

Analysis and Simulation of a Centrifugal Clutch

John McDyer – 2010064

Lecturer:	Rob Goodson
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Task A

Introduction

Clutches are used throughout industry within production lines and driven machinery such as lawnmowers, chainsaws, pump systems, go-karts, cars and any suitable application where power is to be transmitted. They provide a means of transmission of power to the output portion of a system with a usually constant driven input. The main body of this document will explore two specific types of clutch; centrifugal and electromagnetic. Their application, operation and assembly will be discussed along with their advantages and disadvantages. Specifically with electromagnetic clutches their use in modern production lines will be researched and how they can inversely be used as a braking mechanism. The purpose of the literature review is to gain a greater understanding of the use of clutches and to provide the practical knowledge behind the calculations carried out in task b.

Discussion

Centrifugal clutches

Centrifugal clutches have been in use since the early 1900's and have undergone a number of developments in that time and with the advance in materials and designs, centrifugal clutches have increased in capacity and improved efficiency. The early centrifugal clutches were predominately of a spring and shoe design (Fig 1) where the centrifugal force generated by the increase in angular velocity eventually overcomes the spring tension. A further increase in angular velocity would then provide the level of friction required to turn the outer drum and hence the output shaft of the clutch.

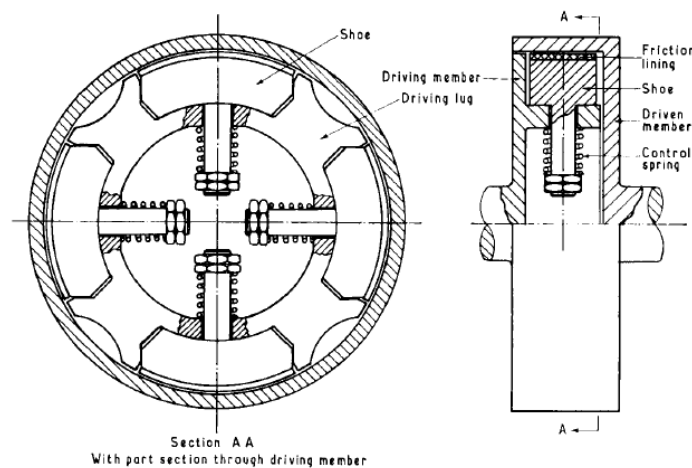


Figure 1

This type of spring force centrifugal clutch was very much favored within the industrial sector however the automotive industry moved to a pivot design of clutch (Appendix 1) to reduce friction between the shoe and driving member. Even with this development in clutch design there was the pitfall of uneven wear on the friction faces due to the pivoting nature of the centrifugal arms. Given the detrimental nature of the wear pattern, the design was further re-worked to incorporate a number of smaller shoes and links known as a multilink trailing shoe clutch (Fig 2).

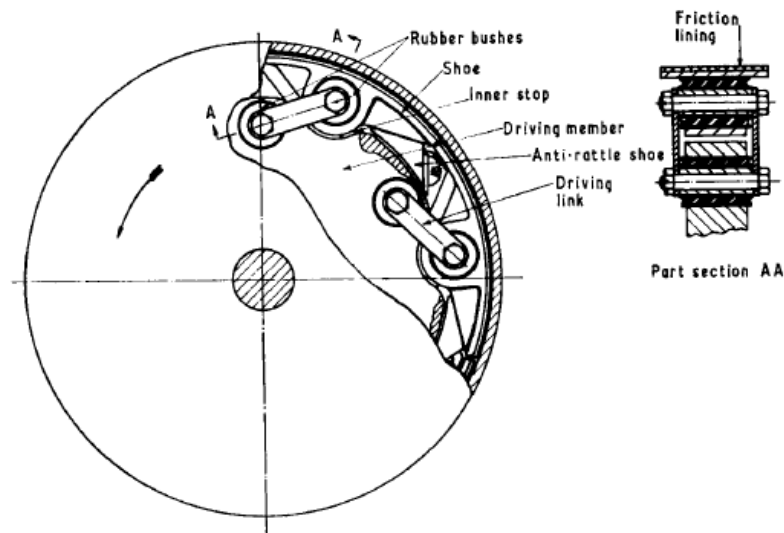


Figure 2 - Multi link trailing shoe clutch

One identified flaw in this type of multi shoe design was an undesirable rattling of the shoes when at idling speeds which caused premature wear and noise. The initial re designs were to fit a sliding ramp arrangement (anti-rattle device) to the clutch shoes ensuring that the shoes engage with the drum at the correct angle for effective transmission of torque (Tridgell, C W;, 1969-70). It is clear to see that all types of centrifugal clutches rely on correct maintenance to ensure they run accurately to their correct torques and engagement speeds. Changes such as worn brake shoes, stiff linkages, high temperatures or ingress of oil / dirt will all effect the working coefficients of the clutch which will have a detrimental effect on the speed / torque relationship.

As with all design aspects of engineering, part count and efficiency are key as highlighted in the design for 'x' approach. The lower the number of parts the easier it is to manufacture and assemble and hence reducing cost (Gatenby & Foo, 1990), and this approach has been taken with the development of the compliant clutch (Fig 3) 'The compliant clutch mechanisms are mechanisms that obtain some or all of their motion from the deflection of their members. They frequently require fewer parts than comparable rigid-body designs since revolute joints are often replaced by flexible segments. The potential energy stored in the flexible segments can replace springs and the reduction in revolute joints reduces problems with backlash and wear (Crane, 2003). In many applications, compliant mechanisms can maintain or even improve performance relative to conventional rigid-body design' (Murphy, et al., 1986).

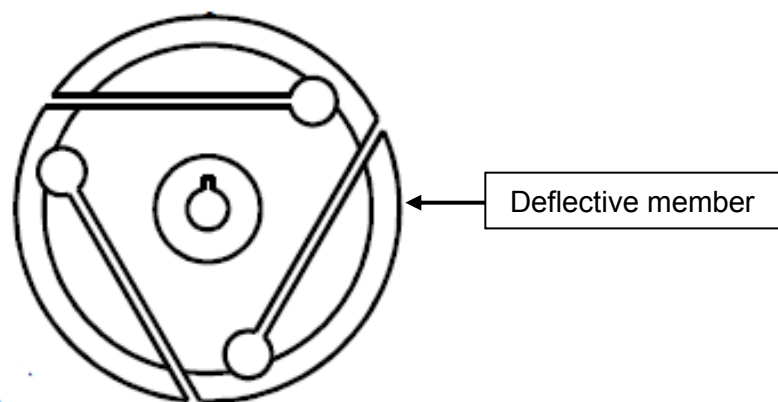


Figure 3 – Compliant clutch (Crane, 2003)

Looking to modern day centrifugal clutches, the basic principle has not drastically changed, however advances in materials have been the major breakthroughs for centrifugal clutches. Initially cast iron outer drums and shoes were utilized alongside asbestos based friction linings (Wemp, 1925). Since the prohibition of importing and use of asbestos based substances in 1985 and subsequently 1999 due to public health was brought into force (Public health England, 2017) alongside the advance in material technology, a change in the material of the components has occurred particularly in the shoe lining. A friction clutch is required to withstand high levels of heat and deformation along with providing a coefficient of friction that is able to transmit the required torque while remaining relatively hard wearing. There are a vast number of friction linings available including organic compounds, non-organic compounds and metallics, however they are commonly made up of a composite of materials to gain the desired characteristics. For instance friction materials are commonly made from sintered lead bronzes or iron powders with an addition of graphite to provide a form of dry lubrication. Organic compounds such as resin can be used however these are not always easy to produce due to having woven fibers and can have a short shelf life (Morris, 2011). In more recent years the use of more exotic materials such as carbon ceramics have been favored in more expensive systems for their ability to withstand extremely high temperatures and having excellent wear rates however, due to the cost of purchase they are very limited within the industrial sector. The material of the clutch drum is predominantly made of far simpler materials such as cast irons or higher carbon steels to produce the hard wearing requirements of the clutch, particularly within the systems that are being considered within this discussion. Clutch drums that are made up carbon steel can also undergo further processing such as case hardening to prolong the life of the drum.

Advantages / Disadvantages

Advantages	Disadvantages
Provide overload protection	Can overheat
Reduce starting torque on AC motors	Wear rates
Reduces the motor size requirement	Load capacity
Reduce load on engines	Space and size of the clutch
Relatively low cost	Small Δ in speed = Large Δ in power
Simplicity	Lack of versatility
Ease of maintenance / Low maintenance	Slips in heavy load conditions

(Mishra, 2019)

Electromagnetic Clutches in Automation

To understand how this particular type of clutch is used in automation a basic understanding of its operation is necessary. As stated in the dictionary of mechanical engineering, an electromagnetic clutch is 'a clutch in which an electromagnet is used to magnetize a rotor that attracts one or more friction discs, either directly or through an armature' (Escudier & Atkins, 2019). So in simple terms a friction disc is held off from a contact face by compression springs or permanent magnet until an electromagnetic field is produced that overcomes the spring force or permanent magnet at which point the clutch faces engage providing transmission (Fig 4). There are a number of types electromagnetic clutches such as a hysteresis clutch, magnetic particle clutch and a standard electromagnetic clutch (Fleming, 2017).

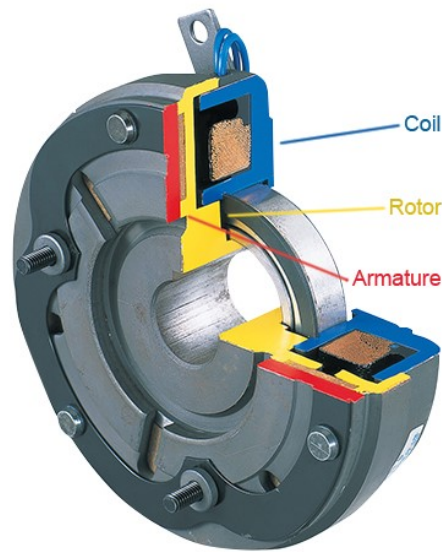


Figure 4 – <https://www.electric-clutches.com/articles.php>

It is also worth noting that the systems can be inversely set up/designed where an electromagnetic clutch could be used as a braking system. For instance on a machine hoist system, the magnetic field releases the friction faces by overpowering the springs. This type of system has inherent safety features as a loss of power would then engage the braking system preventing any loss of control of the load. Electromagnetic clutches are used within automated manufacturing processes as they can be controlled remotely by PLC systems. The PLC system can receive feedback from a number of sensors which can engage or disengage a clutch. For instance on a conveyor drive for a production line, electromagnetic clutches are used for controlling the movement of the conveyor, when a part passes through a proximity sensor the signal of the sensor is fed back to the PLC which runs the program and sends a signal to the electromagnet to engage the clutch. The electromagnetic clutch is very fast at gaining 100% engagement and can be used for very quick and precise applications (Ortlinghaus, 2004). This type of use saves large amounts of energy use in the production process and also decreasing wear on mechanical parts. Electromagnets do have one major limitation which is the relative heat they can produce from the electromagnetic coil, this poses a risk of overheating in applications where clutch cycling is required. Certain types of electromagnetic clutches such as magnetic particle clutches can also provide variable torque and slow engagement which is a useful technology in industries where automated winding applications are required. The variable torque and slow engagement is controlled via a change in voltage to the clutch which variably engages/disengages the clutch, so we can say that the torque is proportional to the voltage applied (Fleming, 2017).

Conclusion

As show in the discussion previously and shown in my research, centrifugal clutches are an extremely effective machine for power transmission, they are relatively inexpensive and the positives far outweigh the negatives. Their ease of maintenance and the overload protection they provide to more expensive parts within the drive chain means they can be a valuable form of linkage in many systems. With the advances in technology and the inception of electromagnetic clutches, the vast control features that can be accessed and the levels of accuracy and variability offered by the magnetic clutch shows it's a far superior form of transmission albeit as a much higher cost. Ultimately the selection of clutch relies on the application of its intended use and features required.

Task B

1) Diameter of outer drum

$$\text{Outer diameter} = (95+9+4) \times 2 = 216 \text{ mm}$$

$$\text{Inner diameter} = (95+9) \times 2 = 208 \text{ mm}$$

2) Engagement speed of the centrifugal clutch

$$r_s (\text{SHOE RADIUS}) = 0.095 \text{ m}$$

$$r_d (\text{DRUM RADIUS}) = 0.104 \text{ m}$$

$$F_c = F_s \quad \text{where} \quad F_s = K \times x$$

$$F_s = 7.5 \times 10^3 \times 0.009 = 67.5 \text{ N}$$

SO,

$$F_s = m \omega^2 r_d$$

$$\omega^2 = \frac{F_s}{m r_d}$$

$$\omega = \sqrt{\frac{F_s}{m r_d}}$$

$$\omega = \sqrt{\frac{67.5}{0.275 \times 0.104}}$$

$$\omega = 48.5812 \text{ rad/s}$$

$$\text{RPM} = \frac{48.5812}{2\pi} \times 60 = 464 \text{ RPM}$$

3) Radial force acting on the clutch

$$\omega = \frac{550}{60} \times 2\pi = 57.5959 \text{ rad/s}$$

$$F_c = m \omega^2 r d$$

$$F_c = 0.275 \times 57.5959^2 \times 0.104$$

$$F_c = 94.8744 \text{ N}$$

* NET FORCE IS EQUAL TO $F_c - F_s$ SO,

$$F_N = 94.8744 - 67.5 = 27.3744 \text{ N}$$

* MULTIPLIED BY 4x SHOES.

$$F_N \times 4 = 109.4976 \text{ N}$$

4) Torque value at 550 RPM

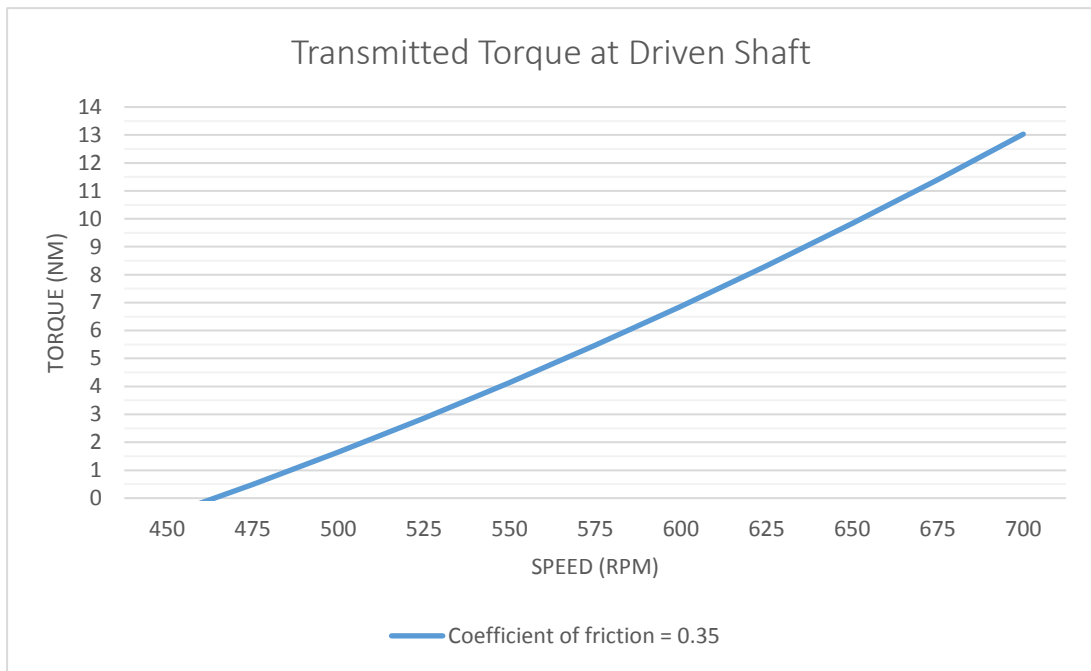
$$T = \mu r d (m r \omega^2 - Kx) \text{ SO,}$$

$$T = \mu r d F_N$$

$$T = 0.35 \times 0.108 \times 109.498$$

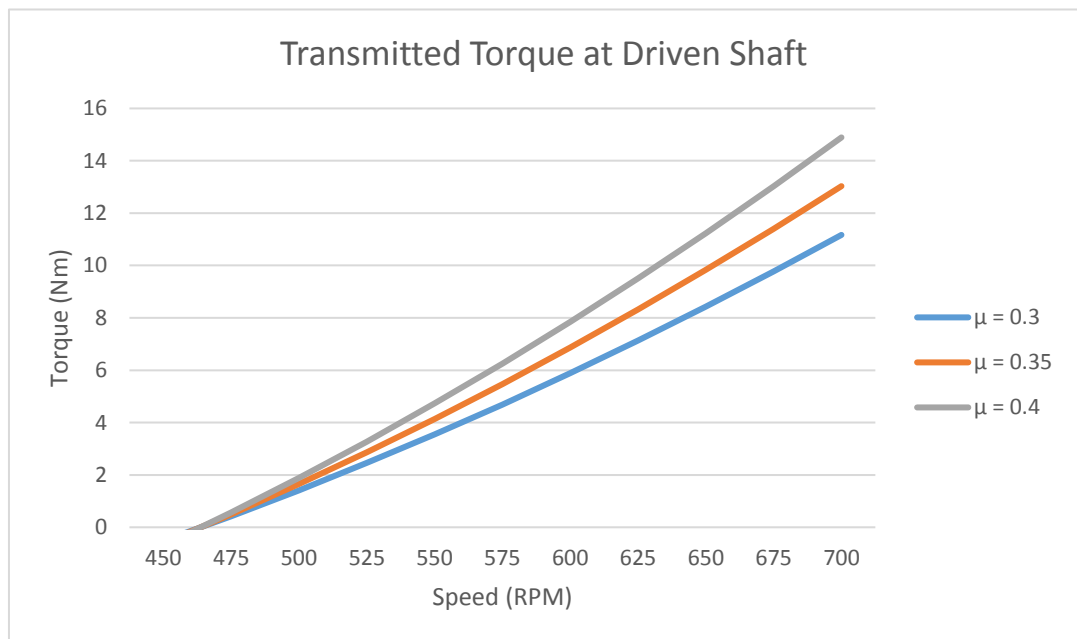
$$T = 4.139 \text{ Nm}$$

5) Transmitted torque at the driven shaft at intervals of 25 RPM between 450 RPM and 700 RPM



*Tables used to produce the graph can be found in Appendix 2

6) Transmitted torque at the driven shaft at intervals of 25 RPM between 450 RPM and 700 RPM with varying coefficients



*Tables used to produce the graph can be found in Appendix 3

As can be seen from the graph containing the varying coefficients of friction, its clear to see that the higher coefficient of friction leads to an increase in torque. In addition to that, the torque increases proportionally to the coefficient of friction. For instance if you double the coefficient of friction you will double your torque. This is because when coefficient of friction is measured, the force required to move the object over the surface is calculated by using the formula $COF = F_{\text{friction}} \times F_{\text{normal}}$. The higher the friction force required the higher the coefficient of friction becomes.

- 7) Angular acceleration to achieve maximum torque.

$$T = I \alpha \quad \text{so} \quad \alpha = \frac{T}{I}$$

$$\text{WHERE } I = \frac{1}{2} m r^2$$

$$I = \frac{1}{2} \times 0.75 \times 0.108^2$$

$$I = 4.374 \times 10^{-3} \text{ kg/m}^2$$

$$\text{ANGULAR ACCELERATION @ 700RPM} + \mu 0.4$$

$$\alpha = \frac{14.89198}{4.374 \times 10^{-3}} = 3404.66 \text{ rad/s}^2$$

- 8) Direct stress at drum using the table from appendix 2 @700RPM

$$\text{STRESS} = \frac{F}{A}$$

$$\sigma = \frac{F}{A}$$

$$\sigma = \frac{86.18045}{0.05 \times 0.02}$$

$$\sigma = 86180.45 \text{ N/m}^2 \therefore 86.18 \text{ kN/m}^2$$

The stress situation is not considered as a complex stress model as at the maximum speed it would be assumed that the speeds are matching between the drum and the shoe so there is no slip between the faces, 100% engagement shall we say. Taking this into account, the only stresses acting on the face of the drum are considered to be perpendicular to the drum face and the load is uniformed across the cross section of the material. In addition to this, in our formula we are only considering the force / area. Given this case it can be considered as a simple stress.

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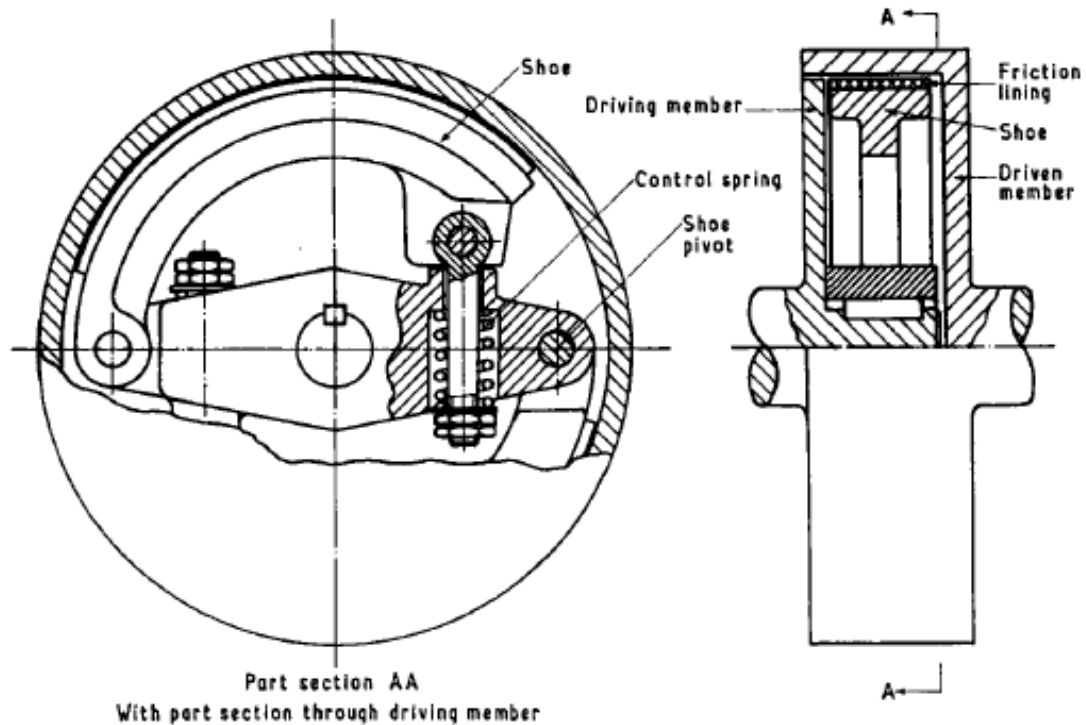
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Appendix

1)



Shoe type centrifugal clutch - (Tridgell, C W.; 1969-70)

2) Table of results for Task 2(5)

No of shoes	μ	m (kg)	Speed (RPM)	ω (rad/s)	Angular velocity	Radius outside of drum I.D	Radius outside of drum O.D	Spring force	Centrifugal force	Net force for 1 shoe	T (N/m)
						rd (m)	rod (m)	Fs	Fc (N)	Fn (N)	
4	0.35	0.275	450	47.1239		0.104	0.108	67.5	63.51079703	-3.989203	-0.603167
4	0.35	0.275	475	49.7418		0.104	0.108	67.5	70.76357323	3.263573	0.493452
4	0.35	0.275	500	52.3598		0.104	0.108	67.5	78.4083914	10.90839	1.649349
4	0.35	0.275	525	54.9778		0.104	0.108	67.5	86.44525151	18.94525	2.864522
4	0.35	0.275	550	57.5958		0.104	0.108	67.5	94.87415359	27.37415	4.138972
4	0.35	0.275	575	60.2138		0.104	0.108	67.5	103.6950976	36.1951	5.472699
4	0.35	0.275	600	62.8318		0.104	0.108	67.5	112.9080836	45.40808	6.865702
4	0.35	0.275	625	65.4498		0.104	0.108	67.5	122.5131116	55.01311	8.317982
4	0.35	0.275	650	68.0678		0.104	0.108	67.5	132.5101815	65.01018	9.829539
4	0.35	0.275	675	70.6858		0.104	0.108	67.5	142.8992933	75.39929	11.40037
4	0.35	0.275	700	73.3038		0.104	0.108	67.5	153.6804471	86.18045	13.03048

3) Table of results for Task 2(6)

No of shoes	μ	m (kg)	Speed (RPM)	ω (rad/s)	rd (m)	rod (m)	Fs	Fc (N)	Fn (N)	T (N/m)
4	0.3	0.275	450	47.1239	0.104	0.108	67.5	63.51079703	-3.989203	-0.517001
4	0.3	0.275	475	49.7418	0.104	0.108	67.5	70.76357323	3.263573	0.422959
4	0.3	0.275	500	52.3598	0.104	0.108	67.5	78.4083914	10.90839	1.413728
4	0.3	0.275	525	54.9778	0.104	0.108	67.5	86.44525151	18.94525	2.455305
4	0.3	0.275	550	57.5958	0.104	0.108	67.5	94.87415359	27.37415	3.54769
4	0.3	0.275	575	60.2138	0.104	0.108	67.5	103.6950976	36.1951	4.690885
4	0.3	0.275	600	62.8318	0.104	0.108	67.5	112.9080836	45.40808	5.884888
4	0.3	0.275	625	65.4498	0.104	0.108	67.5	122.5131116	55.01311	7.129699
4	0.3	0.275	650	68.0678	0.104	0.108	67.5	132.5101815	65.01018	8.42532
4	0.3	0.275	675	70.6858	0.104	0.108	67.5	142.8992933	75.39929	9.771748
4	0.3	0.275	700	73.3038	0.104	0.108	67.5	153.6804471	86.18045	11.16899

No of shoes	μ	m (kg)	Speed (RPM)	ω (rad/s)	rd (m)	rod (m)	Fs	Fc (N)	Fn (N)	T (N/m)
4	0.4	0.275	450	47.1239	0.104	0.108	67.5	63.51079703	-3.989203	-0.689334
4	0.4	0.275	475	49.7418	0.104	0.108	67.5	70.76357323	3.263573	0.563945
4	0.4	0.275	500	52.3598	0.104	0.108	67.5	78.4083914	10.90839	1.88497
4	0.4	0.275	525	54.9778	0.104	0.108	67.5	86.44525151	18.94525	3.273739
4	0.4	0.275	550	57.5958	0.104	0.108	67.5	94.87415359	27.37415	4.730254
4	0.4	0.275	575	60.2138	0.104	0.108	67.5	103.6950976	36.1951	6.254513
4	0.4	0.275	600	62.8318	0.104	0.108	67.5	112.9080836	45.40808	7.846517
4	0.4	0.275	625	65.4498	0.104	0.108	67.5	122.5131116	55.01311	9.506266
4	0.4	0.275	650	68.0678	0.104	0.108	67.5	132.5101815	65.01018	11.23376
4	0.4	0.275	675	70.6858	0.104	0.108	67.5	142.8992933	75.39929	13.029
4	0.4	0.275	700	73.3038	0.104	0.108	67.5	153.6804471	86.18045	14.89198