Gazi University Journal of Science
GU J Sci
23(2): 227-232 (2010)

GUJS
www.gujs.org

Cam Motion Tuning of Shedding Mechanism for Vibration Reduction of Heald Frame

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Received: 20.02.2009 Revised: 20.09.2009 Accepted: 21.09.2009

ABSTRACT

This paper presents a new approach to eliminate the residual vibration of shedding system and the associated mechanism that connects the heddle shafts with the cam mechanisms. Effective cam design method which is based on motion design of the cam pitch curve by "blends" of the traditional profiles, formed by superimposing three functions; a cycloid, a ramped versine, and a ramp called as ramp-plus-ramped cycloid-plus-ramped versine is proposed in this study. This function is reshaped using the system's natural frequency and damping ratio to yield almost zero residual vibration of the heddle shaft. The proposed method is then compared with the well known cam curves and its applicability is shown by simulation and experimental results. The results show that the proposed method is quite effective for fast vibration free motion of the heald frame.

Key Words: *Shedding system, Cam tuning, Motion design, Residual vibration elimination*.

1. INTRODUCTION

Weaving is the process in which yarn is converted into fabric. The process of weaving is divided into three fundamental motions which are; shedding is the dividing longitudinal threads called 'warp' into two sheets; picking is the insertion of transverse thread called 'weft' into the space created by the division of warp sheets and beating is the pulling inserted wefts one after the other to form cloth. However almost in all system cam accomplish shedding and beating mechanisms. Shedding plays a vital role in the conversion of yarn into fabric, its function is to divide the warp sheet into two parts in the form of a shed, so that the weft can pass through the shed.

High-speed, high-precision weaving machines are becoming increasingly important in harsh completive world. Effective cam design methods are required in order to develop such superior machines. Faster machines cause problems such as inaccuracy, vibrations and wear. A major cause of these problems is motion design of the cam-follower mechanism. Generally speaking, the cam is designed for a constant drive speed and a flywheel attached to keep the drive speed as constant as possible. Cam design practice, based on acceleration curves defined by trigonometric, polynomial, or spline functions of the rotation angle, yields required profiles of the output motion [1-6]. The design of cams for dynamic performance depends critically on identifying curvilinear profiles that yield desired characteristics of the output motion. So, a differential analysis requires for proposed cam profile.

In almost all the present weaving systems shedding is mechanical and is done by cams for periodical motions to move heald frame dwell-rise-dwell motions to form a shed. Although the most harmonic possible motions are sought and achieved, nevertheless vibration which is then called residual vibration occurs in the cam shedding system and the associated mechanism. This vibration puts a load on all the elements of the shedding system and causes increased noise, wear, heddle breakage, warp yarn breakage, and operating costs. In

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general, residual vibrations are not desired and need to be avoided. Various concepts with a view to reducing wear in the shedding system and reducing residual vibration have been developed. John Rees Jones [7] has proposed a method to avoid residual vibrations of the elastic cam follower system. This method is merely to give an initial velocity to the system, which generates a residual vibration equal in amplitude, but out of phase to the original oscillations imposed by the gross motion, hence the two canceling each other. Initial velocity is obtained by applying a ramping displacement throughout the gross motion. A ramping displacement is actuated by infinite magnitude-zero duration impacts at the beginning and end of the motion. John Rees Jones [7] calls the modification of the cam profile as such, tuning of' the cam. Alici and Bayseç [8] have presented the applicability of the aforementioned method to avoid the residual vibration of a dynamic load of the manipulator links. Yan et al. [9, 10, 11] has proposed a method to improve kinematic and dynamic characteristic of cam system with integrating computer controlled servomotor. This method requires complicated and very expensive systems. Anderson and Singhose [13] have presented the use of input shaping on cam profiles to reduce vibration. Their method is based on convolving original command signal by sequence of impulses to create a new function that drives system with very little vibration.

This work presents a dynamic analysis of the mechanism of a cam shedding motion of the loom. Approach towards cam design is significantly different from the traditional ones and provides a way to develop machines with high drive speeds. Proposed approach is based on motion design of the cam by "blends" of the traditional profiles, formed by superimposing three functions; a cycloid, a ramped versine, and a ramp. A cycloidal motion profile is commonly used as a high-speed cam profile, continuous in all derivatives throughout the complete cycle. A ramped+versine function is generated by adding a ramp onto a versine function which has no discontinuity in slope at the beginning and end. These two motion templates are superimposed onto a ramp function whose duty is to give an initial velocity to the system. Given the total height to be traveled and the cam angle or transportation time, and using correct system parameters, the amplitudes of three functions are calculable. When this cam profile, the so-called 'cycloid+ramped versine+ramp function', is given to the system, the oscillations of the three functions cancel each other out and an oscillation reduce free motion is obtained.

2. MODELING OF THE SYSTEM

Most elastic mechanical systems are composed of masses moving under the action of position and velocity dependent forces and hence can be modeled by a second order differential equation. To calculate the dynamic response of a mechanical system, a complete dynamic analysis must be carried out. A dynamic model must be created; then the equations of motion for the system can be derived. Typically these are differential equations, which must be solved numerically to calculate the dynamic effects on the output motions. Dynamic modeling is a method of describing a physical system by a mathematical model. It is necessary to understand what a mechanical system is actually doing when the dynamics of the system are introduced. There are many different methods to mathematically model a mechanical system; a lumped parameter model can be described as a simplification of a mechanical system to an equivalent mass, equivalent stiffness and equivalent damping. Simplified model of a complicated system is often used in dynamic analysis [6].

Although the real shedding system depicted in Fig. 1, and the equivalent system are different physically, they can be represented by the same mathematical model. Such systems are called equivalent or analogous systems. In this study, we make use of the mathematical model of an inertia element movable in a single coordinate $x(t)$ under the effect of position and velocity dependent forces and cam displacement $y(t)$. The equation of motion of such a system with damping ratio ζ and natural frequency ω_n can be written as;

$$\ddot{x}(t) + 2\zeta\omega_n\dot{x}(t) + \omega_n^2 x(t) = 2\zeta\omega_n\dot{y}(t) + \omega_n^2 y(t) \qquad (1)$$

where natural frequency is $\omega_n = \sqrt{\frac{k_e}{m_e}}$, and damping ratio is $\zeta = \frac{c_e}{2\sqrt{k_e m_e}}$.

Employing dimensionless quantities into the equation 1 can be written as

$$\chi''(\tau) + 4\pi n\zeta\chi'(\tau) + (2\pi n)^2\chi(\tau) = 4\pi n\zeta y'(\tau) + (2\pi n)^2 y(\tau) \qquad \ldots..(2)$$

where primes will be used as alternative representation for the derivatives with respect to normalized time, τ and the dimensionless parameter, n, is the ratio of the rise of the cam, t_r, to the natural period of vibration of the follower, t_n;

$$n = \frac{t_r}{t_n} \qquad \ldots..(3)$$

Figure 2 depicts the definition of the variables in equation 3.

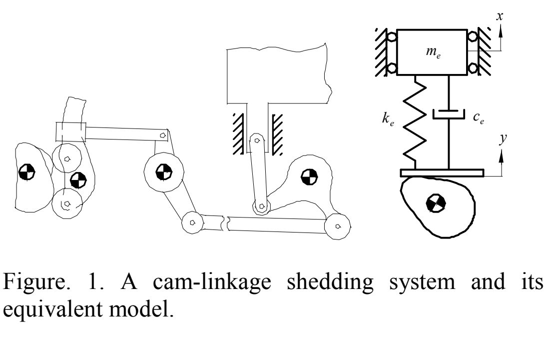


Figure. 1. A cam-linkage shedding system and its equivalent model.

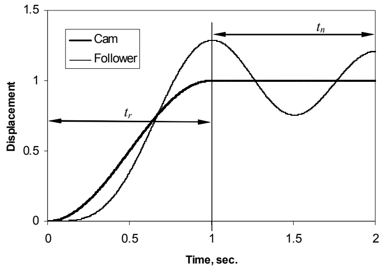


Figure. 2. Cam profile and follower displacement.

3. PROPOSED CAM PROFILE AND TUNING

The proposed cam profile is designed to consist of three functions. The total rise or return distance to be covered from beginning to end of a move within a specified time is the sum of the distances to be traveled by each of the three functions within the same travel time. By adjusting the rise or return distance of each function, vibration can be eliminated provided that the specified move time and the total distance are unchanged. Each component of the reference input creates oscillations such that they cancel each other and no vibration results.

A cam motion profile of a cycloid-plus-ramped versine-plus-ramp function is expressed as;

$$Y(t) = \frac{L_1 \omega t}{2\pi} + \frac{L_2}{2\pi}\left[\omega t - \sin(\omega t)\right] + \frac{L_3 \omega t}{2\pi} + \frac{L_3}{2\pi}\left[1 - \cos(\omega t)\right] \quad \ldots(3)$$

where L_1 is the maximum distance to be traveled by ramp motion profile, L_2 is the maximum distance to be traveled by a ramped cycloid motion profile, L_3 is the maximum distance to be traveled by ramped versine motion profile, t is time into motion, ω angular velocity of the cam. Furthermore, the total distance can be written as $L = L_1 + L_2 + L_3$, and then, arranging the equation above in normalized form,

$$\frac{Y(t)}{L} = \frac{\omega t}{2\pi} - \frac{L_2}{2\pi L}\sin(\omega t) + \frac{L_3}{2\pi L}\left(1 - \cos(\omega t)\right) \quad \ldots..(4)$$

The solution of the equation of motion (2) under the effect of the positional input equation (4) and corresponding velocity yields the following excursion distance values ($\frac{L_1}{L}$, $\frac{L_2}{L}$, and $\frac{L_3}{L}$) for zero residual vibration with zero initial conditions [12].

$$\frac{L_1}{L} = \frac{\omega(\omega - 2\xi\omega_n)}{\omega_n^2} = \frac{t_n(t_n - 2\xi t_r)}{t_r^2} = \frac{1 - 2\xi n}{n^2}$$

$$\frac{L_2}{L} = \left(1 - \frac{\omega^2}{\omega_n^2}\right) = \left(1 - \frac{t_n^{\;2}}{t_r^2}\right) = \frac{n^2 - 1}{n^2} \quad \ldots..(5)$$

$$\frac{L_3}{L} = \frac{2\xi\omega}{\omega_n} = \frac{2\xi t_n}{t_r} = \frac{2\xi}{n}$$

Note that $1 = \frac{L_1}{L} + \frac{L_2}{L} + \frac{L_3}{L}$. Variations of L_1, L_2, and L_3 is possible with angular velocity of the cam to result in an oscillation-free displacement of the system.

4. SIMULATIONS

Here, residual vibration of a shedding mechanism is analyzed by feeding three different cam profile based on the methods of:

- Simple harmonic motion (SHM) rise curve for the cam profile

$$\frac{y(\tau)}{L} = \frac{1}{2}\left(1 - \cos(\pi\tau)\right) \quad \ldots..(6)$$

- Cycloidal motion (CYCM) rise curve for the cam profile

$$\frac{y(\tau)}{L} = \tau - \frac{1}{2\pi}\sin(2\pi\tau) \quad \ldots..(7)$$

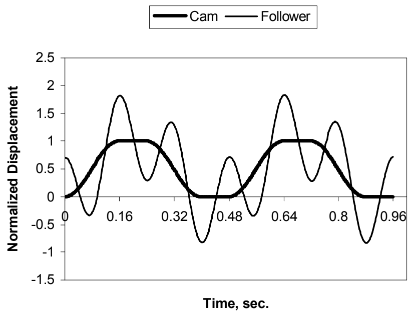
- Optimized excursion distances (equation 5) of the cam rise profile made up from a ramp plus ramped cycloid plus ramped versine

$$\frac{y(\tau)}{L} = \tau - \frac{L_2}{2\pi L}\sin(2\pi\tau) + \frac{L_3}{2\pi L}\left(1 - \cos(2\pi\tau)\right) \quad \ldots..(8)$$

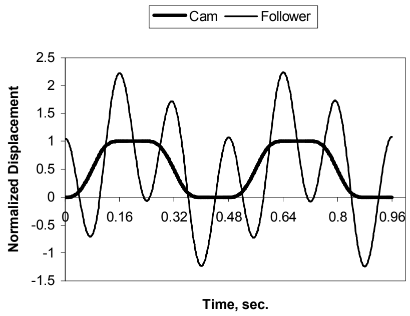
Differential equations of motion given in equations (2) are solved on a digital computer to show the applicability of the method for arbitrarily selected dimensionless time ratio of $n = 1.00$, damping ratio $\xi = 0.045$ and natural period of vibration of the follower $t_n = 0.16$ sec. Cam and follower system design requirement is assumed to satisfy the following condition for illustrative example; a heald frame rises through a displacement of unity for 120°, and dwells while the cam rotates 60°. The heald frame again in 120° degree returns and then dwells for the final 60°. Note that for the speed of the cam is chosen as125 rev/min, period and rise time of the cam described above becomes 0.48 sec. and 0.16 sec., respectively. The same data is used in Section of experimental verification.

Firstly, simple harmonic motion (SHM) curve for the cam profile is used for the motion of heald frame. The steady state response of the system together with the cam profile is depicted in Figure 3a for shedding motion. Secondly, the analysis of the heald frame motion is carried out for cycloidal motion (CYCM) curve for the cam profile. Simulation response of the system together with cam curve is shown in figures 3b. As seen in figures 3a, and 3b, during the dwell periods residual vibration of the heald frame exists. These oscillations are the residual vibrations left in the dwell segments after a rise or a fall.

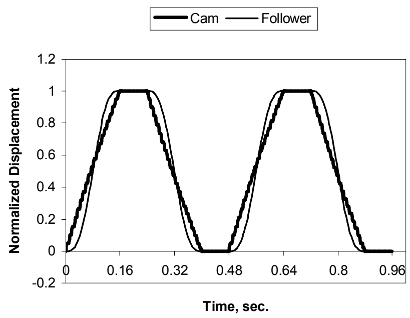
Simulation response of the system together with proposed cam curve is shown in figures 3c. As seen in figures 3c, during the dwell periods residual vibration of the heald frame is eliminated by modifying cam profile with excursion distance values of $\frac{L_1}{L}$, $\frac{L_2}{L}$, and $\frac{L_3}{L}$.



(a)



(b)



(c)

Figure. 3. Simulation results of a cam-linkage shedding system model for time ratio of $n = 1.00$, damping ratio $\xi = 0.045$, natural period of vibration of the follower $t_n = 0.16\,\text{sec}$ and cam rotating at particular speed of 125 revolutions per minute. a) Simple harmonic motion (SHM) cam profile. b) Cycloidal motion (CYCM) cam profile. c) Proposed motion (PM) cam profile by modifying excursion distance values of $\frac{L_1}{L} = 0.9052$, $\frac{L_2}{L} = 0.0051$, and $\frac{L_3}{L} = 0.0898$.

The result in Figure 3 only depicts the cam motion at one particular rotational speed. In some cases, cam speed may be changed in the neighborhood of the desired speed so it is necessary to demonstrate the validity of the cam profile to speed alteration. Figure 4 describes the vibration amplitude produced by the simple harmonic motion curve, cycloidal motion curve and proposed cam curve over a range of operating speed. Residual vibration amplitude is normalized by taking the peak of the overshoot vibration amplitude, after the rise portion of the profile, and dividing by the cam displacement amplitude. A decreasing value of n (time ratio) indicates an increasing operating speed. It should be mentioned here, when *n* time ratio is less than 1, the proposed method can introduce an increase in slope of the cam profiles at rise-return start and stops. This can cause higher contact forces at the cam surface. However, the results indicate that the proposed method is less sensitive to speed variation than the other standard cam curves.

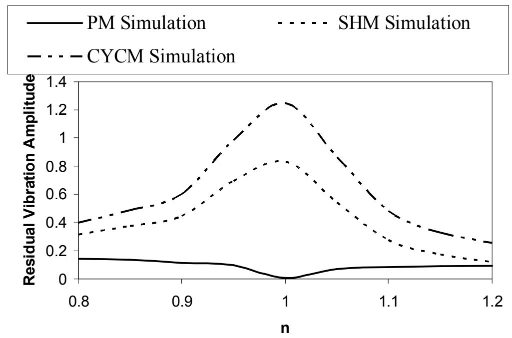


Figure. 4. Residual vibration amplitude produced by the simple harmonic motion (SHM) curve, cycloidal motion (CYCM) curve and proposed motion (PM) cam curve over a range of operating speed.

5. EXPERIMENTAL VERIFICATION

The experimental system used in the present study is depicted in Figure 5. The object of the experimental work has been to illustrate the applicability of the proposed method in preventing residual vibration at cam motion dwell time. The set up consist of a modified cam experiment set up of Techquipment cam analysis machine, a conductive potentiometers to read the displacements of the cam and follower, and hardware to control, command and read the data of the system. The spring stiffness is 3100 N/m and damping ratio is 0.045. The resulting natural frequency of the follower is 39.37 rad/sec. Cams were designed and manufactured by a CNC milling machine. Figure 6 shows the manufactured cams using simple harmonic motion on the top, cycloidal motion on the middle and proposed motion on the bottom. The experimental results of the Figure 3 and Figure 4 are shown in Figure 7 and Figure 8, respectively. The system constants for experiments are the same as those used in the simulation results revealed in Figure 3.

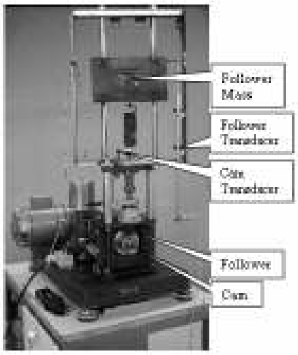


Figure. 5. Experimental set up.

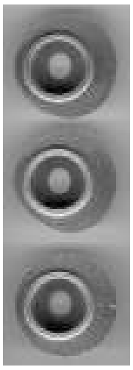
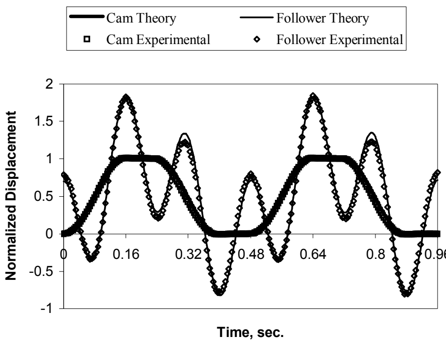
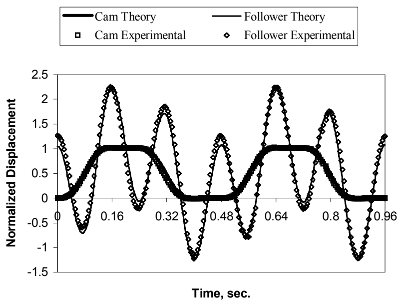


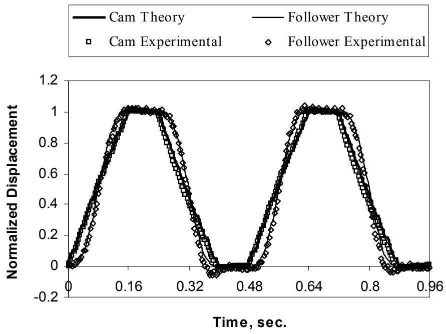
Figure. 6. The manufactured cams using simple harmonic motion on the top, cycloidal motion on the middle and proposed cam profile on the bottom.



(a)



(b)



(c)

Figure. 7. Experimental results of a cam-linkage shedding system model for time ratio of $n = 1.00$, damping ratio $\xi = 0.045$, natural period of vibration of the follower $t_n = 0.16$ sec and cam rotating at particular speed of 125 revolutions per minute. a) Simple harmonic cam profile. b) Cycloidal cam profile. c) Proposed cam profile by modifying excursion distance values of $\frac{L_1}{L} = 0.9052$, $\frac{L_2}{L} = 0.0051$, and $\frac{L_3}{L} = 0.0898$.

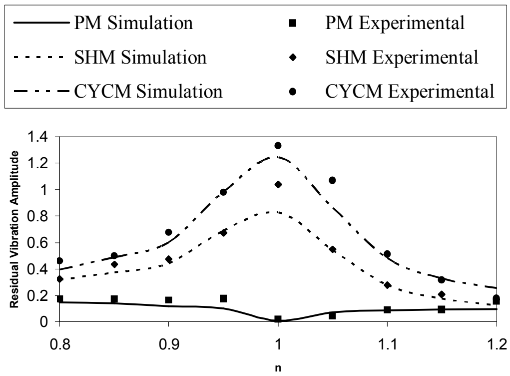


Figure. 8. Experimental residual vibration amplitude produced by the SHM cam, CYCM cam and PM cam over a range of operating speed.

6. CONCLUSIONS

In this study, a method to reduce vibration problems in cam design is discussed. In this method, simplified dynamic model of the shedding cam-linkage mechanism is used to create the equation of motion for the system. This equation relates the cam displacement to the heald frame displacement by using equivalent model. The equation of motion is then rewritten to solve for the cam profile. The desired heald frame motion is defined and the cam displacement needed to obtain that desired heald frame motion is computed. The excursion displacement of mathematical curves originally used to define the cam motion is substituted in as the output motion values. The new cam functions are then being solved for, creating an entirely new cam profile. The dynamic effects systems are being used to solve a new cam profile. It should be noted that the proposed method can cause an increase in slope of the cam profiles at rise-return start and stops when n time ratio is less than 1. This will result of high contact forces at the cam surface. However, the new cam profile compensates for the residual vibrations by removing them for the designed operating speed. The applicability of the method is shown by experimental results.

Acknowledgements

We are grateful to Mr. S. Mehmet Dabanıyastı from Örnek Textile Machinery Ltd. Sti (Gaziantep, Turkey) for their assistance in manufacturing the cams. This study was supported by The Scientific Research Projects of Gaziantep University (GÜBAP).

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