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RESEARCH ARTICLE

Advancing balancing machine technology: A cost-effective solution for laboratory and small-medium enterprises

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ABSTRACT - Small-medium enterprises (SMEs) that provide maintenance services to power plant companies, notably on rotor components, encounter a significant challenge in balancing. The main concern is the high cost associated with outsourcing balancing services. The absence of portable balancing machines requires transporting components back to the workshop, resulting in substantial downtime lasting weeks before reinstallation. This paper presents a DIY-compatible balancing machine design and its assembly process, which can be applied to various balancing processes. The design primarily emphasizes mechanical aspects, excluding measurement and instrumentation components, which can be obtained or developed based on the user's software proficiency. The actual expenditure for developing this balancing machine is approximately RM11,698.00. In the development process, benchmarking, concept scoring, functionality evaluations, dynamic analysis, and ideation for workable balancing machine concepts were all involved. The modal analysis demonstrates that the first natural frequency at 279.5 Hz (equivalent to 16,770 rpm) is significantly higher than the motor's excitation frequency, which reaches a maximum speed of around 2800 rpm (46.67 Hz). This result indicates that the dynamic behaviour of the balancing machine does not significantly affect the dynamic results of the rotor balancing process.

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1.0 INTRODUCTION

Imbalance in rotating components such as rotors, fans, turbines, and crankshafts can develop during production due to variances in material distribution and machining or after prolonged use, resulting in wear and tear, broken, etc. [1]. Therefore, balancing is a crucial process that reduces vibration, prevents structural damage, and enhances the performance and safety of rotating, requiring a balancing machine to measure and correct any imbalance. Two main elements in the balancing machines are the instrumentation for vibration measurement and the structural-mechanical design. While the former often receives considerable study and research with new algorithms proposed by researchers [2, 3], the latter tends to receive less attention. In contrast to instrumentation, which can be developed using free software, structural-mechanical design can be cost-intensive. To tackle this issue, the development of balancing machines has been dramatically enhanced. Many manufacturers now suggest that users can build them independently, providing general guidelines and emphasising the importance of incorporating soft-bearing suspension.

Balancing machines can be categorised into four types: horizontal, vertical, non-rotating, and high-speed balancing machines. Horizontal balancing machines [4, 5] are equipped with bearing pedestals, making them suitable for rotors with two bearing positions. They can handle heavier parts like rotors, crankshafts, and industrial equipment, providing high balancing accuracy. However, their main drawback is that they require a larger footprint, potentially occupying more floor space in the workshop. Vertical balancing machines [6, 7], on the other hand, are ideal for rotors without journals or those with an overhung rotor system, and they are commonly used for single correction plane balancing. The non-rotating balancing machine [8] is ideal for balancing non-rotating components, such as long shafts, spindles, and some non-rotating industrial components. Only single-plane balancing can be performed on balance machines that do not rotate. For rotors with flexible behaviour, a high-speed balancing machine [9, 10] is required. These machines are constructed using high-precision sensors and cutting-edge technology to measure the imbalance of rotating parts precisely.

There is a limited academic publication on the subject of balancing machine development. Nonetheless, several scholarly publications on subjects include balancing algorithms, balancing techniques, and measurement and instrumentation systems. With an emphasis on balancing machines' mechanical and technical development, publications are primarily found in patent collections since they focus deeper into the design and fabrication processes. The balancing machine increased in use and received a patent in the early 1900s. A machine for balancing a gyroscope's rotor was patented by Thomas et al. [11]. A unique characteristic of this design is that the machine includes two drill bits to remove metal from the rotor to achieve proper balance. Ongaro et al. [12] presented another remarkable invention of the balancing machine, in which the balancing process can be performed without the need to capture stroboscopic light or transitory

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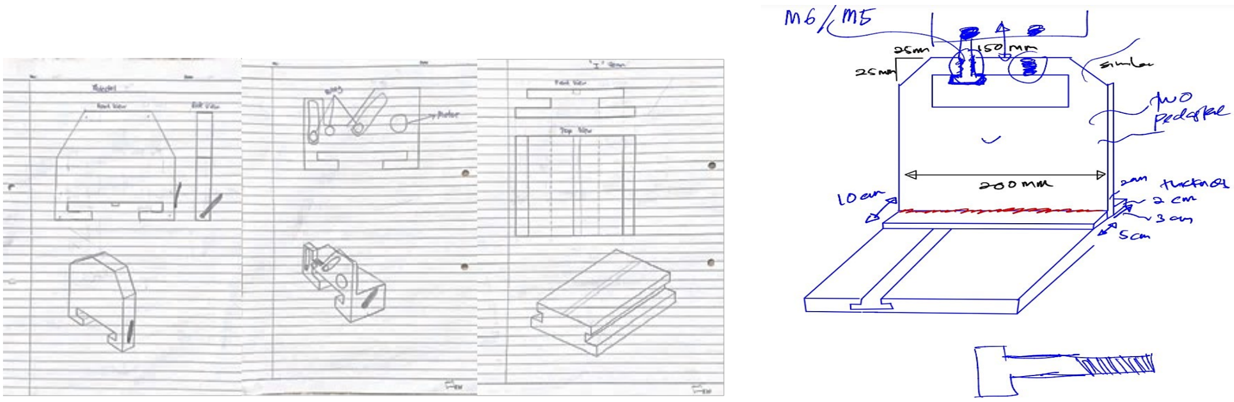
quantitative indications of any metres. The balancing machine that was patented by Nunnikhoven et al. [13] relates to the art of balancing a rotatable body, more specifically to a method and means for locating the unbalanced mass and then removing it while the body is rotating without increasing the bearing forces. This ensures that the material is removed under conditions nearly identical to those in which the rotor would be operating. Reutlinger et al. [14] are credited with solving the common issue of mass corrector performance causing reactions in the other balancing out plane of the rotating body. This issue is solved by setting up so-called centres of oscillations through the axial distribution of masses of the rotary body during oscillatory movement, which centres, in turn, were influenced by the inertia moment of the rotating body about its main axis of inertia oriented perpendicular to the direction of oscillation and to the axis of rotation. Turner et al. [15] have developed an innovative balancing machine that eliminates components that inflate size, weight, and cost while retaining the fundamental functionality of traditional balance machines. The objective is to demonstrate that a more compact, lightweight, and cost-effective machine can do the balancing task with equivalent effectiveness. This feature enhances its affordability for smaller enterprises and improves its portability by facilitating delivery via efficient package services.

In line with the concept proposed by Turner et al. [11], this paper introduces a portable and DIY-friendly horizontal balancing machine designed for easy assembly and disassembly. This innovation aims to aid SMEs in Malaysia by minimizing operational costs, facilitating high-quality balancing services for clients, and supporting research efforts. With a focus on affordability and ease of use, this innovative horizontal balancing machine aligns perfectly with Malaysia's prevalent low-speed rotor operations. The paper provides in-depth details about the design, construction, and potential benefits of this cost-effective horizontal balancing machine, as well as a dynamic analysis.

2.0 DESIGN OF THE HORIZONTAL BALANCING MACHINE

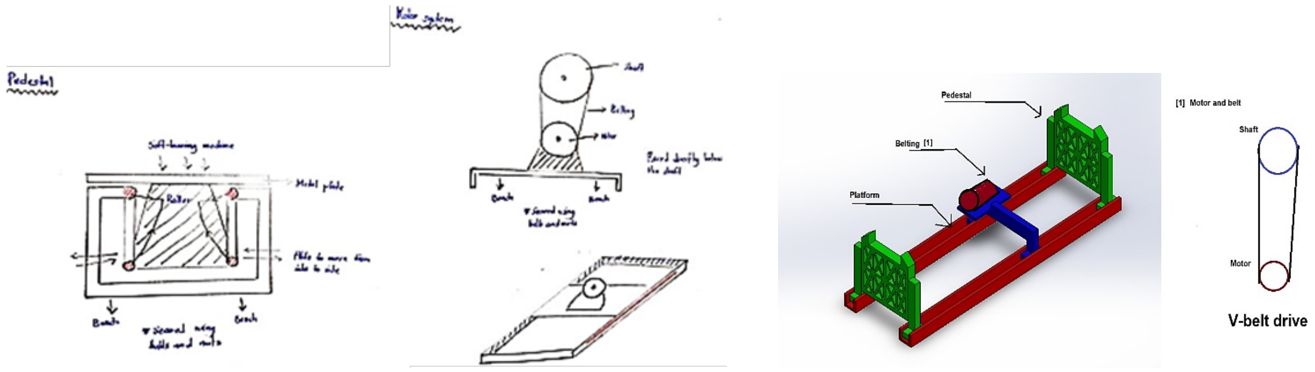
The design of the horizontal balancing machine considers features of durability, ideal size, acceptable materials, safety, cost-effectiveness, convenience of use, portability, and low-speed balancing capabilities. The complete development of the horizontal balancing machines consists of several stages, including concept design, concept scoring, finalising the final design after several improvements and modifications, providing a brief description of the design, and establishing its technical specifications. Figure 1 shows four designs considered during the initial design stage of the horizontal balancing machine. Each of these four designs is distinguished by its own distinctive features. The design process takes into account critical factors such as durability, sizing, and fabrication cost to ensure each design meets its specific requirements. It is important that the product must exhibits exceptional durability to withstand the imposed whether static or dynamic load effectively. Size is also carefully considered, aiming to achieve an optimal balance between functionality and practicality.

In the first design concept (Figure 1(a)), the pedestal is made of aluminum, which could positively impact the machine's durability and its load-bearing capacity. However, there are some concerns regarding the complexity and difficulty in constructing the motor base, along with its lack of safety features. On a more promising side, the '*T*' beam is made from solid stainless steel material, making it an excellent choice for effectively handling the load placed on both the pedestal and the motor base. For the second design (Figure 1(b)), the idea is to use a bed plate as the base. The pedestal is securely locked onto the bed plate using *T*-clamps. Additionally, soft bearings suspension are incorporated on the pedestal to facilitate shaft balancing. This concept is incredibly effective and promising, but the high cost of the bed plate that will be purchased from a bed-plate supplier is a significant downside. In the third design (Figure 1(c)), a retractable steel plate is utilized for the bench, which is secured using bolts and nuts. The base of the design consists of a stainless steel bench, commonly referred to as the *C*-beam, which is designed to support the weight of the pedestal. The pedestal is equipped with soft-bearing suspensions and offers a unique feature that allows it to move from side to side. The fourth design (Figure 1(d)) is a straightforward and minimalistic concept, with its foundation centered around a *U*-shaped slider. This design allows both the pedestal and the belting base feet to move smoothly in a forward and backward motion within the *U*-shaped slider.



(a) (b)

Figure 1. Four concept designs: (a) aluminium-based pedestal, (b) bed-plate concept



(c) (d)

Figure 1. (cont.) (c) retractable steel plat concept and (d) U-shaped slider concept

Table 1 depicts the concept scoring of these four designs. This concept scoring employs weight to differentiate each option. The weight is multiplied by two and three to emphasise the relevance of the concept, as it the main criteria in the development of the horizontal balancing machine. The design with the highest total score was chosen as the final design for the horizontal balancing machine.

Table 1. Concept scoring

Criteria	Weight	Concept Design			
		1	2	3	4
Durability	0-5 (x3)	5	15	4	5
Portabality	0-5	3	4	3	5
Material	0-5	4	5	2	4
Size	0-5 (x2)	6	9	7	6
Safety	0-5	4	5	5	4
Fabrication cost	0-5 (x3)	15	15	15	15
Total score		37	53	36	39

As indicated in Figure 2, the final design has been reached through a rigorous and challenging process of improvement and refining. The result of work incorporates concepts from three other designs, culminating in a comprehensive and ideal design. Various angles were carefully considered throughout the development of this final design. For instance, the high cost of the ready-made bed plate and *T*-clamps led the decision to modify the design. In lieu of these parts, a simpler, in-house-manufacturable bed plate design has been proposed. This bed plate will feature a numbers of evenly spaced taped holes, specifically designed to facilitate the use of steel hex socket cap screws (M4 × 35mm) for securing the pedestal, *L*-base plate, and rail together. This approach aims to achieve a balance between cost-effectiveness and functional efficiency in the design.

The finalized-assembly horizontal balancing machine, as depicted in Figure 2, consists of four main sections:

i. Soft bearing and its support: This section serves to hold the workpiece that needs to be balanced securely. The soft bearing ensures smooth rotation and precise balancing.

ii. Bed plate and railing system: The bed plate provides the stable base for the entire horizontal balancing machine. It is equipped with a railing system that allows the soft bearing to extend and retract in accordance with the length the workpiece.

iii. Motor and pulley-belting system: The motor serves for spinning the workpiece, and it is connected to a pulley-belting system to facilitate smooth and controlled rotation.

iv. Measurement and instrumentation system (future development): This section will be developed in the future and will be responsible for measuring the acceleration response of the soft bearing's horizontal motion during the rotation of the workpiece. This data will be crucial for precise balancing.

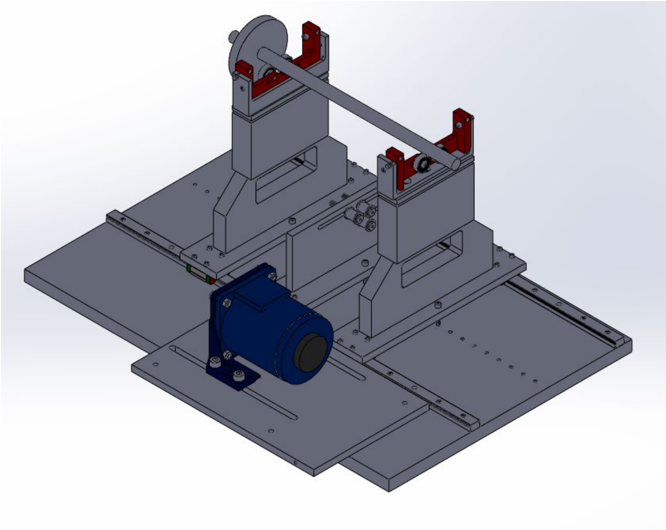
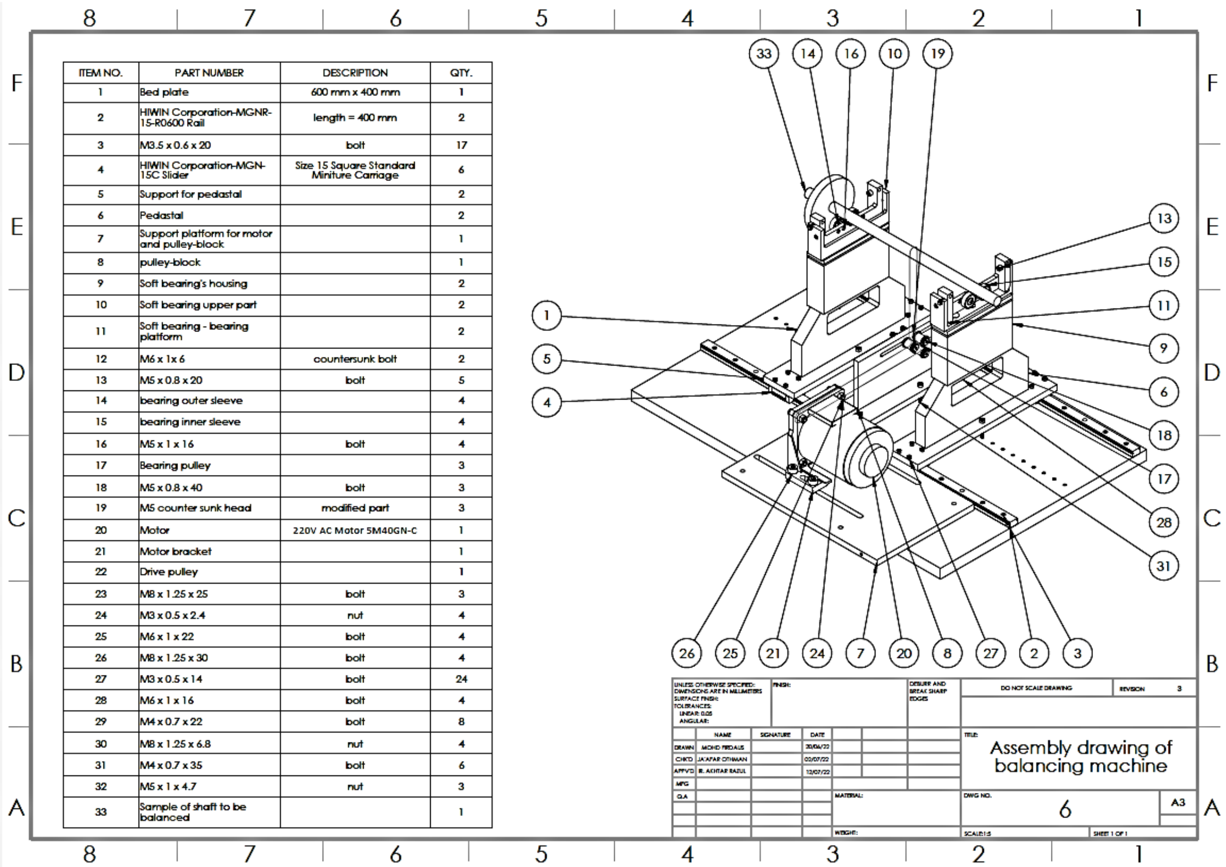


Figure 2. The finalized-assembly design of the horizontal balancing machine

2.1 Description of the Preferred Embodiment

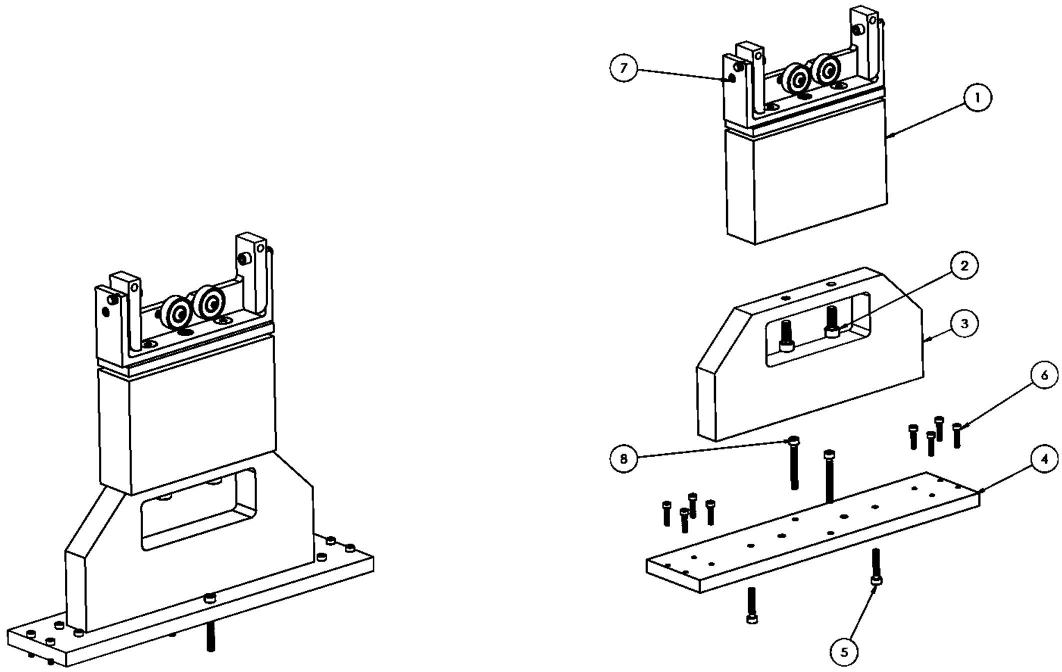
Figure 3 depicts the assembly drawing and bill of materials for the horizontal balancing machine. The bill of materials specifies all components and materials required to build the horizontal balancing machine. For better description of the preferred embodiment, the assembly drawing is divided into three major systems, as follows (It's important to note that the numbering in Figure 4 until 6 is different from the numbering in the drawing found in Figure 3). Figure 4 shows the isometric view of the soft-bearing suspension and its support system. The soft-bearing suspension (1) comprises several components, such as the bearings and their holder, and is attached to the pedestal (3) using two M6 × 1 × 16 bolts (2). The pedestal (3), in turn, is vertically mounted on the pedestal support (4) using two M4 × 0.7 × 22 bolts (5). The pedestal support (4) is fixed onto sliders (9, 11) (see Figure 5), using eight M3 × 0.5 × 14 bolts (6) (four bolts on each side of the pedestal support). To measure acceleration, an accelerometer is mounted at (7) using an adapter. This soft bearing and its support system are locked in place with two M4 × 0.7 × 35 (8) bolts onto the bed plate (13) (Figure 5) into the tapped hole (16) (Figure 5). For balancing a workpiece on the machine, two units of this soft-bearing suspensions and its support system are required on both sides of the bed plate system (see Figure 5).



ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	Bed plate	600 mm x 400 mm	1
2	HIWIN Corporation-MGNR-15-R0600 Rail	length = 400 mm	2
3	M3.5 x 0.6 x 20	bolt	17
4	HIWIN Corporation-MGN-15C Slider	Size 15 Square Standard Miniture Carriage	6
5	Support for pedastal		2
6	Pedastal		2
7	Support platform for motor and pulley-block		1
8	pulley-block		1
9	Soft bearing's housing		2
10	Soft bearing upper part		2
11	Soft bearing - bearing platform		2
12	M6 x 1x 6	countersunk bolt	2
13	M5 x 0.8 x 20	bolt	5
14	bearing outer sleeve		4
15	bearing inner sleeve		4
16	M5 x 1 x 16	bolt	4
17	Bearing pulley		3
18	M5 x 0.8 x 40	bolt	3
19	M5 counter sunk head	modified part	3
20	Motor	220V AC Motor 5M40GN-C	1
21	Motor bracket		1
22	Drive pulley		1
23	M8 x 1.25 x 25	bolt	3
24	M3 x 0.5 x 2.4	nut	4
25	M6 x 1 x 22	bolt	4
26	M8 x 1.25 x 30	bolt	4
27	M3 x 0.5 x 14	bolt	24
28	M6 x 1 x 16	bolt	4
29	M4 x 0.7 x 22	bolt	8
30	M8 x 1.25 x 6.8	nut	4
31	M4 x 0.7 x 35	bolt	6
32	M5 x 1 x 4.7	nut	3
33	Sample of shaft to be balanced		1

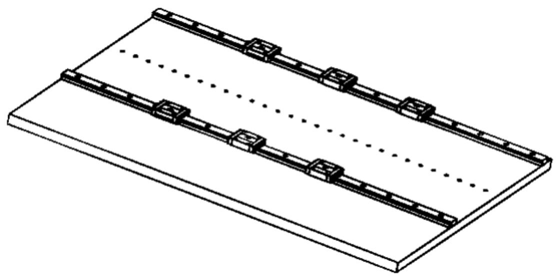
Figure 3. The assembly drawing and bill of materials of the horizontal balancing machine

Figure 5 illustrates the platform of the horizontal balancing machine, which is composed of a bed plate (13), rails (12), and sliders (9, 10, 11). The bed plate (13) is constructed of rigid mild steel, with dimensions of 600 × 400 x 18 mm. The rails (12) are used as a guide for the sliders (9, 10, 11), which can move along them depending on the length of the workpiece mounted on the soft-bearing suspension in Figure 4. The rails (12) are mounted onto the base plate (13) using 18 of M3.5 × 0.6 x 20 bolts (15), with nine bolts securing each rail (12). Four sliders (9, 11) are used to mount two pedestal supports (Figure 4), while two sliders (10) are used to mount the motor and pulley system support (Figure 6). When the positions of the systems in Figure 4 and 6 are in place and the workpiece is positioned, the systems must be locked into the tapped hole (16) to prevent any movement.

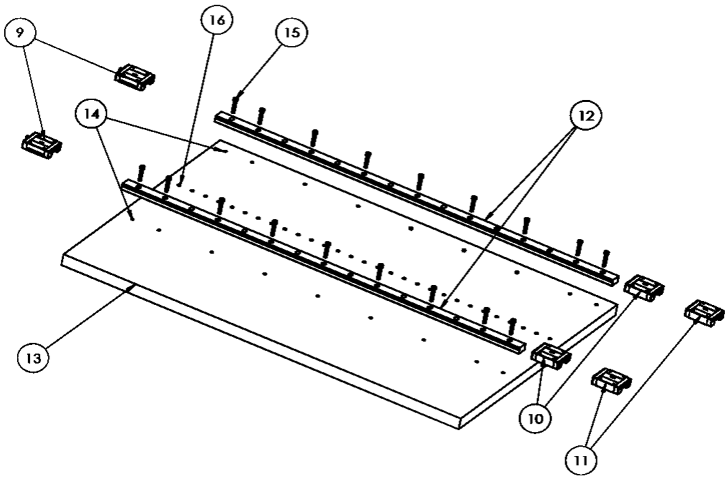


(a) (b)

Figure 4. Soft-bearing suspension and its support system: (a) isometric view, and (b) isometric-exploded view



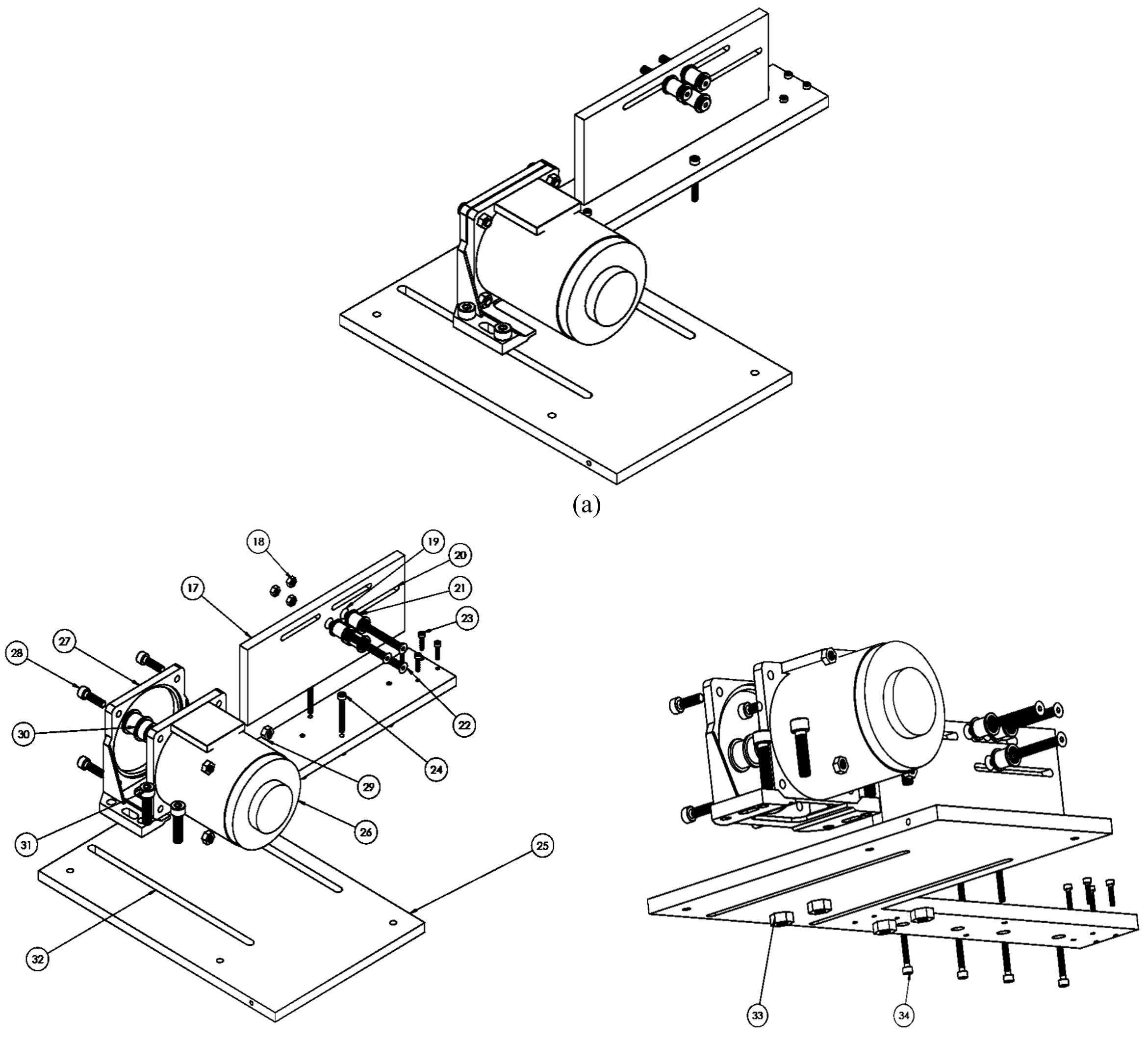
(a)



(b)

Figure 5. Bed plate, rail and slider system: (a) isometric view, and (b) isometric-exploded view

Figure 6 depicts the motor and pulley system and its platform/support. The workpiece will be driven by a motor (26) (220V AC Motor 5M40GN-C complete with speed controller) that can reach a maximum speed of 2800rpm, secured to the bracket (27) with four M6 × 1 × 22 bolts (28) and its nuts (29). This bracket (27) is mounted on the support (25) using four M8 × 1.25 × 30 bolts (31) and its nut (33) through two slots (32), which allow the motor to move freely to suit the pulley system. The motor's shaft uses a fabricated pulley (30) before the belt is installed, connecting it to the ready-made pulley system. The pulley system consists of three pulleys (21) installed on the wall (17) using three M5 × 0.8 × 40 countersunk bolts (22) and nuts (18) as well as three modified M5 × 0.8 × 40 countersunk bolts (19) (to prevent pulley to touch the wall), which can be adjusted in the available slots (20) to accommodate different workpiece diameters. The wall (17) of the pulley system is mounted on the support using four M4 × 0.7 × 22 bolts (34). The platform is mounted on sliders (10) (Figure 5) using eight M3 × 0.5 × 14 bolts (23), four on each side of the platform. Similar to the soft bearing system, the platform in Figure 3.4 is locked with two M4 × 0.7 × 35 (24) bolts onto the bed plate (13) (Figure 5) into the tapped hole (16) (Figure 5).



(a)

(b) (c)

Figure 6. Motor and pulley system and its platform: (a) isometric view, (b) isometric-exploded view, and (c) bottom-exploded view

2.2 Actual Horizontal Balancing Machine and its Technical Description

Figure 7 shows the actual horizontal balancing machine while a sample of workpiece placed on the soft bearings. The following step is procedure for setting-up the horizontal balancing machine as well as some technical aspects:

i. It is important to have all four main components in place: the soft bearing and its support, the bed plate and its railing system, the motor and its pulley-belting system, and the measurement and instrumentation system.

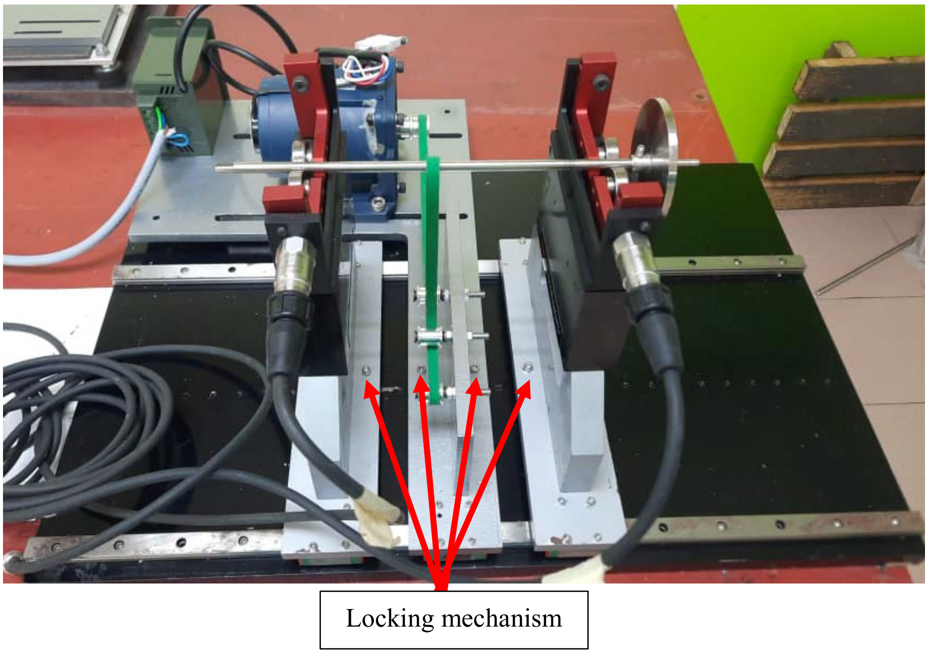
ii. Place the workpiece onto the soft bearing bearings and adjust the pedestal to fit the workpiece. As illustrated in Figure 7(b), the minimum gap between two soft bearings, indicated by the arrow, is 103 mm. This gap allows for a minimum workpiece length of approximately 170 mm.

iii. Once the workpiece is in place, the pedestal and motor supports should be locked to prevent any movement (see Figure 7 (a)).

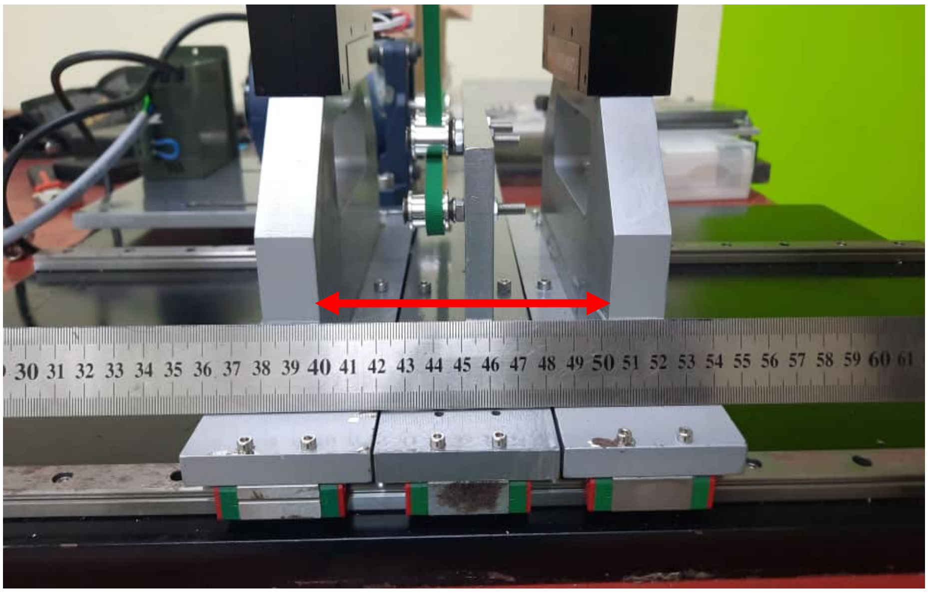
iv. Insert the belt (860 × 7 × 2 mm) and adjust its tension by moving the pulleys along the slot. Once the tension is at the optimal enough, tighten all of the pulleys securely.

v. The accelerometers and tachometer should be properly placed for measurement purpose.

vi. Once everything is in place, the motor can be turned on and controlled using a speed controller.



(a)



(b)

Figure 7. Actual horizontal balancing machine: (a) locking mechanism between pedestal and bed plate and (b) minimum distance between soft bearings

2.3 Cost Analysis of the Balancing Machine

The actual cost of constructing the balancing machine is outlined in Table 2, excluding expenses related to the instrumentation and measurement system. This cost encompasses the machining processes for various components, delivery charges, and does not account for labour pay. In the event of commercialization, where labour costs are essential, they should be factored in, along with considerations for inflation and exchange rates, especially for components bought outside Malaysia. With a total price of RM11,698.00, the pricing is reasonable for small-medium enterprises (SMEs) and commercially viable.

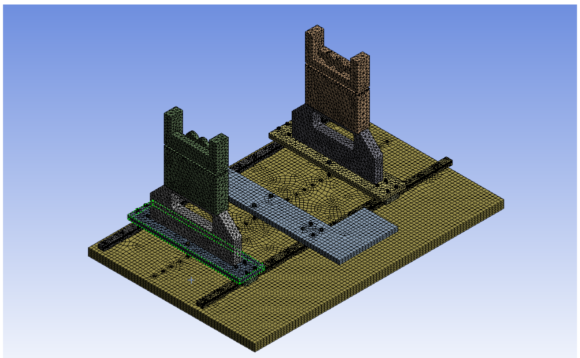
Table 2. Actual cost involved

Criteria	Price per unit (MYR)	Quantity	Total price
Soft bearing	3,949.50	2	7,899.00
Bed plate, pedastal, motor's support (including machining cost)	3,044.00	1	3,044.00
Motor	200.00	1	200.00
Pulley	10.00	3	30.00
Railing system	95.00	2	190.00
Slider block	30.00	4	120.00
Belting	65.00	1	65.00
Bolts and nuts	100.00	(bulk)	100.00
Total cost	-	-	11,698.00

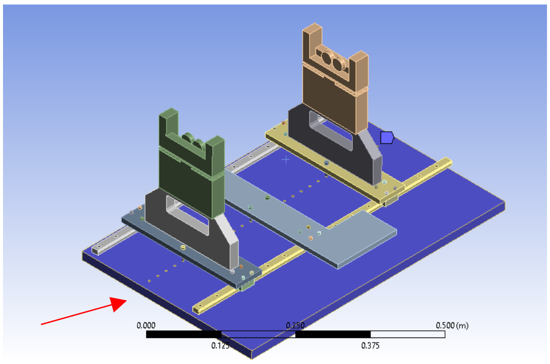
3.0 DYNAMIC ANALYSIS OF THE HORIZONTAL BALANCING MACHINE

To prevent failure resulting from external excitation, it is crucial to understand the dynamic behavior of any mechanical system. Hence, we conducted an analysis of the dynamic behavior of the horizontal balancing machine through free vibration response, specifically employing modal analysis in ANSYS Workbench software. The purpose of this analysis was to ensure that the natural frequencies of the machine do not coincide with the external excitation generated by the motor, which can reach up to 2800 rpm (46.67 Hz). The 3D model of the horizontal balancing machine was modeled using Solidworks. However, it had to be saved in the IGES format instead of the original SLDASM format due to compatibility issues with the FEA software ANSYS Workbench. Additionally, during the initial FEA analysis using the actual model from the assembly drawing, errors were encountered, leading to the need for simplification of the model. Specifically, certain small areas and curves, like those around bolts and nuts, posed challenges for accurate meshing. To address this problem, certain simplifications were made, such as removing the threads from the bolts. In the FEA process, each element is assigned specific material properties, such as Young's modulus, Poisson's ratio, and density. For this anlysis, three different types of materials were used, and the appropriate type was selected based on their respective properties. The materials used were mild steel ASTM A36 (Young's Modulus of 200 GPa, density of 7,800 kg/m3, Poisson's ratio of 0.32), stainless steel AISI 316L (Young's Modulus of 193 GPa, density of 8,030 kg/m3, Poisson's ratio of 0.28), and aluminium 6061-T6 (Young's Modulus of 68.9 GPa, density of 2,700 kg/m3, Poisson's ratio of 0.33).

In FEA, the geometry of the system is divided into a finite number of smaller elements, with each element having a set of nodes. The accuracy and efficiency of the analysis largely depend on the quality of the mesh. However, in this analysis, due to the involvement of numerous small parts such as bolts and nuts, the mesh generation was automatically determined by the software. The sample mesh generation for the horizontal balancing machine is illustrated in Figure 8. The operation of the horizontal balancing machine requires a rigid base or foundation to prevent vibrations from external sources from transferring and disturbing the machine's structure. As a result, the boundary condition of the horizontal balancing machine is defined by setting the base plate to be fixed and leaving the other components free (see right figure in Figure 8).



(a)



(b)

Figure 8. (a) Meshing of the horizontal balancing machine and (b) Fix boundary condition on the base plate (shown by the red arrow)

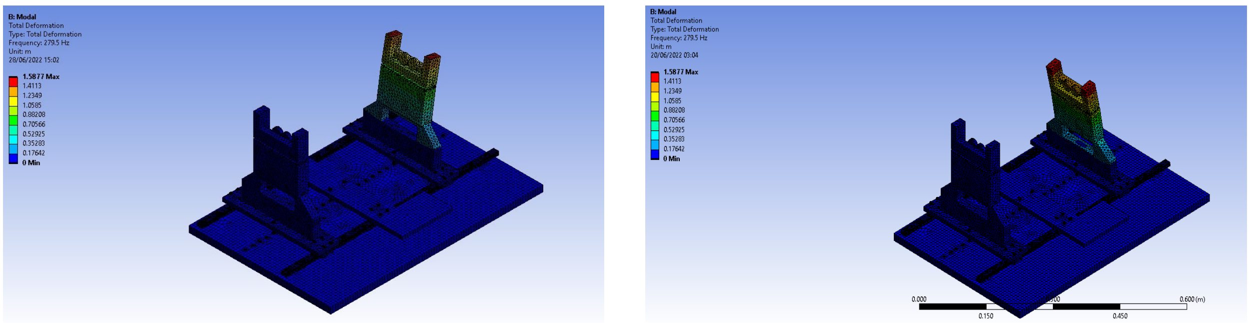
4.0 RESULTS AND DISCUSSION

4.1 Dynamic Behaviour of the Horizontal Balancing Machine

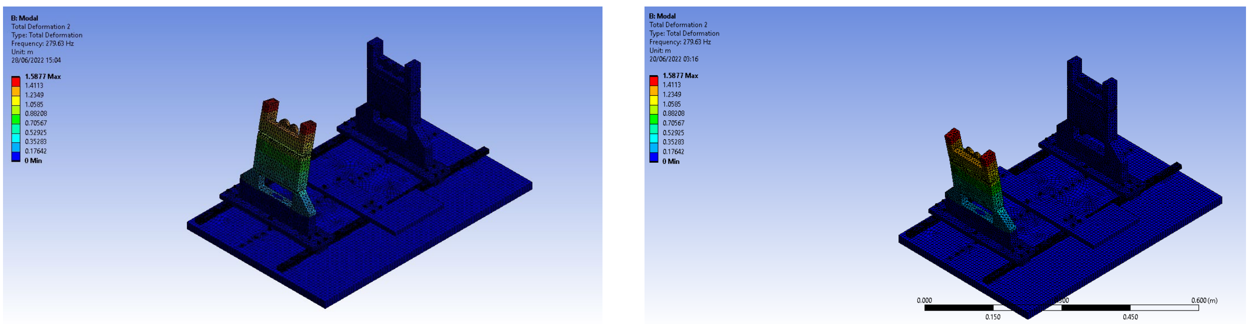
Figure 9 shows the natural frequencies and corresponding mode shapes of the horizontal balancing machine obtained from modal analysis. The natural frequencies are measured at 279.5, 279.63, 428.61, 773.63, and 773.85 Hz. Notably, the first natural frequency at 279.5 Hz (equivalent to 16,770 rpm) is considerably higher than the excitation frequency generated by the motor, which reaches a maximum speed of around 2800 rpm (46.67 Hz). These results indicate that the excitation frequency generated by the motor will not significantly impact the dynamics of the horizontal balancing machine, and thus, will not interfere with the results of the balancing process. It is also observed that because the horizontal balancing machine is almost symmetrical, the occurrence of natural frequencies in pairs is caused by this intrinsic symmetry. Each pair comprises two natural frequencies with identical values but with opposite mode shapes. For example, the first natural frequency at 279.5 Hz corresponds to one soft-bearing suspension experiencing a back-and-forth motion, while the second natural frequency at 279.63 Hz corresponds to the other soft-bearing suspension displaying the same motion. This pattern also holds true for the fourth and fifth natural frequencies. One might argue that there is a slight difference between 279.5 and 279.63 Hz. However, this variation can be attributed to factors such as imperfections, material variations, and boundary conditions, which are inherent in real-world structures. Despite these small differences, the fundamental principle of natural frequencies occurring in pairs with similar values and opposite mode shapes still holds true.

In composite structures, including the one in this case where the horizontal balancing machine is made of mild steel, stainless steel, and aluminum, the mode shape tends to be more dominant in parts/components with lower density. Lower density often leads to lower stiffness, which, in turn, can result in higher deflections under vibration (although there may be exceptions to this general trend). As the soft-bearing suspension in this structure is made of aluminum, it experiences large deflection in all four modes (mode 1, 2, 4, and 5). This is due to the fact that parts with lower density are generally more flexible and prone to experiencing larger deflections during vibration, resulting in a more pronounced mode shape in those parts. The mode shape reflects the characteristic vibration pattern of the structure at specific natural frequencies. However, there is no significant cause for concern because the natural frequencies of these four modes are far away from the excitation frequency of the motor. Hence, the vibrations induced by the motor's excitation will not have a notable impact on the soft bearing component.

The mode shape (3rd natural frequency) also exhibits large displacement at the base where the motor is placed. This raises safety concerns since the motor functions to rotate the workpieces using a belting and pulley mechanism. However, this issue is not critical as the natural frequency at the 3rd mode shape is significantly high, at 428.61 Hz. This indicates that the motor would need to achieve speeds of up to 25,700 rpm, which is practically impossible for a small motor with high speed and average torque. Therefore, it is not feasible for such a motor to spin the workpieces at that speed. Moreover, it iss important to note that this horizontal balancing machine is designed specifically for balancing rotors used in low-speed applications.

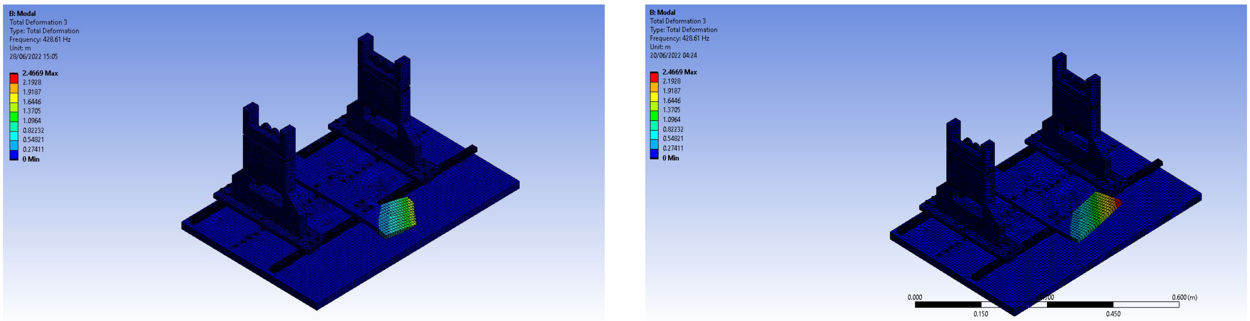


(a)

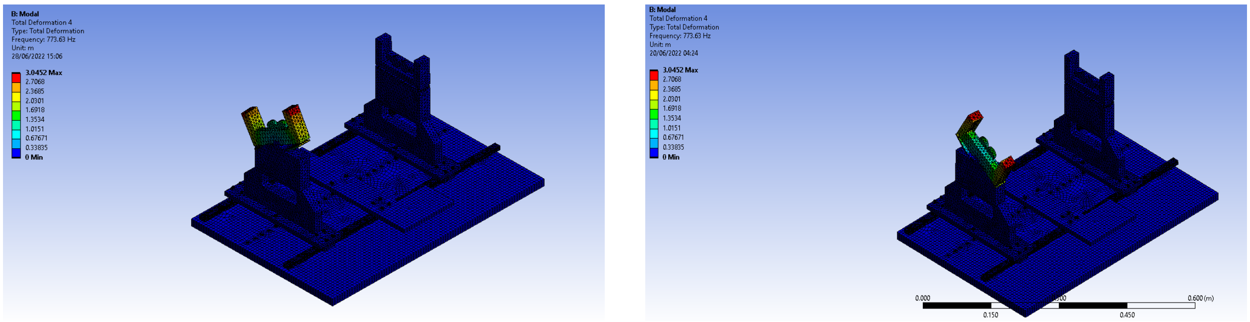


(b)

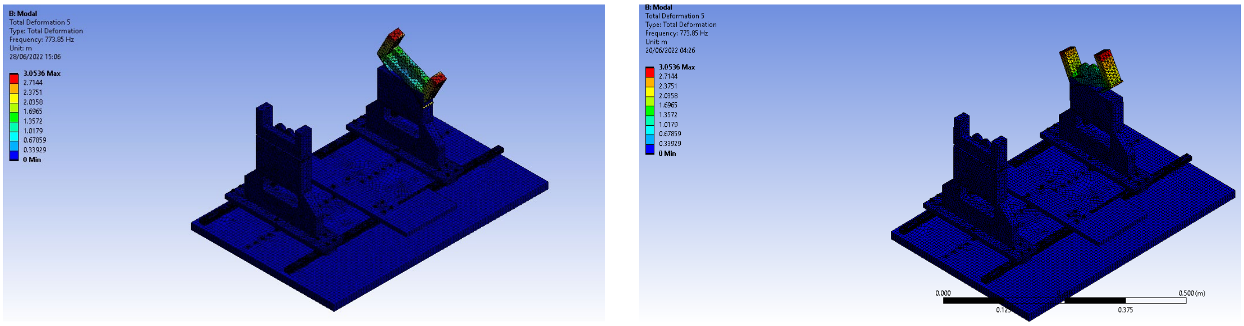
Figure 9. Natural frequencies and mode shapes of the horizontal balancing machine: (a) first mode, (b) second mode



(c)



(d)



(e)

Figure 9. (cont.) (c) third mode, (d) fourth mode and (e) fifth mode

5.0 CONCLUSIONS

The development of the horizontal balancing machine has been a resounding success, taking into consideration factors such as cost-effectiveness, user-friendliness, and do-it-yourself (DIY) adaptability. The primary focus during its development was to cater to the needs of small and medium-sized enterprises (SMEs) and laboratory in Malaysia. The process involved following standard design procedures, including concept design and concept scoring, to ensure an efficient and reliable horizontal balancing machine. One of the key strengths of the developed horizontal balancing machine lies in its clear technical description and comprehensive operating instructions. This clarity empowers others to replicate the horizontal balancing machine, promoting widespread adoption and utilization of this innovation. Moreover, the dynamic analysis performed on the horizontal balancing machine demonstrated exceptional results where the natural frequencies of the machine were found to be well outside the range of external excitation that may arise from the motor or other sources. As a result, the dynamic behavior of the horizontal balancing machine does not interfere with the balancing process, ensuring accurate and consistent performance.

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