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Modeling Narrow Loom Beat-Up System with Cam Transmission Influence

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Abstract: Modeling and dynamic analysis of the narrow loom beat-up mechanism is presented. The equations of motion of the system are formulated considering the influence of the driver cam-follower mechanism as an improved analysis to an earlier study. Numerical results show the influence of the inertia forces due to the cam-follower mechanism. The model tends to account for the torsional and translational transmission of the cam-follower mechanism, the contact force of the roller/cam interface, the return spring force and the load acting on the cam-follower mechanism due to the action of the beater and all the damping, stiffness, mass and inertia of the mechanism components. The results illustrate that the system response decreases in magnitude as damping ratio increases in value. High peak values of the magnitudes are obtained at damping ratio <0.700 which is about the critical damping ratio for this type of system. The damping coefficient requirement for a reasonable control of the system could be obtained as illustrated by the steady state portion of the system response.

Key words: Mechanism, model, system, disc-cam, stiffness, damping

INTRODUCTION

The beat-up system of a narrow loom is a substructure of the narrow loom mechanism which generates a DRRD harmonic motion for beating the weft into the yarn during weaving process as reported by Raji (2000). A cam mechanism with swinging roller follower has been design to drive the beater mechanism as detailed by Raji and Adegbuyi (2003). One of the major factors in the design and operation of a dynamic system is the system stability and this can be influenced by several factors including the effect of the prime mover on the response of the driver mechanism for the system, stability analysis of this kind of dynamic system are discussed by Turhan and Koser (2004), Liu and Sun (2008), Verichev *et al*. (2010), Nieto *et al*. (2010) and Chen and Zou (2010). Analysis of the transient response of the beater system as an isolated system had earlier been discussed by Raji *et al*. (2010). Their study attempt to control the instability of the system by introducing a damping element connected to the beater for purpose of dampening the vibration of the system which result from the impact loading of the beater and the input torque from the prime mover.

The response was observed to attain a steady state within determined range of values for the damping element. The input torque was considered as a continuous turning effect which is constantly delivered on the beater shaft via the swinging roller follower. The beater system was modeled as an isolated system from the whole mechanism neglecting the effect of the possible disturbances that could be induced by the driver cam mechanism of the swinging roller follower and the reducing resultant torque on the input shaft due to the frictional forces, the follower spring forces and the inertias of the follower and cam.

In this communication, we investigate the effect of the cam mechanism as a separate system on the beater system for the same narrow loom set-up. The study of this influence will enable optimal design for the cam mechanism parameters such as optimal selection of design parameters, optimal design of the cam profile, optimal control of the mechanism input speed and optimum selection of damping element for the beater system.

The motion of the system could be affected by the stiffness and damping properties of the composition elements of the cam-follower mechanism. In high-speed cam mechanisms, vibration of the linkage is harmful in that the end motion deviates from the required schedule but more importantly, the vibration may result in the loss of contact between the cam and the follower. This causes an increase in noise levels and excessive wear. Thus, for the narrow loom beat-up system, the rotating features of the cam mechanism could create disturbance forces and moments that can contribute to the vibration of the beater system. The effects of driver mechanism such as the cam follower mechanism on the stability of systems have been studied for some time (Fig. 1).

Nagata *et al*. (2001a, b) developed a recursive formulation of rigid/flexible systems. The formulation based on d'Alembert principle in conjunction with

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recursive kinematical and kinetic expressions, deals with both space and ground-based mechanical systems. The formulation generously accounted for arbitrary level of topological branching, character of the structural members, slewing, deployment, orbital perturbations, shift in the center of mass and joint constraints. A dynamic simulation program was developed to demonstrate the effectiveness of the model.

Turhan and Koser (2004) studied the stability of the parametrically excited torsional vibrations of shafts connected to mechanisms with position-dependent inertia. The shafts were considered to be torsional elastic, distributed parameter systems and discretized through a finite element scheme. The mechanisms are modeled by a linearized equation of motion. A simplified mathematical model was developed to predict the dynamic behavior of a mechanism. Fed by the cam profile and the parameters of the mechanism, a computer program yields the output motion of the mechanism as well as the contact force between the cam and its follower.

Kayumov (2006) applied the concept of parametric controllability to systems of rigid bodies for purpose of refinement of models by accounting for small variability of the links assumed by rigid bodies in any first approximation model. It was shown that taking account of small change in the parameters can ensure the controllability of a mechanism which was not controllable assuming absolute rigidity of the links. Lin *et al.* (2007) developed model for lumped parameter simulation of dynamic characteristic peristaltic micropumps. The model was used to perform a systematic analysis of the impact of geometry, materials and pump loading on device performance.

Wang *et al.* (2008) noted the uncertainties in dynamic model of multibody systems caused by factors such as the joint clearance, friction, lubrication, material non-uniformities and assembly error. A model was developed to determine the stochastic behavior of model parameters on the dynamic response of a mechanical system. Gatti and Mundo (2010) investigated the feasibility of controlling follower motion by applying a secondary force directly onto the follower. Simple active and passive control strategies were investigated and compared to determine the effectiveness and practical feasibility of the technique.

In this present study, an attempt is made to develop a model for the narrow loom beat-up mechanism considering the influence of the inertia involved in the operation of the driver cam-follower mechanism for the beater system and the cam-follower transmission effect across the whole system.

MATERIALS AND METHODS

The improved system model is as shown in Fig. 2. The cam-follower mechanism is connected by a shaft to the beater system. The scheme for the analysis is the use of lumped-parameters assumption, avoiding the possibility of spatial dependence of the system's body masses with the assumption that the camshaft, cam plate, follower and the beater slay-bar are rigid. It is also noted that linear displacement of the cam due to motor torque excitation is possible. The damper b_g represent the damping between the disc-cam and the ground while the stiffness k_g maintain the contact between the disc-cam and the ground. M_c is the mass of the disc-cam, K_c and B_c represents the stiffness constant of the disc-cam and the damping coefficient respectively. k_{rf} is the equivalent stiffness between the roller/follower and the disc-cam, br_f is the equivalent damping coefficient between the roller/follower and the disc cam. k_f maintain the contact

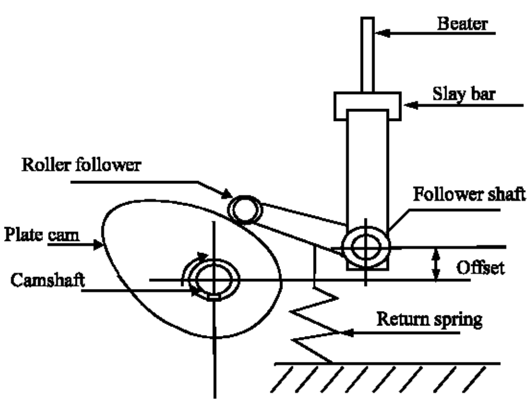

Fig .1: Schematic diagram of the model mechanism (Raji, 2000)

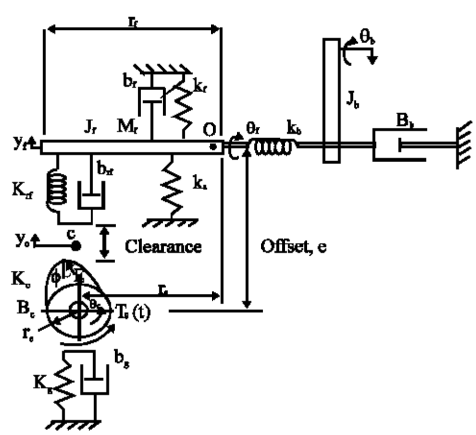

Fig. 2: Model for beat-up system with cam transmission influence

between the follower and the ground, b_f represent the damping between the follower and the ground. F_c is the contact force between the roller and the cam ϕ is the swinging roller follower pressure angle and k_s is the return spring constant. The spring of stiffness k keeps the follower always in contact with the cam plate to avoid bouncing of the follower during operation.

The beat-up system substructure is modeled as described by Raji *et al*. (2010). The system modeled as a torsional system comprises a shaft of torsional stiffness K_b, a beam of mass-moment of inertia J_b which represents inertia body for the beater and slay-bar rotating about the axis of rotation of the shaft, a beam of mass moment of inertia J_f representing the combined follower-roller inertia. Torque T_i is delivered on the beater. The system is damped with a viscous coupling of coefficient, B_b. The corresponding differential equations describing the model are obtained as expressed in Eq. 1-5:

$$M_c\ddot{y}_c + b_g\dot{y}_c + k_g y_c = 0 \quad (1)$$

$$M_f\ddot{y}_f - b_f\dot{y}_f + (k_s - k_f)y_f = 0 \quad (2)$$

$$I_c\ddot{\theta}_c - b_{\Upsilon f}\dot{\theta}_f + (B_c + b_{\Upsilon f})\dot{\theta}_c + (K_c + k_{\Upsilon f})\theta_c - k_{\Upsilon f}\theta_f + F_c\mathrm{Cos}\phi\frac{(\dot{y}_f - \dot{y}_c)}{\dot{\theta}_c} = T_\theta \quad (3)$$

$$I_f\ddot{\theta}_f + b_{\Upsilon f}(\dot{\theta}_f - \dot{\theta}_c) + (k_{\Upsilon f} + K_b)\theta_f + k_b\theta_b - k_{\Upsilon f}\theta_c = 0 \quad (4)$$

$$I_b\ddot{\theta}_b + B_b\dot{\theta}_b + k_b(\theta_b - \theta_f) = 0 \quad (5)$$

Where F_c is the contact force expressed as:

$$F_c = \frac{T_f}{\Upsilon_f \mathrm{Cos}(\phi)} \quad (6)$$

T_f is the total torque of the follower with respect to point O. This include the torque due to the effect of springs and inertias and could be expressed as; $T_f = k_{eqv}.\theta_s$ for an initial displacement of the return spring θ_s and equivalent spring stiffness, k_{eqv}. The pressure angle of the cam is expressed as;

$$\phi = \tan^{-1}\left[\left(\frac{\dot{y}_f - \dot{y}_c}{\dot{\theta}_c} - e\right) \times \frac{1}{(y_f - y_c) + \sqrt{\Upsilon_c^2 - e^2}}\right] \quad (7)$$

The influence of the cam mechanism on the beater system is explicitly exposed by Eq. 5 which may be re-expressed as;

$$\ddot{\theta}_b + 2\zeta\omega_n\dot{\theta}_b + \omega_n^2\theta_b = \omega_n^2\theta_f \quad (8)$$

Where:

$$\zeta = \frac{B}{2\sqrt{JK}}$$

and:

$$\omega_n = \sqrt{\frac{K}{J}}$$

are the damping ratio and natural frequency, respectively for the beater system. The resulting dynamic model is a set non-linear system. Numerical simulation of the dynamic behavior of the system can be obtained by solving the system Eq. 1-5.

RESULTS AND DISCUSSION

A numerical example is presented to describe the model performance. It is desire that the follower exhibits the DRRD motion to obey the following equations as is obtained by Raji (2000):

$$\theta_{\downarrow}f = 1/3[20(_{\downarrow}c^{\uparrow}3 - 25(_{\downarrow}c^{\uparrow}4 + 8(_{\downarrow}c^{\uparrow}5)] \quad (9)$$

Where: $_{\downarrow}c = \theta_{\downarrow}c/\beta$

is a dimensionless ratio for a specified follower maximum rise β. The response was a sinusoidal function which should serve as input to the beater dynamic modeled in Eq. 8. This is a second-order system subjected to sinusoidal input and the solution may be best obtained by Frequency Response Function (FRF) method as follows;

$$G(s) = (\downarrow b/(\downarrow f = (\omega_{\downarrow}n^{\uparrow}2(s^{\uparrow}2 + 2^{\cdot}(\omega\downarrow ns + \omega\downarrow n^{\uparrow}2) \quad (10)$$

The magnitude and phase angle of the FRF may be expressed as:

$$\phi = -\tan^{-1}\frac{2((\omega/\omega n)}{1 - (\omega/\omega_n)^2} \quad (11)$$

A program is prepared in the Microsoft excel environment to generate the Table 1 and 2 for the magnitude and phase plot of the system. The plots for the magnitude and phase response of the system are as shown in Fig. 3 and 4. It is observed from Fig. 3 that the system response decreases in magnitude as damping ratio

Table 1: Magnitude response

Normalized frequency (ω/ω_n)	System magnitude response to sinusoidal input $\zeta = 0.1$	$\zeta = 0.3$	$\zeta = 0.5$	$\zeta = 0.7$	$\zeta = 1.0$
0.1	0.086	0.071	0.043	0.001	-0.086
0.2	0.347	0.287	0.170	0.000	-0.341
0.4	1.475	1.174	0.627	-0.083	-1.289
0.8	8.091	4.437	1.137	-1.411	-4.297
1.0	13.979	4.437	0.000	-2.923	-6.021
1.2	6.000	1.475	-2.131	-4.794	-7.748
1.6	-4.041	-5.257	-6.984	-8.722	-11.029
2.0	-9.619	-10.187	-11.139	-12.263	-13.979
2.4	-13.596	-13.932	-14.536	-15.308	-16.599
2.8	-16.730	-16.956	-17.374	-17.935	-18.929
3.2	-19.334	-19.497	-19.805	-20.230	-21.015
3.6	-21.570	-21.694	-21.931	-22.265	-22.898
4.0	-23.534	-23.632	-23.820	-24.089	-24.609
4.4	-25.287	-25.366	-25.520	-25.741	-26.176
4.8	-26.872	-26.938	-27.065	-27.250	-27.619
5.2	-28.320	-28.375	-28.483	-28.640	-28.956
5.6	-29.652	-29.699	-29.791	-29.926	-30.200
6.0	-30.886	-30.927	-31.007	-31.125	-31.364
6.4	-32.037	-32.072	-32.143	-32.246	-32.457
6.8	-33.114	-33.146	-33.207	-33.299	-33.486
7.2	-34.128	-34.155	-34.210	-34.292	-34.459
7.6	-35.084	-35.109	-35.158	-35.231	-35.382
8.0	-35.990	-36.012	-36.056	-36.122	-36.258
8.4	-36.850	-36.870	-36.910	-36.970	-37.093
8.8	-37.669	-37.687	-37.724	-37.778	-37.891
9.2	-38.450	-38.467	-38.501	-38.550	-38.654
9.6	-39.198	-39.213	-39.244	-39.289	-39.385
10.0	-39.914	-39.929	-39.957	-39.999	-40.086

Table 2: Phase response

Normalized frequency (ω/ω_n)	System phase response to sinusoidal input $\zeta = 0.1$	$\zeta = 0.3$	$\zeta = 0.5$	$\zeta = 0.7$	$\zeta = 1.0$
0.1	91.157	92.959	94.103	93.468	90.149
0.2	92.386	103.317	107.382	103.774	90.000
0.4	95.440	53.043	17.835	145.318	80.558
0.8	113.959	85.694	76.645	10.456	160.629
1	0.000	87.530	76.645	90.000	127.773
1.2	61.393	84.128	38.573	130.099	113.561
1.6	78.409	98.985	101.163	101.564	103.079
2	82.406	93.608	95.660	97.220	99.322
2.4	84.242	92.540	94.126	95.527	97.474
2.8	85.320	92.061	93.387	94.622	96.382
3.2	86.038	91.782	92.943	94.053	95.659
3.6	86.555	91.597	92.644	93.660	95.143
4	86.948	91.463	92.427	93.369	94.754
4.4	87.256	91.361	92.261	93.144	94.449
4.8	87.506	91.281	92.129	92.965	94.203
5.2	87.713	91.215	92.021	92.817	93.999
5.6	87.888	91.160	91.930	92.693	93.827
6	88.037	91.114	91.854	92.588	93.680
6.4	88.166	91.074	91.787	92.496	93.552
6.8	88.278	91.039	91.729	92.416	93.440
7.2	88.378	91.008	91.678	92.345	93.340
7.6	88.466	90.980	91.633	92.282	93.251
8	88.545	90.956	91.592	92.225	93.171
8.4	88.617	90.933	91.555	92.174	93.098
8.8	88.681	90.913	91.521	92.127	93.032
9.2	88.740	90.894	91.490	92.084	92.971
9.6	88.794	90.877	91.462	92.044	92.916
10	88.843	90.862	91.435	92.008	92.864

increases in value. High peak values of the magnitudes are obtained at damping ratio <0.700 which is about the critical damping ratio for this type of system. This is an indication that the system will require an overdamped

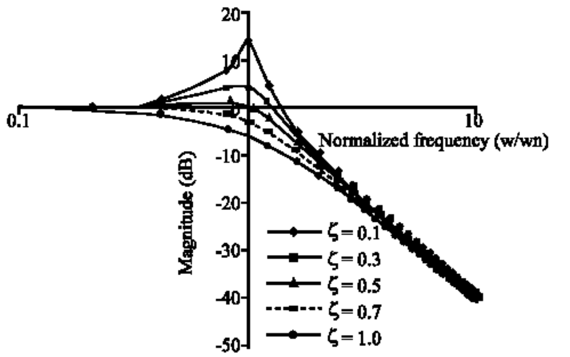

Fig. 3: Bode log magitude plot for the system response

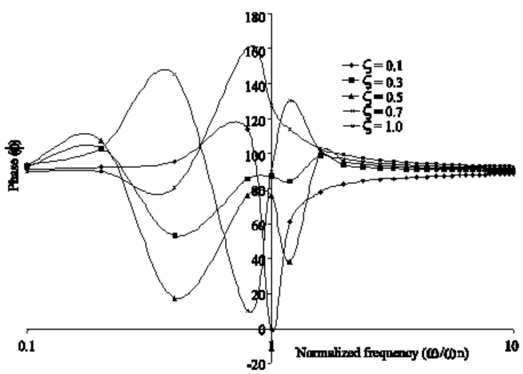

Fig. 4: Bode phase plot for the system response

control method for stability to be obtained. The peak magnitude is reached at the peak frequency which could be estimated from the differentiation of the magnitude expression. The damping coefficient requirement for a reasonable control of the system could then be estimated. It also could be observed in Fig. 4 that the system attained steady state at normalized frequencies of $\frac{\omega}{\omega n} > 4.4$.

CONCLUSION

The present study addresses the need to develop an accurate model for the response of narrow loom beat-up mechanism. The mathematical model of the system structure is realized as a set of non-linear system. The model tends to account for the torsional and translational transmission of the cam-follower mechanism used to drive the narrow loom beater system, the contact force of the roller/cam interface, the return spring force and the load acting on the cam-follower mechanism due to the action of the beater and all the damping, stiffness, mass and inertia of the mechanism components. The model could be used to predict the dynamic response of such system in order to be able to evaluate its stability.

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