MACHINE DESIGN, VOL.4(2012) NO.1, ISSN 1821-1259 PP. 15-20

Preliminary note

DEVELOPING A DYNAMIC LOAD-TRACKING LEARNING-SOFTWARE FOR WINCH LIFT DESIGN

Emmanuel SIMOLOWO [1,*] - **Adeboye GBADEBO** [2]
[1,2] Department of Mechanical Engineering, University of Ibadan, Ibadan, Nigeria

Received (27.12.2011); ***Revised*** (30.01.2012); ***Accepted*** (06.02.2012)

Abstract: *Many engineering applications involve the problem of a dynamic loading property which is often hard to track and critical in design analysis. The winch-lift design learning software developed in this work incorporates the graphical display of dynamic loading on the cable or rope of the winch lift within the first seconds of its operation. The learning software allows the user to generate a profile of this load and its acceleration against time in order to obtain the maximum value of the load to be borne by the whole unit. This feature is underplayed in most engineering software which usually uses the static total nominal load for simplicity.*
The overall program flow is in three sections based on the software architecture. They are the study, practice and the assessment sessions. At the validation stage, the results generated were compared with those obtained by numerical procedures in literature. The results were in the same range with differences in values of the order 0.1%. In conclusion, a learning software for exploring motorized winch systems has been developed in this work.

Key words: *Winch Lift, Dynamic Loading, Learning Software, Results Validation*

1. INTRODUCTION

Machine design, unaided with relevant software application, is generally an interactive process with associated tedium of numerical synthesis and analysis. It is important that designers learn the process designing in the most accurate and yet least involving and time consuming means possible by the use of interactive software thereby constituting an improvement on conventional design computations and design teaching methods. More so, the creativity in engineering design should be a critical function of education [1, 2].
Software design methodology has undergone tremendous transformation from the periods when Pascal/C and SA/SD were mainly used to develop applications to the present times when factors like modeling, globalization, coding, testing, and maintenance are taken into consideration [6].Today, software packages are being used in virtually all fields and at all levels to improve learning skills. Recent Works related to this area of study include; software design for turbo machine parts [5], profile design of cams and followers systems [15, 16, 17] and design of mechatronic systems with varying dynamics [9]
The learning software developed here in performs basic design calculations relating to the design of a winch lift. Winch lifts employ many engineering principles and modern ones combine the use of wheel and axle, gearing and leverage to achieve lifting and thus form the basis for an important design case study for training winch designers. The picture and a simple model of a motorized winch is shown in Figure 1. The mathematical involvements of the major aspects of the winch lift design process are incorporated in the software to allow the user to see the effects of all design modifications on the performance of the lift. It is meant to serve as an example of what can be achieved by correctly employing software in machine design and to emphasize the importance of computer-aided techniques in training and learning.
As a learning aid, this developed software grossly reduces the time required to carry out design computations and gives the users a clearer view of the overall design objectives. Professionally used, it will reduce the effort required of both user and benefactor in performing their respective roles in the process of design.
The Winch lift constitutes a good basis for an important engineering case study and learning aid because it incorporates critical design concepts. It can be used to show the general approach to design and also to monitor, point out and correct faulty design practices. It incorporates; (i) the general approach to machine elements design concepts (ii) winch lift design implications (iii) design of some tribo-elements.

2. NUMERICAL DESIGN

Procedures obtained from earlier works in machine design [7, 8, 9, 11] as well as other design principles [3, 4, 6] have been applied in the development of this learning aid. The description of the mathematical design of the winch lift is in two stages: (i) design of lifting mechanism (ii) gear-link design.

2.1. Design of Lifting Mechanism

The lifting mechanism design stage determines critical parameters needed for the entire design process. The time for each lift cycle called the lift rate. (L_T) is given by equation (1), while equation (2) gives the actual lift time (A_{alt}). N_L is the number of lift actions and L_T is the lift time

* Correspondence author University of Ibadan, Department of Mechanical Engineering, Faculty of Technology, Nigeria, esimmar@yahoo.com

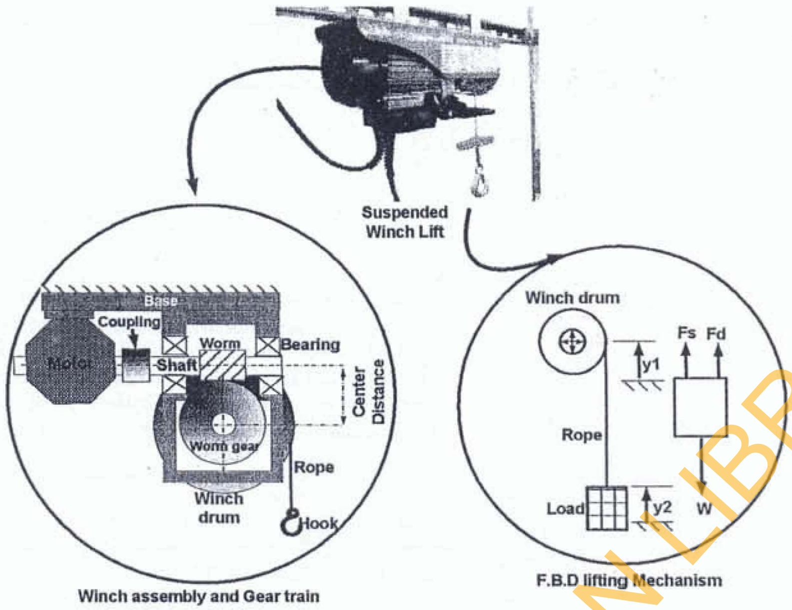

Fig.1. Motor Driven Winch Lift Mechanism

$$L_r = \frac{N_L}{L_T} \tag{1}$$

$$A_{alt} = (L_r)(L_f) \tag{2}$$

Lift fraction L_f is the action part of the total time (lifting, lowering and static) utilized only for the lifting of load. All these values are based on the design requirements. Equation (3) computes the average velocity (A_{VL}), while the live load (L_L) and the total nominal load (T_{NL}) are obtained by equations (4) and (5) respectively. The dead load comprises the weight of rope and hook while the lift height (L_h) is vertical height through which load is conveyed. The weight of item is W_i while N_i is the number of item. In equation (5), L_L is the live load and D_L dead load.

$$A_{VL} = \frac{L_h}{A_h} \tag{3}$$

$$L_L = (W_i)(N_i) \tag{4}$$

$$T_N = L_L + D_L \tag{5}$$

The power required for lifting action (P_L) is obtained by equation (6), while D_{av}, the angular velocity of the drum shown in figure 1 is computed with equation (7), where D_{rad} is drum radius. Also equation (8) gives the gear ratio (G_r) for the worm set where N_s is the motor speed.

$$P_L = \frac{(T_{NL})(L_h)}{A_{Lt}} \tag{6}$$

$$D_{av} = (A_{VL})(D_{rad}) \tag{7}$$

$$G_r = \frac{D_{av}}{N_s} \tag{8}$$

To determine the maximum dynamic load F_{max}, the dynamic loading on the rope or chain must be considered. Based on the schematic dynamic system and free body diagram shown in Figure 1, the presence of the rope which acts as a spring/damper system sets up vibration in the system lasting as long as it takes the load to reach steady state [12]. It is important to estimate the value of this maximum load to ensure that it does not exceed the maximum permissible load on the rope and that the system can provide enough energy to overcome this load. This maximum force and drum torque are from the on going analysis. Specifying weight as W, spring force as F_s and damper force as F_d, Newton's 2nd law for this free body diagram gives.

$$F_s + F_d = \frac{w}{g}\ddot{y}_2 \tag{9}$$

Where, $F_s = k(y_1 - y_2)$; $F_d = c(\dot{y}_1 - \dot{y}_2)$ with $\ddot{y}$ represents acceleration and $\dot{y}$ as velocity. Subject to initial conditions: $t = 0$; $y_1(0) = 0$; $y_2(0) = 0$; $\dot{y}_1(0) = v$; $\dot{y}_2(0) = 0$; equation (9) transforms to equation (10)

$$y_2 = \left[\frac{k}{m}(y_1 - y_2) + \frac{c}{m}(\dot{y}_1 - \dot{y}_2) - g\right] \tag{10}$$

Equation (10) above can be rearranged as (11) to give a 2nd order ODE with constant coefficients.

$$\ddot{y}_2 + \frac{k}{m}y_2 + \frac{c}{m}\dot{y}_2 = \frac{c}{m}\dot{y}_1 + \frac{k}{m}y_1 + g \tag{11}$$

Once it is recognized that for any value of time t, $\frac{c}{m}$ and

$\frac{k}{m}$ are constants and constant coefficients B = $\frac{c}{m}$ and C = $\frac{k}{m}$; then equation (11) is rewritten as equation (12)

$$\ddot{y}_2 + B.y_2 + C.\dot{y}_2 = D(t) \quad (12)$$

$$D(t) = \frac{c}{m}.\dot{y}_1 + \frac{k}{m}.y_1 + g \quad (13)$$

The solution to equation (12) is however dependent on one major assumption: 'The rope spring K is constant over the period for which the vibration exists'. This is not the case in reality as the rope is being coiled up during this period thereby reducing its length. In reality, the rope would have to be treated as a collection (n number) of springs in series such that for springs of constant k', the overall spring constant would be $k = \frac{k'}{n}$. This would allow the designer to account for the coiling up of the rope by removing successive springs from the series based on the speed of coiling. The spring constant would thus change from $\frac{k'}{n}$ at the start to $\frac{k'}{m}$ at some later time where m is number of springs in the series at that time.

The assumption is however a reasonable one since the time period under consideration is small (of the order of a few seconds) and thus the amount of rope coiled up is fairly negligible. A complementary solution to the corresponding homogenous equations $\ddot{y}_2 + B \cdot y_2 + C \cdot \dot{y}_2 = 0$ with constant coefficients is found. The complementary solution is

$$y_2 = \exp(\frac{c}{2m} \cdot t)\{A \cdot Cos((ARG)(t)) + B \cdot Sin((ARG)(t))\} \quad (14)$$

Where, $ARG = \sqrt{\left[\frac{k}{m} - (\frac{c}{2m})^2\right]}$, A and B are arbitrary constants. A particular solution to equation (12) is then found based on the form of the right hand side. Since the right hand side is of the form $D(t) = \frac{c}{m} \cdot \dot{y}_1 + \frac{k}{m} \cdot y_1 + g$, we expect a particular solution of the form $y_2 = (d \cdot t) + e$. The particular solution is found using the initial conditions to be $y_2 = (v \cdot t) - \frac{mg}{k}$. The general solution is obtained as a sum of complementary and particular solutions to give.

$$y_2 = \exp(\frac{c}{2m} \cdot t)[A \cdot Cos((ARG)(t)) + B \cdot Sin((ARG)(t))] + \Delta \quad (15)$$

Where $\Delta = (v \cdot t) - \frac{mg}{k}$

$\ddot{y}_2$ is written as

$$\ddot{y}_2 = \exp(-\frac{c}{2m} \cdot t)[H \cdot Cos((ARG)(t)) + \Psi] \quad (16)$$

Where $\Psi = H1 \cdot Sin(ARG)(t))$

$$H = (-\frac{c}{2m} \cdot D) + ((ARG)(D1));$$

$$H1 = (-\frac{c}{2m} \cdot D1) + (ARGxD);$$

$$D = (-\frac{c}{2m} \cdot A) + ((ARG)(B));$$

$$D1 = (-\frac{c}{2m} \cdot B) - ((ARG)(A)).$$

Expressions for the arbitrary constants A and B are computed based on the initial conditions as $A = \frac{mg}{k}; B = \frac{\{(\frac{cg}{2k}) - v\}}{ARG}$. At this point a complete expression for the acceleration $\ddot{y}_2$ in terms of time t alone is available. A plot of values of the $\ddot{y}_2$ against t is used to obtain the maximum value of $\ddot{y}_2$. Then the maximum load (force) and thus maximum drum torque are found by applying Newton's 1st law as.

$$F_{max} = m\,\ddot{y}_2 \quad (17)$$

$$T_{max} = F_{max} D_{sr} \quad (18)$$

The plot of $\ddot{y}_2$ and F_{max} are displayed in Figure 5. D_{sr} is the Drum Sheave Radius. These maximum values set the requirements which the worm gear set must meet in order to lift the load. The maximum load must not exceed the maximum permissible load on the rope in order to prevent rope failure and the maximum drum torque must be met or exceeded by the worm set output torque value in order for the winch to operate at all [14]

2.2. Tribo-element Design for Winch

This design stage involves using models of each part of the winch lift and appropriate functions to calculate the required parameters [15]. Examples of such parameters are forces, moments, torques, deflections of particular sections and adequate safety factors based on material choices. This forms the main area in which the iterative procedure is employed. The main design includes the sizing of the worm – set speed reducer, the shaft carrying the drum and the bearings within which it rotates.

The complete design of a worm set is herein performed based on design information or parameters obtained from the lift mechanism design. Parameters include the required Torque (M_1), input speed (N_1) and output speed (N_2). The selection of material for worm/wheel and diameter factor (q) is based on standard tables [12]. The gear ratio R_g can be obtained using equation (19), while the center distance (a) between worm and wheel is an assumed parameter at commencement of design. The number of treads on worm (Z_1) and wheel (Z_2) are obtained by equations (20) and (21) respectively where m is the axial module. The pitch diameters for worm (d_1) and wheel (d_2) are calculated using equations (22) and (23). The lead angle (λ) ensuring self locking capability is calculated employing equation (24) in terms of the worm diameter (d_w), worm gear diameter (d_g), number of worm teeth (n_w), number of worm gear teeth (n_g).

$$R_g = \frac{z_1}{z_2} \quad (19)$$

$$Z_1 = \frac{\{7 + (2.4a^{0.5})\}}{R_g} \quad (20)$$

$$Z_2 = R_g z_1 \quad (21)$$

$$d_1 = q \times m = \frac{a^{0.875}}{1.53} \quad (22)$$

$$d_2 = 2\pi P_X = (2.a - d_1) \quad (23)$$

$$\lambda = Tan^{-1}\left(\frac{n_w}{p_d \cdot d_w}\right) \quad (24)$$

The max allowable face width b_a is calculated by equation (25) where P_x the axial pitch of the worm as appearing in equations (26) and q the diameter factor obtained from standard tables [12]. The tangential velocity V_t is obtained by equation (27) where n_1 and n_2 are the rotational speed of worm and wheel respectively.

$$b_a = 2 \,.\, m(q+1)^{0.5} = \text{number} < 0.67.\, d_1 \quad (25)$$

$$M = \frac{P_x}{\pi} \quad (26)$$

$$V_t = \frac{\pi n_1 d_1}{60} \quad (27)$$

The tangential load factors based on materials (C_s), size ratio (C_m) and friction (C_v). Subsequently the tangential load (W_{tg}), friction force (W_f), rated power output (ϕ_o), power loss (ϕ_L), rated power input (ϕ), efficiency of worm set (η) and rated output torque (T_o) are calculated by equations (28), (29), (30), (31), (32), (33) and (34) respectively. The coefficient of friction is μ.

$$W_{tg} = 0.0132\, C_s\, C_m\, C_v\, b_a\, d_2^{0.8} \quad (28)$$

$$\frac{\mu W_{tg}}{Cos(\alpha_n)Cos(\lambda)} \quad (29)$$

$$\phi_o = \frac{n_1 W_{tg} d_2}{R_g} \quad (30)$$

$$\phi_L = V_t W_f \quad (31)$$

$$\phi = \phi_o + \phi_L \quad (32)$$

$$\eta = \frac{\phi_o}{\phi} \quad (33)$$

$$T_o = W_{tg} \times 0.5 \times d_1 \quad (34)$$

The values of rated input power and rated output torque obtained from tribo-element design must meet the requirements for the lift mechanism designed earlier otherwise a redesign is necessary. The design process is brought to an end by the documentation stage which involves the complete specification of all the parameters which accurately define the machine (winch lift) to achieve the particular design criteria [1].

2.3. Sample Winch Lift Design

Shown in Table 1 is a complete list of the results obtained numerically. Equations and processes described in section 2.2 above are applied to a specific case study of winch lift design. The results were obtained in two iterations and were later compared with results obtained using developed software.

3. SOFTWARE DEVELOPMENT

The ideal choice of programming language for the development of the learning software was Visual Basic.Net. The relative ease with which Visual Basic allows a user to create the interface (see Figure 6) made it the most appealing option and so it was chosen. The flow chart for the design stage is shown in Figure 2. The main stages involved in the software algorithm development were:

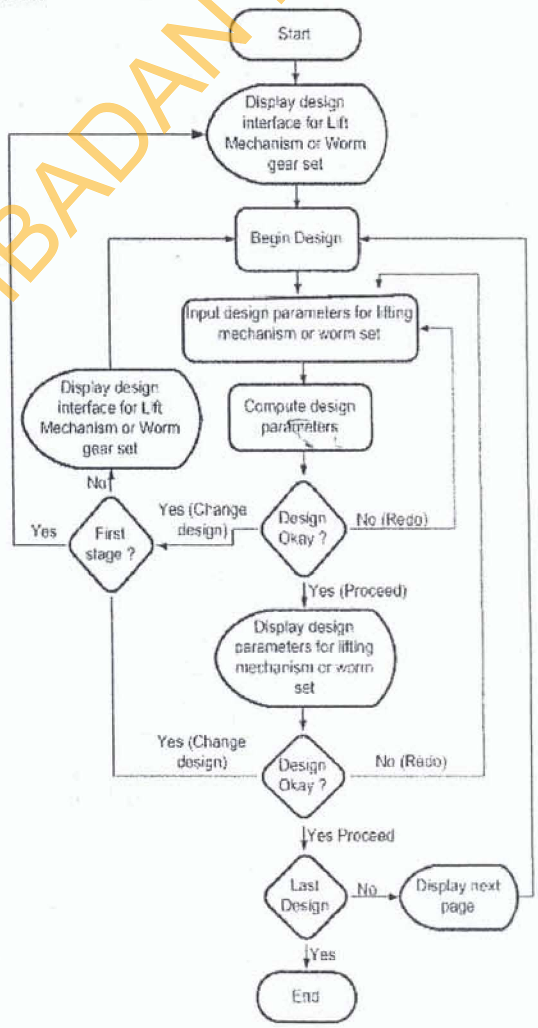

Fig.2. Flow chart of software Practice session

Table 1. Sample Numerical Design

S/N	*Design Parameters*	*Values and specifications*	*Status of Parameter*
	Specifications for case study design		
1.	Items	100	Given values based on design requirements
2.	Time	2000 seconds	
3.	Height	7.3152 m	
4.	Item Weight	22.2109 Kg	
5.	Spring constant K'	49905.21 N/m^2 (based on material)	
6.	Rope Diameter	0.025 m	
7.	Power Source	Motor	
8.	Dead Load	222.486 N	Assumed values
9.	Items per Lift	2	
10.	Sheave Diameter	0.25 m	
11.	Lift Time fraction	0.3	
	Lifting Mechanism design		
12.	LR	40 seconds per cycle	Results obtained
13.	ALT	12 seconds per lift	
14.	AVL	0.6069 m/s	
15.	LL	435.7 N	
16.	TNL	658.2 N	
17.	MDL	3261.2 N	
18.	PR	0.6019 KW	
19.	DAV	2.438 RAD/SEC	
20.	GR	1 : 75	
	Worm gear set design		
	Parameters	Initial Results	Re-design and final results
21.	a	0.139 m	0.150
22.	R_g	1 :75	1 :75
23.	d_1	0.0511 m	0.0546 m
24.	d_2	0.2269 m	0.2454 m
25.	λ	0.0592 rad	0.0598 rad
26.	b_a	0.0342 m	0.0366 m
27.	Fabrication. Type	Chilled casting	Chilled casting
28.	C_s	1000	1000
29.	C_m	0.6256	0.6526
30.	C_v	0.2723	0.2621
31.	V_t	4.6143 m/s	4.9323 m/s
32.	W_{tg}	6156.7 N	6744.8 N
33.	α_n	20^0	20^0
34.	m	0.022	0.0211
35.	W_f	142.65 N	151.42 N
36.	φ_o	1682.4 W	1993.22 W
37.	φ_L	658.22 W	746.88 W
38.	φ	2340.64 W	2740.10 W
39.	η	0.711	0.727
40.	T_o	698.5 Nm	827.55 m

(i) determination of a programme structure (ii) Creation of a user friendly program interface (iii) Development of the codes (iv) Performance testing and tuning. Once the software development was completed, a full evaluation was conducted and validation of its results was performed by comparing it with those based on numerical procedures in literature. Features embedded in the functionality of the developed software algorithm include (i) On-design graphical simulation of a dynamic property. (ii) Two-way interactive and flexible capability.

4. VALIDATION OF SOFTWARE RESULTS

The sample design presented in Table 1 was repeated using the developed learning software. By comparing the values in Table 1 and those obtained from developed learning software, the results are correct and accurate. Shown in Figures 3, 4 and 5 are the results compared for the lifting and gearing designs

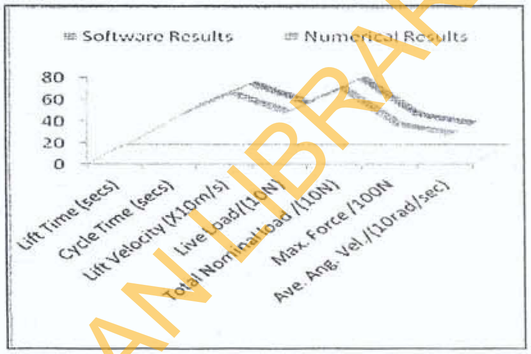

Fig.3. Validation of results for Lift Mechanism

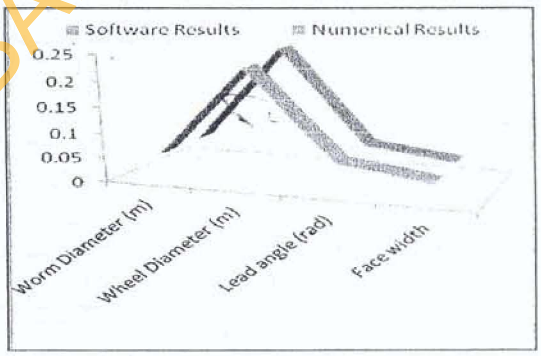

Fig.4. Validation of results for Worm gear design

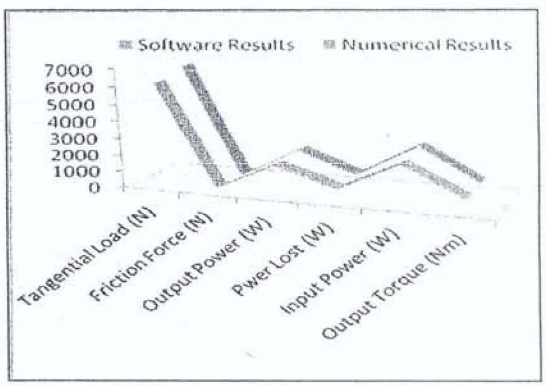

Fig.5. Validation of results for Power requirement

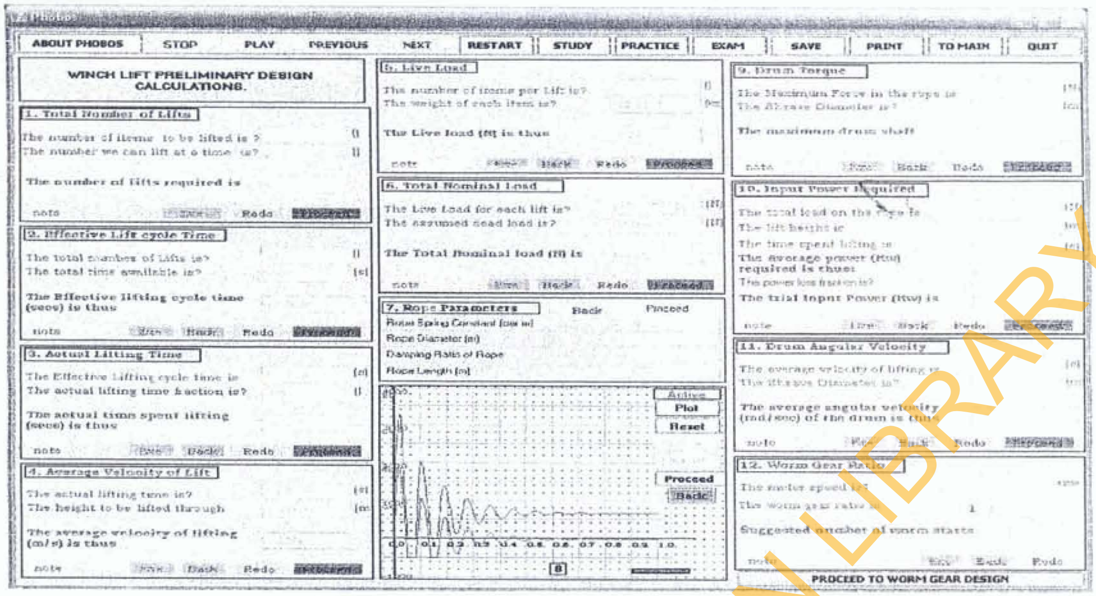

Fig.6. Interface for Design of lifting Mechanism showing the graphical display of dynamic force (----) and acceleration of load (-----) during design process.

5. CONCLUSION

In order to keep up with the modern trend in computer-aided engineering learning practice involving various aspects of machine designs, a dynamic learning application for motorized winch lift design has been developed in this work. The software with the aid of appropriate features takes the user into proper understanding of the dynamic loading and acceleration phenomenon by performing and displaying an on-design simulation of such properties as they relate to the design stages of the winch lift. The software proves reliable when compared with results obtained from numerical procedures. Through this work, further tasks are opened up on the various winch lift systems by extending the algorithm potentialities of the present learning software. This is because the underlying numerical design analysis for dynamic loading of the lifting system is here in captured for software design analyses and demonstration.

REFERENCES

[1] BILJANA, G., DESNICA, E., SUBIC, N., *New Trends in Technology And Higher Education, Engineering Technical Professions (E-Learning)* Machine Design, 2011, Vol.3, No.4, ISSN 1821-1259, pp. 301-304

[2] CHAPLIN, C., *Creativity in Engineering Design* Fellowship of Engineering Publishers, London, 1989.

[3] DHILLON, B. S., *Engineering Design: A modern Approach,* Richard Irwin, Homewood, Illinois 1996.

[4] FRENCH, J., *Invention and evolution: design in nature and engineering,* Cambridge University Press, United Kingdom, 1994.

[5] GEORGIA N., SOTIRIOS S., IOANNIS K., *A software tool for parametric design of turbo machinery blades.* Advances in Engineering Software, 2009, Vol. 40, No. 1, pp 41-51

[6] HONG Z., *Software Design Methodology,* Butterworth Heinemann, 2005.

[7] JON, B., MCCRACKEN, B., *The detail design phase,* SEED, Sharing Experience in Engineering Design, 1993.

[8] JON, B., SMITH, D., *The conceptual design phase,* SEED, Sharing Experience in Engineering Design, 1996.

[9] MAÍRA, M., OLIVIER, B., JAN, S., WIM, D., HENDRIK, V., *Computer-aided integrated design for machines with varying dynamics,* Mechanism and Machine Theory, 2009, Vol. 44, No. 9, pp 1733-1745

[10] MATTHEWS, C., *Case Studies in engineering design,* Arnold Press, Great Britain, 1998.

[11] MISCHKE, C., *Mathematical model building,* Iowa State University Press, 1980.

[12] NORTON, L., *Machine design, an integrated approach,* Prentice Hall, USA, 2006.

[13] PUGH, S., *Total design,* Addison Wesley, 1990.

[14] SHIGLEY, J.E. and MISCHKE, C.R., *Standard handbook of machine design,* McGraw Hill, New York, 1986.

[15] SIMOLOWO, E., BAMIRO, O., *Roller-Cams Systems Design: Development of a profile Analysis Software,* Pacific Journal of Science and Technology, 2009, Vol. 10 No. 1 pp 20-34

[16] SIMOLOWO, E., BAMIRO O., *Software Developed for Polynomial Cams,* The NSE. Technical Transaction. 2010, Vol. 45, No. 3 pp 69-84

[17] SIMOLOWO E., BAMIRO O., *The Development of An Analysis–Intensive Software for Improved Cam Systems Design,* Journal of Science and Technology, 2008, Vol. 28. No 1 pp. 103 – 115