Abstract

Machines are growth engines of the economy; each sector of the economy achieves its demand by the use of machine. They are installed in various establishments for the purpose of using them to perform certain functions or the others. However, as a result of the kind of forces, dynamic and static loads, they transmitted to their adjoining surroundings when used, they are often mounted on supporting structures, foundations or combination of them to achieve adequate or appropriate safe operation and stability. When a machine is operating, it is subjected to several time varying forces and as a result of which it tends to exhibit vibrations. In such a situation or process, certain quantity of this force is transmitted to the foundation, which could undermine the life of the foundation and also affect its performance and the operation of any other machine on the same foundation. Hence, it is of interest to minimize this force transmission. This research aimed at developing an adjustable steel framed structure for supporting the major components of a 5.0 kW micro-steam power unit (steam, turbine and alternator) and evaluating the performance of the unit with or without vibration isolator when they are axially connected with flexible flange coupling or transversely connected with sets of belt and pulley, in succession, respectively. The results showed that reduction in the force transmitted to the supporting structure occurred when the vibration produced by the unit is isolated from its base by the use of vibration isolator, maximum reduction of 99.95 % achieved when axially coupled and 99.91 % when transversely connected with belt and pulley system. The results also showed that better performance would be attained when the steam turbine is axially coupled to the alternator than when connected with belt and pulley; Maximum voltage of 52V and speed of 1000 rpm at 77 dB sound level attained with coupling connection, and voltage of 20 V and speed of 752 rpm at 75 dB with belt and pulley connection.

1. Introduction

Machines are growth engines of the economy. Each sector of the economy fulfils its demand by use of machine. Machines plays vital role in economy. The use of machine since past few decades has been increasing rapidly and it is not only in industrial/rural areas but in small isolated/ rural areas also.

Machines are installed in various establishments for the purpose of using them to perform certain functions or the others. However, as a result of the kind of forces, dynamic and static loads, they transmitted to their adjoining surroundings, they are often mounted on supporting structures, foundations or combination of them to achieve adequate or appropriate safe operation and stability

When a machine is operating, it is subjected to several time varying forces and as a result of which it tends to exhibit vibrations (period and aperiodic oscillations). In the process, certain quantity of this force is transmitted to the foundation – which could undermine the life of the foundation and also affect the operation of any other machine on the same foundation. Hence, it is of interest to minimize this force transmission. Similarly, when a system is subjected to ground motion, part of the ground motion is transmitted to the system In this case, appropriate measure must be taken to minimize the motion transmitted from the ground to the system.

According to Srinivasulu and Vaidyanath (1990); Rajput and Dubey (2012), Codulo *et al*., (2016), Foundations are structural elements that transfer loads from the superstructure to the underlying soil. A structure may be supported on a system of individual foundations, or on a single large foundation. Machine foundations require a special consideration because they transmit dynamic loads to both the supporting structures (frames) and the soil in addition to static loads due to weight of these supporting structures and the machine and accessories. Therefore, the design engineers must consider, in addition to the static loads, the dynamic forces caused by the working of the machine. These dynamic forces are, in turn, transmitted to the foundation supporting the machine, and as such the foundation response measured at some degrees of freedom of the machine supports is mainly affected by the force acting at the same degree of freedom of the response or at least by the forces acting at some degrees of freedom located near the considered one (Srinivasulu and Vaidyanath, 1990; Bhatia, 2006;).

Basically, foundations are designed by considering the impulses (shocks) and dynamic loads (vibrations) resulting from operation of the machine. However, for effective use of machine within tolerance limit integrated approach of both civil and mechanical/electrical engineers is required from planning stage to execution stage for satisfactory performance of machine without causing harm to operator and other structural part.

The history of foundations extends for thousands of years; For example, 4,000 to 5,000 years ago the alpine lake dwellers in Europe used timber piles to support their houses. Also, in the year 55 BCE, Julius Caesar built a pile-supported bridge across the Rhine River to facilitate his conquest of Gaul. With reference to the supports of machinery, Heavy machinery with reciprocating, impacting, or rotating masses requires a support system that can resist dynamic forces and the resulting vibrations, and as such many engineers with varying backgrounds are engaged in the analysis, design, construction, maintenance, and repair of machine foundations (Hodgkinson, 1986; Codulo *et al*., 2016).

Vibration is a repetitive, periodic, or oscillatory response of a mechanical system. The rate of the vibration cycles is termed "frequency." Repetitive motions that are somewhat clean and regular, and that occur at relatively low frequencies, are commonly called oscillations, while any repetitive

motion, even at high frequencies, with low amplitudes, and having irregular and random behaviour

falls into the general class of vibration. Vibrations can occur naturally and may be representation of free and natural dynamic behaviours in engineering systems; it may be good vibration, when it serves a useful purpose, and bad when unwanted, undesirable and destructing. The most common sources of vibration in machinery are related to the inertia of moving parts in the machine. Some parts have a reciprocating motion, accelerating back and forth. In such a case Newton's laws require a force to accelerate the mass and also require that the force be reacted to the frame of the machine. The forces are usually periodic and therefore produce periodic displacements observed as vibration (Alphouse, 2015). Hence, stability of the machine with reference to its supporting structure, in terms of balancing is essential

According to Derek (2006), Balancing is one of the enabling technologies that allow development and production of more powerful and efficient equipment. Balancing is a cost saving, efficiency boosting process. Balancing is a way to reduce vibration and bearing loads to improve performance and reliability. Balancing reduces the loads on the bearings, since the bearing life of any system is proportional to load and speed. Thus, according to Bhatia (2006), by limiting the centrifugal forces to less than 10 % of the static load bearing life is maximized. Balancing reduces vibration. Vibration causes parts to become loose, generates noise, and produces a perception of low quality. Balancing is the correction of manufacturing problems. No manufacturing material or process is perfect and errors in manufacturing and assembly combine and build up with the result that the final product has too much noise, vibration, shaft bending, bearing load or performance loss to meet the final testing requirements. Balancing is an enabling technology and not an additional cost

In power industry, an example of establishment that utilizes rotating machinery in its operation, its various component (turbine, steam generator, steam condenser) including the auxiliaries (pumps, heaters) are generally mounted or supported on supported on foundation structures that are flexible (such as steel frame structure) , instead of connecting directly to the soil structure. Hence, during the operational period and over the operational speed range they are found experiencing certain displacements (vertical, horizontal or combination of them) as a result of the variation in the loads transmitted to the to the supporting structures/ foundations through the fasteners used to connect them to the supporting structures which causes instantaneous variation in the length s of the fasteners. However in order to reduce the vibration caused by these unwanted displacements between the rotating machinery and the foundations, vibration absorbers or vibration isolators are introduced as the interfaces between the base of the machines and the steel-framed structures. While vibration absorber involved the use of external source of power to achieve the vibration cushioning effect (additional costs is required) the latter (vibration isolator) does not require this and as such the use of vibration isolator to achieve this effect would be addressed in this study.

1.1 Considerations on Machine foundations

Different types of foundations are used for different machines depending on their capacity, geometrical sizes and constructional features. According to Teng (1992); Timlinson, 2001; Lee and Golod (2004), the type, configuration, and installation of a foundation or support structure for dynamic machinery may depend on the following factors:

(i) Site conditions such as soil characteristics, topography, seismicity, climate, and other effects;

(ii) Machine base configuration such as frame size, cylinder supports, pulsation bottles, drive mechanisms, and exhaust ducts;

Process requirements such as elevation requirements with respect to connected process equipment and hold-down requirements for piping.

1.2 General Criteria for Design of Machine Foundations

In the course of selecting and designing a foundation to meet the needs of any machine, the following criteria should be satisfied by the machine foundation (Teng, 1992; Derek2006; Gunaratne, 2006):

(i) The foundation should be able to carry the superimposed loads without causing shear failure. The bearing capacity under dynamic loading conditions is generally considered to be less than that for static loading, the reduction factor ranging from 0.25 to 1.0;

(ii) The settlement should be within permissible limits;

(iii) The combined centre of gravity of machine and foundation should be, to the extent possible, in the same vertical line as the centre of gravity of the base line;

(iv) Resonance should be avoided; hence the natural frequency of the foundation-soil system should be far different from the operating frequency of the machine; for low-speed machines, the natural frequency should be high, and vice-versa, likewise, the operating frequency should be high, and vice-versa. The operating frequency must be either less than 0.5 times or greater than 1.5 times the resonant frequency so as to ensure adequate margin of safety.

1.3 Aim of the Research

The aim is to design and fabricate an adjustable bed for a micro-steam turbine generator plant.

1.4 Specific Objectives of the Research:

The specific objectives of this research are to:

(a) design an adjustable bed with damping system for a micro-steam turbine generator plant;

(b) fabricate the system designed in (a); and

(c) evaluate the performance of the system with/without vibration isolator

2. Literature Review

The most effective way of improving system performance is, of course, to improve the efficiency of its components (Ayodeji, 2018). For well over a century, efforts have been made to reduce intensity and causes of vibration in rotating machines as a result of its impact on the overall performance of the system concerned The study of vibration reduction in steam turbine has gained serious momentum during recent years as result of the vacuum it occupies in thermal power and allied industries One of the methods of cushioning effect of unwanted vibration at the machine base-foundation interface is by the use of vibration isolator or absorber.

Over time, the mechanical systems suffer wear and become more inefficient, gradually reducing the quality of the processes of which they integrate. According to Bently *et al*., (2002), excessive mechanical stress is associated with the rotational movement of the axes and that high torsional loads and radial loads culminate in severe conditions that can lead to eventual rotor cracking and immediate shutdown of the machines. However, in order to study the significance of foundation, including the supporting structures when included, on the performance on rotating equipment, various researches had been carried out by different scholars and numerous findings were arrived at.

Allan *et al.,* (2019) presented a method of diagnosing failure in rotary machine using machine learning technique. The researchers utilized a Support Vector machine (SVM) algorithm to study rotational imbalance in rigid shaft rotor of an induction motor using vibration analysis. From the study, the researchers were able to detect various causes of imbalance faults

Vishnu, *et al*., (2015) carried out a research to study reasons for carrying out vibration monitoring, how to describe vibration and measure it effects in machines. From the results of the scheduled analysis, they found that vibration analysis provides data that could assist concerned stakeholders to maintain machinery in good health and improve the life time of the machine.

Prakash and Puri (1981) compared the observed and computed response of a reciprocating compressor foundation subjected to excessive vibrations. The analysis of the foundation was conducted with a linear weightless spring method and the elastic half-space analogs with soil properties for the designed condition and corresponding to the observed vibration amplitudes. From the findings they found that the computed amplitudes obtained by both methods were far in excess of the permissible amplitudes detailed in the manufacturer's specifications. The computed natural frequencies were found to be within about 25 % of the observed natural frequencies in horizontal vibrations. They also found that adequate soil exploration and a realistic determination of soil constants play an important role in the design of machine foundations.

In 2014, Jayarajan and Kouzer utilized a dynamic analysis to calculate natural frequency of vibration under a loaded condition that fell within ±20% of operating frequency and critical speeds. More so, the researchers used SAP 2000 to perform free vibration and forced vibration analysis and developed finite element modelling of the framed foundation structure In the study the researchers were able to highlight dynamic analysis issues related to mathematical modelling of structure, soil and machine, since the finite element method provides an efficient tool for dynamic analysis and modelling concluded that dynamic analysis needs attention to detail, both in the interpretation of results as well as modelling of turbine foundation.

Based on detailed seismic analysis, Fleischer and Trombik (2008) used practical design approach to elaborate significance of spring mounted, table mounted and raft type foundation system on load distribution. From their findings they were able to deduce importance of load distribution over the height apart from local parameters such as soil amplifications and ground accelerations

Fang and Wang (2012) utilized a three viscous spring boundary element method to exploring and analyzing the influence of soil structure interaction on the response of the 1000 MW turbine foundation soil system under excitations from earthquakes and rotor unbalances. From the findings of the study, they reported that acceleration has minimal effect on the magnitude of the internal force and that the presence of soil does affect the displacements of the system under seismic excitations

Prakash and Vijay (2006) used analytical methods to determine the response of foundations due to vibratory loads and design of machine foundation made by idealizing the foundation soil system as a spring mass dashpot model having single or double degrees of freedom. Since most foundations for machine is treated as surface footing, and that the soil spring and damping values can be obtained by following the impedance compliance function approach and also by using the elastic half space analog. complexity to the system. From the findings, they concluded that the safety, performance and stability of machines depend largely on their design, interaction and technique of manufacturing of their supporting frame. More so, they opined that the machine foundation system should be able to withstand the action of earthquake loading up to the safety limit without collapse.

Bhatia (2008) carried out investigation on the dynamics of the machine foundation system and the consideration of earthquake effects on machines as well as on their foundations. While carrying out the investigation, he equated the vertical seismic coefficient with horizontal seismic coefficient in application to machine-foundation in order to get better performance for the systems. From the research, He found that the suitability of machine foundations depends not only on the forces they are subjected to, but also on their behaviours of machine foundation when exposed to dynamic loads, which in turn depends on the natural frequency of the foundation and speed of the machine. Hence, he concluded that vibration analysis is necessary in the design of machine foundations

However, in all the researches carrien out by these scholars/researchers on foundation or supporting structure, flexibility was not considered and this necessitates the reason for delving into this research.

Table 1 shows the stiffness values of typical isolators

Material	Deflection (mm)	Frequency (Hz)
Cork or felt	0.1 – 0.5	50 – 25
Rubber	0.1 – 10	50 – 5
Metal springs	5 – 50	5 – 1

Source: Vehicle Refinement controlling Noise and Operation in Road Vehicles, Harrison (2004)

2.1 Choice of selection of isolation type

The choice of method to reduce transfer of vibration between a system and surrounding varied depending on the applications, properties of isolators, excitation frequency and condition of the machine environment. Table X provides a guide to the isolator type and alongside with the static deflection over given values of operating speed and isolation efficiency. While isolate a system the procedure for using this table involves reading the minimum static deflection against the system disturbing (excitation) frequency and required isolation efficiency. In this study, since the estimated isolation efficiency is $R \geq 80\ \%$ at r =4, and disturbing speed of 1500 rpm, based on this, the choice of isolator that would be considered is rubber mounting

	DISTURBING FREQUENCY		ISOLATION EFFICIENCY 80%		ISOLATION EFFICIENCY 90%		ISOLATION EFFICIENCY 95%		ISOLATION EFFICIENCY 98%	
	RPM	HZ	Ground Floor	Upper Floor	Ground Floor	Upper Floor	Ground Floor	Upper Floor	Ground Floor	Upper Floor
			ISOLATOR STATIC DEFLECTION (mm)							
High Deflection Springs	200	3.3	125	—	—	—	—	—	—	—
	300	5.0	60	90	110	150	—	—	—	—
25mm Deflection Springs	500	8.3	20	35	40	50	70	90	—	—
	700	11.7	11	18	20	27	40	50	100	120
Rubber Mountings	1000	16.7	6	10	10	15	18	25	50	60
	1500	25.0	3	5	5	8	8	11	20	25
Pad Mountings	2000	33.3	2	4	4	6	6	8	11	15
	3000	50.0	0.8	1.5	1.5	3	4	5	7-	10
			NON-CRITICAL AREAS		GENERAL AREAS		CRITICAL AREAS			
			Factories, Workshops, Garages, Warehouses, Laundries, Basements		Schools, Dept. Stores, Supermarkets, Telephone Exchanges, Hotels		Multi-storey Bldgs, Offices, Hospitals — Service Areas, Churches, Schools, Restaurants		Multi-storey Bldgs, Hospitals — Ward Areas, Broadcasting Studios, Theatres, Auditoriums, Libraries	

Source: Lifetime Reliability Solution (www/lifetime-reliability.com**)**

3. Methodology

The method adopted in the fabrication of an adjustable bed for micro-steam turbine generator plant comprises:

(i) marking out of the required parts out of the 8inches U-channel materials using scriber and engineering marker;

(ii) Mounting of the turbine on the developed Bed for adjustments

(iii) Installation of instrumentations required for monitoring the stability of the turbine with reference to the developed turbine bed for the purpose of achieving the research stated objectives, experimentations, and performance evaluation, and

(iv) Selection and insertion of the vibration and of vibro-isolator, based on estimated properties,

3.1 Material

The following materials (instrumentations) were used in the course of this research:

U-channel Steel metal, Couplings, Pulley and Belt , Cutting Discs , Electrode and Bolts and Nuts.

List of digital instruments used are:

* Vibration meter, range of acceleration capacity 0.1-199.9m/s^2
* Digital tachometer, test range over 1000rpm
* Digital sound level meter, measuring capacity, 40-130dBA

3.2 Design Analysis

3.2.1 The Power transmission system

For effective power transmission between the main components of the micro-steam thermal power plant (steam generator and alternator) and engagement/disengagement of these components to facilitate their orientation with reference to their supporting steel-framed structure which serves as the bed, the two power transmission systems considered in this study to achieve this purpose are: flexible flange coupling and sets of belt and pulley system.

3.2.1.1 Coupling Design and Selection

Coupling is a mechanical device used to connect two (2) shafts together at their ends for the purpose of transmitting power. Basically, it can be classed as rigid and flexible couplings. Rigid couplings are used to draw two shafts together tightly so that no relative motion can occur between them. However, in case there is any misalignment of shafts, either axially, radially or angularly, that might occur which could affect the accuracy and performance of the entire system, flexible coupling permits this without altering the required function of the system. Thus, in case of any misalignment, the shafts adjacent to the coupling are subjected to tension, but the misalignment causes no axial or bending loads. Hence, flexible coupling is considered for this study as it permits misalignment. The misalignment in this study might be due to: difference in the eyes of the shafts of the steam generator and the alternator from their bases, imperfection during welding/ construction due to unequal contraction or expansion of the materials used for developing the adjustable bed which could cause deviation in height of the developed machine bases from the desired value.

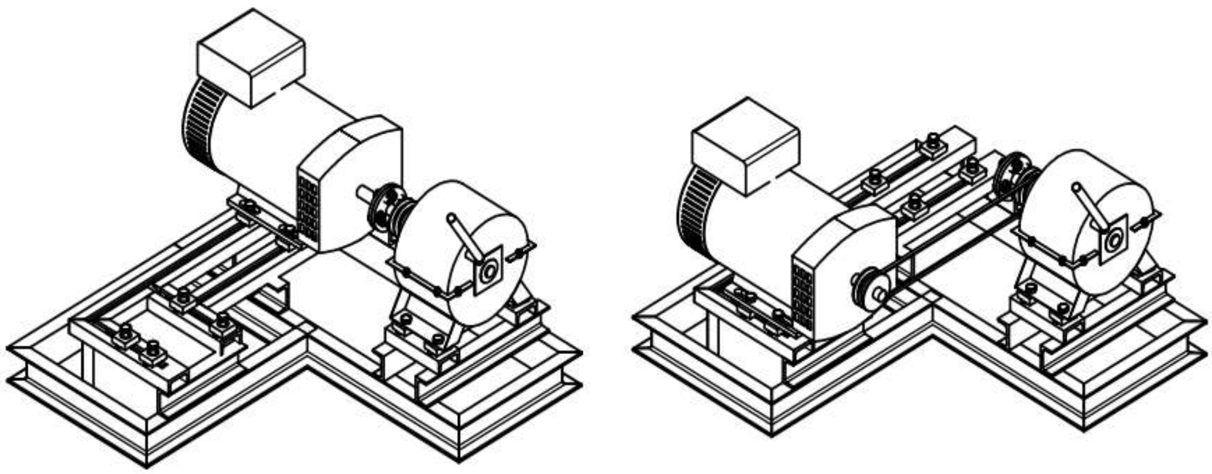
The design and selection of the coupling for this research, with reference to its components (key, hub, bolts, and flange) were done based on the under-listed assumptions in-line with the following criteria and standards ASME Code, IS: 2693-1980, IS: 2292 and 2293-194 (also reaffirmed in 1992);

Table 1 presents summary of the equations used, including the results (calculated) obtained and adopted, according to Sadhu, (2004);Klebanov *et al*., (2008); Khurmi and Gupta (2010); Ullman, (2010); Sadhu, (2010); Radzevich, (2018);

Table 1: Design summary of the equations used, including the results for coupling selection

S/N	Particulars	Equation	Calculated	adopted		Remark
1	Mean Torque (N-m)	$T_{mean} = \frac{60000P_r}{2\pi N}$	31.83.	31.83		
2	Maximum Torque (N-m)	$T_{max} = \frac{\pi}{16} d_{sh}^3 \tau_s$	502.65	502.65		
	Design of Key					
1	Length of the key (mm)	$l_k = 1.5d_{sh}$	60.00	60.00		

		Design of Hub				
1	Diameter (mm)	d_h = 1.75d_{sh} + 6	76. 00	80.00 mm		
2	Induced shear stress, τ_h (MPa)	$\frac{\pi}{16}\left(\frac{d_h^4 - d_{sh}^4}{d_{sh}}\right)\tau_h = T_{max}$	3.32	3.32		
3	Thickness (mm)	$\frac{1}{2}(d_h - d\ \ sh) = t_h$	18.00	18.00		
4	Length (mm)	$l_h = l_k$	60.00	60.00		
		Design of Bolts				
1	Number of bolt	$n_b = \frac{4d_{sh}}{150}$ +3	4.07	5		
2	Radius of bolt circle, r_{bc} (mm)	1.5d_{sh}	60.00	60.00		
3	Diameter of bolt, d_b (mm)	$n_b . \frac{\pi}{4} . d_b^2 . \tau_s = \frac{T_{max}}{r_{bc}}$	7.30	M10		As per IS:1364
		Design of Flange				
1	Thickness, t_f (mm)	t_f = 0.5t_h +6	15.00	15.00		
2	Diameter of head of socket wrench (mm)	d_w = 1.85d_b + 8	26.5	30.00		
3	Thickness of protecting flange (mm)	0.5t_f = t_{pf}	7.50	7.50		
4	Outside diameter of flange (mm)	$2\left[\frac{d_h}{2} + d_w + t_{pf} + 2c_B\right]$	82.00	90.00		



AN ADJUSTABLE BED ACCOMMODATING BOTH AXIAL AND TRANSVERSE COUPLING OF A MICRO-STEAM TURBINE PLANT

3.2.1.2 Belt and Pulley system Design and Selection

Table 2 presents summary of the equations used, including the results (calculated) obtained and adopted for selecting the size of belt, number of belts and pulley grooves and the diameters of pulley, according to Sadhu, (2004); Khurmi and Gupta (2010); Ullman, (2010); Sadhu, (2010);

Table 2: Design summary of the equations used, including the results for Belt and Pulley Selection

S/N	Particulars	Equation	Calculated		Adopted	Remark
1	Pulley Pitch diameters, Turbine/Alternator (mm)	$D_p = 1200\sqrt[3]{\frac{P}{N}}$	179.26		280.00 100.00	IS:2122 (Part I)-1973
2	Belt nominal pitch length (mm)	$L_p=\frac{\pi}{2}\left(D_{1p}+D_{2p}\right)+2C+\frac{\left(D_{1p}-D_{2p}\right)^2}{4C}$	1300.73		1331.00	IS:2494-1974
3	Actual pulleys center-center (mm)	$A+\sqrt{A^2-B}$	355.66		356	IS:2494-1974
4	Groove angle for V-Belt	2β			40^o	IS:3142-1965

5	Angle of contact (radian)	$\theta = 2.Cos^{-1}\left[\frac{D_{1p}-D_{2p}}{2C}\right]$	2.63		2.63	
6	Centrifugal Tension (N)	$F_c = \rho_b.V_b.A_b$	80.28		80.28	IS:9515-1980
7	Number of Belt, n_b	$n_b(F_i - F_c)\left[\frac{e^{\mu\theta}-1}{e^{\mu\theta}}\right] = \frac{10^3 P.k_a}{V_b}$	2.46		3	IS:9517-1980

Steam turbine-Alternator and Vibration Isolator Parameters

Detailing below are the properties of the steam turbine, alternator and the isolator

Mass of machine:

Turbine (m_t): 10.0 kg

Alternator (m_{al}): 15. 0 kg

Number of buckets (n_b): 15

Total mass of the buckets 1.0 kg

Steam consumption (m_{sl}): 68 kg.hr (1.133 kg/min)

Balance quality factor (Q) G2.5

Dimensions of the isolator (each) 100 mm x 50 mm x25 mm

Length of 100 mm, breadth or width 50 mm and height of 25 mm

Unbalanced rotating mass (m_u): 2.133 kg

Non-rotating mass (M-m_u) 23.9997 kg

F_e 87.71611 N

Number of mounts (n_p) 4

Stiffness of the vibration isolator 141.91 N/MM

3.2.1.2 Experiment Data and analysis

The experimental data obtained in the course of the experimentation are as detailed in table T and R below. Table T shows the data recorded with the attached instrumentations when the two main components (steam turbine and alternator) test rig were connected transversely with belt and pulley system whilst Table R shows the experimental data when they are axially connected with flexible flange coupling.

Table T: Experimental data obtained with the Test rig connected together using Belt and pulley system

	With Test rig mounted on vibration isolator					Test rig not mounted on vibration isolator				
s/n	S (mm)	v (mm/s)	a (mm/s^2)	V (V)	N (rpm)	S (mm)	v (mm/s)	a (mm/s^2)	V (V)	N (rpm)
1	2.7	63.0	1.4	20.0	752	1.2	75	0.9	10.0	656
2	1.7	47.0	1.1	5.0	549	1.0	55	0.6	8.0	599

Table R: Experimental data obtained with the Test rig connected together using Flange coupling system

	With Test rig mounted on vibration isolator					Test rig not mounted on vibration isolator				
s/n	S (mm)	v (mm/s)	a (mm/s^2)	V(V)	N (rpm)	S (mm)	vel (mm/s)	a (mm/s^2)	V(V)	N (rpm)
1	1.33	43.1	12.2	52	1000	1.175	43.4	12.37	52	1000
2	1.192	43.3	17.7	52	1000	1.122	44.1	18.2	52	1000

Displacement	Vel. (mm/s)	Mmls Acc	Voltap	8pm speed	Displacement	vel	acc	volt	Speed	
940/3/2.7	6.3	1.4	20.0	752	1.2	7.5	0.9	10v	656	
17	4.4	1.1	50	549	10	5.5	0.6			

1.263	41.8	11.1	560	950	1.211	4.5		53	1030	
1.33	43.1	12.2	5.2	1000	1.175	43.4		52	1000	
1.193	43.3	17.7	5.2	1000	1.121	43.1		52	1000	

3.2.13 RESULTS AND DISCUSSIONS

This study aimed at evaluating the contributory effect of vibration isolator on the performance of the major components of micro-steam thermal unit mounted on an adjustable steel framed bed before mounted on a concrete floor.

This section presents the results and discussions of the analysis of the experimental data obtained in the course of the experimentations, when the steam turbine and alternator were axially connected with a flange coupling or transversely connected with sets of belts and pulley system and when they were supported at their bases with vibration isolator before mounting on the developed adjustable steel framed bed and when mounted directly on the steel frame without the insertion of vibration isolator, in succession, respectively.

. For the purpose of evaluating the effect of vibration isolator on the performance of the steam turbine-alternator unit, the parameters utilized for comparing the performance are: the ratios of the damped frequency to the undamped frequency of the test rig when with/without vibration isolators and vibration isolator reduction efficiency .

Table Q presents results of the Performance of the test rig when mounted on the developed adjustable steel-framed bed with/without insertion of vibration isolator with respect to the ratio of the damped frequency to the undamped frequency

And vibration reduction efficiency

Table Q presents summary of the calculated results with reference to the experimental data obtained

s/n	Particular	Equation	Calculated values (Belt and Pulley system)		Calculated values (coupling system)	
			With isolator	Without isolator	With isolator	Without isolator
1	Ratio of consecutive amplitude	$\frac{x_1}{x_{n+1}}$	1.588235294	1.20	1.114836547	1.048171276
	Logarithmic decrement	$\log_e\left[\frac{x_1}{x_{n+1}}\right] = a$	4.895102885	3.320116923	3.049069757	2.852430037
	Operating frequency (rad/s)	$\omega = \frac{2\pi N}{60}$	50π	50π	50π	50π
	Natural frequency (rad/s)	$\omega_n = \frac{\omega}{fr}$	25π	$12{,}5\pi$	$12{,}5\pi$	$12{,}5\pi$
	Damping frequency ()rad/s)	$\omega_d = \sqrt{\omega_n{}^2 - a^2}$	78.38711712	39.12930502	39.15135836	39.16617585
	Isolator resistance to vibration (Ns/m)	$c = 2.a.m_m$	244.7551443	166.0058462	152.4534879	142.6215019
	Damped-undamped frequency ratio	$\frac{\omega_d}{\omega_n} = f_{d/u}$	0.4990278866	0.9964195702	0.9969811538	0.9973584782
	Periodic time (second)	$t_p = \frac{2\pi}{\omega_d}$	0.1612577416	0.1605749273	0.1604844779	0.1604237629
	Static deflection (mm)	$\frac{m.g}{k_{is}} = \delta_{st}$	1.728208019		1.728208019	

	Transmitted force Reduction (%)	$\left\{\frac{\left[\left(\frac{2\pi N_o}{60}\right)^2 \frac{m}{k_{is}}\right] - 2}{\left[\left(\frac{2\pi N_o}{60}\right)^2 \frac{m}{k_{is}}\right] - 1}\right\} = R$	99.90838315 99.82796470		99.94821063	
	Force transmitivity (%)	$1 - R = T_f$				
	Transmitivity coefficient					

Table Q presents a details of comparison of the observed experimental when the combined steam-turbine-alternator was mounted on the developed adjustable bed with vibration isolator inserted and when not inserted between them and the bed.

From the Table it was observed that the resistance offered by the isolator when used to cushion, the effect of vibration of the machine against the supporting adjustable bed, is more than when the machine is directly mounted on the supporting bed. The resistance offered with the use of sets of belt and pulley system as a power transmission is greater than when a flange coupling is used; the reason is that with the use of belt and pulley, both torsional and bending moments are experienced by the machine as the steam turbine shaft and alternator shaft are transversely positioned to each other, but when a flexible flange coupling is used as the power transmission system, only the two components of the micro power plant is subjected to only torsional moment.

The results also showed that the use of vibration isolator reduction the quantity of the force transmitted to the supporting bed is reduced, with the maximum reduction achieved when the unit was axially connected with coupling than when transversely connected with sets of belt and pulley system.

However, maximum power was generated with the use of flange coupling than when the machine was connected with belt and pulley system; this could be attributed to the fact that: with the use of coupling the two components (steam turbine and alternator) worked same speed ratio of unity (1.0) or speed of 1500 rpm, hence little or no opposing force was offered by the alternator to the motion of the steam turbine, but with the use of belt and pulley system the speed ratio of turbine to alternator is 1 : 3 (or speed of 1500 rpm to 4500 rpm) a value outside the range of e design property of the manufacturer design and working capacity of the alternator hence the opposing force offered by the alternator was higher than with coupling and this accounted for reduction in the power generated when belt and pulley was utilized as the power transmission system.

5 CONCLUSION

The primary objective of this study was the performance assessment of micro-steam turbine-alternator mounted on developed adjustable steel-framed bed when their shafts were either axially connected together with flexible flange coupling or transversely with sets of belt and pulley power transmission. Also, the evaluation was done, in succession, when these components were isolated from the bed by the insertion of vibration isolator between their bases and developed bed and when they were mounted directly on the bed without the use of vibration isolator respectively. To achieve this aim, experimental data were obtained, with the aids of the attached instrumentations, when they were connected with coupling, or belt and pulley system and isolated from the developed bed with vibration isolator, and when connected with any of these two power transmission systems in succession and mounted directly on the developed bed without the use of vibration isolator respectively. The results showed that performance of the components of the micro steam thermal unit is not only being influenced by the kinds of power transmission system used to connect them together, but also whether the vibration produced by them is isolated/cushioned or transmitted directly to their supporting structure. Based on this, the following conclusions have been made from the experimental investigation

(i) reduction in the force transmitted to the supporting bed was when the two components isolated from their supporting structure with the insertion of vibration isolator;

(ii) better performance was achieved when the components were axially connected with coupling than transversely with sets of belt and pulley system. Maximum voltage of 52V and speed of 1000 rpm at 77 dB sound level attained with coupling connection, and voltage of 20 V and speed of 752 rpm at 75 dB with belt and pulley connection

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