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Design and Mathematical Modeling of Fluid Coupling for Efficient Transmission Systems

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ABSTRACT

In the present day scenario, 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. (1)Effective dampening of shocks, load fluctuations and torsional vibrations. (2) Smooth and controlled acceleration without jerks in transmission of the vehicle. (3) 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. The objective of this research work is to highlight the study conducted on the torque transmission efficiency of a basic-fluid coupling without vanes and a modified-fluid coupling [with vanes]. This is followed by fabricating and then experimentally testing a fluid coupling design based on the recommendations extracted from this computational study. The computational study was helpful in making 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 was fabricated and tested. The experimental study was conducted on disks with radial vanes and with micro clearance between the driver and driven disks. The experiments showed very high torque transmission efficiency as a result of the introduction of radial vanes with micro clearances. Based on the findings from the above computational and experimental studies, further research work is currently under progress for the design of a highly efficient modified fluid coupling for application in the Two-Wheeler automobile industry of Gear-less Scooters.

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KEY WORDS

Fluid coupling, impeller, torque converter ,input disk torque.

1. INTRODUCTION

Old mobiles 1940 models featured hydra matic 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 packards 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.

2. WORKING OF A TORQUE CONVERTER

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

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3. TORQUE CONVERTER ELEMENTS

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.

4. EXPERIMENTAL STUDY ON FLUID COUPLING

The recommendations of using radial vanes and micro clearances between the disks from the computational analysis were tested experimentally. The experiments were carried out by fabricating a conventional fluid coupling with radial vanes and with a micro clearance between the vanes of the two disks. Figures, 2, 3, 4 and 5 show the photographs of the experimental setup of this fluid coupling. The driver shaft powered by induction motor drives the pump. The turbine is coupled to the driven shaft on which the brake load is applied for conducting the brake test. The pump and the turbine consist of impellers with radial vanes as shown in figure 5.

Fig 2. Induction motor with fluid coupling

Fig 3. Fluid coupling

Fig 4. Wattmeter connected to motor

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Fig 5. Impeller of pump and turbine

The casing housing is filled with oil with the following specifications given in Table1.

Table 1: Specifications of the oil used in fluid coupling.

Oil name	Castrol Hyspin VG46
Density at 15^0 C	879 kg/m^3

5. CALCULATIONS OF TORQUE AND EFFICIENCY

The torque from the driver shaft T_i is calculated using the power equation

$P_i = (2*\pi*N_i*T_i) / 60$ (1)

The input power P_i is read from the wattmeter. The input shaft R.P.M. N_i is read using a tachometer. The torque captured by the driven shaft "T_o" is again calculated using the power equation

$P_o= (2*\pi*N_o*T_o) / 60$ (2)

The output power "P_o" is calculated from the brake test. The output shaft R.P.M. "N_o" is read using a tachometer. Torque transmission efficiency η_T == (Output torque / Input torque) = (T_o/T_i).

Table 2: Experimental Results

S. No.	Brake Load gms	Induction Motor			Fluid Coupling		Torque transmission Efficiency η_T
		Power Pi watts	Speed N_i RPM	Torque T_i N-m	Speed N_o RPM	Torque T_o N-m	
1	210	280	1122	2.383	1147	2.33	97.8
2	310	280	1298	2.059	1330	2.01	97.6
3	410	272	1370	1.896	1400	1.85	97.6
4	510	260	1370	1.812	1403	1.77	97.7
5	610	260	1370	1.812	1403	1.77	97.7
6	710	252	1375	1.75	1407	1.71	97.7
7	810	248	1381	1.71	1410	1.68	98.2

Variation of torque transmission efficiency with change in speed of driver shaft is shown in Fig 6

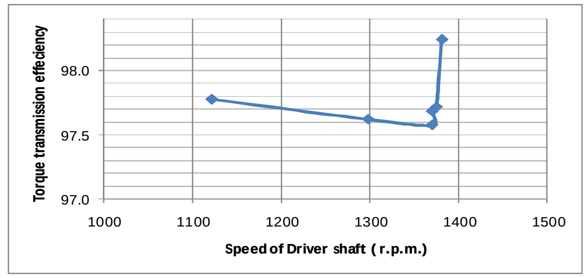

Fig 6: Variation of torque transmission efficiency with speed of driver shaft

6. EXPERIMENTAL VALIDATION

The experimental data collected from various test runs are analyzed from the **MINITAB SOFTWARE,** for the development of mathematical modeling of the fluid coupling. The various parameters like input disk torque, input shaft speed with and without shaft coupling are considered.

C1	C2	C3	C4	C5	C6	C7
S.No	Brake Load gms	Power Watts	Speed RPM	Torque N-m	Speed with Coupling RPM	Fluid Coupling Torque N-m
1	210	280	1122	2.383	1147	2.33
2	310	280	1298	2.059	1330	2.01
3	410	272	1370	1.896	1400	1.85
4	510	260	1370	1.812	1403	1.77
5	610	260	1370	1.812	1403	1.77
6	710	252	1375	1.750	1407	1.71
7	810	248	1381	1.710	1410	1.68
8	910	232	1398	1.680	1425	1.65
9	1010	225	1410	1.575	1435	1.60

Fig 7: Data sheet of experimental data in MINITAB SOFTWARE

Regression Analysis: Input disk torque versus Speed RPM, Power Watts,

The regression equation is given by

Input disk torque N-m =4.1596-0.00176*Speed+0.00093*Power-0.000303*Brake Load (3)

Table 3: Coefficients of Regression equation obtained from MINITAB

Predictor	Coef	SE Coef	T	P
Constant	4.1596	0.6859	6.06	0.002
Speed	-0.0017558	0.0001646	-10.67	0.000

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RPM				
Power Watts	0.000926	0.002510	0.37	0.727
Brake Load gms	-0.0003029	0.0001997	-1.52	0.190

S = 0.0239570 R-Sq = 99.4% R-Sq(adj) = 99.0%

Analysis of Variance

7. BOX-BEHNKEN DESIGN

The following are the factors and test runs are considered in the design table for determining the range of optimum values of the input disk torque with desired efficiency.

Table 4: Box-Behnken design table

1	Factors =1	Replicates = 1
2	Base Runs = 15	Total runs = 15
3	Base block = 1	Total blocks: 1

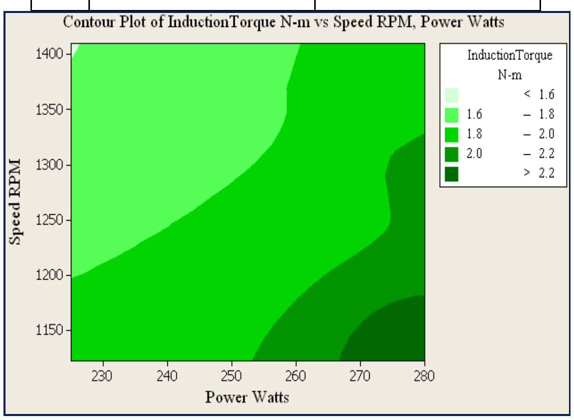

Fig 8: Contour plots of Input disk torque

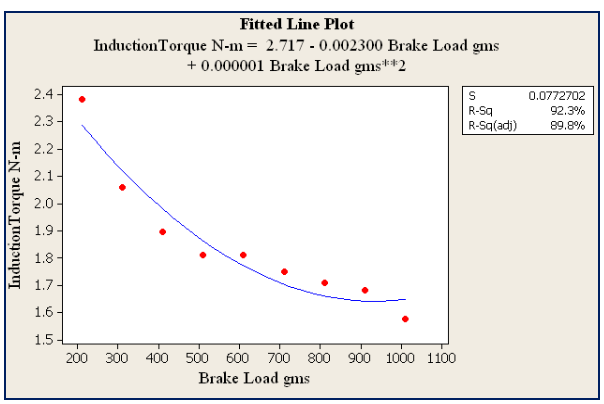

Fig 9: Fitted Line plot for Input disk torque

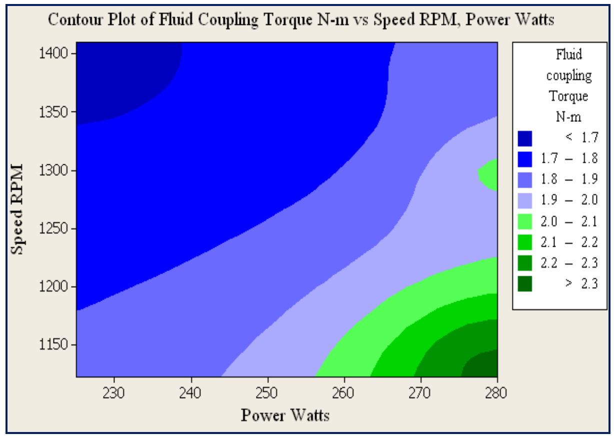

Fig 10: Contour plots of Output disk torque

Regression Analysis: Fluid Coupling versus Speed with C, Brake Load g.

The regression equation is given by Output disk torque =4.00-0.00164* Speed with Coupling -0.000332* Brake Load +0.00101 Power

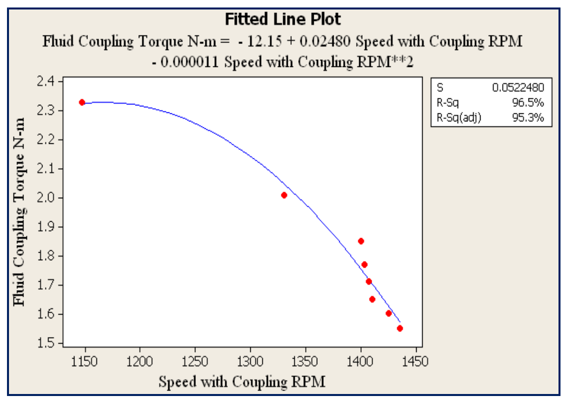

Fig 11: Fitted Line plot for Output disk torque

Table 5: Coefficients of Regression equation obtained from MINITAB

Predictor	Coef	SE Coef	T	P
Constant	4.0018	0.4684	8.54	0.000
Speed with Coupling RPM	0.0016409	0.0001080	-15.20	0.000
Brake Load gms	0.0003316	0.0001361	-2.44	0.059
Power Watts RPM	0.001007	0.001719	0.59	0.583

S = 0.0162948 R-Sq = 99.7% R-Sq(adj) = 99.5%

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8. CONCLUSION

The recommendations from the computational study were successfully tested by constructing a conventional fluid coupling prototype with radial vanes and micro clearance between the disks. The results showed that the above design approach gives a significantly high torque transmission efficiency of over 95 %. This design concept can therefore be taken forward for developing a highly efficient hybrid fluid coupling.

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