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Analysis of Vibration Severity of Paddy Thresher of Centrifugal Type

Zaimar*1, Mursalim2, H. Abbas3, Supratomo4

[1]Department of Agro Industry, Agricultural Polytechnic of Pangkep State, South Sulawesi, Indonesia
[2,4]Department of Agricultural Engineering, Hasanuddin University, Makassar, Indonesia
[3]Department of Mechanical Engineering of Hasanuddin University, Makassar, Indonesia

ABSTRACT

A common problem occurs on rotating machines is vibration. This vibration would affect machine performance. This vibration should be tested and analyzed for ensuring machine runs smoothly that can extend the life of the machine. Therefore, the purpose of this research was to analyze of vibration severity of the paddy thresher with centrifugal type. The experimental results indicated that the velocity amplitude of the vertical direction is higher than the velocity amplitude of vertical and horizontal at all levels of the impeller speeds except in the range of 400 to 800 RPM. The highest velocity amplitude reached about of 7.2 mm/s in the vertical direction at 1120 RPM. Based on the statistical analysis of the RMS, crest factor, and shape factor methods, the operating condition for ensuring the equipment run accordingly the criterion is *good* or *allowance* then it is recommended choosing in the range of the impeller speed of 480 to 800 RPM.

Keywords: Centrifugal Fan, Impeller Speed, Paddy Thresher, Velocity Amplitude, Vibration Severity

I. INTRODUCTION

A common problem with rotating machines is the issue of vibration. This vibration will affect machine performance and operator comfort. In reference [12] when the system vibrates, the energy in the system alternately changes back and forth between the kinetic and potential energy. In the absence of a mechanism to take out energy from the system, theoretically the machine will vibrate forever when un-damped. Damping is the energy that converts the kinetic energy and the potential becomes heat and then takes energy from the vibrating system [1].

Equipment with rotating components will due to vibration with the amount of vibration that occurs due to the effect of factors cumulatively, such as unbalanced and misalignment of the rotating engine components, operating factor, and the dynamic characteristics of the complete assembly [6].

Design errors or improper mounting factors in the location and position of the impeller and the shaft that are inconsistent with the center of the rotation mass, it is possible causing unbalance the rotating impeller, cyclic vibration, and ultimately lead a fatigue on the machine [5]. The fan operating factor in which the fan is run under unsuitable electrical current conditions such as the lower current that can cause the peak pressure so that can occur an instability of the rotating fan [9].

A vibration can arise from the transfer of cyclic forces through the elements of the machine, where it interacts with each other's and energy dissipated through the structure in the form of vibration. Damage or wear and deformation will alter the dynamic characteristics of the system and tend to increase vibration energy [14].

Vibration analysis is used for ensuring fans run smoothly and can extend the life of the fan. The two components of vibration that concerned by the fan manufacturer are their amplitude and frequency [5]. Monitoring of machine components, especially in rotating parts such as bearing and shaft is essential to be done in stages to ensure the reliability of the rotating engine so as to minimize damage such as an excessive wear and optimize equipment life. The excessive wear occurs due to imbalance, misalignment and the occurrence of resonance and all of these can cause the machine failure [10]. The mechanical vibration analysis already proved as the most successful predictive tool, both in increasing equipment availability and reliability [8].

The reduction of engine vibration in all points tested from the redesign is a positive sign of reducing engine fatigue that results in longer life because it could be balancing the third directions horizontal, vertical, and axial. Redesigned for decreasing engine vibration would yield advantages: optimal instrument readiness, damage to minimal, maximum bearing life, low maintenance cost, smoother engine operation, decreased fuel consumption, improved engine efficiency [13].

Therefore, the purpose of this research was to analyze of vibration severity of the paddy thresher with centrifugal type.

II. METHODS AND MATERIAL

This research used the main tools: the paddy thresher uses a modified centrifugal fan with the serrated blade impeller that function as a thresher component. The impeller was driven by an electric motor of 0.75 HP. The secondary tools were a vibration meter (model GM63A), a level meter, an analog tachometer, tang ampere-meter, and calculator.

Vibration measurement applied by seven levels of the impeller speeds (400; 600; 800; 960; 1120; 1200; and 1400 RPM). The vibration meter puts at the bearing that connected to the impeller [14] with three dimensions of X (horizontal), Y (vertical), and Z (axial) axes for measuring of velocity amplitude of each level of the impeller speed as presented in Fig. 1.

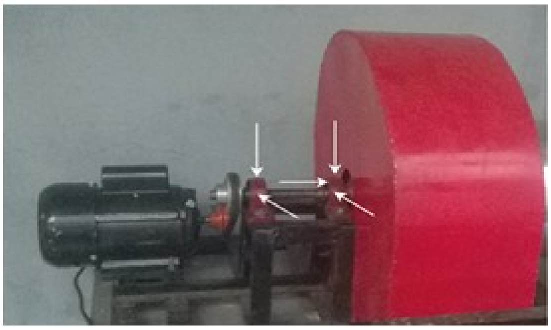

Figure 1: The position of the vibration meter on both the bearing for equipment

Measurements performed as much as ten times repetition at each speed of the impeller. The replications are obtained of each axis of measurement then analyzed and plotted on the graph of the relationship between the impeller speeds with an RMS of velocity amplitude. The velocity amplitude measured by a vibration meter. The results of velocity amplitude could compare with the standard of vibration severity.

To take a proper measurement of the fan vibration is in the bearing of the shaft of the impeller because through this component the primary vibration is transmitted and the point or location of the vibration measurement that requires a space and not covered or avoiding the risks on the rotating or moving parts [11].

Statistical Analysis

1. RMS (Root Mean Square)

A simple approach for determining the vibration severity is RMS method. The average of the vibration velocity is determined as the root mean square (RMS) of the velocity amplitude of horizontal, vertical, and axial direction [2] according to the equation (1).

$$RMS_v = \frac{1}{N}\sqrt{\sum_{i=1}^{N}(v_{xi}^2 + v_{yi}^2 + v_{zi}^2)} \quad (1)$$

Where:

RMS_V = RMS of velocity amplitude (mm/s)

v_X = velocity amplitude of horizontal direction (mm/s)

v_Y = velocity amplitude of vertical direction (mm/s)

v_Z = velocity amplitude of axial direction (mm/s)

As guiding in determining the vibration severity level of centrifugal fan could use the following Table 1. It is the broadest and most general reference, but provided a valid starting point [11]. Standard of ISO 2732 of vibration severity for machine with power below of 20 HP [7] can be used for determining the level of the vibration severity as presented in Table 1 or Table 2 below.

TABLE 1
THE STANDAR OF VIBRATION SEVERITY FOR CENTRIFUGAL FAN

Range of RMS Vibration velocity (mm/s)	Criteria
0.000 - 0.127	Extremely smooth
0.127 - 0.254	Very smooth
0.254 – 0.508	Smooth
0.508 – 1.016	Very good
1.016 – 2.032	Good
2.032 - 4.064	Fair
4.064 – 8.001	Slightly rough
8.001– 16.002	Rough
Above of 16.002	Very rough

TABLE 2
THE ISO 2732 FOR VIBRATION SEVERITY OF MACHINE BELOW 20 HP

Range of RMS Vibration Velocity (mm/s)	Criteria
0.28 – 0.71	A (good)
0.71 – 1.80	B (allowable)
1.80 – 4.50	C (Tolerable)
Above of 4.5	D (Not permissible)

2. Crest Factor

A good approach for determining the vibration is a crest factor method. The crest factor (CF) is a dimensionless factor defined as the ratio of the peak value to the RMS value of velocity amplitude. The crest factor was calculated using the formula [2] as below.

$$CF = \frac{\text{Peak}}{\text{RMS}} \tag{2}$$

Where:

CF = Value of crest factor

Peak = Maximum amplitude of velocity amplitude (mm/s)

RMS = Root mean square of velocity amplitude (mm/s)

3. Shape Factor

Shape factor (SF) is a dimensionless defined as the ratio between values of RMS and average values of velocity amplitude [2]. The shape factor is one of symptom parameters. In the damage diagnosis the symptom parameters used to identify conditions machine, as it indicates the information by signal of measurement that determined with the formula (3) below.

$$SF = \frac{\text{RMS}}{\mu} \tag{3}$$

Where:

SF = value of shape factor

μ = average of velocity amplitude total (mm/s)

III. RESULTS AND DISCUSSION

The results of velocity amplitude measurement of the level of the impeller speed can be presented in the following Fig. 2 as below.

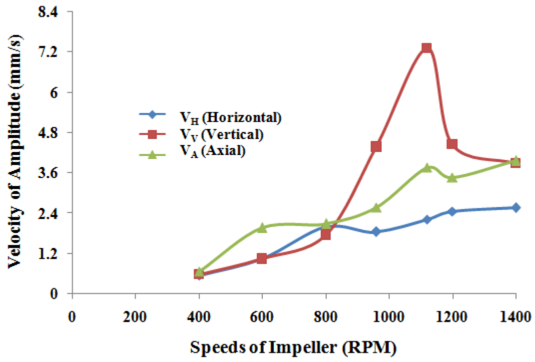

Figure 2: The velocity amplitude of horizontal, vertical and axial direction for equipment

In Fig. 2 shows that at lower speed of the impeller speed reached the velocity amplitude with a relatively small increase. In reference [3] show that at lower speed, this vibration could ignore, but at moderate and higher speed this vibration becomes larger and various parts of machines no longer move smoothly due to the vibration.

The velocity amplitude of horizontal and axial direction tend to increase linearly toward the increase the impeller speeds of operation in the range of 400 to 1400 RPM except for the velocity amplitude of vertical increase quadratic in the range of 800 to 1200 RPM. The highest velocity amplitude reached about of 7.2 mm/s in the vertical direction with the impeller speed of 1120 RPM. In reference [1] the higher vibration amplitude even when the exciting force is due to a small imbalance. Another characteristic of such a system is the large changes in the vibration level occur with little change in input frequency.

The velocity amplitude of the vertical direction is higher than the velocity amplitude of horizontal and axial in the range of 800 to 1400 RPM (Fig. 2). This might due to because the speed of the impeller causes the centrifugal forces of the fan, which results in the higher aerodynamic pressure and greater rotation toward the vertical direction. In reference [11] this is because the vibration reading of the specific axes gives the best an indication of a particular problem. For example, an imbalance is generally indicated a high vibration readings in the horizontal or vertical direction. Nonconformist indicated with the high vibration readings in the axial direction.

To reduce the high velocity amplitude then done by checking and re-installing the all components in accordance with the data of this test result. In reference [13] the reduction of vibration amplitude on each vertical, horizontal, and axial direction with a proper redesign of the machine component is a positive method for reducing engine fatigue resulting in longer life.

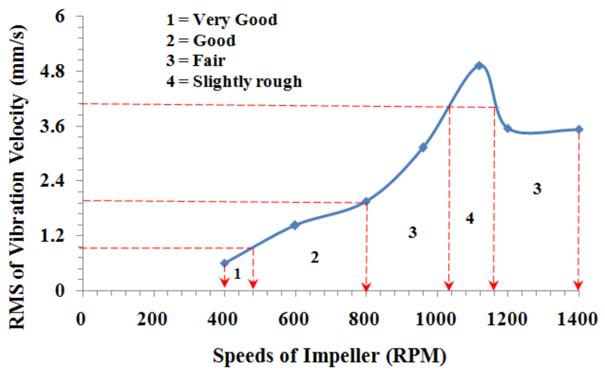

Figure 3: The performance of vibration severity in the equipment with the RMS method

In Fig. 3 shows that the vibration severity of the paddy thresher of centrifugal type in the range of 400 to 480 RPM is *very good*, in the range 480 to 800 RPM is *good*, and the range of 600 to 1040 RPM is *fair*. The operation of the rotating of the impeller above 1000 RPM on the condition of the paddy thresher of centrifugal type is not possible because it is very risky at vibration which could cause the equipment damaged periodically.

The calculating results of the RMS, crest factor, and shape factor used the formula (1), (2), and (3) could be presented as in Table 3 below.

TABLE 3
THE RESULT OF THE RMS, CREST FACTOR AND SHAPE FACTOR OF VIBRATION VELOCITY

Speed of Impeller (RPM)	RMS (mm/s)	CF	SF
400	0.598	1.103	1.008
600	1.416	1.385	1.051
800	1.952	1.066	1.003
960	3.122	1.403	1.064
1120	4.921	1.487	1.110
1200	3.555	1.254	1.028
1400	3.526	1.123	1.017

I ne range of 400 to 800 RPM and 1200 to 1400 RPM, meanwhile increased in the range of 800 to 1120 RPM. This range occurs the stability of vibrating with different of velocity amplitude very small compare with the range other is the high value of the crest factor of 1.487 (Table 3). The higher value of the crest factor occurs the peak of velocity vibration from input the vibration velocity.

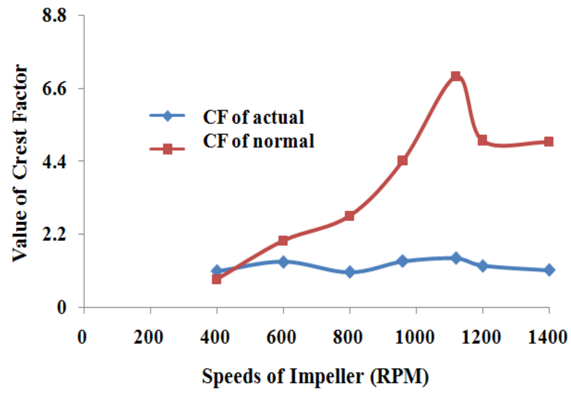

Figure 4: Value of the crest factor

In the normal operating condition, a sinusoidal motion with the normal value of the RMS is about 71% of the peak value or the value of crest factor is √2 or 1.414 times the RMS value so that obtained the normal value of the crest value of 0.87 to 6.95. The actual value of the crest factor indicated that the values of the crest factor in all the impeller speeds is below of the normal condition, except in the range of 400 to 480 RPM with the value of the crest factor exceeds from the normal value (Fig. 4). In reference [2] a value of the crest factor can indicate a typical vibration signal from a machine with imbalance.

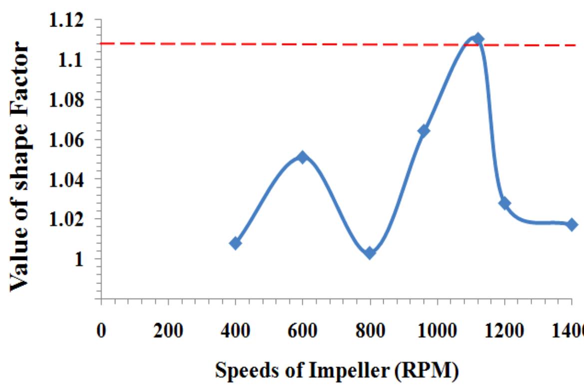

Figure 5: Value of the shape factor

A sinus wave or vibration that measured in a normal condition had an average of velocity amplitude or the RMS is 0.707 times the peak value and the average of the velocity amplitude (μ) is 0.637 times the peak value so that can determine the normal standard of the shape factor is 1.11 that obtained from 0,707 divided by 0.637 [2, 4].

In Figure 5 shows that this is two signals of vibration with a peak at the impeller speed of 1120 RPM which describe the form of vibrating signal exceeds from the normal position. At all the impeller speed had the value of the shape factor is below the normal standard except in the range of 1120 to 1140 RPM with the value of the shape factor of 1.101. In reference [2] a value of the shape factor also present changes under unbalance and misalignment.

IV. CONCLUSION

The experimental result indicated that the velocity amplitude of the vertical direction is higher than the velocity amplitude of vertical and horizontal at all levels of the impeller speeds except in the range of 400 to 800 RPM. The highest velocity amplitude reached about of 7.2 mm/s in the vertical direction with the impeller speed of 1120 RPM.

Based on the statistical analysis of the RMS, crest factor, and shape factor methods, this operating condition for ensuring the equipment run accordingly the criterion is *good* or *allowance* then it is recommended choosing in the range of the impeller speed of 480 to 800 RPM.

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