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ΔΙΠΛΩΜΑΤΙΚΗ ΕΡΓΑΣΙΑ

Design of a Magnetorheological Fluid Clutch

ΚΟΥΡΤΗΣ ΛΑΜΠΡΟΣ

[1069357]

ΠΑΝΤΕΛΗΣ ΝΙΚΟΛΑΚΟΠΟΥΛΟΣ –ΚΑΘΗΓΗΤΗΣ

ΠΑΤΡΑ,

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Η παρούσα διπλωματική εργασία παρουσιάστηκε

από τον

ΛΑΜΠΡΟΣ ΚΟΥΡΤΗΣ

1069357

την [ημέρα, μήνα, έτος]

Η έγκριση της διπλωματικής εργασίας δεν υποδηλοί την αποδοχή των γνωμών του συγγραφέα. Κατά τη συγγραφή τηρήθηκαν οι αρχές της ακαδημαϊκής δεοντολογίας.

Σχεδιασμός Μαγνητορεολογικού Συμπλέκτη

ΚΟΥΡΤΗΣ ΛΑΜΠΡΟΣ-1069357

ΕΠΙΒΛΕΠΩΝ:ΠΑΝΤΛΗΣ ΝΙΚΟΛΑΚΟΠΟΥΛΟΣ - ΚΑΘΗΓΗΤΗΣ

ΠΕΡΙΛΗΨΗ

Οι συμπλέκτες είναι το στοιχειό μηχανής που συνδέει την κινητήρια μονάδα ενός μηχανολογικού συστήματος με το υπόλοιπο σύστημα. Κατά τη διάρκεια της σύμπλεξης, ο οδηγός δίσκος, ή δίσκοι μεταφέρει/ρουν την ενέργεια του κινητήρα μέσω της ροπής που στρέψης που ασκούν στον/στους οδηγούμενο/ους δίσκο/δίσκους. Ένας μαγνητορεολογικός συμπλέκτης χρησιμοποιεί μαγνητορεολογικά ρευστά για να επιτύχει τη σύμπλεξη και τη μεταφορά ροπής. Τα μαγνητορεολογικά ρευστά είναι υγρά στα οποία βρίσκονται βυθισμένα φερομαγνητικά σωματίδια. Όταν τα ρευστά βρεθούν εντός μαγνητικού πεδίου τα σωματίδια σχηματίζουν αλυσίδες στη διεύθυνση του πεδίου, αλλάζοντας δραστικά τις ιδιότητες του ρευστού. Η μεγαλύτερη αλλαγή έχει να κάνει με το όριο ροής. Τα ρευστά αυτά, ρέουν αφού έχει ξεπεραστεί ένα συγκεκριμένο όριο ροής, αυξάνοντας έτσι τη διατμητική τάση που ασκούν. Το μοντέλο ροής για αυτά τα έξυπνα ρευστά είναι το μοντέλο του Bingham. Ένας μαγνητορεολογικός συμπλέκτης δημιουργεί μέσω ηλεκτρομαγνητισμού ένα εξωτερικό μαγνητικό πεδίο, με σκοπό να διεγείρει το ρευστό και να μεταφέρει την απαιτούμενη ροπή. Στη παρούσα εργασία εξετάστηκε ένας μαγνητορεολογικός συμπλέκτης σε διάφορες γεωμετρικές διαμορφώσεις και ρεύματα εισόδου. Αρχικά επιλύθηκε το μαγνητικό πρόβλημα χρησιμοποιώντας το λογισμικό ANSYS Magnetostatic. Στη συνέχεια αφού το μαγνητικό πεδίο έγινε γνωστό, χρησιμοποιήθηκε προκείμενου να βρεθεί το όριο ροής του ρευστού. Τέλος πρόβλημα ροής του ρευστού επιλύθηκε με τη χρήση του ANSYS Fluent. Μέσω της διατμητικής τάσης του ρευστού βρέθηκε η ροπή που μπορεί κάθε φορά να μεταφέρει ο συμπλέκτης. Έπειτα κατασκευάστηκαν οι καμπύλες Ροπή-Ρεύμα για κάθε διαμόρφωση. Επίσης κατασκευάστηκαν καμπύλες Ροπή-Ακτίνα Δίσκου για να βρεθεί μια βέλτιστη διαμόρφωση. Τέλος υπολογίστηκε η ενέργεια που καταναλώνει ο συμπλέκτης. Συμπερασματικά οι μαγνητορεολογικοί συμπλέκτες μπορούν ελεγχόμενα να μεταφέρουν μεγάλα ποσά ροπής.

Λέξεις Κλειδιά: Μαγνητορεολογικος Συμπλέκτης, Όριο ροής, Μεταφερόμενη Ροπή, Μοντέλο Bingham, Μαγνητορεολογικά Ρευστά

Design of a Magnetorheological Fluid Clutch

KOURTIS LAMPROS-1069357

SUPERVISOR: PANTELIS NIKOLAKOPOULOS – PROFESSOR

ABSTRACT

Clutches are the machine elements which connect the engine of a system with the rest of the system. During the engagement, the driving disc or discs transfer the power of the engine to the driven disc through the torque. A Magnetorheological Fluid Clutch utilizes magnetorheological fluid to achieve the engagement and the torque transfer. Magnetorheological fluids are liquids in which ferromagnetic particles are immersed. When the fluids are placed within a magnetic field the particles form chains in the direction of the field, drastically changing the properties of the fluid. The biggest change has to do with the yield stress. These fluids flow after a certain yield stress has been exceeded, thus increasing the shear stress they exert. The flow model for these smart fluids is Bingham's model. In a MRF clutch an external magnetic field is generated in order to stimulate the MRF. In this diploma thesis, a magnetorheological clutch in various geometric configurations and input currents has been investigated. Initially the magnetic problem was solved using ANSYS Magnetostatic software. Then after the magnetic field was known, it was used to find the yield stress. Finally fluid flow problem was solved using ANSYS Fluent. The shear stress of the MRF used to calculate the torque the clutch can transfer. Then the Torque Current curves are made for every configuration. Moreover Torque Radius curves were made in order to determine the best configuration. Finally it was calculated the power consumption for each configuration. In conclusion, the MRF clutches can controllably transfer large amount of torque.

Keywords: Magnetorheological Fluid Clutch, Yield Stress, Transferred Torque, Bingham Model, Magnetorheological Fluid

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NOMECLATURE

τ_y……………………………………Yield stress (Pa)

τ……………………………………Shear Stress (Pa)

ω_1……Driving Disc Angular Velocity (rad/s)

ω_2…….Driven Disc Angular Velocity (rad/s)

Δω……..Angular Velocity Difference (rad/s)

r………………………………………Disc Radius (m)

r_{in}……………………………Inner Disc Radius (m)

r_{out}…………………………Outer Disc Radius (m)

μ………………..Newtonian Viscosity (Pa*sec)

ρ………………………………MRF density (kg/m^3)

h……………………………….MRF gap width (m)

$\dot{\gamma}$……………………………………Shear Rate(1/s)

P………………………………Fluid’s Pressure (Pa)

T………………………………………….Torque (Nm)

B…………………….Magnetic Flux Density (T)

I……………………………….Electric Current (A)

J……..……….Electric Current Density (A/m^2)

H……………..Magnetic Field Intensity (A/m)

A……………………..Wire’s Cross Section (m^2)

N……………………………………………Coil Turns

σ…………………….Wire’s Conductivity (S/m)

μ_0……………….Magnetic Permeability (H/m)

μ_0…………….Relative Magnetic Permeability

P_c…………………Power Consumption (Watt)

MRF………………..Magnetorheological Fluid

R…………………Coil’s Internal Resistance (Ω)

h_r……MRF gap width inside the groove (m)

a…………………………Height of the Groove (m)

τ_{yr}………..Yield Stress inside the Groove (Pa)

λ……………………Energy Coefficient (Nm/P_C)

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CHAPTER 1

INTRODUCTION

Clutches are machine elements which connect the engine of a system with the transmission system. During the operation of some mechanical systems it may occur the need to interrupt its operation for a while, however it is not desirable the shut off of the engine. This problem is solved using clutches. Clutch works as a switch. When clutch is on the engagement state, it allows the energy the engine produces to ‘’pass’’ to the transmission system, while on the disengagement state the energy does not flow to the transmission system. A classic example of a clutch function is the clutch of car. During driving the clutch engages engine and transmission system. However when the driver wants to change gear, presses the clutch pedal, disengages the engine so can change the gear and then he releases the clutch pedal to reengage engine with transmission system.

TYPES OF CLUTCHES

Many types of clutches exist. The suitable type of clutch depends on the demands of the system they are going to be part of it. The vast majority of clutches try to utilize the friction which is created between two contacted surfaces. For this reason most of the clutches are consisted of two or more discs which during the engagement contact each other. By this contact the necessary friction is generated which transfers the power and torque from the engine to the rest of the system. The simplest configuration consists of just 2 discs. This configuration includes 2 friction surfaces. However on applications with higher needs for transferred torque there are configurations which include more than two discs.

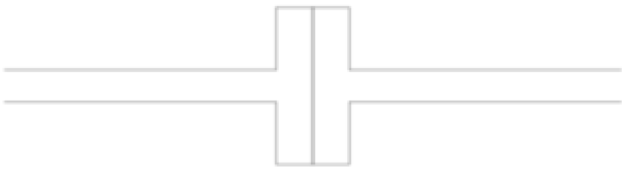


Fig1 Simple Display of a Friction Clutch with 2 Discs on Engagement

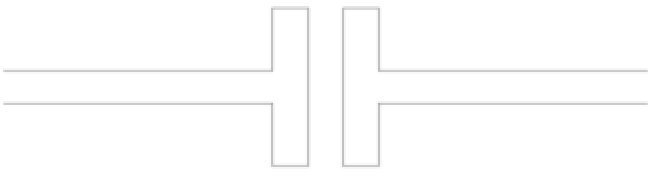


Fig 2 Simple Display of a Friction Clutch with 2 Discs on Disengagement

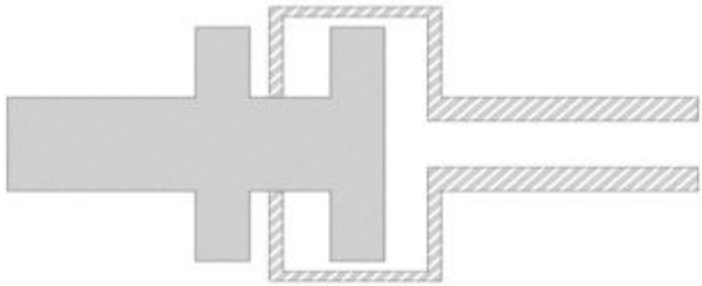


Fig 3 Simple Display of a Friction Clutch with 4 discs (2 for input and 2 for output)

Other friction clutch configuration is the cone clutch. Cone clutch is made of by 2 parts, one ''male'' and one ''female'' part. During the engagement the ''male'' part inserts into the ''female'' part and their surfaces come to contact, generating the necessary friction, for transferring the torque (**15**).

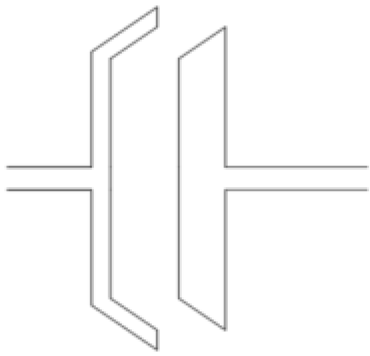


Fig 4 Cone Clutch

One more friction clutch configuration is the centrifugal clutch. This configuration includes an outer ring which is connected with the driven shaft. On the interior of the ring there are the friction blocks. Friction blocks, are connected with the driven shaft. Those blocks are held on their positions with coils. As the rotating speed of the driving shaft is getting larger the centrifugal force the blocks experience is increasing too. On some velocity, the centrifugal force becomes larger than the coils force and the blocks touch the ring. Through this contact torque is generated which transfers the power from driving to the driven shaft (**15**).

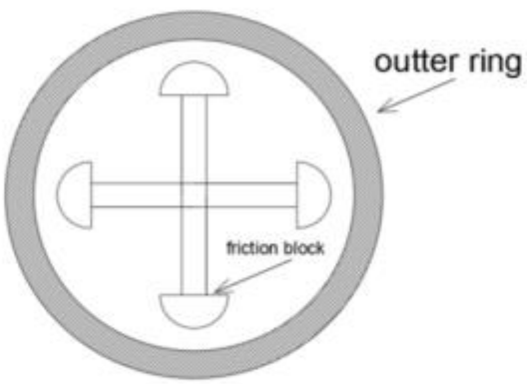


Fig 5 Centrifugal Clutch

In cases where high temperatures are generated, wet clutches are used. A wet clutch has the same configuration as a friction clutch with 2 or more discs. However in the gap between the discs there is lubricant oil. The presence of oil contributes to the decreasing of the temperature. However this comes with the cost of less transmitting power. The fluid decreases the friction coefficient of the discs, so during their contact the friction is not as high as on dry touch. Other way to transfer power via fluids is the hydraulic clutch. Hydraulic Clutches are divided into 2 categories, fluid couplings and torque converter. Fluid coupling uses a lubricant oil to transfer the power. The input is an impeller, which increases the energy of the fluid, while at the exit there is a turbine which extracts the energy from the fluid. Torque converter works at same way. The difference between the 2 types is that in torque converter there is an extra part between the impeller and the turbine, the stator. Stator is a stationary part with fins. Its fins change the direction of the fluid, while returns from turbine to the impeller, and lead it to the outer of the configuration. This gives to the fluid more energy so the power in the exit is increased (**15**).

Additional clutch's configurations include the magnetic clutch, which uses magnetic force to bring in contact the discs and the hysteresis powered clutch which uses magnetic force to rotate the driven shaft (**15**).

Last major type of clutch is the magnetic particle clutch. This type of clutch achieves engagement with the usage of materials which change their properties under the influence of magnetic field. Those materials may be ferromagnetic powder or magnetorheological fluids or other materials with controllable properties (**15**).

MAGNETORHEOLOGICAL FLUIDS

Magnetorheological Fluids (MRFs) are a subcategory of the ''smart'' fluids. They are made by fluid (carrier) in which ferromagnetic particles are submerged. The carrier fluid is typical oil, synthetic oil water or glycol (https://www.shoplordmr.com/). The carrier fluid itself has no magnetic properties. In absence of an external magnetic field, the fluid behaves as a typical Newtonian fluid and the ferromagnetic particles are randomly distributed inside the fluid currier. However, when the fluid is inside of a magnetic field, the ferromagnetic particles form ¨chains¨ in the direction of the magnetic lines of the field. That is portrayed in **Fig 6** and **Fig 7**. By this procedure the fluid's yield stress and its viscosity change drastically and fluid inserts into a solid like situation. That means that by applying an external magnetic field and varying it, we can controllably change the properties of those fluids, bringing them on a desired state. The relation between the critical yield stress and the magnetic field intensity is founded by experimental data. Some manufacturers provide the B-H curve and the τ_y-H curve too. Moreover the whole procedure can be reversed. The MRF can return to its previous state immediately, after the applied magnetic field ceases. The transition from the liquid phase to the solid like and reverse takes place in few milliseconds. According to (**17**) for the MRF 132 DG of Lord the response time is 0.8-1.4ms. Those attributes of MRFs make them eligible for a variety of applications. Dezheng Hua , Xinhua Liu, Zengqiang Li , Pawel Fracz 3, Anna Hnydiuk-Stefan and Zhixiong Li on their work (**11**), refer many applications where MRF are used, such as dampers, brakes, bearings, in surfaces finishing, medical applications and of course clutches.

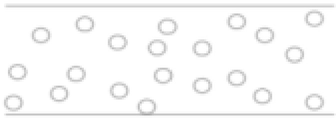


Fig 6 Ferromagnetic Particles of MRF inside the fluid in absence of Magnetic Field

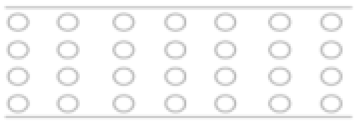


Fig 7 Ferromagnetic Particles of MRF inside the fluid in presence of Magnetic Field

MAGNETORHEOLOGICAL FLUID CLUTCH

MRF clutches belong to the type of Magnetic Particles Clutch. MRF clutches consist of at least two or more friction discs. Between the discs, there is a gap which is filled with the MRF. In the disengagement state, no magnetic field is applied to the clutch and the MRF flows like Newtonian fluid. During the engagement, an external magnetic field is applied and then the MRF enters the semi-solid condition. The input discs rotate and by this rotation the MRF is sheared. By this way, the torque is transferred from the driving shaft to the driven shaft. The magnetic field is created by exploiting the fundamental principle of the electromagnetism; when current passes through a wire, magnetic field is generated around the wire. For that reason, the coil is vital part for the function of a MRF clutch. Current passes through the coil and the magnetic field is created. When it is desired for the clutch to enter the disengagement state, the current stops passing form the coil, the magnetic field disappears and the MRF returns to its fluid state. A solenoid coil with its core is around the discs. Usually the discs are in the internal of the solenoid. However, there are configurations where the coil is in a different position, for example in the external side of one disc (Jin Huang , Wenjian Chen , Ruizhi Shu and Jing Wei) (4). Other configurations include more than one coil in different angles around the gap (Nicholas Bira1 , Pallavi Dhagat2 and Joseph R. Davidson) (6), in an effort to reduce the power required by the coil. Also (Emmanuel Mbondo Binyet Jen-Yuan Chang) (9) have proposed a configuration with a permanent magnet and coils. The coils, on that case, create a counter field during the disengagement state, in order to cancel the permanent field. In (**16**) Manish Kumar Thakur, Chiranjit Sarkar tried various groove patterns in an attempt to enhance the field in the working gap.

Other parts of the clutch are the bearings which support the two shafts. The whole device is inside a cover to protect it from damage and also to reduce the intensity of the field outside the clutch.

The main advantage of the MRF clutches is that their exit torque is absolutely controlled. Without changing any other system's parameter, such as input velocity, number of discs, radius of discs, gap between the discs, materials etc, the clutch provides the desired torque and power to its exit. The only parameter is needed to be varied is the magnetic field. Magnetic field is directly linked to the input current, so by just varying input current, we can control the torque in the output. Furthermore the fast response time of the MRF to the changes of the field makes the procedure really fast. Moreover in a MRF clutch the discs do not need to move to each other to accomplish engagement or disengagement, like in a traditional friction clutch. This makes the system simpler and more reliable.

OBJECTIVES OF DIPLOMA THESIS

This diploma thesis investigates the behavior of the MRF clutch under certain currents. Then the behavior of the clutch is investigated in various MRF gaps and in various discs radius. Also it is examined the response of the clutch in high values of currents. Moreover it will be examined with grooved surfaces in the discs and finally it is calculated the power the clutch consumes. The numerical solution of the problem is achieved using ANSYS Magnetostatic for the magnetic field and ANSYS Fluent for the flow of the MRF. This diploma thesis wills exam the MRF clutch in order to define:

1. Exit torque for different current inputs
2. Exit torque for different distances between the discs
3. Exit torque for different discs radius
4. Exit torque for high values of current, in an effort to approach saturation point for all examined radius and gaps
5. Exit torque for grooved surfaces of discs
6. Power consumption of the coil in each configuration

Objective of this diploma thesis is to contribute to an easier and more effective choice of parameters in a MRF clutch design, such as geometry, dimensions and current inputs.

CHAPTER 2

ASSUMPTIONS

To apply the governing equations and obtain the final results, some assumptions have been made. A real world problem is not ideal and those assumptions may not be valid.

1. The MRF gap has been divided into zones. In each zone the magnetic field is constant.
2. The flow of the MRF in each zone is independent and it is not affected by the flow in neighboring zones.
3. The problem is considered isothermal. This means that the magnetic properties of the used materials are constant and the viscosity of the MRF is a function only of the magnetic field.
4. It is supposed uniform distribution of the ferromagnetic particles, so the MRF has the same properties in each point.
5. The gap is fully filed with the MRF. No air is present.
6. Radial clearness is not taken into consideration.
7. The friction surfaces are totally smooth, no roughness profile exists.
8. No slip condition is considered. The fluid has the same velocity as the discs.
9. During the CFD analysis the field forces of the fluid have been neglected.
10. $\frac{h}{r} \ll$ meaning that Pressure and Magnetic Field do not vary in the direction of h

GOVERNING EQUATIONS

The problem is divided in 2 parts. The one part is the flow problem and the other part is the magnetic problem.

For the analysis of the magnetic field the laws of Maxwell for the magnetic field have been used.

The Gauss law is the following

$\vec{\nabla}\vec{B} = 0$ (1)

The Ampere's law with Maxwell's correction is:

$\vec{\nabla} \times \vec{H} = \mu_0 \vec{J} + \mu_0 \varepsilon_0 \frac{\theta E}{\theta t}$ (2)

$\vec{B}$ and $\vec{H}$ are related with the equation:

$\vec{B} = \mu_0 \mu_r \vec{H}$ (3)

Electric current density J is the input for the Ansys magnetostatic. J is given by the equation:

$J = \frac{NI}{A}$ (4)

With N is the turns of the coil, I is the current and A is the cross section of the coil's wire.

The flow problem can be modeled as flow between 2 rotating discs.

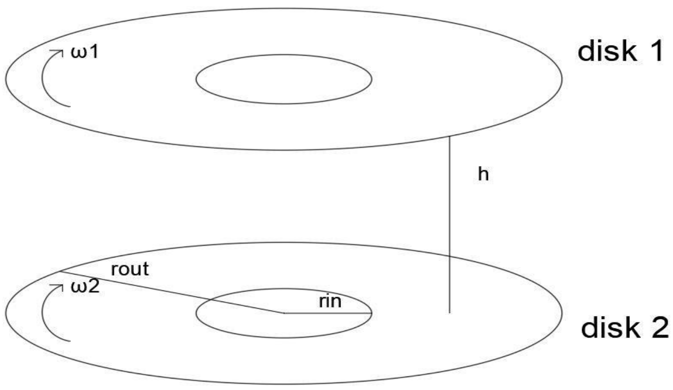


Fig8 Model of flow between 2 rotating discs

The flow of the MRF is described by the Navier – Stokes equations. The momentum conservation equation is the following:

$$\frac{\theta(\rho\vec{V})}{\theta t}+\vec{\nabla}(\rho\vec{V}\vec{V})=\rho\vec{g}+\vec{F}+\vec{\nabla}\sigma_{ij} \text{ (5)}$$

With $\rho\vec{g}$ being the gravitational force the MRF experiences and $\vec{F}$ being other external forces.

σ_{ij} is the stress tensor which is given by the following equation:

$$\sigma_{ij}=\begin{bmatrix}-P & 0 & 0\\ 0 & -P & 0\\ 0 & 0 & -P\end{bmatrix}+\begin{bmatrix}\tau_{xx} & \tau_{xy} & \tau_{xz}\\ \tau_{yx} & \tau_{yy} & \tau_{yz}\\ \tau_{zx} & \tau_{zy} & \tau_{zz}\end{bmatrix} \text{ (5a)}$$

The mass conservation equation is:

$$\frac{\theta\rho}{\theta t}+\vec{\nabla}(\rho\vec{V})=0 \text{ (6)}$$

The MRF behaves as Newtonian fluid when no external magnetic field is applied and it obeys the Newton's law for viscosity $\tau=\mu\dot{\gamma}$ (7) with $\dot{\gamma}$ being the shear rate and μ describes the Newtonian viscosity. For flow between rotating discs the $\dot{\gamma}$ is: $\dot{\gamma}=\frac{\Delta\omega}{h}r$ (8).

So the viscosity of the shear stress of MRF when no external magnetic field is applied is: $\tau=\mu\frac{\Delta\omega}{h}r$ (9)

However when the magnetic field is applied the MRF stops behaving as Newtonian fluid. It has been shown that the Bingham rheology model is which describes the flow of the MRF in the best way. The equation of the Bingham model is the following:

$$\tau=\tau_y(B)+\mu\dot{\gamma} \text{ (10)}$$

Substituting Eq. (4) to (6) the Bingham Equation becomes:

$\tau = \tau_y(B) + \mu\frac{\Delta\omega}{h}r$ (11)

T_y(B) is the critical yield stress of the fluid, as a function of the magnetic field.

The resulting torque of the clutch is given by the following equation:

$T = \int r\tau(r)dA$ (12)

With dA=2πrdr the torque equation, for Bingham flow becomes:

$T = 2\pi\int_{r_{in}}^{r_{out}}(\tau_y(B)r^2 + r^2\frac{r\Delta\omega\mu}{h})dr$ (13).

By integrating the equation, it is obtained the final equation for the torque, which is: $T = \frac{2\pi\tau_y}{3}\left(r_{out}^3 - r_{in}^3\right) + \frac{\pi\mu\Delta\omega}{2h}(r_{out}^4 - r_{in}^4)$ (14).

For Newtonian flow the torque equation becomes:

$T = 2\pi\int_{r_{in}}^{r_{out}} r^2\frac{r\Delta\omega\mu}{h}dr$ (15).

By integrating the equation, it is obtained the final equation for the torque, which is: $T = \frac{\pi\mu\Delta\omega}{2h}(r_{out}^4 - r_{in}^4)$ (16).

By comparing equations (14) and (16) it can be observed that the torque for Bingham flow is greater than the torque for Newtonian flow by a factor of$\frac{2\pi\tau_y(B)}{3}\left(r_{out}^3 - r_{in}^3\right)$.

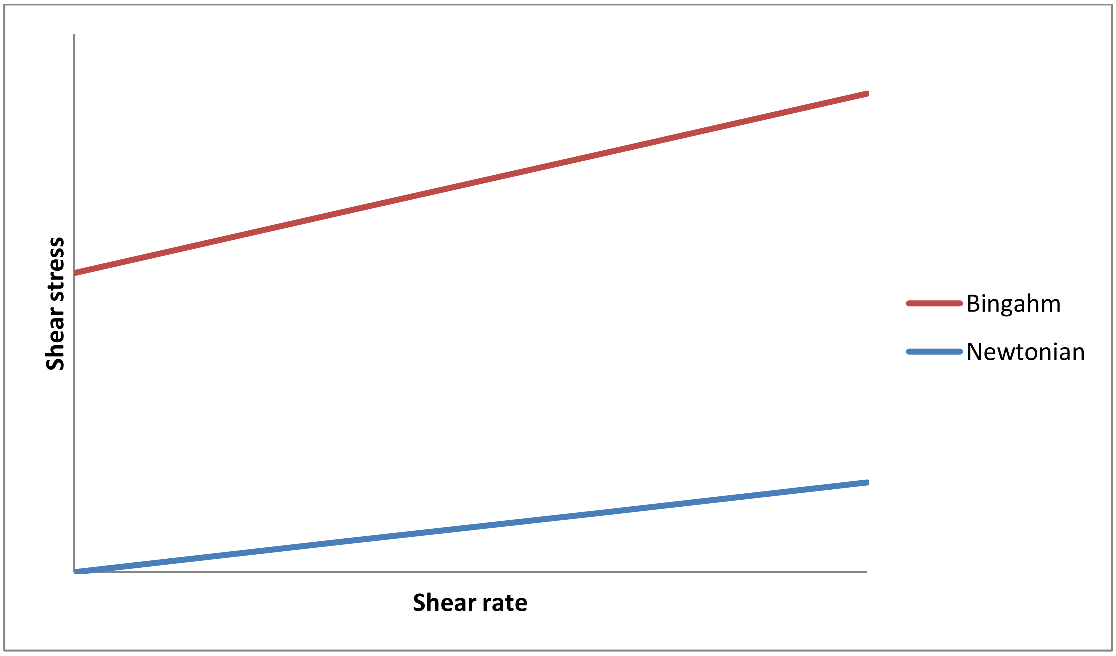
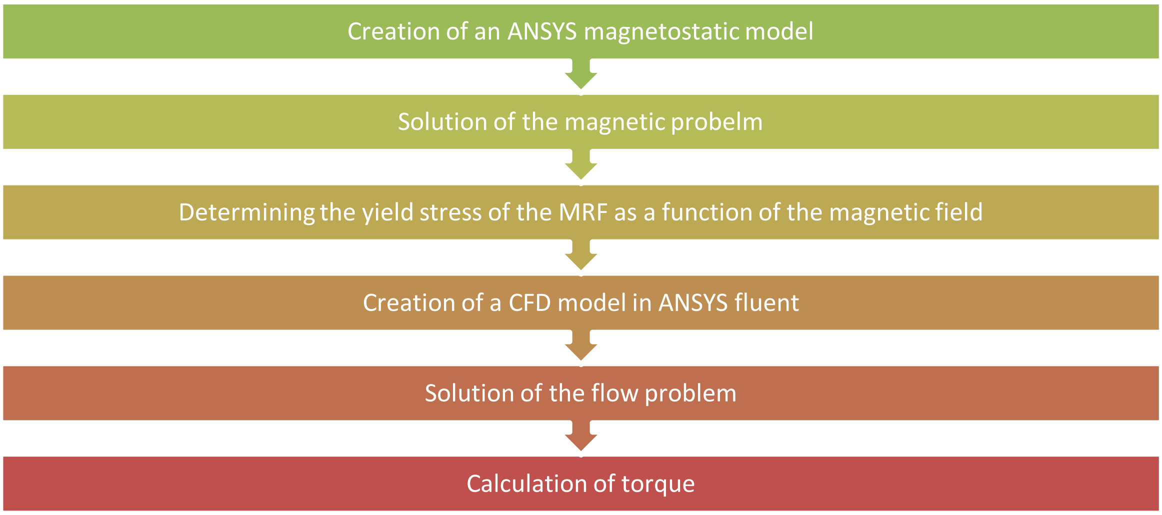


Fig 9 Comparison of Newtonian and Bingham model

For the power consumption of the coil it is used the relationship $P_c = I^2R$ (17). R is given by the formula:$R = \frac{L}{\sigma A}$ (18) where σ is the electrical conductivity of the coil and L is the length of the coil and A is the cross section of the coil's wire. The length of the coil is $L = N2\pi r$ (19) with N the number of turns. From the above equations the final relationship for the power consumption is: $P_c = \frac{N2\pi r}{\sigma A}$ (20).

SOLUTION METHOD

The problem, as it has already been mentioned, is a problem of two parts. Firstly the magnetic part of the problem must be solved, in order to determine the magnetic field in the clutch and more specific, the magnetic field intensity in the gap between the discs. Using the values for the magnetic field intensity, the yield stress of the MRF is determined. With the yield stress of the MRF known, it follows the solution of the flow problem. With shear stress known it can be calculated the torque of the clutch using equations (14) and (16). For the numerical solution of both problems it has been used the ANSYS software. For the magnetic problem, it was created a model of the clutch in ANSYS magnetostatic, while for the CFD problem it was created a model in ANSYS Fluent. In the following flow chart is shown detailed the process solution of the problem.



ANSYS MAGNETOSTATIC MODEL

To solve the magnetic problem, it is necessary to create a detailed model of a clutch. For this purpose, the clutch used by Kumar (2) is also used in the present diploma thesis. The geometry was created in ANSYS design modeler. A 2D draw of the clutch is shown in

Fig 10 2D draw of the used clutch

. The dimensions are in mm.

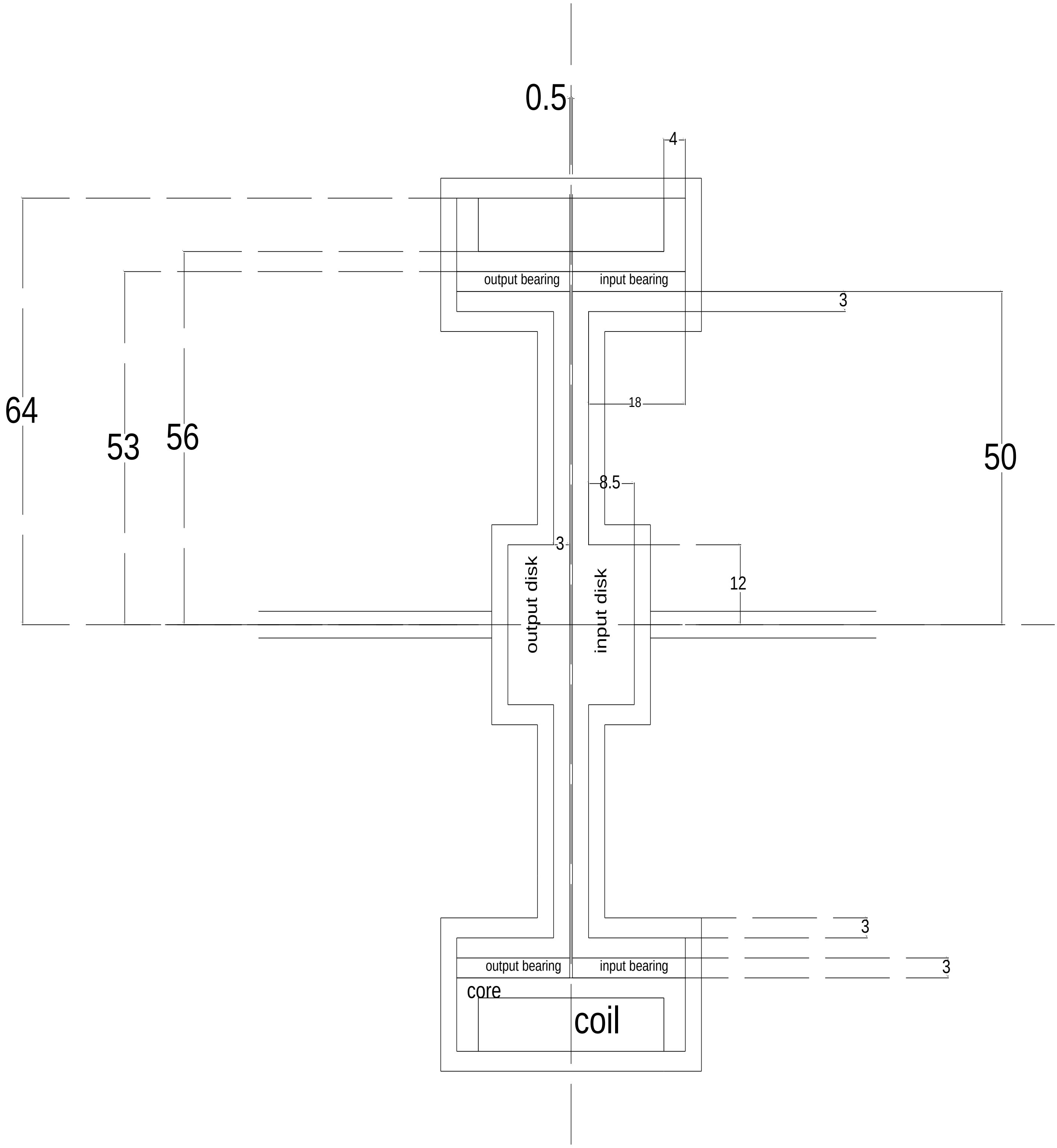


Fig 10 2D draw of the used clutch

The geometry is consisted of two discs, the input disc and the output disc. Between them there is the MRF gap. Both discs are supported by two bearings, one for each disc. In the bearings outer side stands the coil's core. The devise is enclosure by cover. The used materials are inserted in the ANSYS model from ANSYS libraries. The MRF used is the MRF 132 DG of the LORD. The magnetic properties of the MRF (curve B-H) are provided by the manufacturer (**14**). For the solution of the problem a 3D model of the clutch was created in ANSYS magnetostatic (**Fig 11** The ANSYS magnetostatic geometry) and (**Fig 12** Cross section of the geometry).

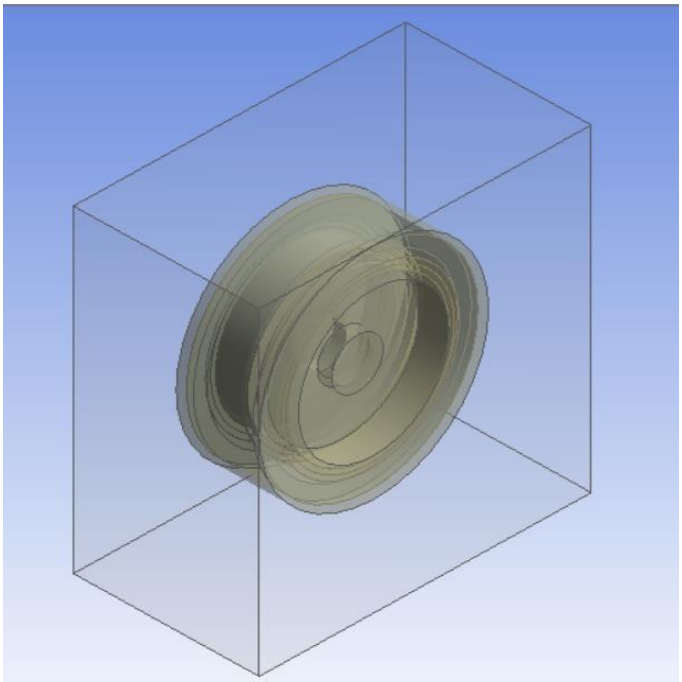


Fig 11 The ANSYS magnetostatic geometry

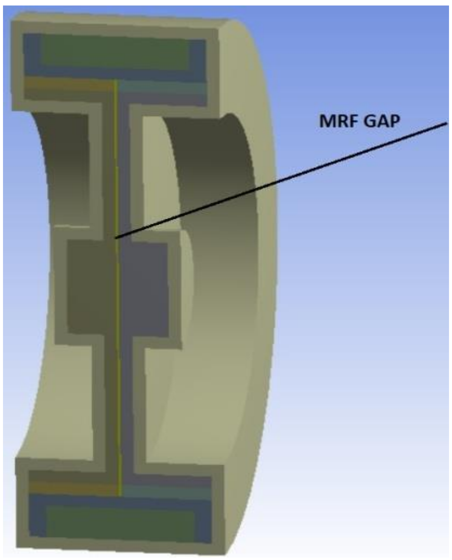


Fig 12 Cross section of the geometry

The materials from the ANSYS libraries used in the model are concentrated on the following table.

PART	MATERIAL
INPUT DISC	Cold low carbon steel strip
OUTPUT DISC	Cold low carbon steel strip
INPUT BEARING	Steel
OUTPUT BEARING	Steel
MRF	MRF 132 DG LORD
CORE	Aluminum Alloy
COIL	Copper Alloy
AIR	Air
COVER	Cold low carbon steel strip

Table 1 Materials of the clutch

The coil has 300 turns. The cross section of the wire is $4.1*10^{-7}$ m^2. The clutch is surrounded by air. The element size for meshing was chosen to be 0.005m. For the MRF gap, where it is needed more accuracy because it is the region of high interest, the element size was chosen to be 0.002mm. The result is a mesh of 409037 elements and 643123 nodes (**Fig 13** Meshed geometry (MRF gap)). The average skewness of the mesh is 0.36. Surrounding air is set as boundary condition. The magnetic field in the boundary is zero, meaning that any external magnetic fields are neglected. Input of the system was set the current of the coil. The current is inserted in the ANSYS magnetostatic model through the electric density (Eq 4).

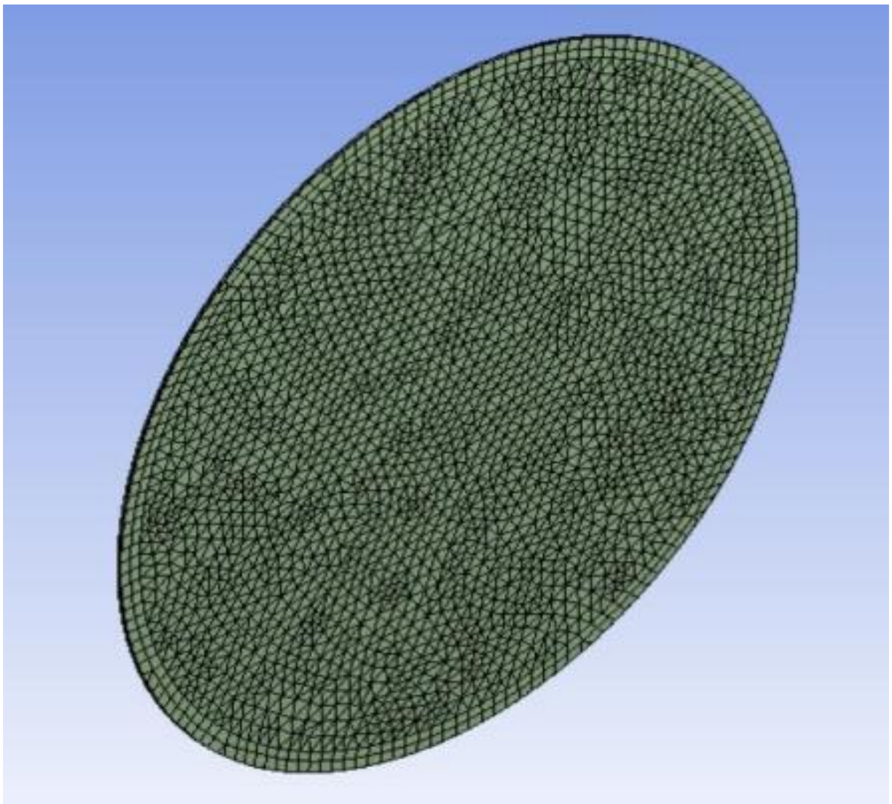


Fig 13 Meshed geometry (MRF gap)

VALIDATION

In the following Figures (**Fig 14**) and (**Fig 15**) is shown the magnetic field intensity and the magnetic flux in the MRF gap. It can be observed that both quantities are uniform distributed around the center of the gap.

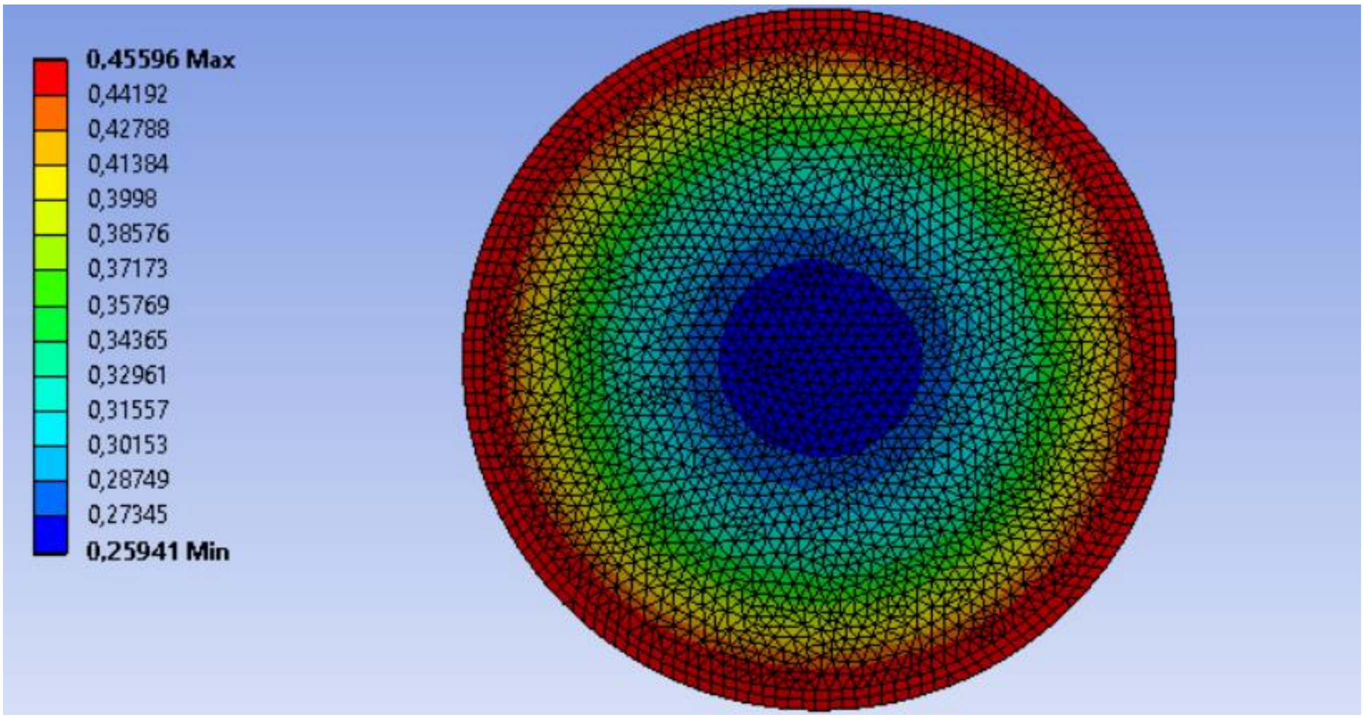


Fig 14 Total Magnetic Flux Density in MRF gap for I=1A

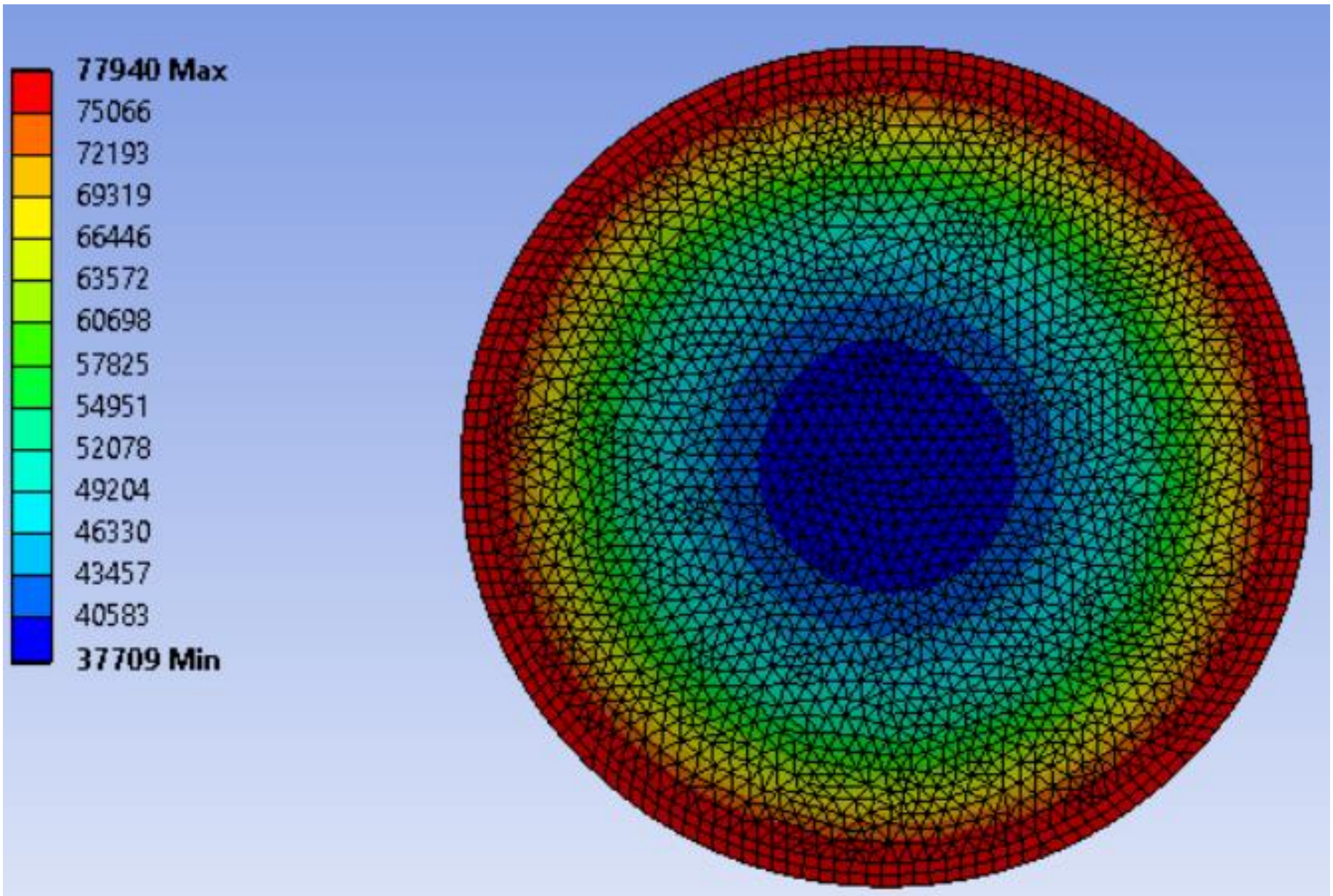


Fig 15 Total Magnetic Field Intensity in MRF gap for I=1A

The above results are validated from the Kumar (2) paper. In the next diagram is portrayed the magnetic flux density of the present thesis, for currents 1A and 2A and the magnetic flux density, for the same values of current of Kumar's paper (2).

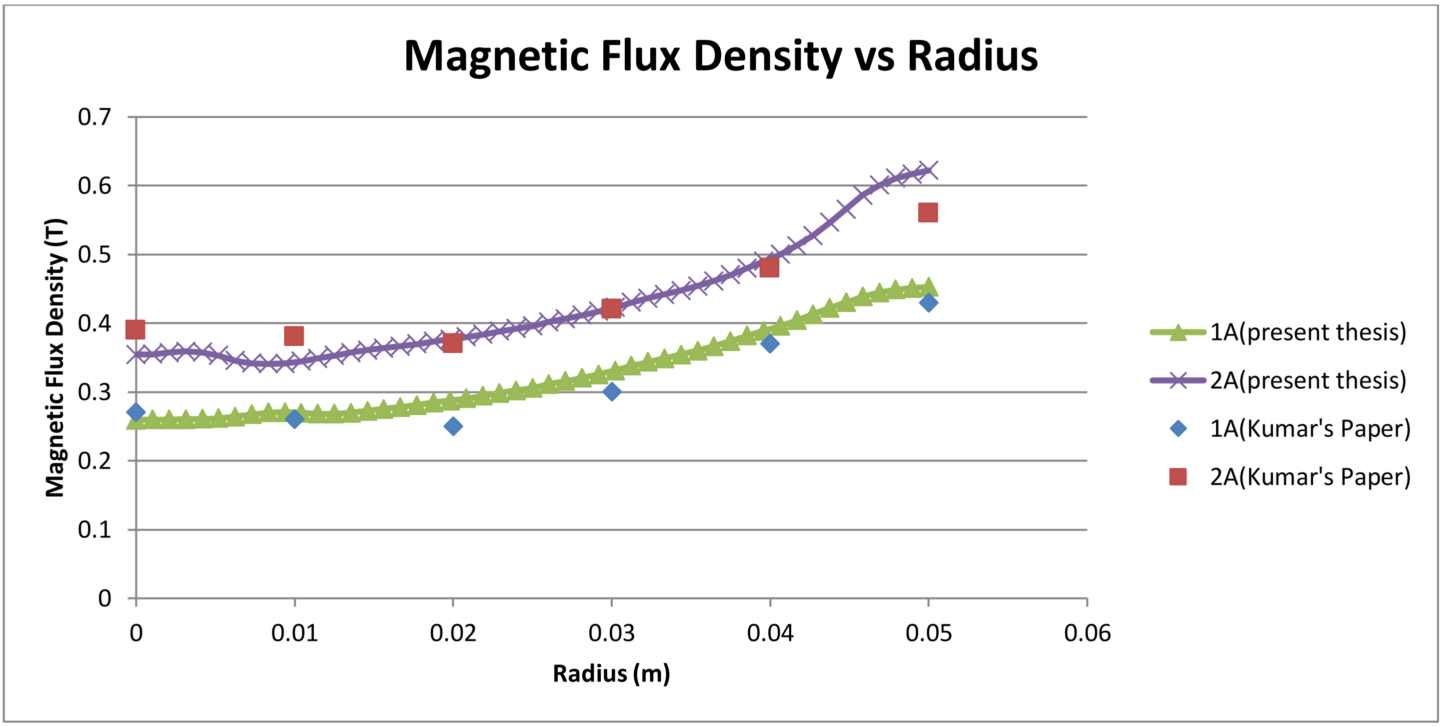


Fig 16 Validation of the ANSYS Magnetostatic Model

The reasons for the declinations between the present work and the Kumar's paper (2) are probably due to different meshing quality and method and in differences in the magnetic properties of the used materials, and especially in the MRF which is used in both occasions. However, in both works, the distributions of the magnetic field have the same shape.

ANSYS FLUENT MODEL

In (**Fig 14**) and (**Fig 15**) it is shown that the distribution of the magnetic field has a wide range. As a consequence, the yield stress of the MRF, which is a function of the field, has a wide range too. The assumption of constant yield stress in the whole flow domain would lead to significant errors in the calculation of the shear stress and the torque too. In order to avoid those mistakes, the MRF gap was divided in radial zones. The magnetic field is measured in 50 points, all equally distributed among the radius of the MRF gap. The division is taking place approximately every 10kA/m. In each zone it is considered that the magnetic field is constant. Each zone includes a number of the 50 points. The value of the field in each zone is average value of the measurement points included in each zone. That value is the input in the function which gives the yield stress. As a result, the yield stress in each zone is constant. Adopting this approach, the number of radial zones are needed in each case, varies.

For the solution of the flow problem it was created a 3D model of the flow domain in ANSYS Fluent. The geometry was created in ANSYS design modeler. As it can be shown in

Fig 10 2D draw of the used clutch

, the clutch has 2 discs with r_{in}=0mm, r_{out}=50mm and h=0.5mm. The flow domain is a cylinder with height equal to the MRF gap. The geometry is consisted of many independent zones, according to the needs for each case.

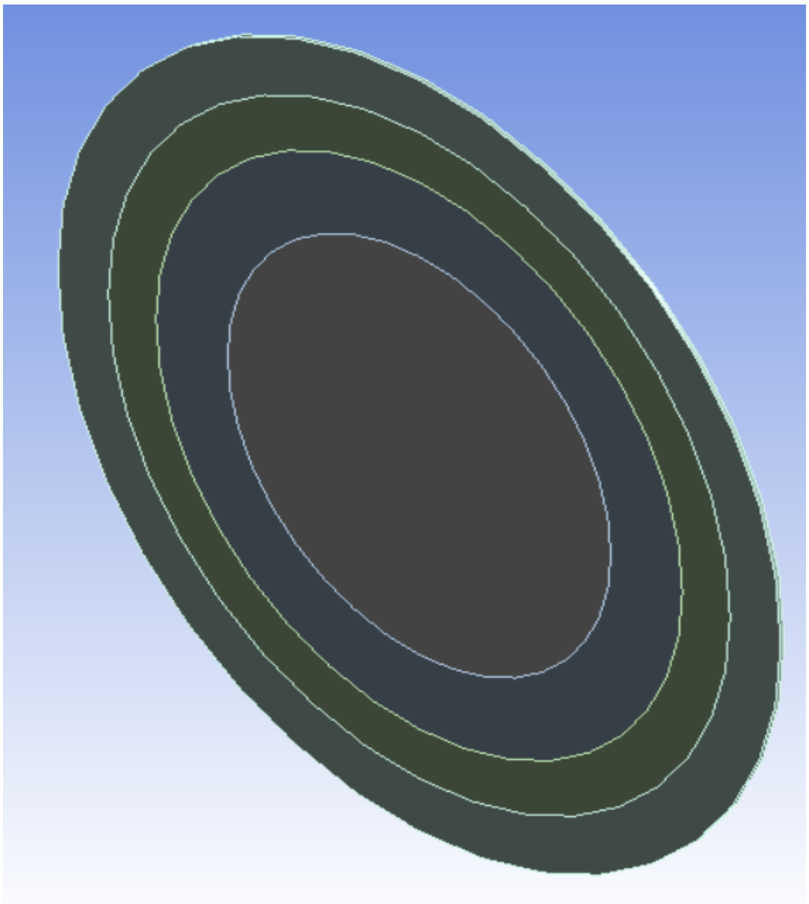


Fig17 The flow domain for I=1A

For the meshing, independently of the number of zones, it has been set element size of 0.0002m. The result is a mesh of 661233 elements and 895708 nodes (**Fig 18**). The average skweness is 0.11.

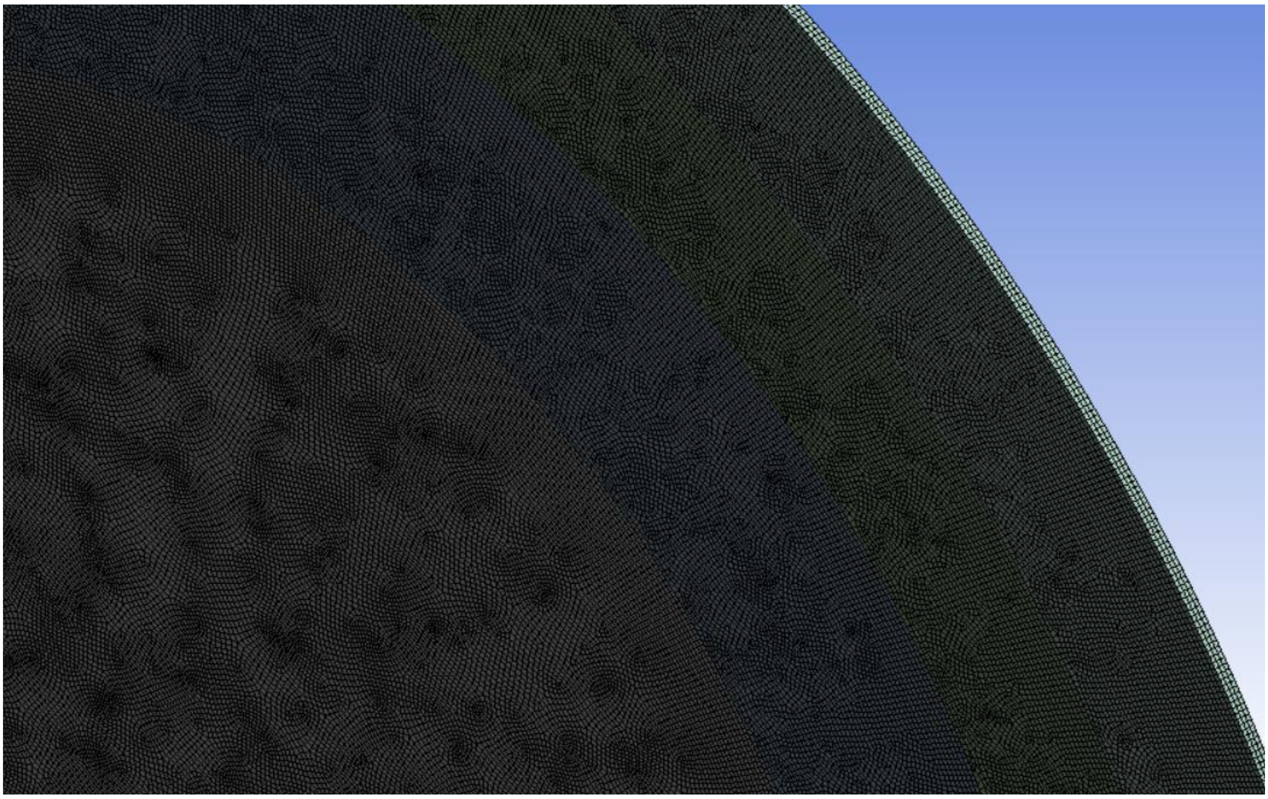


Fig 18 Meshed CFD geometry

For the boundary conditions, the input side is set as velocity inlet and the other 2 sides, output and wall, are set as stationary wall, as it is shown in **Fig 19**.

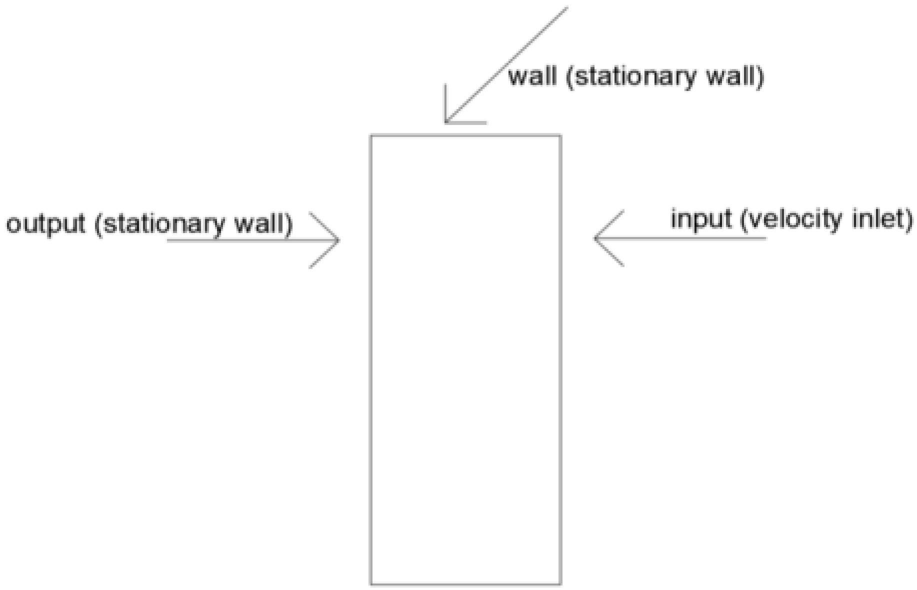


Fig 19 CFD model

The input velocity is 100 rad/s. The flow is laminar and the simulation is taking place in atmospheric pressure conditions. The fluid, as it has been already mentioned is the MRF 132 DG of LORD. The Newtonian viscosity of the fluid is 0.112 Pa*s and the density is

3050kg/m^3.The Bingham flow model is inserted in ANSYS through Herschel-Bulkley model by setting the exponential factor n equal to 1. Moreover, for ANSYS, the Bingham model is a model with 2 zones, one with plastic viscosity μ_p and one other with typical Newtonian viscosity μ_f. According to Bompos and Nikolakopoulos (**13**), the μ_p is: $\mu_p = 100\mu_f$ (21). Residuals for the simulation were set at 10^{-6} for all the equations.

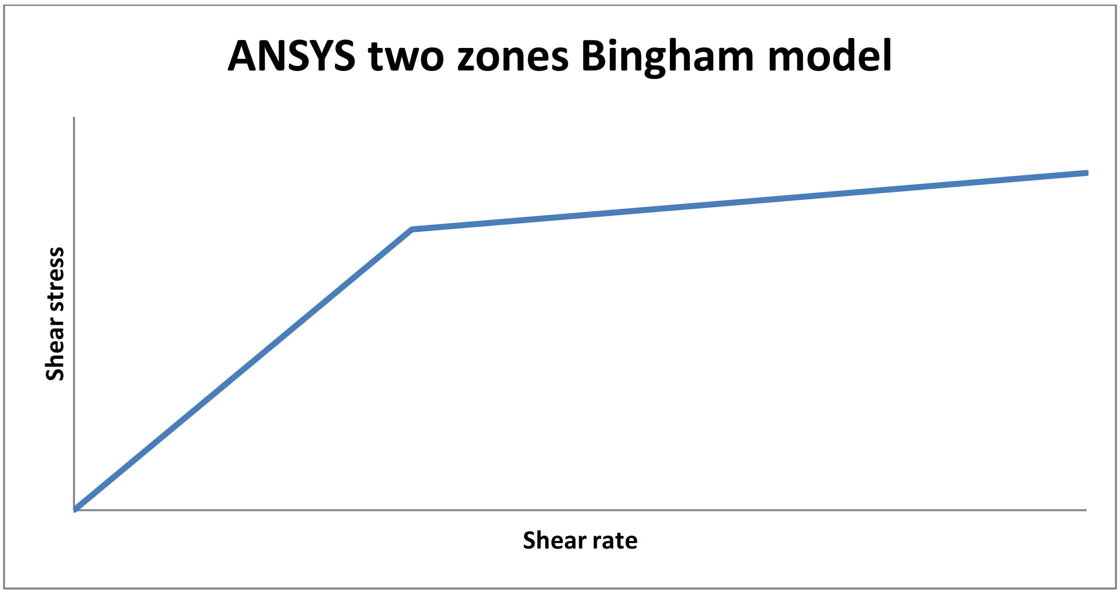


Fig 20 ANSYS two zones Bingham model

In the next table are shown the parameters used in the CFD model

r_{in}	0mm
r_{out}	50mm
h	0.5mm
Newtonian viscosity	0.112 Pa*s
Plastic viscosity	11.2 Pa*s
ρ	3050 kg/m^3
Input angular velocity	100 rad/s
Output angular velocity	0 rad/s

Table 2 Parameters of the CFD model

VALIDATION

In **Fig21** is shown the shear stress distribution among the surface of the output disc, for I=1A. It is uniformly distributed around the center of the disc. In **Fig 22** the shear stress is plotted as a function of the radius. They can be observed 4 distinctive areas in both Figures, as the zones the flow domain divided based on the magnetic field.

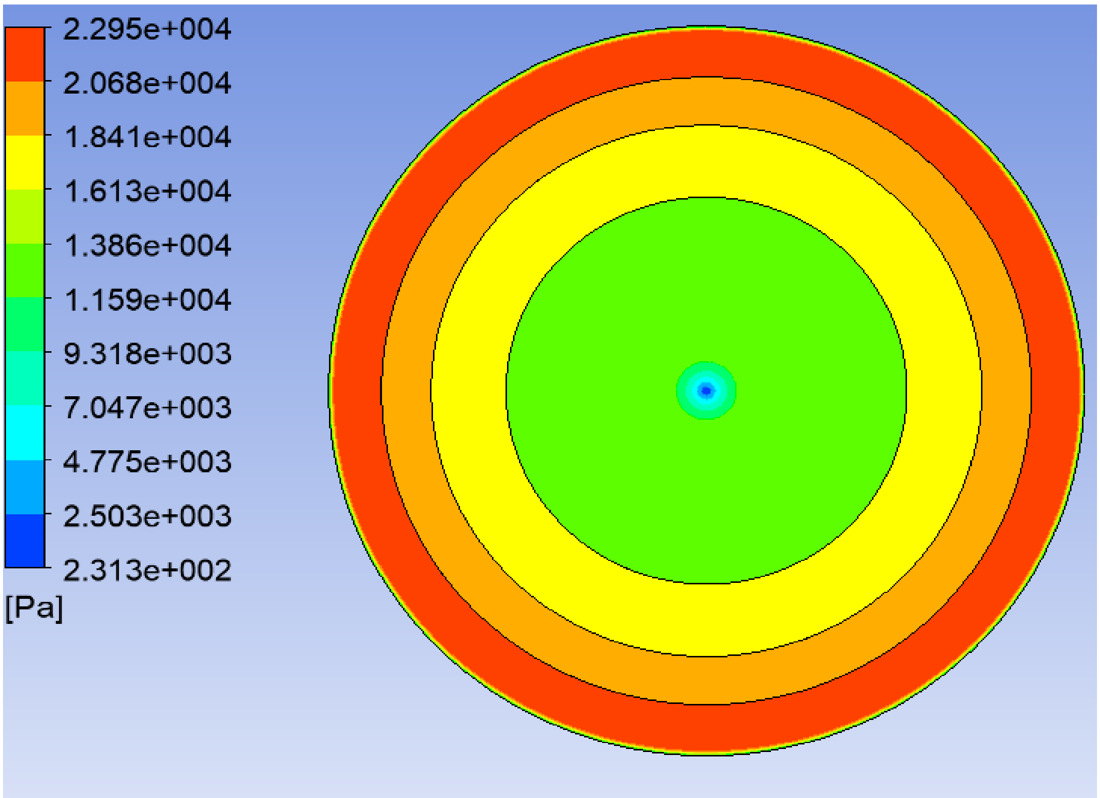


Fig21 Shear Stress for I=1A

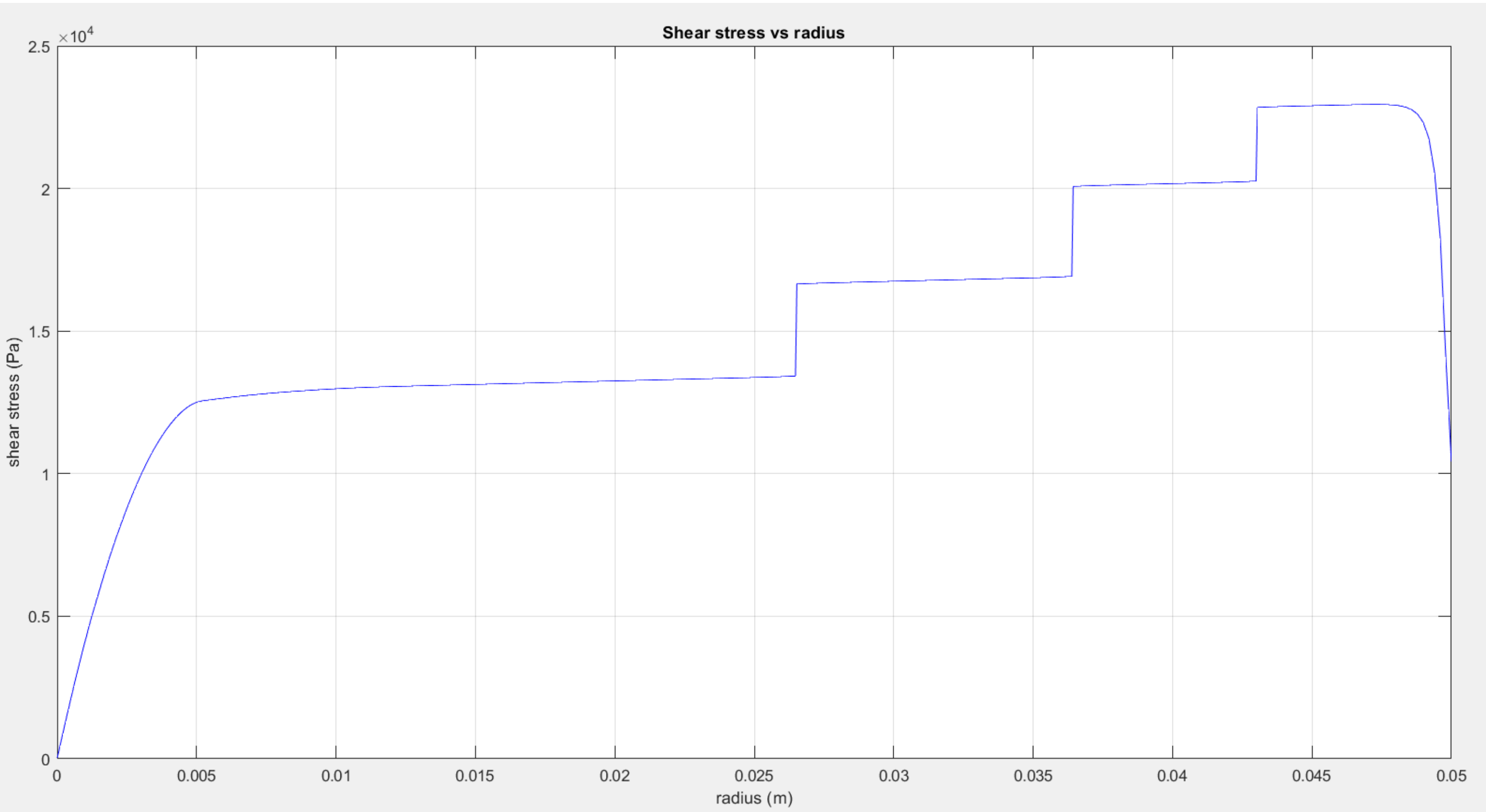


Fig 22 Shear stress vs radius (I=1A)

In **Fig 23** is shown the comparison between the numerically calculated shear stress and the analytically calculated, from Eq.11 shear stress. The only divergences are observed, are occurred in the center and in a very small radial zone in the outer radius of the disc, meaning that the model is validated.

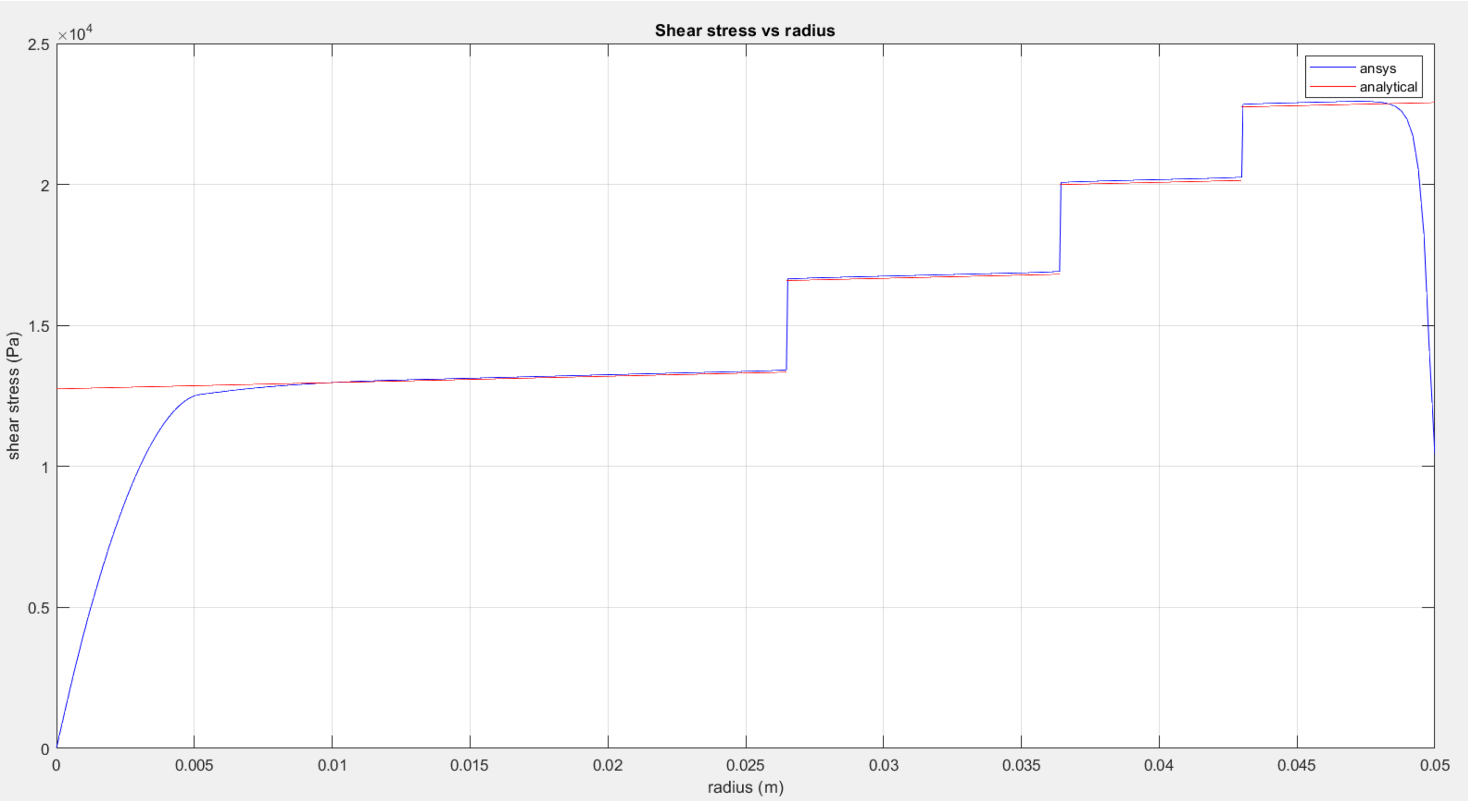


Fig 23 Comparison between analytically and numerically calculated shear stress

In **Fig 24** is shown a surface plot for velocity of the fluid inside the MRF gap. By comparing the present plot with Kumar's (2) it can be observed that both plots have the same shape.

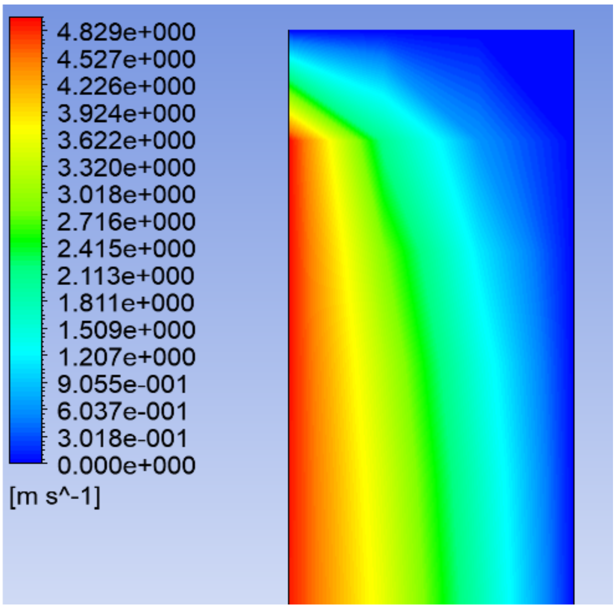


Fig 24 Surface Plot of velocity of the MRF

CHAPTER 3

TORQUE FOR DIFFERENT CURRENTS

The clutch tested in various current inputs, from 0A to 3A with a step of 0.5A. The magnetic field intensity is shown in the following Figure (**Fig25**).

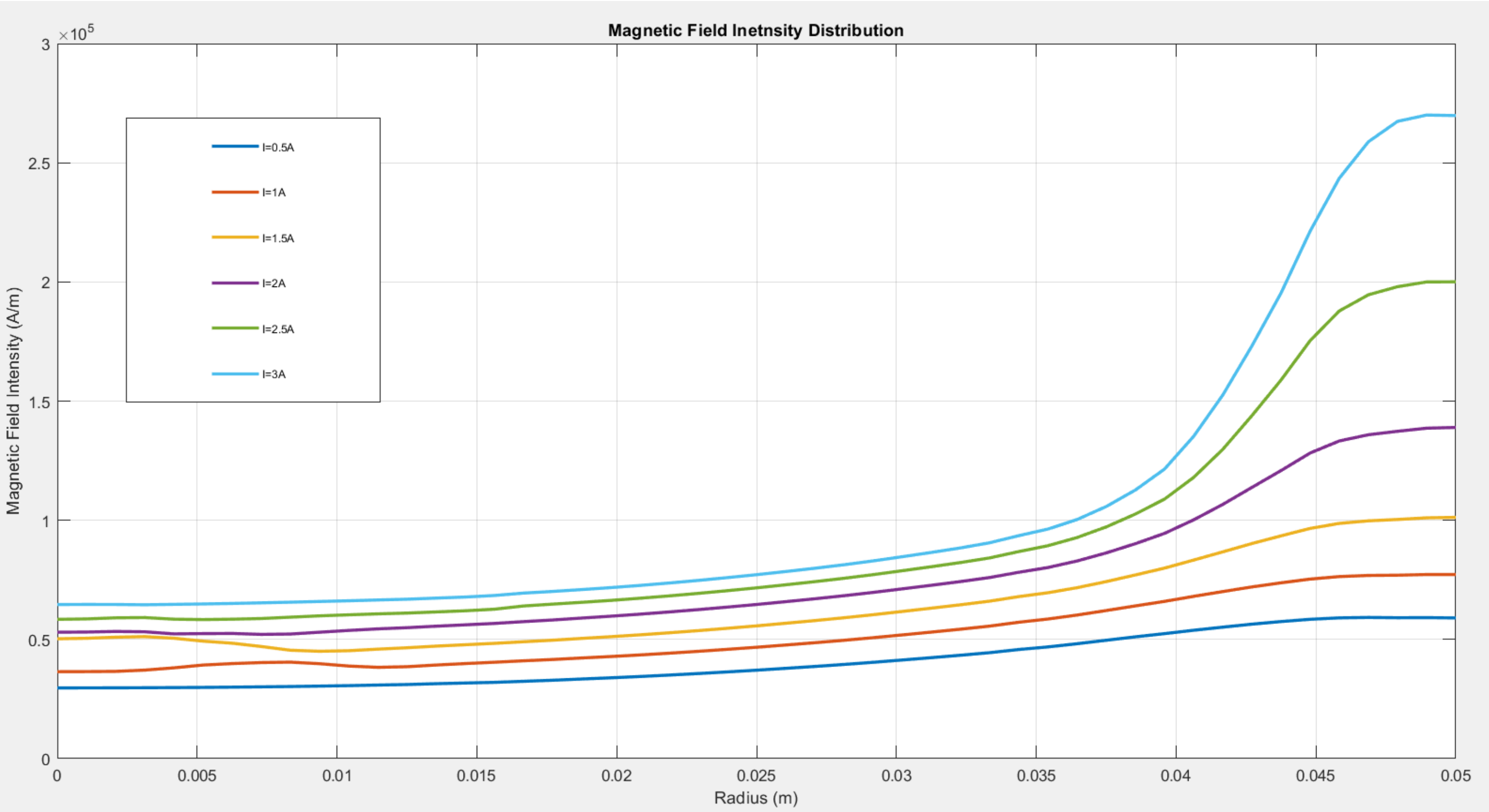


Fig25 Magnetic Field Intensity vs Radius (h=0.5mm)

From the above diagram it is shown that the intensity of the field is increase as it increases the electric current. Moreover on high current values, the distribution of the field becomes wider, meaning that more zones for the CFD models are needed. The magnetic field intensity becomes larger away of the center of the discs and near the coil.

Using Eq 14 and Eq 16 the transferred torque is calculated. In **Fig 26** is plotted the torque as a function of the current. It is obvious that the electric current and as a consequence the magnetic field, enhance the torque transferring ability of the clutch. For higher current values, the torque increases. However, the torque does not increase at the same ratio at every current. For current values greater than 2.5A the ratio decreases significantly.

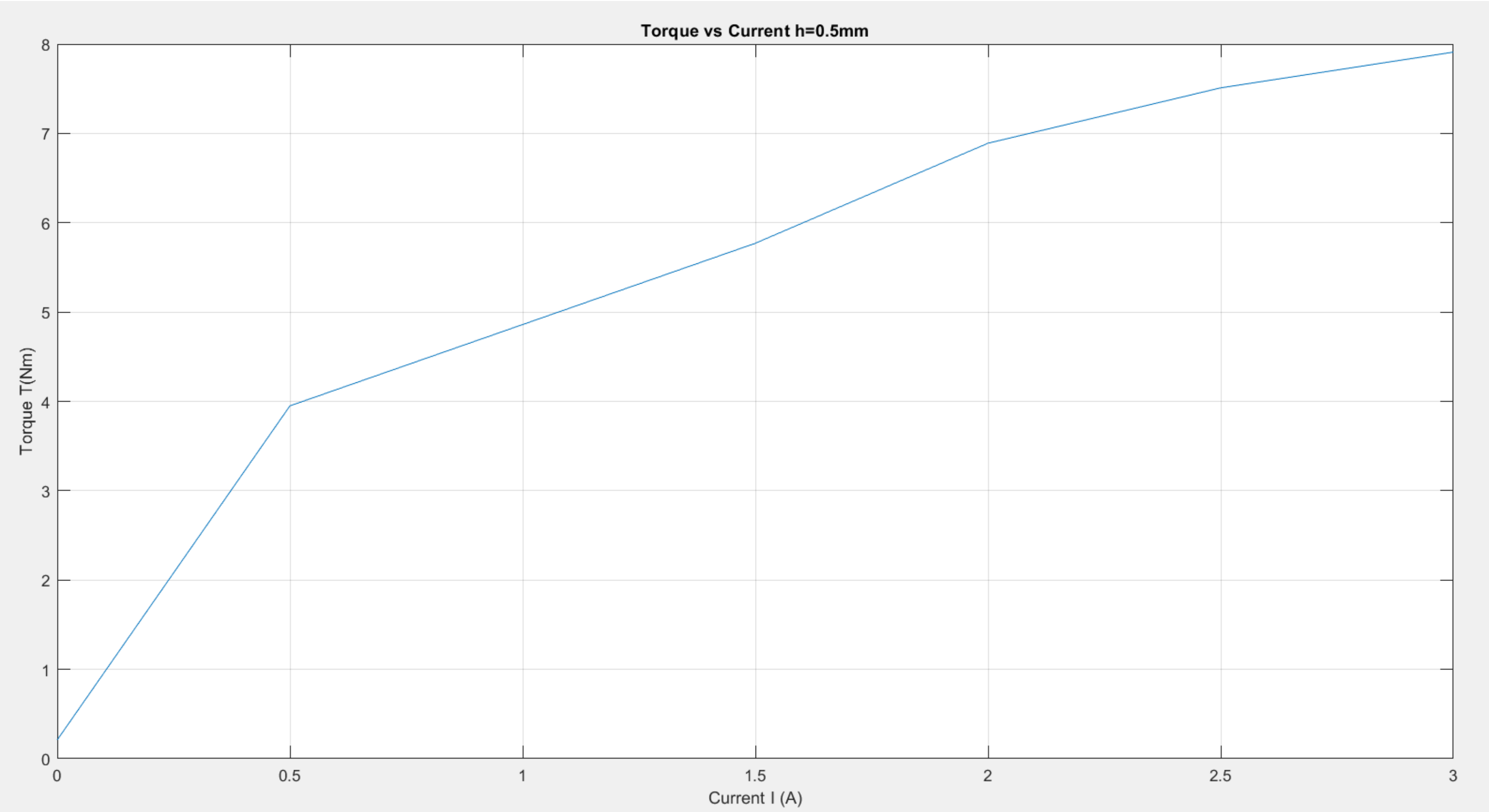


Fig 26 Torque vs Current (h=0.5mm)

INFLUENCE OF MRF GAP IN TORQUE

The clutch examined in various MRF gaps. The examined gaps are, 0.4mm, 0.3mm, 0.2mm. The current inputs, for all the examined gaps are the 0A to 3A with step of 0.5A. Varying the gap between the discs also affects some other dimensions. Those are the width of the coil, core and cover. Dimensions which do not affected by changing the r_{out} (for example the width of the discs 3mm) are constant for each configuration. Materials and wire's properties do not change for each gap configuration.

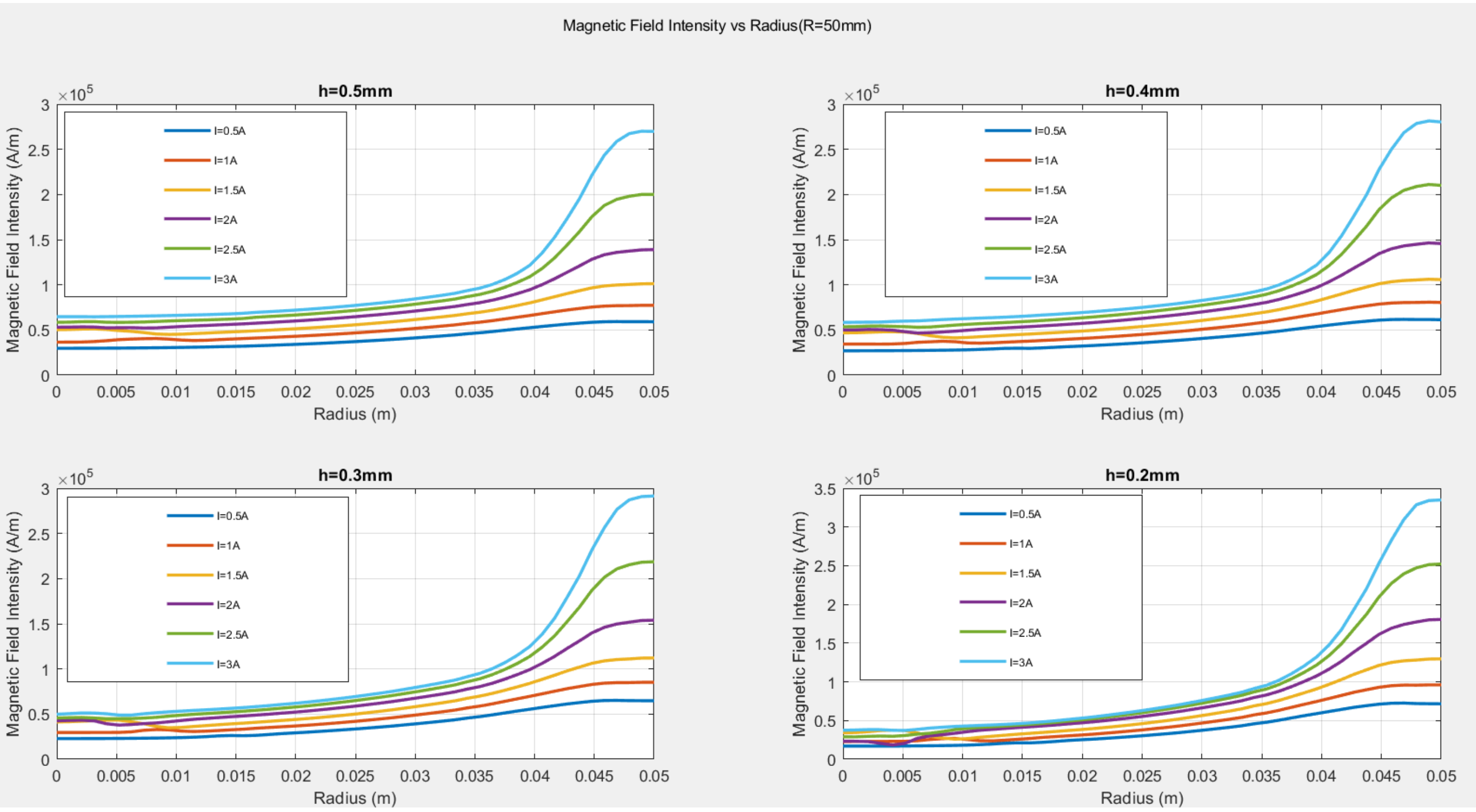


Fig 27 Distribution of the Magnetic Field Intensity for various MRF gaps (Current 0.5A to 3A)

In **Fig 27** it is displayed the distribution of the magnetic field intensity for different gaps between the 2 discs. It is observed that for smaller gaps the magnetic field intensity becomes greater. The torque the clutch can transfer for each gap is shown in the next diagram **Fig 28**.

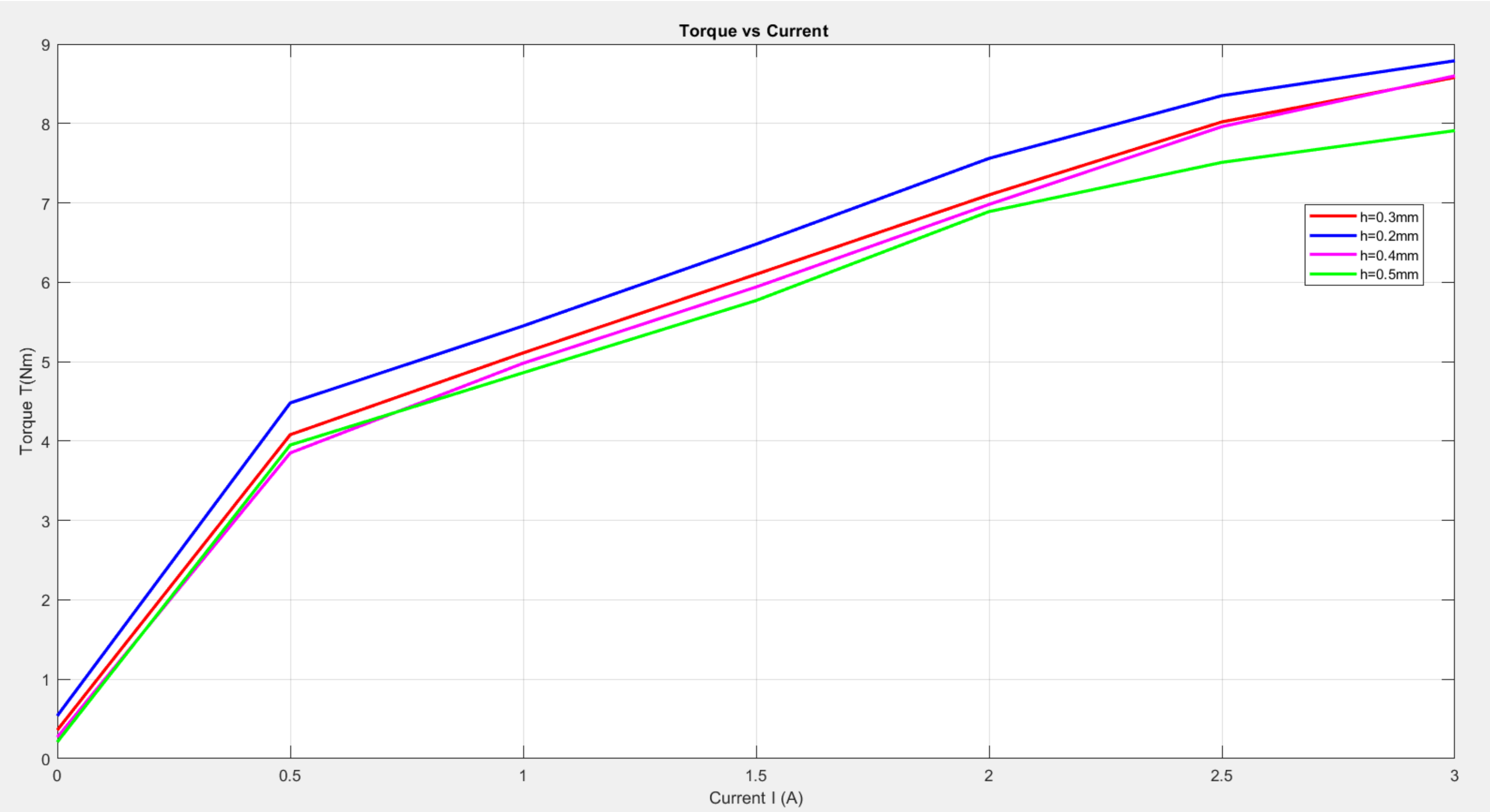


Fig 28 Torque vs Current for various MRF gaps

For every gap it is observed that the magnetic field enhances the torque significantly. For smaller gaps between the discs, it is observed an increase of the torque capability of the clutch. That happens for two reasons. First of all the magnetic field is increased for smaller gaps leading to higher yield stress for the MRF. Secondly from Eq14 and Eq16 the torque is increasing for smaller gaps between the discs. However, it is observed that for each gap the rate of increase becomes lower for higher values of currents.

MAGNETIC SATURATION

Theoretically with the increasing of the input current, the MRF will magnetizing and will increase the yield stress to infinity. However, it is known, that there is no material which can increase its magnetizing to infinity. There is a limit, beyond which there is no significant change in the magnetic situation of the material. This limit is known as saturation point. For the examined MRF, the 132 DG of Lord, the manufacturer informs that for magnetic field intensity greater than 300kA/m, the MRF inserts into magnetic saturation. On that situation the yield stress does not further increase with the increasing of the field, but remains constant. The yield stress for saturation point, according to M I Varela-Jiménez , J L Vargas Luna , J A Cortés-Ramírez and G Song (8) is 49100 Pa. This leads to the conclusion that there is an upper limit in the torque transfer capability of the clutch. This limit is achieved when the magnetic field intensity is higher than 300kA/m at every point of the MRF gap and for all the volume of MRF the yield stress is 49.1kPa.

MAXIMUM POSSIBLE TORQUE FOR EACH MRF GAP	
MRF GAP h (mm)	Maximum Possible Torque (Nm)
h=0.5	13.074
h=0.4	13.093
h=0.3	13.18
h=0.2	13.36

Table3 Maximum Torque for each MRF gap

The torque has been calculated with the procedure it was followed before. The only difference is that there is no division in radial zones for areas where the magnetic field intensity is larger than 300kA/m. The clutch simulated for inputs of 5A, 10A, 15A, 20A and 25A.

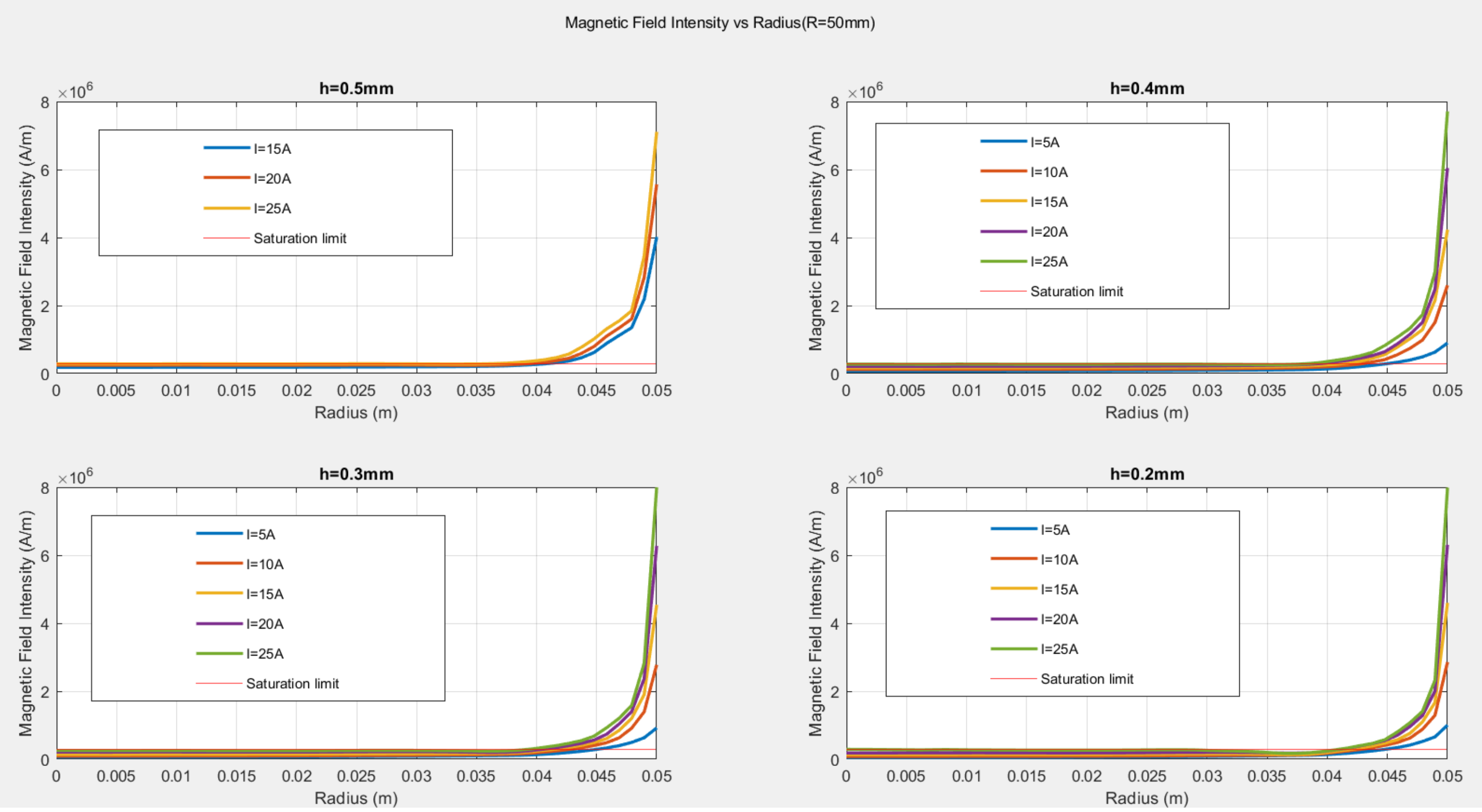


Fig29 Distribution of the Magnetic Field Intensity for different MRF gaps (Current 5A to 25A)

In **Fig29** it is shown how the magnetic field intensity varies for different currents, in different gaps between the discs. It is shown that the field takes larger values for higher currents, as it was observed in the previous sector. However, it exceeds the saturation limit in small radial zones, near the end of the discs.

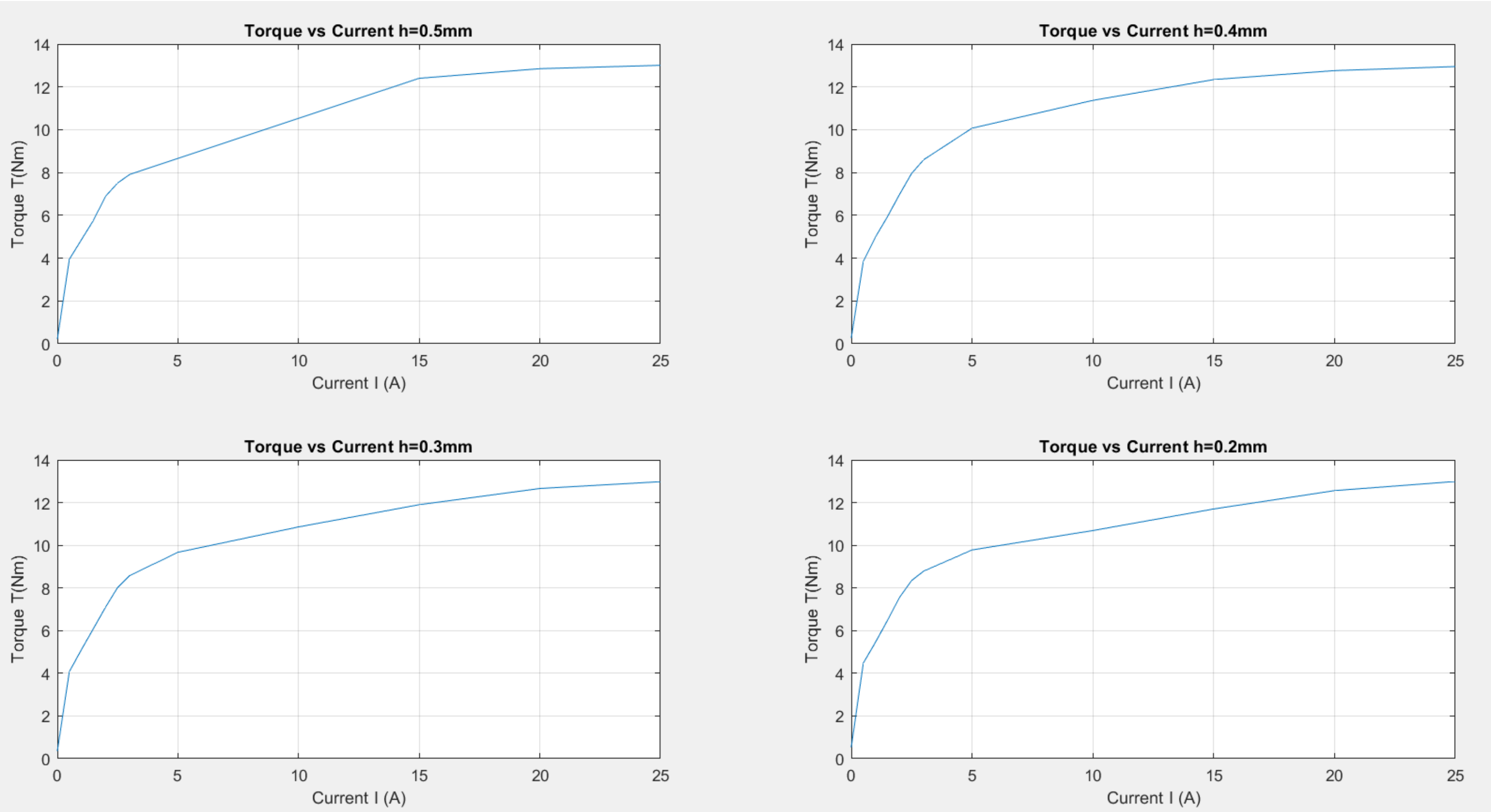


Fig 30 Torque vs Current for various MRF gaps

In **Fig 30** is shown how the torque varies with respect to current, for different gaps between the discs. By increasing the current, the torque increases as well. However as the currents values become higher, the torque does not increase significantly. The ratio of increase for torque drops for high currents. In addition, the torque for all gaps converges to its maximum value for each gap. In the next table (**Table 4**) are shown the maximum calculated torque for each gap.

MAXIMUM CALCLUTATED TORQUE FOR EACH MRF GAP		
MRF gap h (mm)	Maximum Calculated Torque (Nm)	Maximum torque capacity coverage
h=0.5	13.008	99.4%
h=0.4	12.95	98.9%
h=0.3	12.98	98.4%
h=0.2	12.99	97.2%

Table 4 Maximum calculated torque for each gap

Comparing the values of **Table 4** with those of **Table3** it is concluded that for input of 25A torque approaches very close to its upper limit. More specific for h=0.5mm the torque at 25A reaches at 99.4% of the maximum torque and the maximum torque capacity coverage achieves its highest value. The smallest value is 97.2% and it occurs for h=0.2mm. Despite the fact that the magnetic saturation limit is exceeded for a very small radial area, the torque almost reaches at the maximum value of each MRF gap configuration.

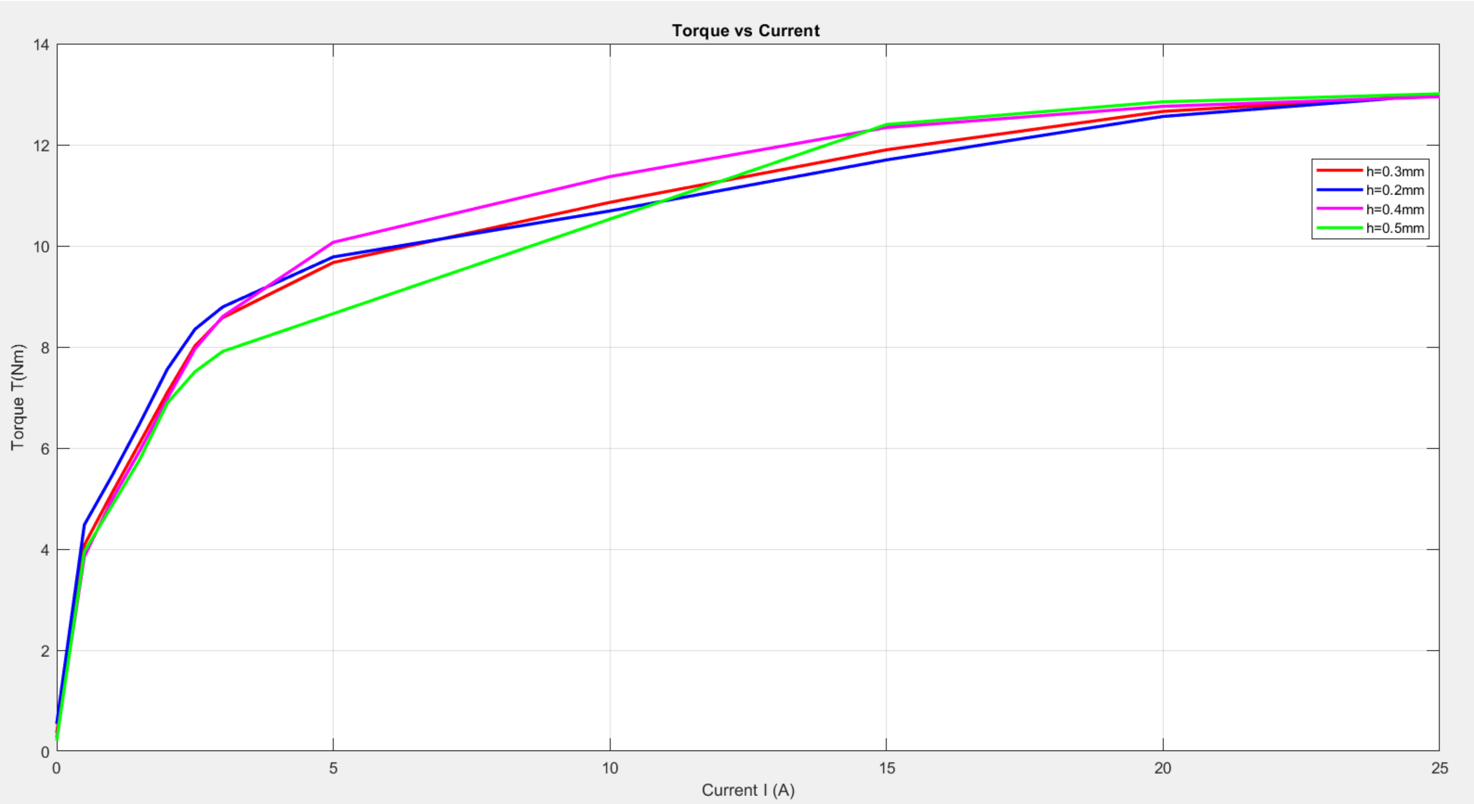


Fig31 Torque vs Current included saturation

In **Fig31** are shown in one diagram the torque curves for each gap, included higher currents values, leading to saturation. The diagram can be divided in 3 areas. In first area, from 0A to 3A, torque is larger for smaller gaps. In the second area from 3A to 20A torque is larger for bigger gaps between the discs. In the last area, from 20A to 25A torque curve for every gap converges to the maximum value for each gap. The torque is almost the same for each configuration.

INFLUENCE OF MAXIMUM RADIUS OF DISCS

After it was investigated the influence of the gap between the discs, the next step is to investigate how the maximum radius of the discs affect the torque the clutch can transfer. The clutch is investigated for maximum radius of: 70mm, 90mm and 110mm for all the gaps. The shape of the clutch remains the same in each configuration (**Fig 10**). The distance from the center of the clutch for bearings, core, coil and cover change accordingly in each case. Dimensions which do not affected by changing the r_{out} (for example the width of the discs 3mm) are constant for each configuration. Materials and wire's properties do not change for each configuration.

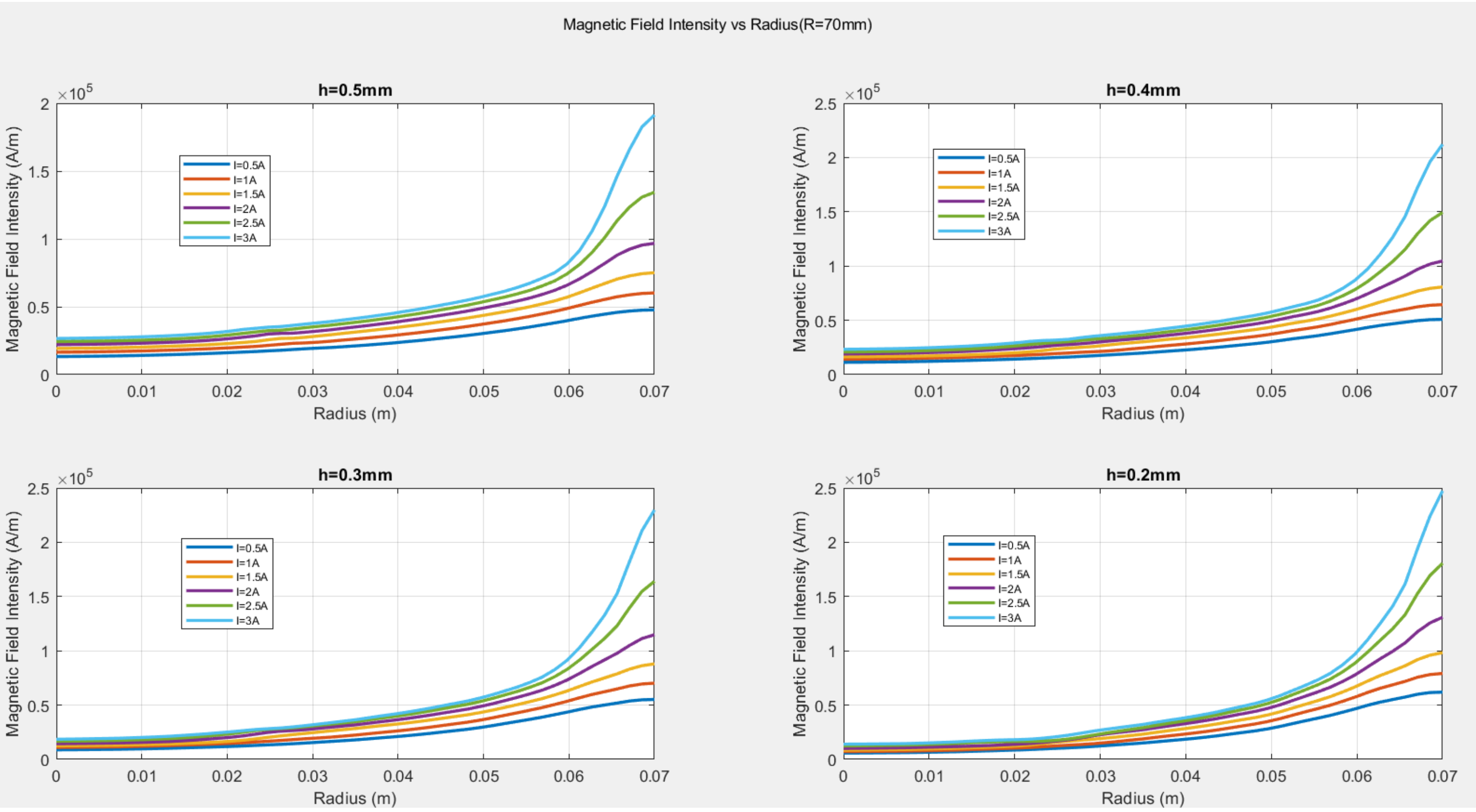


Fig32 Magnetic Field Distribution for various gaps (Current 0.5A to 3A and maximum disc radius 70mm)

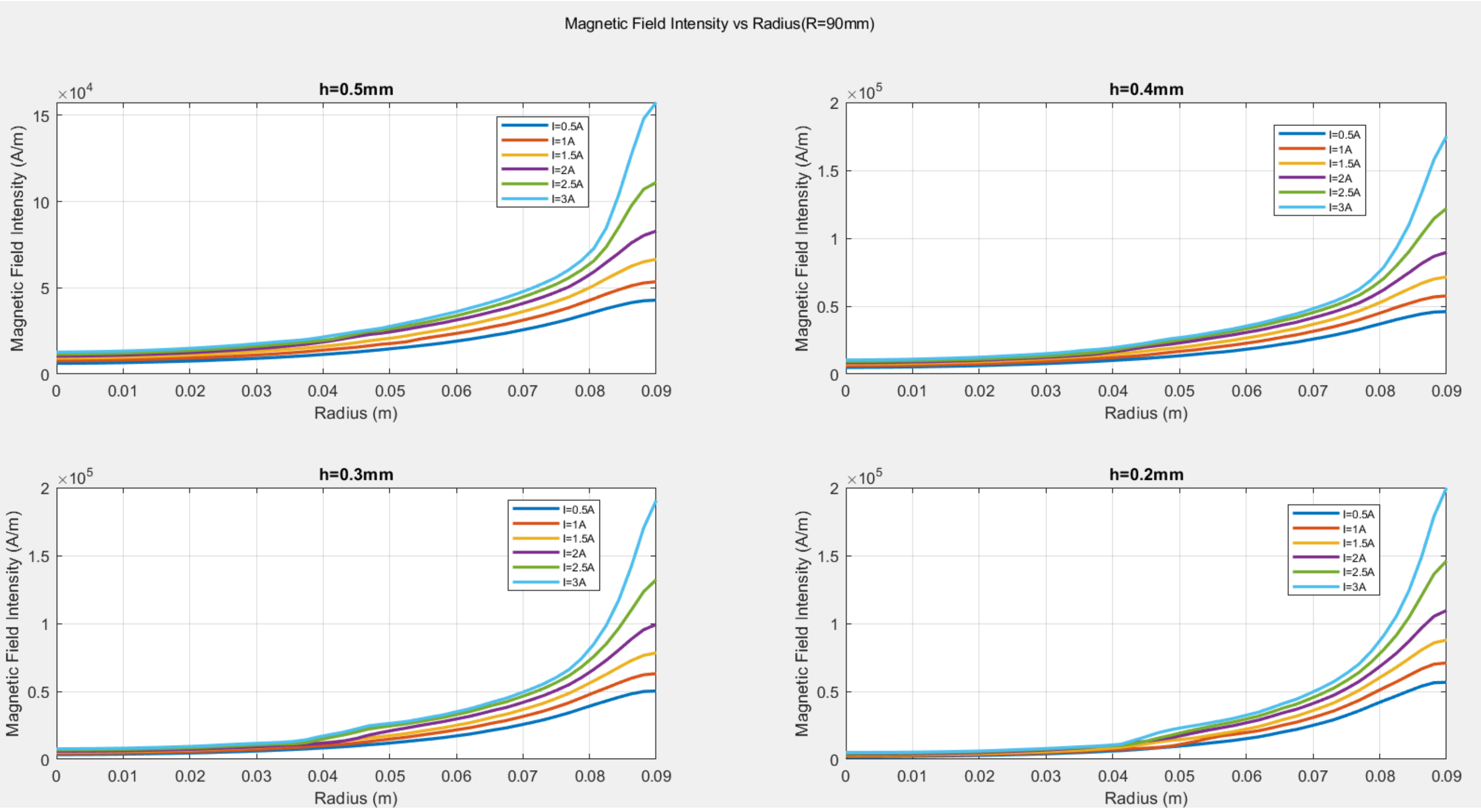


Fig 33 Magnetic Field Distribution for various gaps (Current 0.5A to 3A and maximum disc radius 90mm)

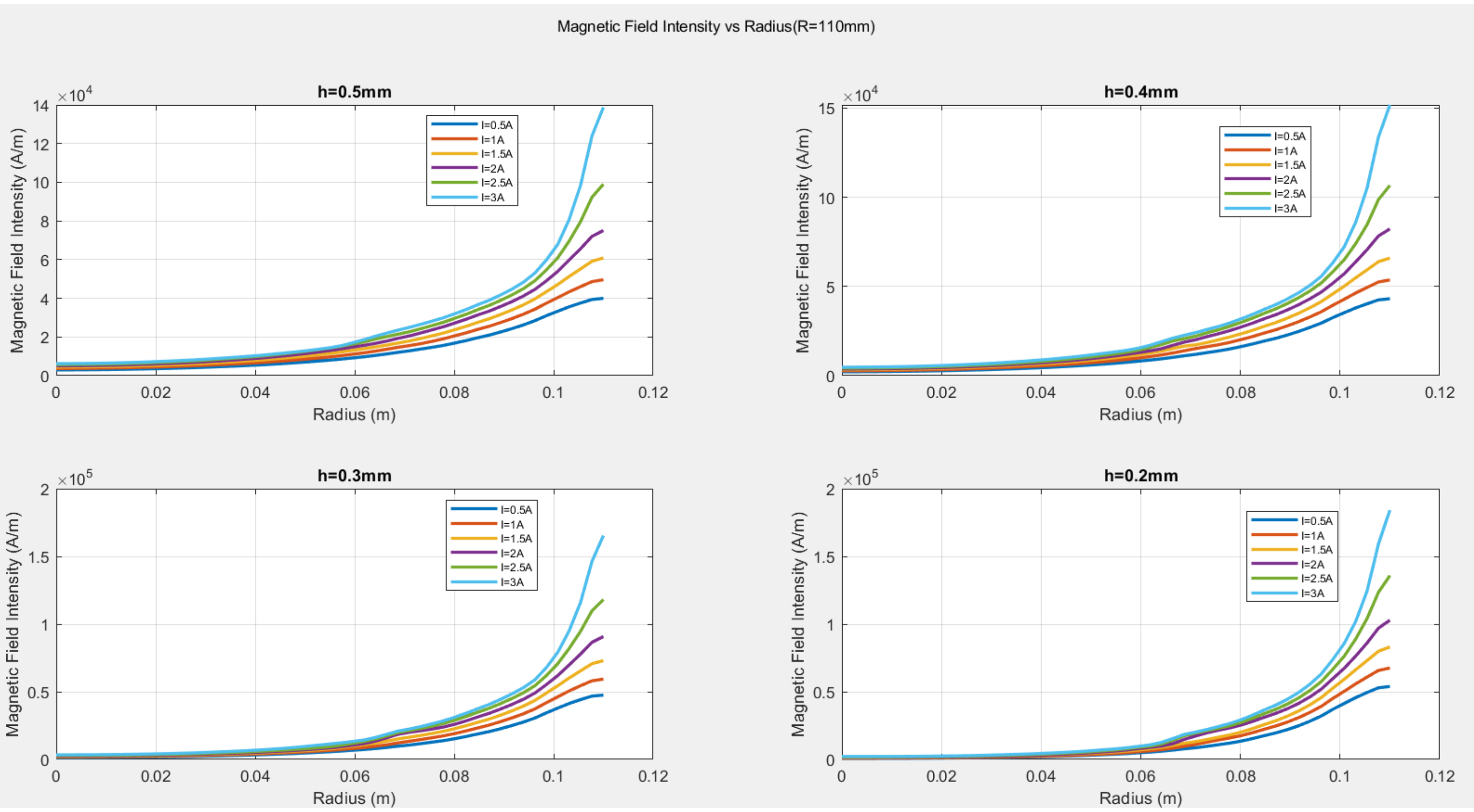


Fig 34 Magnetic Field Distribution for various gaps (Current 0.5A to 3A and maximum disc radius 110mm)

In **Fig32**, **Fig 33** and **Fig 34** is shown the magnetic field intensity distribution for different gaps, for each radius. The magnetic field intensity inside the MRF decreases near the center of the discs as the maximum disc radius becomes bigger. However at the end of the discs the magnetic field intensity takes for every r_{out} almost the same value. That means that the distribution of the field becomes wider as the r_{out} increases. This happens for every gap. The explanation for this is when the disc's radius becomes bigger the magnetic circuit becomes bigger, increasing the magnetic reluctance.

In the next 3 figures (**Fig 35**, **Fig36**, **Fig37**) it is shown the distribution of the magnetic field for higher currents inputs, in an effort to approach saturation point.

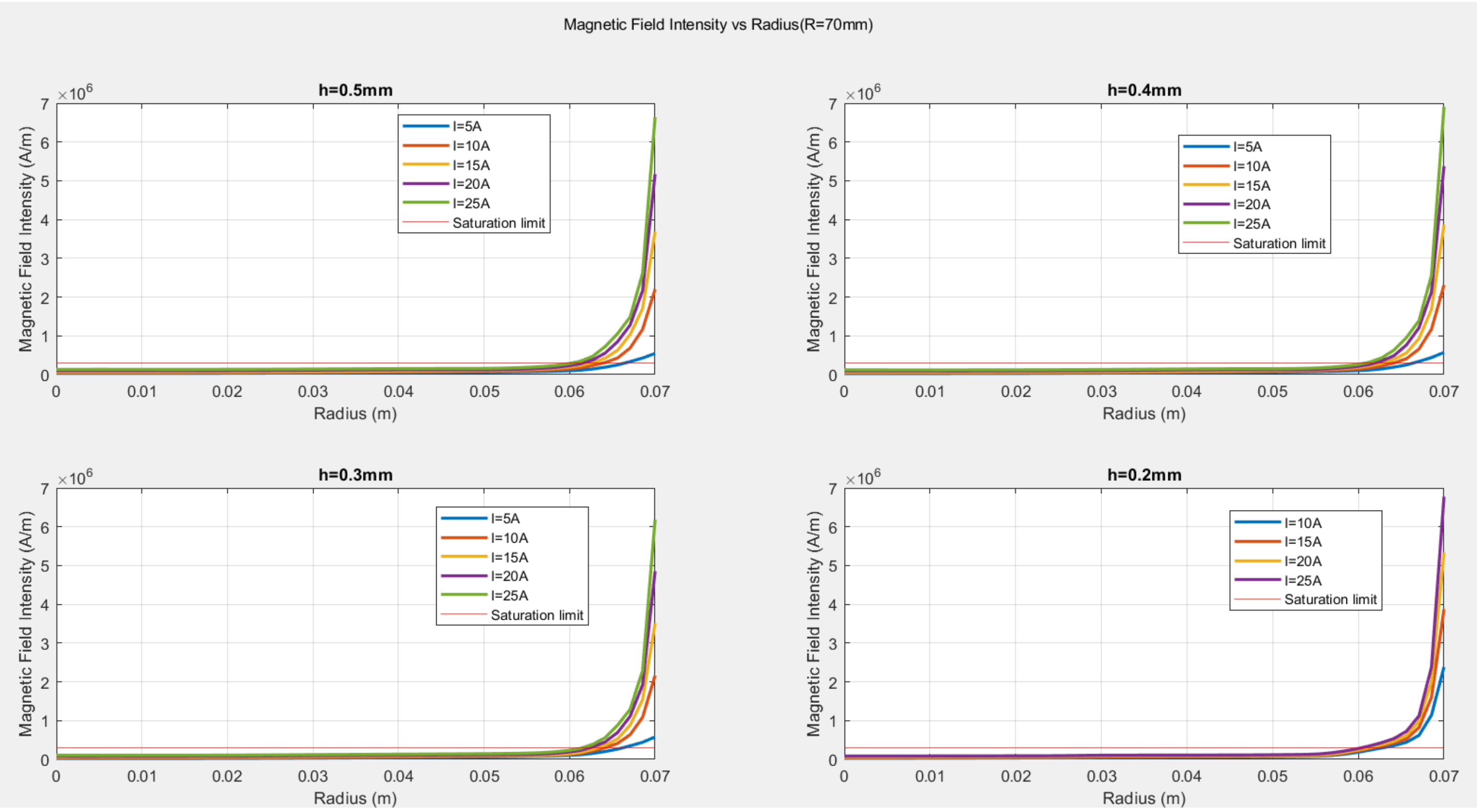


Fig 35 Magnetic Field Distribution for various gaps (Current 5A to 25A and r_{out} 70mm)

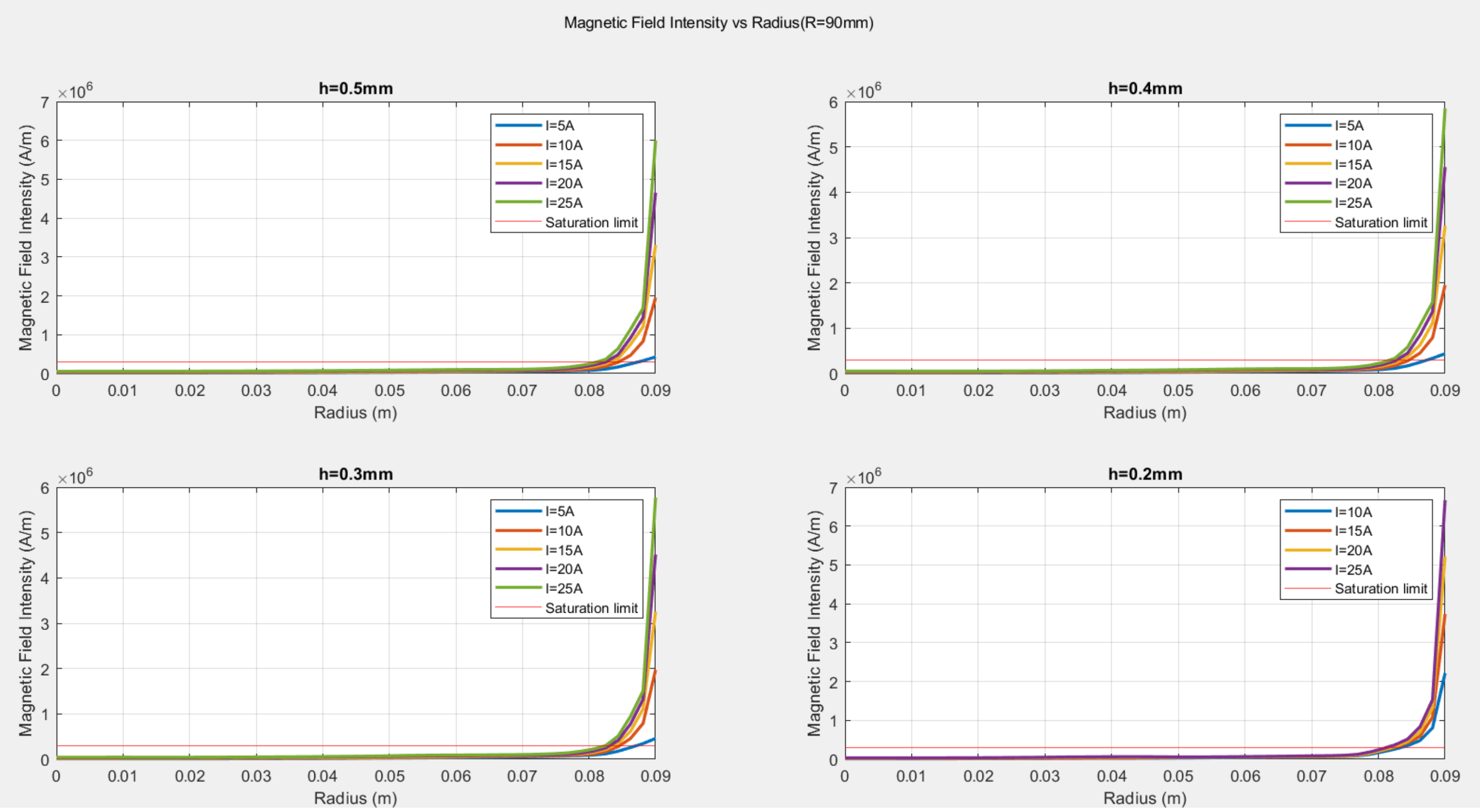


Fig36 Magnetic Field Distribution for various gaps (Current 5A to 25A and r_{out} 90mm)

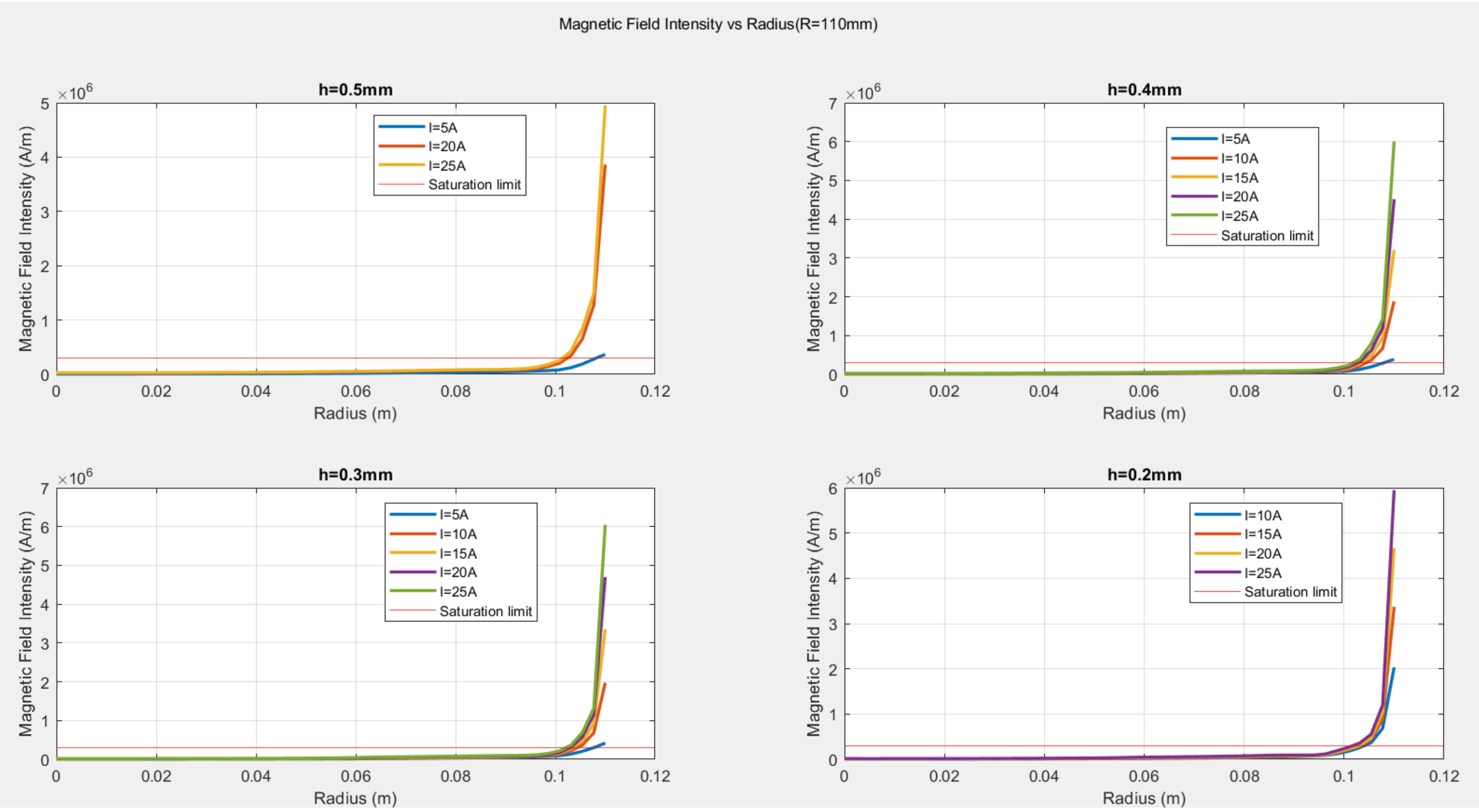


Fig37 Magnetic Field Distribution for various gaps (Current 5A to 25A and r_{out} 110mm)

It is observed that the magnetic field intensity exceeds the saturation limit only for small radial zones near the end of the disc. However for smaller radius the saturation zone becomes larger. Again it is observed that for bigger discs the magnetic field intensity is lower.

The maximum possible torque that can be transmitted by the clutch for each geometric configuration is summarized in the following table.

MAXIMUM POSSIBLE TORQUE FOR EACH GEOMETRIC CONFIGURATION		
	MRF GAP h (mm)	Maximum Possible Torque (Nm)
Maximum r_{out}=70mm	h=0.5	36.11
	h=0.4	36.32
	h=0.3	36.68
	h=0.2	37.38
Maximum r_{out}=90mm	h=0.5	77.27
	h=0.4	77.85
	h=0.3	78.81
	h=0.2	80.73
Maximum r_{out}=110mm	h=0.5	142.02
	h=0.4	143.31
	h=0.3	145.45
	h=0.2	149.75

Table5 Maximum torque for each geometric configuration

From **Table5** it is obvious that maximum possible torque is larger as the discs radius becomes bigger, something absolutely reasonable, if we look at Eq14. However it is interesting to test, how the clutch responses to larger discs, based on the fact that the field decreases.

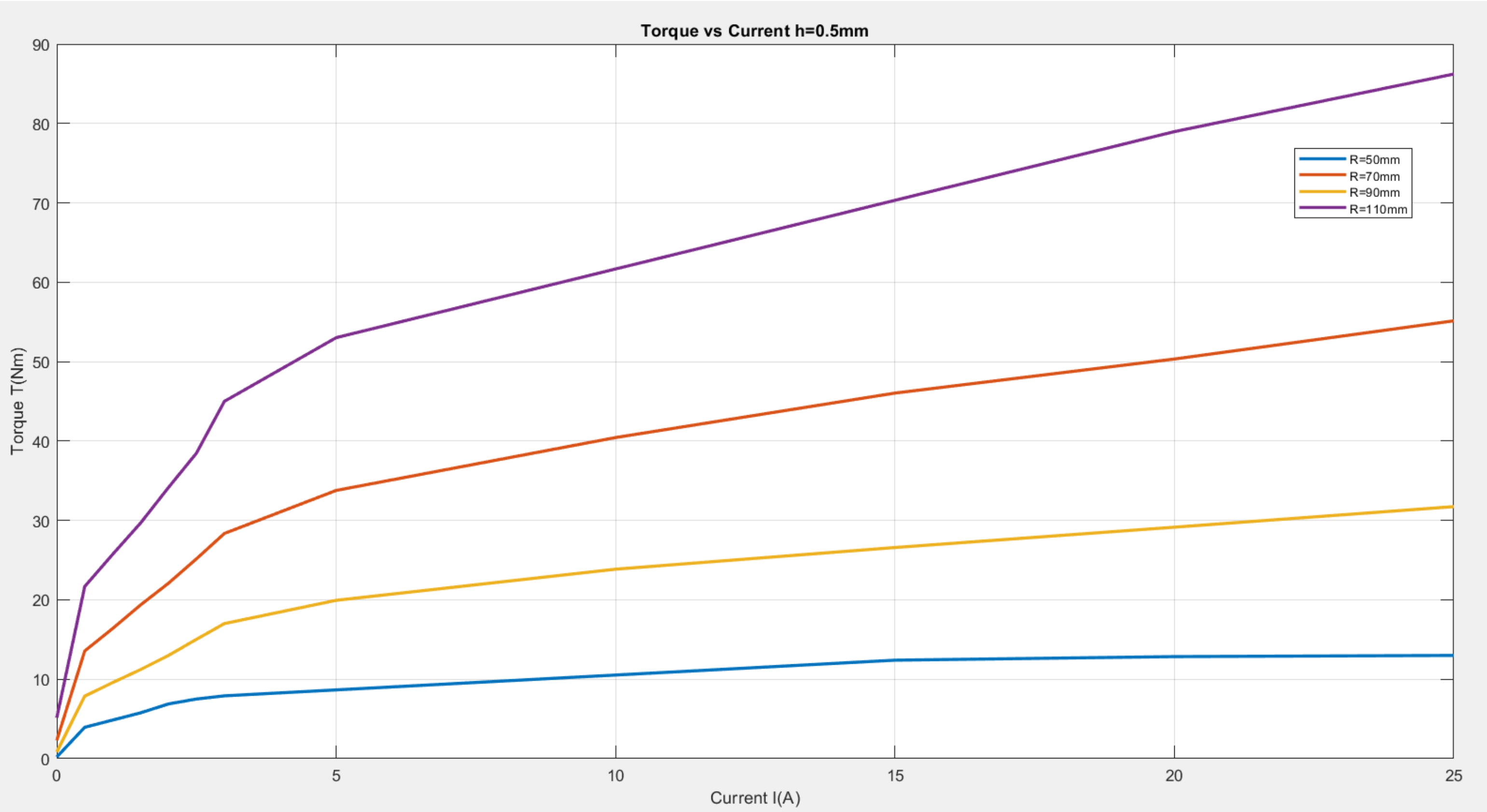


Fig38 Torque vs Current for h=0.5mm

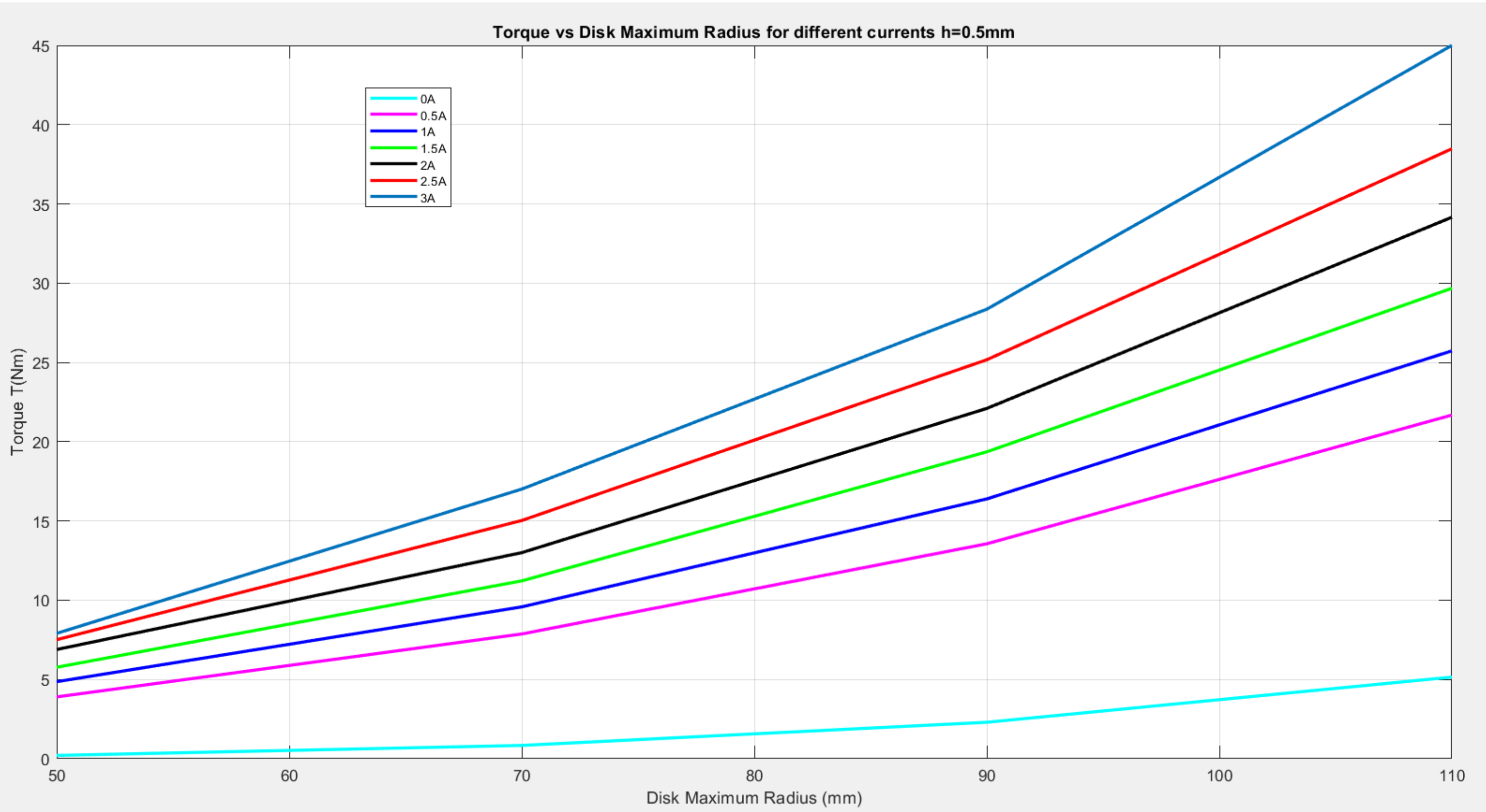


Fig 39 Torque vs Radius for h=0.5mm

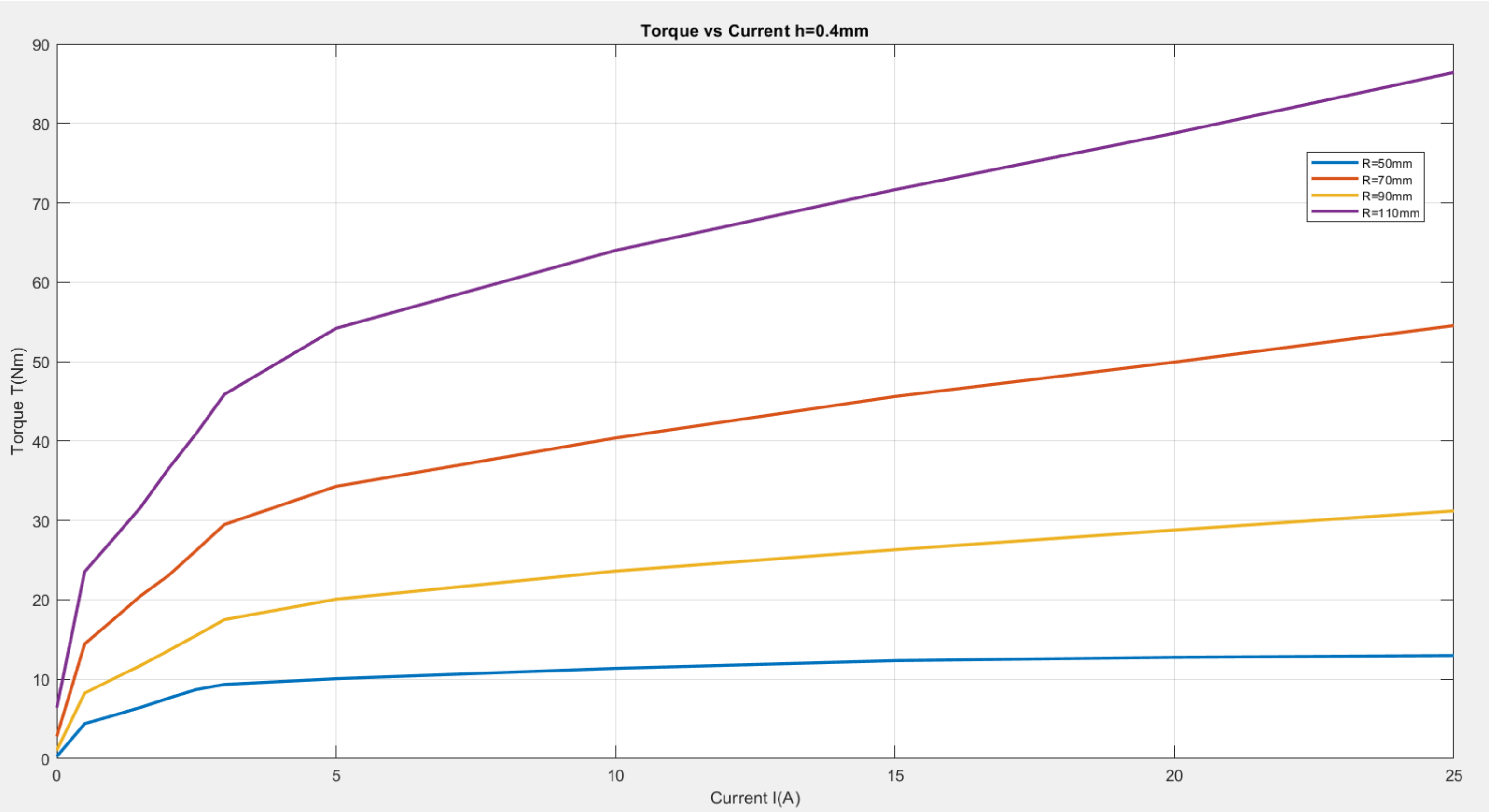


Fig40 Torque vs Current for h=0.4mm

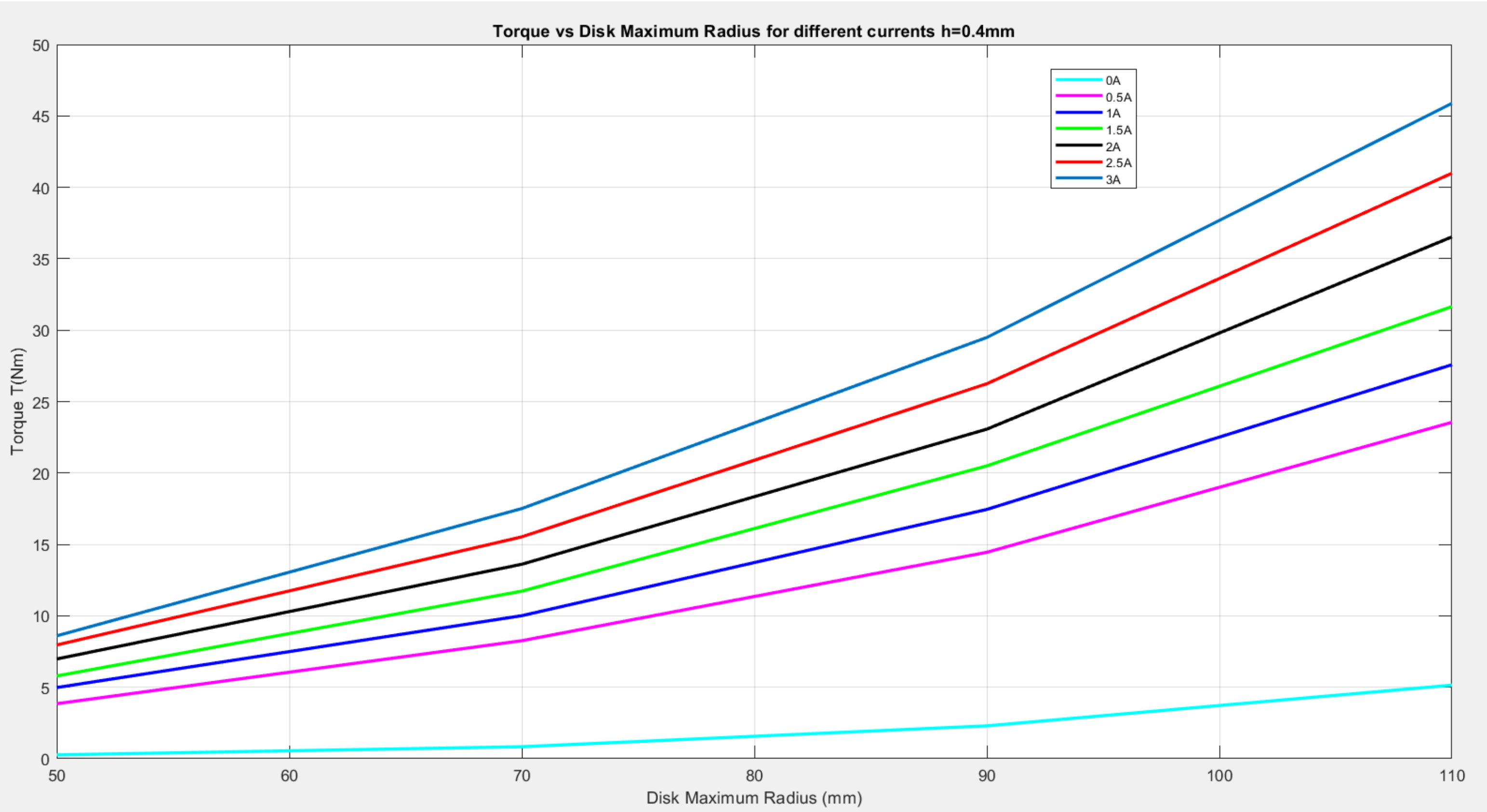


Fig41 Torque vs Radius for h=0.4mm

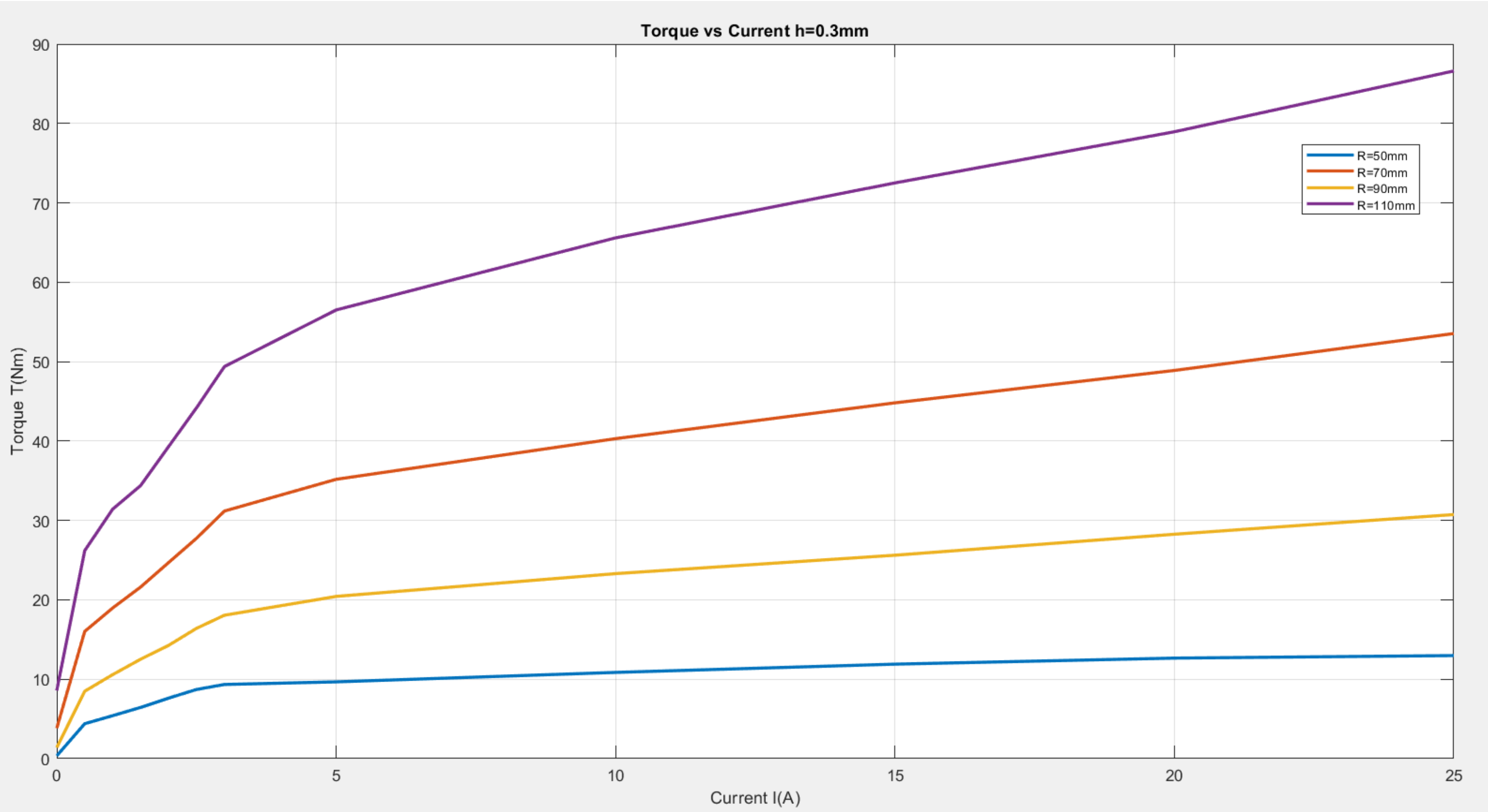


Fig 42 Torque vs Current for h=0.3mm

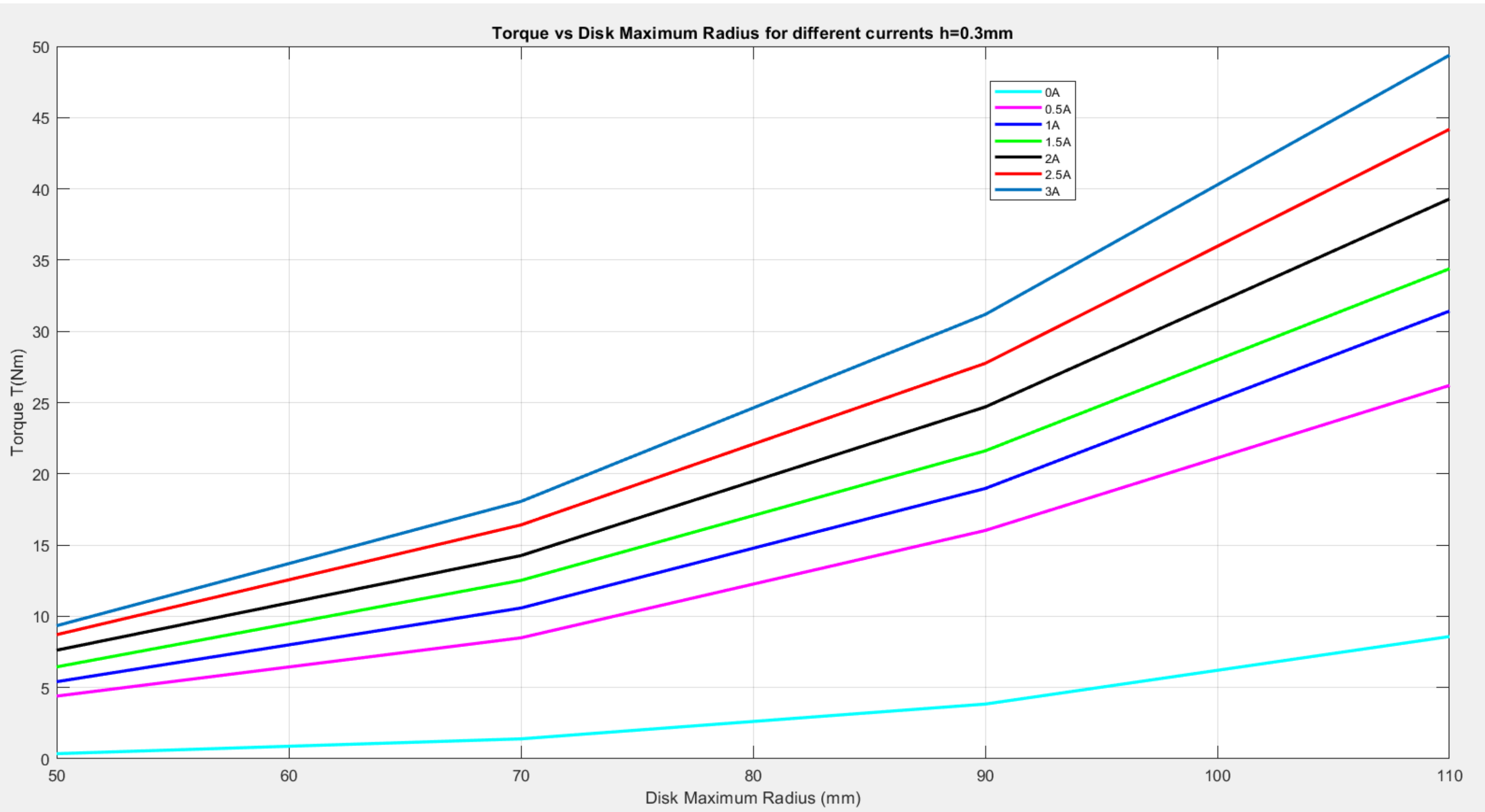


Fig 43 Torque vs Radius for h=0.3mm

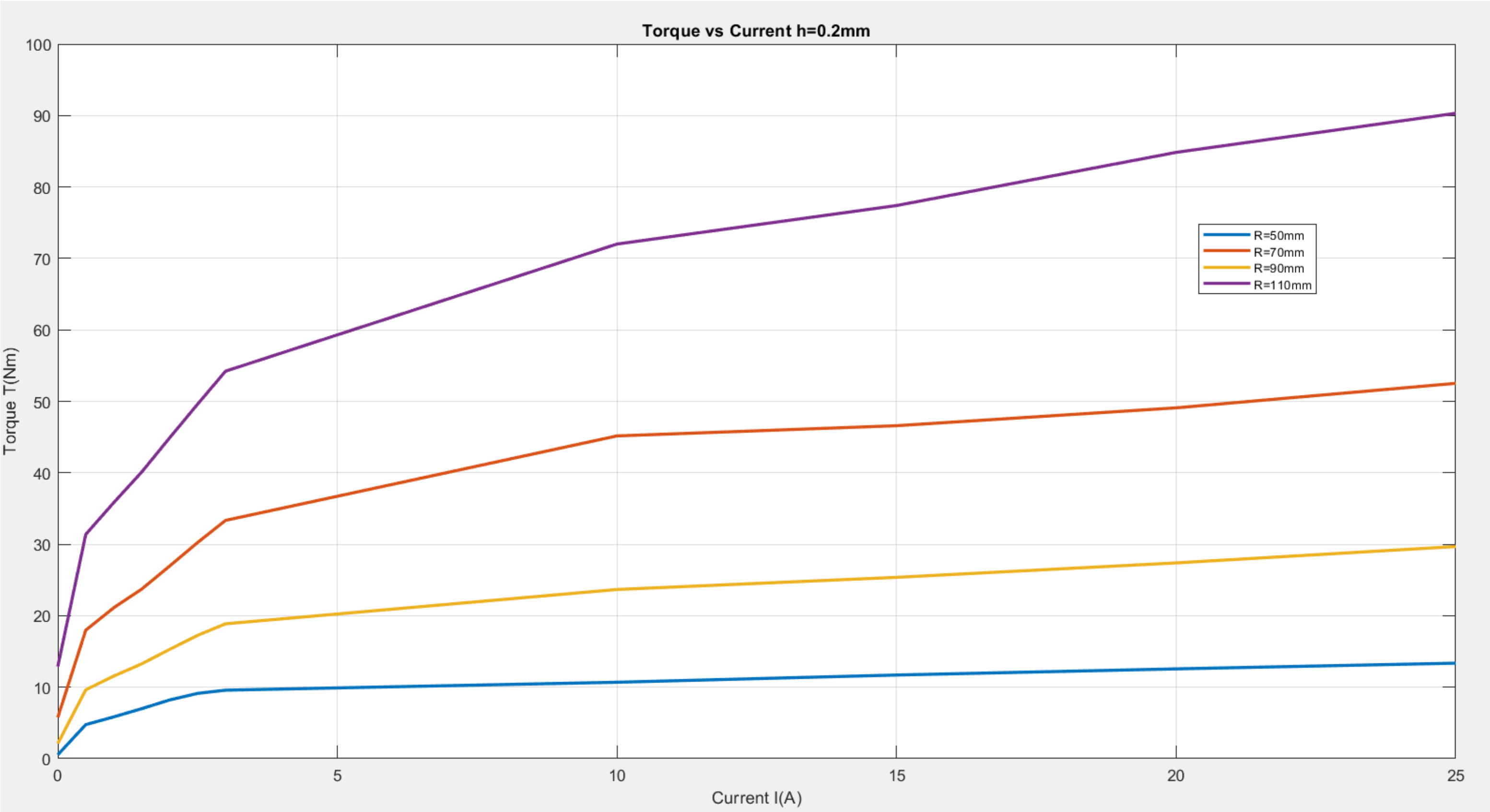


Fig 44 Torque vs Current for h=0.2mm

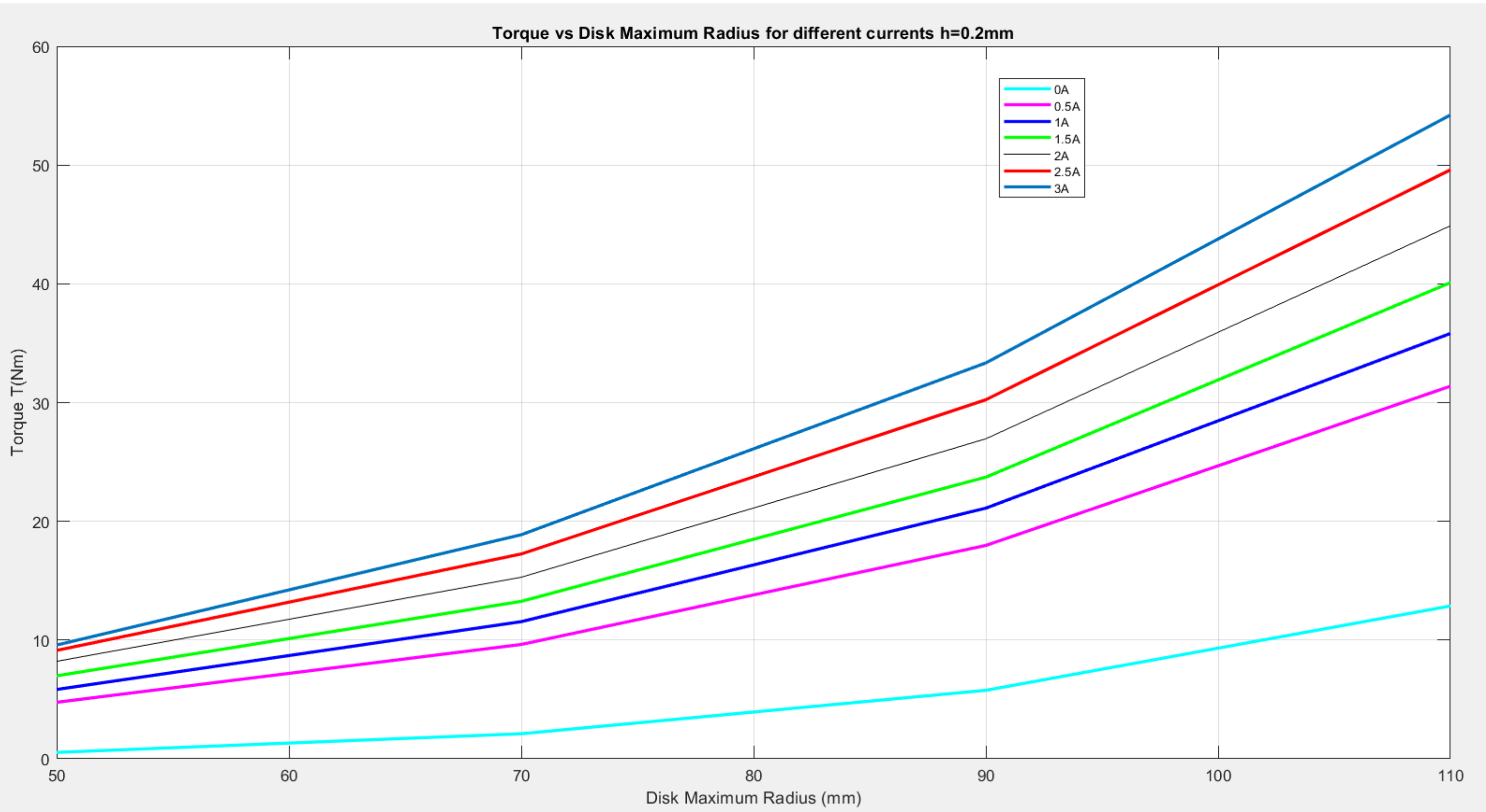


Fig45 Torque vs Radius for h=0.2mm

In **Fig38**, **Fig40**, **Fig 42** and **Fig 44** is shown the torque as a function of current, for different gaps between the discs. The torque is plotted for all the examined disc maximum radiuses. It is shown that for specific gap between the discs, the torque is increased with the increasing of the discs. For every gap the maximum torque occurs for r_{out}=110mm. **Fig 39**, **Fig41**, **Fig 43** and **Fig45** show how the torque varies with respect to disc maximum radius, for different currents. The diagrams confirm that for bigger discs the torque increases. Also for higher radius value the increase ratio for the torque becomes greater. This happens despite the significant drop of the magnetic field intensity and the MRF's yield stress, meaning that for a magnetorheological clutch the geometry and the discs size are more important than the yield stress.

Next it will be compared the torque each gap can transfer keeping for certain r_{out}.

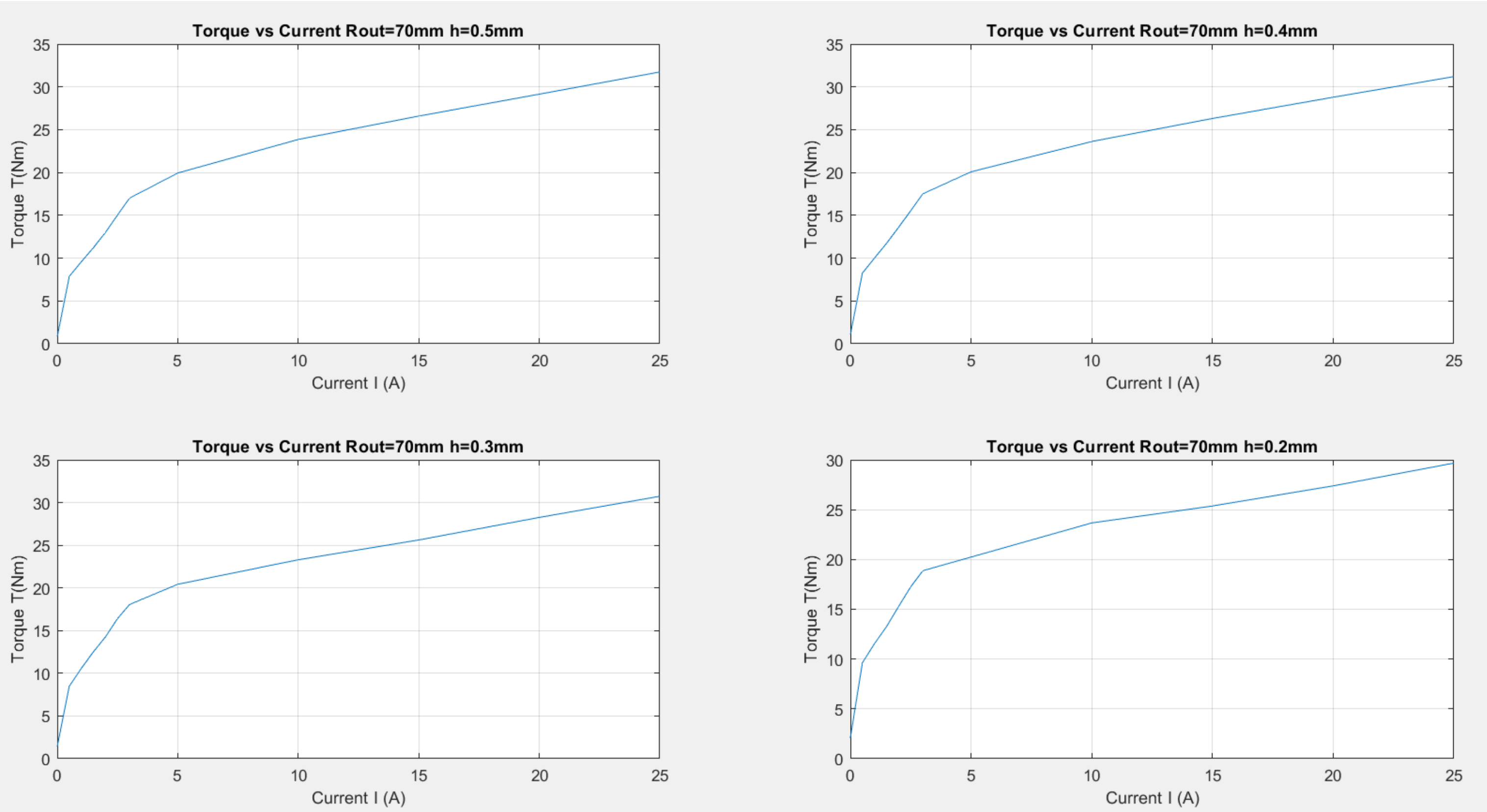


Fig 46 Torque vs Current for various gaps (r_{out}=70mm)

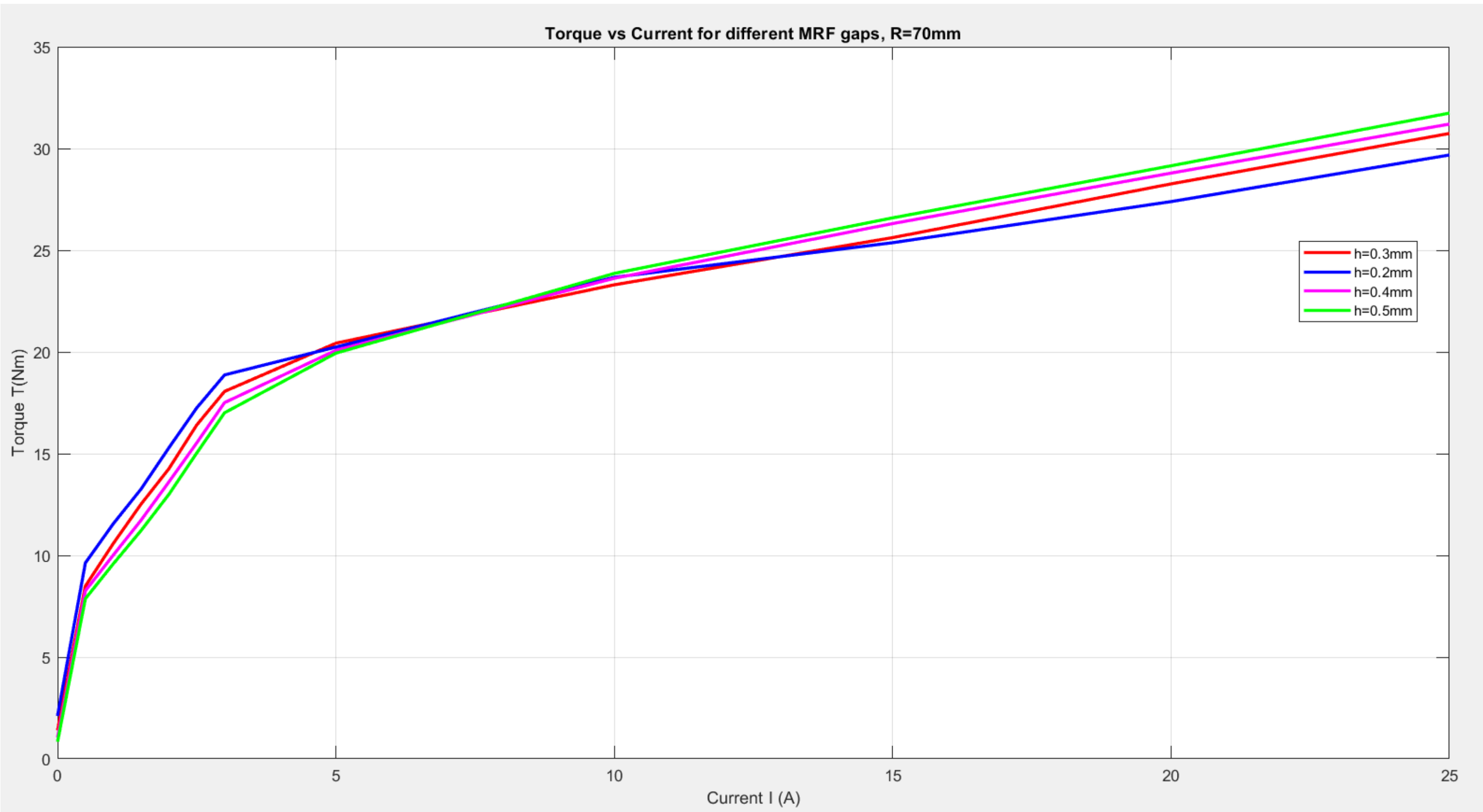


Fig 47 Torque vs Current (comparison between various gaps r_{out}=70mm)

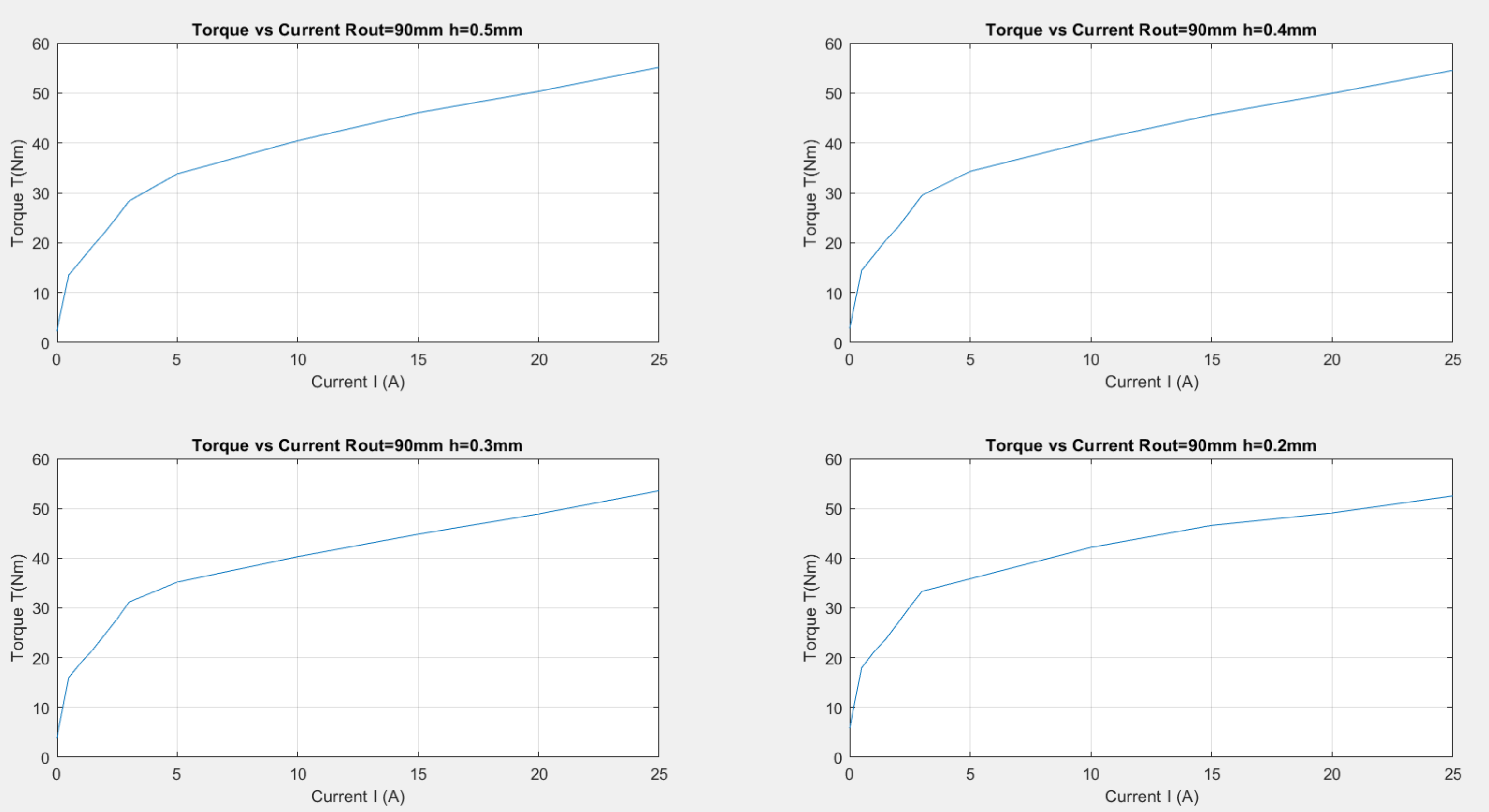


Fig48 Torque vs Current for various gaps (r_{out}=90mm)

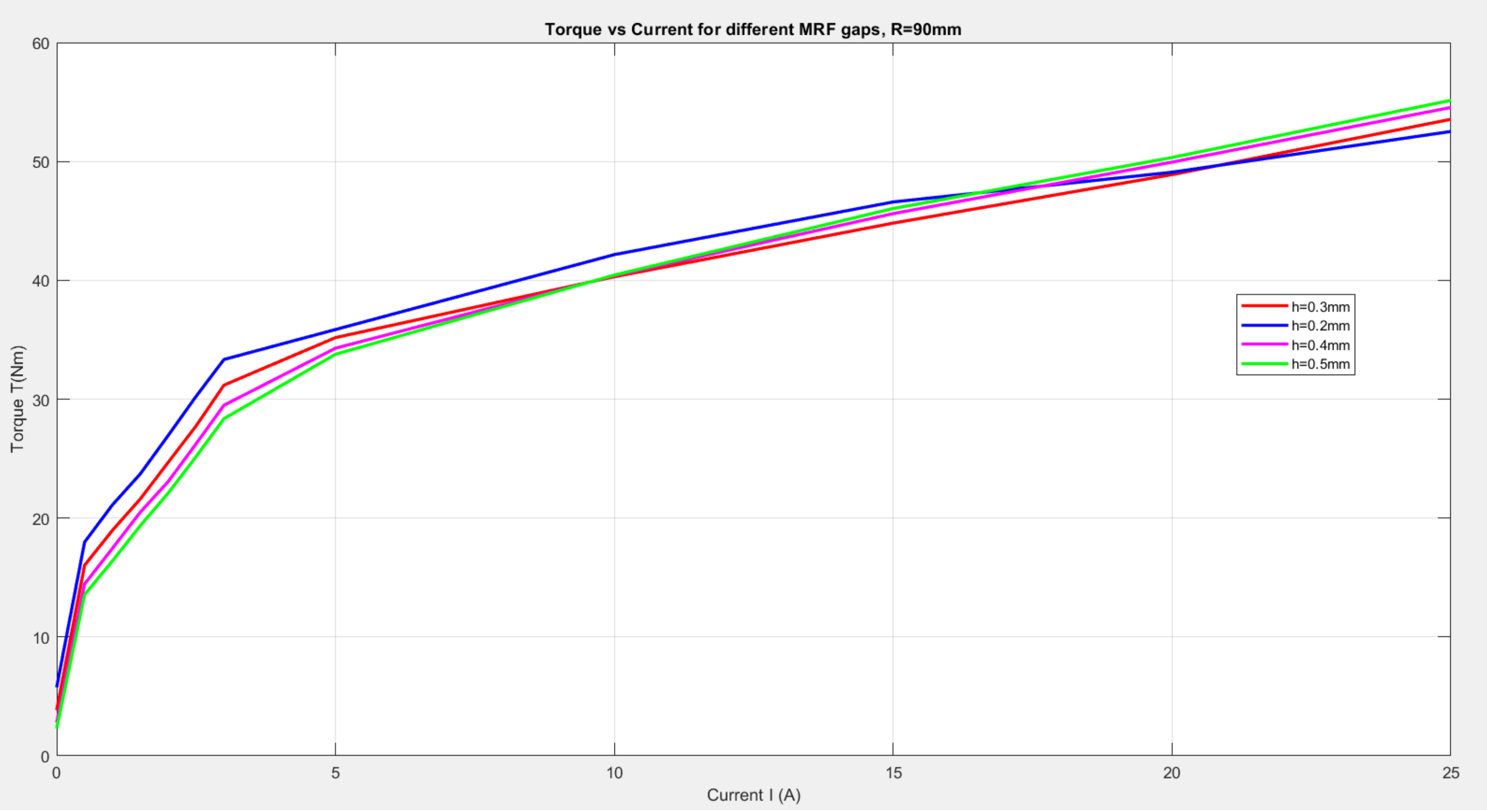


Fig49 Torque vs Current (comparison between various gaps r_{out}=90mm)

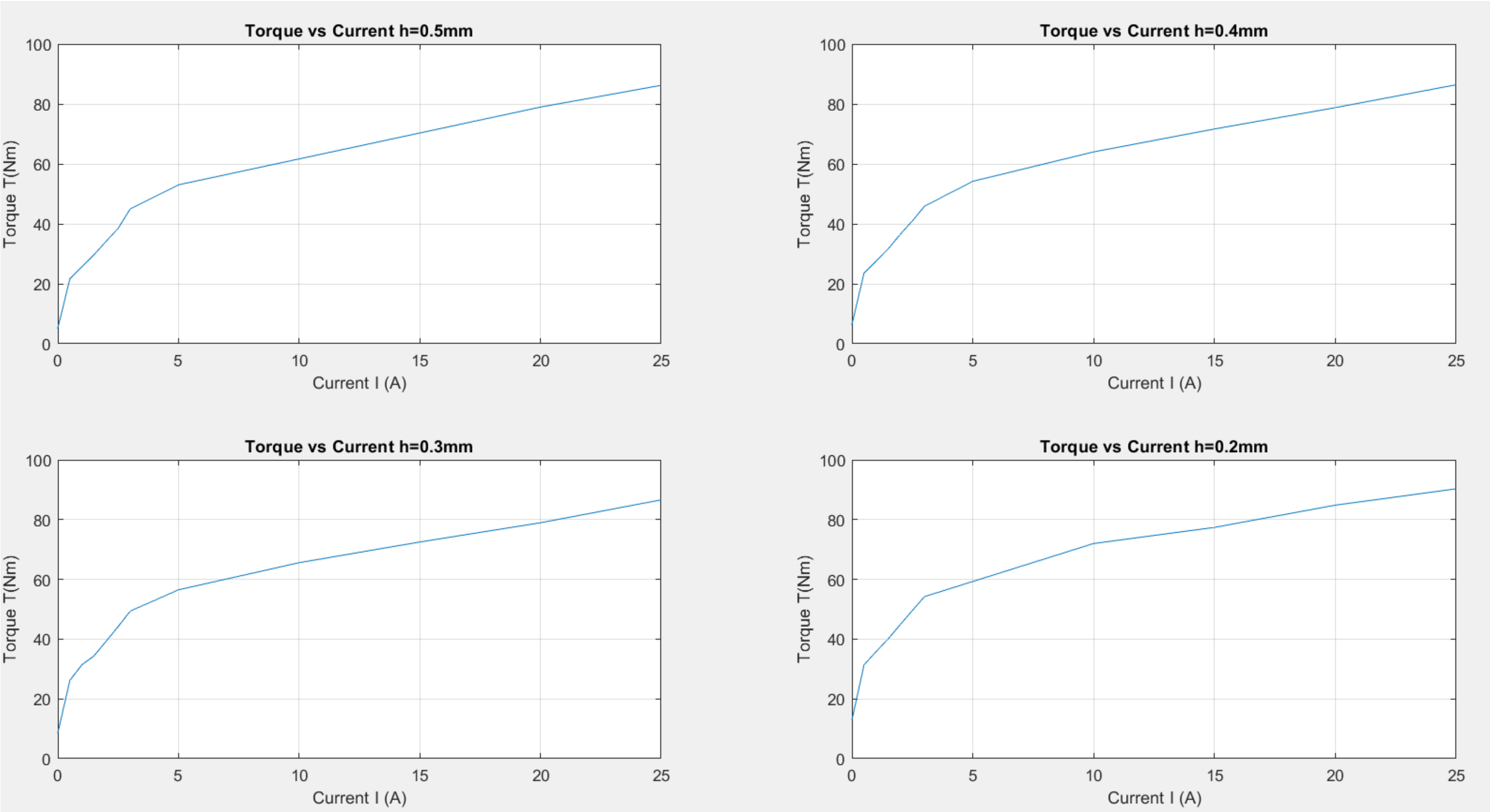


Fig 50 Torque vs Current for various gaps (r_{out}=110mm)

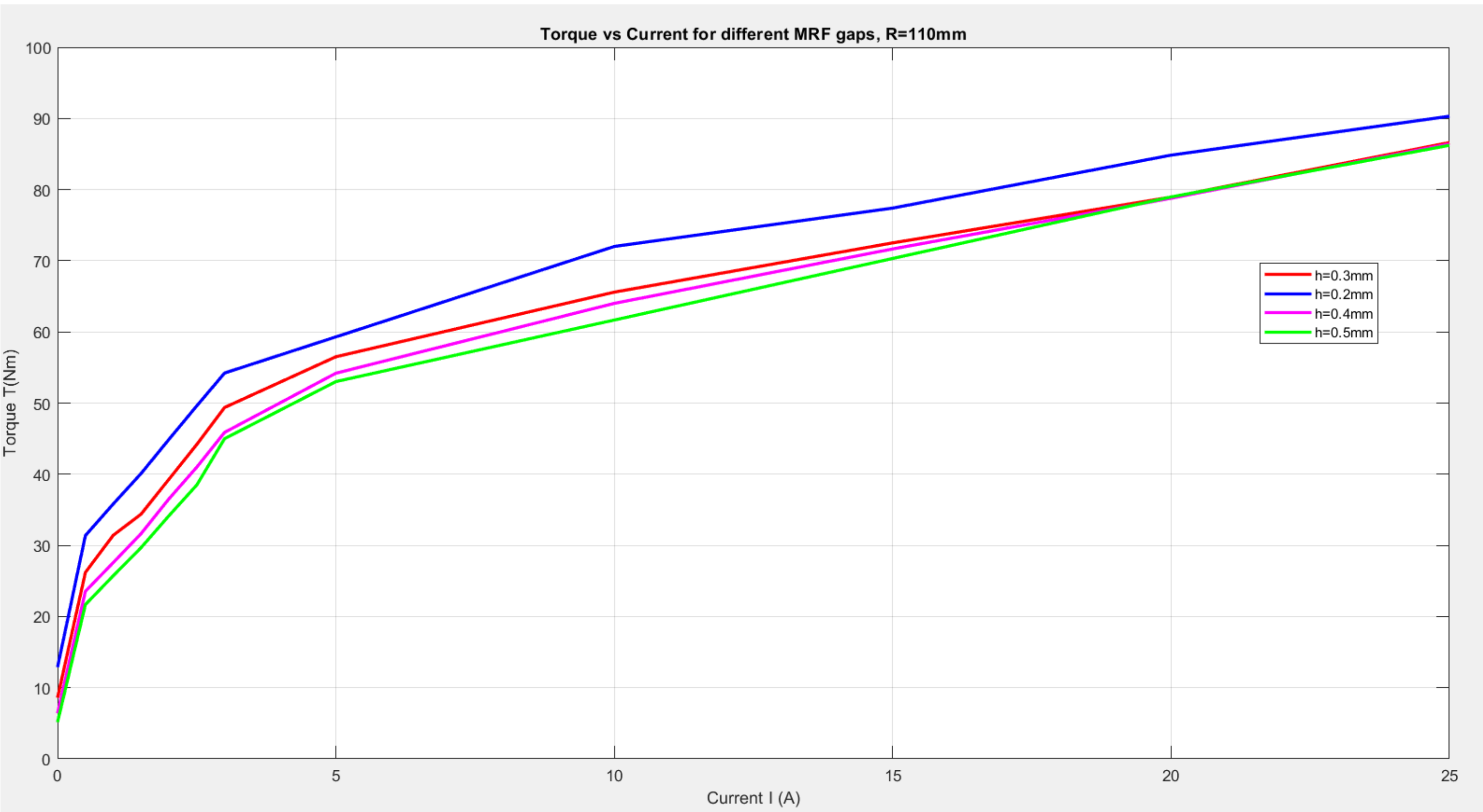


Fig 51 Torque vs Current (comparison between various gaps r_{out}=110mm)

In **Fig** 46, **Fig48** and **Fig** 50 are shown the torque current curves, separately for each gap, with r_{out}70mm, r_{out}90mm and r_{out}110mm, respectively. In **Fig** 47, **Fig49** and **Fig** 51 the torque-current curves are plotted on the same diagram in an attempt to compare them. It is concluded that each geometric configuration achieves its maximum torque transfer capability when the current input is the highest; at this case 25A. The maximum values for each configuration are concentrated in **Table 6** . By comparing the diagrams, included the **Fig** 28, we can divide them in 4 areas. The range of each ''operating'' area becomes bigger for larger discs. In the first area, from 0A to some point, the smaller gaps transfer higher amount of torque. In this area a small change in input current, results a big increase of the torque. Then it follows the second area. This area is transitional. In transitional area for smaller gaps the ratio of increase drops while for bigger gaps this rate does not vary. Here the bigger gaps between the discs start to have better performance than smaller gaps. In the third area, the transition is completed and now the bigger gaps transfer higher amount of torque. In that area ratio of increase for the torque starts to drop. Finally in the last area all curves converge to the maximum possible torque, each gap can transfer. Here the clutch is in saturation situation and significant increase in current input, does not change significantly the torque. The last area is shown only in **Fig** 28. In the other diagrams, where the r_{out} values are higher, 25A are not enough to put the clutch in saturation situation, despite the fact that there are disc areas where the magnetic field intensity is larger than 300kA/m. This occurs because of the wide range distribution of the field. The MRF's yield stress reaches at its maximum value, near the end of the discs, but near the center of the discs the yield stress is still very low.

MAXIMUM CALCULATED TORQUE FOR EACH GEOMETRIC CONFIGURATION			
	MRF GAP h (mm)	Maximum Calculated Torque (Nm)	Maximum torque capacity coverage
Maximum r_{out}=70mm	h=0.5	31.74	87.90%
	h=0.4	31.20	85.90%
	h=0.3	30.74	83.81%
	h=0.2	29.68	79.40%
Maximum r_{out}=90mm	h=0.5	55.14	71.36%
	h=0.4	54.54	70.06%
	h=0.3	53.54	67.94%
	h=0.2	52.52	65.06%
Maximum r_{out}=110mm	h=0.5	86.22	60.71%
	h=0.4	86.42	60.30%
	h=0.3	86.61	59.55%
	h=0.2	90.31	60.31%

Table 6 Maximum Calculated Torque for each configuration

From the table above it is obvious that for bigger discs the 25A is not value able to make the clutch transfer its maximum possible torque. Especially, for r_{out}=110mm, for input of 25A the clutch is still in the first operating area.

GROOVED DISC SURFACES

As it has already been mentioned, for lower current inputs the clutch performs better for smaller gaps between the discs. In this section, it is attempted to enhance the intensity of the magnetic field inside the gap by creating groove in the surfaces of both discs. The groove is rectangular in the center of both discs.

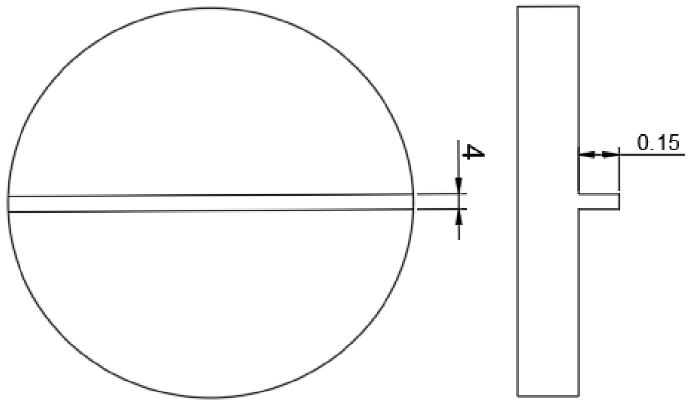


Fig 52 Grooved disc

In **Fig 52** is shown the geometry of the disc and the dimensions. This configuration chosen in an effort to decrease the MRF gap near the center of the discs, because it has been approved before that in smaller gaps the field's intensity becomes higher.

The clutch simulated for r_{out}=50mm and h=0.5mm. That means that in the groove the MRF gap is h=0.2mm. The input currents were 0.5A to 3A with step of 0.5A. For the calculation of the shear stress used the Eq11. For the torque calculation used the Eq12.

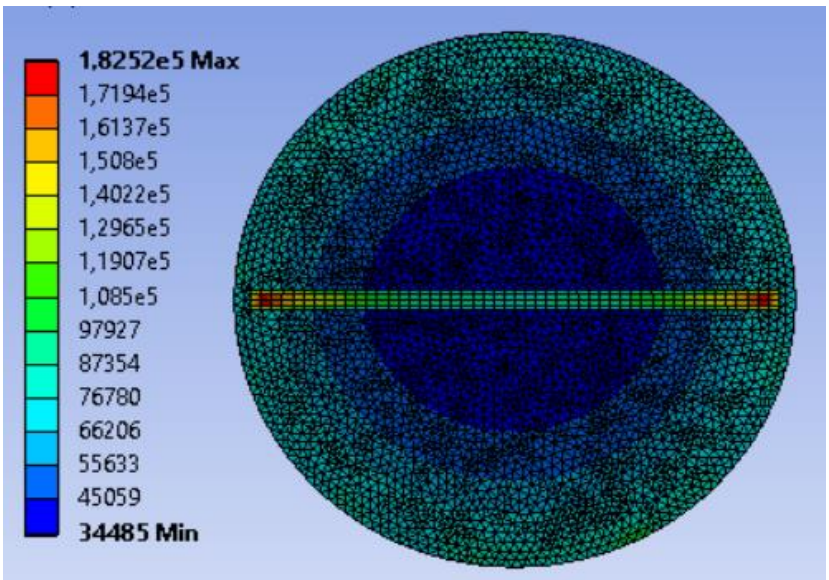


Fig 53 Magnetic Field Intensity for grooved surface discs (I=1A)

In **Fig 53** is shown the magnetic field intensity for I=1A. It is observed that the intensity inside the MRF is distinguished in two separate areas. The distribution of the field in both areas is uniform around the center. The first area is inside the groove and the second is the rest of the MRF. Inside the groove the field has larger value. In the present thesis it is assumed that the field inside the groove varies only with respect to the distance of the center.

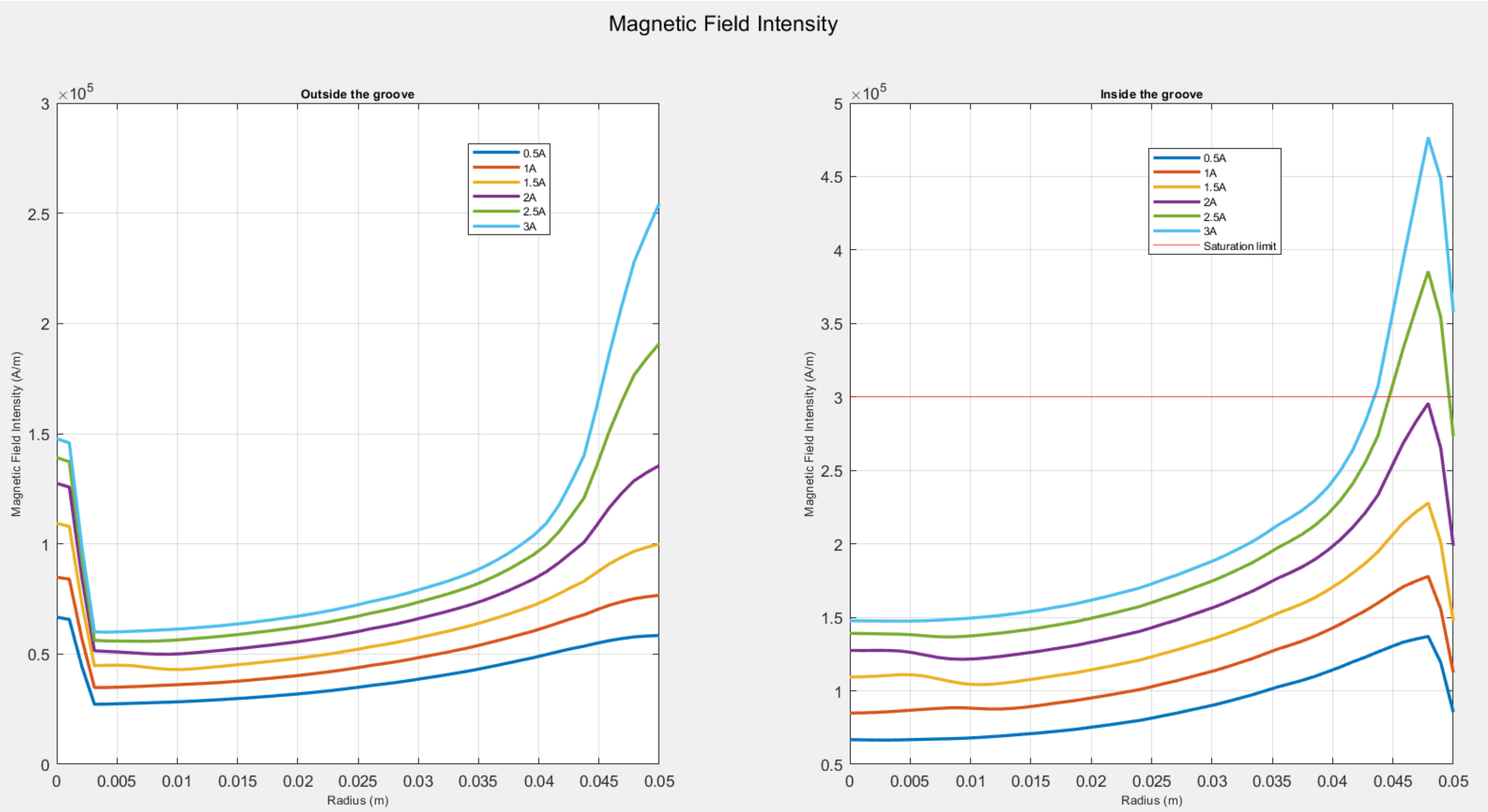


Fig 54 Distribution of the Magnetic Field Intensity inside and outside the groove

From the diagrams in **Fig 54** it is shown that the magnetic field is more intense inside the groove than outside. Moreover for 2.5A and 3A the field exceeds the saturation limit inside the groove. The distribution outside the groove is similar to the distributions for no grooved disc. Inside the groove the field increases away from the center of the disc, because it is closer to the coil. However near the end of the disc, the field decreases significantly, obviously affected by the fact that it approaches regions of the fluid with lower field intensity.

As it described before for the calculation of the torque used the Eq12. For a disc with a rectangular groove, the torque equation splits in two parts: one for the rectangular $T_r$and one for the rest of the disc T_d. The total torque is the sum of the$T = T_r + T_d$ (22). For the calculation of τ_y was followed the same procedure it was followed before. T_r and T_d are given below.

$$T_r = \frac{\alpha\tau_{yr}}{2}(r_{out}^2 - r_{in}^2) + 2\frac{\alpha\mu\Delta\omega}{3h_r}(r_{out}^3 - r_{in}^3) \text{ (22a)}$$

with a being the height of the groove ,τ_{yr} being the yield stress inside the groove and h_r the MRF gap inside the groove.

$$T_d = \frac{2\pi\tau_y}{3}(r_{out}^3 - r_{in}^3) + \frac{\pi\mu\Delta\omega}{2h}(r_{out}^4 - r_{in}^4) - [\frac{\alpha\tau_y}{2}(r_{out}^2 - r_{in}^2) + 2\frac{\alpha\mu\Delta\omega}{3h}(r_{out}^3 - r_{in}^3)] \text{ (22b)}$$

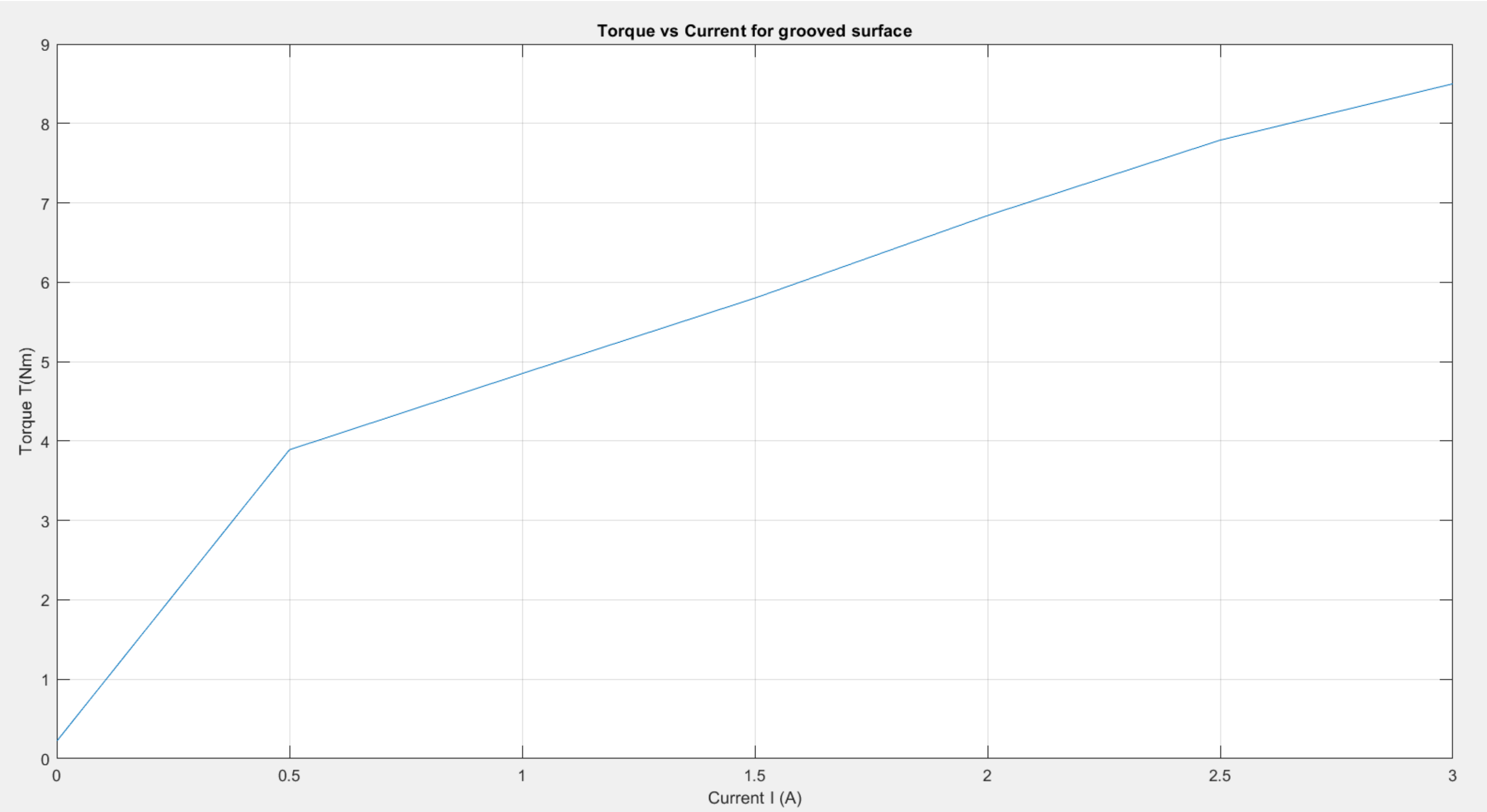


Fig 55 Torque vs Current for grooved surface

In **Fig 55** is shown the T-I curve for grooved surface. The curve has the same shape as the curve for plane discs. In the next figure (**Fig 56**) both curves, for plane disc and grooved disc are compared.

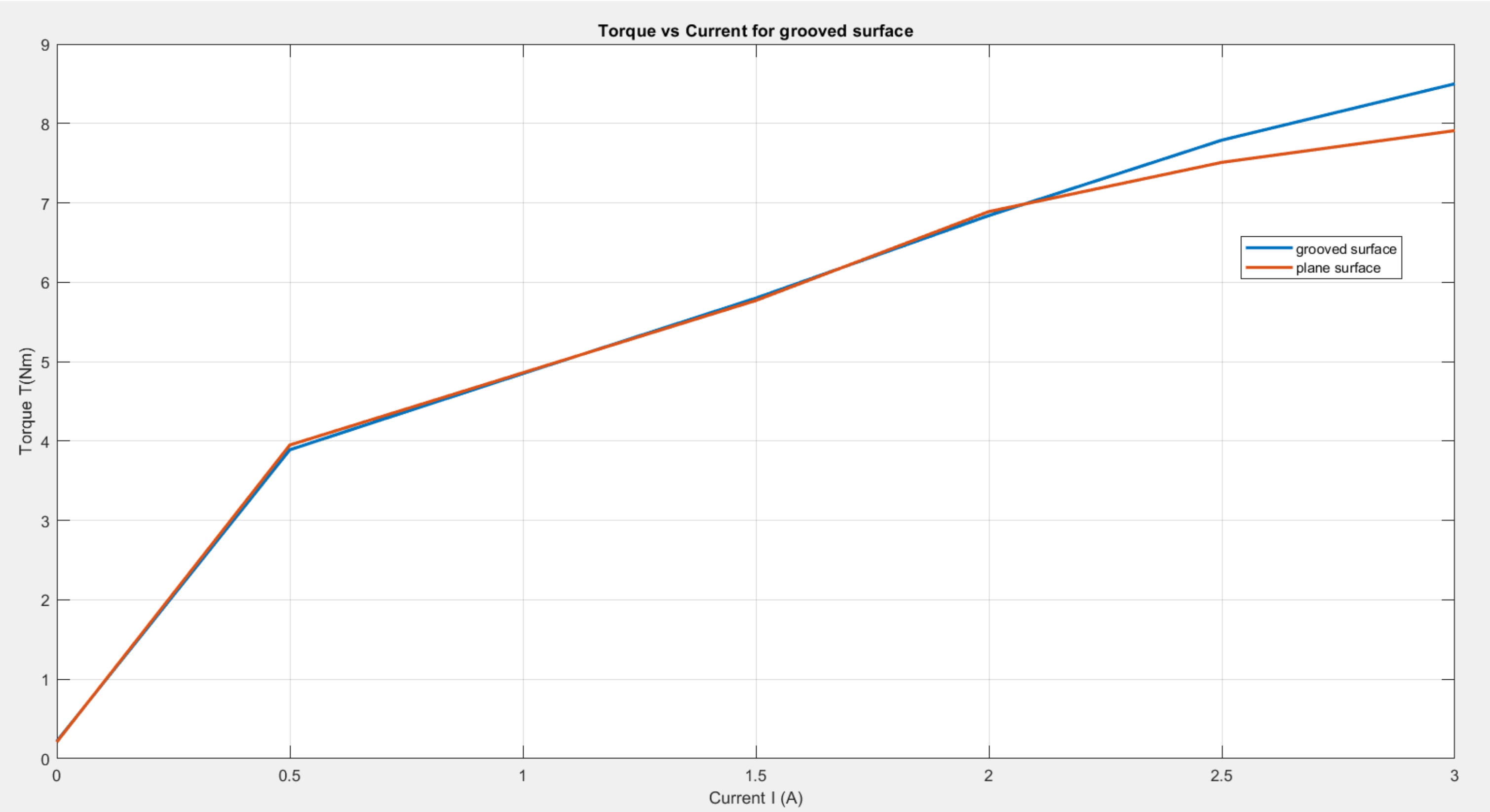


Fig 56 Comparison of T-I curves for grooved and plane surface

From the above diagram it can be concluded that both surfaces have almost the same behavior. The grooved surface seems to perform better for bigger current inputs (more than 2A). An explanation for this behavior is that the groove occupies only a small percent of the total surface, and it can not affect significantly the performance of the clutch. The area where a small divergence starts to occur is where the groove enters in saturation situation.

The maximum theoretical possible torque the grooved surface disc clutch can have is 13.074Nm, almost the same as the plane. Again the explanation is that the groove occupies only a small percent of the total surface. The maximum calculated torque is 8.50Nm at 3A. The Maximum Torque Capacity Coverage is 65% for the grooved discs clutch at 3A, while for plane discs clutch the Maximum Torque Capacity Coverage is 60% at 3A, meaning that grooved discs clutch is 5% more efficient than the plane discs clutch.

POWER CONSUMPTION OF THE CLUTCH

The most essential element for the operation of a MRF clutch is the electric current. Without current no magnetic field can be generated and as a consequence the MRF does not enter to its ''solid-like'' situation. In this section it is studied the power consumption of the electric circuit of the clutch. Like every electric circuit, the clutch's circuit consumes an amount of energy due to internal resistance of the wire. The power consumption of the wire is given by Eq17, while the internal resistance of the coil is given by Eq18. For this diploma thesis the wire's cross section is $A=4.1*10^{-7}$ m^2 and the wire's conductivity is $\sigma=6*10^7$ S/m and coil's turns are 300.

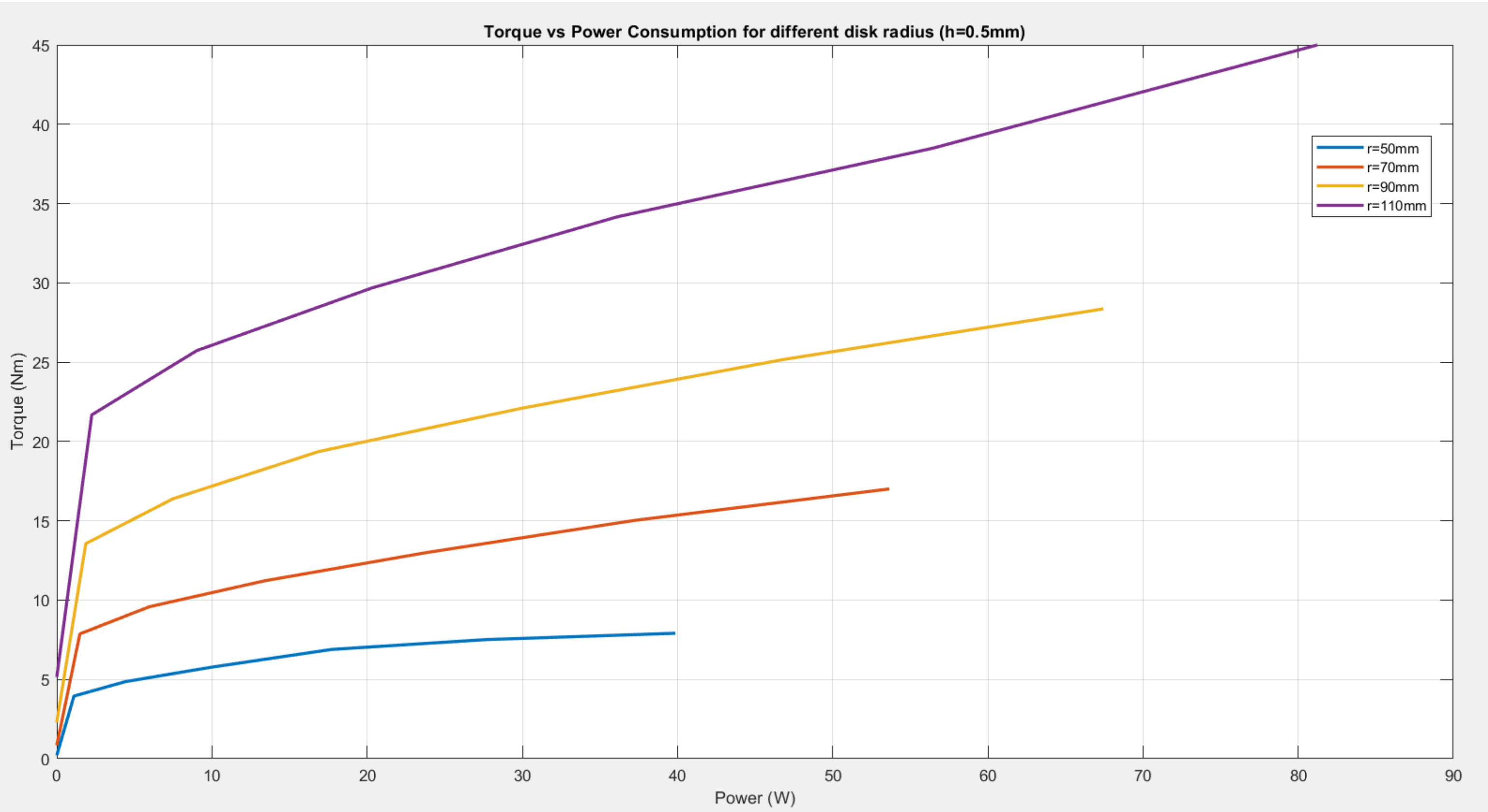


Fig57 Torque vs Power Consumption for various maximum radiuses

In **Fig57** is shown the power consumption for each geometric configuration. Smaller discs require less energy. In the diagram above the power consumption calculated only for currents from 0A to 3A. According to Eq17 higher current input results higher power consumption. This is illustrated in the diagram. Moreover bigger discs consume more power, which is absolutely reasonable because the coil has bigger diameter and as a consequence the wire is need is longer. This is also shown in the diagram. For r_{out}=50mm the power consumption for 3A is 39.8W while for r_{out}=110mm the power consumption for 3A is 81.24W.

In the previous diagram are shown the T-P_c curves for all examined r_{out} only for currents from 0A to3A. In the next **Fig58** diagram are presented the T-P_c curves included all the examined currents which make them more integrated.

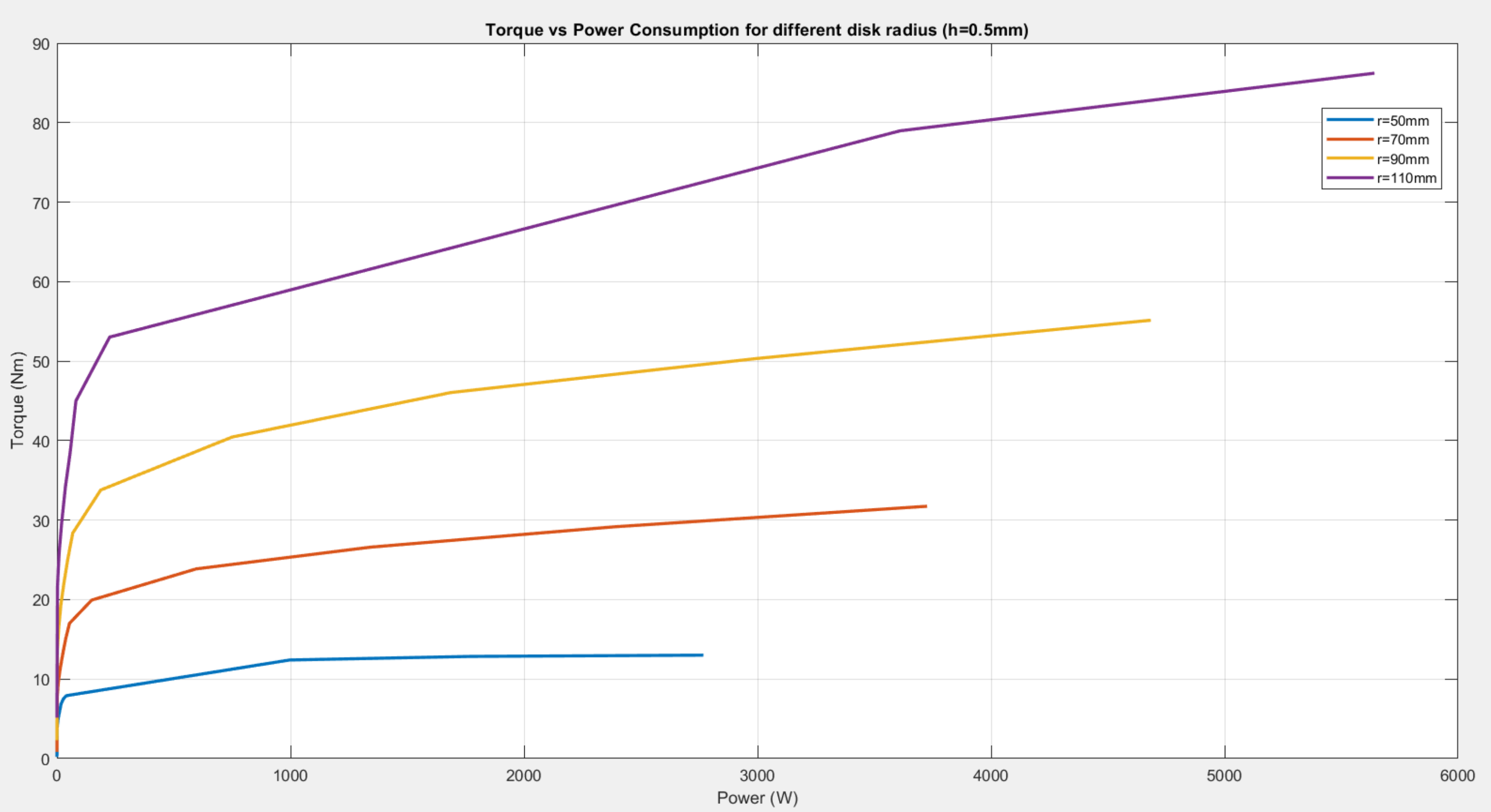


Fig58 T-P_c curves for all the currents

It is observed that for high values of input current the power consumption of the clutch, for each configuration, becomes enormous. The torque for r_{out}=50mm at 3A is 7.91Nm and the power consumption is 39.8W. At 25A the torque is 13.008 and the power consumption is 2768W. That means that for an increase of 5.098 Nm or 64% of torque the power must increase by 6854%. For r_{out}=110mm an increase in torque from 45Nm to 86.22 or 91.6% of requires a power increase by 6844%.

In order to estimate which clutch configuration is more energy efficient is introduced the coefficient $\lambda = \frac{T}{P_c}$(Nm/W) (23). From the observation of the diagrams it is obvious that bigger discs have better λ than smaller discs. That means that for the same amount of energy bigger discs have larger exit torque. In the next table are summarized the λ coefficients for each r_{out} for h=0.5mm.

r_{out} (mm)	Current I (A)	$\lambda (Nm/P_c)$
50	0.5	3.56
	1	1.09
	1.5	0.58
	2	0.39
	2.5	0.27
	3	0.20
70	0.5	5.28
	1	1.60
	1.5	0.84
	2	0.55
	2.5	0.40
	3	0.32
90	0.5	7.24
	1	2.19
	1.5	1.15
	2	0.74
	2.5	0.54
	3	0.42
110	0.5	9.61
	1	2.85
	1.5	1.46
	2	0.95
	2.5	0.68
	3	0.55

Table 7 λ coefficient for all r_{out} (0.5A-3A)

In the table above it is confirmed that bigger discs are more energy efficient. They require less energy to achieve the same amount of torque in comparison to smaller discs. Moreover it is also shown that lower input currents have better λ than higher inputs.

CHAPTER 4

CONCLUSIONS

In this diploma thesis it was attempted to illustrate the behavior of a MRF clutch. In general it was demonstrated how easily the clutch can change controllably the exit torque by varying only the current input.

At first examined a simple clutch with specific r_{in}, r_{out} and gap between the 2 discs. The first major conclusion comes from the T-I curve (**Fig 26**). The curve shows that torque increases by the increasing of the input current. This is reasonable because higher current has as result higher magnetic field intensity which means higher yield stress for the MRF.

Comparing the T-I curves for different MRF gaps (**Fig 28**) we can see that for every working gap the torque increases by the increasing of the current. Moreover it is shown that smaller gaps have better performance. A reason for that is that smaller gaps create smaller magnetic circuits with lower magnetic resistances which create more intense fields.

At specific Δω the clutch's ability to transfer torque is limited. This limit is set by the magnetic saturation of the MRF. For every MRF, its yield stress after a certain value of magnetic field intensity is not further increased. For this diploma thesis the chosen MRF was the LORD 132 DG with maximum yield stress of 49.1kPa at 300kA/m. When the current input increases, the MRF enters in saturation, meaning that torque does not significantly increases. This is visualized in **Fig31**, where are shown the T-I curves for each gap. All the 4 curves converge to their maximum value. Despite the fact that the field exceeds the saturation limit only in small radial area, all the curves approach very close to their maximum value for input of 25A. In **Table 4** are summarized the maximum calculated values for each gap and on what percent they cover the maximum theoretical torque. In addition, from **Fig31** it is shown that when clutch is in saturation state, bigger gaps tend to be more efficient.

Furthermore, it has been approved that bigger discs have better performance, despite the fact that the field is becoming weaker as the r_{out} increases. That means that for a MRF clutch the disc's surface plays more important role, than the field. In addition for bigger discs the entrance into saturation situation comes for larger inputs, despite the fact that there are areas beyond the saturation limit.

For the grooved design which was studied in the present diploma thesis, it was shown that no there is no significant change in torque comparing with plane discs. However it should be noticed again that the shear stress calculated with analytical expression (Eq11).

Finally for power consumption of the clutch it has been shown that bigger discs are more energy efficient, because they can transfer a certain amount of torque with less power consumption (**Table 7**). Moreover for every r_{out} it is shown that for current input larger than 3A the power consumption is enormous without important gains in torque (**Fig58**).

In conclusion from this diploma thesis it is obvious that according to the limitations of the design, a MRF clutch should be as big as possible. In order to be affordable its operation, it should not operate with currents larger than 3A. Because the clutch is proposed to work for low current inputs the gaps between the discs it is proposed to be as small as possible. The surface of the discs, affect the resulting torque, however the pattern was examined in the present thesis does not contribute significantly to the exit torque. Furthermore if for some reason the clutch is intended to operate in saturation state, bigger gaps between the discs should be preferred.

FURTHER WORK

In the present diploma thesis it was examined the MRF clutch under many assumptions. One very important assumption is that the problem is isothermal. One proposition for future work is to take into consideration the thermal problem and how it affects the MRF.

In addition it is very important to investigate a clutch with more than 2 discs. This configuration creates a total different magnetic field inside the working gap. More than two discs are able to transfer higher amount of torque, however the magnetic field perhaps is not strong enough and the corresponding yield stress not so high.

Also, in order to obtain more accurate results it is important to study how the ferromagnetic particles are distributed inside the fluid carrier. In this thesis it is assumed that the particles are uniformly distributed inside the fluid carrier and the MRF has the same magnetic properties everywhere. This has to be investigated because the centrifugal forces may push the ferromagnetic particles away from the center of the discs, changing the magnetic properties of the MRF.

Moreover there are many roughness profiles and groove patterns they can be tested to determine if there is someone with better performance.

One more field which is not studied widely is the influence of the time on a MRF clutch. It is very important to for a designer to have good knowledge of the MRF's behavior after working circles for some time and the life expectancy of the MRF.

Last but not least is the study of the clutch's materials (discs, bearings, coil, core etc). The materials must be appropriate for usage in a magnetic circuit (as high as possible μ_r) and at the same moment should be able to withstand the loads they will experience (magnetic forces, pressure from the fluids, rotational forces).

All the previous suggestions for further investigation have as primary target to increase the energy efficiency of the clutch in order to spend less energy with more benefit in torque exit.

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