IJCRT.ORG ISSN : 2320-2882

INTERNATIONAL JOURNAL OF CREATIVE RESEARCH THOUGHTS (IJCRT)

An International Open Access, Peer-reviewed, Refereed Journal

DESIGN ANALYSIS AND TESTING OF SHAFT MOUNTED SPEED REDUCER FOR COIL WINDING MACHINE.

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Abstract: The gearbox is a device which is used to transmit the power from one shaft to the other shaft within the required gear ratio. But, most of the times space limitation becomes the major problem of system. Also efficiency of the device is one of the important parameter. The device should have maximum efficiency. The conventional gear box gives us the required power and speed ratio but, they require the larger space for their working. Also they possess large number of parts and become bulky. In some applications, the space limitation is the important factor while designing the device. The aim of this project is to design a shaft mounted speed reducer which requires the less space and gives required speed ratio. In this thesis the analytical and numerical methods are used to design the parts of shaft mounted speed reducer. Also the experimentation is done to check validation of the work. The advantage of this project is, it requires less space and gives high efficiency. The maximum efficiency obtained by this speed reducer is 93.28%.

***Index Terms* - Timer Pulley, RH Sleeve, Internal Gear Ring**

I. INTRODUCTION

A Shaft mounted speed reducer is a device which is used to reduce speed of a machine from input speed to the required speed. In this device an internal external gear arrangement is used for speed reduction. The external gear is engaged with the internal gear but the external gear is eccentric with the internal gear. Because of such an arrangement reduction of speed can be achieved as per the requirement. We can change output speed by only changing the eccentric distance between the external gear and internal gear. The shaft mounted speed reducer is a small cylindrical unit that mounts directly on thedrive shaft and transmits power to the driven shaft (not shown) via a v -belt drive. Thecentre section on the speed reducer consists of a steel sleeve-A, internal gear –B and pinion-C. Internal gear –B is pressed into the steel sleeve whereas the pinion-C which is keyed to drive shaft-D meshes with the gear.

II. LITERATURE SURVEY

Govada Tejaswini & G. Chandra Mohan Reddy [1]: They compared present technologies of the gearbox and the result of their calculated efficiency. From the theory and results, it can be concluded that, com-pared to the conventional drives, the drives studied in this paper are highly efficient when compared to the conventional drives when designed for a diameter of 50 mm. As the inertia is quiet less in the initial stages of torque transmission, the efficiency will definitely be high. Using cycloid and harmonic drives, the gear reduction ratios of up to 1:6500 (max.) are attainable. Introduction of these compact drives help in reduction of power input to be supplied to the gear system thus, reducing the overall cost. Neither cycloid nor harmonic drives are universally superior for all applications and conditions. However, cycloid drives should be considered for applications in robots and aerospace applications, where the need for precision and applications of back drivability are important. The harmonic drives especially can be recommended for those applications in which size, inertia, and efficiency take precedence over backlash and torque ripple. From the comparison of various drives, they found that two stage cycloidal drives is having high efficiency (i.e. 92.7%) as compared to the other drives.

Padmanabhan.S., Chandrasekaran.M. and Srinivasa Raman. V [2] Gears are used in almost all mechanical devices and do numerous essential jobs and important it provides a gear reduction. This is vital to ensure that even though there is enough power there is also enough torque. Gear box has to produce maximum power with minimum weight. In many real-life problems, objectives under consideration conflict with each other, and optimizing a particular solution with respect to a single objective can result in unacceptable results with respect to the other objectives. Multi-objective formulations are realistic models for many complex engineering optimization problems. A reasonable solution to a multi-objective problem is to investigate a set of solutions, each of which satisfies the objectives at an acceptable level without being dominated by any other solution. In this paper, Ant Colony Optimization was developed specifically for a Worm gear drive problem with multiple objectives. Within the various design variables available for a worm and worm wheel design, the power, weight, efficiency and centre distance have been considered as objective functions and bending stress, compressive stress as vital constraints to get an efficient compact and high power transmitting drive.

Chiu-Fan Hsieh[3]: They proposed a new transmission design for an eccentric speed reducer for which the internal and external gear profiles are constructed via a gear mathematical model and stress tested using a system dynamics analysis model. They proposed a design for eccentric speed reducers that differs from that used with a traditional cycloid speed reducer. The main difference, other than the input and output shafts, is that it uses the internal gear as its fixed part and transmission between the external gear and output shaft occurs via pins connected to a drive plate. In this paper, gearing theory is used to construct a mathematical model of the involute external and internal gears, based on which the trajectory equations, component geometry, and reducer kinematics can be derived. A dynamics analysis model is also constructed and used to test the feasibility of two types of drive plate designs a cross piece and a round disc, each with a single-gear and double-gear design. The stress results show the infeasibility of the cross drive plate versus the feasibility of the round disc drive plate. An additional kinematic analysis provides evidence that the design of the gear tooth profile also affects the machine transmission and can lead to vibration and stress fluctuation. This stress analysis indicates that when a cross piece is used as a drive plate, the cross piece groove and pins operate in a straight-line tangent movement that is likely to produce yield and result in stress that exceeds the yield strength of the material. As a result, the reducer parts will soon be damaged. If the drive plate is changed to a round disc design, however, the pins and holes operate in a circular tangent movement, eliminating the yield problem. The stress variation is also well below the yield strength. Comparison between the single-gear round disc and the double-gear round disc further shows that the double-gear round disc design can reach dynamic balance with even stress distribution, thereby reducing fluctuation and lowering the stress value to well under the yield strength. This design, therefore, is highly feasible. A final kinematic analysis, based on a comparison of the theoretical and simulation calculations, provides clear evidence that the design of the gear tooth profile has a significant effect on the machine transmission and can lead to vibration and stress fluctuation.

Wan-Sung Lin a, Yi-Pei Shih b, Jyh-Jone Lee a [4]: The cycloidal gear reducer is a compact, high-ratio, and low-backlash speed reduction device. It has been commonly used for transmitting motion and torque in machinery. They proposed the design of a new two-stage cycloidal speed reducer with tooth modifications. The topological structure of cycloidal drives have been discussed and analyzed with the aid of graphs. New cycloidal gear reducers are enumerated through the topological analysis and a new two-stage cycloidal gear reducer with simpler structure is then proposed. The design of the proposed cycloidal gear reducer is also performed, including profile generation and modifications. Subsequently, kinematic errors are analyzed by using the tooth contact analysis, and the results caused by different combinations of the gear profile modifications have been presented quantitatively. Finally, based on the analysis, a mock-up of the cycloidal gear drive is constructed to validate the feasibility of the new mechanism. The topological analysis provides the designer more realization of the characteristics of the cycloidal speed reducers. Two new structures of two-stage cycloidal speed reducers are then enumerated after topological analysis. A new configuration with the least number of links and joints is chosen to establish the design procedure and a mock-up is fabricated to validate the feasibility of this type of two-stage cycloidal drive. The angular velocity analysis of the gear drive is straightforwardly obtained through the help of graph representation. Modifications of the cycloidal gear profile and transmission errors are analysed quantitatively rather than qualitatively. It is shown that a smaller quantity of modification in a single type of modification yields smaller kinematic errors. However, compound modifications may yield smaller kinematic errors than a single type of modification. Further, if variables for compound modification are appropriately selected, a larger value of modification can still acquire smaller kinematic errors. Based on the results obtained, this new cycloidal drive is realizable as a compact, high speed reduction ratio, and high accuracy speed reduction device.

Hong-Sen Yan & Ta-Shi Lai [5] proposed Geometry design of an elementary planetary gear train with cylindrical tooth-profiles. They present a concept of elementary gear trains such that the tooth-profiles of the pinion are cylindrical. A mechanism is termed a planetary mechanism if it contains at least one rigid member that is required to rotate about another axis. For examples, planetary gear trains, Ferguson Hi-Range speed reducers, and cycloid drives are planetary mechanisms. Planetary gear trains are compact, light-weight devices capable of producing high speed reduction as well as high mechanical advantage in a single stage. They are widely used in speed reduction or transmission devices. The cycloid drive is more compact, light-weight devices capable of producing high speed reduction than planetary gear trains as well as high mechanical advantage in a single stage. Above, it has high precision pointing, so that it is an attractive candidate for many applications today.

Bernd-Robert Hohn et.al,[6]Although mechanical gearboxes used as torque and speed converters have already very high efficiency it is not only a task in automotive applications to further decrease gearbox power losses but also in many industrial applications. Different methods are discussed for power loss reduction in a gearbox. No load losses can be reduced, especially at low temperatures and part load conditions when using low viscosity oils with a high viscosity index and low oil immersion depth of the components. This in turn influences the cooling properties in the gear and bearing meshes. Bearing systems can be optimized when using more efficient systems than cross loading arrangements with high preload. Low loss gears can contribute substantially to load dependent power loss reduction in the gear mesh. Low friction oils are available for further reduction of gear and bearing mesh losses. All in all a reduction of the gearbox losses in average of 50 % is technically feasible. The challenge is substantial power loss reduction with only minor impact on load carrying capacity, component size and weight and noise generation.

Todd R. Bobak[7]:In the power transmission industry, shaft mounted speed reducers provides one possible Solution to meet the speed reduction/power generation needs for an application. As the name implies, a shaft-mounted speed reducer (SMSR) is a speed reducer mounted directly onto, and statically supported by, a driven shaft—the type of setup that may be found on a conveyor. Typically, the SMSR incorporates a hollow bore (with keyway, tapered bushing or some other coupling mechanism) to facilitate mounting onto the driven shaft. The SMSR may be ―fixed‖ to the machine using an output flange or, in certain instances; its housing may be bolted directly onto the machine. However, situations exist where direct attachment of the SMSR is either not possible or desirable. In such situations, the SMSR is supported only by the shaft that it is intended to drive. When the SMSR is mounted directly onto a driven shaft with no other external support, it must have a torque arm attached to it. A torque arm is a pivoted; connecting link between the reducer and a fixed anchor point intended to resist the torque developed by the reducer. Quite simply, a torque arm transmits the reaction torque produced by the SMSR into the structure of the machine, thereby preventing the counter-rotation of an SMSR during operation. Most manufacturers of shaft mounted speed reducers have designed, and offer for purchase, standard torque arms for their products. Situations may exist, however, where a manufacturer‘s torque arm does not meet the needs of a certain application (due to space limitations, for example). In such situations, the machine designer may be required to design his or her own torque arm to fi t within the constraints of the application. This article provides some design guidelines for such a situation. Although seemingly simple in concept, a torque arm is an important component when considering a shaft-mounted speed reducer for an application. Before selecting a potential torque arm design, designers should evaluate the offering supplied by the manufacturer of the shaft-mounted speed reducer. In doing so, they may determine that the standard offering fits into the application constraints.

M. Chandrasekaran et.al.[8]Gears are used in almost all mechanical devices and they do several important jobs, but most important, they provide a gear reduction. This is vital to ensure that even though there is enough power there is also enough torque. Gear box has to produce maximum power with minimum weight. In many real-life problems, objectives under consideration conflict with each other, and optimizing a particular solution with respect to a single objective can result in unacceptable results with respect to the other objectives. Multi-objective formulations are realistic models for many complex engineering optimization problems. A reasonable solution to a multi-objective problem is to investigate a set of solutions, each of which satisfies the objectives at an acceptable level without being dominated by any other solution. The genetic algorithms developed specifically for a single speed gear box problem with multiple objectives.

David W. Pessen [9] invented a quick release mechanism for self-locking mating worms. This invention is modification of a self-locking mating worm drive. The modification consists of a nut mounted on one of the worm shafts with a very large pitch angle, such that it is not self-locking. This nut keeps the two worms in mesh, and axial motion of the nut is prevented by a tapered key. When this key is removed, the axial pressure of the worm pushes the nut sideways, until the two worms disengage. The self-locking property of the worm drive can thus be removed at will.

This invention relates to a release mechanism for self-locking mating worms. An objective of the invention was to provide a simple mechanism whereby the self- locking characteristic of the mating worms can be quickly removed when so desired. A further objective was to easy reinstatement of the self-locking characteristic, to permit lowering of a load in the case of mating worms with so called second order self-locking which ordinarily would not permit lowering of the load.

Bingkui Chen, HuiZhong, JingyaLiu, Chaoyang Li and Tingting Fang [10]: In this paper,Bingkui Chen et.al presented a new cycloid drive with double contact lines between one tooth pair. The new conjugated tooth profile has been generated by applying double-enveloping gear theory in cycloid drives. Based on coordinate transformation and gear geometry theory, the meshing equation for this new cycloid drive has been established at first, and then the equation of tooth profile and meshing line and the formula of induced normal curvature are also derived. The meshing characteristics have been investigated by theoretical analysis and numerical examples. The double contact characteristic is revealed by that the meshing function can be resolved into two independent factors. The contact lines at different instant and meshing line are illustrated based on a numerical example. The superior characteristics of the new conjugate tooth profiles have been represented by comparison of induced normal curvature with conventional cycloid drives. The physical prototype has been trial-produced and the transmission error of the prototype has also tested, which shows that double-enveloping theory can be applied for cycloid drives. The meshing function can be resolved into two independent factors, which reveal the double contact characteristic of new cycloid drive. Corresponding to the two independent factors, the new double-enveloping tooth profile is composed by two portions. One portion is the same as the part of original cylindrical pin tooth profile which takes part in meshing in conventional cycloid drive, the other is the new tooth profile generated in the second enveloping motion, the two portions are tangent with each other at the meshing limit line of the original cylindrical pin tooth, and the tangent line is also the limit line which determines the points on double-enveloping tooth profile that can take part in meshing once or twice. The physical prototype is trial- produced and the transmission error of the prototype is also tested. The transmission error of the prototype is 41 s, which is relatively high transmission precision. The success of trial production of the physical prototype shows that double-enveloping theory can be applied for cycloid drives.

III. RESEARCH METHODOLOGY

A. Problem Statement

Worm gear boxes are popular choices for high speed reduction in many cases where there is limitation on space for power transmission. The worm gear box though gives high transmission ratios in small space the inhereent disadvantage is of very low transmission efficiency. The efficiency of worm gear depends on the coifficient of friction and lead angle. In order to obtain a worm gear with high efficiency it is recommended to use the lead angle in the range between 100 to 300. But to obtain a self locking worm gear the efficiency should be less than 50% and the respective lead angle will be 10 to 50. This causes the effincency to drop below 50% but results in a large energy loss and intesive heating of the worm gear. Self locking worm gears can operate only under small losses and interval work periods. Also the worm gearbox can not be suitable for haigh speeds and long working periods. Thus we can conclude that if worm gear drives when used for lifting applications with self-locking as the primary objective for safety considerations the drives are extremely in-efficient. Hence there is a need of special purpose drive that will provide better transmission efficiency in self-locking condition so as to reduce power consume by the device i.e. lowering the running cost of device.

B. Need Of The Project

The selection of transmission system for a variable speed transmission shall be governed by the following factors.

1. Ability to deliver reduced speed ratio.
2. Space available.
3. Service conditions.
4. Compactness of drive.
5. Speed ratio; etc.
6. Cost of drive.

- It is possible to change speed ratio using a belt drive using step pulley arrangement;
  a) But it will again be limited to the corresponding ratio of diameters on the steps in either of the drive pulleys.
- More over as the range of speeds needed increases the no. of steps on pulley will also have to be increased. This makes the drive large, bulky & uneconomical. Hence it is not possible to use the belt drive alone.
  c) Another difficult is that it is not possible to shift form one speed to another while machine is in running condition.
  d) So also any intermediate speed between that provided by the step pulleys cannot be achieved.

The above points prove the limitations of belt drive being put to limited in the variable speed transmission.

- Chain drives can be used for the transmission but again the speed ratio depends upon the ratio of no of father driver and driven sprockets which will remain fixed for a given application. This sprocket arrangement is not capable to transmit any other speed ratio. Hence these chain drives because of their above limitation & size, bulk & cost cannot be used.
- Gear drives when arranged in the form of gear train are capable to transmit different ratio of speeds such arrangements are called ‗Gear Box'.

  Gear boxes can be single stage (speed) or multistage (speed). The different Speeds can be obtained by shifting the gear or change or gear shift lever. But the different speeds available from the gear box are dependent on the number of teeth on the gears in the gear train; which will remain fixed once the gear box is constructed. This limits the variable speeds that can be obtained from a particular gear box.

More over as the no. of stages is increased the size of the gear box and correspondingly the cost of the gear box. These factors thus influence the use of gear box for a particular variable speed transmission.

Considering the above factors; it is clear that there is need of a special transmission unit having following qualities:-

a) Compactness of system.
b) Low weight.
c) Low cost.
d) Easy mounting
e) Change in ratio of speeds with minimum modification.

So this shaft mounted speed reducer can be used for the machines which require the above requirements.

C. Objective

- Design of internal external gear mechanism for minimum space spot reduction.
- Design of internal external gear reduction gear for optimal efficiency for reduced power consumption.
- Design of various parts of speed reducer such as :
  1. Input shaft
  2. LH –Sleeve
  3. Input timer pulley
  4. RH-Sleeve
  5. External gear
  6. Internal gear
  7. Output shaft
- Mechanical design of above components using theoretical theories of failure after selection of appropriate materials
- 1. 3-D modelling of set-up using Creo-3.0
- 2. CAE of critical component and meshing using Ansys 14.5 i.e. the pre- processing part.
- 3. Mechanical design validation using ANSYS 14.5 critical components of the system will be designed and validated
- 4. Validation of strength calculations of critical components using ANSYS 14.5
  i.e. the post processing.
- Creation of Prototype:
  The selected mechanism and machine along with the damper will be designed using following machines:
  1. Centre lathe
  2. Milling machine
- 3. DRO – Jig Boring machine
- Electrical Arc Welding
- Experimental validation :
  The experimental validation part of the lifting force developed by the system is validated using test-rig developed.
  Following characteristics will be plotted
  1. Torque Vs. Speed
  2. Power Vs. speed
  3. Power consumption of motor under rated load.
  4.Efficiency of system Vs. speed
-

IV. WORKING

In experimental method, the model of shaft mounted speed reducer is manufactured using various machines (i.e. Lathe, milling, Centre lathe Milling machine, DRO – Jig Boring machine, Electrical Arc Welding.). The model is tested with the test rig developed i.e. rope brake dynamometer. Following test are conducted during experimentation.

a) Torque Vs. Speed Characteristics

b) Power Vs. Speed Characteristics

c) Efficiency Vs. Speed Characteristics

In order to conduct trial, a dynobrake pulley cord, weight pan are provided on the output shaft.

1) Procedure:-

1) Start motor
2) Let mechanism run & stabilize at certain speed (say 1440 rpm)
3) Place the pulley cord on dyno-brake pulley and add 1 KG weight into the pan, note down the output speed for this load by means of tachometer.
4) Add another 2KG cut & take reading.
5) Tabulate the readings in the observation table
6) Effective Radius of dyno-brake pulley = 30 mm
7) Plot Torque vs. speed characteristic and Power vs. speed characteristic.

• Apparatus required for experiment:

Rope-brake dynamometer, weight hanger, dead weights, tachometer.

The observations taken during the experiment are shown in table no. 8. For experimental method it is necessary to take only the reading of weight and speed. So, only these readings are measured and tabulated in the following table.

Table 8: Observations of experiment

Sr. No.	Weight (Kg)	Speed Rpm
1	1	1100
2	2	1090
3	3	1075
4	4	1062
5	4.5	1040

After taking the readings, the results are obtained by using the required formulas. The table no.9 shows the result of experiment. From the obtained observations the torque and power are obtained and then the efficiency is calculated by taking the ratio of output and input power.

Table 9: Results of experiment

Sr. No.	Load (Kg)	Speed (Rpm)	Torque (N-m)	Power (Watt)	Efficiency (%)
1.	1	1100	0.2943	33.90	18.33
2.	2	1090	0.5886	67.19	36.32
3.	3	1075	0.8829	99.40	53.73
4.	4	1062	1.1772	130.93	70.77
5.	4.5	1040	1.32435	144.25	77.97
6.	5.0	1032	1.4715	159.05	85.97
7.	5.5	1018	1.61865	172.58	93.28

2) Experimental validation:

The experimental validation part of the lifting force developed by the system is validated using test-rig developed. Following characteristics will be plotted

a) Torque Vs. Speed
b) Power Vs. speed
c) Power consumption of motor under rated load.
d) Efficiency of system Vs. speed

I have done the experimentation on the set up developed. The above photo was clicked during the experimentation. By calculating the results of experiment, the following graphs are plotted using Microsoft excel.

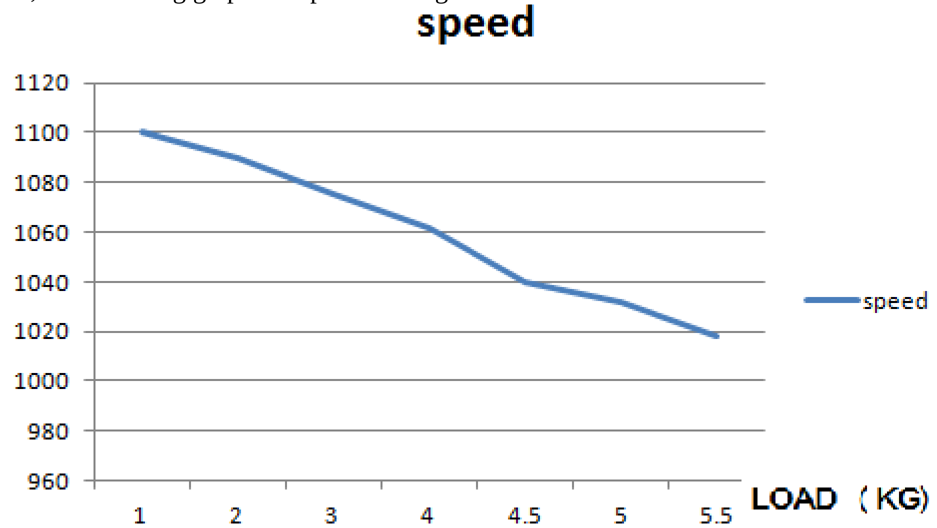

Figure 1: Graph of load vs. speed.

Figure shows the graph of load vs. speed. From the graph it can be observed that speed is inversely proportional to the load. That is as load increases, the speed reduces by respective amount.

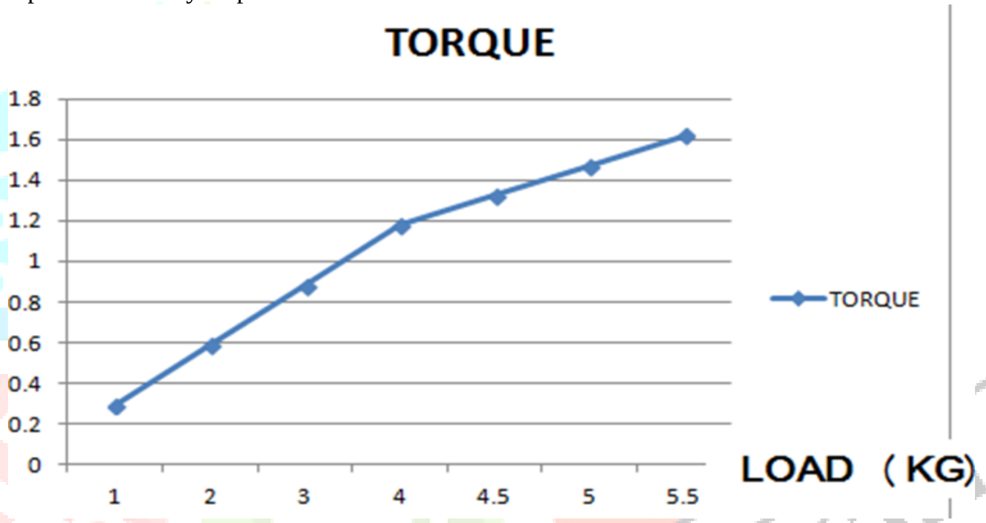

Figure 2: Graph of load vs. speed.

Figure shows the graph of load vs. Torque. From the graph it can be observed that torque is directly proportional to the load. That is as load increases, the torque also increases by respective amount.

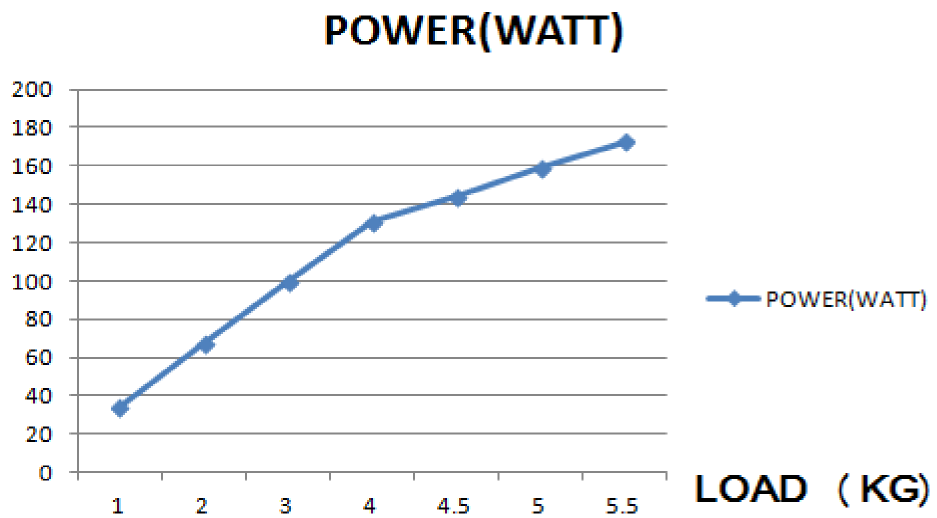

Figure 3: Graph of load vs. power.

Figure 49 shows the graph of load vs. power. From the graph it can be observed that power is directly proportional to the load. That is as load increases, the power also increases by respective amount.

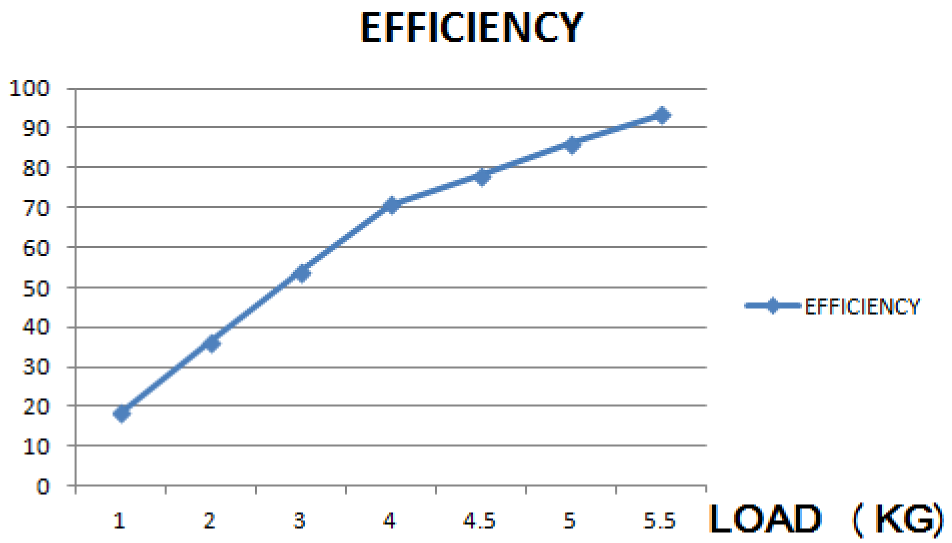

Figure 4: Graph of load vs. efficiency.

Figure shows the graph of load vs. efficiency. From the graph it can be observed that efficiency is directly proportional to the load. That is as load increases, the efficiency also increases by respective amount. The maximum efficiency obtained is 93.28%.

3) Details of model manufactured:

Figure 2: Photograph of model

Figure no.5 shows the model of shaft mounted speed reducer. The detailed bill of material is mentioned in table no.10 and the costs required for manufacturing also mentioned below

V. RESULT

Critical parts are validated by analytical calculations as well as FEA i.e. numerical methods. Following table shows comparison of analytical & numerical stress results induced in critical parts of shaft mounted speed reducer.

1. Maximum stress for input shaft calculated by theoretical method and Numerical method are well below the allowable limit. The percentage error between two results is 0.49 %. This error is very small. Also Input shaft shows 2300×10-6mm deformation which is negligible; hence the input shaft is safe.
2. Maximum stress for LH- Sleeve calculated by theoretical method and Von-mises stress are well below the allowable limit. The percentage error between two results is 9.63 % which is considerable. But this error is less than 10%, so it can be neglected. Both the values of stresses are well below the allowable stress; also the LH sleeve shows 4.00×10-6mm deformation which is very negligible. Hence the LH sleeve is safe.
3. Maximum stress for Timer pulley calculated by theoretical method and Numerical method are well below the allowable limit. The percentage error between two results is 7.91%. This error is very low. Also Timer pulley shows 2.37×10-6mm deformation which is very negligible; hence the timer pulley is safe
4. Maximum stress for RH-Sleeve calculated by theoretical method and Numerical method are well below the allowable limit. The percentage error between two results is 7.88%. This error is less than 10% and can be

accepted. Also RH-sleeve shows 7.77×10-6mm deformation which is very negligible; hence the RH Sleeve is safe.

5. Maximum stress for External gear calculated by theoretical method and Numerical method are well below the allowable limit. The percentage error between two results is 6%. This error is also less than 10% so can be acceptable. Also External gear shows 8.4 ×10-6 mm deformation which is negligible; hence the External gear is safe.
6. Maximum stress for internal gear calculated by theoretical method and Numerical method are well below the allowable limit. The percentage error between two results is 5.75%. This error is very low. Also internal gear shows 2.884 ×10-6mm deformation which is very negligible; hence the internal gear is safe.
7. Maximum stress for Output shaft calculated by theoretical method and Numerical method are well below the allowable limit. The percentage error between two results is 3.89%. This error is very small. Also Input shaft shows 3400×10-6mm deformation which is negligible; hence the input shaft is safe.
8. Stresses calculated by both the methods for the components are well below the allowable stress. So design of all the components is safe.
9. The device can give the maximum efficiency up to 93.28%.
10. The results obtained from both i.e. theoretical and numerical method are nearly similar for all the parts and have a percentage error less than 10%.

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Table 3: Comparison of analytical & Numerical stress results

Sr. No.	**Name of Part**	**Max. Allowabl e Stress** (N/mm^2)	**Comparison parameter**		**%Erro r**	**Total Deformatio n (mm)**
			Analyticall y calculated stress (N/mm^2)	**FEA Obtained stress (Equivalen t Stress)** (N/mm^2)		
1	Input Shaft	144	✓ 5.0750	✓ 5.1000	0.49	2300×10^{-6}
2	LH-sleeve	144	✓ 0.0244	✓ 0.0270	9.63	4.000×10^{-6}
3	Timer pulley	144	✓ 0.0664	✓ 0.0721	7.91	2.370×10^{-6}
4	RH-sleeve	144	✓ 0.0608	✓ 0.0660	7.88	7.770×10^{-6}
5	External gear	144	✓ 0.5640	✓ 0.6000	6.00	8.400×10^{-6}
6	Internal gear	95	✓ 0.0754	✓ 0.0800	5.75	2.884×10^{-6}
7	Output shaft	144	✓ 5.4013	✓ 5.6200	3.89	3400×10^{-6}

VII CONCLUSION.

1] The analysis of all the parts has been carried out using two methods i.e. analytical method and numerical (FEA) method. The results obtained from both the methods are well below the allowable stress. So the design is safe and hence device is safe.
2] The device exhibits reduction of speed from 1400 to 1100 with no slip at moderated load condition
3] The device exhibits maximum efficiency of 93.28%
4] The device gives maximum torque of 1.62 N-m.
5] The device can thus safely handle power of 185 watt necessary for coil winding application
6] Device exhibits increase in transmission efficiency with increase in load with marginal drop in speed, maximum efficiency being 93.28%
7] Device is modular and can be used for increased ratio of transmission upto 900 rpm with slight modification in LH sleeve.

ACKNOLEDGEMENT.

I would like to express my gratitude, sincere thanks and deep sense of appreciation to my project guide **Dr. S. S. Patil**, Head Of The Department, Department of Mechanical Engineering, SVPM's College of Engineering, Malegaon (Bk.) for continuously assessing my work, providing great and valuable guidance.

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