DYNAMIC STIFFNESS OF SHAFT FOR THRESHER IN COMBINE HARVESTER

[1]THAN THAN HTAY, [2]HTAY HTAY WIN, [3]ZIN EI EI WIN

Department of Mechanical Engineering, Mandalay Technological University, Mandalay, Myanmar
E-mail: [1]phil.tth@gmail.com

Abstract— This paper deals dynamic stiffness of shaft for thresher in Combine Harvester. In this research, the amplitude, phase angle and stiffness can then be determined in a dynamic sense. Especially, the most synchronous excitation force in rotating machinery is the residual unbalance in the shaft. Therefore, the value of original response vector, new response vector, response due to calibration weight alone, applied force, synchronous dynamic stiffness, modal stiffness, modal mass of unbalance weight, modal damping, influence vector and response vector can be solved based on Hooke's Law equation. Result of applied force is 1.6636 N for calibration weight alone. Modal mass (m) of unbalance weight is 1.2575 kg. Mass of shaft is 34.68 kg with 75 mm diameter and 7850 kg/m^3 density of carbon steel. Finally, outcome of modal stiffness for unbalance weight (K) is 209.03 N/m, and stiffness of shaft (K_t) is 3.9716×10^9 based on rotating speed (1351 rpm) and engine speed (2800 rpm).

Index terms— Amplitude (Response), Combine Harvester, Synchronous Dynamic Stiffness, Shaft, Speeds

I. INTRODUCTION

This combine harvester was produced from KUKJE Machinery Co.,ltd. (Korea). All real physical structures, when subjected to loads, behave dynamically. If the loads are applied very slowly, a static load analysis can be justified. This research has been developed to understand what dynamic stiffness is, why it is important and how to use it. So, dynamic stiffness is the static shaft stiffness of the mechanical system complemented by the dynamic effects of mass and damping.

It is useful to see how Dynamic Stiffness changes as a result of perturbation frequency. Hence, dynamic analysis is a simple extension of static analysis. All rotating shafts deflect during rotation. The magnitude of this deflection is dependent on the shaft speed and the stiffness of the shaft.

The purpose of dynamic stiffness a shaft is to help ensure that the machinery is safe and reliable. This is achieved when the shaft mass and length are as close to equal as possible. Excessive unbalance can cause vibration and stress in the shaft or attached pieces. Excessive vibration can also cause wear in machinery components and reduce the life expectancy of these parts.

In addition, there are many dynamic analysis involving modal analysis, harmonic, transient, dynamic stiffness etc. Among them, dynamic stiffness was analyzed for a shaft of thresher in combine harvester. The mass of the shaft system can be accurately estimated to resist excitation force as unbalance weight (mass).

During site balancing, it is easy to make a weight change. It is inexpensive to make a run, and there is less physical risk to the machinery or the operator. In this case of field balancing, it is often difficult to change weights, and it is generally expensive to make a full speed run. Balancing behavior is based on the fundamental relationship between the shaft response and applied force [1].

$$\text{Response} = \frac{\text{Force}}{\text{Restraint}} = \frac{\text{Force}}{\text{DynamicStiffness}} \qquad (1)$$

This general expression has many specific applications in rotor dynamic. Within the balancing rule, the response is the measured shaft and the force is the magnitude of the unbalance force. The restraint is the dynamic stiffness. The variables in Equation (1) are vector quantities.

II. PROCEDURE FOR DYNAMIC STIFFNESS

A. Dynamic Stiffness

Dynamic Stiffness is a function of the excitation frequency. It is relationship between machine parameters and measured vibration response. It can be used to monitor the dynamic forces acting in a machine. A dynamic force is a force is a force that changes in magnitude or direction with time. The dynamic input force will cause dynamic output motion.

Because of this dynamic motion, both the Quadrature Stiffness due to damping and the mass stiffness effects come into play. The force, F, and the response, R, are vectors, and they have both magnitude and direction. This equation (2) based on Hooke's Law for spring [2].

$$\vec{K}_{DS} = \frac{\vec{F}}{\vec{R}} \qquad (2)$$

By Hooke's Law, equation (3) can be expressed.

$$F = K x \qquad (3)$$

Fig. 1 shows the static spring stiffness of the mechanical system complemented by the dynamic effects of mass and damping.

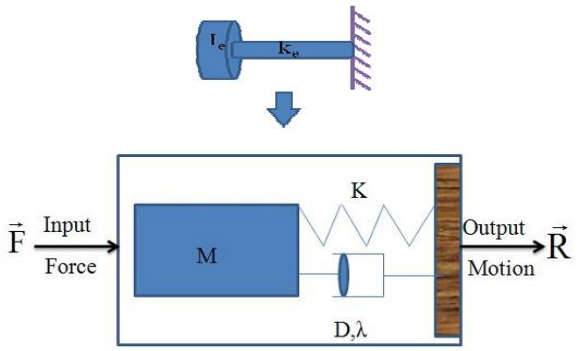

Fig. 1 Mathematical Modeling of Dynamic Stiffness

There are two types of dynamic stiffness. These are synchronous and nonsynchronous dynamic stiffness. Fig. 2 presents procedure for developed Dynamic Stiffness.

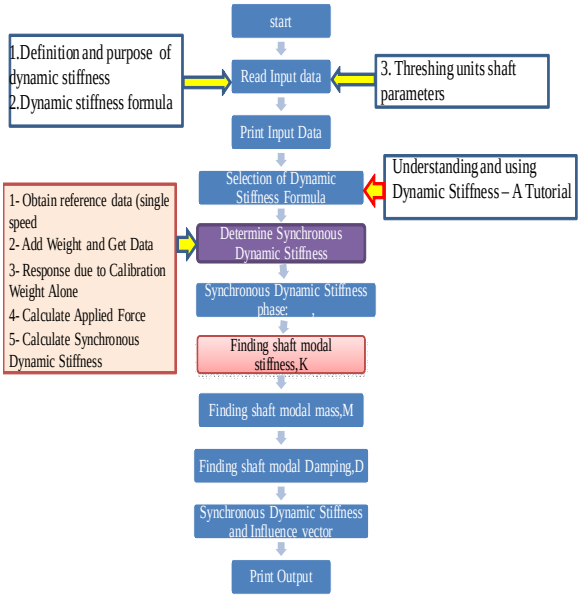

Fig. 2 Flow chart of the program for the developed Dynamic Stiffness

B. Synchronous Dynamic Stiffness

Synchronous means that the input force is at the same frequency as the machine's rotational speed. Synchronous Dynamic Stiffness is a special case of Nonsynchronous Dynamic when the excitation frequency equals the rotating speed of the machine.

When Synchronous Dynamic Stiffness can be calculated for a single speed, both the direct and quadrature parts are functions of the rotational frequency, ω. The direct dynamic stiffness is a parabola with y-intercept equal to the radial spring stiffness, K. The zero crossing is important and occurs in equation (4).

$$\omega = \sqrt{\frac{K}{M}} \qquad (4)$$

C. Nonsynchronous Dynamic Stiffness

It refers to any excitation force that is independent of the rotational speed of the machine.

III. MACHINE CONFIGURATION

This machine (Combine Harvester) consists of harvester, conveyor, thresher, blower, sieve and concave, storage tank, auger, shaft as main components. The vital component one is thresher. It threshes the grain from the straw. So, to be less in threshing losses needs. Therefore, designer who creates thresher must consider carefully before inventing. Fig. 3(a) shows thresher of combine harvester.

To be a good quality, thresher has composed of cylinder, drums, tooth, comb tooth that fitted by bolts and nuts. Shaft is fitted middle of thresher. This type of shaft is similar to many threshers. Fig. 3(b) and (c) are views of shaft inside thresher.

Although each of the thresher segments is component balanced, any minor shift between elements will produce a synchronous unbalance force. Since, this unit operates at 1351 rpm, a few grams of unbalance or mil or two of eccentricity will result in excessive shaft vibration and a strong potential for machine damage.

On the positive view, this machine is a compact design that yields a high thermal efficiency. Therefore, when the unit is properly assembly and balanced, it is very cost-effective to operate [3].

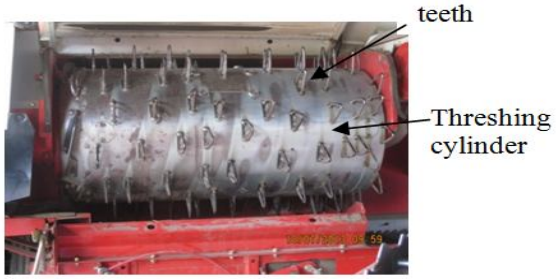

Fig. 3(a) Thresher composed of tooth

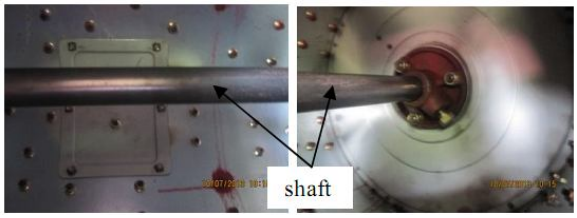

Fig. 3 (b) and (c) View of Shaft inside thresher

IV. SOME REALITIES OF FIELD BALANCING AND METHOD

D. Balancing of Dynamic Stiffness

During site balancing, it is easy to make a weight change. It is inexpensive to make a run, and there is less physical risk to the machinery or the operator. In this case of field balancing, it is often difficult to change weights, and it is generally expensive to make a full speed run. Balancing behavior is based on the fundamental relationship between the shaft response and applied force expressed by [4].

The force is proportional to the static spring deflection (or response), x, where K is the spring

stiffness. The Dynamic Stiffness, $\vec{K}_{DS}$, is the shaft dynamic equivalent of the simple, static, linear spring in the Hooke's Law equation.

Equation (5) is for finding Synchronous Response Motion ($\vec{R}$). Values of both sides are vector quantity. The synchronous Dynamic Stiffness can control the shaft response to the unbalance force. In equation (6), K-MΩ^2 refers to Direct term and jD(1-λ) Ω refers to Quadrature term.

$$\text{Synchronous Response Motion}(\vec{R}) = \frac{\text{UnbalanceForce}}{\text{SynchronousDynamicStiffness}} \quad (5)$$

$$= \frac{(mr_u\Omega^2)\angle\delta}{(K - M\Omega^2) + jD(1-\lambda)\Omega} \quad (6)$$

The purpose of balancing shaft is to help ensure that the machinery is safe and reliable. Excessive unbalance can cause vibration and stress in the shaft. Dynamic forces can also be transmitted to supporting structures. Then, excessive vibration can cause wear in bearing and reduce the life expectancy of these parts. For this machine, λ is zero for bronze bearings.

It is occasion where the measurement systems could be mixed such as grain-inches depending on the type of scale or weight that are available. Fig. 4 represents the centrifugal force produced by unbalance depends on the mass and radius as well as rotational speed of the shaft [4].

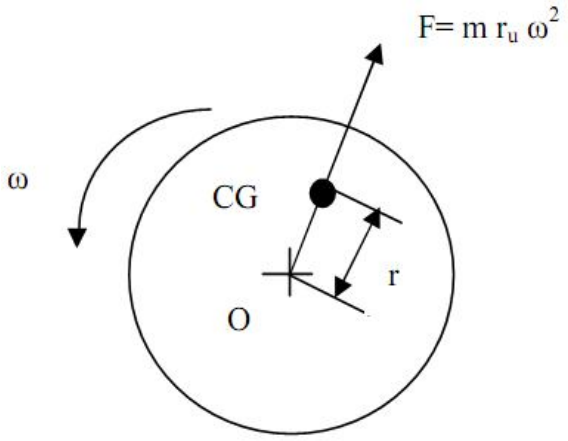

Fig. 4 Rotor section with CG displaced due to unbalance

E. Modal Balancing

At resonance, the shaft deflection will be similar to the mode shape of the natural frequency. It depends on the physical characteristics of the shaft and the bearing properties. However, unbalance near the supports would create very little vibration. The actual vibration amplitudes at resonance will depend on the unbalance and damping in the system [5].

Shaft vibration readings consist of amplitude and phase data. The phase angles are determined from a once-per-revolution. The angular location is measured opposite shaft rotation. And, Fig. 5 is the unbalance location on the shaft [3].

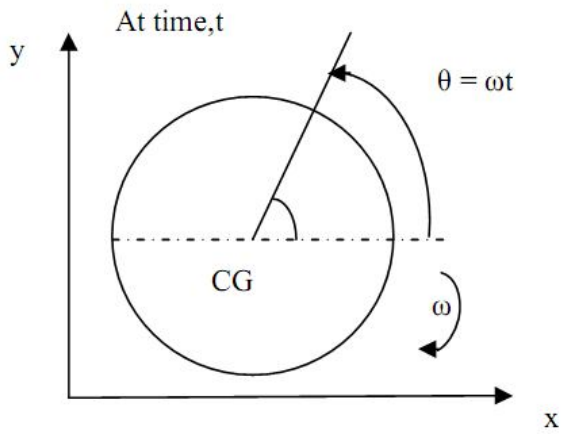

Fig. 5 The unbalance location on a shaft

F. Balancing Specification

Response of vibration is measured by MIL-STD-167 (military specification). It deals with the mechanical vibration of rotating machinery. Equation (7) can be applied to operate at its maximum continuous speed within the specified operating speed range, the peak-to-peak amplitude of vibration [6].

$$A = \sqrt{\frac{12000}{N}} \quad (7)$$

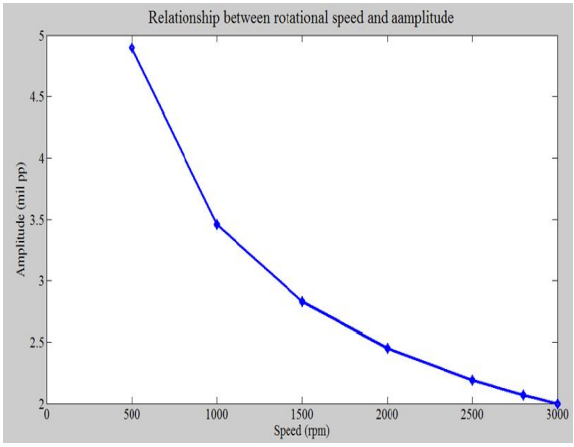

Fig. 6 Relationship between rotational speed and amplitude

Operating at maximum continuous speed within the specified operating speed, the peak-to-peak amplitude of vibration in any plane was found in Fig. 6. The procedure involves the addition of a known calibration weight to the shaft and determining the response vector C due to that calibration weight, so, new vibration response vector of the shaft is defined as the $\vec{O} + \vec{C}$ [7].

The balancing technique of add a weight take data, and do vector calculations generally produces acceptable results on many machines. However, this technique may fail on a complex rotor [1].

Balance of rotating shaft can be found at low speed when vibration occurs at high speeds (mode shape). However, for a constant angular velocity of the shaft, ω, the location of the unbalance is searched by

equation (8).

$$\theta = \omega t \qquad (8)$$

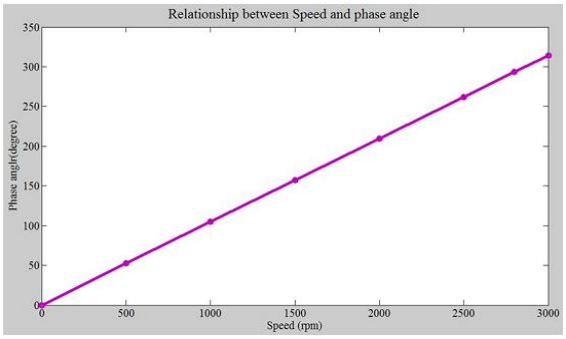

Fig. 7 Relationship between rotational speed and phase angle

Then, Fig. 7 shows phase angle is changing based on various operating speeds at time one second. The values of original and new response vector are taken from Fig. (7). Fig. 6 and Fig.7 are created by MATLAB according to variable speeds.

V. RESULTS AND DISCUSSION

The value of original response and new response vectors required for considering dynamic stiffness basically. So, Fig (6) and (7) are used to obtain that reference data. After reading and applying, the value of original and new response vector on a polar plot can be determined as shown in Fig. 8. (a) and 8 (b).

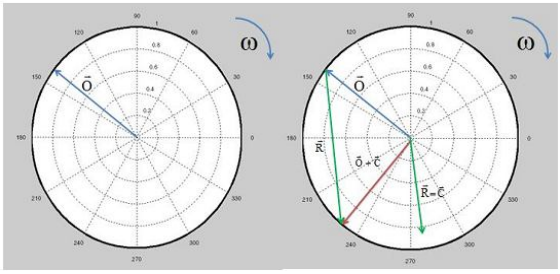

Fig. 8 (a) Original Response vector ($\vec{O}$) on a polar plot

(b)New Response vector ($\vec{O}+\vec{C}$) and response vector ($\vec{R}=\vec{C}$) on a polar plot

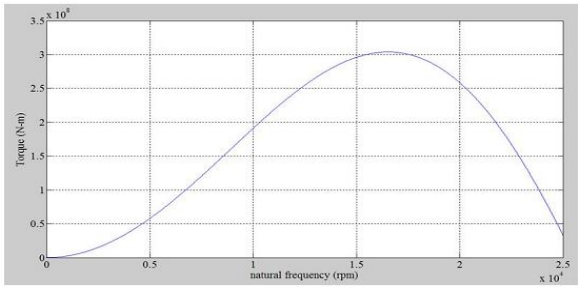

Fig. 9 Modes shapes by Holzer's Method
(Under 25,000 rpm)

The value of mode shapes and natural frequencies is calculated by using Holzer's Method. By using MATLAB, the values of mode shapes are found where they happen due to relation between natural frequencies and torques.

Fig. 9 represents mode shapes do not happen fewer than 25,000rpm operating speed. Unless the shaft weight and its speed are corresponding, maximum amplitude will be found in everywhere. i.e; every rotating speed. Mode shapes become the ratio of amplitudes. Although limiting of speed for this machine is 2800 rpm, operators can operate safely this machine till 25,000rpm because the shaft is considered to resist external force (unbalance weight) before doing design.

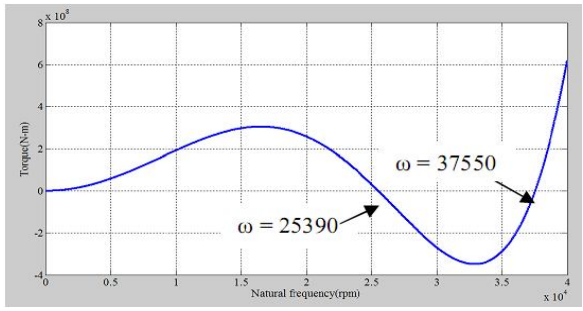

Fig. 10 Mode shapes by Holzer's Method
(under 40,000 rpm)

In Fig. 10 represents mode shapes happening during 40000rpm. Mode shapes can be seen at zero level where intersection of natural frequency and torque. Maximum vibration happens 25390 rpm and 37550 rpm in Fig. (10). Therefore, less than 25000rpm natural frequency do not create maximum vibration and mode shapes cannot create. Response becomes during acceptable limit. Also, in Fig. (11), mode shapes because of responses are at over 25000 rpm. So, the shaft satisfies to use for thresher over maximum engine speed (2800 rpm).

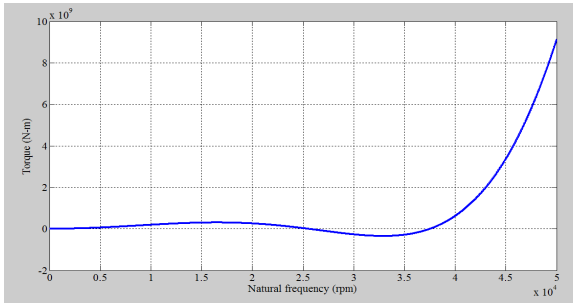

Fig. 11 Modes shapes by Holzer's Method
(less than 50,000 rpm)

In this paper, dynamic stiffness is analyzed. Also, it evaluated unbalance force, modal mass, and influence vector including the dynamic stiffness. Based on various speeds, relationship between speed and amplitude, and then phase angle can be calculated.

Also, the result values in table I are monitored. Comparing of modal stiffness and mass with equivalent stiffness of shaft, using shaft is more strength than for unbalance weight. So, the following table II shows comparison of modal and actual mass stiffness values.

Table I
Result of Unbalance Weight Alone

Items	Parameters	Symbols	Values
1	Original response(mil p-p)	$\vec{O}$	$2.98 \angle 141.37°$
2	New response (mil p-p)	$\vec{O}+\vec{C}$	$3.8 \angle 231.37°$
3	Response due to calibration weight	$\vec{R}=\vec{C}$	$4.83 \angle 269.48°$
4	Applied force of calibration weight	$\vec{F}$	$1.6636\ N \angle 90°$
5	Synchronous dynamic stiffness	$\vec{K}_{DS}$	$688.89 \angle 179°$
6	Modal mass (kg)	M	0.0104
7	Modal stiffness (N/m)	K	209.03
8	Modal damping (N.s/m)	D	0.041
9	Influence vector (mil p-p)	$\vec{H}$	$528.22 \angle 179°$
10	Response vector	H	$0.0268 \angle 179°$

Table II
Comparison of Modal Mass and Stiffness of Shaft and Unbalance

Items	Modal mass	Stiffness
Shaft	34.68 kg	3.9716×10^9 N/m
unbalance	1.2575kg	209.03 N/m

CONCLUSION

In this research, the required data is taken from design calculation. In design consideration, the value of operating speed is determined and compared with actual speed of thresher. Therefore, in this research, authors expressed only the main point for paper's title. The dynamic stiffness has been tested by using rotor dynamic theory. The actual dynamic stiffness of shaft for thresher is strength to apply. It can resist unbalance weight. And, mode shapes have been found at over 40000 rpm. During maximum engine speed, 2800 rpm, mode shapes cannot be create so that vibration (response) is less in operating rotational speed. Thus, the modes did not appear in the vibration data due to the high dynamic stiffness for each mode. Summing up, the shaft for this thresher is strength, resist to unbalance weight so that it is stiffness dynamically.

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Nomenclatures

A	vibration level (amplitude or response), mil p-p
CG	Center of Gravity
D	damping (N.S/m)
F	Centrifugal Force (N)
K	radial direct stiffness (N/m)
M	Mass of shaft (kg)
m	unbalance mass (kg)
mil	one thousandth of an inch (0.001")
N	maximum continuous speed, rpm
O	shaft axis
p-p	peak-to-peak
rpm	revolutions per minute
r	radius (m)

Greek Letters

λ	Fluid circumferential average velocity ratio
δ	The phase lag of unbalance mass (degree)
ω	Frequency (Hertz)
Ω	Rotative speed of unbalance weight (rad/s)

Subscripts

u	unbalance
n	natural

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