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Design and Analysis of Modified Fluid Coupling for Gearless Two Wheeler

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Abstract: *There is an increased attention towards the development of Gear-less two-wheeler automobiles [scooters] which have automatic-transmission systems, using centrifugal-clutch systems. These clutch systems have high wear and tear and hence the present research work is taken up to replace the said clutch system with a fluid coupling which may be more effective in wear-free transmission and it will provide a smooth & controlled acceleration with effective damping of shocks, load fluctuations and torsional vibrations. The attention is therefore laid on developing a highly-efficient fluid coupling. This fluid coupling would capture the mechanical power from the main source, namely the I.C. engine and then transmits it to the rear wheels via an automatic gear box. The fluid coupling has an advantage over the mechanical coupling in the following areas. Effective dampening of shocks, load fluctuations and torsional vibrations. Smooth and controlled acceleration without jerks in transmission of the vehicle. Wear-free power transmission because of absence of mechanical connection [no metal-to-metal contact] between the input and output elements. But as the Conventional-fluid couplings have relatively low transmission efficiency, the challenge lies in developing an efficient modified fluid coupling which would transfer the mechanical power with minimum transmission losses in the case of Two-Wheeler automobiles especially the Gear-less Scooters. Presently these vehicles are using a Centrifugal-clutch which has maximum wear and tear frequently. Using the solid works 2008 software a fluid coupling was modelled by adjusting the gap between the driver and driven shaft and an estimate of the efficiency was also known. Based on the recommendations of a computational study an initial estimate of the efficiency of a basic modified-fluid coupling design without vanes and consequently in making recommendations on scope for further design improvements i.e. using vanes on the two discs].Based on the recommendations from the computational study a conventional fluid coupling with one driver shaft and one driven shaft will be fabricated and tested. Using the ANSYS 11.0 software the fluid coupling was analysed and various stresses and strains were calculated and an estimate of the efficiency was also known. Based on the recommendations of analysis further work was taken forward for the experimental work.*

I. Introduction

Old mobiles 1940 models featured hydramatic drive the first mass production fully automatic transmissions. Initially an old exclusive hydra matic had a fluid coupling (not a torque converter) and three planetary gear sets providing four speeds plus reverse. Hydramatic was subsequently adopted by Cadillac and Pontiac, and was sold to various other automakers, including Bentley, Hudson, Kaiser, Nash and Rolls-Royce. From 1950 to 1954 Lincoln cars were also available with GM hydra matic. Mercedes Benz subsequently devised a four speed fluid coupling transmission that was similar in principle to hydra matic but did not share the same design. The first torque converter automatic Buicks dyna flow was introduced for the 1948model year. It was followed by Chevrolets power glide and Packard’s ultramatic for the1950 model year. Each of these transmissions had only two forward speeds relying on the torque converter for additional gear reduction. In the early 1950s Borg-warner developed a series of three speed torque converter automatics for ford motor company. Studebaker and several foreign independent makes Chrysler was late in developing its own true automatic, introducing the two-speed torque converter power in 1953 and the three speed power flite in 1956. By the late 1960s most of the fluid coupling four speed and two speed transmissions had disappeared in favor of three speed units with torque converters. By the early 1980s these were being supplemented and eventually replaced by over drive equipped transmissions providing four or more forward speeds. Many transmissions also adopted the lock-up torque converter to improve fuel economy. As the engine computers became more and more capable and even more of the valves body functionality was offloaded to them. These transmissions introduced in the late 1980s and early 1990s remove almost all of the control logic from the valve body and place it in into the engine computer. In this case solenoids turned on and off by the computer control shift patterns and gear ratios rather than the spring loaded valves in the valve body. ZF Friedrichshafen AG and BMW were responsible for introducing the first five speed automatic and the first six speed (ZF 6HP26 in the 2002 BMW E65 7 series). Mercedes Benz was the first seven speeds in 2003 with Toyota motor company introducing an 8 speed in 2007 on the Lexus LS.

In modern usage, a torque converter is generally a type of hydrodynamic fluid coupling that is used to transfer rotating power from a prime mover, such as an internal combustion engine or electric motor, to a rotating driven load. The torque converter normally takes the place of a mechanical clutch in a vehicle with an automatic transmission, allowing the load to be separated from the power source. It is usually located between the engine's flex plate and the transmission. The key characteristic of a torque converter is its ability to multiply torque when there is a substantial difference between input and output rotational speed, thus providing the equivalent of a reduction gear. Some of these devices are also equipped with a temporary locking mechanism which rigidly binds the engine to the transmission when their speeds are nearly equal, to avoid slippage and a resulting loss of efficiency.

Fig 1: Torque Converter

A fluid coupling is a two element drive that is incapable of multiplying torque, while a torque converter has at least one extra element—the stator—which alters the drive's characteristics during periods of high slippage, producing an increase in output torque. In a torque converter there are at least three rotating elements: the impeller, which is mechanically driven by the prime mover the turbine, which drives the load and the stator, which is interposed between the impeller and turbine so that it can alter oil flow returning from the turbine to the impeller. The classic torque converter design dictates that the stator be prevented from rotating under any condition, hence the term *stator*. In practice, however, the stator is mounted on an overrunning clutch, which prevents the stator from counter-rotating with respect to the prime mover but allows forward rotation. Modifications to the basic three element design have been periodically incorporated, especially in applications where higher than normal torque multiplication is required. Most commonly, these have taken the form of multiple turbines and stators, each set being designed to produce differing amounts of torque multiplication. For example, the Buick Dyna flow automatic transmission was a non-shifting design and, under normal conditions, relied solely upon the converter to multiply torque. The Dyna flow used a five element converter to produce the wide range of torque multiplication needed to propel a heavy vehicle. Although not strictly a part of classic torque converter design, many automotive converters include a lock-up clutch to improve cruising power transmission efficiency and reduce heat. The application of the clutch locks the turbine to the impeller, causing all power transmission to be mechanical, thus eliminating losses associated with fluid drive.

An automatic transmission uses a fluid coupling torque converter to replace the clutch to avoid engaging/disengaging clutch during gear change. A completed gear set, called planetary gears, is used to perform gear ratio change instead of selecting gear manually. With the invention of the automatic transmission, a driver no longer needs to worry about gear selection during driving. It makes driving a car much easier, especially for a disabled or new driver. However the indirect gear contact of the torque converter causes power loss during power transmission and the complicated planetary gear structure makes the transmission heavy and easily broken.

Unlike a manual transmission system, automatic transmission does not use a clutch to disconnect power from the engine temporarily when shifting gears. Instead, a device called a torque converter was invented to prevent power from being temporarily disconnected from the engine and also to prevent the vehicle from stalling when the transmission is in gear. Consider two fans facing each other: when one of them is turned on and starts spinning, the other one will also start spinning at lower speed(see figure). That's because the first fan moves the air to drive the second fan to spin. This is the same idea as the torque converter of an automatic transmission system, except that it uses fluid instead of air as the transportation media.

Automatic transmission fluid (ATF) is the fluid used in vehicles with self shifting. It is typically coloured red or green to differentiate it from motor oil in the vehicle. The fluid is highly specialised oil optimized for the special requirements of a transmission. Modern fluids typically contain a wide variety of chemical compounds intended to provide required properties of a particular fluid specification. Most fluids contain some combination of additives that improve lubricating qualities such as wear resistant, corrosion resistant inhibitors, detergents, dispersants and surfactants (which protect and clean metal surfaces

Fluid coupling fluid is blended with the finest solvent refined hydro-finished which is with high viscosity index up to 100% paraffin base stocks available. These solvent refined hydro finished paraffin base stocks provide simplex fluid coupling fluid with superior oxidation resistance and also excellent thermal stability.

II. Modelling And Analysis Of Modified Fluid Coupling

1. Modified Fluid Coupling and Experimental Set Up Modelling

Solid modelling of modified fluid coupling and the entire experimental setup is done using solid works 2008 software [1]. Following are the sub sequent images for the model of the experimental set up.

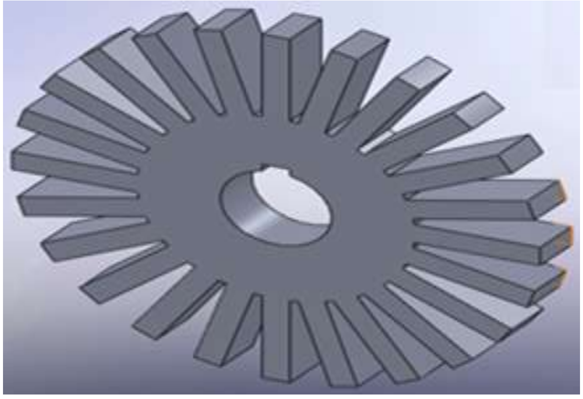


Fig 2: Impeller Straight

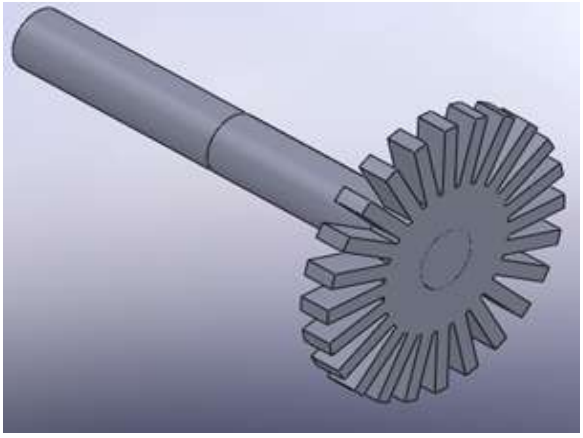


Fig 3: Assembly of Straight Vane Impeller

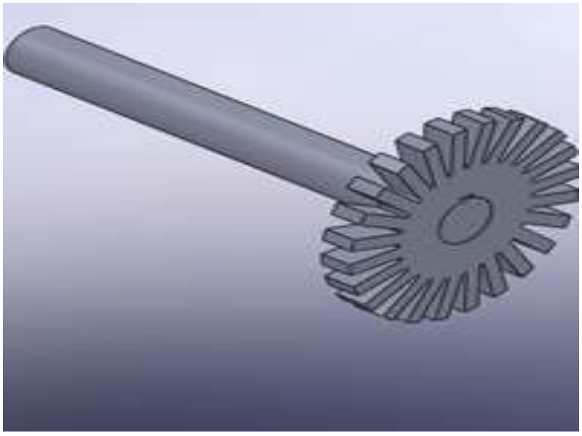
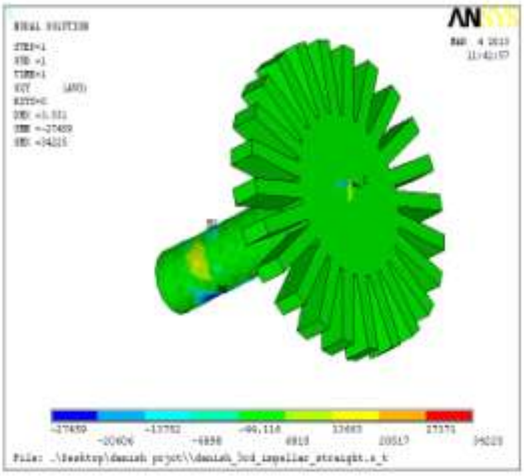


Fig 4: 15 Degree Assembly Impeller with Shaft

2. Analysis of the set up in ANSYS 11.0

Figures 5,6,7,8 shows that after application of load in the turbine runner we analyse the set up. Before the analysis various values such as shear stress and the material properties values were given in the Ansys .The load was applied tangentially in the face of the vanes and after analysis for various loads it was observed that the turbine runner was able to take the load of the corresponding fluid that is Castrol hyspin VG46 and was able to with stand the load.It is proved as the runner is in green colour.



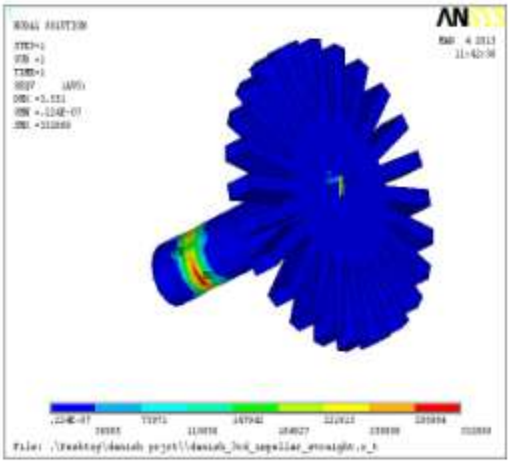
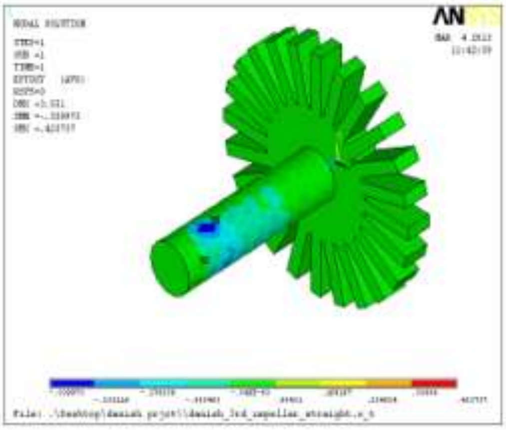


Fig 5: Shear Stress of Straight Impeller **Fig 6: Von mises Stress acting on the Straight Impeller**

The load was applied tangentially in the face of the vanes and after analysis for various loads it was observed that the turbine runner was able to take the load of the corresponding fluid and was able to with stand the load. The blue colours in the figures indicate that there is not much application of pressure or strain in the runner.



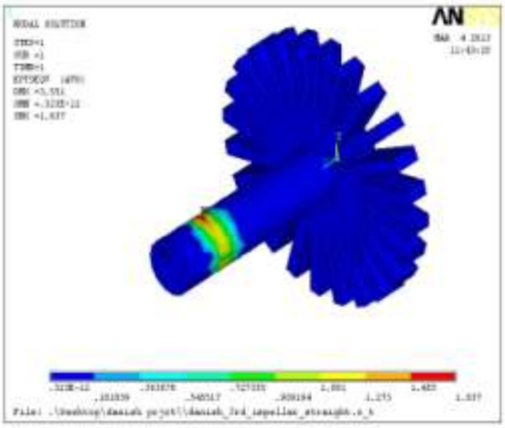
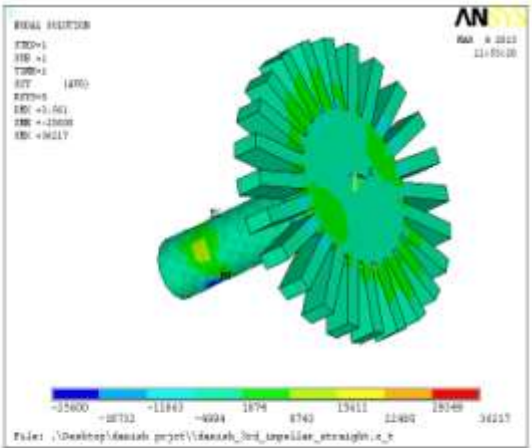


Fig 7: Shear Strain acting on the Straight Impeller **Fig 8: Von mises Strain at Impeller Straight**

Figures 9,10, shows that after application of load in the turbine runner we analyse the set up. Before the analysis various values such as shear stress and the material properties values were given in the Ansys .The load was applied tangentially in the face of the vanes and after analysis for various loads it was observed that the turbine runner was able to take the load of the corresponding fluid that is Castrol hyspin VG46 and was able to with stand the load. It is proved as the runner is in light green colour which indicated the setup is safe.



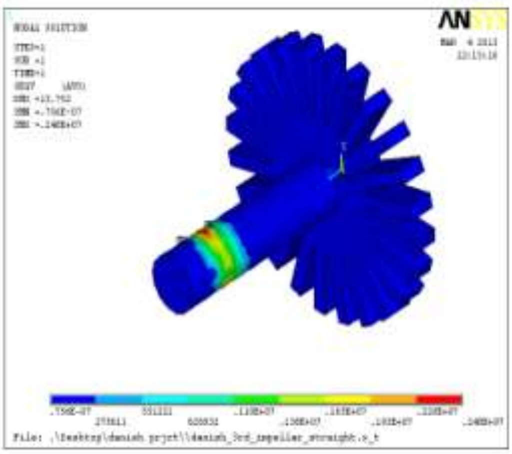


Fig 9: Shear Stress of 15^{o} inclined Impeller **Fig 10: Von mises Stress acting on the 15^{o} Impeller**

The load was applied tangentially in the face of the vanes and after analysis for various loads it was observed that the turbine runner was able to take the load of the corresponding fluid and was able to with stand the load. The blue colour in the figures indicate that there is not much application of pressure or strain in the runner

III. Results And Discussion

After the analysis various shear stresses, shear strain and deflection values for straight, 15 degree tilt and 30 degree tilt vane angle were compared. After comparison it was found that the radial or straight vane which was selected for is a optimised one and gives improved readings compared to 15 degree and 30 degree vane angle. The values of deflection obtained for 15 degree and 30 degree vane angle are more compared to the values of deflection obtained for 15 degree and 30 degree vane angle. The values of shear stress obtained for 15 degree and 30 degree vane angle are less compared to the values obtained for straight or radial vane angle. The values of shear strain obtained for 15 degree and 30 degree vane angle are less compared to the values of shear strain obtained for straight or radial vane angle. It was observed that as the load was increasing the deflection was also increasing for all the three cases that is radial, 15 degree, 30 degree vane angles. It was observed that the shear stress and shear strain values were taking a high at a particular range of loads for all the three cases that is radial or straight, 15 degree vane angle, 30 degree vane angle. Finally after considering all the results we decided to take radial or straight vanes for our research work. The number of vanes in the pump impeller and turbine runner are 24.

TABLE 1: STRAIGHT VANE ANGLE

S.NO.	LOAD (N)	DEFLECTION (mm)	SHEAR STRESS (N/mm^2)	SHEAR STRAIN
1	2.05	0.003	362	0.448
2	3.04	0.005	464	0.563
3	4.02	0.0069	578	0.695
4	5	0.0086	698	0.736
5	5.98	0.0102	814	0.834
6	6.96	0.0122	578	0.536
7	7.94	0.0137	432	0.412
8	9.80	0.0169	303	0.33

TABLE 2: VANE ANGLE AT 15 DEGREES

S.NO.	LOAD (N)	DEFLECTION (mm)	SHEAR STRESS (N/mm^2)	SHEAR STRAIN
1	2.05	0.005	298	0.312
2	3.04	0.007	310	0.444
3	4.02	0.0089	412	0.501
4	5	0.011	612	0.613
5	5.98	0.0124	712	0.713
6	6.96	0.0141	432	0.455
7	7.94	0.015	312	0.213
8	9.80	0.0187	221	0.245

TABLE 3: VANE ANGLE AT 3O DEGREES

S.NO.	LOAD (N)	DEFLECTION (mm)	SHEAR STRESS (N/mm^2)	SHEAR STRAIN
1	2.05	0.007	191	0.234
2	3.04	0.0089	210	0.321
3	4.02	0.0105	280	0.422
4	5	0.0124	423	0.541
5	5.98	0.0135	512	0.623
6	6.96	0.0159	312	0.321
7	7.94	0.0176	235	0.134
8	9.80	0.0205	135	0.145

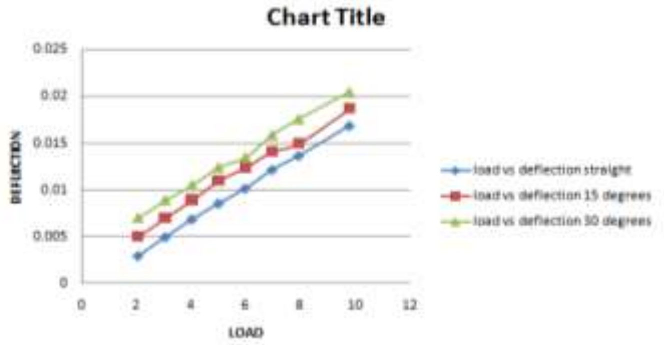


Fig 11: load vs deflection

Figure 11 shows that deflection for straight vane angle is less than compared to 15 degree and 30 degree vane angle. It shows that as the load increases the deflection also increases.

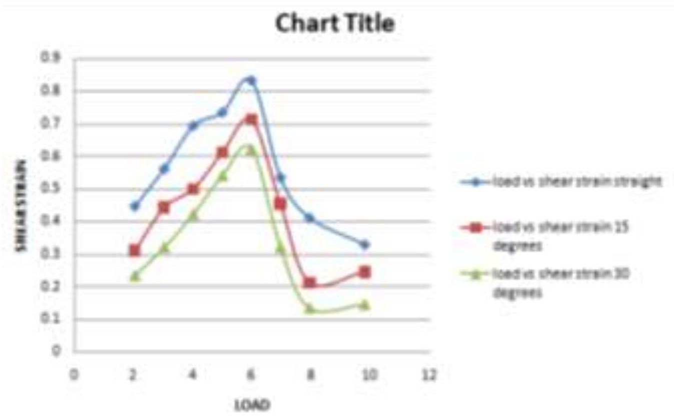


Fig 12: load vs shear strain

Figure 12 shows that shear strain for straight vane angle is more compared to 15 degree and 30 degree vane angle. It shows that at a brake load of 6 newtons the shear strain reaches a high value.

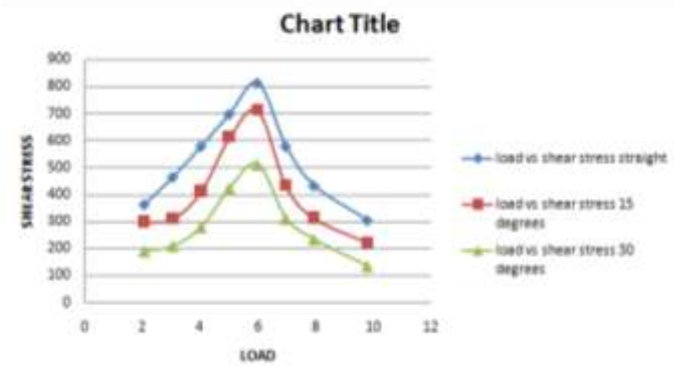


Fig 13: load vs shear stress

Figure 13 shows that shear stress for straight vane angle is more compared to 15 degree and 30 degree vane angle. It shows that at a brake load of 6 newtons the shear stress reaches a maximum high

IV. Conclusion

- The deflection, shear stress and shear strain were obtained for straight angle, vane angle at 15 degrees and vane angle at 30 degrees.
- The deflection for straight vane angle is less than compared to 15 degree and 30 degree vane angle.
- The shear strain for straight vane angle is more compared to 15 degree and 30 degree vane angle
- That shear stress for straight vane angle is more compared to 15 degree and 30 degree vane angle

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