IMPROVED DESIGN OF A PORTABLE BALCONY CRANE

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ABSTRACT

Moving heavy and large household loads such as furniture and fridges, in or out of apartments at the upper floors of multiple floor residential buildings is a tedious task, most especially where there is steep and narrow staircase that makes it impossible for one to follow the basic rules of lifting and carrying. A portable balcony crane designed to create an alternative to that long and stressful process, was modified in this work. The modified balcony crane is incorporated with a rotating arm that would enable loads to be directly lifted into or from the balcony, without moving the whole crane. The results of the performance testing of the crane confirmed that a maximum load of 150kg can be lifted safely, using ten 20-liter plastic buckets of water as counterweight. The crane, which was designed and fabricated with locally available standard materials, would help reduce manual lifting of heavy, large and unbalanced loads in and out of apartments in multi floor residential building. This would in turn reduce the exposure of residents to musculoskeletal diseases and trauma. The total cost of producing one unit of the balcony crane is Sixty-Four thousand, Eight Hundred and Fifty Naira (₦64,850:00) only), which is more than 62% cost reduction compared to the existing balcony crane.

Key Words: *Rotary arm crane; Household loads, Counterweights; Multi-floor buildings; Musculoskeletal diseases*

1. INTRODUCTION

The current trend in most cities all over the world is the development of multi-floor residential buildings, mainly to overcome the challenges of urban over population and ensure the optimal use of scarce land resources (Aliyu et al., 2015; Streen, 2002). Staircase and elevators are required for vertical movement in such buildings, though the use of elevators is recommended for buildings having more than three floors (Otis, 2015). However, it is obvious that apart from some hotels and corporate buildings in major Nigerian cities, most residential buildings of even more than three floors do not have elevators. Thus, the staircase is the only option through which large and heavy household items like couches, freezer, fridge, mattresses, wooden cabinets, etc., can be moved in or out of the apartments on the upper floors of such building. Moving these large and heavy loads through the staircase is always stressful, especially in buildings with narrow and steep staircase. This is because, traversing the stairs while carrying large and heavy loads makes it impossible for one to follow the basic rules of lifting and carrying. Improper load lifting and carrying can induce musculoskeletal disorders (MSDs), which are injuries in the body's joints, ligaments, muscles, nerves, tendons, and structures that support limbs, neck and the spine, due to uneven loading of the muscles, joints and nerves. MSDs are degenerative diseases and inflammatory conditions that cause pain and impair normal activities of an individual (Marjolein, 2016; Côté et al., 2013; Kuorinka, 1987).

Perhaps, lifting operations in multi-floor residential buildings could be much easier for an occupant if there is portable hoisting equipment in place, to lift large and heavy household load in and out of the apartment at the upper floors through the balcony. Reviewed literatures identified some unique existing models of hoisting equipment that can be used for lifting operations in multi-floor buildings. These include the "Small Indoor Rotating Crane Hoist" and Boom crane pulley. The small indoor rotating crane hoist requires a ceiling over the balcony to work, and it is mostly used in construction sites for lifting loads from the ground to the upper floors (b2bfoo, 2017). The second model which is the "Boom Crane Pulley" is designed to be installed and used in multi floor buildings that has a roof deck (Instructables, 2017). Both models of hoisting equipment can only be used if properly secured to a structure and for this reason, most individuals who may wish to have such equipment are being discouraged, since it may not be possible for them to modify an existing building to accommodate the installation of such equipment.

To eliminate the need for securing the hoisting equipment on a structure, Vander (2013) developed a portable balcony crane (Fig. 1), which has a maximum load lifting capacity of 136 kg using eight 20-litre plastic buckets of water as counter weight. However, in order to drop or lift a load on or from the balcony floor, the whole crane would have to be moved on its base to properly position the inclined arm before lowering or lifting the load. Though, caster wheels are mounted to the base of this portable balcony crane for mobility but moving the whole crane with the hoisted load and counter weights is stressful and unsafe. Also, the balcony may not be spacious enough to allow such movement, therefore, to eliminate this stressful activity, this work modified the portable balcony crane developed by Vander (2013), to a rotating arm balcony crane, which will encourage the adoption of the equipment in lifting operations among occupants in multi floor residential buildings.

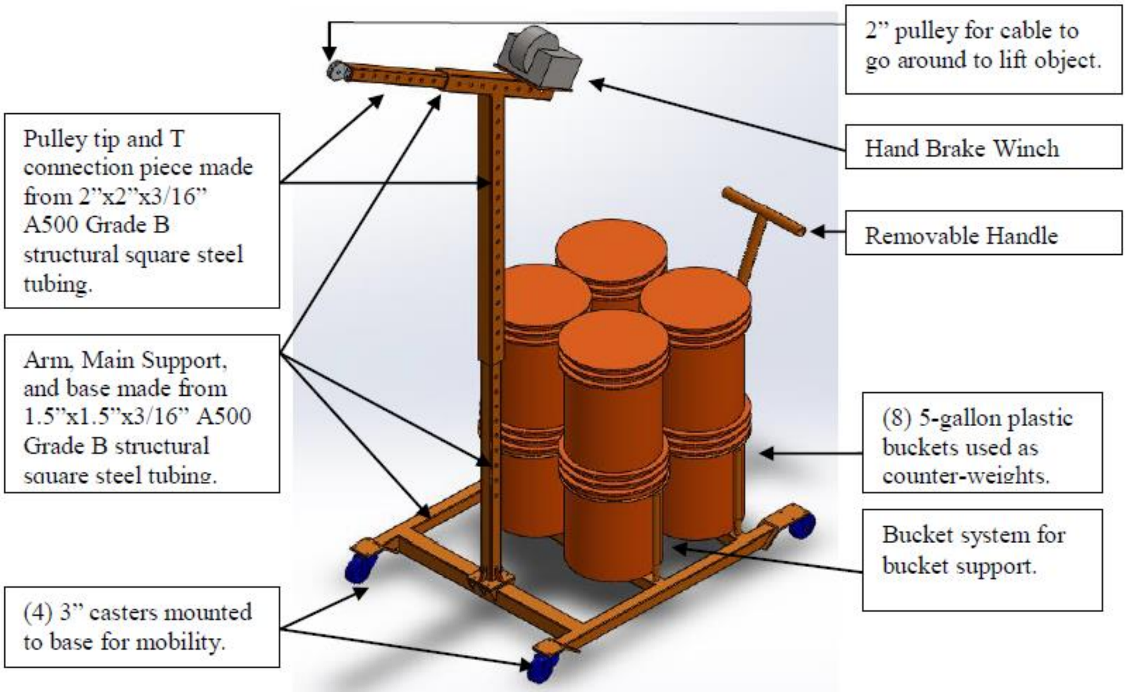

Figure 1: Portable Balcony Crane (Source: Vander, 2013)

2. DESIGN METHODOLOGY AND ANALYSIS

2.1 Design Modification Concept and Considerations

The following concepts and considerations brought about the modification of the existing balcony crane:

i. The modified balcony crane is designed for buildings not exceeding five floors (about 18 meters). Though, the mass of most household items/appliances like couches, freezer, washing machine, wooden cabinets, mattresses, etc., is less than 100kg, however, in order to account for the weights of the wire rope, hook and wind effect, the improved machine is designed to lift 140 kg load.

ii. To enable the load bearing arm of the improved balcony crane to be supported and rotated about the main support, the arm is welded to a vertical pipe that functions as a journal bearing at the upper part of the main support.

iii. To make the fabrication of the balcony crane easy, the main support and the rotating arm were designed to be constructed from standard mild steel pipes, while its base structure will be constructed from mild steel channel section beam sourced locally to encourage mass production of the crane.

iv. The height and width of a standard balcony in multi floor residential buildings in Nigeria are 2.5m and 1.2m respectively, while the height of the railing is 1.3m. Thus, to ensure that there is enough space to operate and move the crane within the balcony, the total height of the crane is 2.3m, while the length of its base is 1.1m. The winch will be mounted on base positioned at shoulder height (1425mm).

v. For safe operation of the crane, the manual winch selected for the operation of the improved balcony crane is incorporated with pawl and ratchet mechanism to avoid the free fall of loads during lifting operation, which may be catastrophic.

vi. The crane has to be strong enough to withstand the forces acting on it without tipping over or failing, but also lightweight for it to be lifted and assembled by two persons (50th percentile male). Thus, the components of the crane were designed to be safe, bolted and easily assembled within reasonable time.

2.2 Description of the modified balcony crane

The main parts of the portable rotating arm balcony crane (Fig. 2) include the base, counterweight platform, main support, rotating arm, and the winch assembly.

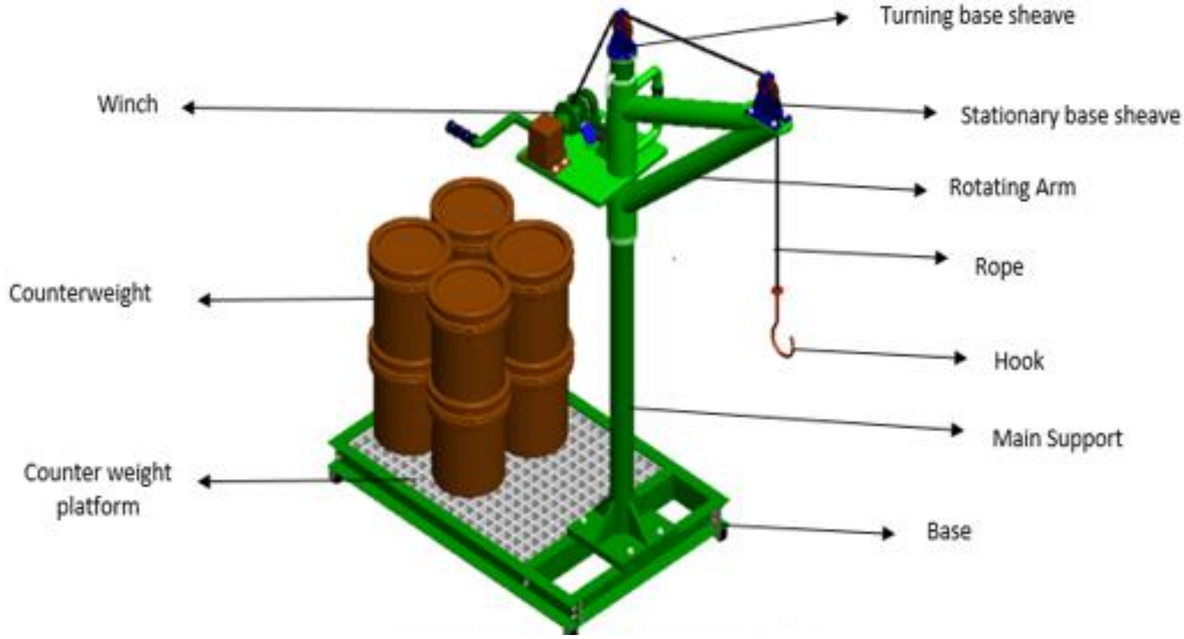

Figure 2: Portable Rotating Arm Balcony Crane

The rectangular base is the main supporting structure upon which every other component of the crane and the required counterweight are supported. It is a bolted section comprising of two long frames and three crosspieces fabricated from C-channel section beam. Between the mid crosspiece and the back crosspiece is a square cavity on which the counterweight platform, fabricated from steel grating is placed. Bolt holes are drilled on the top surface of each of the two front crosspieces through which the main support is bolted to the base. The main support is constructed from a steel pipe, with the lower end welded to an end plate with bolt holes for bolting down the main support to the base structure, while at the upper part of the main support is a journal on which the rotating arm of the crane is supported. A collar is welded below the journal to ensure that the rotating arm does not slide down along the main support, while above the journal, a hole is drilled through the main support, in which a pin is inserted to stop the rotating arm from sliding upwards. The rotating arm comprising of the plain bearing, inclined arm and rib, is a welded section fabricated from two different sizes of pipes. Also, welded to the main support is a platform on which the winch is mounted. The winch assembly comprises of the winch, sheave, rope and hook. One end of the rope is fitted to the hook, on which the load is hanged, while the order end of the rope runs through the sheaves to the hoisting drum of the winch, where it is securely fixed. As the winch is manually cranked, the rope winds round the turning hoisting drum, thereby lifting the load. With the load lifted up to the clearance of the balcony, the rotating arm with the hoist assembly is now swiveled to position the arm to the required position before lowering the hoisted load.

2.3. Determination of the Required Counterweight for the Crane.

The free body diagram of the crane is shown in Fig. 3, while the lengths of the structural members of the crane is shown in Fig. 4. The lengths of the rib (l_1) and the arm (l_2) were respectively determined from Equations (1) and (2) as 0.894m and 1.281m, while their angles of inclination (θ_1) and (θ_2) from the horizontal were respectively determined from Equations (3) and (4) as 26.5^o and 51.3^o. Since the maximum design mass (m_L), to be lifted by the crane is 140kg, therefore the rated design load lifting capacity (W_L) of the crane was determined from Equation (5) as 1400 N. Thus, considering the angle of inclination (θ_2) of the arm, the stability of the crane is governed by the equilibrium equation given in Equation (6), from which the counterweight (W_{CW}) required for the stability of the crane during lifting operation was estimated as 2010 N. Since the mass of water (m_w) water contained in a 20 litre plastic bucket is 20 kg, therefore, neglecting the weights of empty buckets, the number of 20 litre plastic buckets required as counterweight for the crane was determined as 10.04.

$$l_1 = \sqrt{(x_2)^2 + (y_1 - y_2)^2} \quad (1)$$

$$l_2 = \sqrt{(x_2)^2 + (y_1 - y_3)^2} \quad (2)$$

$$\theta_1 = \tan^{-1}\left(\frac{y_1 - y_2}{x_2}\right) \quad (3)$$

$$\theta_2 = \tan^{-1}\left(\frac{y_1 - y_3}{x_2}\right) \quad (4)$$

$$W_L = m_L \times g \quad (5)$$

$$(W_L \times x_2)\sin(\theta_2) = W_{CW} \times a\left(x_1 - \frac{a}{2}\right) \quad (6)$$

$$a = n_1 \times D_{bucket} \quad (7)$$

$$n_{buckets} = \frac{W_{CW}}{m_w \times g} \quad (8)$$

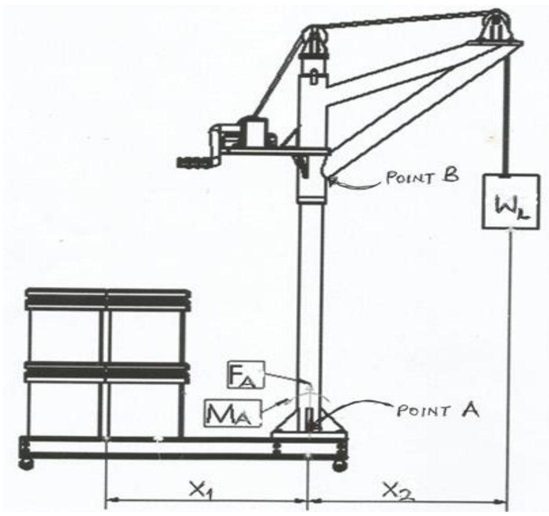

Figure 3: Free Body Diagram of the Crane

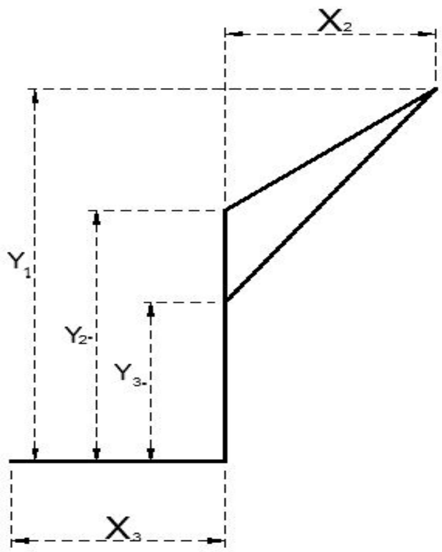

Figure 4: Dimension of the structural members of the Crane

Where; $x_1 = 1.0m$ and $x_2 = 0.8m$ are the back-span and swing of the improved balcony crane respectively; $x_3 = 1.2m$ is the length of the base structure; a, $n_1 = 2$, and $D_{bucket} = 0.32m$ are the effective length of the counterweight platform, number of buckets along the length of the counterweight platform, and diameter of each bucket respectively; $g = 9.81m/s^2$ is gravitational acceleration.

2.4. Selection of Materials for the Main Support and Rotating Arm of the Crane.

During lifting operation, as the load (W_L) is lifted from the ground as shown in Figure 2, the rotating arm is subjected to bending stress only, which will be maximum at point B, while the main support is subjected to combined tensile and bending stresses, which will be maximum at point A. Since the material for the main support and rotating arm is mild steel, and both components are likely to be subjected to uncertain stresses, thus, using a factor of safety (FS_1) of 4 as recommended by Sharma and Aggarwal (2006), the design load (W_{Ld}) for the main support and rotating arm was determined from Equation (9) as 5600 N. Consequently, the bending moment (M_A) at point A of the main support and bending moment (M_B) at point B of the rotating arm were respectively determined from Equations (10) and (11) as 4480 Nm and 3482 Nm. Thus, considering the effect of shock and fatigue loads, the design bending moments on the main support (M_1) and rotating arm (M_2) were determined from Equations (12) and (13) as 6270 Nm and 5223 Nm. Consequently, the section moduli (Z_1) and (Z_2) of the pipes required for fabricating the main support and rotating arm were respectively determined from Equations (14) and (15) as 57×10^3 mm^3 and 47.5×10^3 mm^3. From standard mild steel pipes property table (Steel, 2017), the tabulated section modulus closer to the calculated value for the main support is 65.8×10^3 mm^3, corresponding to a standard pipe with outer diameter and wall thickness of 101.6 mm and 8.0 mm, respectively. While, the tabulated section modulus closer to the calculated value for the rotating arm is 47.6×10^3 mm^3, corresponding to a standard pipe with outer diameter and wall thickness of 88.9 mm and 7.6 mm, respectively. Hence, these standard pipes were used in fabricating the main support and rotating arm. To enable the arm to be rotated on the main support whose outer diameter is 101.6 mm, a standard pipe with inner and outer diameters of 102.3 mm and 114.3 mm, respectively, was selected as the pipe for the plain bearing of the rotating arm. Since the plain bearing is vertically supported on the main support, it will be subjected to a maximum bending moment equal to that of the main support, and thus, the section modulus of the material must be equal to the calculated value for the main support. The tabulated value of the section modulus of the pipe selected for the plain bearing is 66.2×10[3]mm[3], which is higher than the calculated value of 57×10[3] mm[3], therefore, the selected pipe is safe.

$$W_{Ld} = W_L \times FS_1 \quad (9)$$
$$M_A = W_{Ld} \times x_1 \quad (10)$$
$$M_B = W_{Ld} \sin\theta_2 \times x_1 \quad (11)$$
$$M_1 = k_b \times M_A \quad (12)$$
$$M_2 = k_b \times M_B \quad (13)$$
$$Z_1 = \frac{M_1}{\sigma_b} \quad (14)$$
$$Z_2 = \frac{M_2}{\sigma_b} \quad (15)$$

Where; k_b (1.5) is the shock and fatigue factor for steady loads; σ_b (110 MPa) is allowable bending stress for mild steel (Sharma and Aggarwal, 2006).

2.5. Selection of Materials for the Structural Members of the Crane Base

The "through bolt" method of bolting will be used in coupling the structural members of the crane base and in bolting down the main support of the crane to the base. Thus, there will be need for easy access to the bolts and nuts during assembly and dis-assembly of the portable crane. Hence, a channel section beam was considered for the fabrication of the structural members of the base of the improved balcony crane. Since the load on the main support is equally shared by the two front crosspieces of the base on which the main support is bolted down on, thus, the various loads acting on each of the front crosspiece are depicted in the free body diagram (Fig. 5). The direct load (F_A) at point A on each crosspiece was determined from Equation (16) as 2800 N, while the moment at point A on each crosspiece is equal to M_A (4480 Nm). *Beamboy* software was used to analyse the effects of these loads on the crosspiece. The result of the analysis (Fig. 6) shows that the maximum shear force (F_{A1}) on the bolt used for bolting the crosspiece to the side frame of the base structure is 7370 N, while the maximum bending moment (M_{A1}) on the crosspiece is 2760Nm. Considering the effect of shock and fatigue loads, the design bending moments (M_3) on each of the crosspiece was determined from Equation (17) as 4140 Nm. Thus, the section modulus (Z_3) required for the material of the crosspiece was determined from Equation (18) as 16.8×10[3]mm[3]. However, due to its availability confirmed from market survey, the "$C100 \times 10.8$" type of channel section beam, which has a section modulus of 37.5×10[3] mm[3] and 5.74×10[3] mm[3] along its $X - X$ and $Y - Y$ axis respectively (Beer and Johnston, 1992), was selected as the material for the structural members for the base of the improved crane. Since the section modulus (Z_3) required for the crosspiece material is greater than the section modulus of the selected material along its $Y - Y$, but less than its section modulus along the $X - X$ axis, therefore, the channel section beam for the crosspiece will be bolted on the side frame in a way that the direct load (F_A) at point A of the crosspiece will act along the $X - X$ axis of channel section beam.

$$F_A = \frac{W_{Ld}}{2} \quad (16)$$
$$M_3 = k_b \times M_{A1} \quad (17)$$
$$Z_3 = \frac{M_3}{\sigma_{ys}} \quad (18)$$

Where; $\sigma_{ys} = 247$ MPa is the yield stress of mild steel.

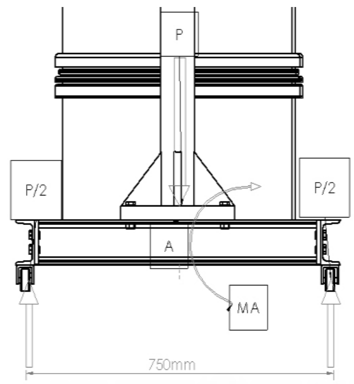

Figure 5: FBD of each front Crosspiece

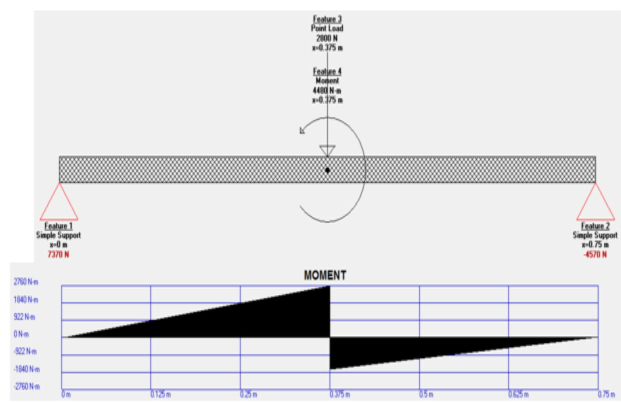

Figure 6: Force Analysis and Bending Moment of the cross piece

Similarly, Fig. 7 shows the free body diagram of the side frame for the base structure of the crane. The load (F_{A2}) at point C and D of each side frame of the base was determined from Equation (19) as 1842.5 N, while the load at point E is due to the counterweight ($W_{CW} = 2010$ N). *Beamboy* software was also used to analyze the effect of these loads on the side frame of the base structure and the caster wheels fixed under the frame. The results (Fig. 8) of this analysis shows that the maximum load on each of the front caster wheel placed under the side frames is 3840N, while the maximum load on each of the rear wheels is 1850N. Figure 8 also, shows that the maximum bending moment (M_C) on each side frame is 685 Nm, and considering the effect of shock and fatigue loads, the design bending moment on the side frame (M_4) was determined from Equation (20) as 1027.5Nm. Thus, the section modulus (Z_4) required for the material of the side frame of the crane base was determined from Equation (21) as 4.16×10^3 mm^3. which is less than the section modulus of the "$C100 \times 10.8$" type of channel section beam selected for the fabrication of the base, both along its $Y-Y$ and $X-X$ axis. However, to enable the caster wheels to be bolted properly due to the dimension of its base plate, the channel section beam for the side frame will coupled to the crosspiece in a way that the load (F_{A2}) at point C and D of the side frame will act along the $X-X$ axis of channel section beam.

$$F_{A2} = \frac{F_A}{4} \quad (19)$$

$$M_4 = k_b \times M_C \quad (20)$$

$$Z_4 = \frac{M_4}{\sigma_{ys}} \quad (21)$$

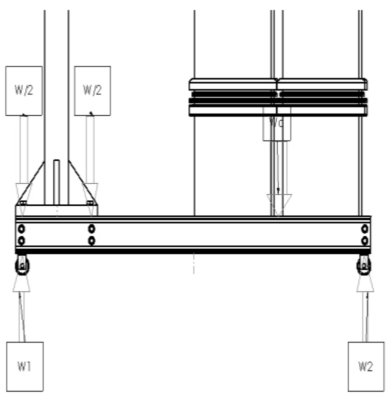

Figure 7: FBD of the Side Frames

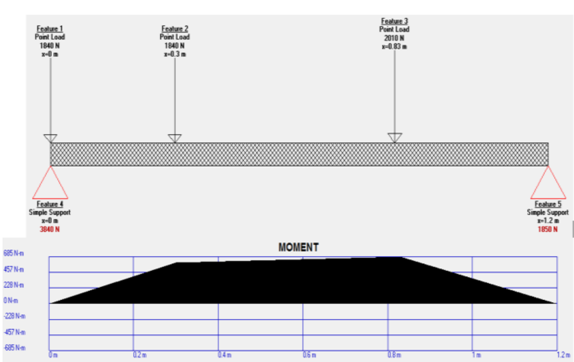

Figure 8: Force Analysis and Bending Moment on the side frame

2.6 Selection of Hoisting Rope, Sheave and Winch for the Crane

Among all the fibrous ropes used in hoisting applications, nylon rope is most preferred, because it has superior strength and remarkable stretching capabilities, and also allows a smaller size of rope to be used. Due to its availability and size, a 16 mm diameter nylon rope (8 strand) was selected as the hoisting rope for the crane because the factor of safety (FS_r) and the safe load (F_r) for the size of nylon rope selected is 12 and 3.30kN respectively (Cordage, 2006) which exceeds the maximum load lifting capacity of the crane (1.4kN). Thus, the maximum safe mass (m_r) for the selected rope was determined from Equation (22) as 336.40kg, which is two times greater than the rated load (140kg) of the portable crane, hence the selected rope is safe for the operation of the crane. Since nylon ropes are elastic and abrasion resistant, they can run on smaller sheaves compared to other fibrous ropes. Thus, the minimum pitch diameter (PD_s) of the sheave selected for the rope is 150 mm. According to Sharma and Aggarwal (2006), the best groove angle which provides greatest adhesion without slipping, prevents undue wedging and offers least resistance to the rope while leaving the groove is 45^o, therefore the groove depth (G_d), from the centre of the rope, radius of the groove (G_r) at the bottom of the sheave, and groove width (G_w) of the sheave were determined from Equations (23), (24) and (25) as 16 mm, 4.0 mm and 17.33 mm, respectively (Sharma and Aggarwal, 2006). For increased pressure velocity and load bearing capacity, a straight walled sheave with roller bearing was considered (NYLATECH, 2017), thus, the minimum thickness (t_s) and outer diameter (OD_s) of the required sheave were respectively determined as 58 mm and 182 mm using the relations in Equations (26) and (27). Furthermore, the diameter (d_s) of the shaft on which the sheave is mounted was determined as 23.15mm, using the maximum stress relation given by Burr and Cheatham (2002); Khurmi and Gupta (2005); Sharma and Aggarwal (2006) in Equation (28). Therefore, a standard 25 mm diameter shaft will be required for the sheave, consequently, a standard roller bearing sheave for 16 mm rope, with outer diameter OD_{s1}, bearing bore (d_{s1}), and sheave thickness (t_{s1}) of 203mm, 25.4mm and 60 mm (Loosco, 2017), was selected for the operation of the crane. The maximum pressure (P_{sb}) on the bearing of the sheave was calculated as 0.94 N/mm^2, using the relation in Equation (31) given by Sharma and Aggarwal (2006). Since this pressure did not exceed the maximum 6.0 N/mm^2 recommended for the lowest peripherial speed in manual hoists, therefore the selected sheave is satisfactory for the operation of the crane.

$$m_r = \frac{F_r}{g} \quad (22)$$
$$G_d = d_r \quad (23)$$
$$G_r = 0.25 d_r \quad (24)$$
$$G_w = d_r + \frac{d_r}{12} \quad (25)$$
$$t_s = 0.25 + 3.59 d_r \quad (26)$$
$$OD_s = PD_s + 2d_r \quad (27)$$
$$d_s = \left[\frac{16}{\pi\tau_2}\left(\sqrt{(M_s)^2 + (T_s)^2}\right)\right]^{\frac{1}{3}} \quad (28)$$
$$M_s = (W_L + W_r) \times \frac{t_s}{2} \quad (29)$$
$$T_s = (W_L + W_r) \times \left(\frac{PD_s}{2}\right) \quad (30)$$
$$P_{sb} = \frac{(W_L + W_r)}{t_{s1} \times d_{s1}} \quad (31)$$
$$W_r = w_{r1} \times H \quad (32)$$

Where; $w_{r1} = 0.147 kg/m$, is the weight of the nylon rope per unit length (engineeringtoolbox.com, 2017); $H = 20m$ is the total length of the nylon rope; $\tau_2 = 55Mpa$, is the permissible shear stress for a shaft without key way (Sharma and Aggarwal, 2006).

Also, it is obvious that apart from lifting and lowering of loads, the hoist should be able to pull the loaded nylon rope between the grooves of the fixed and rotating sheaves. Thus, a hand crank winch suitable for lifting, lowering and pulling within the rated capacity, with a self actuating brake mechanism, that ensures the suspension of loads and 'jerk-free' movement of the rope when lowering a load under pressure, was considered for the operation of the crane. The length of the crank handle (L_{ch}) and diameter of the winch drum (d_{wd}) that can accommodate the total length of fiber rope required for the opertion of the crane are 250 mm and 100 mm (Atlantic, 2015). Thus, the number of revolutions (N_{wd}) made by the winch drum to raise the load from the ground floor to the balcony of the last floor of a five story building and the ideal velocity ratio (VR_1) of the required winch were respectively determined from Equations (33) and (34) as 180 and 15.7:1. Consequently, the effort (P) required on the crank handle to lift the load was determined as 90.96N, using the relations in Equation (35) given by Khurmi (2013). However, a standard hand crank winch (spur gear type), Model LBW650NC (Atlantic, 2015) with gear ratio (VR_2), rated load lifting capacity, and maximum cable capacity of 15:1, 7357 N, and 22mm respectively, was selected for the operation of the crane.

$$N_{wd} = \frac{L_d}{d_{wd}} \quad (33)$$
$$VR_1 = \frac{2\pi L_{ch} N_{wd}}{L_d} \quad (34)$$
$$P = \frac{(W_L + W_r)}{VR_1} \quad (35)$$

Where; $L_d = 18m$, is the load distance (equivalent to the height of a standard 5-storey building.

2.7 Selection of Bolts and Base Plates

Figure 8 shows the base plate with bolt holes, through which the main support is bolted down to the base structure of the crane. According to Khurmi and Gupta (2005), the direct load (w_{t1}) carried by each bolt and the load (w_1) in a bolt per unit distance were respectively determined from Equations (36) and (37) as 1400 N and 52.7N/mm. Since the heavily loaded bolt is at a distance, L_2 from the tilting edge of the base plate, therefore the load (w_{t2}) on the heavily loaded bolt was determined from Equation (38) as 10541.18N. Thus, the maximum tensile load (w_t) on the heavily loaded bolt was determined as 11941.18 N using the relation in Equation (39). Mild steel coarse series bolts were used for the bolting, therefore, with a factor of safety (FS_b) of 5 for the bolts (Khurmi and Gupta, 2005), the core diameter (d_{c1}) of the bolts was estimated from Equation (40) as 19.898 mm. From the standard bolt table (coarse series) in Khurmi and Gupta, (2005), the standard core diameter of the bolt is 20.320 mm and the corresponding size of bolt selected for bolting down the main support to the base structure of the crane is M24. As shown in Figure 5, the bolts used in coupling the crosspiece to the frame of the base is subjected to a maximum shear force (F_{A1}), which is equal to 7370 N. Thus, using Equation (41), the core diameter (d_{c2}) of the bolts was determined as 15.632. Similarly, from standard table (IS: 4218-1996) for coarse series, the standard core diameter of the bolt is 16.933 mm and the corresponding size of bolt selected for coupling the crosspiece to the side frame of the base structure of the crane is M20.

$$w_{t1} = \frac{W_{Ld}}{n_1} \quad (36)$$
$$w_1 = \frac{M_A}{2[(L_1)^2 + (L_2)^2]} \quad (37)$$
$$w_{t2} = w_1 \times L_2 \quad (38)$$
$$w_t = w_{t1} + w_{t2} \quad (39)$$
$$d_{c1} = \sqrt{\frac{w_t}{\frac{\pi}{4} \times \frac{\sigma_{tb}}{FS_b}}} \quad (40)$$
$$d_{c2} = \sqrt{\frac{F_{A1}}{\frac{\pi}{4} \times \frac{\sigma_{tb}}{FS_b}}} \quad (41)$$

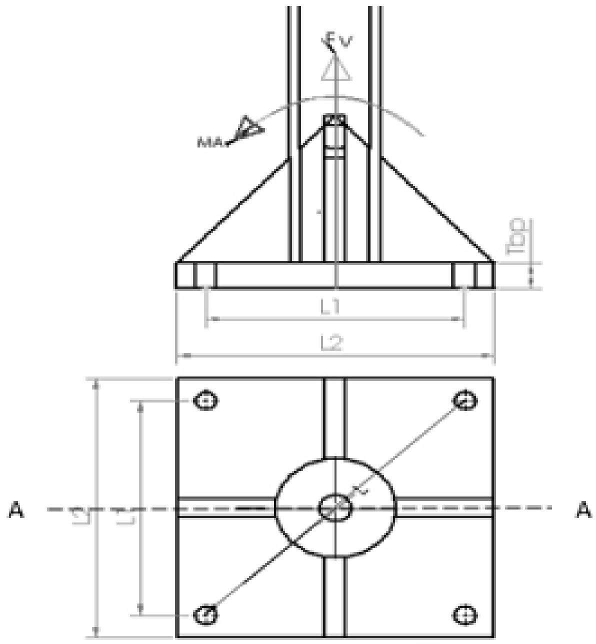

Figure 8: Diagram of the Base Plate for Bolting the Main Support

Where; $n_1 = 4$, is the number of bolts on the base plate; L_1 =50 mm; $L_2 = 200$; σ_{tb} =192 N/mm^2, is the allowable tensile stress of grade 4.6 mild steel bolts (Engineering Spreadsheets, 2003)

The force trying to separate the base plate of the main support from the crosspiece is resisted by four bolts, and this force is equal to the direct load ($w_{t1} = 1400$ N) carried by each bolt (Equation 35). The bending moment (M_{p1}) due to the force in two bolts was calculated from Equation (42) as 350 kN-mm. Mean radius (r_m) of the arc from section A-A over which the load due to tensile force on the main support may be taken to be concentrated was determined from Equation (43) as 50.8 mm, and the centroid ($\bar{x}$) of this arc from A-A was calculated as 32.34 mm, using the relation in Equation (44), given by Khurmi and Gupta (2005). Thus, the bending moment (M_{p2}) due to the tensile load on the main support was estimated from Equation (45) as 90.552 kN-mm. Since the bending moments M_{p1} and M_{p2} are in opposite direction, therefore net resultant moment (M_p) on the base plate about section A-A was determined from the relation in Equation (46) as 119.45 kN-mm. The width (b) of the base plate at section A-A was determined from Equations (47) as 259.45 kN-mm. Thus, the minimum thickness (t_{bp}) of the base plate was estimated as 14.60mm, using the relation given by Khurmi and Gupta (2005) in Equation (48). Therefore, a standard 15 mm thick mild steel plate was used in fabricating the base plate of the main support and the base on which the winch is mounted.

$$M_{p1} = 2w_{t1} \times \frac{L_3}{2} \tag{42}$$

$$r_m = \frac{D_1}{2} \tag{43}$$

$$\bar{x} = 0.6366 r_m \tag{44}$$

$$M_{p2} = 2w_{t1} \times \bar{x} \tag{45}$$

$$M_p = M_{p1} - M_{p2} \tag{46}$$

$$b = L_3 - D_1 \tag{47}$$

$$t_{bp} = \sqrt{\frac{M_p}{\frac{1}{6} \times b \times \frac{\sigma_{yt}}{FS_b}}} \tag{48}$$

Where; $D_1 = 101.6$ mm, is the outer diameter of the pipe for the main support;

Furthermore, the free body diagram of the hoisting drum of the winch assembly is shown in Figure 9. Since the drum shaft is simply supported on its bearing, therefore, taking moments about B, the force (F_A) parallel to the wire rope on bearing A was determined from Equation (49) as 3136 N, while the angle of inclination (θ) of the wire rope was determined from Equation (50) as 25^o. Thus, the vertical (F_{AV}) and the horizontal (F_{AH}) components of the force (F_A), were determined from Equations (51) and (52) as 2842.20 N and 1325.33 N. Due to the load F_{AV}, bolts at bracket A will be in direct tension only, thus, the tensile load (F_{AV1}) per bolt was estimated from Equation (53) as 1421.10 N. The load F_{AH} will cause both tension and direct shear on the bolts. The tensile load (w_2) in a bolt per unit distance caused by F_{AH} was determined from Equations (54) as 6.36 N/mm, therefore, the tensile load (w_{t3}) on the heavily loaded bolt was determined from Equation (55) as 1113.30 N. Thus, the total maximum tensile load (w_{th}) on each bolt was estimated from Equation (56) as 2534.40 N. The direct shear load (w_{s2}) caused by F_{AH} was determined from Equations (57) as 1267.18 N. Combining the tensile and shear loads, the equivalent tensile load F_e on each bolt was calculated as 3059.30 N, using Equation (58). While, the core diameter (d_{c2}) of the bolts was estimated from Equation (59) as 10.072 mm, thus, from standard table (IS: 4218-1996) for coarse series, the core diameter of the bolt is 11.546 mm and the corresponding size of bolt selected for bolting down the winch of the crane is M14 bolts.

$$F_A = \frac{W_{Ld} \times l_4}{l_5} \tag{49}$$

$$\theta = 90 - \left[\tan^{-1}\left(\frac{y}{x}\right)\right] \tag{50}$$

$$F_{AV} = F_A \cos\theta \tag{51}$$

$$F_{AH} = F_A \sin\theta \tag{52}$$

$$F_{AV1} = \frac{F_{AV}}{n_3} \tag{53}$$

$$w_2 = \frac{F_{AH} \times h_1}{[(l_8)^2 + (L_9)^2]} \tag{54}$$

$$w_{t3} = w_2 \times L_9 \quad (55)$$

$$w_{th} = F_{AV1} + w_{t3} \quad (56)$$

$$w_{s2} = \frac{w_{th}}{n_3} \quad (57)$$

$$F_e = \frac{1}{2}\left[w_{th} + \sqrt{(w_{th})^2 + 4(w_{s2})^2}\right] \quad (58)$$

$$d_{c2} = \sqrt{\frac{F_e}{\frac{\pi}{4} \times \frac{\sigma_{yt}}{FS_b}}} \quad (59)$$

Where; $n_3\ (2)$, is the number of bolts on each side of the winch bracket; x (300 mm) is the horizontal distance from the main support to the centre of the hoisting drum shaft; y (640 mm) is the vertical distance from the turning sheave to the hoisting drum shaft; $L_4 = 140mm$; $L_5 = 250mm$; $L_6 = 200mm$; $L_7 = 150mm$; $h_1 = 150mm$; $L_8 = \frac{L_6 - L_7}{2}$; $l_9 = L_7 + L_8$

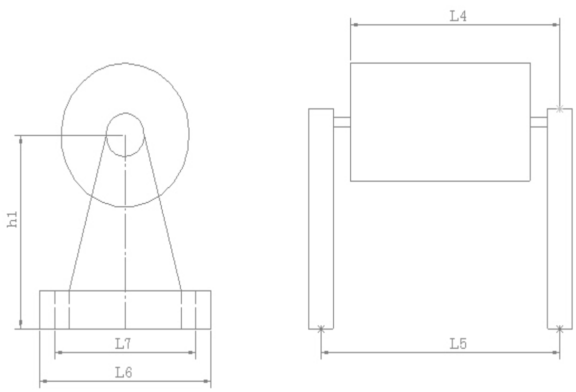

Figure 9: Free Body Diagram of the Hoisting Drum of the Winch

2.8 Performance Evaluation Procedure

The modified balcony crane was evaluated after its fabrication to determine the time taken and effort required to lift varying household loads. In evaluating the crane performance, the standard procedure for testing and examining lifting appliances as recommended by mardep.gov (2018) was followed whereby the counter weight was kept constant at ten 20L plastic buckets of water. The load was varied from 50kg to 150kg and to get uniform load capacity, bags of rice weighing 50kg each was used in testing the machine. The time taken to carry the bags of rice to the fifth floor of the residential building was taken using a stop watch and the time taken to lift the same bags of rice was also recorded. In order to determine the effort expended in lifting the load using the portable crane, the load was varied from 200N to 1400N and the effort required to lift the load was computed using Equation (60).

$$P_{actual} = \frac{(W_L + W_r)}{\eta_t \times VR_2} \quad (60)$$

Where; η_t (= 75%,) is the transfer efficiency of spur gear type of winch (Atlantic, 2015).

2.9 Bill of Engineering Materials and Evaluation/Cost Comparison

The materials required for the fabrication of the improved balcony crane and the labour costs involved were quantified and presented in Table 1. The total cost (C_2) of producing one unit of the improved balcony crane is Sixty-Four thousand, Eight Hundred and Fifty Naira (₦64,850:00) only). The percentage cost reduction ($\%C$) in the improved balcony crane compared to the cost of the balcony crane developed by Vander (2013), was calculated as 62.91 % using the relation in Equation 61.

$$\%\ C = \frac{C_1 - C_2}{C_1} \quad (61)$$

Where; C_1 (= 480.95 USD = ₦172,142:00), is the cost of the balcony crane developed by Vander (2013), excluding importation cost.

Table 3.1: Bill of quantity and labour cost

S/No	Description	Quantity	Unit Price ₦ K	Amount ₦ K
1	Mild steel pipe (Ø101.6 mm, 8 mm thick)	2.3 m	3,500:00	3,500:00
2	Mild steel pipe (Ø88.9 mm, 7.6 mm thick, 2.2 m Length)	2.2 m	2,500:00	2,500:00
3	Mild steel pipe (Ø102.3 mm, 6 mm thick, 0.6 m Length)	0.6 m	1,300:00	1,300:00
4	Mild steel Channel section beam ($C100 \times 10.8$ mm)	4.6 m	8,000:00	8,000:00
5	8 Strands Nylon Rope (Ø 16 mm)	20 m	200:00	4,000:00
6	Roller bearing Sheave (for 16 mm nylon rope)	2	2,000:00	4,000:00
7	Hand Crank Winch (Model No.: LBW650NC)	1	15,000:00	15,000:00
8	Bolts and nuts (Size: M24, M20 and M14)	24 pcs	50:00	1,200:00
9	Mild steel Plate (400mm×800mm×15mm)	1	2,000:00	2,000:00
10	Hoisting hook	1	1,500:00	1,500:00
11	Caster wheels	4	1,000:00	4,000:00
17	Labour cost	LS		15,000:00

18	Miscellaneous and Transportation	LS	2,850:00
	TOTAL		**₦ 64,850:00**

3. RESULTS AND DISCUSSION

Performance of the portable hoisting crane as compared to manual lifting of household load is shown in Figure 10. It is shown that there is 62.5 percent time savings in the use of the improved portable balcony crane for load lifting compared to manual lifting of the loads using the staircase. For 150kg load (equivalent of 3 bags of rice), the occupant had to walk through the staircase at least thrice to move the loads to the upper floors of the building.

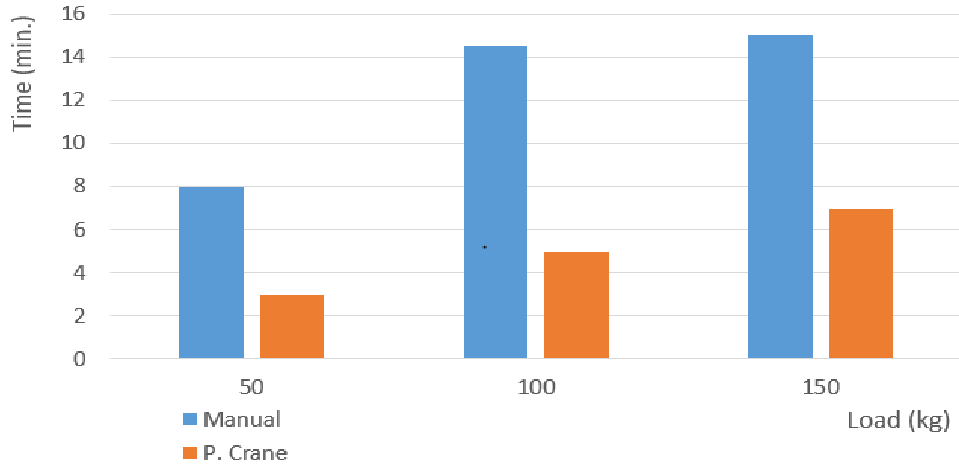

Figure 10: Lifting time for Loads

Figure 11 is the graph for both the ideal and actual effort applied against the lifted loads. It is evident that the effort progress linearly with the load and the actual mechanical advantage obtained from the graph is 11.02. This shows that the portable crane lifts loads that are eleven times heavier than the required effort.

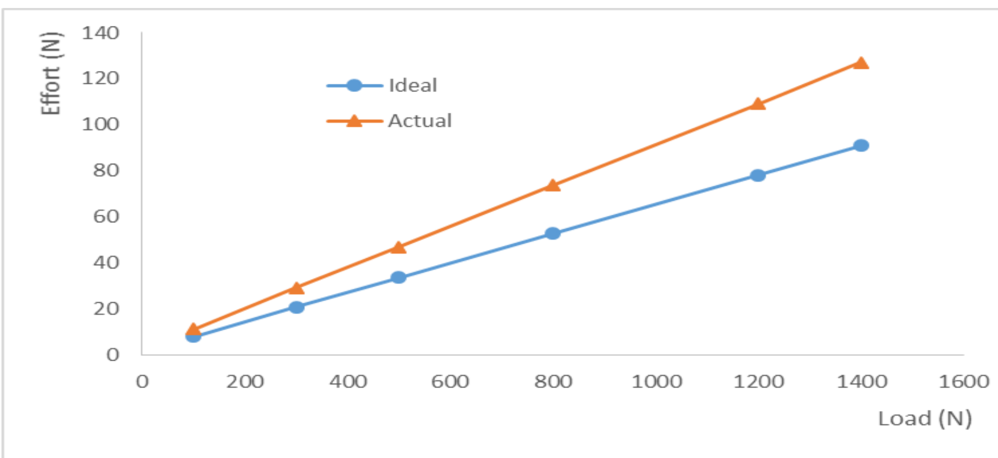

Figure 11: Mechanical Advantage of the Crane

4. CONCLUSION

A portable rotating arm balcony crane for lifting large and heavy household items in or out of apartments on the upper floors of multi floor residential buildings was developed in this study. The results of the performance testing of the crane confirmed that a maximum load of 150kg can be lifted safely, using ten 20 liters plastic buckets of water as counterweight. The use of this portable hoisting equipment will reduce the manual lifting of heavy and large household items/loads like furniture, fridges, through narrow staircase whose ripple effect would in turn reduce damage to these items and musculoskeletal related diseases. Also, compared to the existing portable balcony crane developed by Vander (2013), there is about 62.91 percent cost reduction in adopting the improved balcony crane. Thus, due to its low cost, easy installation, operation and maintenance, this portable balcony crane is recommended to commercial homeowners and their tenants. The crane can also be used in building construction sites to handle loads that are not safe to be lifted manually.

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