

DESIGN AND MANUFACTURING OF SELF ACTUATING TRACTION DRIVES WITH SOLID AND HOLLOW ROLLERS

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ABSTRACT

The friction drive speed reducer proposed by Flugrad and Qamhiyah in 2005 was mainly investigated in this paper. That self actuating traction drive uses six intermediate cylindrical rollers to transmit motion. Those rollers fail by fatigue. So, this research built a numerical simulation model to find the optimum size of those rollers which give the least contact stresses and so the longest fatigue life. Then those rollers were replaced by hollow ones. The numerical simulation results showed that the contact stresses values decreased in case of having the rollers hollow, which means longer fatigue lives of those rollers. The hollow rollers were found to live more than 30 times the solid ones under same loading conditions.

To validate the simulation results and practical application of the proposed model by Flugrad and Qamhiyah, two traction drives were manufactured from aluminum with the optimum dimensions found numerically. The hollow rollers were manufactured with 60% percentage of hollowness, that the ratio of the internal diameter to the external diameter was 60%. The hollowness was filled with rubber which has very low modulus of elasticity, which is very close to represent the hollow space. The two friction drive models were experimentally tested and could successfully transmit motion and reduce speed.

KEY WORDS

Fatigue life, Hollow rollers, Traction drive speed reducer.

1. INTRODUCTION

Traction drives with cylindrical rollers have high radial load capacity. They contain rollers that are cylindrical in shape. Machine elements in concentrated rolling motion (rolling element bearings) may fail for a variety of causes, like wear, what is known as galling failure, localized plastic deformation, over loading and overheating. Most or all of these causes may be avoided or reduced by careful system design and manufacture. Even so, the system may then fail eventually by fatigue. In rolling element bearings, the fatigue occurs where there are rolling contacts that suffer from repeated loading. To date, there is no proper examination of the damage in the rolling contacts. It is just known that repeated stressing of the machine elements causes irreversible changes in them, which results in the formation of cracks. These cracks occur after a random number of load cycles based on the stress distribution inside the machine element. The sign of the crack propagation

is formation of some damage in the contact surface known as pitting or spalling [2]. Since the stress distribution within the machine element is a main factor of determining fatigue life, some design changes should be made to redistribute the contact stresses of the rolling elements. Making rollers hollow might be a good technique to redistribute those contact stresses.

Using hollow rollers in the roller bearings had the interest of some designers because of their advantages over solid rollers. These include reducing the material used in making the rollers, less weight for the roller bearing, and the ability to preload the hollow roller element, giving it more stability with less noise and vibration. Hollow roller bearings are single or double row radial bearings with an inner ring, outer ring and hollow or thin wall rollers. In traction drives, two or more sets of rollers are used in contact between the inner race and the outer ring. They utilize traction or friction to transmit torque and power. Ai [3,4] indicated that traction drives have unique characteristics which are not present in gears. Like the high mechanical efficiency, little or no backlash, low noise and vibration. In friction drives, the power is transmitted between the contacted surfaces using Coulomb friction. No lubricant is used in friction drives. A good example of traction drives using cylindrical roller bearings is the self actuating traction drive designed by Flugrad and Qamhiyah [4]. This friction drive is used as a basis for this work for testing the fatigue life of hollow rollers in traction drives. Even so, the results can be generalized for cylindrical rollers used in roller bearings and other traction drives.

Ai [3, 4] developed planetary traction drive transmission with zero spin. Traction drives with their two types; the fixed ration transmission (FRT) and the continuous variable speed (CVT) have the problem of the spin motion that causes rotational sliding between the contacted surfaces about an axis normal to the surfaces and thus must be oil-lubricated [3]. The spin motion causes power loss and component wear. The zero spin design of Ai [3,4] improved the mechanical efficiency of the traction drives when operated with lubricant and running dry. He found that lubricant churning-loss is an important power sink. So, traction drives running dry might have higher mechanical efficiency than running with lubricant.

Using traction drives has many advantages over gears, such as less noise, easier to manufacture, and easier and cheaper to maintain. On the other hand, one of the main disadvantages of using the traction drives over gears is its weight. For the same load application, the required traction drive is heavier than the

required gear system. So, this paper interest is to find a solution for the two problems of the cylindrical rollers; their fatigue life and their heavy weight. A solution for both these problems can be found by using hollow rollers instead of solid rollers in the traction drives. Using the hollow rollers means less weight. And so the problems are partially solved. This works investigating the fatigue life of the traction drive compared to traction drive with solid rollers.

In Abu Jadayil [5], numerical simulations of two rollers under pure rolling contact with normal and tangential loads, have been made to study the fatigue life of hollow rollers, and to compare it to solid rollers. The results of that work showed that hollow rollers have longer fatigue life than solid ones. The optimum percentage of hollowness with the longest fatigue life was found to be around 50% in case of combined normal and tangential loading.

This paper is trying to theoretically and experimentally verify the results achieved by Abu Jadayil [5] by building a numerical simulation models for the self actuation traction drive proposed by Flugrad and Qamhiyah [1], a model with solid rollers and another with hollow rollers. Then the manufactured traction drives with optimum dimensions were tested experimentally to investigate their rollers' fatigue lives.

2. LITERATURE REVIEW

The distribution of the tangential loading in the area of contact was studied by Mindlin in 1949 [6]. He assumed one elastic body sliding over the other, and he assumed the tangential loading value at the point of contact did not exceed the product of the coefficient of friction between the two bodies and the normal loading value. The value of the coefficient of friction he used in his analysis was 0.33. He found that the tangential loads generate infinite stress at the boundaries of the contact area. In the same year, Poritsky [7] obtained a solution using two different methods for the same problem, but with coefficient of friction of 0.3.

Smith and Liu [8] extended the solution obtained by Poritsky [7] to consider the effects of these stresses in causing failure by inelastic yielding and fatigue. They assumed the Hertzian distribution of the normal and tangential loads over the area of contact. The resulting stresses of applying the normal and tangential loads were represented in a closed form. A coefficient of friction of 0.33 as a proportionality factor between the tangential and normal loads was used.

The interest in using hollow rollers in bearings started in the 19th century. Many patents were issued for different designs of hollow roller bearings. In 1897 Miller [9] received a patent for a hollow roller bearing in which the rollers were formed of spirally wound strips of metal. Eight years later Fownes [10] patented a thin walled hollow roller bearing. In his invention he tried to provide more flexibility and soften the shocks the vehicle was subjected to on rough roads. In the next year a more flexible roller bearing was invented by Canre [11]. He used slots in the walls of his thin walled tabular rollers. After that many designers received many patents for new hollow

roller designs, like Lockwood [12], Steenstrup [13] and Lockwood [14]. One of the best designs for hollow roller bearings was made by Steffenini [15] in 1947. He discussed the use of preloaded hollow rollers, including cylindrical, tapered, wrapped and layered rollers with complete and partial hollowness.

Many researchers started to discuss the advantages and disadvantages of hollow rollers. Hanau [16] described the advantages of roller bearings with over 50 percent hollowness in high speed applications. Given [17] pointed out the advantages of hollow roller bearings in driving turbine shafts at the required speed. Harris and Aaronson [18] utilized this advantage in his patent with 60 to 80 percent hollowness. In his design, he increased the hollowness to be able to increase the preloading and so reduce the roller skidding.

Zaretsky [19] reported many experimental tests made between 1967 and 1978 to investigate the fatigue life of hollow balls, hollow rollers and cylindrical hollow balls. He indicated that calculations made by Harris in 1968 [20] showed that using hollow balls in the bearings causes the reduction of the centrifugal loading as a result of reducing the mass of the rolling element. He believes that to have significant effect on the bearing fatigue life, the mass should be reduced at least 50% by using hollow rolling elements.

Bowen utilized the advantages of preloading the hollow rollers and their superior characteristics in his patents in 1976 [21] and 1977 [22]. Many unique characteristics of the hollow roller were discussed by Bowen and Bhateja [23]. They noted that in low speed hollow rollers, the limiting factor of the fatigue life is the contact stresses. These stresses can be reduced by increasing the number of the preloaded hollow rollers sharing the load. At high speeds, Bowen and Bhateja thought the lighter weight of the rollers and the complete preloading and roller's compliant nature, all tend to increase the contact surface fatigue life. So the bending stresses, especially those happening at the bore surface would be the fatigue life limiting factor. Bowen and Bhateja [23] experimentally investigated the 50 to 80 percent hollowness. They found that rollers with a hollowness range from 60 to 70 percent are the best to use for most applications.

The fatigue life of the hollow roller bearings compared to solid roller bearings has been studied by many researchers. The general trend has been to agree that the fatigue life is improved in the case of hollow roller bearings. However, they used different methods to prove that.

The contact problem of hollow cylinders was analyzed by Hong and Jianjun [24]. They used three hollowness percentages in their analysis; 50%, 60% and 70%. As defined earlier for the hollow rollers, the hollowness percentage is the ratio of the diameter of the hole to the outer diameter of the cylinder. In their experimental results, they found that the contact stresses of 50% hollow cylinders are 33%-35% less than contact stresses of solid cylinders. That was related to the increase of the contact area of hollow cylinders. In their theoretical analysis they found that in rollers with 50% hollowness, most of the deformation occurs at the contact region, and the amount

of bending deformation is very small. This shows why they behave like solid cylinders. Hong and Jianjun [24] concluded that when hollowness is less than 70%, fatigue life of contacting bearings can be improved. That is not related to the change in the contact width, but to the decrease of the contact stresses.

Since running rollers without lubricant increases the mechanical efficiency as indicated by Ai [2,3], numerical simulation in this work assumes no lubricant present between the two rollers in contact. Moreover, running the rollers dry increases the fatigue life as demonstrated by Way [25].

3. TRACTION DRIVE SELECTION

Most speed reducer in use today utilizes gears to produce an output speed different than the input speed. There are, however, other types of speed reducers in use, including traction drive speed reducer. These depend on friction between rolling elements to transmit torque from the input member to the output member. The rolling elements are held together with a prescribed normal force to generate the required friction force based on the power to be transmitted by the device. However, these devices are not self-actuating. Further, these devices often require a separate clutch to allow the output to be disengaged from the input. It is therefore a principal object of this invention proposed by Flugrad and Qamhiyah [1] is to provide a traction drive speed reducer which is self-actuating. A further object of this invention is to provide a self-actuating traction drive speed reducer wherein the normal force on the roller members is only present when needed to permit the rolling elements to be operationally disengaged. A still further object of this invention is to provide a self-actuating traction drive speed reducer which can be easily engaged and disengaged in response to changing speed requirements.

The speed reducer proposed here is designed so that the configuration of the rolling elements creates the needed normal force in response to the torque exerted back on the system by the downstream loading. Thus the device is self-actuating. Since the normal force is only present when needed, the rolling elements of the device can readily be disengaged, thus eliminating the need for a separate clutch in the drive system. This feature can be exploited to design a transmission with several distinct speed ratios which can be engaged and disengaged in response to changing speed requirements.

Many traction drives were developed by researchers. A cone roller toroidal traction drive was developed by Tanaka in 1988 [26]. In that work the a traction drive CVT of toroidal rolling elements changes its speed ratio by the control of the attitude angle of the power roller of which control force is induced from a little side slip. Design scheme for multidisk Beier traction variators was produced by Younes in 1992 [27]. The unique design feature of the Beier traction variator is that it can provide a higher power capacity per unit volume by using multiple thin disk pairs in a planetary system. Compared to these traction drive designs, the self actuating traction drive of Flugrad and Qamhiyah [4] has unqiues charcateristics; mainly being self actuating, that rollers do not need to be in contact all

the time. That would result in significant increase in the fatigue life of the rollers. The list of products in which this new self-actuating traction drive speed reducer might be used is seemingly endless. Currently, gear-driven units are used for many of these applications. Since this new traction drive design consists of cylindrical-shaped rollers rather than complex shaped gear teeth, it will be simpler to manufacture, easier to assemble, and will run quieter.

4. PROBLEM STATEMENT AND SOLUTION TECHNIQUE

4.1. Problem Statement. Much of the research has been centered on the use of hollow rollers in roller bearings. Research in this area has shown hollow rollers to have advantages in accuracy of rotation and stiffness, even at high speeds. A related area of interest is the use of hollow rollers in pure rolling contact with another roller. One main advantage of using hollow rollers in a roller bearing is the additional sharing of load between rollers as the rollers deflect more than solid rollers do under the same load, the reduction in stress is seen when the area of contact between the rollers expands under load.

One of the main disadvantages of friction drive systems compared to gear drive systems is the size required. The size of a friction drive system must be larger to account for the stress induced due to the normal force required to prevent slip. Using hollow rollers in a friction drive system can decrease the stresses in the rollers, thereby allowing smaller rollers to be used. Since the need for all inherent advantages in traction drive speed reducer instead of geared one, the configuration must be studied very well, installed and assembled in such away that all these benefits and improvements in the design can be implemented and provided.

The study here will discuss every improvement in the traction drive, how can we implement it and the needed design and configuration in order to be exploited. Figure 1 shows most of the parts in this embodiment.

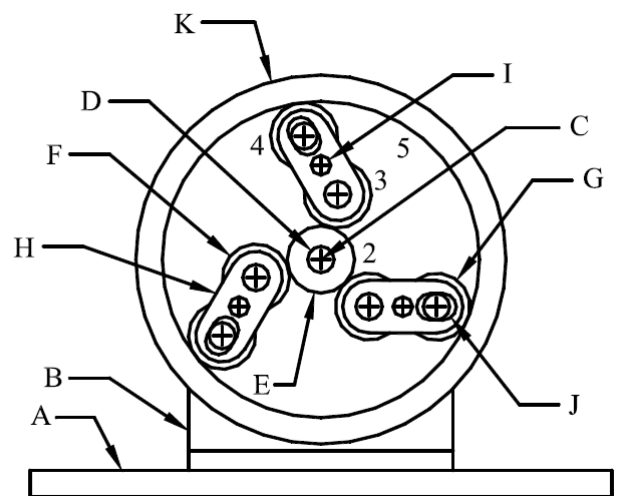


Figure 1 : Speed Reducer Embodiment [1]

Part **A** is the mounting plate on which the assembly is mounted. Part **B** is the back plate, which is securely fastened to part **A**. The back plate holds a bearing which defines the axis of rotation for the input and output members. This axis is identified as **C** in the figure. The input shaft is marked as **D** and the input member, part **E** is rigidly attached to **D** and rotates with it. There are three pairs of intermediate rollers shown in the figure, but there could be more or less than three sets. A typical inner roller is labeled **F**, and **G** is an outer roller.

Inner and outer rollers are held in proximity to one another by the roller plate, **H**. The outer hole, **J**, through which the outer roller shaft protrudes, is slightly elongated. This allows the inner and outer rollers to press firmly against one another, generating sufficient friction to transmit torque through the unit. The roller plate, along with the inner and outer rollers, is free to rotate about pin, **I**. The mid plate, in which the three pins are mounted, is shown in Figure 2. The slots are elongated to allow the roller plates and inner and outer rollers to seek a configuration with the inner roller pressed firmly against the input member, and the output roller pressed firmly against the output member, part **K**.

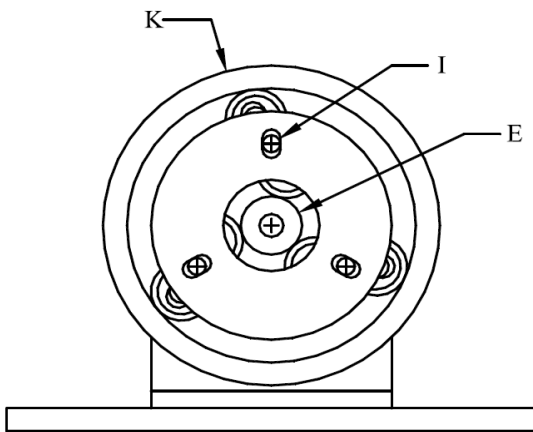


Figure 2: Speed Reducer Embodiment; Including the Mid Plate [1]

By using hollow rollers in the friction drive system have the advantages in accuracy of rotation and stiffness, even at high speeds, which leads to:

- Decrease the stresses in the rollers, thereby allowing smaller rollers to be used, due to the additional sharing of load between rollers as the rollers deflect more than solid rollers do under the same load, the reduction in stress is seen when the area of contact between the rollers expands under load.
- Decrease the device weight (some of the material is being removed).
- Since the contact stresses between the cylindrical rollers are decreased, the fatigue lives of those rollers are expected to increase, which is the main concern of using hollow rollers in that traction drive instead of solid rollers.

4.2. Solution Technique. The work of this research passes through three main stages. The first stage is the design stage of

the traction drive. The second stage is the fabrication of the traction drive, one with hollow rollers and another friction drive with solid rollers. The last stage is the testing of the speed reducers to make sure they are working well in transmitting the motion with speed reduction and to investigate their fatigue lives.

4.2.1. The Design Stage. Using the ProEngineering software, a CAD model was constructed for the traction drive proposed by Flugrad and Qamhiyah [1]. Initial values were chosen for the traction drive parts' dimensions. The ProEngineering contains a Pro-Mechanica package that enabled us to assign forces and reactions to the model parts. The main objective of the design stage was to get the optimum dimensions for the parts with the least values of the contact stresses.

One of the main disadvantages of friction drive systems compared to gear drive systems is the weight of the embodiment, hence, using a lightweight material will overcome the weight issue, such as using aluminum (AL), as it is suggested in this study. Aluminum has a density of 2.71g/cm³, modulus of elasticity of 73.08 GPa, and Poisson's ration of 0.33. Since aluminum is a ductile material, it will allow the rollers and the whole embodiment surfaces to get more flattened, which means distributing the forces applied on the surfaces normally on a larger area, so that the pressure on the parts will be decreased, allowing more fatigue life cycles for the rollers. While in hollow rollers the rubber was used to fill the hollowness since the axels and the pins need a rigid body to be fixed with, selecting the rubber was based on it's low density and modulus of elasticity ($\rho=0.92 \text{ g/cm}^3$, $E=0.05 \text{ GPa}$, $\nu=0.5$). Based on initial load of 100 N.m, initial dimensions were suggested for the rollers as follows: $r_2 = 55 \text{ mm}$, $r_3 = r_4 = 40 \text{ mm}$ and $r_5 = 183 \text{ mm}$. Based on those dimensions, it was found that:

$$\Phi = 70.69^\circ, \alpha_1 = 19.23^\circ, \alpha_2 = 12.64^\circ, \psi=38.83^\circ, \text{ and } R=114.62 \text{ mm.}$$

By trial and error strategy, the dimension values were varied until the minimum contact stresses on the solid rollers were obtained. Those minimum contact stresses resulted in optimum dimensions of: $r_2 = 31.25 \text{ mm}$, $r_3 = r_4 = 33 \text{ mm}$ and $r_5 = 157.5 \text{ mm}$. Based on those dimensions, it was found that:

$$\Phi = 34.18^\circ, \alpha_1 = 11.45^\circ, \alpha_2 = 5.88^\circ, \psi=16.85^\circ, \text{ and } R= 93.4 \text{ mm.}$$

Where:

- r_2 : the radius of the input rod
- r_3 and r_4 : the radii of the cylindrical rollers
- r_5 : the radius of the outer ring.

The angles are shown in Figure 3.

Based on these geometrical values, the coefficient of friction (μ) was estimated using equation 1 to be at least 0.30.

$$\mu \geq \frac{\sin(\alpha_1 + \alpha_2)}{\frac{(r_5 - r_2)}{R} \cos \alpha_2 + \cos(\alpha_1 + \alpha_2)} - 1 \quad \dots\dots\dots 1$$

after they have been deformed and all external forces have been removed.

Pro-Engineer software uses these two criteria to analyze the forces applied on the parts and illustrate the stress distribution on each element of the model, in order to find the areas with high stress concentration.

After the forces on each element are being analyzed the software will obtain the stress distribution as a multicolored figure, each color represent a range of stress values, as shown in Figure 9.

Same optimum dimensions obtained for the solid rollers were chosen for the hollow rollers. The traction drive with hollow rollers was subjected to the same loading conditions of the traction drive with solid rollers. The percentage of hollowness used was 60%.

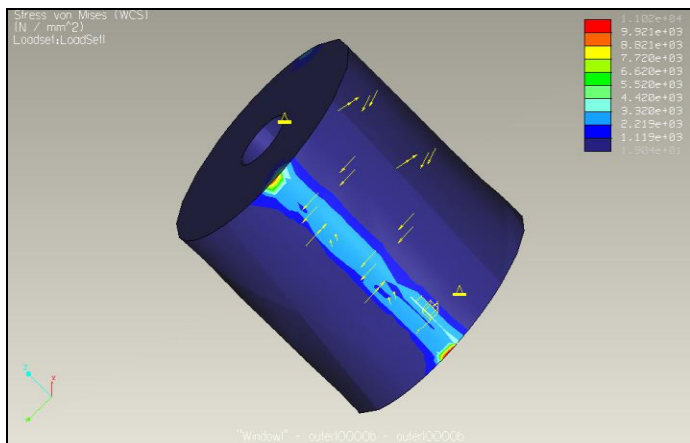


Figure 9: Distributed Stresses in Colors

4.2.2. Manufacturing Stage. The parts of the traction drives were fabricated based on the optimum dimensions obtained from the simulation models in the design stage. The output member of the traction drive that is shown in Figure 10 was fabricated by sand casting process. Casting is most often used for making complex shapes that would be otherwise difficult or uneconomical to make by other methods. Moreover casting is the easiest way to manufacture parts. In manufacturing the traction drive it is required to have the outer ring and output shaft as one piece to avoid any slippage and stress concentration there. The easiest and most economical way to do it is by casting. Midplate shown in Figure 11 was fabricated by casting process because of its complex shape. To separate the motion of input member from the fixed midplate, a bearing was used. The cylindrical rollers were fabricated from extruded rods of Aluminum. The extruded rod was cut to the specified dimension of the rollers and using the lathe machine the solid rollers were drilled for the center pin, and the hollow rollers were drilled up to 60% percentage of hollowness. The hollow rollers were filled with rubber as shown in Figure 12. Figure 13 shows the manufactured roller plate with the optimum design dimensions using lathe machine.



Figure 10: Casted Output Member



Figure 11: Casted midplate and input member

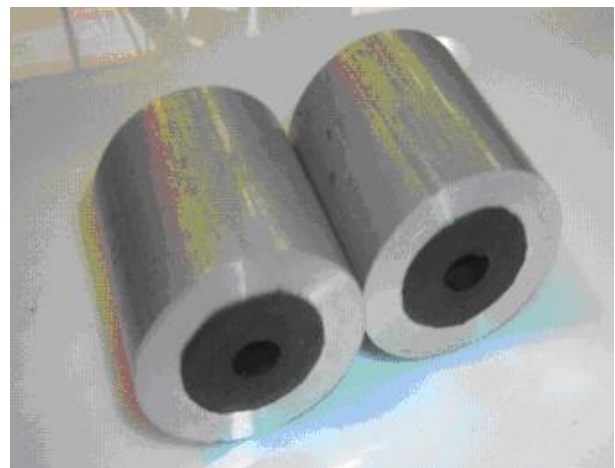


Figure 12: Hollow rollers reshaped at lathe machine

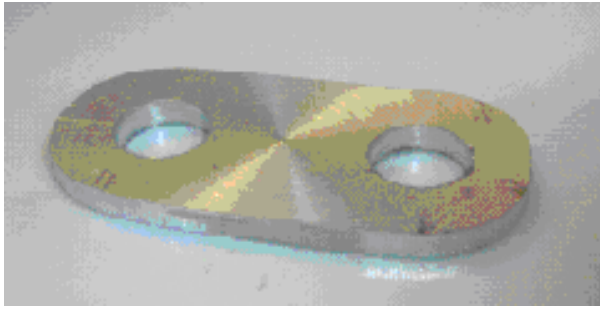


Figure 13: Roller plate after being drilled and refinished

Assemblage Stage. After the fabrication of all arts of both friction drives, the one with solid rollers and the one with hollow rollers, parts were assembled together as shown in Figure 14.

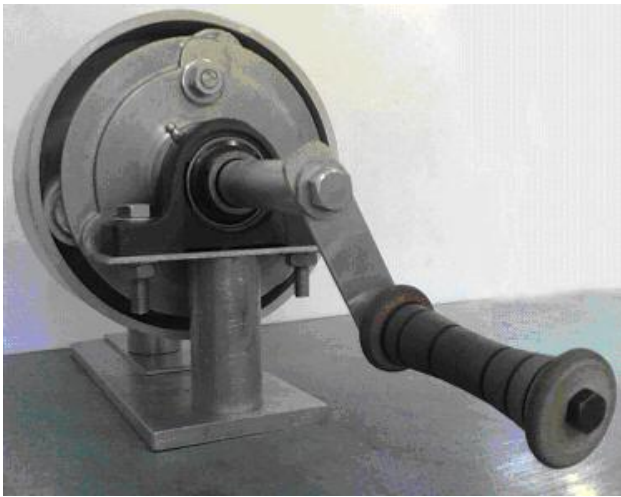


Figure 14: The speed reducer after assembled

4.2.3. Testing Stage. Manually the manufactured friction drives passed the test and could transfer motion to the output member successfully. Then they were tested using input motor to generate input rotational motion, and output brake acting as output load on the outer member. Speed and torque transducers were fixed on the input and output shafts. The output load were fixed at a constant value of 100 N.m in both testing cases, the testing of the friction drive with solid rollers and the testing of the friction drive with hollow rollers. The two friction drives with hollow rollers and with solid rollers could successfully transmit motion from input shaft to the output shaft with reducing the speed at fixed ratio. No problems appeared during their operation. The friction drives with hollow rollers could work in parallel and in series with the solid rollers friction drive.

5. DISCUSSION OF THE RESULTS

As mentioned before, Pro-Engineer software offers a practical way to preview the results of applying load on each part of the

two models of the traction drives. Each part of the two models –solid rollers model, and hollow rollers model- was subjected to different values of load, in order to make a clear comparison between the two models. Figure 15 shows a comparison between the solid roller used in the solid rollers traction drive and the hollow roller used in the hollow roller traction drive when subjected to same loading conditions. It can be clearly seen that stresses are distributed more in the body of the hollow roller, which resulted in lowering the maximum value of the stress. In case of the solid roller the stresses are more concentrated in the contact region of the roller with the other parts, and so the values of these stresses are very high. The red color indicated the areas of the highest stress value. The value of the red color is much higher in case of the solid roller as the figure shows. It is 54.51 MPa for the solid roller and it is 44.03 MPa in case of the hollow roller when both subjected to a load of 100 N.m. The flexibility of the hollow rollers resulted in expanding the contact patch of the roller with the other parts in contact and so distributing the contact stresses on larger area. That is why the maximum stress is lower in case of hollow roller.

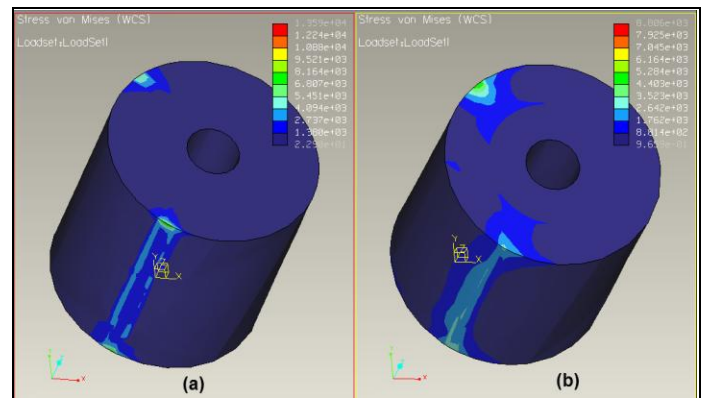


Figure 15: Distribution of stresses on: (a) A Solid roller
(b) A Hollow roller

The effects of having the cylindrical rollers hollow on the other parts of the traction drive were also studied. Figure 16 shows the outer ring of traction drive with solid rollers and the traction drive with hollow rollers. The outer ring of the hollow rollers traction drive was subjected to less contact stresses in very limited areas and the values of those stresses were much lower than the contact stresses on the outer ring of the solid rollers traction drive. The maximum stress value on the outer ring of the solid roller traction drive was 14.51 MPa compared to 2.494 MPa maximum stress value on outer ring of the hollow rollers traction drive. So, the flexibility of the hollow rollers made its contact patch with the outer ring larger and so less contact stress value was detected. That makes the fatigue life of the outer ring used in the hollow rollers traction drive much higher.

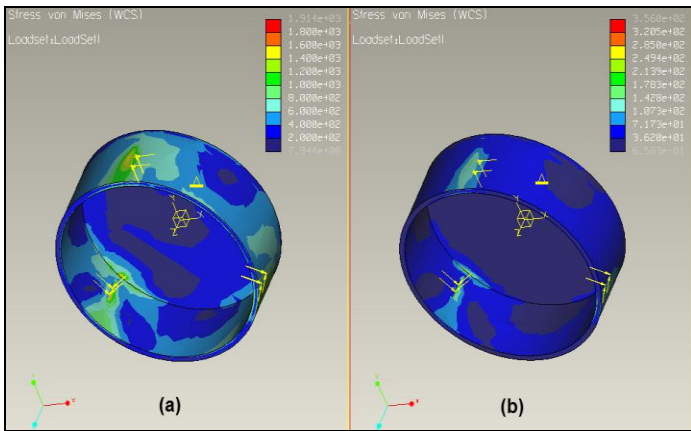


Figure 16: Distribution of stresses on output member, when using: (a) Solid rollers, (b) Hollow rollers

When investigating all parts of the traction drives, it was found that the maximum stress values were found on the roller plates. As Figure 17 shows, the maximum stresses on the roller plates of the hollow rollers traction drive was 75.54 MPa, but the values of the maximum stress values increased to 120 MPa in the roller plates of the solid rollers traction drive. These high values of the stress make the roller plates the weakest part in the traction drive for constant loading. But since these plates are not subjected to fatigue loading, they are not going to fail in fatigue and they can survive with a factor of safety of 2.67 in case of solid rollers and 4.21 in case of hollow rollers based on a yield stress value of 320 MPa for aluminum.

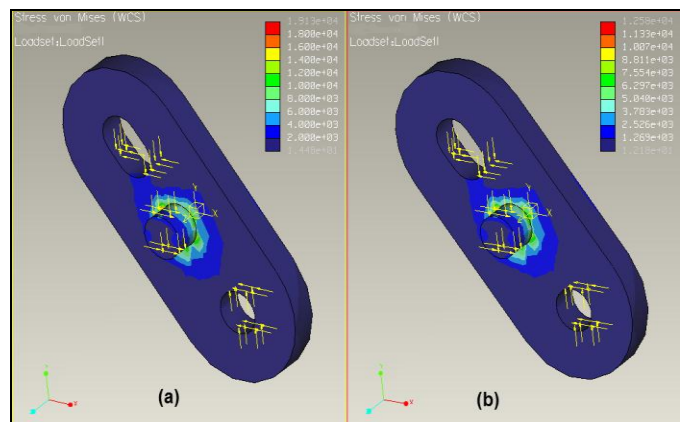


Figure 17: Distribution of stresses on the roller plate, when using: (a) Solid rollers, (b) Hollow rollers

Table 1 compares the maximum Von Mises stress values of solid rollers traction drive parts and the hollow rollers traction drive parts. Based on the S-N curve of Aluminum, the number of cycles each part can survive before fatigue failure is presented in the table. So, the hollow rollers when used in the traction drive, they can theoretically survive around 30 times the solid rollers. The outer rotating ring in contact with hollow roller is going to have much longer fatigue life than the ring in contact with solid roller. As Table 1 shows the outer ring of the hollow traction drive has a fatigue life of 2 million times the fatigue life of the outer ring of the solid rollers traction drive.

These results completely agree with the findings of Abu Jadayil [5]. Also other stationary parts in contact with those cylindrical rollers are going to survive longer when they are in contact with hollow rollers rather than being in contact with solid rollers.

Table 1: The maximum Stresses and the number of cycles of traction drive parts

Part Name	Maximum Von Mises Stress (MPa)	No. of Cycles (N)
Solid Cylindrical Roller	54.51	$3 * 10^9$
Hollow Cylindrical Roller	44.03	$9 * 10^{10}$
Output Ring with Solid Cylindrical Roller	14.51	$5 * 10^{15}$
Output Ring with Hollow Cylindrical Roller	2.494	$1 * 10^{22}$

6. CONCLUSIONS AND RECOMMENDATIONS

Based on the research results, the main conclusions can be summarized in the following points:

- The self actuating traction drive proposed by Flugrad and Qamhiyah [1] can be easily fabricated and assembled. Practically it is very successful in transmitting motion and reducing speed by a fixed ratio. It can be used with solid cylindrical rollers or with hollow cylindrical rollers.
- When comparing the stress distribution on both hollow and solid rollers when subjected to combined normal and tangential loading, it was found that better stress distribution was resulted for hollow rollers with lower peak stress values. So, it is expected that hollow rollers, with 60% percentage of hollowness, have longer fatigue life than identical solid rollers.
- Replacing solid rollers by hollow ones filled with rubber in traction drives, reduces their weights and increases their fatigue lives. That means time and money savings.
- Using hollow rollers instead of solid cylindrical rollers in the self actuating traction drive proposed by Flugrad and Qamhiyah [1] increases the fatigue lives of other parts in contact with the cylindrical rollers which subjected to fatigue loading, and factor of safety for stationary parts in contact with the cylindrical rollers.

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