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Development of an Adjustable Bed for a Micro-Turbine Electric Power Generator Plant

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Abstract

An adjustable bed is a supporting structure designed to provide a seating base or platform, for a 5.0 KW micro steam turbine generator plant prior to its mounting on a block-type concrete foundation. The design of the bed frame and flanges was carried out by considering the predetermined weights of the turbine and generator (alternator) [1]. On this basis, steel materials of U-channels and angle irons were used in the fabrication of the generator bed. The bed was designed to be adjustable by accommodating direct coupling of the turbine with the generator, and the belt drive. Fabrication was carried out by welding, machining, and assembly. During assembly, the bed was made to accommodate damping materials in order to reduce the vibration of the plant [2]. 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 a 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 a vibration isolator, maximum reduction of 99.95% achieved when axially coupled and 99.91 % when transversely connected with belt and pulley system [3].

Keywords

Coupling, Dynamic Load, Forces Transmission, Pulley, Static Load, Steel Frame Structure, Vibration Isolator

1. Introduction

Machines are the growth engines of the economy. Each sector of the economy fulfils its demand by the use of machines. Machines play a vital role in the economy [4]. The use of machines in the past few decades has been increasing rapidly and it is not only in industrial/rural areas but in small isolated/rural areas also [5].

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, and dynamic and static loads, they transmitted to their adjoining surroundings, they are often mounted on supporting structures, foundations, or combinations of them to achieve adequate or appropriate safe operation and stability [6].

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, a 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 [7]. 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 measures must be taken to minimize the motion transmitted from the ground to the system [8].

According to Srinivasulu and Vaidyanath (1990); Rajput and Dubey (2012), and 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 special consideration because they transmit dynamic loads to both the supporting structures (frames) and the soil in addition to static loads due to the weight of these supporting structures and the machine and accessories [9]. 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) [10].

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 [11]. The forces are usually periodic and therefore produce periodic displacements observed as vibration (Alphouse, 2015) [12]. Hence, stability of the machine with reference to its supporting structure, in terms of balancing is essential. The specific objectives of this research are to design an adjustable bed with a damping system for a micro-steam turbine generator plant and fabricate the system designed [13].

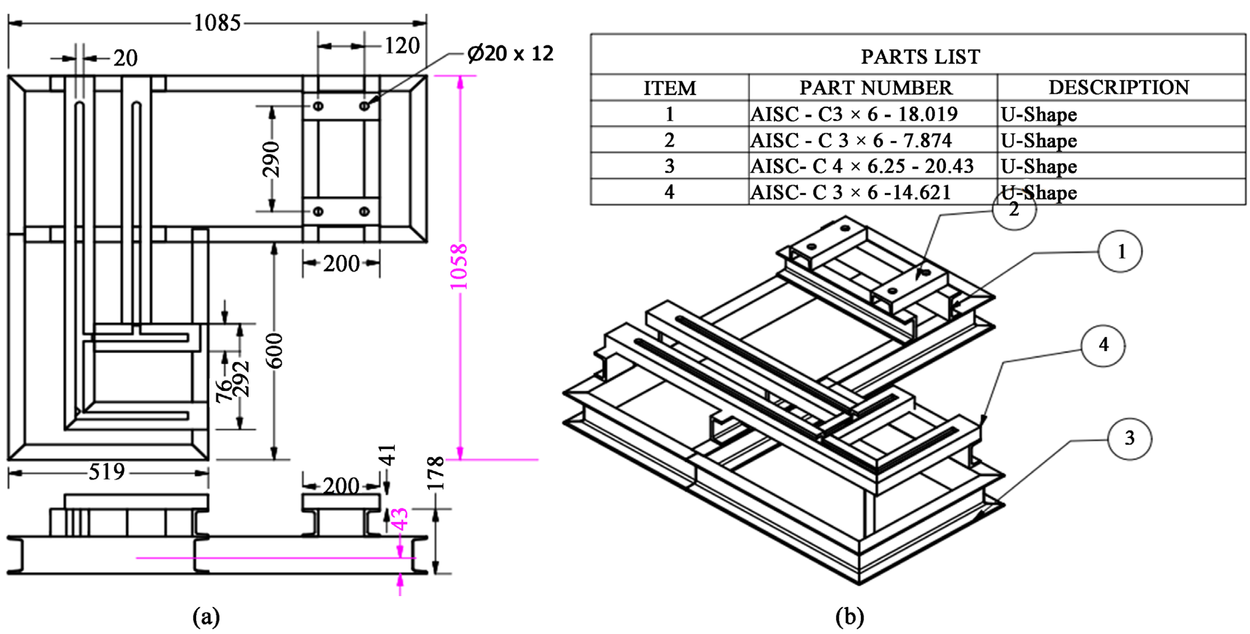
According to Derek (2006), Balancing is one of the enabling technologies that allow the development and production of more powerful and efficient equipment. Balancing is a cost-saving, efficiency-boosting process. Balancing is a way to reduce vibration and bearing loads to improve performance and reliability

[14]. 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. [15].

The scope of the research would be limited to the development of an adjustable steel-frame structure, as a supporting structure for mounting a 5.0 kW-h micro-steam turbine-alternator, prior to its mounting on a block-type concrete foundation. The steel-frame supporting structure was made adjustable to facilitate adjustment of the major components of the micro-steam thermal unit in cases when they are axially coupled together with flexible flange coupling or transversely connected together with sets of belts and pulley [16]. The vibration investigation and analysis would be of a single degree of freedom type (1 dof) as it would be limited to the isolation of the vibration transmitted by the machine (combined steam generator and alternator) to the developed adjustable steel-framed supporting bed only (vibration transmitted to the concrete box-type foundation would not be considered). Due to the fact reduction of the force transmitted to the supporting structure can be monitored/investigated at varied damping factors and frequency ratios. The selection of the vibration isolator (pads) to cushion the vibration effect would be done based on the values of the damping factor and frequency ratio that gives the least force transmissibility factor [17].

2. Materials and Methods

2.1. Design Concepts



PARTS LIST		
ITEM	PART NUMBER	DESCRIPTION
1	AISC - C3 × 6 - 18.019	U-Shape
2	AISC - C 3 × 6 - 7.874	U-Shape
3	AISC- C 4 × 6.25 - 20.43	U-Shape
4	AISC- C 3 × 6 -14.621	U-Shape

Figure 1. (a) Orthographic drawing of the Bed; (b) Isometric drawing of the Bed.

Table 1. Presents summary of the calculated design results with reference to the experimental data obtained on amplitude, operating frequency, natural frequency, damping frequency, isolator resistance to vibration, Damped-undamped frequency ratio, periodic time, static deflection, and transmitted force reduction.

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)	$\omega_n=\frac{\omega}{f_r}$	25π	12,5π	12,5π	12,5π
	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)	$c=2\cdot a\cdot m_m$	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)	$\frac{mg}{k_{is}}=\delta_{st}$	1.728208019		1.728208019	
	Transmitted force Reduction (%)	$\left\{\frac{\left(\frac{2\pi N_o}{60}\right)^2\frac{m}{k_{is}}-2}{\left(\frac{2\pi N_o}{60}\right)^2\frac{m}{k_{is}}-1}\right\}=R$	99.90838315 99.82796470		99.94821063	

2.1.1. The Power Transmission System

Figure 2 shows variances between the coupling system with and without insulators while Figure 3 shows variances between the belt and pulley system when with and without insulators.

We considered flexible flange coupling and sets of belt and pulley system for effective and efficient 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 as shown in Figure 5 and Figure 6.

2.1.2. Coupling Design and Selection

Coupling is a mechanical device used to connect two shafts together at their ends for the purpose of transmitting power, see Figure 5. Basically, it can be

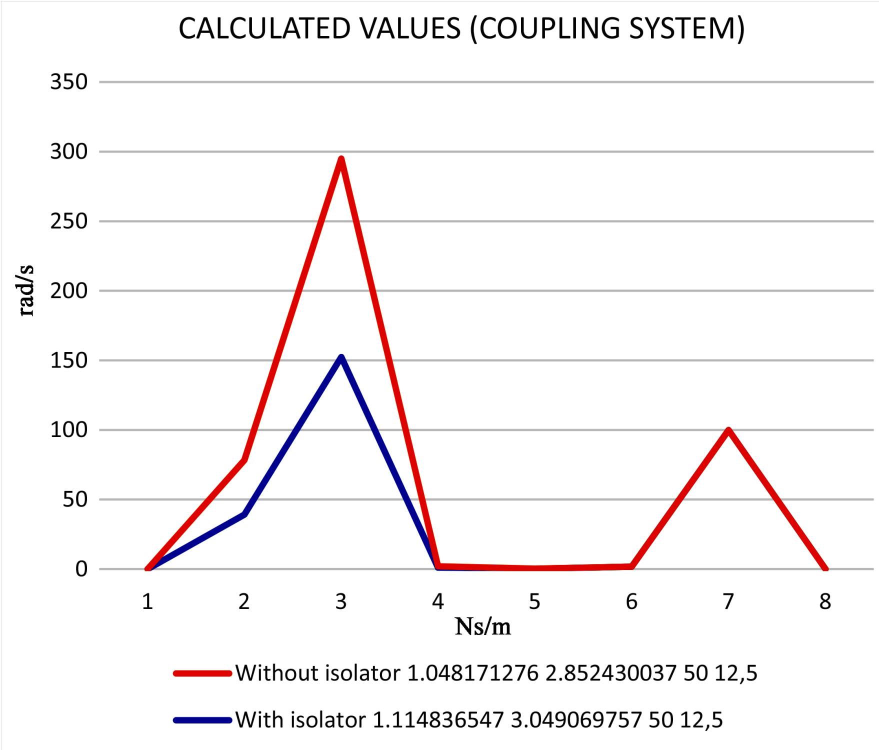


Figure 2. Shows variance between the calculated values of coupling system with vibration isolator and without vibration isolator.

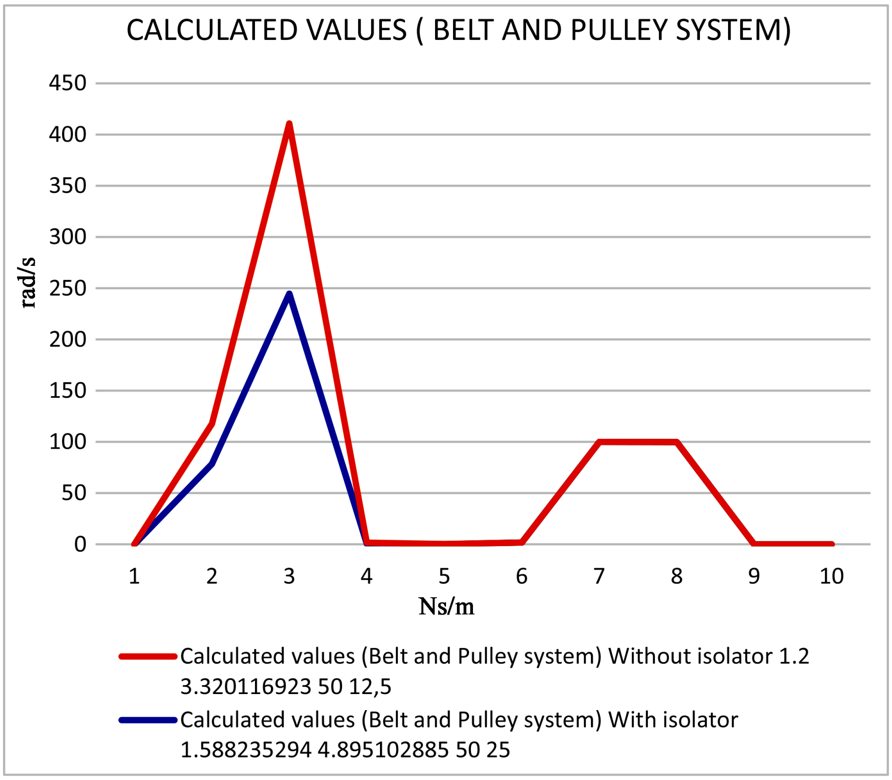


Figure 3. Shows variance between the calculated values of belt and pulley system with vibration isolator and without vibration isolator.

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°	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)	$F_c = \rho_b \cdot V_b \cdot A_b$	80.28	80.28	IS:9515-1980
7	Number of Belt, n_b	$n_b\left(F_i - F_c\right)\left[\frac{e^{\mu\theta} - 1}{e^{\mu\theta}}\right] = \frac{10^3 P \cdot k_a}{V_b}$	2.46	3	IS:9517-1980

Table 3. 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)	$\frac{1}{2}\left(d_h - d_{sh}\right) = t_h$	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)	$n_b \cdot \frac{\pi}{4} \cdot d_b^2 \cdot \tau_s = \frac{T_{max}}{r_{bc}}$	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	

Continued

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

Design Nomenclatures: ω: Operating frequencyy (rad/s); ω_n: Natural frequency (rad/s); D_p: Pulley pitch diameter (mm); C: Center distance of belt (mm); T_f: Force transmissibility (%); l_k: length of the key (mm); L_p: belt nominal pitch belt; β: groove angle for V belt; θ: Angle of contact (radian).

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 [18].

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).

2.1.3. Material Selection

The selection of materials for parts of the adjustable bed depends on the strength, tension/compression the component can withstand, force transmissibility and deflection so that the total weight will be minimal. In selecting materials, high consideration was given to moderate weight materials that can withstand stress and strain of that particular component compared to heavy or bulky materials, see **Table 4**. After thorough analysis, the materials chosen with estimated cost for the components and their proposed fabrication methods is in **Table 4**.

2.1.4. Design and Construction of the Bed

Design of bed frame and flanges was carried out by considering the predetermined weights of turbine and generator (alternator). On this basis steel materials of AISC—C 3 * 6—18.0, AISC—C 3 * 6—7.874, AISC—C 4 * 6.25—20.43, AISC—C 3 * 6—14.621 were used in the fabrication of the generator bed. The bed was designed to be adjustable by accommodating direct coupling of the turbine with generator, and the belt drive. Fabrication was carried out by welding, machining

Table 4. Materials and its cost (in Nigeria currency, Naira).

COMPONENTS	MATERIAL USED	FABRICATION METHOD	COST (#)
Frame	Mild Steel	Welding	22,500
Coupling	Cast-Iron	Casting	18,000
Pulley	Cast-Iron	Casting	12,000
Electrode	Mild steel		2500
Bolt & Nut	Steel	Fastening	1500
Belt			3000
Insulator	Rubber	Fastening	5000
			64,500

and assembly. During assembly the bed was made to accommodate damping materials in order to reduce the vibration of the plant and for this see **Figure 1** as it shows the orthographic and isometric design drawing of the bed.

Vibration meter (Fluke 805) was used to check the vibration level of the micro-steam turbine generator plant when mounted on the adjustable bed with, and without damping materials incorporation. This was carried out on the basis of direct coupling and by belt drive. The two categories of vibration data obtained (with or without a damping material) were tested using statistical paired T-test to determine the significant difference [18].

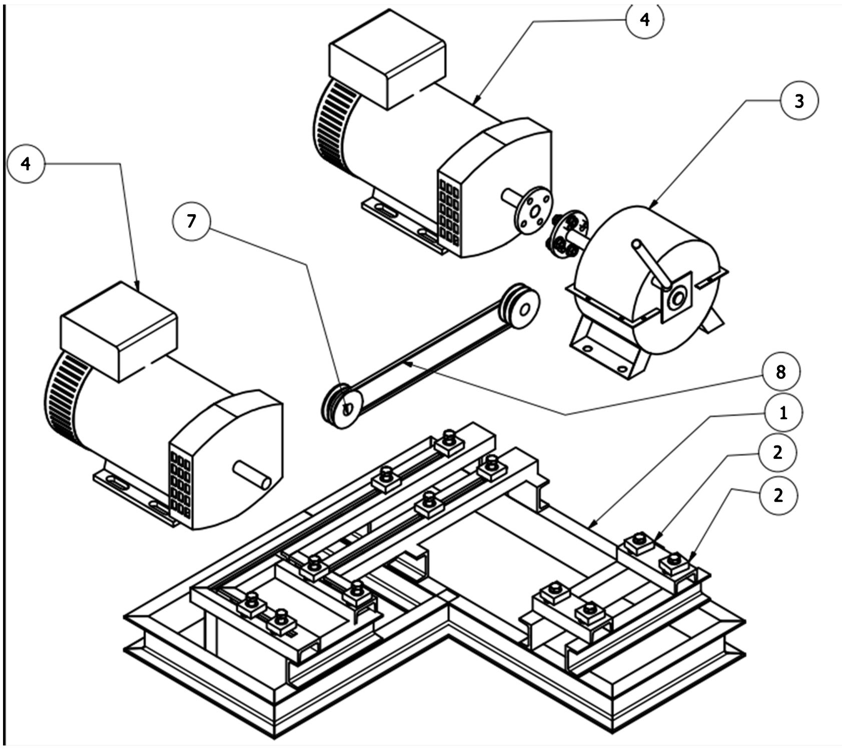
3. Results and Discussion

This study presents the 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 [18].

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 [19]. **Figure 4** shows the exploded view of the adjustable bed.

Table 1 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 [19].

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



PARTS LIST		
ITEM	QTY	PART NUMBER
1	1	frame
2	12	damper
3	1	Turbine assembly
4	2	alternator assembly
5	1	flange
6	8	ANSI B18.2.4.4M - M8 x 1.25
7	1	pulley
8	1	V-Belt

Figure 4. Isometric drawing of the Adjustable bed dimensions: Length of 100 mm, breadth or width 50 mm and height of 25 mm.

Table 5. Comparison between single flat bed and adjustable bed.

S/N	SINGLE FLAT BED	ADJUSTABLE BED
1	Only one direction of assembly is allowed	Both axial and transverse assembly can be allowed
2	Cost of fabrication is lower	Cost of fabrication is low.

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 a vibration isolator reduced 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 belts and pulley system. Table 5 shows the T-paired comparison advantage between the single flat bed that can only give room for one assembly form and the new adjustable bed that can accommodate both the axial and transverse assembly of the alternator and the turbine.

However, maximum power was generated with the use of flange coupling than when the machine was connected with a belt and pulley system; this is attributed to the fact that: with the use of coupling the two components (steam turbine and alternator) worked at 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 the turbine to the alternator is 1:3 (or speed of 1500 rpm to 4500 rpm) a value outside the range of design property of the manufacturer design and working

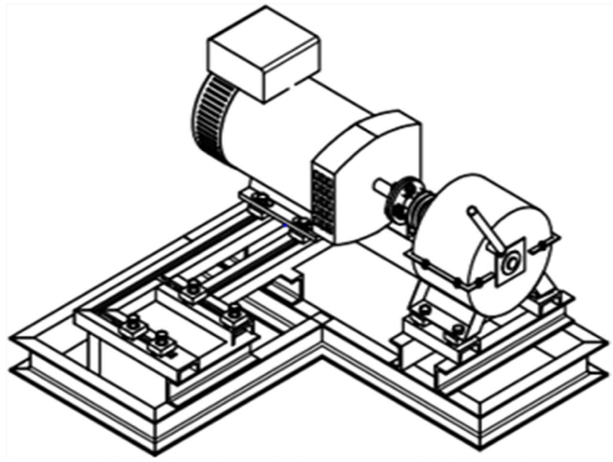


Figure 5. Axially coupled.

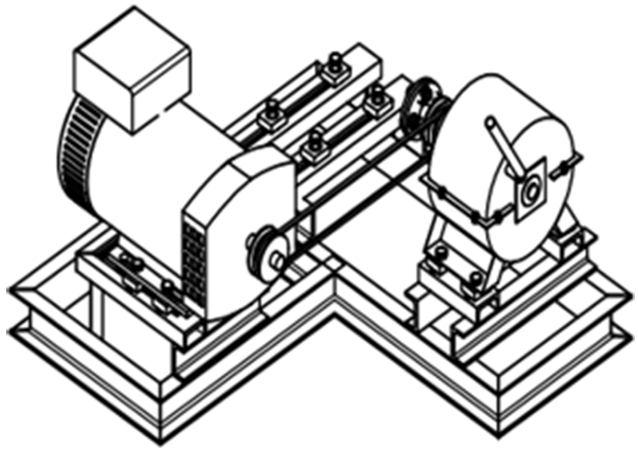


Figure 6. Transversely coupled.

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 were utilized as the power transmission system [19].

4. Conclusions

The results showed that the 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, and also by 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.

1) Reduction in the force transmitted to the supporting bed was when the two components were isolated from their supporting structure with the insertion of a vibration isolator.

2) Better performance was achieved when the components were axially connected with coupling than transversely with sets of belt and pulley system. The maximum voltage of 52 V 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.

The primary objective of this study was the performance assessment of micro-steam turbine-alternator mounted on the 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. The components of the adjustable bed (steel frame structure) can be produced using machining and welding techniques and the materials can be sourced locally (**Table 4**). The cost of production is N64,500 or $129 in US Dollars (**Table 4**). The cost of production can be greatly reduced by 5% under a mass production regime.

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Conflicts of Interest

The authors declare no conflicts of interest regarding the publication of this paper.

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