present work is about driving mechanism of a novel dobby design. Anaytical, mathematical and experimental approaches are used to explain the driving unit of the 14 framed rotational dobby.

2. ANALYSIS OF ROTARY DOBBY DRIVING MECHANISM

Basic difference that distinguishes the rotary dobbies from the other known dobby constructions is the 180° intermittently rotational movement of main shaft. This constructional difference allows rotary dobbies not to employ any shedding search mechanisms. Principle working positions of the geared-cam driving mechanism are shown in Figure 1 (8,9) schematically,

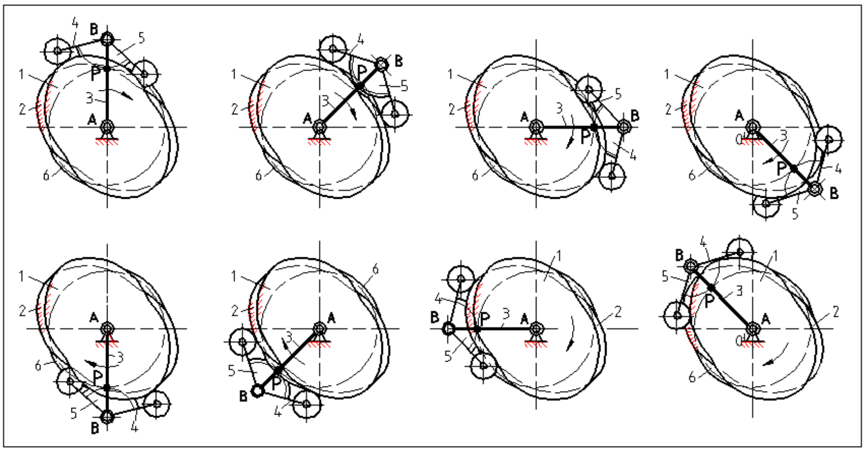

Figure 1. Driving mechanism of rotary dobby and principle working positions.

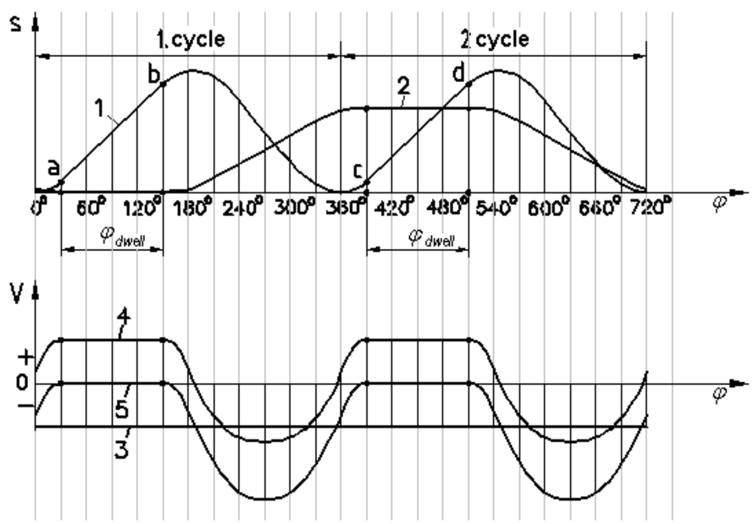

Figure 1. Displacement and velocity diagrams of rotational dobby mechanical parts: 1:Displacement diagram of follower 3; 2:Displacement diagram of gear 6; 3:Velocity diagram of oscillating follower 3 at the (AB-R_5) point 4:velocity diagram of gear 5; 5: velocity diagram of gear 6

Main parts of the mechanism are; dual cams **1** and **2**-also known as conjugate cams- which are stabilized at the A rotation center; oscillating follower **3**, roller **4** - mounted at the **A** and makes constant speed rotational movement around the cams, gear transmission of **5** and **6** mounted at the **A** center and **B** movable junction. Since cams **1** and **2** are inert, roller **4** is capable to make two different motions, one is rotational movement around A point and the other is oscilational motion resulting movable connection to B junction. Displacement and velocity diagram of the mechanism elements are shown in Time utilization for one whole period of the dobby main shaft is equal to two whole period of weaving machine. a-b and c-d parts of the follower **3** displacement diagram has been designed to make linear motion to maintain constant angular velocity of ω_3 –line 3 of velocity diagram. Mechanism's degree of freedom is;

$$W=3n - 2p_5 - p_4 = 3x3 - 2x3 - 2 = 1.$$

Velocity vectors of oscillating follower **3** and roller **4** are opposite to each other resulting rotational motion with stand-by positioning of gear **6**-line 5 of (10)

When the diameters of gear **5** and gear **6** are defined as R_5 and R_6 respectively and rotation angle of follower **3** and roller **4** are defined as φ_3 and φ_4 respectively, below equation can be written considering the stand-by stipulation of mechanism.

$$R_5\ \varphi_4 = R_6\ \varphi_3 \text{ or } i_{tr} = \varphi_4 / \varphi_3 \qquad (1)$$

Since value of φ_3 and φ_4 are known, gear transmission ratio of i_{tr} is determined as seen above formula *(1)*. $i_{5\text{-}6}$ is determined as 0.5, when dwell angle of loom main shaft is 120° (dwell angle of cam main shaft is 60°) and rotation angle of roller **4** is 30°. In order to maintain dwell positioning of gear **6**, velocity vector (V_5) of gear **5** and velocity vector (V_P) of oscillating follower **3** should be equal and opposite to each other.

Velocity vector of (V_5) and (V_P) are shown in as curves 3 and 4 respectively. Velocity curve of (V_6) ($\vec{V}_6 = \vec{V}_5 + \vec{V}_P$) is also shown as curve 5 in. Integral of this curve is displacement law of the frame elevation (second graph in). Line course of this diagram is similar to those of common shedding mechanisms of weaving machines. (7,11) It is known that one way stand by rotational movement of rotary dobby main shaft does not have any relation to shedding stipulations of the loom (12) It can be concluded that 180° oscilational motion of the rotary dobby main shaft does not cause any interruption in the accomplishment of shedding motion.

As results of the kinematic and constructional analysis, below listed themes can be summarized concerning comparison of stand by positioned oscilational motion and one way stand by positioned rotational motion;

- The number of mechanism that employs 180° equally timed, intermittently, rotational motion mechanism is limited while use of such rotational mechanisms that make equally timed, double stand-by positioned oscilational motion in every 180° is common.
- Constructional structure of 180° equally timed, intermittently rotational motion mechanisms is complicated manufacturing, their manufacturing process is expensive, and they require high technology.
- Program reading and processing units of 180° intermittently rotational motion dobby constructions are quite complicated units. Dobby constructions, employing oscilational motioned with stand-by positioned mechanisms, require simple programming and processing unit, similar to those of employed on the common double stroked dobby mechanisms.
- Intermittently rotational motion dobbies require special mechanical parts and rollers in the motion transmission mechanism to the frames. Oscilational motioned with stand-by positioned mechanisms make possible use of standard machine parts and rollers.

3. SYNTHESIS OF PROPOSED DRIVING MECHANISM

Since the frame elevation mechanism of shedding motion is designed based on well known motion laws its constructional calculation phases is not described in detail. Synthesis works of the oscilational motion double cam mechanism is explained in detail (4) (Figure 3)

Principle parameters of dobby construction are designated considering below mentioned preliminary conditions;

- Mechanism should be designed to be placed between main shaft of the loom and main shaft of the motion transmittance mechanism to the frames.
- Dimension of the mechanism should be compatible with the other mechanisms.
- Adjustment, assembling and de-assembling of the mechanism should be easy to handle.
- Mechanism should not employ any complicated and expensive transmission mechanisms.

Mechanism has been built up employing dual cams **1** and **2**, oscillating follower **3** carrying rollers **4** and **5** and gear transmission of **6** and **7**. Gear **6** has been assembled to main shaft of the dobby, it makes 180° rotational movement. Kinematical relations of mechanical parts;

$\varphi_3 = \varphi_6 \ / \ i_{6\text{-}7}$

or

$i_{6\text{-}7} = \varphi_6 \ / \varphi_3 = R_7 / R_6 \quad (2)$

Gear transmission between gears of **6** and **7** has been designed as internal transmission in order to minimize the dimension of the mechanism and be positioned in the dimensional limits of A and B points.

Working principle of the mechanism can be explained as follows; (Figure - b) dual cam **1** and **2** are driven by main shaft of the weaving machine with the rotational ratio of 1:2 and as result of it oscillating follower **3** makes oscilational movement around the B axis. Gear **6**, assembled to the follower **3**, rotates with the follower and transmits the rotational motion to the gear **7** which is mounted on to the dobby main shaft. Working principle of rotational dobby mechanisms guide us to choose 180 ° as oscillation range of gear **7**; and 30°- 45° oscilational motion range for cam mechanisms which carry oscillating follower. Transmission rate of gears **6** and **7** therefore can be either φ_3 = 30° for $i_{6\text{-}7}$ = 6 or φ_3 = 36° for $i_{6\text{-}7}$ = 5 or φ_3=40° for $i_{6\text{-}7}$ = 4,5 or φ_3 = 45° for $i_{6\text{-}7}$ = 4. Rise and return angle of cam are wide enough allow us to choose value of φ_3 at the range of its upper limit. Value of φ_3 has been chosen as 45° in the designed construction.

4. DESIGN AND MANUFACTURING OF THE MECHANISM

Experimental set of the designed mechanism has been built up using commonly used, non-expensive machine elements. Instead of using dual cam mechanism and internal gear transmission mechanisms, which were expensive and complicated to construct, shape closed cam mechanism and external gear transmission mechanism have been employed. (Figure 2) Mechanism is built up using a shape closed cam **1**, follower **2**, and gear transmission mechanism of **4** and **5**.

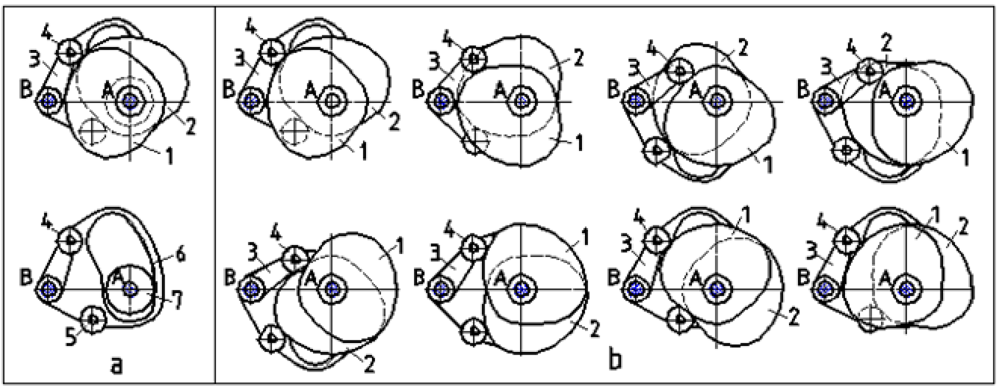

Figure 3. Principle schemes of oscilational motion driving mechanism
a – main parts of mechanism, b –positions explaining working principle of the mechanism

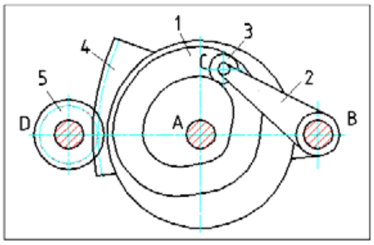

Figure 2. Driving mechanism of experimental set

In the mechanism, oscilating angle of follower is φ_2= *30°*; engagement angle $\gamma \leq 40°$; transmission ratio of $i_{4\text{-}5}$ = 1/6; adjustment range of AB distance is ± 1mm; adjustment range of BC distance is ± 0,5mm; and displacement formula is determined as follows;

$$s = S_{maks}\left[\frac{\varphi_i}{\varphi_y} - \frac{1}{2\pi}\sin\left(2\pi \cdot \frac{\varphi_i}{\varphi_y}\right)\right] \qquad (3)$$

4.1. Synthesis Work about Main Parameters of the Mechanism

Determination of distance between cam center and rotation axis of follower –AB- was carried out as it is explained in the following parts. Velocity of experimental set was assumed below 500m/min and sinusoidal displacement law was employed and minimum radious of cam was calculated as $R_0 \geq 1,46s$. C_1 and C_2 points are determined drawing a intersection line from the cam rotation center A to $R_{internal}$ ($R_{int}= R_0+r$) and $R_{external}$ ($R_{ext}= R_0+r+s$) curves of *1* and *2*. (Figure 3)

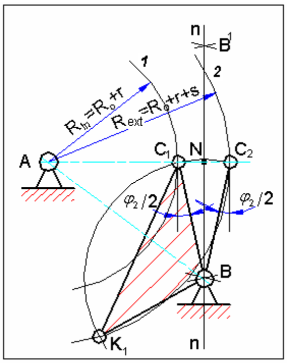

Figure 3. Synthesis of cam mechanism

Since cam mechanism follows the symmetric motion law, oscilational axis of the follower is placed on the line perpendicular to the line of C_1 and C_2 . Oscillation center of the follower is called B and it is determined by crossing the lines of $\varphi_2/2$ angle to verticals from the C_1 and C_2 points. Boundary conditions of the mechanism are determined using the lines from C_1 and C_2 points to B point. Intersection of B centered BC_1 curve with the curve 2 determines the place for second shoulder of follower, which is named N.

Below equations are concluded from Figure 4;

$$AB = \sqrt{AN^2 + NB^2} \qquad (4)$$

$$AN = R_0 + r + s/2\,,$$

$$s = 2CB\sin(\varphi_2/2)\,,$$

$$NB = CB\cos(\varphi_2/2) \qquad (5)$$

In the equations, φ_2 and radius of the roller (r = 26 mm) have been known; and value of the *s* is determined depending on length of the BC distance and φ_2 angle. (Figure.6) In the figure, d_1 and d_2 are axis diameters; D_1 and D_2 are bearing diameters; b_1 and b_2 are housing thicknesses.

Below formula is used to calculate length of BC distance.

$$BC \geq d_1/2 + d_2/2 + D_1/2 + D_2/2 + b_1/2 + b_2/2 + \Delta \qquad (6)$$

In the equation, Δ is an additional length determined depending on construction. Other parameters of the equation are determined using kinetostatical calculation. However during the synthesis of the mechanism, instead of using such calculations, empirical values are used. Analyses of other shedding mechanisms have brought the following empirical values for calculation of BC arm of Figure 6; $d_1 \geq$ 30mm, $d_2 \geq$ 20mm, $D_1 \geq$62mm, $D_2 \geq$47mm, $b_1 \geq$8mm, $b_2 \geq$6mm and Δ =0mm and length of arm (BC ≥ 69) can be determined using equation *(6)*. In the experimental set length of the BC is assumed as 70mm; using equations *(4)* and *(5)* with the values of

s =36.24 and

r = 26 mm

R_0 =1,46 x 36.24 =52.91 mm; *NB* = 67,62 mm; *AN*=97.03mm; and

AB = 118.26 mm

are determined.

AB value is then assumed as 120 mm; and R_0 = 55mm, and shoulder angle of the follower is β = 112°

Technical drawing of the constructed cam mechanism is seen in Figure 7.

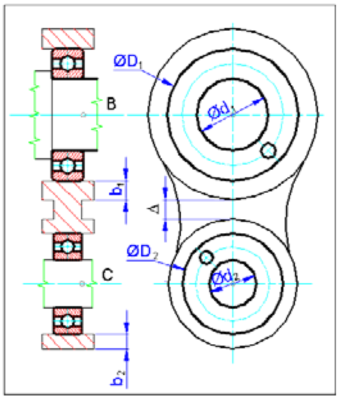

Figure 6. Determination of follower length

5. EXPERIMENTAL WORK

Experimental set has been constructed sum of three main parts; driving mechanism, program reading and locking unit; and mechanism for elevating motion to the frames. 1:1 scale of the constructed experimental set (Figure 8) allows us to assembly it to weaving machine. Mechanisms are assembled to the body of the machine and it is driven by 0.6kw speed controlled electric engine.

Three main sets of experiments have been conducted in order to evaluate the possibility of weaving using proposed dobby construction. These experiments are;

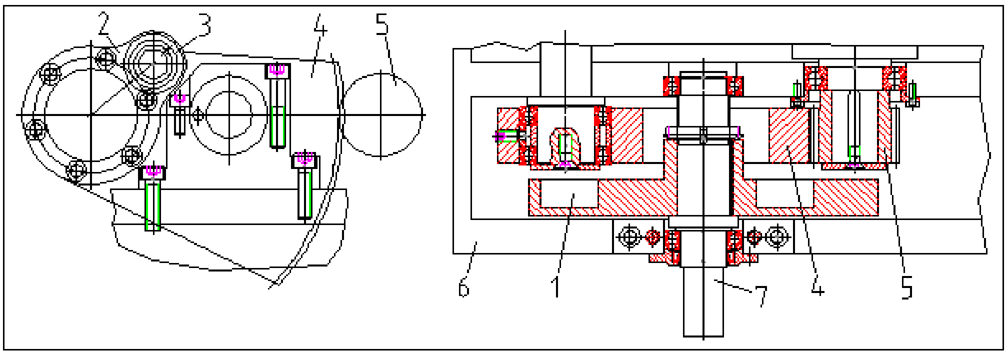

Figure 7. Construction of dual cam, cam **1**, follower **2**, roller **3**, gear **4**, pinion gear **5**, main body **6**, cam shaft **7**

- measurement of main shaft oscillation angle and determination of adjustment parameters,
- evaluating consistency of the mechanisms to displacement diagrams,
- determinations of velocity limits of the mechanism

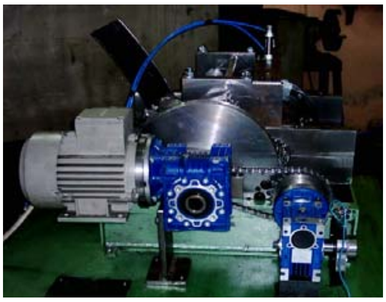

Figure 8. Picture of the experimental set

5.1. Measurement of oscillation angle and determination of follower arm adjustment parameter

Driving unit of the frame elevation mechanism is continuously moving mechanism that makes one way rotational movement, oscillational movement or rotational oscillation movement constantly. Continous moving of driving mechanism and velocity changes of main shaft causes some small gaps at the mechanical junctions. Some of these gaps should not exceed the tolerance limits of the locking mechanism (T_{loc}). It has been reported that tolerance limit of such mechanical gaps should be between 0,6-1,0 mm.' (9, 13) Length of the follower arm has been designed adjustable to correct any mechanical gaps of the mechanism.

Adjustment of oscillation angle has been controlled by two ways rotational movement possibility of eccentric. Rotation of main shaft has been controlled manually. It has been experimentally determined that 180° ± δ oscillational motion of main shaft can be provided by ± 0,5 mm adjustment value of follower arm.

Maximum mechanical gap distance of the whole mechanism has been reduced to the knife, ignoring the deformation of the linkages, and below formula has been used to determine the distance;

Δ_{red}= (R_{gear} / r_{fol} x δ_c + δ_{oil}) x R_{knife} / r (6)

Δ_{red} = [(216 x 0.02)/70+0.02] x 123.5 / 32 = 0,315 mm

Where ;

R_{gear} : radious of the gear, r_{fol} : radious of the follower, δ_c = working and oiling gap distance between follower and cam nesting, δ_{oil} : oiling gap in gear clamp, R_{knife} : osilation radious of knife, r : radious of pinion gear.

When the $\Delta_{red} < T_{loc}$ is constituted, it is known that locking condition is provided. In experimental works, it is seen that locking process has been completed.

5.2. Evaluating consistency of the mechanisms to displacement diagram

Although consistency evaluation of the mechanism is not directly related to constructional features of the driving mechanism, it is important to understand endurance, stability, and working speed limits of the mechanism.

Purpose of the experiment is to understand the relationship between working phases of the locking mechanism and make precise statements of adjustment parameters. At the first stage of the experiments, program reading and locking mechanism has been activated and main shaft of the mechanism is manually driven to make timing adjustments of the mechanism. Surface of the programming disk is clefted and adjustment is made loosing the connection bolts of program disk and shaft (14).

Experimental findings of the evaluation are shown in Figure 9. Appropriate time range for running the locking unit are shown as squared areas of φ_k ranges, t_2, on the diagram. Locking mechanism starts to be active from the points *1,* and stays active upto the points *2.* Program reading and practicing processes should be completed in this time interval; locking mechanism should be turned off, if it was turned on or vice versa. φ_{po} range, t_1, is reserved as safety time interval for program reading and φ_e range, t_3, is reserved as safety time interval for complete the locking up process. Since the locking mechanism has been built up using electro-mechanic elements, φ_{po} and φ_e time intervals of total φ_{dwell} range are reserved as wide as possible to tolerate any presumptive mechanical detention (12).

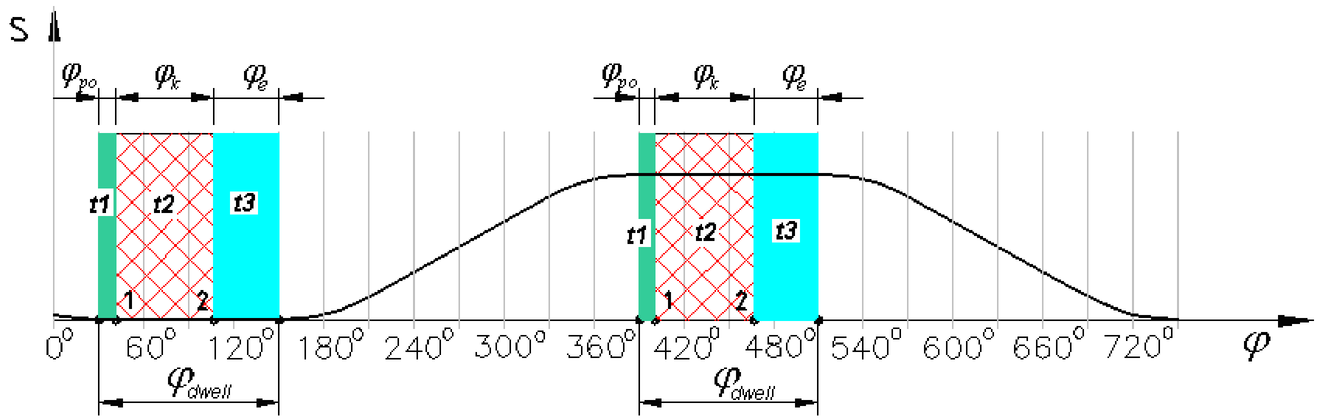

Figure 9. Displacement diagram

Table 1. Results of experimental works to find out t_1 reserve time of program reading, *a- program reading performance is not good enough, b- program reading performance is acceptable, c- program reading performance is satisfactory.*

Trial number	Rotational angle of main shaft									
	1°	2°	3°	4°	5°	6°	7°	8°	9°	10°
1	a	a	b	b	b	b	c	c	c	c
2	a	a	b	b	b	c	c	c	c	c
3	a	b	a	b	c	c	c	c	c	c
4	b	a	b	b	b	c	c	c	c	c
5	a	a	a	b	b	c	c	c	c	c
6	a	b	b	b	c	b	c	c	c	c
7	a	b	b	b	b	c	c	c	c	c
8	a	a	b	b	c	c	c	c	c	c
9	b	b	a	b	c	c	c	c	c	c
10	a	a	b	b	b	b	c	c	c	c

Timing adjustments has been accomplished rotating the program disk in clockwise or counter clockwise direction at the inert condition of the main shaft. (12)Each experiment set has been repeated three times for each one degree of angle in the range of 1° to 10°. Experimental findings, in Table.1, show us that t_1, φ_{po} range, should be equivalent toβ_1 > 6° rotation of cam shaft.

Reserved time, t_3, φ_e range, for accomplishment of the locking process should be long enough to complete mechanical boundary conditions of the locking mechanism. Principle of the locking mechanisms can be conducted using pnoumatic cylinders or mechanical spring cyclinders, which influence t_3, reserved time interval of locking up process.

Determination of reserved time;

Proposed program reading mechanism is employing pneumatic cylinders. Designation of t_2 locking time requires the displacement distance of locking snag, which is max. 10 mm; piston velocity, which is higher than 1mm/s; therefore t_2 locking time is determined as less than 0,01s. For the 1000rpm working speed of the dobby mechanisms, rotation angle of the cam shaft can be designated as 60°. t_3 locking accomplishment time of the mechanism is equivalent to subtraction of t_1 and t_2 . Since dwelling time of the cam shaft is 120°, $t_3 \leq$54°.

5.3. Velocity trials

Purpose of the velocity trial is to test synchronized working condition of the mechanisms and locking mechanism. Since mass balancing mechanism has not been employed on the exprimental set, velocity trials has been limited up to 400t/min. β_1 has chosen as parameter for the experiment and value of the β_1 and rotation speed level have been changed in the range of 1° - 10°, and 10-400rpm respectively. Performance of the machine is visually recorded. Evaluation of the visual records has shown that $\beta_1 \geq 7°$ gives the best performance of mechanism concerning shedding process and reliable working conditions of the machine elements.

6. RESULTS AND CONCLUSION

1- Performance of the rotary dobby shedding mechanisms is not related to the one way, stand-by rotational motion of the main shaft, it has been proven that 180° oscilational, stand-by motion of the main shaft does not also influence the performance of shedding process.

2- Design parameters of a novel driving mechanism are designated.

3- Accomplished experimental works on the constructed model of the proposed dobby have shown that working conditions of driving mechanism and other units of the dobby are reliable, stabilized and substantive.

4- Proposed dobby design can be considered to be manufactured and used on most of the standart weaving machines.

Experimental and theoretical results of the 14 framed novel rotary dobby mechanisms are found quite optimistic. Constructional and mechanical possible problems are seems solved from the view of experimental measurements on the model dobby construction.

Any further work about the rotational dobby construction might be the manufacturing of real sized prototype dobby construction.

ACKNOWLEDGEMENTS

The authors are grateful to TUBITAK, DPT, Suleyman Demirel University and Pamukkale University for their financial and technological supports.

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*Bu araştırma, Bilim Kurulumuz tarafından incelendikten sonra, oylama ile saptanan iki hakemin görüşüne sunulmuştur. Her iki hakem yaptıkları incelemeler sonucunda araştırmanın bilimselliği ve sunumu olarak **"Hakem Onaylı Araştırma"** vasfıyla yayımlanabileceğine karar vermişlerdir.*