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By Than Than Htay, Htay Htay Win, Zin Ei Ei Win & Myint Thein
*Mandalay Technological University, Myanmar*

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*GJRE-A Classification :* *FOR Code: 240599p*

EFFECTOFDYNAMICSTIFFNESSONPERFORMANCEOFPADDYGRAINLOSSESINAXIAL-FLOWTHRESHER

*Strictly as per the compliance and regulations of:*

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Than Than Htay [α], Htay Htay Win [σ], Zin Ei Ei Win [ρ] & Myint Thein [ω]

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## I. Introduction

Performance test has based on dynamic analysis of the thresher. There are many sorts of the threshers to use for Combine Harvesters. This axial flow thresher is applied because it can give good performance for threshing and the least losses. Then, performance of that thresher is used by based on the shaft stiffness for this one.

The shaft is matched at the centre of it. While the shaft is operating with three forward engine speeds, the thresher will also do at the same condition. Based on the speeds, how the link of threshing losses and speeds at any positions that placed the unbalance weight on the shaft is considered.

Because of the high operating speeds and the performance from the shaft to the thresher, dynamic stiffness becomes a major design consideration. The need for dynamic analysis is especially important in the thresher of the shaft where an effective and efficient strength shaft is crucial in expending the shaft life. In the highest engine speed, the total threshing losses are very high.

It is therefore essential to be able to estimate the dynamic stiffness and ensure that the shaft can withstand such high model enables the thresher-shaft designer to modify the strength configuration for the optimum rate at high speeds level.

The result is impressive in that analysis but mathematical and dynamic model are complicated to consider. It is therefore proposed in this study a technique with consideration of any unbalance weight attachment in various speeds.

*Author: Department of Mechanical Engineering, Mandalay Technological University, Mandalay, Myanmar*
*e-mail: phil.tth@gmail.com*

## II. Methodology

*a) Machine Configuration*

There are many sorts of the threshers in threshing the grain. This combine harvester operated axial-flow thresher was produced from KUKJE Machinery Co., ltd. (Korea). This thresher performs based on the shaft stiffness in this study. The shaft is matched at the centre of this thresher. While the shaft is operating with three forward engine speeds, the thresher also operates at the same condition.

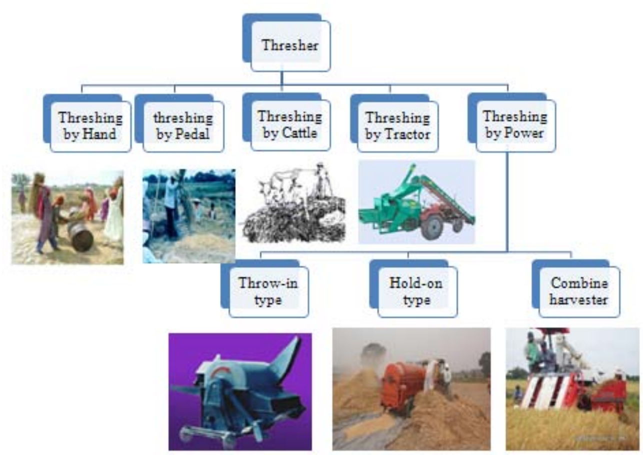


*Figure 1* : various kinds of threshers as period

Various kinds of threshers are shown in Fig. 1. As well as harvesting method, threshing is the important practice which can affect the quantitative and qualitative losses of rice. In Myanmar's rice fields, four main types of paddy thresher are used, i.e; manual, tractor operated cross-flow type, small thresher equipped with wire loop threshing drum and combine harvester operated axial-flow thresher.

Recently, DKC-685 combine harvester operated axial-flow thresher adopted in many rice fields because of its easy application and better output for paddy thresher. It has wire-loop type peg-tooth. And, four comb types is attached entry of the thresher. All peg-teeth and drum (cover) are bolted by nuts at the threshing cylinder. It threshes the grain by axially. These are used for threshing the paddy crop. And, many different kinds of peg-tooth design and the shaft for this are applied to get good performance.

Based on the speeds, the un-threshed and total grain losses also relate by any positions that placed the unbalance weight on the shaft. Dynamic Stiffness is a function of the excitation frequency. Hence, dynamic analysis is a simple extension of static analysis. All rotating shafts deflect during rotation.

Also, using thresher and its shaft in this machine are shown in Fig. 2 (a) and (b).


*Figure 2 :* (a) Thresher and (b) Shaft

b) *Experimental Field performance*

This study was carried out during 2013. And, Mechanization Training Centre (Meiktila) was chosen as a site to perform the paddy field. Two different performance systems were used to thresh paddy grain, namely, the DKC-685 Combine Harvester, with storage type (tank) , harvester to cut the crop and Thresher with axial wire loop (peg-tooth) type.

The specification of the used machine tabulates in Table (1). Results data of the thresher from the design consideration are expressed in Table (2) to apply for the next determination. The evaluation of threshing systems involves a number of experiment approaches and the dynamic stiffness into the following categories.

*Table 1 :* Specification of machine

| Parameter | Dimension | unit |
|---|---|---|
| L×W×H | 4430× 1860×2330 | mm |
| Total displacement | 2392 | cc |
| Power / Revolution | 52 (70)/2800 | kW(hp) /rpm |
| Number of reaping lines | 4 | Row |
| Reaping width | 1485±50 | mm |

*Table 2 :* Parameters of Thresher (DKC-685 Combine Harvester)

| Type | Values | unit |
|---|---|---|
| Outside diameter of Thresher | 0.43 | m |
| Length of thresher | 0.6576 | m |
| Diameter of shaft | 0.075 | m |
| Length of shaft | 0.2334 | m |
| Thresher Weight | 105.6931 | N |
| Threshing Speed | 5.31 | m/s |
| Threshing Power | 2.1141 | kW |
| Threshing Torque | 14.94 | N-m |
| Torsional Moment | 14.9433 | N-m |
| Total Weight (UD) | 69.5038 | N |
| Total Mass | 7.085 | kg |

i. *Thresher performance*

This thresher performance for all different types under study was evaluated measuring an un-threshed grain losses and total grain losses. A local long-grain paddy variety widely cultivated in Myittar Township was used for the performance. Physical characteristics of the variety are list in Table (3). The crop was cut 45-55 cm above the ground and collected for the experiments. The paddy moisture content at harvesting and threshing was measured using moisture meter. Four levels of paddy moisture contents of 25, 21.5, 20 and 17 % (w.b.) were considered for the tests. Determination of grain moisture content accurately is important before decision of harvesting, storage and milling as shown in Table (4) [10Ath].

*Table 3 :* Physical characteristics of paddy variety used in the experiment

| Item | Description | Unit |
|---|---|---|
| Paddy grain | 3438.61 | kg/acre |
| Plant height | 80.8 | cm |
| 100 grains mass | 2.62 | g |
| Length of panicle | 6.9 | cm |
| Length of grain | 8.0 | mm |
| Width of grain | 1.9 | mm |
| Slenderness ratio | 3.4 | - |

*Table 4 :* Importance of measuring moisture content (Courtesy: IRRI)[10Ath]

| Operation | Desired Moisture Content | Primary losses |
|---|---|---|
| Harvesting | 20-25 % | Shattering if grain is too dry |
| Threshing | 20-25 % for mechanical threshing<br>< 20% for hand threshing | Incomplete threshing<br>Grain damage and cracking/ breakage |
| Drying | Final moisture content is 14% or lower | Spoilage, fungal damage, Discoloration |
| Storage | <14% for grain storage<br><br><13% for seed storage<br><9% for long term seed preservation | Fungle, insect& rat damage<br>Loss of vigor<br>Loss of vigor |
| Milling | 14% | Grain cracking and breakage over milling |

At each level of paddy moisture, six level of drum speed 5.89, 7.07, 8.25, 9.425, 10.603 and 11.781 m/s were examined. The drum speed was measured with a digital tachometer (Lutron DT-2236). At each test operate; five bundles of paddy crop were fed to the threshing chamber at a constant rate.

To obtain the percentage of broken grain, 10 samples of 100g were randomly chosen from the outlet of the thresher. The broken grains were separated by hand from the whole paddy grains and the weight of the broken grain was recorded. In order to determine the percentage of cracked grain, at each test runs, 10 samples of 50 grains were randomly selected from the outlet of the thresher and manually husked. The husked paddy grains (Ma Naw Thu Kha) were put on a crack tester and the number of cracked kernels was recorded [10Ali].

ii. *Workability*

The second parameter is workability, which is calculated consideration the dynamic stiffness of the shaft for thresher, mainly, the stiffness and mass of threshing period and potential threshing process.

iii. *Percentage of total grain losses (Tgl)*

Visual investigation and manual separation of 10 samples each of 100 grams were used to calculate percentage of damaged and un-threshed grains. And, grain yield was estimated by manual harvesting 5plots each of (1×1 m) with high care from random locations.

The harvested plants were threshed by the thresher; the threshed grains were weight for each sample.

The percentage of total grain losses was calculated from Equation (1), and (2) to determine of threshing efficiency, and then (3) for specific consumed energy: [09Els]

$$T_{gl} = P_d + U_{th} + P_{gl} \tag{1}$$

Where,

(Pd) - Percentage of damage

(Uth) - un-threshed grains

(Pgl) - Percentage of grain losses

iv. *Threshing Efficiency (ηth)*

$$\text{Efficiency} = \left[\frac{(W_1 - W_2)}{W_1}\right] \tag{2}$$

where,

W1 - weight of pure grain output (kg/hr),

W2 - weight of residual grain in the straw (kg/hr).

v. *Specific consumed energy (Se)*

The energy consumed was evaluated from the following formula,

$$S_e = 3.163 F_c / A_p \tag{3}$$

Where:

Fc = fuel consumption, (L/h)

Ap = Actual system productivity = Wg × Pr , (kg/hr)

*c) Performance Analysis with Dynamic Stiffness*

Dynamic stiffness generally creates images of complicated equations with limited practical value. Vibration is merely a response to other conditions in a machine [09Els].

$$\text{Observed Vibration (Response)} = \frac{\text{Force}}{\text{DynamicStiffness (Restraint)}} \tag{4}$$

As Equation (4) shows, vibration can only change as the result of two things: a change in force or a change in stiffness (or both). Also, dynamic stiffness is essential for the machinery specialist.

A change in unbalance is a force changing in a machine. When vibration is viewed as a ratio of forces to stiffness, the perspective changes and the focus becomes what has changed in the machine, the forces acting on its stiffness. A sudden reduction in vibration could signify an increased stiffness. If the excitation force acting on the shaft becomes higher, the Dynamic Stiffness of the shaft must also be increased by checking size of the shaft. Forces and responses (vibration) are vector quantities [08Moh].

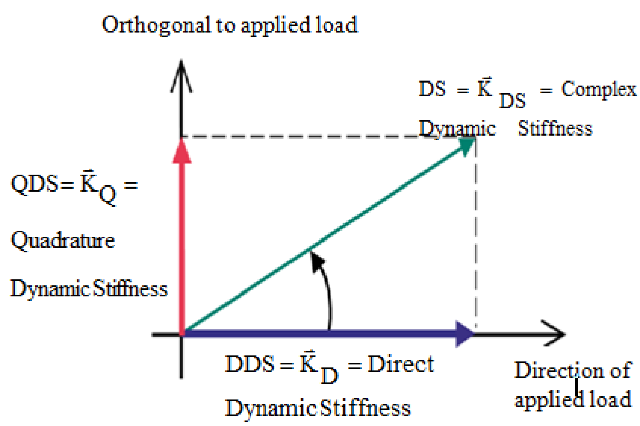


*Figure 3 :* The relationship between Complex, Direct, and Quadrature Dynamic Stiffness

It concern only with synchronous excitation forces in this study. The two orthogonal components of dynamic stiffness, Direct Dynamic Stiffness (DDS) determine how far the shaft moves in the direction of the applied force and Quadrature Dynamic Stiffness (QDS) determines how far the shaft moves to the side (orthogonal to the applied force). Fig. 3 shows the relationship between Complex, Direct and Quadrature Dynamic Stiffness.

i. *Synchronous Dynamic Stiffness*

Dynamic stiffness is the static spring stiffness of the mechanical system complemented by the dynamic effects of mass and damping. The thresher, shaft and damper are represented by mathematical modeling of dynamic stiffness in Fig. 4.

In Fig. 4, M refers to mass, D means damping, K also refers to spring, and λ is the circumferential average velocity ratio. Because of this dynamic motion, both the Quadrature Stiffness due to damping and the mass stiffness effects come into play.

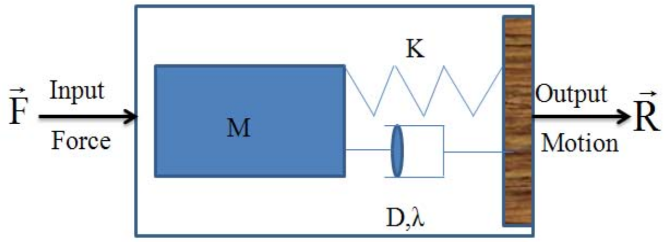


*Figure 4* : Mathematical Modeling of Dynamic Stiffness

The force, F, and the response, R, are vectors, and they have both magnitude and direction. This Equation (5) based on Hooke's Law for spring [07Cha].

$$\vec{K}_{DS} = \frac{\vec{F}}{\vec{R}} \tag{5}$$

By Hooke's Law, Equation (5) can be expressed by (6).

$$F = K\,x \tag{6}$$

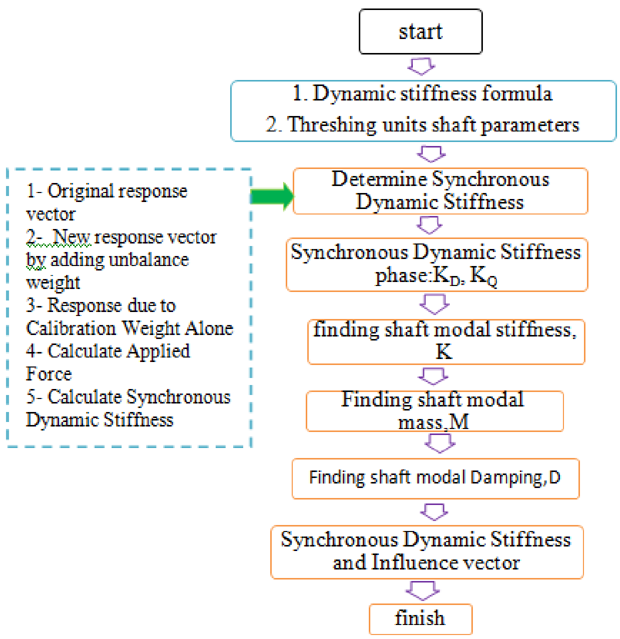


*Figure 5* : Flow chart of the program for the developed Dynamic Stiffness

Dynamic stiffness can be used to estimate the dynamic forces acting in the thresher. There are five basic steps involved in determining Dynamic Stiffness. The above Fig. 5 is step by step flow chart to determine the dynamic stiffness and modal mass including modal damping.

*d) Relation between dynamic stiffness and threshing losses based on positions and engine speeds*

To determine natural frequency for the shaft with dynamic analysis, the total value of stiffness and mass of the shaft must be known. The natural frequency was calculated according to the following Equation (7) [05Joh].

$$\omega_n = \sqrt{\frac{K}{M}} \tag{7}$$

Natural frequency is denoted by ωn, K means spring stiffness of shaft and M refers to mass of shaft. The total value means adding the value from unbalance weight and design consideration. Variable frequency ratio is assumed for evaluate the operating frequencies. The following Equation (8) is used to calculate operating speed [08Rji].

$$V = 2\pi\omega r \tag{8}$$

Where,
V = angular velocity, m/s
ω = operating speed of shaft, rad/s
r = radius of shaft, m

Optimum threshing operations as well as good systems is needed to minimize the loss and obtain maximum efficiency. So, the relation between dynamic stiffness and threshing losses based on operating speeds is shown in Fig. (6).

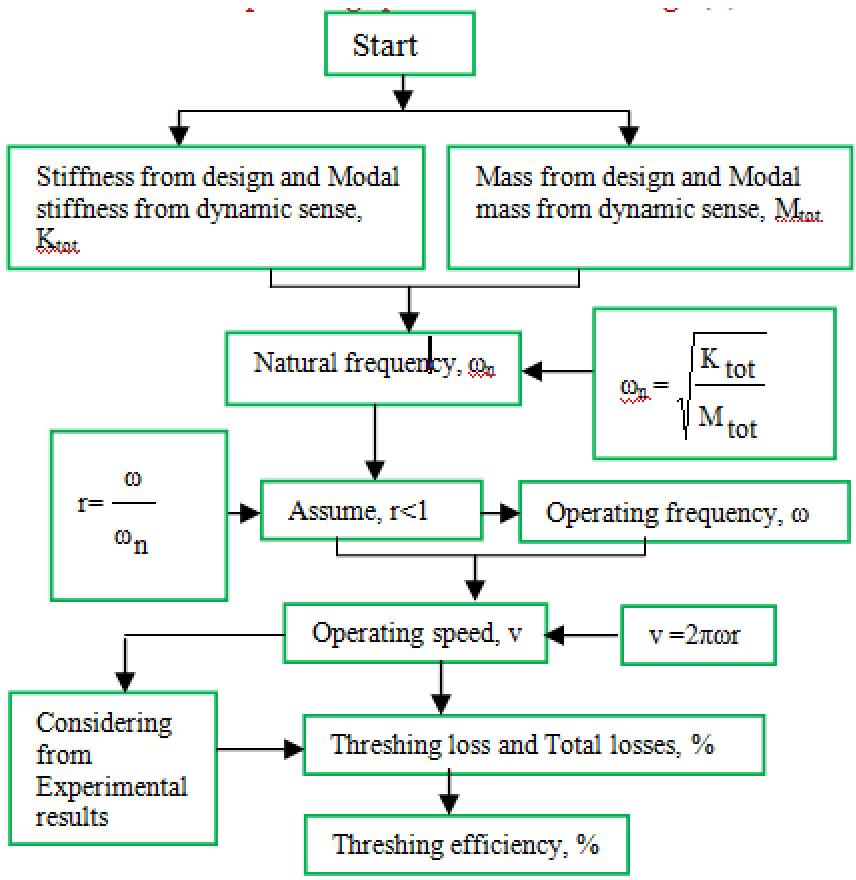


*Figure 6* : Flow chart to determine for the relationship of dynamic stiffness and losses

## III. Results and Discussion

To determine how to relate dynamic stiffness and grain losses is the main contribution. It is important to be stiff because the shaft is attached inside the thresher. Operating the shaft, the thresher also operates in same time. It threshes the grain from the straw as the speed of the shaft. When the speed becomes higher suddenly, the threshing grain can be crush and damage. Also, if the stiffness of the shaft is weak, the shaft can twist and cannot operate well. So, the grain losses can be found due to weak performance.

Therefore, it is vital to be stiff. In this paper, the stiffness of the shaft due to the attaching mass at any positions is determined. Moreover, the relationship of dynamic stiffness and grain losses are shown in the Fig (14, 15 and 16). To plot these, the relation of dynamic stiffness and losses via operating speeds is expressed in Fig (6). The required bode and polar plots are used to examine the response (vibration), dynamic stiffness and modal mass. These are shown in Fig. 7, Fig. 8 and Fig 9.

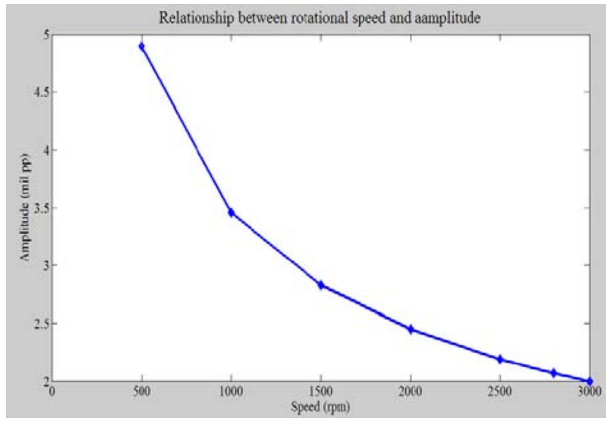


*Figure 7 :* Relationship between rotational speed and amplitude

Operating speeds are selected in relative natural frequency and frequency ratio that exhibit amplitude and phase variations that are acted by placing at various positions attached the unbalance weight.

$A = \sqrt{\frac{12000}{N}}$ and $\theta = \omega t$ are used to do the following plots

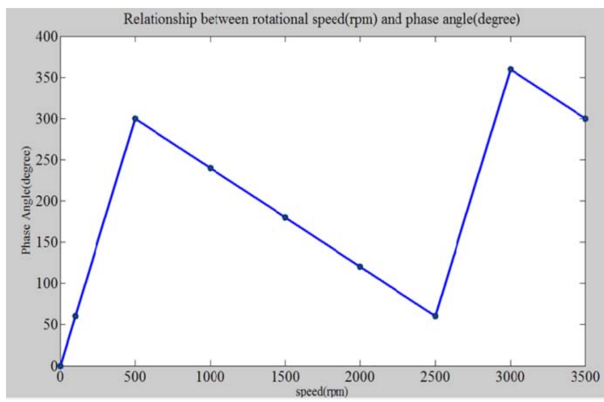


*Figure 8 :* Relationship between angular velocity and phase angle

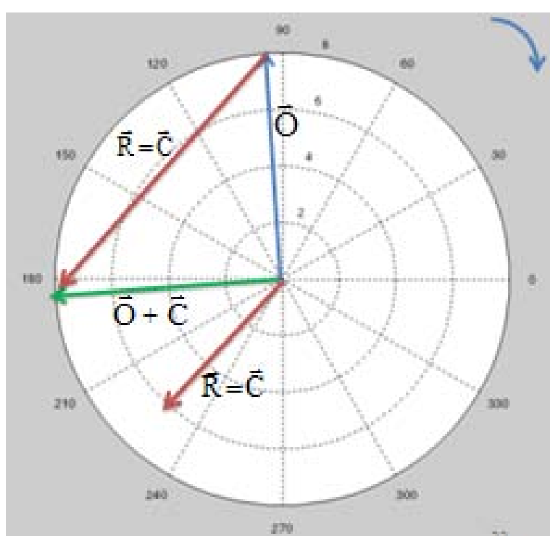


*Figure 9 :* Denote on polar plot

Performance for the un-threshed grain losses, total grain losses, and threshing efficiency is determined based on experimental performance and dynamic analysis. Table (5) shows experimental performance result. These results based on 17% moisture content. If the machine threshes at 20% (w.b) moisture content, the results of grain losses will be decrease.

Un-threshed weight correction locations are been modally effective. If the unbalance weight is increased, an unexpected result will be produced. When adding calibration weights, check the response vector C. Small vector changes indicate a lack of sensitivity to the weight. Care should be taken when analytically modeling the shaft.

*Table 5 :* Result of Experimental Performance

| No. | Items | Values | units |
|---|---|---|---|
| 1 | Actual performance rate, $P_r$ | 0.5625 | acre/h |
| 2 | Percentage of damage, $P_d$ | 19.231 | % |
| 3 | Un-threshed grains, $U_{th}$ | 2 | % |
| 4 | Percentage of grain losses, $P_{gl}$ | 5.263 | % |
| 5 | Percentage of total grain losses,$T_{gl}$ | 12.263 | % |
| 6 | Field efficiency, $\eta_t$ | 77.49 | % |
| 7 | Cutting efficiency, $\eta_c$ | 62.87 | % |
| 8 | Cleaning efficiency, $\eta_{cl}$ | 90 | % |
| 9 | Threshing efficiency, $\eta_{cl}$ | 97.77 | % |
| 10 | Specific consumed energy, $S_e$ | $7.8\times 10^{-3}$ | kW.h /kg |
| 11 | Fuel consumption per hour | 3.08 | gal/ hr |
| 12 | Fuel Cost per Acre | 31,197 | kyats/acre |
| 13 | Labour Cost par Acre | 13750 | Kyat/acre |

The results at 90 degree position are shown in Table (6). These results are determined based on Fig 7, 8 and 9. Table (7) is to compare for the values of stiffness and modal mass from design and unbalance weight condition.

*Table 6 :* Result of Unbalance Weight at 90° position

| Items | Parameters | Symbols | Values |
|---|---|---|---|
| 1 | Original response (mil p-p) | $\vec{O}$ | $2.98\angle 90.6°$ |
| 2 | New response (mil p-p) | $\vec{O}+\vec{C}$ | $3.8\angle 180.6°$ |
| 3 | Response due to calibration weight | $\vec{R}=\vec{C}$ | $4.8\angle 218.71°$ |
| 4 | Applied force of calibration weight | $\vec{F}$ | $0.2902\angle 90°$ |
| 5 | Synchronous dynamic stiffness | $\vec{K}_{DS}$ | $4730.9\angle 129°$ |
| 6 | Modal mass (kg) | M | 0.0451 |
| 7 | Modal stiffness (N/m) | K | 903.4586 |
| 8 | Modal damping (N.s/m) | D | 12.539 |
| 9 | Influence vector (mil p-p) | $\vec{H}$ | $26.83\angle 129°$ |

*Table 7 :* Comparison of Modal Mass and Stiffness of Shaft and Unbalance (90° position)

| Items | Modal mass | Stiffness |
|---|---|---|
| Shaft | 34.68 kg | $3.9716\times 10^{9}$ N/m |
| unbalance | 0.1398kg | 2799 N/m |

For all Fig. 7 to Fig. 10, 90 degree condition result of unbalance weight is only considered for the relationship of dynamic stiffness and losses. This position is applied as the operating speed of the shaft approaches to the critical speed, the center of rotation begins to shift toward the CG. The phase angle between the exciting force (direction of the unbalance) and the actual vibration will be 90 degree. In this particular case, the vibration (response) amplitude lags the unbalance by 90°. So, dynamic stiffness for dynamic analysis is determined at 90° position calibration weight.

At 90 degree position, the values of stiffness decrease slightly in each engine speed (9.425, 10.2102 and 10.996 m/s. So, 903.47 N/m of stiffness can be accepted for this engine speed, 10.996 m/s.

*Table 8 :* Results of Stiffness for (Unbalance) calibration weight

| Engine speeds (m/s) | 9.425 | 10.2102 | 10.996 |
|---|---|---|---|
| Position (degree) | Stiffness (N/m) | Stiffness (N/m) | Stiffness (N/m) |
| 30 | 3143.427 | 2941.34 | 2799 |
| 60 | 1696.7251 | 1587.6 | 1511 |
| 90 | 1014.6117 | 949.33 | 903.47 |
| 120 | 600.5195 | 561.934 | 534.81 |
| 150 | 287.5591 | 269.0757 | 256.15 |
| 180 | 20.0446 | 18.7543 | 17.851 |

Table (8) refers to the effect of stiffness at various unbalance weight position by considering for each engine speeds. These relationships are shown by bar chart in Fig. 10. In this bar chart, the values of dynamic stiffness decrease steadily in the forward engine speeds (9.425, 10.2102 and 10.996 m/s). Also, these values become low from 30 degree to 180 degree. The highest dynamic stiffness can be found at the lowest speed.

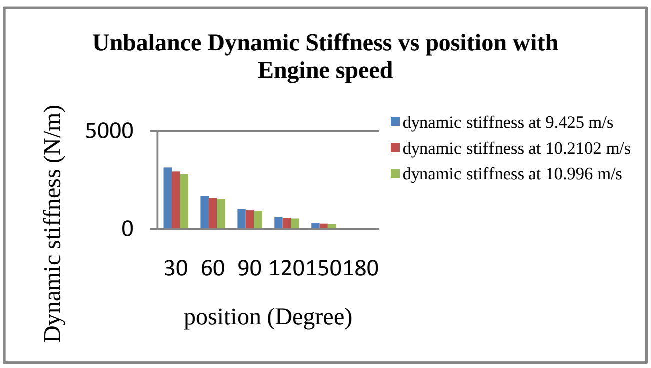


*Figure 10 :* Independent effect of dynamic stiffness (unbalance) on position

*Table 9 :* Results of Modal mass for (Unbalance) calibration weight

| **Engine speeds (m/s)** | **9.423** | **10.2102** | **10.996** |
|---|---|---|---|
| **Position (degree)** | **Modal mass (kg)** | **Modal mass (kg)** | **Modal mass (kg)** |
| 30 | 0.157 | 0.1469 | 0.1398 |
| 60 | 0.0848 | 0.0793 | 0.0755 |
| 90 | 0.0507 | 0.0474 | 0.0451 |
| 120 | 0.03 | 0.0281 | 0.0267 |
| 150 | 0.0144 | 0.0134 | 0.0128 |
| 180 | 0.001001 | 0.000937 | 0.000892 |

Table (9) shows the value of modal mass for (unbalance) calibration weight on various attached position. In this condition, Modal mass means trial mass, which is used during balancing to make temporary mass distribution on the shaft.

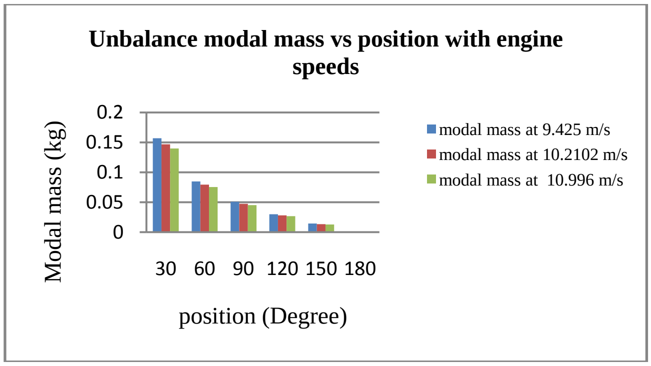


*Figure 11 :* Independent effect of modal mass (unbalance) on position

Fig. 11 can shows clear the change of modal mass for unbalance at various positions with engine speeds. There are three various modal mass values for each 9.425, 10.2102 and 10.996m/s. The graph is decreasing slightly from the position of 30 to until 180 degree.

*Table 10 :* Results of Total Stiffness including calibration weight

| Engine speeds (m/s) | 9.425 | 10.2102 | 10.996 |
|---|---|---|---|
| Position (degree) | Stiffness×1 0 9 (N/m) | Stiffness×1 0 9 (N/m) | Stiffness×10 9 (N/m) |
| 30 | 3.971603143 | 3.971602941 | 3.971602799 |
| 60 | 3.971601697 | 3.971601588 | 3.971601511 |
| 90 | 3.971601015 | 3.971600949 | 3.971600903 |
| 120 | 3.971600601 | 3.971600562 | 3.971600535 |
| 150 | 3.971600288 | 3.971600269 | 3.971600256 |
| 180 | 3.97160002 | 3.971600019 | 3.971600018 |

Table (10) shows total stiffness including the value of calibration weight and shaft from design consideration depends upon engine speeds. The shaft for this thresher withstands strength, resist to unbalance weight so that it is stiffness dynamically. In dynamic analysis, the value of operating speed is determined based on natural frequency and frequency ratios.

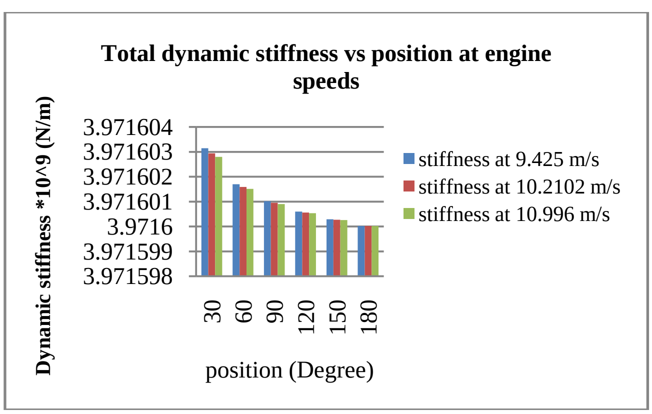


*Figure 12 :* Desired output of stiffness versus position

In Fig. 12, the total dynamic stiffness increased with decreasing engine speeds (9.425 m/s) for the reason that less feed rate into the threshing drum resulted in less impact force on the material. The forward engine speed of machine had significant effect on decreasing quality dynamic stiffness as speed increased.

*Table 11 :* Results of Total Mass including calibration weight

| Engine speeds (rpm) | 9.425 | 10.2102 | 10.996 |
|---|---|---|---|
| Position (degree) | Modal mass (kg) | Modal mass (kg) | Modal mass (kg) |
| 30 | 34.84 | 34.8269 | 34.82 |
| 60 | 34.7648 | 34.7593 | 34.76 |
| **90** | **34.7307** | **34.7275** | **34.7251** |
| 120 | 34.71 | 34.7081 | 34.7067 |
| 150 | 34.6944 | 34.6934 | 34.6928 |
| 180 | 34.681 | 34.68094 | 34.681 |

Table (11) shows the result of total mass (kg) with against to engine speed (rpm) and unbalanced weight position attachment (degree) on the shaft. The design requirement must be nearly the same with the critical speed for the operating speed to approach the C.G point of the shaft.

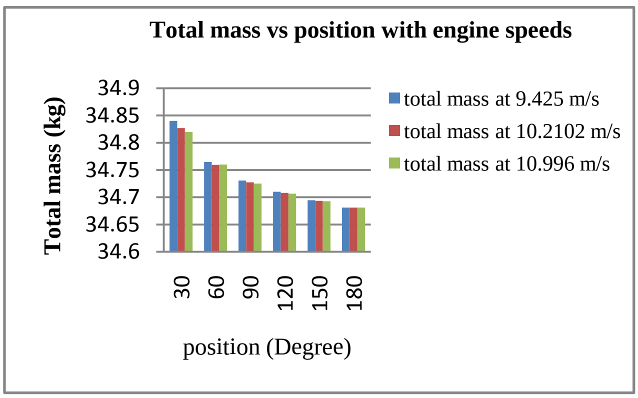


*Figure 13 :* Desired output of mass versus position

In this Fig. 13, it can see clear the change of total mass at various position with engine speeds. There are three various total mass values for each 9.425, 10.2102 and 10.996m/s. The graph is decreasing slightly from the position of 30 to until 180 degree.

During forward engine speeds (9.425, 10.2102 and 10.996 m/s), the values of un-threshed and total grain losses by changing operating speeds are shown in Fig. 14. According to this result chart, grain losses become increased steadily in each speed by dynamic stiffness consideration. These values also depend upon frequencies ratio. As the frequency ratio increases, the grain losses will follow. So, the grain losses need to adjust balance condition for frequency ratio.

At various speeds, losses are not different, nearly equal and the least in percentage in losses. So, it is satisfied to apply as a shaft of thresher in this combine harvester. Operating speeds, un-threshed and total losses are same each various positions attaching unbalance weight in three forward engine speeds.

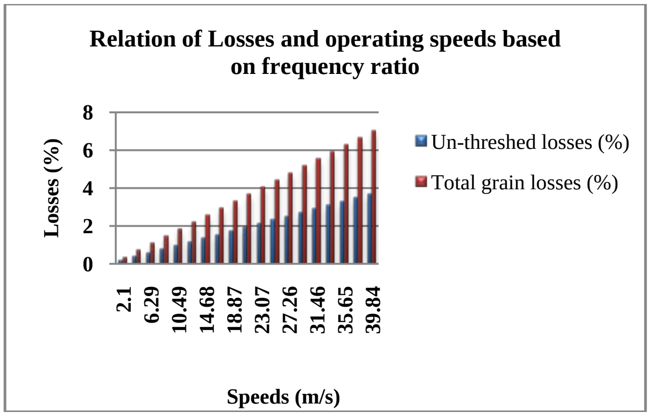


*Figure 14 :* Desired output of losses on frequency ratio

Fig. 14 refers to the information of whatever the speed changes, all the operating speed and losses are still nearly equal at any attachment of unbalanced weight position. It is only for 90 degree position unbalanced weight at each operating speed.

It can be seen that at each level of drum speed tested, the un-threshed and total grain losses increased significantly as the drum speed increased from 2.1 m/s to 39.84 m/s. However, higher value of grain losses was obtained at higher drum speed. The most un-threshed and total grain losses are 3.7 % and 7.05 % at 39.84 m/s, and then the least value of 0.19 % and 0.37 % by dynamic stiffness consideration were observed at drum speed of 2.1 m/s.

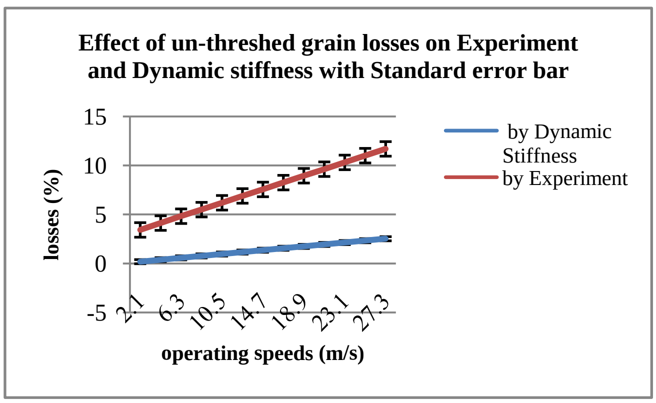


*Figure 15 :* Effect of Standard error bar for un-threshed grain losses by Experiment and Dynamic Stiffness condition

All the value of un-threshed and total grain losses at each operating speeds with standard error bars are expressed in Fig. 15 and 16. These values are equal for various unbalance weight positions with experiment and dynamic analysis. The effect of drum speed on the value of un-threshed grain losses for both experiment and dynamic stiffness are shown in Fig. 15 and the value of total grain losses for both experiment and dynamic stiffness are shown in Fig. 16.

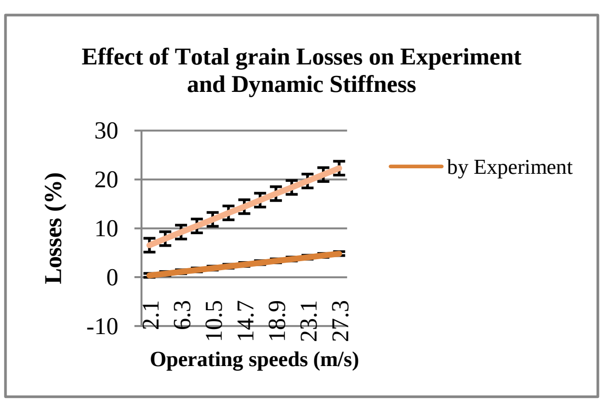


*Figure 16 :* Effect of Standard error bar for Total grain losses by Experiment and Dynamic Stiffness condition

The results revealed that un-threshed grain losses increased steadily in the dynamic stiffness. By testing dynamic stiffness, the result of un-threshed grain losses increased significantly in experiment as the paddy moisture content decreased at 21.5, 20% and 17 % (w.b). It was observed that at each level of drum speed tested. But, the lower un-threshed grain losses observed at higher paddy moisture content with given drum speeds.

## IV. Conclusion

The paddy moisture content and the drum (or shaft) speeds significantly affected the total losses during paddy threshing by placed the shaft in the axial-flow thresher tested. The maximum total grain losses were obtained at shaft speed 39.84 m/s, frequency ratio 0.95 and moisture content 17 % in Fig (14). The values of total grain losses are 12.263 % and 7.05% for each theory with experiment in the paddy field and dynamic stiffness at 90 degree unbalance weight position.

The grain losses decrease in the suitable moisture content 20% (w.b). So, threshing losses was more increase than determining by dynamic stiffness. Comparing these two results, the total grain losses due to the dynamic stiffness is more satisfied than experimental field condition.

In order to minimize the effect of shaft or drum speed on total grain losses in the axial-flow thresher, it is recommended that the threshing operation should be performed immediately after crop harvesting. Performances due to dynamic analysis of the shaft for thresher have been reviewed in this paper. The two main topics include: experimental measurement techniques and dynamic stiffness of the shaft with various operating speeds.

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## References Références Referencias

1. [10Ath] Dr. Athapol Noomhorm, 2010, “Rice Harvesting and post-harvest Technologies in Myanmar, A Training Manual, The Asian Institute of technology of Thailand (ATT), Thailand.
2. [10Ali] M.R. Alizadeh, 2010, “ Effect of Threshing Drum Speed And Crop Moisture Content on The Paddy Grain Damage In Axial-Flow Thresher,” Department of Agricultural Engineering, Rice Research Institute of Iran, Rasht, Iran.
3. [09Els] Elsaied, G.H., 2009, “ Studying a New Combine Threshing Rotor Design,” Agric. Eng. And Director of Ag. Eng. Res. Institute, Egypt.
4. [08Moh] Mohamed Sayed Omran, 2008, “A Comparative Study of The Most Widely Harvesting Systems for Wheat Crop,” Egypt.
5. [08Raj] Dr. Rajiv Tiwari, 2008, “ Analysis of Simple Rotor Systems,” Department of Mechanical Engineering, Indian Institute of Technology, Guwahati 781039.
6. [07Cha] Charles Phelps, “ Basic Concepts of Rotor Dynamics,” Field Engineer, GE Energy, 312 Thompson Ave, Lehigh Acres, FL 33972.
7. [05Joh] John E. Judd, 2005, “ Basics of Acceleration Measurements,”: Dynamic Measurement Consultants, LLC Hamden, CT.
8. [05Sor] Sorin Weissman, Alfa Wasserman, Inc., 2005, “ A Pratical Review of Rotating Machinery Critical Speeds and Modes,” West Caldwell, New Jersey.
9. [04Tro] Troy D.Feese, P.E. and Phillip E. Grazier, Senior Project Engineer, 2004, “ BALANCE THIS! CASE HISTORIES FROM DIFFICULT BALANCE JOBS,” Engineering Dynamic Incorporated, San Antonio, Texas.