AUGUST 2019 M.Sc. in Mechanical Engineering ALI AL MASHHADANI

REPUBLIC OF TURKEY

GAZİANTEP UNIVERSITY

GRADUATE SCHOOL OF NATURAL & APPLIED SCIENCES

DESIGN AND MANUFACTURING OF A NEW TYPE TORQUE CONNECTION

M.Sc. THESIS

IN

MECHANICAL ENGINEERING

BY

ALI AL MASHHADANI

AUGUST 2019

DESIGN AND MANUFACTURING OF A NEW TYPE TORQUE CONNECTION

M.Sc. Thesis

in

Mechanical Engineering

Gaziantep University

Supervisor

Asst. Prof. Dr. Abdullah AKPOLAT

Co-Supervisor

Prof. Dr. Nihat YILDIRIM

by

Ali AL MASHHADANI

August 2019

REPUBLIC OF TURKEY
GAZİANTEP UNIVERSITY
GRADUATE SCHOOL OF NATURAL & APPLIED SCIENCES
MECHANICAL ENGINEERING

Name of the Thesis : Design and Manufacturing of a New Type Torque Connection

Name of the Student : Ali AL MASHHADANI

Exam Date : 19.08.2019

Approval of the Graduate School of Natural and Applied Sciences

Prof. Dr. A. Necmeddin YAZICI
Director

I certify that this thesis satisfies all the requirements as a thesis for the degree of Master of Science.

Prof. Dr. Mustafa GÜNAL
Head of Department

This is to certify that we have read this thesis and that in our consensus opinion it is fully adequate, in scope and quality, as a thesis for the degree of Master of Science.

Prof. Dr. Nihat YILDIRIM
Co-Supervisor

Asst. Prof. Dr. Abdullah AKPOLAT
Supervisor Examining

Committee Members	Signature
Assoc. Prof. Dr. Bülent AKTAŞ	…………………...
Assoc. Prof. Dr. Ahmet ERKLİĞ	…………………...
Asst. Prof. Dr. Abdullah AKPOLAT	…………………...

I hereby declare that all information in this document has been obtained and presented in accordance with academic rules and ethical conduct. I also declare that, as required by these rules and conduct, I have fully cited and referenced all materials that are not original to this work.

Ali AL MASHHADANI

ABSTRACT

DESIGN AND MANUFACTURING OF A NEW TYPE TORQUE CONNECTION

AL MASHHADANI, Ali
M.Sc. in Mechanical Engineering
Supervisor: Asst. Prof. Dr. Abdullah AKPOLAT
Co-Supervisor: Prof. Dr. Nihat YILDIRIM
August 2019
83 pages

Torque connection (coupling) is termed as a device used to connect two shafts to each other at their ends for the purpose of transmitting power. The couplings can be divided into two main groups, first is rigid coupling second type is flexible coupling. Rigid coupling used to connect two shafts which are perfectly aligned and flexible coupling used to join two rotating parts and allow some degree of misalignment. The flexible coupling also falls into two main categories: Material Flexing and Mechanical Flexing. The material flexible types obtain their flexibility from stretching or compressing a resilient material or from the flexing of thin metallic discs. The mechanical flexing couplings accept misalignment from rocking, rolling or sliding of metal surfaces. In this thesis, design and simulation of a modified coupling was studied. Design of coupling was conducted by solid modeling CAD software (SOLIDWORKS 2018). After preparing the 3D model in SOLIDWORKS, the numerical analysis was obtained with the use of ANSYS 16.2 software. Manufacturing of each part of the new design was explained in details. Comparisons made between the new design and standard flange coupling with respect to load capacity and misalignment. The new design is an attempt to reach a better performing coupling. By combining the two types of the main categories: Material Flexing and Mechanical Flexing type of couplings, increasing the critical part’s area to decrease the stress that may be occurring, giving some parts the ability to slide and with the presence of the resilient material, high misalignment and high torque capacity with more reliability were obtained.

Key Words: Couplings**,** Torque Connections, Misalignment, Stress Analysis, FEM.

ÖZET

YENİ TİP BİR TORK AKTARIM ÜNİTESİ TASARIM VE İMALATI

AL MASHHADANİ, Ali
Yüksek Lisans Tezi, Makine Mühendisliği
Danışman: Dr. Öğr. Üyesi Abdullah AKPOLAT
İkinci Danışman: Prof. Dr. Nihat YILDIRIM
Ağustos 2019
83 sayfa

Tork bağlantısı (kaplin), iki şaft veya dönen parçaları, güç iletmek amacıyla uçlarından birbirine bağlamak için kullanılan bir cihaz olarak adlandırılır. Kaplinler iki ana gruba ayrılabilir, birincisi rijid bağlantı, ikinci tür ise esnek bağlantıdır. Rijid kaplinler iki makine elemanını birbirine çok hassas olarak bağlamak için kullanılır. Esnek kaplinler ise iki dönen parçayı bir dereceye kadar hizalama hatalarına izin vermek için kullanılan esnek bir bağlantıdır. Esnek bağlantılar da malzeme ve mekanik esnekliği olmak üzere iki ana kategoriye ayrılır. Esnek malzeme türleri esnekliklerini, kauçuk gibi esnek bir malzemenin gerilmesinden veya sıkıştırılmasından veya ince metalik disklerin veya ızgaraların esnemesinden alırlar. Mekanik esneme kavramaları metal yüzeylerin sallanmasından, yuvarlanmasından veya kaymasından hizalama hatalarını kabul eder. Bu tez çalışmasında, yeni geliştirilen bir kaplinin tasarımı ve benzeşimi üzerinde durulmuştur. Kaplin tasarımı, katı modelleme CAD yazılımıyla (SOLIDWORKS 2018) gerçekleştirilmiştir. Sayısal analizler ise ANSYS 16.2 yazılımı kullanılarak elde edilmiştir. Yeni tasarımın her bir bölümünün imalatı detaylı olarak açıklanmıştır. Yeni tasarlanan kaplin ile standart flanş kaplin arasındaki karşılaştırmalar yük kapasitesine ve hizalama hatası değerlerine göre yapılmıştır. Yeni tasarım daha iyi performans gösteren bir bağlantıya ulaşma çabasıdır. Ana kategorilerin iki tipini birleştirerek: Malzeme Esneme ve Mekanik Esneme tipi kaplinler, oluşabilecek gerilimi azaltmak için kritik parçanın alanını arttırarak, bazı parçalara kayma kabiliyeti ve esnek malzemenin varlığı ile birlikte hizalama hatası değeri ve daha fazla güvenilirlik ile yüksek tork kapasitesi elde edilmiştir.

Anahtar Kelimeler: Kaplin, Tork Bağlantısı, Hizalama Hatası, Gerilme Analizi, SEM.

- To my family, My parents, my borthers and sisters.

- To my teachers form primery school to engneering college

- To all my teachers at Gaziantep University

- To my all my friends in Iraq and Turkey

ACKNOWLEDGEMENTS

I would like to express my special appreciation and gratitude to my supervisors, Assist. Prof. Dr. Abdullah AKPOLAT and Prof. Dr. Nihat YILDIRIM, for all their help, patience, valuable advice, always providing and guiding me in the right direction. I'm very grateful and proud to work under their academic guidance.

Special thanks to my friend Bahadır Karba and his assistance in the ANSYS software.

I wish to express my special thanks to my second family in Turkey, my friends who for their help and support through this research.

I would like to express heartfelt appreciation to The Republic of Turkey and their people for hospitality and generosity throughout my study which cannot be denied.
Special thanks to my family. Words cannot express how grateful I am to my father and my mother for all of the sacrifices that she made for me. Also to all my siblings, for the many years of encouragement and support they gave me and which enabled me to complete this thesis.

TABLE OF CONTENTS

ABSTRACT .... **v**
ÖZET .... **vi**
ACKNOWLEDGEMENTS .... **viii**
LIST oF TABLES .... **xii**
LIST OF FIGURES .... **xiv**
LIST OF SYMBOL .... **xvii**
CHAPTER 1: INTRODUCTION **1**
1.1 Introduction 1
1.2 Misalignment 1
1.3 Good Shaft Couplings Requirements and Its Common Uses 3
1.4 Couplings Categories 3
1.5 Coupling Maintenance and Failure 5
1.6 Checking The Coupling Balance 7
1.7 Objectives of this Study 8
CHAPTER 2: LITERATURE REVIEW **9**
2.1 Introduction 9
2.2 Literature Review 9
CHAPTER 3: THEORY **15**
3.1 Introduction 15
3.2 Couplings Types 15
3.2.1 Rigid Coupling: 15
3.2.2 Flexible Coupling 17
CHAPTER 4: DESIGN **23**
4.1 Introduction 23
4.2 Modified Flange Coupling Calculation 24
4.2.1 Shaft Design 24
4.2.2 Pin Design 25

4.2.3 Hub Design ... 26

4.2.4 Key Selection ... 27

4.2.5 Bolt Design ... 27

4.2.6 Flange design ... 27

4.3 Pin Stress Analysis ... 28

4.4 CAD Modeling ... 28

CHAPTER 5: ANSYS RESULTS ... 30

5.1 Introduction ... 30

5.2 Meshing ... 30

5.3 Contacts Type ... 31

5.4 ANSYS Results (3MA, Three Misalignment) ... 32

5.4.1 Shear Stress ... 32

5.4.2 Equivalent Stress ... 32

5.5 Modified Coupling (2MA, Two Misalignment) ... 33

5.5.1 Equivalent Stress ... 33

5.5.2 Shear Stress ... 34

5.5.3 Total Deformation ... 34

CHAPTER 6: MANUFACTURING ... 36

6.1 Introduction ... 36

6.2 Technical Drawings ... 36

6.2.1 Middle Holder ... 36

6.2.2 Major Middle Part ... 36

6.2.3 Secondary Middle Part ... 37

6.2.4 Pin ... 38

6.2.5 Plate ... 38

6.2.6 Pin Holder ... 39

6.2.7 Polyurethane ... 39

6.3 Manufacturing Process ... 40

6.3.1 Middle Holder ... 40

6.3.2 Major Middle Part ... 41
6.3.3 Secondary Middle Part ... 43
6.3.4 Pin ... 45
6.3.5 Pin Holder ... 45
6.3.6 Polyurethane ... 46
CHAPTER 7: COMPARISON ... 47
7.1 Introduction ... 47
7.2 Design of Couplings ... 48
7.2.1 Calculations ... 48
7.2.2 CAD Modeling ... 51
7.2.3 Meshing ... 52
7.2.4 Standard Flange Coupling ... 53
7.3 Misalignment Comparison ... 56
7.4 Experimental Work ... 63
7.5 Results ... 65
CHAPTER 8: CONCLUSION ... 66
8.1 Conclusion ... 66
8.2 Recommendation for Future Work ... 67
REFERENCES ... 68
APPENDICES ... 72
APPENDIX A ... 73
APPENDIX B ... 78
APPENDIX C ... 81
APPENDIX D ... 82

LIST OF TABLES

Table 1.1 Examples of the coupling failure and it’s causes 6
Table 4.1 Material names of each part 24
Table 4.2 Design specification of modified coupling 28
Table 5.1 Mesh statistics of model 30
Table 5.2 ANSYS contact definition 31
Table 5.3 Shear stress values 32
Table 5.4 Minimum and maximum equivalent stress values 33
Table 5.5 Minimum and maximum equivalent stress values 34
Table 5.6 Shear stress values 34
Table 5.7 Total deformation values 35
Table 5.8 Physical properties, applied load and boundary condition 35
Table 7.1 Part’s material 48
Table 7.2 Design specification of standard flange coupling 51
Table 7.3 ANSYS inputs for standard flange coupling model 53
Table 7.4 Minimum and maximum shear stress values 54
Table 7.5 Minimum and maximum equivalent stress values 55
Table 7.6 Minimum and maximum total deformation values 56
Table 7.7 Comparison between Spiderflex coupling and the new design 57
Table 7.8 Comparison between Crownpin coupling and the new design 57
Table 7.9 Comparison between Tyreflex coupling and the new design 58
Table 7.10 Comparison between Chainflex coupling and the new design 59
Table 7.11 Comparison between FFX Tyre coupling and the new design 60
Table 7.12 Comparison between ED Double Engagement coupling and the new design 61
Table 7.13 Comparison between ET Spacer Tube coupling and the new design 61
Table 7.14 Comparison between RB type coupling and the new design 62
Table 7.15 Comparison between Universal joints double coupling and the new design 63

Table 8.1 Comparison between the modified coupling and standard flange coupling. 65

LIST OF FIGURES

Figure 1.1 Radial misalignment. 2
Figure 1.2 Angular misalignment. 2
Figure 1.3 Axial misalignment. 2
Figure 1.4 Couplings Categories. 4
Figure 3.1 Unprotected flange. 16
Figure 3.2 Protected flange coupling. 16
Figure 3.3 Marine flange coupling. 17
Figure 3.4 Sleeve or Muff-coupling. 17
Figure 3.5 Pin bush flexible coupling. [12] 18
Figure 3.6 Jaw couplings. 19
Figure 3.7 Tire coupling. 19
Figure 3.8 Disc coupling. 20
Figure 3.9 Gear coupling. 20
Figure 3.10 Grid coupling. 21
Figure 3.11 Roller Chain. 21
Figure 3.12 Universal coupling. 22
Figure 4.1 Initial design. 23
Figure 4.2 Final design. 23
Figure 4.3 Technical drawing of the new coupling. 29
Figure 5.1 ANSYS CAD model. 30
Figure 5.2 Meshed model. 31
Figure 5.3 The suspended contact. 31
Figure 5.4 Shear stress results. 32
Figure 5.5 Maximum shear stress position. 32
Figure 5.6 Equivalent stress results. 33
Figure 5.7 Maximum equivalent stress position. 33
Figure 5.8 Maximum equivalent stress position (2MA). 34
Figure 5.9 Maximum shear stress position (2MA). 34

Figure 5.10 Total deformation results. 35
Figure 6.1 Technical drawing of middle holder. 36
Figure 6.2 Technical drawing of middle part. 37
Figure 6.3 Technical drawing of secondary middle part. 37
Figure 6.4 Technical drawing of the pin. 38
Figure 6.5 Technical drawing of the plate. 38
Figure 6.6 Technical drawing of the pin holder. 39
Figure 6.7 Technical drawing of the polyurethane. 39
Figure 6.8 Middle holder bulk. 40
Figure 6.9 CNC milling proccess for middle holder. 40
Figure 6.10 Final shape for middle holder. 41
Figure 6.11 Hole EDM. 41
Figure 6.12 EDM machine. 42
Figure 6.13 EDM tool. 42
Figure 6.14 CNC machining. 42
Figure 6.15 Major middle part final shape. 43
Figure 6.16 CNC milling process for the secondary middle part. 43
Figure 6.17 Original bulk after the wire cutting process. 44
Figure 6.18 The EDM process for the secondary middle part. 44
Figure 6.19 The final shape of the secondary middle part. 44
Figure 6.20 The final shape of the secondary middle part focusing on therounded surface. 45
Figure 6.21 Shows the pin in the assembly. 45
Figure 6.22 CNC milling for Pin holder. 45
Figure 6.23 The final shape for the pin holder. 46
Figure 6.24 Mold for polyurethane casting. 46
Figure 6.25 Final shape for polyurethane. 46
Figure 7.1 Technical drawings of standard coupling. 51
Figure 7.2 3D CAD model of standard coupling. 51
Figure 7.3 Standard flange coupling. 52
Figure 7.4 Standard flange meshing. 52
Figure 7.5 Hub and flange meshing. 52
Figure 7.6 Bolts meshing. 53
Figure 7.7 Shear stress results for standard flange. 54

Figure 7.8 Maximum shear stress position. 54
Figure 7.9 Equivalent stress result. 55
Figure 7.10 Maximum equivalent stress position. 55
Figure 7.11 Total deformation results. 55
Figure 7.12 Spiderflex coupling. 56
Figure 7.13 Spiderflex technical drawing. 56
Figure 7.14 Crownpin coupling. 57
Figure 7.15 Crownpin technical drawing. 57
Figure 7.16 Tyreflex coupling. 58
Figure 7.17 Technical drawing for Tyreflex coupling. 58
Figure 7.18 Chainflex coupling. 59
Figure 7.19 Technical drawing for Chainflex coupling. 59
Figure 7.20 FFX Tyre coupling technical drawing. 60
Figure 7.21 ED Double Engagement coupling technical drawing. 60
Figure 7.22 ET Spacer Tube coupling technical drawing. 61
Figure 7.23 RB type coupling technical drawing. 62
Figure 7.24 Universal joints double. 62
Figure 7.25 Universal joints double technical drawing. 62
Figure 7.26 CAD model for the designed mechanism. 63
Figure 7.27 The manufactured mechanism. 63
Figure 7.28 (a) The coupling before the tests (b) The coupling after the first test (c) The coupling after the second and third tests. 65

LIST OF SYMBOL

T	Applied torque
d	Shaft diameter
τ_{σ}	Shear stress
d_{pc}	Pin diameter
n	Number of pins or bolts
d_{p}	Pin diameter with the resilient material
P_{D}	Pin pitch diameter
p_{total}	Total load
p	Load on each pin
D	Hub diameter
L	Hub length
w	Key width
t	Key thickness
D_{pb1}	The pitch diameter of the first group
$Dbolt_{1}$	Bolt diameter of the first group
D_{pb2}	The pitch diameter of the second group
$Dbolt_{2}$	Bolt diameter of the second group
D_{f}	Flange diameter
t_{f}	Flange thickness
J	Polar moment of inertia

CHAPTER 1

INTRODUCTION

1.1 Introduction

The standard shaft length is usually up to 7 meters, due to difficulties in transport. In case that a length more than 7 meters is needed, it’s necessary to join more than one piece by the use of a coupling [1]. Coupling or torque connections are a mechanical element used to connect two rotating parts and enables motion transporting. Shaft couplings are usually used in machines for some purposes, the primary uses of couplings are to connect two rotating parts while giving the ability to tolerate some misalignment or the movement that may occur between the two ends or both.

The two main categories of couplings are rigid coupling and flexible coupling. Rigid couplings are used to transfer high torque between two shafts with perfect alignment while flexible couplings are used to transfer torque, which lower than that one in rigid coupling with an allowance of some misalignment. Flexible coupling falls into two main categories according to the way of gaining the ability to tolerate the misalignment. The first type of flexible coupling is obtaining the ability to tolerate the misalignment due to the material properties like rubber in bushed pin coupling. The second type of flexible coupling is obtaining the ability to tolerate the misalignment due to the clearance between the meshing parts like gear, chain coupling.

1.2 Misalignment

The variation of relative position of the shaft from the collinear rotational axis under operating in normal conditions can be termed as misalignment. [2]. Misalignment of shafts occurs when there is no matching between the center lines of coupled shafts [3].

Misalignment can be as three types:

Radial Misalignment Can also be termed as parallel misalignment due to the two shafts, or axes, are parallel, but with different center line [3]. (Figure 1.1)

Figure 1.1 Radial misalignment.

Angular Misalignment in this type of misalignment, the angle is involved. It happens when the two shaft's axes are not on the same angle of rotation [3]. (Figure 1.2)

Figure 1.2 Angular misalignment.

Axial Misalignment in this type, the two ends of each shaft don't meet. They are parallel, but when fully extended, the ends do not come together. This kind of misalignment can be also referred to an end float due to the unconnected ends of the shafts "float", meaning they can move in and out [3]. (Figure 1.3)

Figure 1.3 Axial misalignment.

Axial misalignment is also defined as the deviation in the axial distance between the ends of shafts. The power consumption that caused due to Radial (Offset) misalignment is higher than the one that may be caused by the angular misalignment. The misalignment components are additive without considering that they are horizontal or vertical [4]. For driver and driven, the process of positioning the previous shaft's center lines to create collinear shafts can be defined as shaft alignment, where coupled shafts rotational center lines are intersecting (like a single shaft) and parallel.

1.3 Good Shaft Couplings Requirements and Its Common Uses

Shaft couplings, most common uses listed are the following:

1. To provide for the connection of shafts of units that are manufactured separately, provide for disconnection for repairs or alterations.

2. To provide for misalignment of the shafts or to introduce mechanical flexibility.

3. To reduce the transmission of shock loads from one shaft to another.

4. To introduce protection against overloads [1].

Good Shaft Coupling Requirements are:

1. It should be easy to connect or disconnect.

2. It should transmit the full power from one shaft to the other shaft without losses.

3. It should hold the shafts in perfect alignment.

4. It should reduce the transmission of shock loads from one shaft to another shaft.

5. It should have no projecting parts [1].

1.4 Couplings Categories

Couplings can have a huge variety, but all can be arranged according to these categories as shown in Figure 1.4.

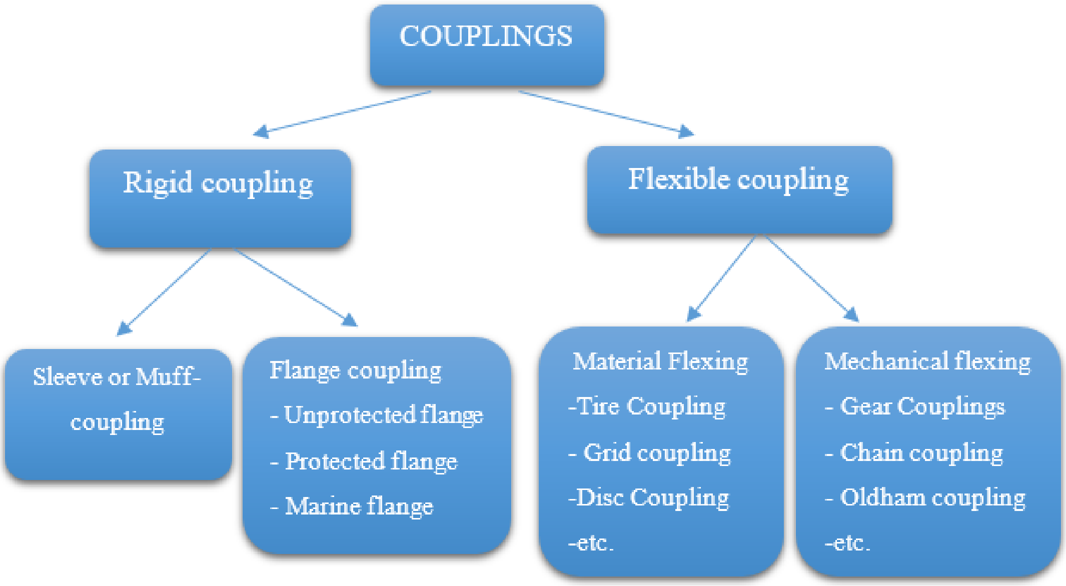


Figure 1.4 Couplings Categories.

Coupling in general can be arranged in two main categories,

1) Rigid coupling,
2) Flexible coupling.

Flexible coupling can also divide into two types according to its ability to tolerate the misalignment

- Material Flexing
- Mechanical Flexing

The first type of flexible coupling is obtaining the ability of tolerating the misalignment due to the material properties like rubber in bushed pin coupling and the grid in grid coupling.

The second type of flexible coupling is obtaining the ability of tolerating the misalignment due to the clearance between the meshing parts through rocking, rolling or sliding of metal surfaces. Like gear coupling and chain coupling [5].

1.5 Coupling Maintenance and Failure

Coupling maintenance requires a regularly scheduled inspection of each coupling. It consists of:

- Performing visual inspections,
- Checking for signs of wear or fatigue
- Cleaning couplings regularly
- Checking and changing lubricant regularly if the coupling is lubricated. This maintenance is required annually for most couplings and more frequently for couplings in adverse environments or in demanding operating conditions.
- Documenting the maintenance performed on each coupling, along with the date.

Even with proper maintenance, however, couplings can fail. Underlying reasons for failure, other than maintenance, include:

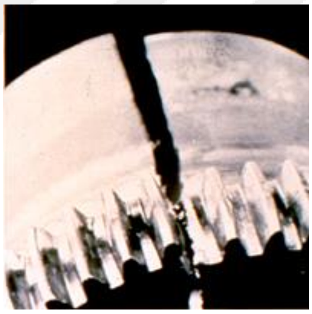
- Improper installation
- Poor coupling selection
- Operation beyond design capabilities.

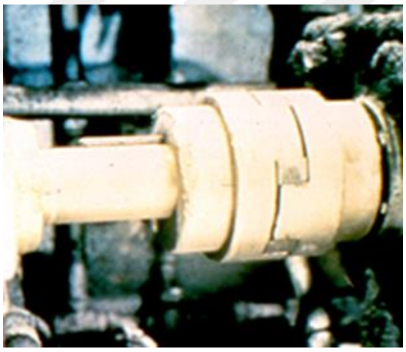
The only way to improve coupling life is to understand what caused the failure and to correct it prior to installing a new coupling. Some external signs that indicate potential coupling failure include:

- Abnormal noise, such as screeching, squealing or chattering
- Excessive vibration or wobble
- Failed seals indicated by lubricant leakage or contamination.

In addition, examination of damaged parts helps provide an explanation of what caused the failure. Table 1.1 shows some examples of the failure and it's causes.

Table 1.1 Examples of the coupling failure and it's causes

Failed part	Cause
Ruptured flexible element 	Excessive torque loading due to improper coupling selection or system changes
Cracked flexible element 	Hardened rubber due to chemical contamination
Split hub 	Improper installation
Key roll in seat 	Possibly due to improper hub size selection

Burst hub 	Improper size selection, coupled with overload
Elongated bolt holes 	Excessive torsional vibration or loading. In properly installed couplings, torque is carried by contact friction between coupling surfaces rather than by the bolts. Excessive vibration can loosen bolts, causing torque to be applied to bolts.
Accelerated wear and early failure 	Lack of maintenance

1.6 Checking The Coupling Balance

Couplings are normally balanced at the factory prior to being shipped, but they occasionally go out of balance in operation. Balancing can be difficult and expensive, and is normally done only when operating tolerances are such that the effort and the expense are justified. The amount of coupling unbalance that can be tolerated by any system is dictated by the characteristics of the specific connected machines and can be determined by detailed analysis or experience.

1.7 Objectives of this Study

Many types of couplings were designed and manufactured by companies and manufacturers. Each type of couplings has it’s own properties (torque capacity, dimensions, and misalignment allowances). Each coupling has advantages and disadvantages. The principal purpose of this research is to design and manufacture a coupling with better performance with more reliability and misalignment capacity. This can be reached by merging the material flexing type with mechanical flexing type. The designed coupling has the properties of mechanical flexing due to the sliding movement of the pins and the movement of the rounded head of the same pin in the grove due to tolerance. The design also has the properties of material flexing due to the presence of resilient material. This combination can give relatively high misalignment capacity comparing with the other flexible couplings. The design will also carry the ability to keep up with the rigid coupling in torque capacity field. The objectives of this study are to prove that the new design is more reliable according to developed stress with the use of FEM software, has better misalignment capacity comparing with other flexible coupling.

CHAPTER 2

LITERATURE REVIEW

2.1 Introduction

Rigid flange coupling is one of the most commonly used type. It is usually used for high torque, low-speed applications. Some researches and studies were made about flange coupling stress analysis (theoretical and numerical) as it will be discussed next.

2.2 Literature Review

A study was made about stress analysis for a flange coupling under torsional load with the use of ANSYS software [6]. The obtained stress from the theoretical calculation and ANSYS software were slightly different. Since the results achieved from ANSYS meet theoretical calculations, the design is safe. In another study, a model was made to reduce the maximum shear stress by selecting a suitable material for flange coupling [7]. Solidworks was used in the modeling process of the rigid-flange coupling, ANSYS Workbench for analysis. The analysis result of the keys and bolts of flange coupling that obtained from ANSYS is much better than the analytical method of calculation. From this comparison, it can be concluded that designing the coupling with the use of Solidworks is more suitable and safe.

In [8,9], both studies investigated the ability to reduce bolt’s stress that acts on it by making it uniform strengthen. The threaded part stress of the bolt is expected to be higher than shank’s stress. As a consequence, the threaded part’s region will be receiving the greater portion of the stress which may cause a fracture in the threaded portion due to it’s small length. At the bolt’s center, an axial hole has been drilled through the head as a threaded portion so that the stress is distributed uniformly along the bolt's length. The result from keys and bolts analysis of flange coupling using ANSYS workbench is better than the result from analytical method.

The obtained shear and crushing stress from the theoretical calculation is slightly higher than the one obtained from ANSYS. A bushed pin which is a kind of flexible coupling, was designed with the use of the design standard equations [10]. Applying amount of torque for the purposes of testing in order to find the main deformation in the coupling. In a try to enhance the design, to maximize the bearing effect for the coupling body. Solidworks simulation is used to visualize the failure points on the body. Thus, to find the main deformation, that occurs in the main body of the coupling, different bushes were used like brass, rubber, and aluminum or remove it completely. Brass bush tells that both of coupling sides deform with small amount of change because of the twisting, while in the rubber bush, the deformation is smaller and the torque loss is higher as acts. The aluminum bush does not give any difference comparing with the brass, as it also causes much deformation and stress for the coupling, so the best results can be obtained by using both of brass and rubber together. In [11], this study analyzed flange coupling with no clearance by using Generative Structural Analysis workbench of CATIA. Two unlike systems were analyzed. The system was a flange coupling with six bolts working under nominal conditions. The second system was a flange coupling with five bolts, working in some failure conditions, with five bolts only distributed in six holes of the flange coupling. However, the five bolts which are non-uniformly distributed around flanges six holes will create some other undesired phenomena. The flange coupling five bolts analysis allows observing that all the stresses and deformations will be non-symmetrical and for the flange coupling which is a rotating coupling, this can be one reason for beats phenomenon.

In [21], reducing maximum shear stresses by adding a new material between shaft and hub. The modeling of rigid flange coupling is done in CREO 2.0 and analysis of rigid flange coupling is carried out with the help of ANSYS 15 Software for calculated torque. The result is that the shear stresses developed in the flange coupling with Ti alloy ring are less than the shear stresses developed in coupling without Ti alloy ring.

In a try to improve the design to maximize the bearing effect for the coupling body [22]. Solidworks simulation was used in order to visualize the failure spots on the body. The maximum applied torque was 64 Nm, rubber and polyurethane were used.

For the other parts of the tire coupling (Alloy steel) was selected. Two materials were compared (Rubber and Polyurethane) to find the optimum design. The authors found that the rubber bears more stress than the polyurethane and this means that the rubber is better than polyurethane. Study that deals with the possible causes of failure for a flange coupling [23]. Flange coupling failure mostly occurs in the areas of contact. Analysis of such failures and suggestions to minimize the failures occurred thereof. In this regard, a model of the flange coupling was prepared with the help of an analysis workbench, proper material was assigned to that particular flange coupling and points of failure was noted accordingly. Secondly, a different material in fact, a better one which would withstand more amount of stress was assigned and failure results were noted again in order to compare between the two materials.

In [24], a concept of flexible type of coupling for power transmission in light load conditions. This coupling allows parallel, angular, torsional and axial misalignment. Modification in conventional coupling as a flexible coupling reduces its weight and radial space for installation considerably. As the inertia of coupling decreases, so the performance is better. During power transmission, such coupling is calm and quite. As there is no direct contact between two flanges so no problem of contact stresses as well as heat production. Also, compared to previously available belt coupling the axial distance is reduced without compromise in its strength. And this coupling provides radial rigidity and angular flexibility.

In [25], constitutes the failure analysis of a ‘Flange’ that had been welded to a high-water transmission pipeline. The flange had failed during the operation and that was the major point to take care. The analysis was conducted by using ANSYS Workbench 11.0 the integrity of the flange as well as that of the weld joint. The failure occurred along the weld on the flange side. The flange, which was a having material as structural steel, was expected to exhibit good weld ability. However, the analysis revealed that the structural steel had a less strength to allow the pressures. Hence, the new material alloy steel is suggested for the same and around 3 iterations are taken on Flange model in ANSYS.

In [26], experimental studies were performed on a rotor dynamic test apparatus to predict the vibration spectrum for rotor unbalance. A self-designed simplified 3 pin type flexible coupling was used in the experiments. The rotor shaft accelerations were measured at four different speeds using accelerometer and dual channel vibration analyzer (ADASH) under the balance (baseline) and unbalance conditions. The experimental and numerical (ANSYS) frequency spectra were also obtained for both base line and unbalanced condition under different unbalanced forces. The experimental predictions are in good agreement with the ANSYS results. Both the experimental and numerical (ANSYS) spectra show that unbalance can be characterized primarily by one times (1X) shaft running speed.

In [27], the pressure distribution on spline's coupling teeth has been investigated by means of analytical approaches and experimental techniques. In particular, the contact pressure distribution between the teeth in axial direction, obtained by means of analytical models available in literature, has been compared with both experimental data and finite element method models. The experimental pressure distribution has been obtained by special film, capable of changing color intensity with the pressure variation, and a dedicated software relating the color intensity to the pressure level. A dedicated workbench has been set-up and a special spline shaft has been created in order to well fit the pressure measurements. The pressure distribution along the teeth of spline couplings has been analyzed, above all from the experimental point of view. In particular, a new experimental technique presenting a good potentiality has been set-up to make quick and easy tests. Experimental results have been compared to the corresponding obtained by means of both analytical and FEM models. Good agreement has been showed between numerical and experimental trends in both low and high contact pressure zones. Also a good agreement has been observed with respect to the analytical results, once a fitting procedure has been done referring to the model coefficients related to the tooth deformation. Finally, it may be concluded that the experimental setup utilized in this research allows us to quantitatively measure the contact pressure in spline couplings and also gives us the possibility of correctly calibrate both analytical and numerical models utilized to that aim.

In [28], this study presents FEM analysis of universal coupling with the help of ANSYS for different torque or load condition and verify it by manual calculation. Fracture analysis of a universal joint yoke and a drive shaft of an automobile power transmission system were carried out. The result focus on the relationship between the manufacturing cost and joint angle performance measures of an automotive universal joint, these results illustrate that an increase in the drivable joint angle requires a corresponding increase in manufacturing cost.

In [29], this study presents a theoretical model and a simulation analysis of flange and bolted joint deformation, stresses. The flange and Nut-Bolts force and contact stiffness factor are considered as parameters which are influencing the joint deformation. The flange joint was modelled and simulated using ANSYS 14 Software. The finite element analysis procedure required in ANSYS simulation is presented as a predefined process to obtain accurate results.

In [30], this study about static & dynamic analysis were made for flange mounted motor which is also known as B5 type motor by using FEA software ANSYS 15 and modeling is done with the help of CATIA V5 R20. The study is for special problem which is observed by Laxmi Hydraulics Pvt. Ltd Solapur, Maharashtra, India. Which has been reported frequent failures of this type of flange. The obtained results were that maximum equivalent stress is 18.708 MPa for static and 0.4307 for dynamic conditions which is safe for the given material by considering the allowable strength (200 MPa) of the C.I. by taking safety factor into account.

In [31], this study was made about the effect of misalignment with flexible flange coupling was studied using the both experimental and simulation. The vibration spectra were obtained through experimental and simulation for various excitation frequencies and found to be in close agreement. The results showed that misalignment can be characterized primarily by second harmonics ie. 2X of shaft running speed for parallel misalignment and 1X of the shaft speed for angular misalignment. In case of aligned system, vibration spectrum showed low amplitude of vibration as compared to misaligned system in both experimental and simulation study. It is also seen that as the excitation frequency increases amplitude of vibration also increases.

In [32], an attempt has been made to integrate commercially available package Autodesk Inventor with Microsoft Excel spreadsheet for creation of modeling and manufacturing drawing. Various product variants of the flange coupling have been executed by parametric designing concept in Inventor. The user can give necessary input data in an Excel spreadsheet. Then using one feature crates in Inventor software the 3D modeling and manufacturing drawings will be generated automatically and efficiently. 3D models for flange type coupling and related dimension database in Microsoft Excel have been prepared. The Excel sheet has been linked with Autodesk Inventor to transfer data and relate to respective features of the part. The User can update the model just by modifying the sheet. The modeling takes comparatively very less time to generate complex part models with respect to generating them individually. The modeling and drawing automation technique was more suitable and more simple than any other techniques. This automation can further be proceeded by exporting models to the analysis or CAM package.

CHAPTER 3

THEORY

3.1 Introduction

Coupling is an important power transmission element. It is used for motion transmission from one rotating shaft to another one. Many coupling types are available and manufactured by companies and workshops. Although of the variety of couplings, it still can be considered under the main categories as explained in chapter one.

In this chapter, couplings main types will be explained briefly, according to some companies and catalogs. Rigid coupling and it's types, flexible coupling and it's types will be explained with mentioning some of it's characteristics.

3.2 Couplings Types

3.2.1 Rigid Coupling:

In the rigid coupling the two common types are the flange coupling and the muff coupling [1].

- **Flange Coupling:**

1- Unprotected Flange: In the unprotected type of flange coupling, as in Figure 3.1, both shafts are connected to the main flange by a counter sunk key, in the other hand the two flanges are coupled together with the use of bolts [1].

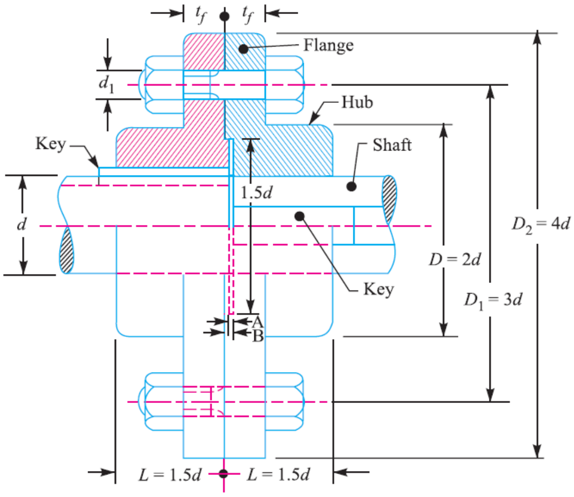


Figure 3.1 Unprotected flange.

2- Protected Flange Coupling: In the protected type of flange coupling, as in Figure 3.2. The eminent bolts and nuts are protected by the flanges on the two halves of the coupling, in order to avoid risk to the workman [1].

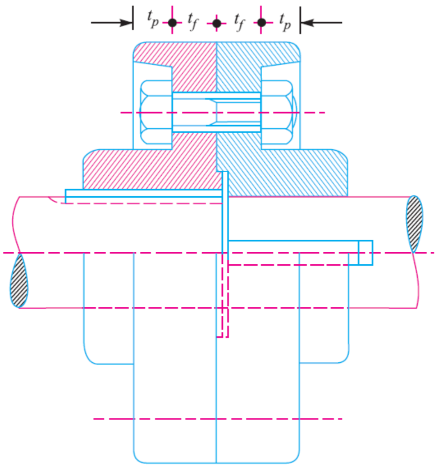


Figure 3.2 Protected flange coupling.

3- Marine Flange: In the marine type of flange coupling, the flanges are formed integrally with the shafts as in Figure 3.3. The flanges are held together by the use of tapered headless bolts. The bolts number starts from 4 to 12 depending on shaft diameter [1].

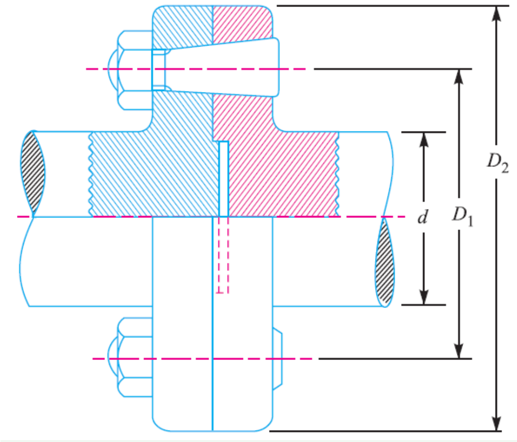


Figure 3.3 Marine flange coupling.

- **Sleeve or Muff-Coupling:**

It is a simple type of rigid coupling, the used material is cast iron. It's a hollow cylinder part with inner diameter same to shaft diameter. It is fitted over the ends of the two shafts with a gib head key, as in Figure 3.4. The power is transmitted from the first shaft to the other shaft by the use of a key and a sleeve [1].

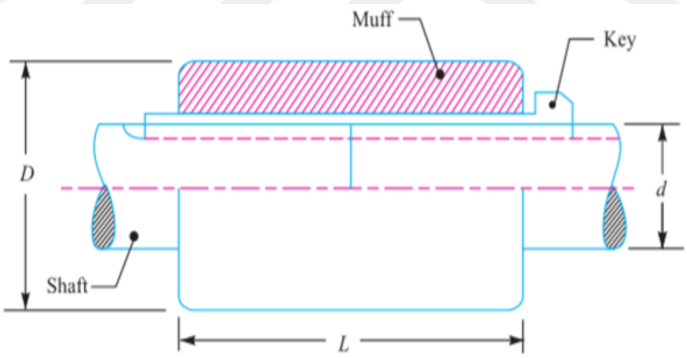


Figure 3.4 Sleeve or Muff-coupling.

3.2.2 Flexible Coupling

For the flexible coupling, the two main categories are the material flexing and the mechanical flexing**.**

Usually for the material flexing couplings, the lubrication is not needed and it can work under shear or compression, these couplings types are able to accept angular, parallel and axial misalignment. The mechanical flexing couplings have the ability to tolerate misalignment from rocking, rolling or sliding of metal surfaces. Lubrication is required for all metal mechanical flexing couplings. The most common types are:

1. Oldham coupling
2. Beam coupling
3. Elastic coupling
4. Constant velocity joint
5. Diaphragm coupling
6. Disc coupling
7. Fluid coupling
8. Gear coupling
9. Grid coupling
10. Rag joint
11. Magnetic coupling
12. Twin Spring coupling
13. Others

- **Pin Bush Coupling**

Pin bush flexible coupling has the ability to absorb shock loads with small misalignments, it can transfer high torque at higher speeds. It consists of two dissimilar halves, one is holding the rubber bushes and the other one holds the steel pins. The number of pins depends on the amount of the power to be transmitted [20]. Pin bush flexible coupling is shown in Figure 3.5.

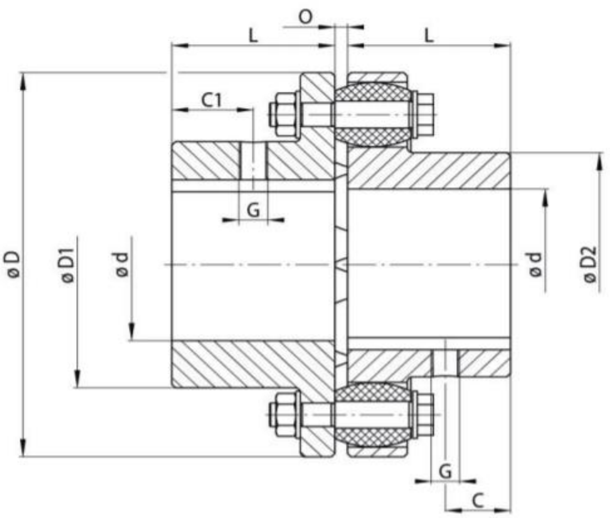


Figure 3.5 Pin bush flexible coupling. [12]

- **Jaw Couplings**

The jaw coupling can be considered as a material flexing type, that transmits torque by compression of the elastomeric spider placed between the two intermeshing jaws. Resilient element material commonly made of polyurethane, NBR, hytrel or bronze. Accommodates misalignment, transmits torque, used for torsional dampening (vibration), low torque and general purpose applications [5]. Jaw coupling is shown in Figure 3.6.

Figure 3.6 Jaw couplings.

- **Tire Coupling**

These couplings consist of a polyurethane or rubber element connected to two hubs. The rubber element transmits torque in shear. Some of the tire coupling characteristics are that it can reduce transmission of shock loads or vibration and it has high misalignment capacity [5]. Tire Coupling is shown in Figure 3.7.

Figure 3.7 Tire coupling.

- **Disc Coupling**

The disc coupling can transmit torque through flexing disc elements. It operates through tension and compression of chorded segments on a common bolt circle, bolted alternately between the drive and driven side. These couplings are typically consisting of two discs packs, two hubs and a center member. A single disc pack can accommodate angular and axial misalignment. Two disc packs are needed to tolerate parallel misalignment [5]. Disc Coupling is shown in Figure 3.8.

Figure 3.8 Disc coupling.

- **Gear Coupling**

Gear coupling has the ability to transmit the highest amount of torque in the smallest diameter of any flexible coupling. Gear coupling consists of two hubs with crowned external gear teeth. The hubs mesh with two internally splined flanged sleeves that are bolted together. Gear coupling has the ability to tolerate angular and axial misalignment by the rocking and sliding of the crowned gear teeth against the mating sleeve teeth. Parallel misalignment is accommodated by having two adjacent hub/sleeve flex points. Periodic lubrication is required depending on the application [5]. Gear Coupling is shown in Figure 3.9.

Figure 3.9 Gear coupling.

- **Grid Coupling**

Grid couplings consist of two radials slotted hubs that mesh with a serpentine strip of spring steel. The grid provides torsional damping and flexibility of an elastomer but the strength of steel. Grid couplings transmit torque and accommodate angular, parallel and axial misalignment from one hub to the other through the rocking and sliding of a tapered grid in the mating hub slots [5]. Grid coupling is shown in Figure 3.10.

Figure 3.10 Grid coupling.

- **Roller Chain**

Roller chain type couplings consist of two radially sprocketed hubs that engage a strand of double pitch roller chain. Chain couplings are used for low to moderate torque and speed applications. The meshing of the sprocket teeth and chain transmits torque and the associated clearances accommodate angular, parallel and axial misalignment. Periodic lubrication is required depending on the application [5]. Roller Chain is shown in Figure 3.11.

Figure 3.11 Roller Chain.

- **Hooke’s Coupling**

Universal coupling, universal joint, Hardy-Spicer joint, U-joint, Hooke's joint or Cardan joint, these are the other names for Hooke’s couplings. Hooke’s couplings are used in order to connect two shafts has intersected axes at a small angle. The inclination of the two shafts may be constant. Motion is varying when it is transmitted from one shaft to another [1]. Universal coupling is shown in Figure 3.12.

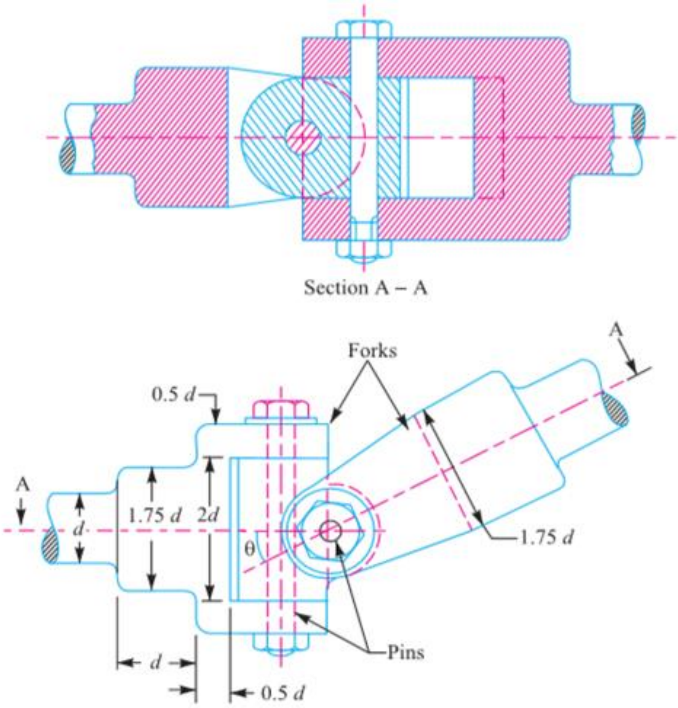


Figure 3.12 Universal coupling.

CHAPTER 4

DESIGN

4.1 Introduction

As in any design, there are stages of development. The initial design was simple, bigger in size with some extra machining required. The development stages are shown in Figure 4.1 and 4.2.

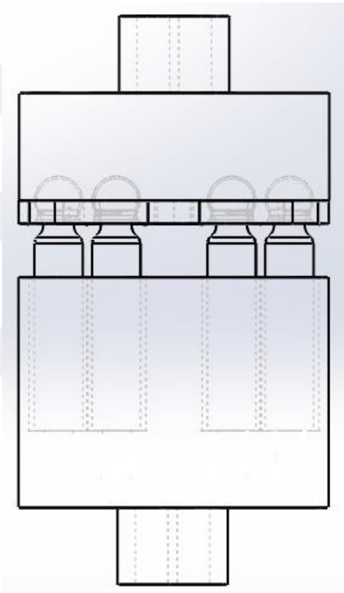


Figure 4.1 Initial design.

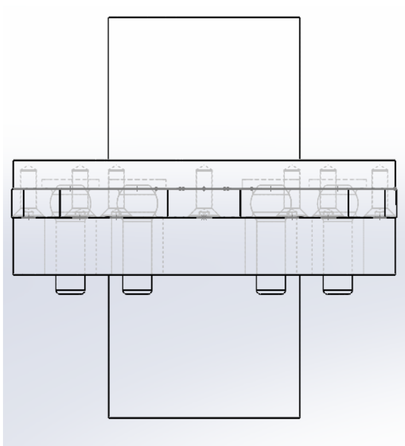


Figure 4.2 Final design.

The theoretical design was done with the use of the equations in the textbook [1]. The applied torque for coupling will be 1662 Nm. Stainless steel was selected as material for shafts. From engineering data tables in ANSYS, the yield strength of the stainless steel is 207 Mpa [13]. Using a safety factor as 1.5, the allowable shear strength will be near to 80 MPa.

Grey cast iron ASTM grade 20 or 150 (ISO 150, EN-JL 1020, F11401) was selected as material for the flanges with shear strength of 180 MPa [14]. This type of material will be used only for the flanges, because the flange is not a critical part, in the case of failure, other parts have to fail first.

Table 4.1 Material names of each part

Part name	Material name
Flange	Gray cast iron
Shaft	Stainless steel
Bush	Polyethylene
Key, Bolts	Stainless steel

4.2 Modified Flange Coupling Calculation

Number of bolts for each group is 6.
Number of pins is 6.

4.2.1 Shaft Design

The shaft diameter is calculated according to shear stress as in Eq. 4.1.

$$T = \frac{\pi}{16} \times \tau_s \times d^3 \tag{4.1}$$

$$1662 \times 10^3 = \frac{\pi}{16} \times 80 \times d^3 \quad d = 47.30 \Rightarrow 50mm. \text{ Standard}$$

4.2.2 Pin Design

Considering that the pin and the shaft are made from the same material. Pin's diameter can be found as in Eq. 4.2. Since the pins and shaft can be made from different material, the pin's diameter must be checked according to shear stress as given in Eq. 4.7 and the higher value can be considered.

$$d_{pc} = \frac{0.5d}{\sqrt{n}} \tag{4.2}$$

$$d_{pc} = \frac{0.5 \times 50}{\sqrt{6}} = 10.20 \Rightarrow 15mm.$$

The pin diameter with the flexible material, considering that the flexible material thickness is 6 mm as in Eq. 4.3.

$$d_p = d_{pc} + (6 \times 2)_{\text{flexible material } .thickness} \tag{4.3}$$

$$d_p = 15 + (6 \times 2)_{\text{flexible material } .thickness} = 27mm.$$

Pin pitch diameter is calculated as in Eq. 4.4.

$$P_D = 100 + 27 + 6 \times 2 = 139 \Rightarrow 140mm. \tag{4.4}$$

$$\tau = 80MPa. \;\; d(pitch) = 140mm.$$

The total load can be obtained as in Eq. 4.5.

$$p_{total} = \frac{T}{r(pitch)} \rightarrow p_{total} = \frac{1662000}{70} = 23742.85N. \tag{4.5}$$

The load on each pin can be found by dividing the total load over the number of pins as in Eq. 4.6.

$$p = \frac{p_{total}}{n} \rightarrow p = \frac{23742.85}{6} = 3957.14N. \quad (4.6)$$

The diameter of pins can be found directly from the direct shear stress as in Eq. 4.7.

$$\tau = \frac{p}{\frac{\pi}{4} * d^2} \rightarrow 80 = \frac{3957.14}{\frac{\pi}{4} * d^2} = 7.93mm. \quad (4.7)$$

4.2.3 Hub Design

The outer diameter and the length of the hub are calculated according to Eq. 4.8 and 4.9 respectively.

$$D = 2d = 100mm. \quad (4.8)$$

$$L = 1.5d = 75mm. \quad (4.9)$$

- **Hub Analysis**

The shear stress in hub is calculated, considering the hub as hollow shaft according to Eq. 4.10.

$d = 50mm.\ \ D = 100mm.\ \ l = 75mm.\ \ G_{cast.iron} = 69Gpa.$

$$\tau = \frac{Tr}{J} \quad (4.10)$$

$$J = \frac{\pi}{32}(D^4 - d^4) \quad (4.11)$$

$$\tau = \frac{T * \frac{D}{2}}{\frac{\pi}{32}(D^4 - d^4)} \rightarrow \tau = \frac{16 * T * D}{\pi(D^4 - d^4)}$$

$$\tau = \frac{16 * 1662000 * 100}{\pi(100^4 - 50^4)} = 9.03MPa.$$

- **Angular Deflection**

Angular deflection in the hub is calculated by the use of Eq. 4.12.

$$\alpha = \frac{32 * l * T}{G * \pi * (D^4 - d^4)} \quad (4.12)$$

$$\alpha = \frac{32*0.075*1662}{69*10^9*\pi*(0.1^4-0.05^4)} = 1.96*10^{-4}\,rad. \rightarrow 0.01\deg.$$

4.2.4 Key Selection

The standard key selected according to the shaft diameter as given in [1]. Width, w is 16 mm and height, t is10 mm.

4.2.5 Bolt Design

For the bolt design, two groups of bolts will be designed for the new coupling. Each group will be designed to carry the total load by itself. The pitch diameter of the first group is calculated as in Eq. 4.13.

$$D_{pb1} = 3d = 3\times 50 = 150mm. \tag{4.13}$$

The bolt diameter of the first bolts group is calculated as in Eq. 4.14 according to shear stress.

$$T = \frac{\pi}{4}\times\tau_s\times Dbolt_1^{\,2}\times n\times\frac{D_{pb1}}{2} \tag{4.14}$$

$$1662\times10^3 = \frac{\pi}{4}\times 80\times Dbolt_1^{\,2}\times 6\times\frac{150}{2} = 7.66mm. \Rightarrow 8mm.$$

- **The second group design**

Assuming pin pitch diameter is 182 mm. Bolt diameter according to shear stress is calculated as in Eq. 4.15.

$$T = \frac{\pi}{4}\times\tau_s\times Dbolt_2^{\,2}\times n\times\frac{D_{pb2}}{2} \tag{4.15}$$

$$1662\times10^3 = \frac{\pi}{4}\times 80\times Dbolt_2^{\,2}\times 6\times\frac{182}{2} = 6.96mm. \Rightarrow 8mm.$$

4.2.6 Flange design

The outside diameter of the flange is calculated as in Eq. 4.16.

$$D_f = 4d = 200mm. \tag{4.16}$$

The thickness of the flange is calculated as in Eq. 4.17.

$$t_f = 0.5d = 25mm. \approx 30mm. \tag{4.17}$$

4.3 Pin Stress Analysis

$d = 15mm.$ $d(pitch) = 140mm.$

Total load can be obtained from the division of the torque over pitch radius as in Eq. 4.18.

$$p_{total} = \frac{T}{r(pitch)} \rightarrow p_{total} = \frac{1662000}{70} = 23742.80N. \tag{4.18}$$

The load on each pin can be obtained from the division of the total load over the number of pins as in Eq. 4.19.

$$p = \frac{p_{total}}{n} \rightarrow p = \frac{23742.8}{6} = 3957.14N. \tag{4.19}$$

Shear stress can be obtained from the direct shear stress formula as in Eq. 4.20.

$$\tau = \frac{p}{\frac{\pi}{4} * d^2} \rightarrow \tau = \frac{3957.14}{\frac{\pi}{4} * 15^2} = 22.40MPa. \tag{4.20}$$

4.4 CAD Modeling

CAD modeling of the modified coupling is done with the use of Solidworks, according to the obtained dimensions from the previous calculations as described in Table 4.2.

Table 4.2 Design specification of modified coupling

Part name	Dimension (mm)
Shaft diameter	50
Hub outside diameter	100
Hub length	75
Key width	16
Key thickness	10
First group Bolt diameter	8
Second group Bolt diameter	8
First group Bolts pitch diameter	150
Second group Bolts pitch diameter	182
Pin diameter	15
Pin pitch diameter	140
Rubber thickness	6
Flange thickness	30
Flange outside diameter	200

4.4.1 2D Drawings

2D drawing is shown in the following figure. Figure 4.3 shows the technical drawing of the new coupling.

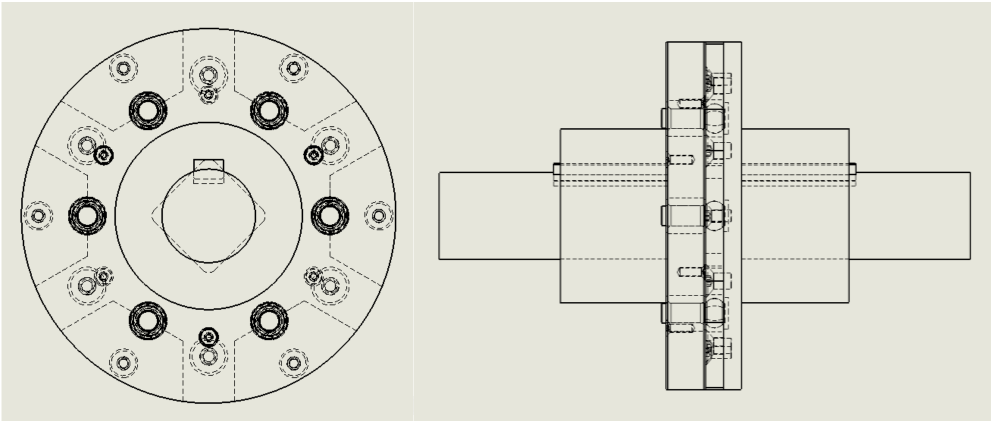


Figure 4.3 Technical drawing of the new coupling.

CHAPTER 5

ANSYS RESULTS

5.1 Introduction

In this section, a 3D CAD model of the new coupling simulated and analysis are carried out using ANSYS 16.2 software. The coupling will be submitted under torque equals to 1662 Nm. The following part consists of complete detail of implemented work. Figure 5.1 shows ANSYS CAD model.

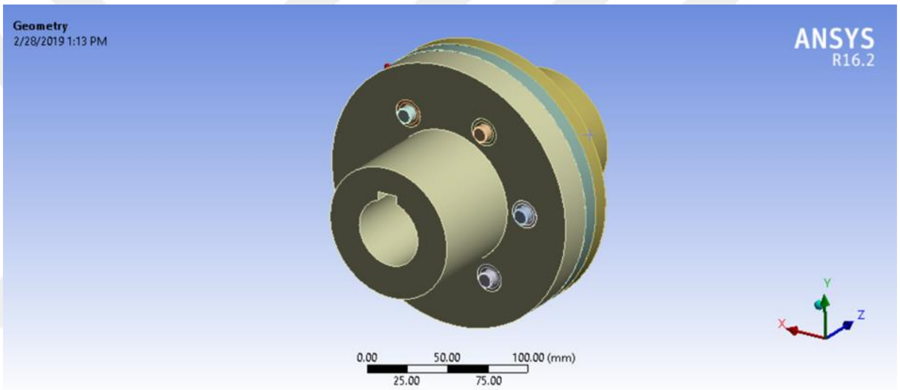


Figure 5.1 ANSYS CAD model.

5.2 Meshing

The mesh statistics shown in Table 5.1. Figure 5.2 shows the meshed model.

Table 5.1 Mesh statistics of model

Statistics	
Nodes	369909
Elements	186686
Mesh Metric	None

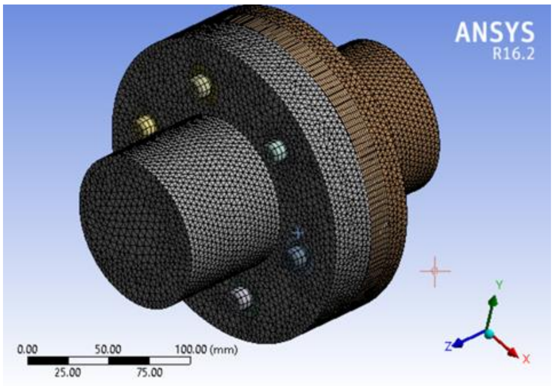


Figure 5.2 Meshed model.

5.3 Contacts Type

For the ANSYS contact definition, the used contact was the bonded contact as shown in Table 5.2. The used contact definition with the same behavior were made for all the parts in this study also the same one that will be used in chapter 7 for the standard flange coupling. The contacts were automatically generated by the software itself. All the contacts were bonded and the contact between the two flanges in both designs was suspended in order to focus on the critical part as shown in Figure 5.3.

Table 5.2 ANSYS contact definition

Definition	
Type	Bonded
Scope Mode	Automatic
Behavior	Program Controlled
Trim Contact	Program Controlled
Trim Tolerance	0.88256 mm
Suppressed	No

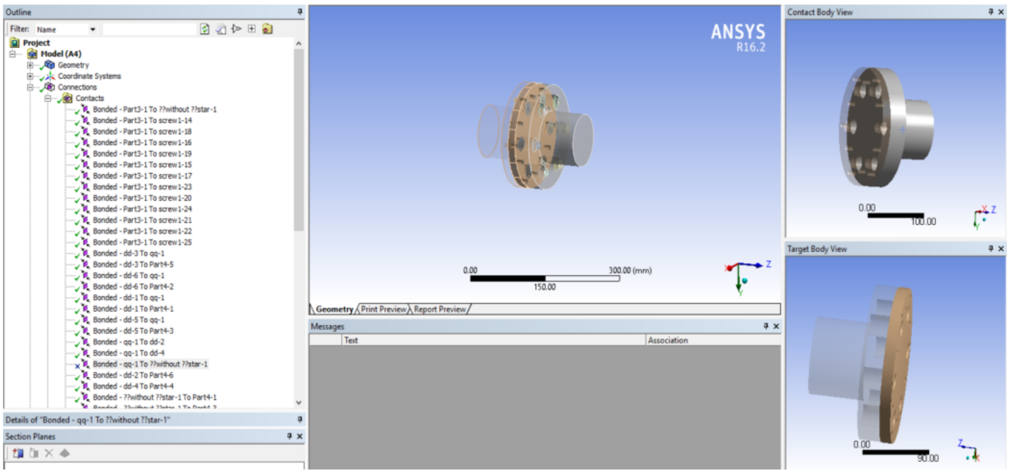


Figure 5.3 The suspended contact.

5.4 ANSYS Results (3MA, Three Misalignment)

5.4.1 Shear Stress

The maximum shear stress will occur in the pin with a value of 21.89 MPa. Shear stress results are shown in Figure 5.4. The maximum shear stress position is shown in Figure 5.5. Table 5.3 shows the minimum and maximum shear stress values.

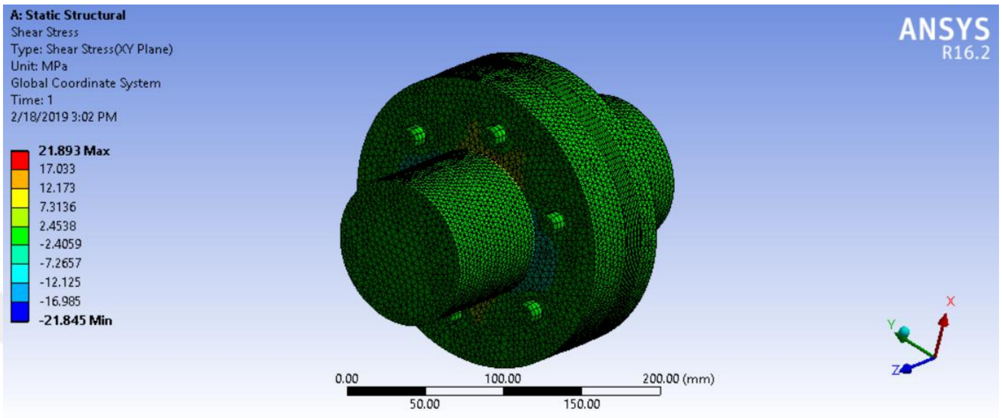


Figure 5.4 Shear stress results.

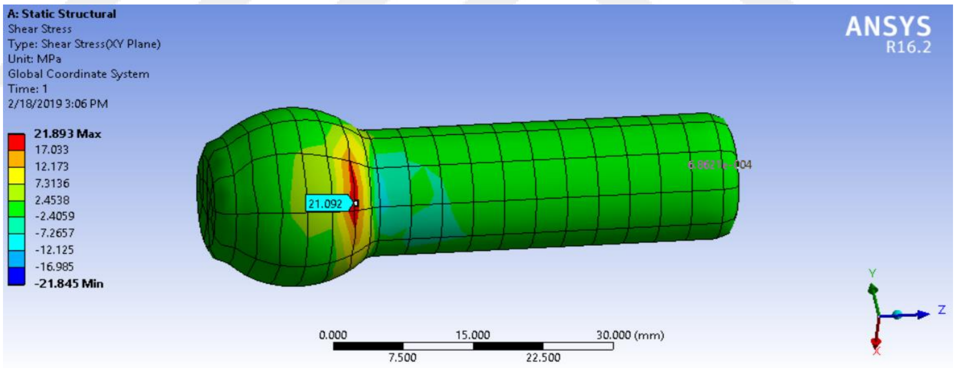


Figure 5.5 Maximum shear stress position.

Table 5.3 Shear stress values

	Time [s]	Minimum [MPa]	Maximum [MPa]
Numerical	1.0	-21.845	21.893
Theoretical	--	--	22.40

5.4.2 Equivalent Stress

The maximum equivalent stress will occur in the pin with a value of 145.96 MPa. Equivalent stress results are shown in Figure 5.6. The maximum equivalent stress position is shown in Figure 5.7 Table 5.4 shows the minimum and maximum equivalent stress values.

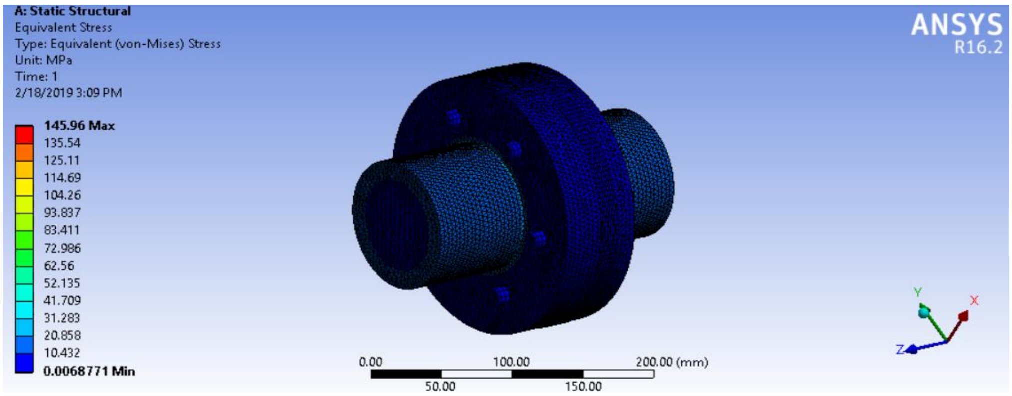


Figure 5.6 Equivalent stress results.

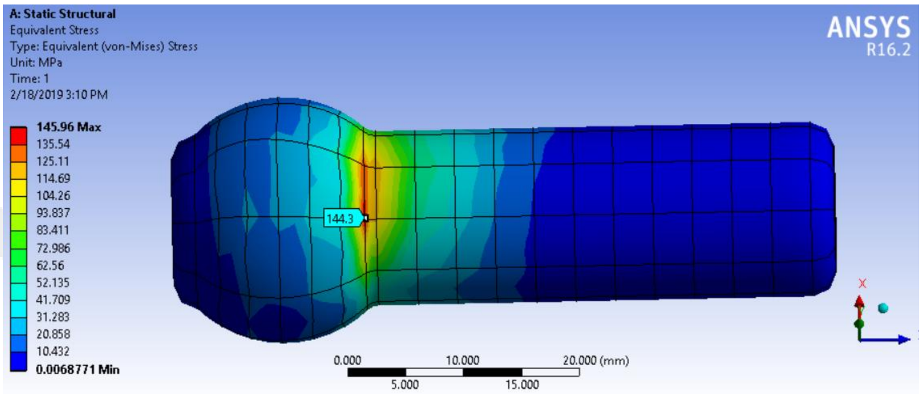


Figure 5.7 Maximum equivalent stress position.

Table 5.4 Minimum and maximum equivalent stress values

Time [s]	Minimum [MPa]	Maximum [MPa]
1.0	6.8771e-003	145.96

5.5 Modified Coupling (2MA, Two Misalignment)

5.5.1 Equivalent Stress

The modified coupling has three misalignments, axial misalignment from the pin sliding. The angular misalignment from combining the sliding and rounded pin's head. The parallel misalignment from the presence resilient material.

Canceling the presence resilient material and replace it by gray cast iron, only two misalignments will be obtained. The equivalent stress that may develop in the critical part (pins) will be reduced with keeping the shear stress with very small difference. The maximum equivalent stress will occur in the pin with a value of 74.124 MPa. The equivalent stress results are shown in Figure 5.8. Table 5.5 shows the minimum and maximum equivalent stress values.

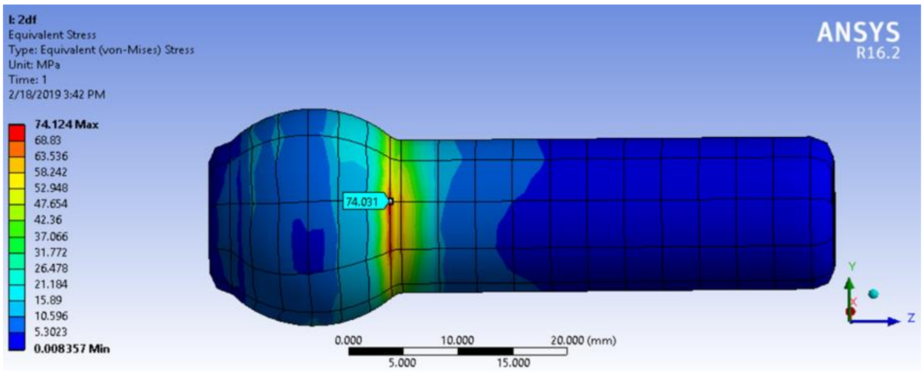


Figure 5.8 Maximum equivalent stress position (2MA).

Table 5.5 Minimum and maximum equivalent stress values

Time [s]	Minimum [MPa]	Maximum [MPa]
1.0	8.357e-003	74.124

5.5.2 Shear Stress

The maximum shear stress will occur in the pin with a value of 21.78 MPa. Shear stress results are shown in Figure 5.9. Table 5.6 shows the minimum and maximum shear stress values.

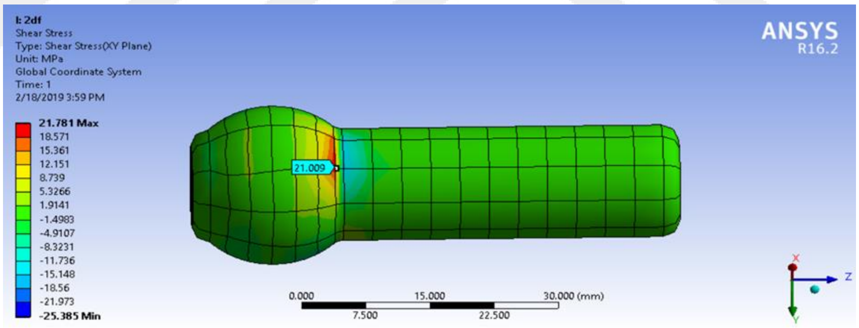


Figure 5.9 Maximum shear stress position (2MA).

Table 5.6 Shear stress values

Time [s]	Minimum [MPa]	Maximum [MPa]
1.0	-25.385	21.781

5.5.3 Total Deformation

The maximum total deformation is 0.073454 mm. Total deformation results are shown in Figure 5.10. Table 5.7 shows the minimum and maximum total deformation values. The physical properties of the design with the applied load and boundary condition are shown in Table 5.8.

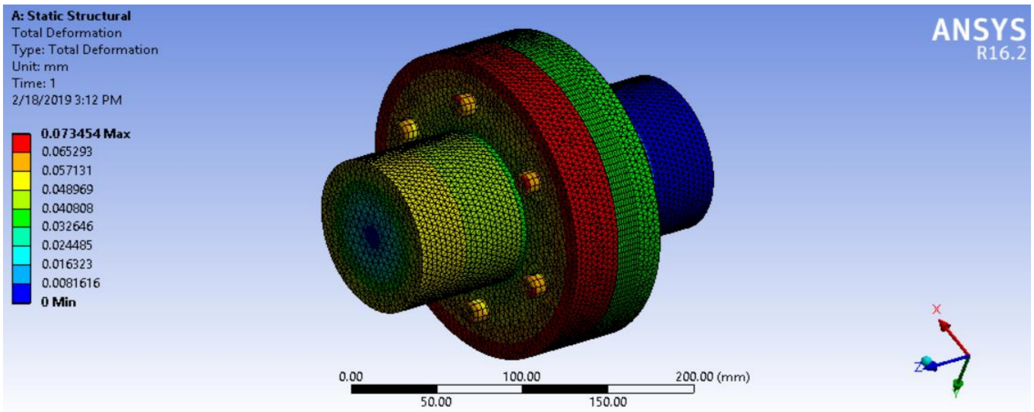


Figure 5.10 Total deformation results.

Table 5.7 Total deformation values

Time [s]	Minimum [mm]	Maximum [mm]
1.0	0.0	7.3454e-002

Table 5.8 Physical properties, applied load and boundary condition

<table>
<tr><td colspan="2">Unit System</td><td colspan="2">Metric (mm, kg, N, s, mV, mA) Degrees rad/s Celsius</td></tr>
<tr><td colspan="2">Angle</td><td colspan="2">Degrees</td></tr>
<tr><td colspan="2">Rotational Velocity</td><td colspan="2">rad/s</td></tr>
<tr><td colspan="2">Temperature</td><td colspan="2">Celsius</td></tr>
<tr><th colspan="4">Properties</th></tr>
<tr><td colspan="2">Volume</td><td colspan="2">3.0539e+006 mm³</td></tr>
<tr><td colspan="2">Mass</td><td colspan="2">21.591 kg</td></tr>
<tr><td colspan="2">Scale Factor Value</td><td colspan="2">1.</td></tr>
<tr><th colspan="4">Statistics</th></tr>
<tr><td colspan="2">Bodies</td><td colspan="2">27</td></tr>
<tr><td colspan="2">Active Bodies</td><td colspan="2">27</td></tr>
<tr><td colspan="2">Nodes</td><td colspan="2">369909</td></tr>
<tr><td colspan="2">Elements</td><td colspan="2">186686</td></tr>
<tr><td colspan="2">Mesh Metric</td><td colspan="2">None</td></tr>
<tr><th colspan="4">Definition</th></tr>
<tr><td>Type</td><td colspan="2">Moment</td><td>Fixed Support</td></tr>
<tr><td>Define By</td><td colspan="2">Vector</td><td></td></tr>
<tr><td>Magnitude</td><td colspan="2">1.662e+006 N·mm (ramped)</td><td></td></tr>
<tr><td>Direction</td><td colspan="2">Defined</td><td></td></tr>
<tr><td>Suppressed</td><td colspan="3">No</td></tr>
<tr><td>Behavior</td><td colspan="2">Deformable</td><td></td></tr>
</table>

CHAPTER 6

MANUFACTURING

6.1 Introduction

In this chapter manufacturing of the new designed coupling will be explained. Small prototype with 130 Nm torque capacity and maximum outside diameter is 50 mm. The used material was AISI 4130 alloy steel.

6.2 Technical Drawings

6.2.1 Middle Holder

The technical drawing of the middle holder is shown in Figure 6.1.

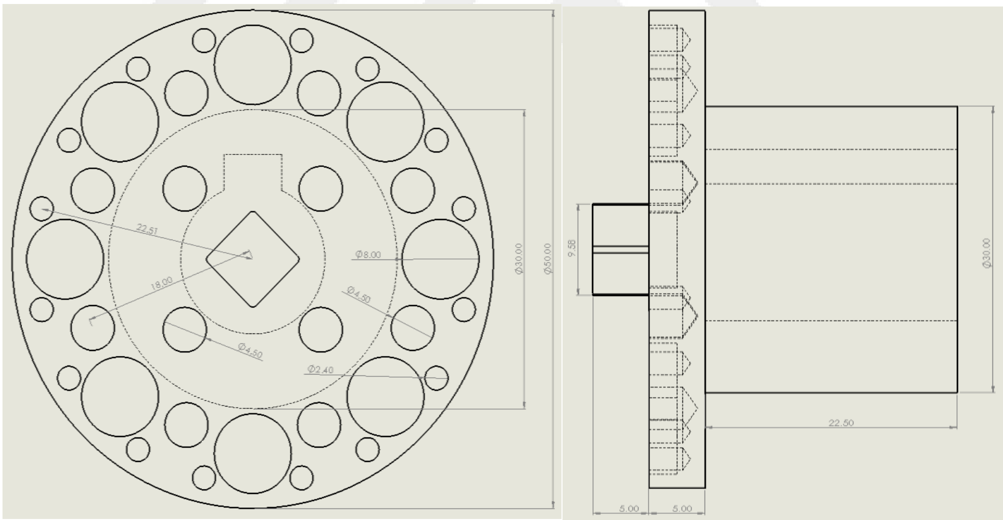


Figure 6.1 Technical drawing of middle holder.

6.2.2 Major Middle Part

The technical drawing of the major middle part is shown in Figure 6.2.

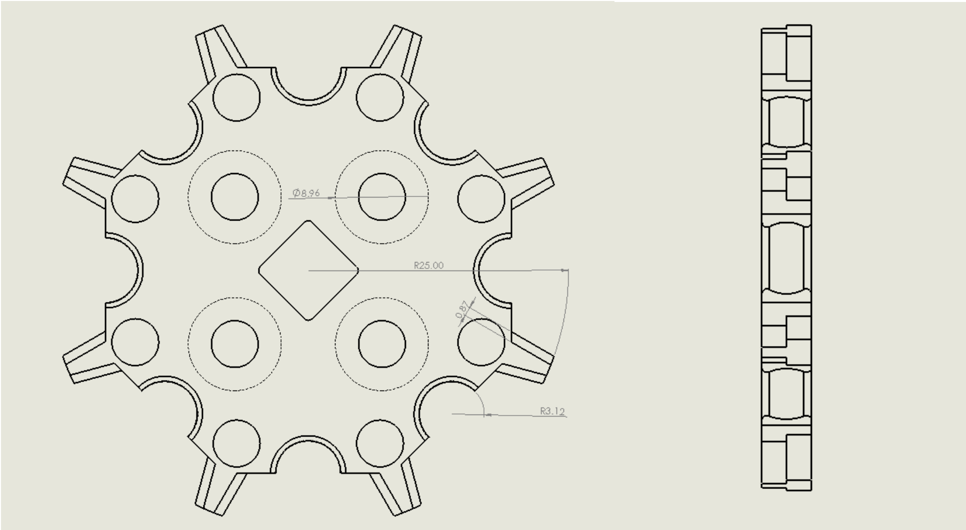


Figure 6.2 Technical drawing of middle part.

6.2.3 Secondary Middle Part

The technical drawing of the secondary middle part are shown in Figure 6.3.

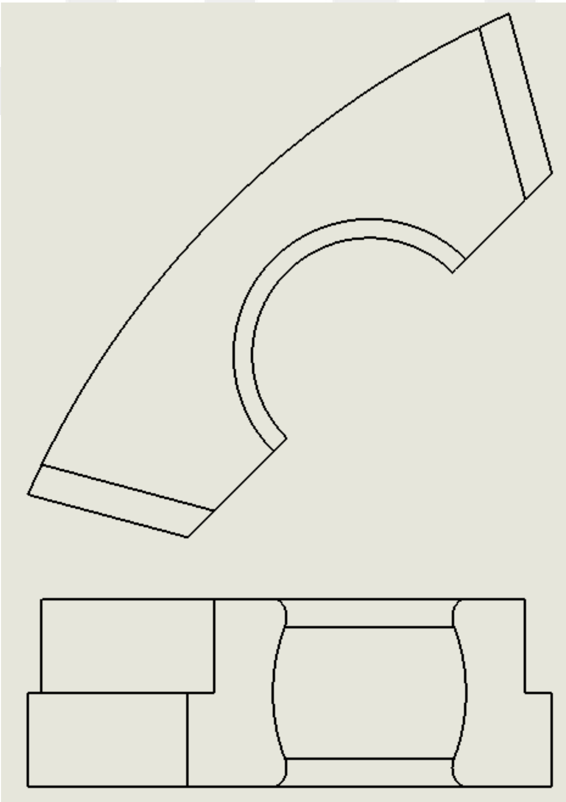


Figure 6.3 Technical drawing of secondary middle part.

6.2.4 Pin

The technical drawing of the pin is shown in Figure 6.4.

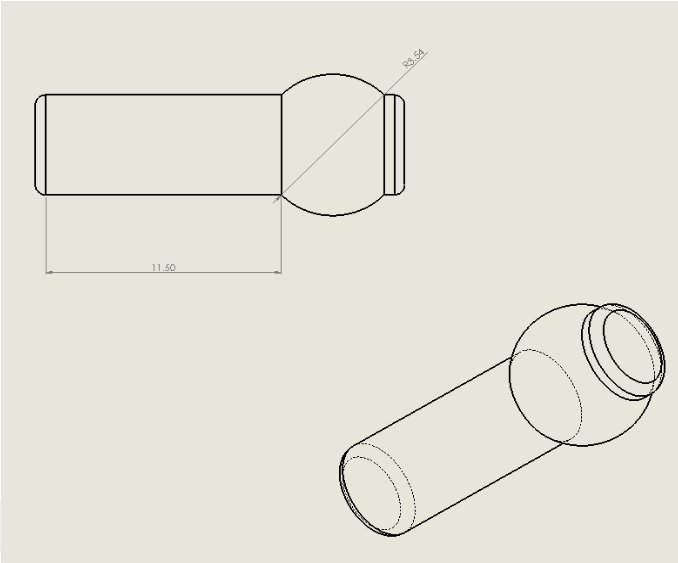


Figure 6.4 Technical drawing of the pin.

6.2.5 Plate

The technical drawing of the plate is shown in Figure 6.5. Plate thickness is 1 mm.

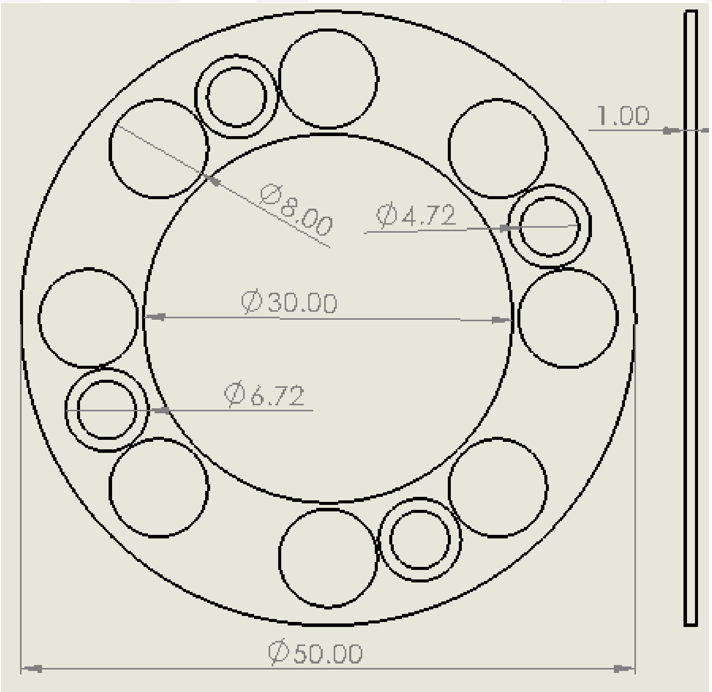


Figure 6.5 Technical drawing of the plate.

6.2.6 Pin Holder

The technical drawing of the pin holder is shown in Figure 6.6.

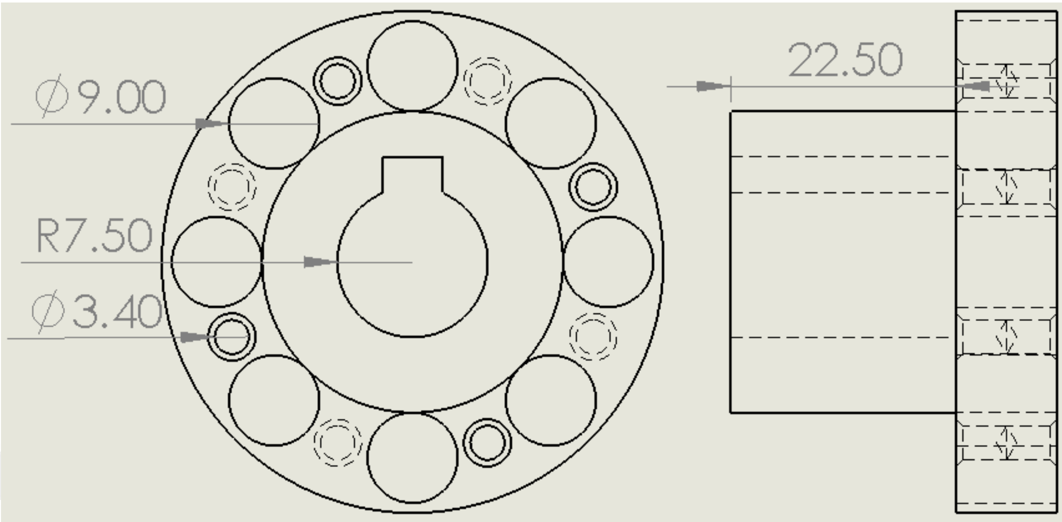


Figure 6.6 Technical drawing of the pin holder.

6.2.7 Polyurethane

The selected material for this design was the polyurethane. The technical drawing of the polyurethane is shown in Figure 6.7.

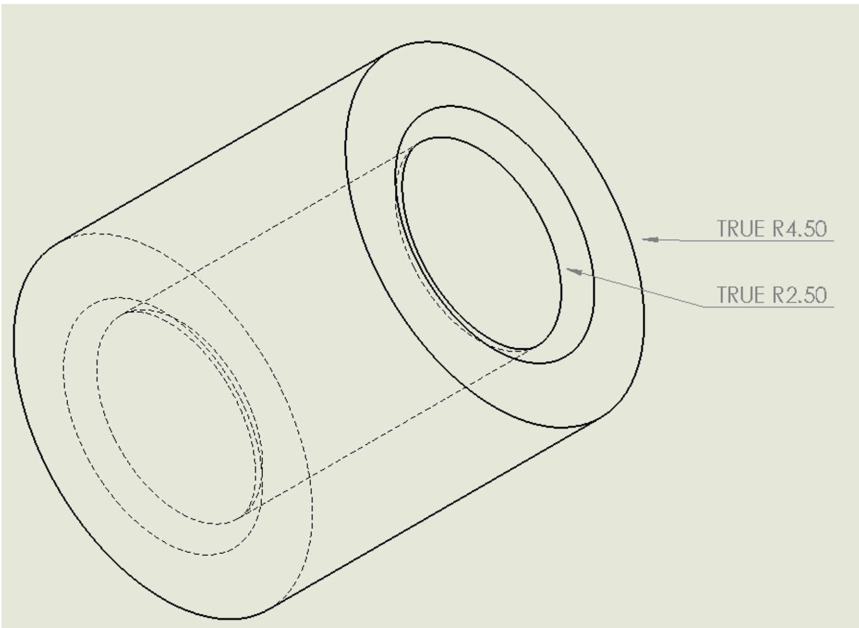


Figure 6.7 Technical drawing of the polyurethane.

6.3 Manufacturing Process

6.3.1 Middle Holder

By the use of the conventional lathe and milling machine the bulk was obtained as in Figure 6.8. Using the CNC milling as in Figure 6.9. The final shape was obtained as in Figure 6.10.

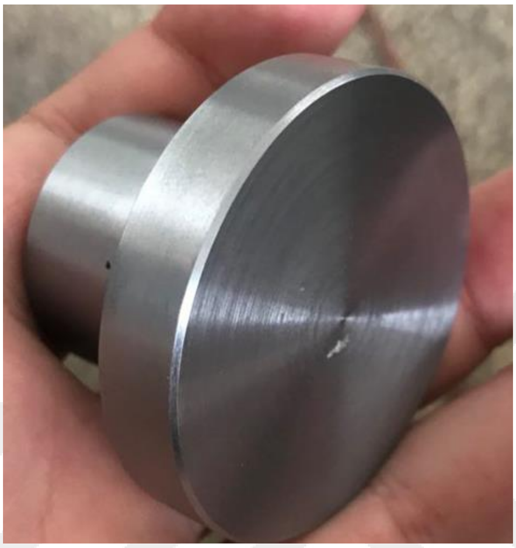


Figure 6.8 Middle holder bulk.

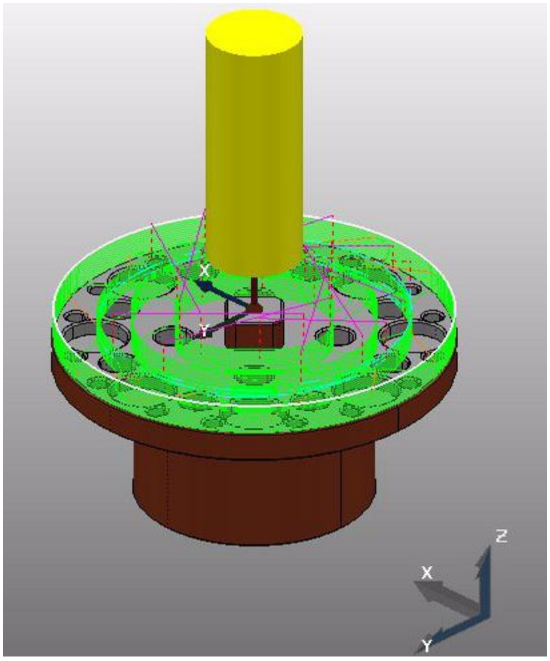


Figure 6.9 CNC milling proccess for middle holder.

Figure 6.10 Final shape for middle holder.

6.3.2 Major Middle Part

For this part, the bulk was obtained with the use of hole EDM and wire cut EDM as in Figure 6.11. For the rounded surfaces an EDM machine was used as in Figure 6.12 with the tool as in Figure 6.13. For the finned part of major middle part, CNC machine was used as in Figure 6.14. The final shape is shown in Figure 6.15.

Figure 6.11 Hole EDM.

Figure 6.12 EDM machine.

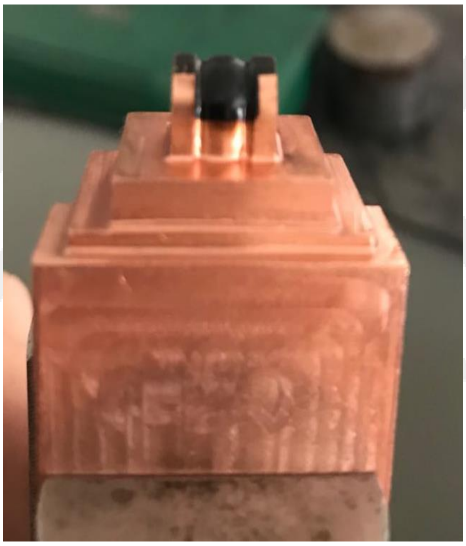


Figure 6.13 EDM tool.

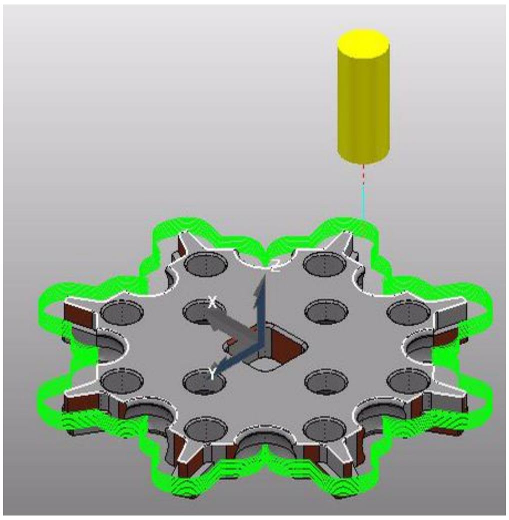


Figure 6.14 CNC machining.

Figure 6.15 Major middle part final shape.

6.3.3 Secondary Middle Part

For this part a CNC milling machine was used as in Figure 6.16. EDM wire cut was used in the purpose of separating the parts from the original bulk as in Figure 6.17. For the rounded surfaces the same EDM machine and tool as in the previous Figure 6.13. The EDM process is shown in Figure 6.18. Figures 6.19 and 6.20 show the final shape.

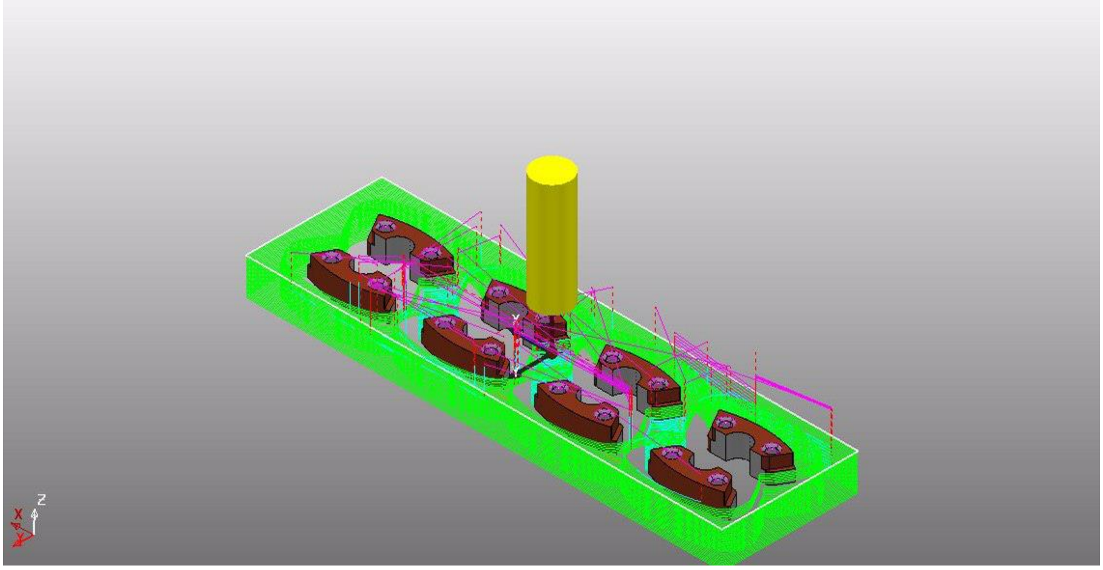


Figure 6.16 CNC milling process for the secondary middle part.

Figure 6.17 Original bulk after the wire cutting process.

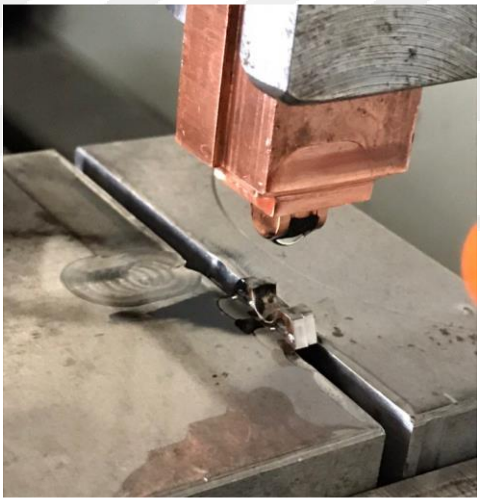


Figure 6.18 The EDM process for the secondary middle part.

Figure 6.19 The final shape of the secondary middle part.

Figure 6.20 The final shape of the secondary middle part focusing on the rounded surface.

6.3.4 Pin

8 pins were manufactured with the use of CNC lathe machine. The final shape is shown in Figure 6.21.

Figure 6.21 Shows the pin in the assembly.

6.3.5 Pin Holder

The bulk was obtained in the same way as in Figure 6.8. For the holes, CNC milling machine was used as in Figure 6.22. The final shape is shown in Figure 6.23.

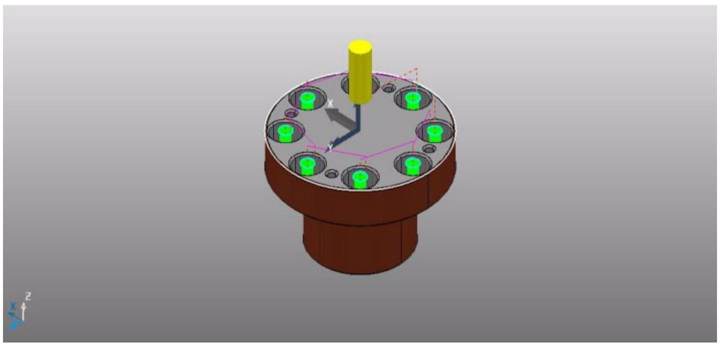


Figure 6.22 CNC milling for Pin holder.

Figure 6.23 The final shape for the pin holder.

6.3.6 Polyurethane

Polyurethane material was manufactured with the use of a mold as shown in Figure 6.24. The final shape is shown in Figure 6.25.

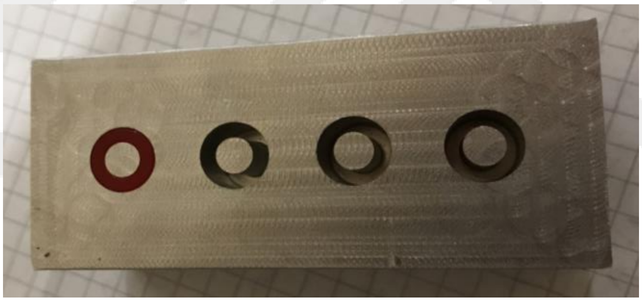


Figure 6.24 Mold for polyurethane casting.

Figure 6.25 Final shape for polyurethane.

CHAPTER 7

COMPARISON

7.1 Introduction

As described before, there are two main types of couplings: rigid coupling and flexible coupling. Rigid coupling usually used for high torque applications with two perfect aligned shafts, the flexible couplings are the type of couplings which can accept some degrees of misalignment. Flange coupling is one of the most common types of rigid couplings, this type of coupling consists of two similar flanges, fixed with a group of (4 or 6) bolts, it is used for high torque applications with no misalignment. In the new design, the same torque capacity with less stress values were gained, besides high misalignment capacity.

A comparison between the new design and standard flange coupling was done, where both of the two coupling types were designed theoretically according to some standard formulas which have been used in several papers and textbooks. The two types of couplings were designed as a CAD model with the use of solid modeling CAD software (SOLIDWORKS 2018), after conducting the 3D model from Solid-works, these two models were imported to (ANSYS 16.2) in order to be checked with the use of finite element method.

The aim is to compare between the stresses which have occurred in the critical parts of each design of the two couplings, also to prove that the stresses in the new design were less than the one in the standard flange coupling. According to misalignment capacity, another comparison was made, between the new design and other couplings from manufactures catalogues was made.

7.2 Design of Couplings

7.2.1 Calculations

The applied torque for both couplings will be 1662 Nm. Stainless steel was selected as material for shafts, keys and bolts. From engineering data tables given in ANSYS 16.2 [13]. The yield strength of the stainless steel is 207 MPa. Using a safety factor as 1.5, the allowable shear strength will be near to 80 MPa.

Gray cast iron ASTM grade 20 or 150 (ISO 150, EN-JL 1020, F11401) was selected as material for the flanges with shear strength of 180 MPa [14]. This type of material will be used only for flanges and because of the flange is not a critical part, in the case of failure, other parts have to fail first.

7.2.1.1 Standard Flange Coupling Calculations

Number of bolts is 6. Bolts are considered as critical parts, the material of each part is shown in Table 7.1.

Table 7.1 Part's material

Part name	Material name
Flange	Gray cast iron
Shaft	Stainless steel
Key, bolts	Stainless steel

- **Shaft Design**

The shaft diameter is calculated according to shear stress as in Eq. 7.1.

$$T = \frac{\pi}{16} \times \tau_s \times d^3 \tag{7.1}$$

$$1662 \times 10^3 = \frac{\pi}{16} \times 80 \times d^3$$

$$d = 47.30 \Rightarrow 50mm. \ \underline{\text{Standard}}$$

- **Hub Design**

The outer diameter and the length of the hub are calculated as in Eq. 7.2 and 7.3 respectively.

$$D = 2d = 100mm. \tag{7.2}$$

$$L = 1.5d = 75mm. \tag{7.3}$$

- **Hub Analysis**

The shear stress in the hub is calculated considering the hub as a hollow shaft according to Eq. 7.4.

$d = 50mm.\ D = 100mm.\ l = 75mm.\ G_{cast.iron} = 69GPa.$

$$\tau = \frac{Tr}{J} \tag{7.4}$$

$$J = \frac{\pi}{32}(D^4 - d^4) \tag{7.5}$$

$$\tau = \frac{T * \frac{D}{2}}{\frac{\pi}{32}(D^4 - d^4)} \rightarrow \tau = \frac{16 * T * D}{\pi(D^4 - d^4)}$$

$$\tau = \frac{16 * 1662000 * 100}{\pi(100^4 - 50^4)} = 9.03MPa.$$

Angular deflection in the hub is calculated by the use of Eq. 7.6.

$$\alpha = \frac{32 * l * T}{G * \pi * (D^4 - d^4)} \tag{7.6}$$

$$\alpha = \frac{32 * 0.075 * 1662}{69 * 10^9 * \pi * (0.1^4 - 0.05^4)} = 1.96 * 10^{-4} rad. \rightarrow 0.01 \deg.$$

- **Key Selection**

The standard key selected according to the shaft diameter [1]. Width, w is 16 mm and height, t is 10 mm.

- **Bolt Design**

Pitch diameter of bolts is calculated as in Eq. 7.7.

$$D_{pb1} = 3d = 3 \times 50 = 150mm. \tag{7.7}$$

Diameter of each bolt is calculated according to shear stress as in Eq. 7.8.

$$T = \frac{\pi}{4} \times \tau_s \times Dbolt_1^{\ 2} \times n \times \frac{D_{pb1}}{2} \tag{7.8}$$

$$1662 \times 10^3 = \frac{\pi}{4} \times 80 \times Dbolt_1^{\ 2} \times 6 \times \frac{150}{2} = 7.66 mm. \Rightarrow 8mm.$$

- **Flange Design**

The outside diameter of the flange is calculated as in Eq. 7.9.

$$D_f = 4d = 200mm. \tag{7.9}$$

The thickness of the flange is calculated as in Eq. 7.10.

$$t_f = 0.5d = 25mm. \tag{7.10}$$

- **Critical Part Stress Analysis**

$$d = 8mm. \quad d(pitch) = 150mm.$$

Total load can be obtained from the division of the torque over pitch radius as in Eq. 7.11.

$$p_{total} = \frac{T}{r(pitch)} \tag{7.11}$$

The load on each bolt can be obtained from the division of the total load over the number of bolts as in Eq. 7.12.

$$p_{total} = \frac{1662000}{75} = 22160N.$$

$$p = \frac{p_{total}}{n} \rightarrow p = \frac{22160}{6} = 3693.30N. \tag{7.12}$$

Shear stress can be obtained from the direct shear stress formula as in Eq. 7.13.

$$\tau = \frac{p}{\frac{\pi}{4} * d^2} \rightarrow \tau = \frac{3693.3}{\frac{\pi}{4} * 64} = 73.50MPa. \text{ For each bolt} \tag{7.13}$$

7.2.2 CAD Modeling

7.2.2.1 CAD Modeling of Standard Flange Coupling

CAD model of standard flange coupling is drawn in Solidworks according to the obtained dimensions from the previous calculations as described in Table 7.2.

Standard flange coupling side view, technical drawing is shown in Figure 7.1. The 3D CAD model is shown in Figure 7.2.

Table 7.2 Design specification of standard flange coupling

Part name	Dimension (mm)
Shaft diameter	50
Hub outside diameter	100
Hub length	75
Key width	16
Key thickness	10
Bolt diameter	8
Bolts pitch diameter	150
Flange thickness	25
Flange outside diameter	200

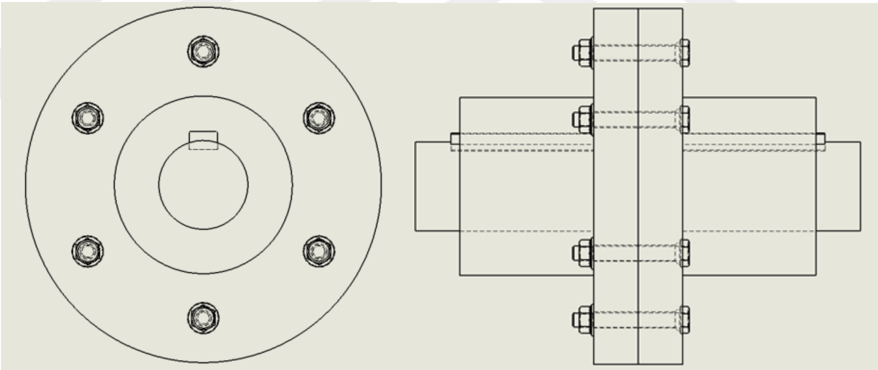


Figure 7.1 Technical drawing of standard coupling.

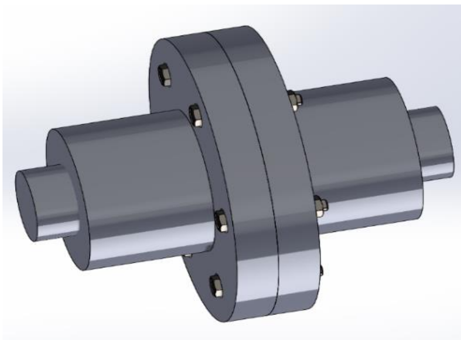


Figure 7.2 3D CAD model of standard coupling.

7.2.2.2 ANSYS CAD Model

The ANSYS CAD model of the standard flange is shown in Figure 7.3.

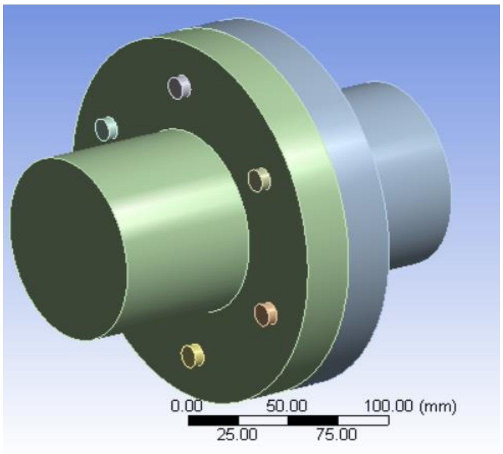


Figure 7.3 Standard flange coupling.

7.2.3 Meshing

7.2.3.1 Standard Flange Coupling

The meshed standard flange coupling model is shown in Figure 7.4 and it’s parts are shown in Figures 7.5 and 7.6.

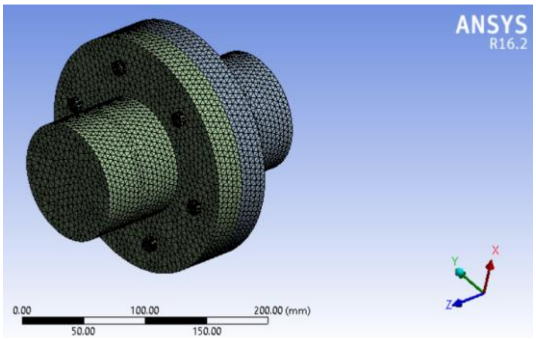


Figure 7.4 Standard flange meshing.

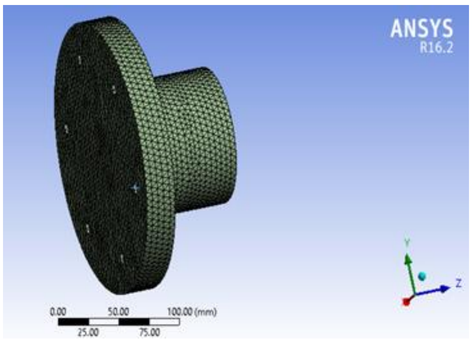


Figure 7.5 Hub and flange meshing.

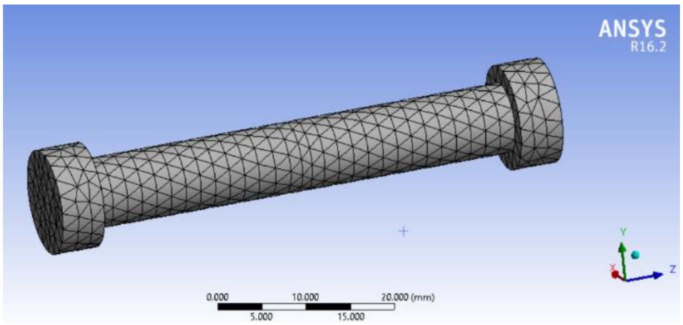


Figure 7.6 Bolts meshing.

a) Standard Flange Coupling

In Table 7.3, the unit system, physical properties, mesh statistics, load and boundary conditions are shown.

Table 7.3 ANSYS inputs for standard flange coupling model

<table>
<tr><td colspan="2">Unit System</td><td>Metric (mm, kg, N, s, mV, mA)
Degrees rad/s Celsius</td></tr>
<tr><td colspan="2">Angle</td><td>Degrees</td></tr>
<tr><td colspan="2">Rotational Velocity</td><td>rad/s</td></tr>
<tr><td colspan="2">Temperature</td><td>Celsius</td></tr>
<tr><th colspan="3">Properties</th></tr>
<tr><td colspan="2">Volume</td><td>2.7557e+006 mm³</td></tr>
<tr><td colspan="2">Mass</td><td>19.853 kg</td></tr>
<tr><td colspan="2">Scale Factor Value</td><td>1.</td></tr>
<tr><th colspan="3">Statistics</th></tr>
<tr><td colspan="2">Bodies</td><td>8</td></tr>
<tr><td colspan="2">Active Bodies</td><td>8</td></tr>
<tr><td colspan="2">Nodes</td><td>157192</td></tr>
<tr><td colspan="2">Elements</td><td>100358</td></tr>
<tr><td colspan="2">Mesh Metric</td><td>None</td></tr>
<tr><th colspan="3">Definition</th></tr>
<tr><td>Type</td><td>Moment</td><td>Fixed Support</td></tr>
<tr><td>Define By</td><td>Vector</td><td></td></tr>
<tr><td>Magnitude</td><td>1.662e+006 N·mm (ramped)</td><td></td></tr>
<tr><td>Direction</td><td>Defined</td><td></td></tr>
<tr><td>Suppressed</td><td colspan="2">No</td></tr>
<tr><td>Behavior</td><td>Deformable</td><td></td></tr>
</table>

7.2.4 Standard Flange Coupling

7.2.4.1 Shear Stress

The maximum shear stress will occur in the bolts with value of 66.4MPa. Shear stress results are shown in Figure 7.7. The maximum shear stress position is shown in Figure 7.8. Table 7.4 shows the minimum and maximum shear stress values.

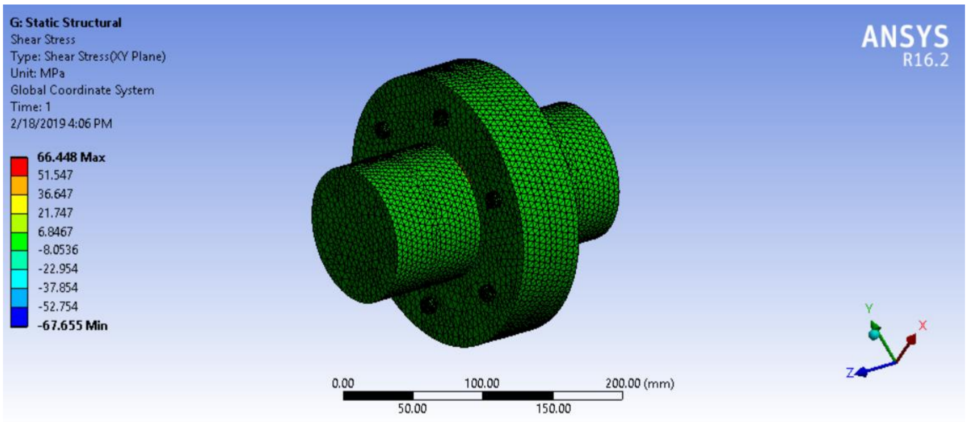


Figure 7.7 Shear stress results for standard flange.

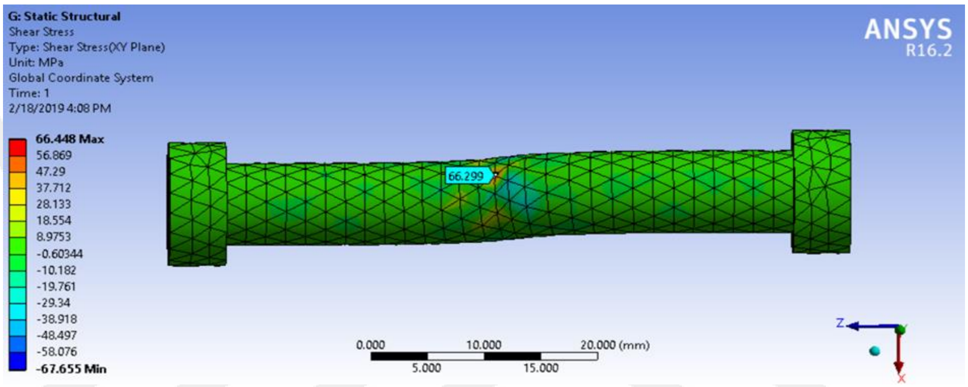


Figure 7.8 Maximum shear stress position.

Table 7.4 Minimum and maximum shear stress values

	Time [s]	Minimum [MPa]	Maximum [MPa]
Numerical	1.0	-67.655	66.448
Theoretical	--	--	73.50

7.2.4.2 Equivalent Stress

The maximum equivalent stress will occur in the bolts with value of 179.11MPa. Equivalent stress results are shown in Figure 7.9. The maximum equivalent stress position is shown in Figure 7.10. Table 7.5 shows the minimum and maximum equivalent stress values.

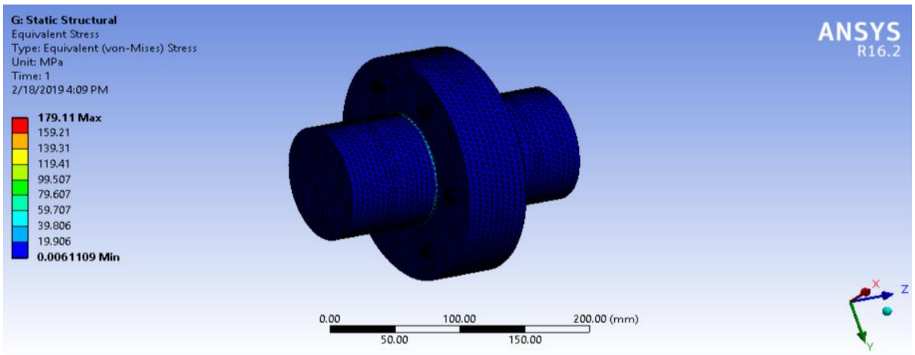


Figure 7.9 Equivalent stress result.

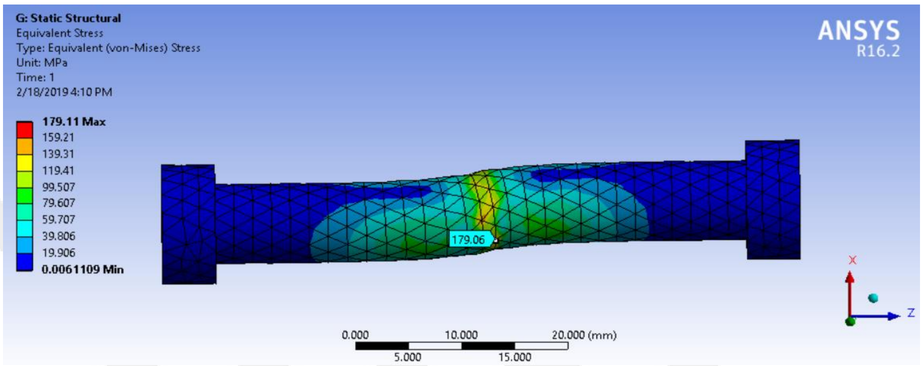


Figure 7.10 Maximum equivalent stress position.

Table 7.5 Minimum and maximum equivalent stress values

Time [s]	Minimum [MPa]	Maximum [MPa]
1.0	6.1109e-003	179.11

7.2.4.3 Total Deformation

The maximum total deformation is 0.064084 mm. Total deformation results are shown in Figure 7.11. Table 7.6 shows the minimum and maximum total deformation values.

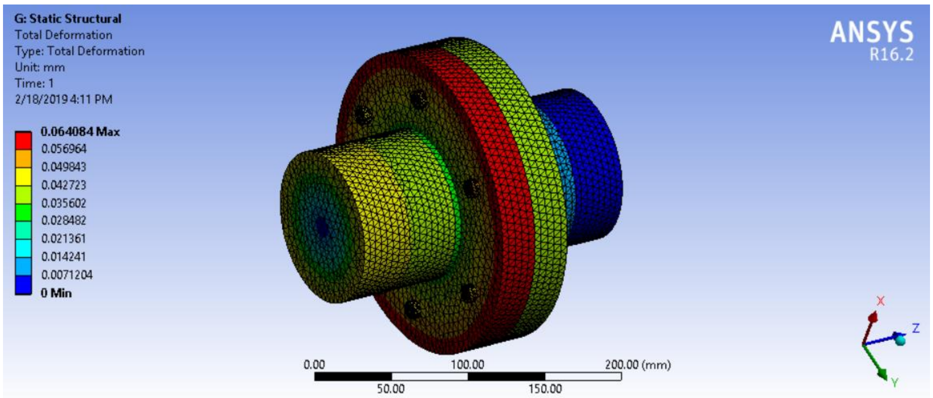


Figure 7.11 Total deformation results.

Table 7.6 Minimum and maximum total deformation values

Time [s]	Minimum [mm]	Maximum [mm]
1.0	0.0	6.4084e-002

7.3 Misalignment Comparison

A comparison was made between the new design and some different flexible couplings from different manufactures and companies' catalogues.

- **Renold Couplings (Resilient and Soft Start Couplings). [15]**

A. Spiderflex

The coupling is shown in Figure 7.12. The technical drawing for the same coupling is shown in Figure 7.13. Table 7.7 shows the comparison with respect to torque capacity, outside diameter and misalignments.

Figure 7.12 Spiderflex coupling.

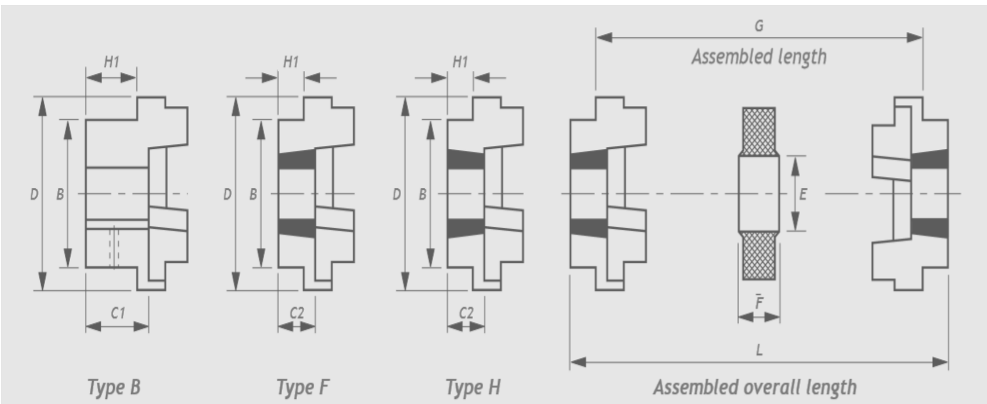


Figure 7.13 Spiderflex technical drawing.

Table 7.7 Comparison between Spiderflex coupling and the new design

Coupling type		Nominal torque Nm	Outside dia. (mm)	Max. Misalignment		End float (mm)
				Offset (mm)	Angular (deg.)	
Spiderflex	RSC90 # # #	80	85	0.3	0.5	0.5
	RSC110 # # #	160	112	0.3	1.0	0.6
New design		130	50	3.0-3.5	5.0	2.0

B. Crownpin

The coupling is shown in Figure 7.14. The technical drawing for the same coupling is shown in Figure 7.15. Table 7.8 shows the comparison with respect to torque capacity, outside diameter and misalignments.

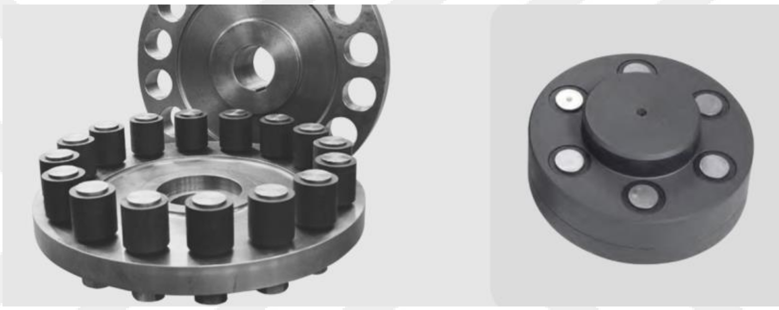


Figure 7.14 Crownpin coupling.

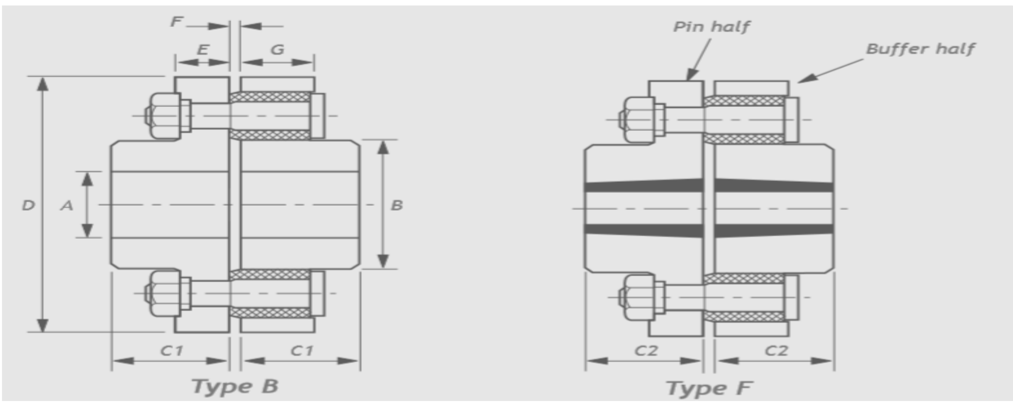


Figure 7.15 Crownpin technical drawing.

Table 7.8 Comparison between Crownpin coupling and the new design

Coupling type		Nominal torque Nm	Outside diameter (mm)	Max. Misalignment		End float (mm)
				Offset (mm)	Angular (deg.)	
Crownpin	CP48BB4	71	112	0.13	0.15	-
	CP48BB8	142	112	0.13	0.15	-
New design		130	50	3.0-3.5	5.0	2.0

C. Tyreflex

The coupling is shown in Figure 7.16. The technical drawing for the same coupling is shown in Figure 7.17. Table 7.9 shows the comparison with respect to torque capacity, outside diameter and misalignments.

Figure 7.16 Tyreflex coupling.

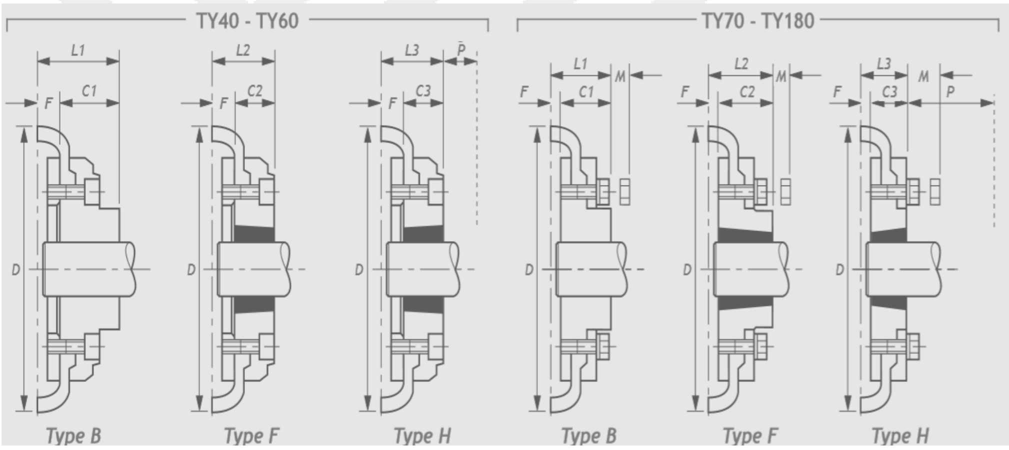


Figure 7.17 Technical drawing for Tyreflex coupling.

Table 7.9 Comparison between Tyreflex coupling and the new design

Coupling type		Nominal torque Nm	Outside dia. (mm)	Max. Misalignment		End float (mm)
				Offset (mm)	Angular (deg.)	
Tyreflex	TY60 # #	127	165	1.6	4.0	+2.0
New design		130	50	3.0-3.5	5.0	2.0

D. Chainflex

The coupling is shown in Figure 7.18. The technical drawing for the same coupling is shown in Figure 7.19. Table 7.10 shows the comparison with respect to torque capacity, outside diameter and misalignments.

Figure 7.18 Chainflex coupling.

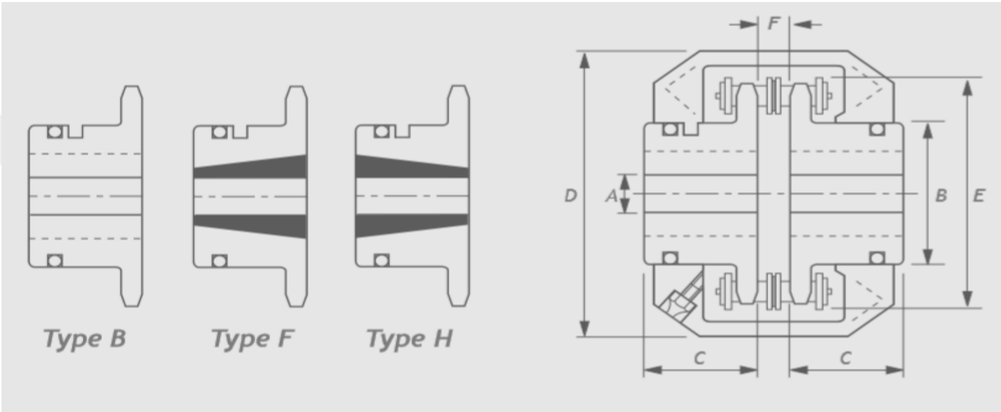


Figure 7.19 Technical drawing for Chainflex coupling.

Table 7.10 Comparison between Chainflex coupling and the new design

Coupling type		Nominal torque Nm	Outside diameter (mm)	Max. Misalignment		End float (mm)
				Offset (mm)	Angular (deg.)	
Chainflex	C33BBK	95.5	83	0.25	1.0	+1.0
New design		130	50	3.0-3.5	5.0	2.0

- **CHALLENGE Couplings [16]**

A. FFX Tyre Coupling

The coupling's technical drawing is shown in Figure 7.20. Table 7.11 shows the comparison with respect to torque capacity, outside diameter and misalignments.

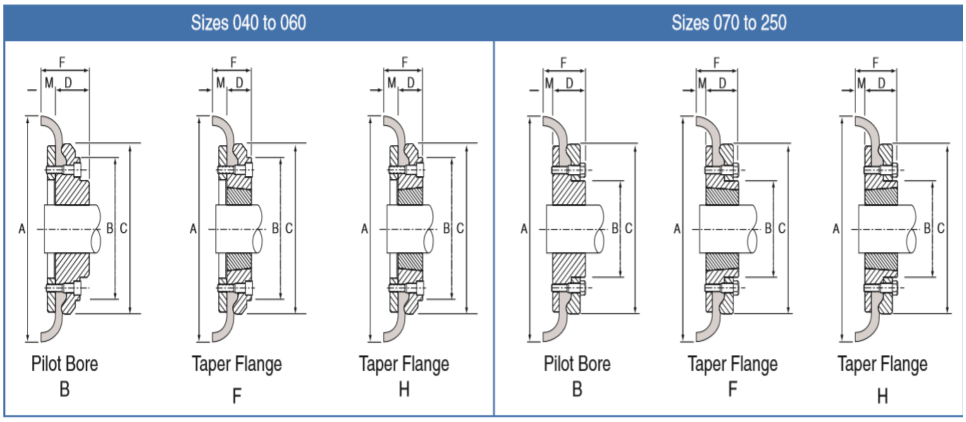


Figure 7.20 FFX Tyre coupling technical drawing.

Table 7.11 Comparison between FFX Tyre coupling and the new design

Coupling type		Nominal torque Nm	Outside diameter (mm)	Max. Misalignment		End float (mm)
				Offset (mm)	Angular (deg.)	
Tyreflex	060 B	127	165	1.6	4.0	+2.0
New design		130	50	3.0-3.5	5.0	2.0

- **ELIGN GEAR TYPE COUPLINGS**

A. ED Double Engagement [17]

The coupling's technical drawing is shown in Figure 7.21. Table 7.12 shows the comparison with respect to torque capacity, outside diameter and misalignments.

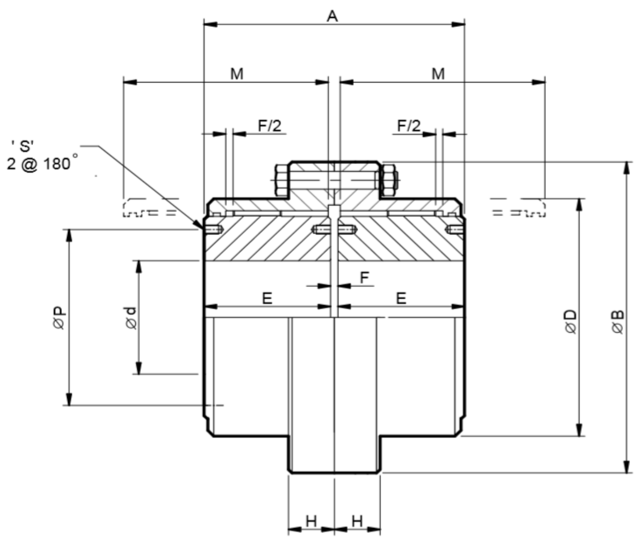


Figure 7.21 ED Double Engagement coupling technical drawing.

Table 7.12 Comparison between ED Double Engagement coupling and the new design

Coupling type		Nominal torque Nm	Outside diameter (mm)	Max. Misalignment in		End float (mm)
				Offset (mm)	Angular (deg.)	
ED Double Engagement	ED130	130	111	0.35	2*0.75	-
New design		130	50	3.0-3.5	5.0	2.0

B. ET Spacer Tube

The coupling's technical drawing is shown in Figure 7.22. Table 7.13 shows the comparison with respect to torque capacity, outside diameter and misalignments.

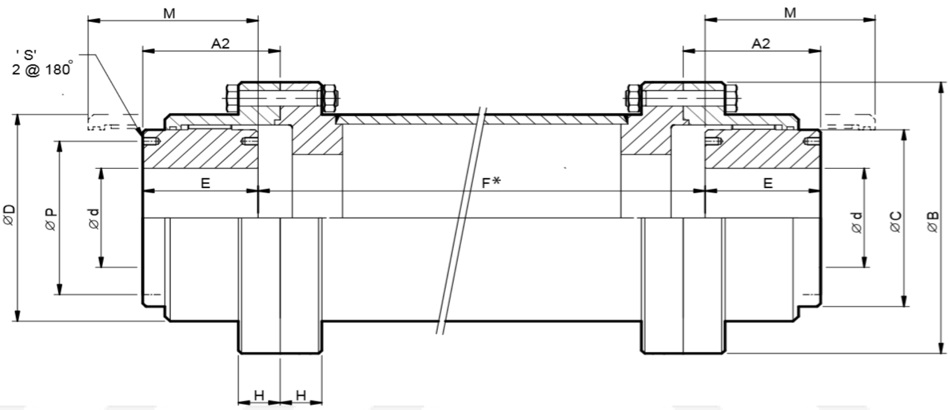


Figure 7.22 ET Spacer Tube coupling technical drawing.

Table 7.13 Comparison between ET Spacer Tube coupling and the new design

Coupling type		Nominal torque Nm	Outside diameter (mm)	Max. Misalignment in		End float (mm)
				Offset (mm)	Angular (deg.)	
ET Spacer Tube	ET130	130	111	1.5	2*0.75	-
New design		130	50	3.0-3.5	5.0	2.0

- **Lovejoy [18]**

A. RB type

The coupling's technical drawing is shown in Figure 7.23. Table 7.14 shows the comparison with respect to torque capacity, outside diameter and misalignments.

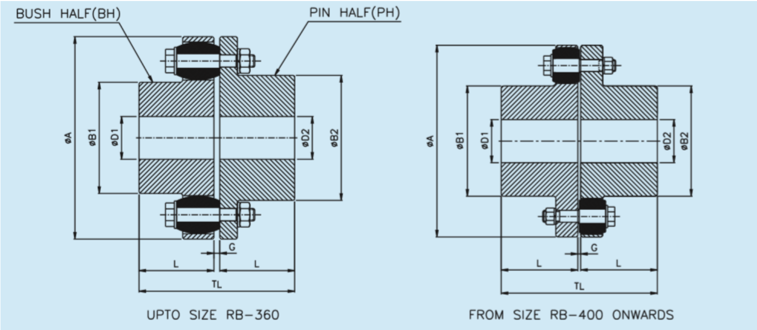


Figure 7.23 RB type coupling technical drawing.

Table 7.14 Comparison between RB type coupling and the new design

Coupling type		Nominal torque Nm	Outside diameter (mm)	Max. Misalignment in		End float (mm)
				Offset (mm)	Angular (deg.)	
RB	RB-105-3	95	105	0.3	1.0	2.0
	RB-116-4	143	116	0.3	1.0	2.0
New design		130	50	3.0-3.5	5.0	2.0

- **Norelem [19]**

A. Universal joints double

The coupling is shown in Figure 7.22. The technical drawing for the same coupling is shown in Figure 7.25. Table 7.15 shows the comparison with respect to torque capacity, outside diameter and misalignments.

Figure 7.24 Universal joints double.

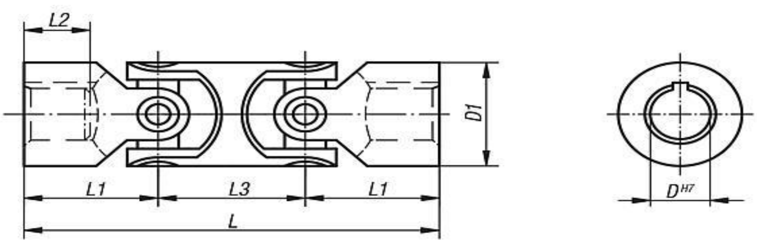


Figure 7.25 Universal joints double technical drawing.

Table 7.15 Comparison between Universal joints double coupling and the new design

Coupling type		Nominal torque (Nm)	Outside diameter (mm)	Max. Misalignment in		End float (mm.)
				Offset (mm.)	Angular (deg.)	
Universal joints double	23407-125160	130	50	-	5	-
New design		130	50	3.0-3.5	5	2

7.4 Experimental Work

A few experiment was made for the coupling that manufactured in chapter 6. A test system was designed and manufactured in mechanical engineering workshop. The components of the test mechanism were (1m lever, two block ball bearings, two shafts, two keys and one fixture). The designed mechanism with Solidworks is shown in Figure 7.26. The manufactured mechanism is shown in Figure 7.27.

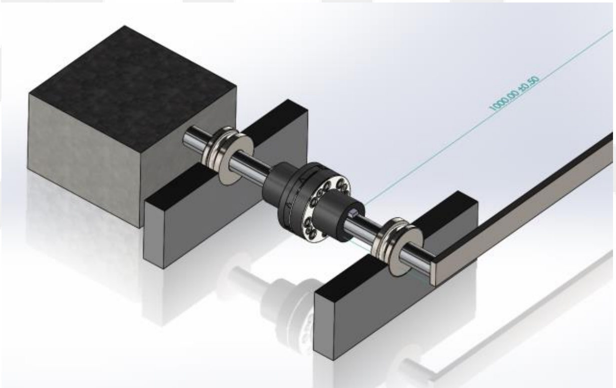


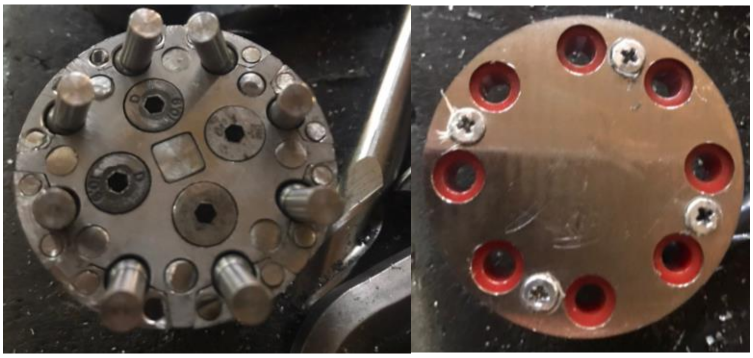
Figure 7.26 CAD model for the designed mechanism.

Figure 7.27 The manufactured mechanism.

The torque value was 200Nm. The coupling tolerated the load without any deformation in the critical part or the main body. The effected part was the resilient part (polyurethane), a small amount of deformation without any rupture was noticed at the 200Nm although at the 150Nm there was no deformation. The deformation can be referred due to low quality of the used material and the small thickness. This problem can be fixed easily by selecting a good quality polyurethane (the same that are using in the pin bushed coupling) and increase the thickness with giving it curvature shape.

Two more tests were made. The second test was with a value of 80 Nm held on the coupling for 3.5 hours. The third test was with value of 150 Nm. It was held for 3.25 hours. The main body were remaining with no damage or deformation. The polyurethane's deformation was increased. Figure 7.28 a, b and c shows the coupling before and after each test.

a



b

c

Figure 7.28 (a) The coupling before the tests (b) The coupling after the first test (c) The coupling after the second and third tests.

7.5 Results

A comparison between modified coupling and standard flange coupling was made for the stresses. Change in developed stresses on the modified coupling with respect to the stresses occurred in the standard flange coupling is shown in Table 7.16. The (-) sign means decreasing, (+) sign means increasing value.

Table 7.16 Comparison between the stresses of the modified coupling and standard flange coupling.

<table>
<tr><td rowspan="2">Object Name
Comparison</td><td colspan="3">Modified coupling</td><td colspan="2" rowspan="2">Standard flange coupling</td><td colspan="2" rowspan="2">Percentage change</td></tr>
<tr><td colspan="2">Numerical</td><td rowspan="2">Theoretical</td></tr>
<tr><td rowspan="2">Shear stress (MPa)</td><td>3MA</td><td>2MA</td><td>Numerical</td><td>Theoretical</td><td>3MA</td><td>2MA</td></tr>
<tr><td>21.89</td><td>21.78</td><td>22.40</td><td>66.45</td><td>73.48</td><td>-67.02 %</td><td>-67.22%</td></tr>
<tr><td rowspan="3">Equivalent Stress (MPa)</td><td colspan="3">Numerical</td><td colspan="2" rowspan="2">Numerical</td><td rowspan="2">3MA</td><td rowspan="2">2MA</td></tr>
<tr><td colspan="2">3MA</td><td>2MA</td></tr>
<tr><td colspan="2">145.96</td><td>71.12</td><td colspan="2">179.11</td><td>-18.50%</td><td>-60.29%</td></tr>
<tr><td rowspan="2">Total Deformation (mm)</td><td colspan="3">Numerical (3MA)</td><td colspan="2">Numerical</td><td colspan="2" rowspan="2">+14%</td></tr>
<tr><td colspan="3">7.3454e-002</td><td colspan="2">6.4084e-002</td></tr>
</table>

CHAPTER 8

CONCLUSION

8.1 Conclusion

Coupling is an important power transmission element. As any other design, the critical part must exist. The critical parts are the first parts to fail in any power transmission system, to avoid the damage in high-cost parts. In the new design, the pins are considered as critical parts. From the stress analysis according to ANSYS 16.2 for the new design and the standard flange coupling, the maximum shear stress, which will occur in the new design is near to 32 % of the maximum shear stress in the standard coupling. The maximum equivalent stress in the new design (two misalignment) is 39 % of the standard flange and the (three misalignment) is 81 % of the standard flange coupling which is expected due to the presence of the resilient material. For the same load, the stresses were occurred in the new design are less than the stresses in the standard flange coupling. In addition, the experimental tests that were made gave a good result. The coupling was designed for 130 Nm and it was tested on 80,150 and 200 Nm without any deformation in the critical parts or the main body of the coupling except the polyurethane which deformed in a small amount. Polyurethane's deformation is normal due to the high load and the small thickness. From all the above, it can be considered that the new design is more reliable than the standard flange coupling according to developed stresses.

The misalignment comparison between the new design and some coupling from company's catalogs shows good advantages for the new design over the other couplings in the torque capacity, dimensions and the ability to tolerate three types of misalignment at the same time. Although the design seems complex, but the manufacturing is not requiring any special equipment, it can be fully manufactured by normal CNC machines, the most complicated part is the rounded groove of major and secondary middle part as in Figure 6.27 and it can be manufactured easily with EDM machining.

From all the above, it can be considered that this design has smaller stress for the same torque value, relatively has a smaller size than the other flexible couplings with greater misalignment capacity in the three types of misalignment. New designed coupling can be manufactured with available machines.

8.2 Recommendation for Future Work

In this study, selecting a good quality polyurethane (the same that are using in the pin bushed coupling) and increase the thickness and giving it a curved shape. For future work, dynamic studies are needed in FEM and experiment. Since this design still under development and can be considered as concept design. For the calculations, it is only covering the main parts but it will be better if it can involve all the small details in order to get an accurate design. For the MATLAB script also can be developed in order to cover all the design and to give a direct calculation with the designed drawings.

REFERENCES

[1] Khurmi, R.S. Gupta, J.K. (2005). A Text Book of Machine Design. First Multicolour Edition. Ram Nagar New Delhi-110 055: Eurasia Publishing House (pvt.) ltd.

[2] Fixturlaser Canada, Laser Alignment & Vibration Specialist. Available at: http://www.fixturlaser.com/en-CA/Shaft-Alignment/Benefits-of-Shaft-Alignment/Identifying-Misalignment. Accessed 14.05.2019.

[3] Mar-Dustrial Sales Serving the Commercial Shipping and Fishing Industries. (1947). Available at: https://www.mardustrial.com/blog/falk-couplings-and-shaft-misalignment-101-radial-angular-and-axial-07-25-2011. Accessed 14.05.2019.

[4] Giannone, J. (1995). Laser-Optical Shaft Alignment Cuts Energy Costs, *World Pumps, Oxford*, **340**, 22–24.

[5] Rexnord Industrial Services. Available at: https://www.rexnord.com/blog/articles/types-of-couplings. Accessed 14.05.2019.

[6] Vijaya, V. G. (2013). Analysis of Rigid Flange Couplings, *International Journal of Innovative Research in Science, Engineering and Technology*, **2**. 7118-7126.

[7] Rajgadia, S., Das, D., Jaiswal, P., Basnet, A., Jha, A. R., Jaiswal, R., Karki, A., Barman, R. N. (2015). Design and Stress-Analysis of a Rigid Flange Coupling Using FEM, *International Journal of Innovative Research in Science, Engineering and Technology*, **4**. 9599-9609

[8] Babu, K. N., Chandpasha, S., Dattaterya, K., Jathin, K. (2017). Design and Failure Analysis of Flange Coupling with Uniform Strengthen Bolts, 400-406.

International Journal of Innovative Research in Science, Engineering and Technology, **6**.

[9] Tech, M., Hod. (2018). Design and Analysis of Rigid Flange Coupling, *Anveshana's International Journal of Research in Engineering and Applied Sciences*, **3**. 90-95

[10] Khidir, T. C. (2017). Design and Analysis of Bushed Pin Flexible Coupling, *International Journal of All Research Education and Scientific Methods,* **5.** 38-44.

[11] Zahariea, D. (2013). Structural Analysis of a Flange Coupling Without Clearance, *Applied Mechanics and Materials*, **371.** 652-656.

[12] Ringspann Installation and Operating Instructions for Elastic Pin and Bush Coupling REB…DCO. Available at: https://www.ringspanncorp.com/en-us/files/InstallationInstructions-REB-DCO-4780.pdf. Accessed 14.05.2019.

[13] ANSYS Is the Global Leader in Engineering Simulation, ANSYS® Engineering Data, Release 16.2, ANSYS, Inc.,

[14] Make It from Is a Database of Engineering Material Properties. Available at: https://www.makeitfrom.com/material-properties/ASTM-Grade-20-or-150-ISO-150-EN-JL-1020-F11401-Grey-Cast-Iron. Accessed 14.05.2019.

[15] Renold Resilient and Soft Start Couplings Catalogue. Available at: https://www.renold.com/media/162120/couplings_catalogue_2012_2.pdf. Accessed 14.05.2019.

[16] Challenge Coupling Catalogue. Available at: https://www.overveldtechniek.com/applications/weemen/overveld/files/Documenten/open_aandrijvingen/Challenge_koppelingen.pdf. Accessed 14.05.2019.

[17] Benzlers & Radicon Coupling Catalogue. Available at: https://www.radicon.com/~radicon/_docs/BR-Series-X.pdf. Accessed 14.05.2019.

[18] Rathi Coupling Catalogue. Available at: http://rathicouplings.com/wp-content/uploads/2018/07/pin-bush-rb.pdf. Accessed 14.05.2019.

[19] Norelem Universal Joints Double with Needle Bearing Catalogue. Available at: https://www.norelem.com/xs_db/DOKUMENT_DB/www/NORELEM/DataSheet/en/23/23407_Datasheet_12381_Universal_joints_double_with_needle_bearing_DIN_808--en.pdf. Accessed 14.05.2019.

[20] MSB India Is a Manufacturer & A Worldwide Supplier of Power Transmission Products Such as Couplings, Pulleys, Conveyor Belts Etc. Available at: http://www.msbindia.co.in/pinbush_coupling.htm. Accessed 14.05.2019.

[21] Lodam, A. B. (2015). Static Analysis of Rigid Flange Coupling, *Proceedings of International Conference on Mechanical and Production Engineering*. 295-297.

[22] Abduljabbar, A. A., Ismael, A. M., Yousif, I. E. (2018). Design and Analysis of Tire Coupling, *International Journal of Mechanical Engineering and Technology*. **9.** 1176-1182.

[23] Babu, K. N., Suneel, D. (2015). Failure Analysis of Flange Coupling with Two Different Materials, *International Journal of Engineering Research & Technology*. **4**. 587-589.

[24] Powar, R. S., Deshmukh, N. S., Suryawanshi, S. D. Flexible Coupling a New Approach, *Journal of Mechanical and Civil Engineering (IOSR-JMCE)*. 1-6.

[25] Jaiswal, S. B., Pasarkar, M. D. (2012). Failure Analysis of Flange Coupling in Industry, *International Journal of Emerging Technology and Advanced Engineering.* **2**. 405-409.

[26] Hariharan, V., Srinivasan, P. S. (2010). Vibrational Analysis of Flexible Coupling by Considering Unbalance, *World Applied Sciences Journal*. 1022-1031.

[27] Cuffaro, V., Cura, F., Mura, A. (2012). Analysis of the pressure distribution in spline couplings, *Proceedings of the Institution of Mechanical Engineers, Part C: Journal of Mechanical Engineering Science*. 2852–2859.

[28] Sheikh, S.M. (2012). Analysis of Universal Coupling Under Different Torque Condition, *International Journal of Engineering Science & Advanced Technology.* **2**, 690-694.

[29] Chavan, S.G. (2012). Stress Analysis of Flanged Joint Using Finite Element Method, *International Journal of Science and Research (IJSR).* 1187-1191.

[30] Pritish, B., Avinash, L. (2015). Static & Dynamic Analysis of Motor Flange, *International Journal for Scientific Research & Development.* **3,** 1209-1211.

[31] Khot, S. M., Pallavi, K. (2015). Simulation and Experimental Study for Diagnosis of Misalignment Effect in Rotating System, *Journal of Vibration Analysis, Measurement, and Control.* **3**. 165-173.

[32] Shah, D.B. (2013). Parametric Modeling and Drawing Automation for Flange Coupling Using Excel Spreadsheet, *International Journal of Research in Engineering & Technology (IJRET).* **1**. 187-192.

APPENDICES

APPENDIX A

SK is a MATLAB script (M file) for designing the new coupling with respect to the standard shafts, keys and bolts :

```
T=input('please input the torque in N.mm     ');
sf=input('please input the safety factor for the shaft     ');
sfkey=input('please input the safety factor for key    ');
sfpin=input('please input the safety factor for pin    ');
sfbolt=input('please input the safety factor for bolt    ');
Sh=input('please input the shear stress for the shaft Mpa.   ');
Shpin=input('please input the shear stress for the pin Mpa.   ');
Shbolt=input('please input the shear stress for the bolt Mpa.   ');
Shkey=input('please input the shear stress for the key Mpa.   ');
Schkey=input('please input the crashing stress for the key Mpa.    ');
n=input('please input bolts number  ')

rth=input('please input rubber thickness    ');

S=Sh/sf
Skey=Shkey/sfkey
Sckey=Schkey/sfkey
Sbolt=Shbolt/sfbolt
Spin=Shpin/sfpin
y='  the shaft diameter in mm. is :'
g=((T*16)/(S*3.14))^(1/3)
('the stander (d) that near to your calculation is    ' )
if ((g>1)&&(g<=6))
    disp('6')
elseif ((g>6)&&(g<=8))
    disp('8')
elseif ((g>8)&&(g<=10))
     disp('10')
elseif ((g>10)&&(g<=12))
     disp('12')
    elseif ((g>12)&&(g<=17))
      disp('17')
elseif ((g>17)&&(g<=22))
     disp('22')
     elseif ((g>22)&&(g<=30))
         disp('30')
           elseif ((g>30)&&(g<=38))
    disp('38')
 elseif ((g>38)&&(g<=44))
 disp('44')
```

```
elseif ((g>44)&&(g<=50))
  disp('50')
 elseif ((g>50)&&(g<=58))
     disp('58')
  elseif ((g>58)&&(g<=65))
    disp('65')
 elseif ((g>65)&&(g<=75))
  disp('75')
  elseif ((g>75)&&(g<=85))
  disp('85')
 elseif ((g>85)&&(g<=95))
  disp('95')
elseif ((g>95)&&(g<=110))
  disp('110')
elseif ((g>110)&&(g<=130))
   disp('130')
elseif ((g>130)&&(g<=150))
 disp('150')
elseif ((g>150)&&(g<=170))
 disp('170')
elseif ((g>170)&&(g<=200))
disp('200')
 elseif ((g>200)&&(g<=230))
 disp('230')
elseif ((g>230)&&(g<=260))
 disp('260')
 elseif ((g>260)&&(g<=290))
  disp('290')
 elseif ((g>290)&&(g<=330))
 disp('330')
  elseif ((g>330)&&(g<=380))
   disp('380')
 elseif ((g>380)&&(g<=440))
     disp('440')
end
d = input('choose the d you want   ')

'pins design in mm.'
Dpc=((0.5*d)/(n)^0.5)
Dp=Dpc+(2*rth)
PD=2*d+Dp+2*n
'critical pin design '
'total load '
loadtotal=T/(PD/2)
' load on each pin '
load=loadtotal/n
('pins critical dia. is ')
pindia=((load)/((3.14/4)*Spin))^0.5
pindia1=input('plrase input pin dia that suitble in mm.  ' )
```

```
'pin rounded head dia. is   '
prr=((((pindia1/2)^2)+((0.125*d)^2)))^0.5

(' hub design in mm. ')
D=2*d
L=1.5*d
(' key design in mm.')
widht=(2*T)/(L*Skey*d)
thickness=(4*T)/(L*Sckey*d)
( 'the stander key that near to your shaft dia  is     ' )
if ((g>1)&&(g<=6))
    disp('w=2 / t=2')
elseif ((g>6)&&(g<=8))
    disp('w=3 / t=3')
elseif ((g>8)&&(g<=10))
     disp('w=4 / t=4')
elseif ((g>10)&&(g<=12))
     disp('w=5 / t=5')
    elseif ((g>12)&&(g<=17))
      disp('w=6 / t=6')
elseif ((g>17)&&(g<=22))
     disp('w=8 / t=7')
     elseif ((g>22)&&(g<=30))
         disp('w=10 / t=8')
           elseif ((g>30)&&(g<=38))
                disp('w=12 / t=8')
 elseif ((g>38)&&(g<=44))
 disp('w=14 / t=9')
elseif ((g>44)&&(g<=50))
  disp('w=16 / t=10')
 elseif ((g>50)&&(g<=58))
     disp('w=18 / t=11')
  elseif ((g>58)&&(g<=65))
    disp('w=20 / t=12')
 elseif ((g>65)&&(g<=75))
  disp('w=22 / t=14')
  elseif ((g>75)&&(g<=85))
  disp('w=25 / t=14')
 elseif ((g>85)&&(g<=95))
  disp('w=28 / t=16')
elseif ((g>95)&&(g<=110))
  disp('w=32 / t=18')
elseif ((g>110)&&(g<=130))
   disp('w=36 / t=20')
elseif ((g>130)&&(g<=150))
 disp('w=40 / t=22')
elseif ((g>150)&&(g<=170))
 disp('w=45 / t=25')
elseif ((g>170)&&(g<=200))
disp('w=50 / t=28')
```

```
 elseif ((g>200)&&(g<=230))
 disp('w=56 / t=32')
elseif ((g>230)&&(g<=260))
 disp('w=63 / t=32')
 elseif ((g>260)&&(g<=290))
  disp('w=70 / t=36')
 elseif ((g>290)&&(g<=330))
 disp('w=80 / t=40')
  elseif ((g>330)&&(g<=380))
   disp('w=90 / t=45')
 elseif ((g>380)&&(g<=440))
     disp('w=100 / t=50')
end


('bolt design')

('1st group ')

('pitch diameter in mm. for the 1st bolts is')

DPb1=3*d
Dbolt=((8*T)/(n*3.14*Sbolt*DPb1))^0.5
if ((Dbolt>1)&&(Dbolt<=3))
    disp('M3 with D = 3 mm ')
elseif ((Dbolt>3)&&(Dbolt<=4))
    disp('M4 with D = 4 mm ')
elseif ((Dbolt>4)&&(Dbolt<=5))
      disp('M5 with D = 5 mm ')
elseif ((Dbolt>5)&&(Dbolt<=6))
     disp('M6 with D = 6 mm ')
 elseif ((Dbolt>6)&&(Dbolt<=8))
    disp('M8 with D = 8 mm ')
elseif ((Dbolt>8)&&(Dbolt<=10))
   disp('M10 with D = 10 mm ')
elseif ((Dbolt>10)&&(Dbolt<=12))
       disp('M12 with D = 12 mm ')
 elseif ((Dbolt>12)&&(Dbolt<=14))
     disp('M14 with D = 14 mm ')
      elseif ((Dbolt>14)&&(Dbolt<=16))
   disp('M16 with D = 16 mm ')
   elseif ((Dbolt>16)&&(Dbolt<=20))
 disp('M20 with D = 20 mm ')
   elseif ((Dbolt>20)&&(Dbolt<=24))
  disp('M24 with D = 24 mm ')
end

'2st group'
'pitch diameter in mm. for the 2st bolts is'
DPb3=(4*d)*0.92
```

```
{'you can change the pitch diameter for the second bolt group according ' ; 'to your
design and the program will give the suitable bolt dia '}
DPb2=input('please input the pitch diameter that is suitable for you design in mm
');
Dbolt2=((8*T)/(n*3.14*Sbolt*DPb2))^0.5

if ((Dbolt2>1)&&(Dbolt2<=3))
    disp('M3 with D = 3 mm ')
elseif ((Dbolt2>3)&&(Dbolt2<=4))
    disp('M4 with D = 4 mm ')
elseif ((Dbolt2>4)&&(Dbolt2<=5))
      disp('M5 with D = 5 mm ')
elseif ((Dbolt2>5)&&(Dbolt2<=6))
     disp('M6 with D = 6 mm ')
 elseif ((Dbolt2>6)&&(Dbolt2<=8))
    disp('M8 with D = 8 mm ')
elseif ((Dbolt2>8)&&(Dbolt2<=10))
   disp('M10 with D = 10 mm ')
elseif ((Dbolt2>10)&&(Dbolt2<=12))
       disp('M12 with D = 12 mm ')
 elseif ((Dbolt2>12)&&(Dbolt2<=14))
     disp('M14 with D = 14 mm ')
      elseif ((Dbolt2>14)&&(Dbolt2<=16))
   disp('M16 with D = 16 mm ')
   elseif ((Dbolt2>16)&&(Dbolt2<=20))
 disp('M20 with D = 20 mm ')
   elseif ((Dbolt2>20)&&(Dbolt2<=24))
  disp('M24 with D = 24 mm ')
end

'flange design in mm.'
Df=4*d
'flange thickness is :'
Tf=0.5*d
```

APPENDIX B

The SK output from MATLAB program with the input data are shown.

```
>> Sk
please input the torque in N.mm     810000
please input the safety factor for the shaft     2
please input the safety factor for key    2
please input the safety factor for pin    2
please input the safety factor for bolt    2
please input the shear stress for the shaft Mpa.   119.439
please input the shear stress for the pin Mpa.   144.25
please input the shear stress for the bolt Mpa.   144.25
please input the shear stress for the key Mpa.   144.25
please input the crashing stress for the key Mpa.    250
please input bolts number  6

n =
     6
please input rubber thickness    4
S =
   59.7195
Skey =
   72.1250
Sckey =
   125
Sbolt =
   72.1250
Spin =
   72.1250
y =
    '  the shaft diameter in mm. is :'
g =
  41.0380


ans =
    'the stander (d) that near to your calculation is     '
44
choose the d you want   44
d =
    44
ans =
    'pins design in mm.'
Dpc =
```

```
   8.9815
Dp =
  16.9815

PD =
 116.9815
ans =
   'critical pin design '
ans =
  'total load '
loadtotal =
  1.3848e+04
ans =
   ' load on each pin '
load =
  2.3081e+03
ans =
   'pins critical dia. is '
pindia =
   6.3848
plrase input pin dia that suitble in mm.  10
pindia1 =
   10
ans =
   'pin rounded head dia. is   '

prr =
   7.4330
ans =
   ' hub design in mm. '
D =
   88
L =
   66
ans =
   ' key design in mm.'
widht =
  7.7345
thickness =
   8.9256
ans =
   'the stander key that near to your shaft dia  is    '

w=14 / t=9
ans =
   'bolt design'
ans =
   '1st group '
ans =
   'pitch diameter in mm. for the 1st bolts is'
```

```
DPb1 =
   132
Dbolt =
    6.0106
M8 with D = 8 mm
ans =
    '2st group'


ans =
    'pitch diameter in mm. for the 2st bolts is'
DPb3 =
  161.9200
ans =
  2×1 cell array
    {'you can change the pitch diameter for the second bolt group according '}
    {'to your design and the program will give the suitable bolt dia '       }
please input the pitch diameter that is suitable for you design in mm    160
Dbolt2 =
    5.4594
M6 with D = 6 mm
ans =
    'flange design in mm.'

Df =
   176
ans =
    'flange thickness is :'
Tf =
    22
>>
```

APPENDIX C

MTLAB script of Stress analysis for pins and the bolt groups:

```
T=input('please input the torque in N.mm     ');
D1g=input('please input the PITCH dia for the first group in mm.     ');
ng1=input('please input the no. of bolts  for the first group       ');
dpg1=input('please input the dia. of bolts  for the first group in mm     ');
D1g2=input('please input the PITCH dia for the 2th group in mm.     ');
ng2=input('please input the no. of bolts  for the 2th group       ');
dpg2=input('please input the dia. of bolts  for the 2th group in mm     ');
D1a=input('please input the PITCH dia for the pins in mm.     ');
na1=input('please input the no. of pins        ');
dag1=input('please input the dia. of bolts  for the first group in mm     ');
ff11=' the total on the bolts is   '
loadtotal=(T/(D1g/2))
ff1='the load on each bolt is  '
loadoneachp1g=(loadtotal/ng1)
ff31=' the shear stress on each bolt is    '
shearg1=(loadoneachp1g/(0.7853*(dpg1^2)))

ff1=' the total on the bolts is   '
loadtota2=(T/(D1g2/2))
ff='the load on each bolt is  '
loadoneachp2g=(loadtota2/ng2)
ff3=' the shear stress on each bolt is    '
shearg2=(loadoneachp2g/(0.7853*(dpg2^2)))

ff21=' the total on the bolts is   '
loadtotal3=(T/(D1a/2))
ff121='the load on each pin is  '
loadoneachp3g=(loadtotal3/na1)
ff3211=' the shear stress on each pin is    '
shearga1=(loadoneachp3g/(0.7853*(dag1^2)))
```

APPENDIX D

The output of Stress analysis for pins and the bolt groups:

The used unite for the load is N. and for the stress is MPa.

```
>> totalscript
please input the torque in N.mm     810000
please input the PITCH dia for the first group in mm.     132
please input the no. of bolts  for the first group       6
please input the dia. of bolts  for the first group in mm     8
please input the PITCH dia for the 2th group in mm.     160
please input the no. of bolts  for the 2th group       6
please input the dia. of bolts  for the 2th group in mm     6
please input the PITCH dia for the pins in mm.     114
please input the no. of pins         6
please input the dia. of bolts  for the first group in mm     10
ff11 =
    ' the total on the bolts is   '
loadtotal =
   1.2273e+04
ff1 =
    'the load on each bolt is  '
loadoneachp1g =
   2.0455e+03
ff31 =
    ' the shear stress on each bolt is    '
shearg1 =
   40.6981
ff1 =
    ' the total on the bolts is   '
loadtota2 =
       10125
ff =
    'the load on each bolt is  '
loadoneachp2g =
   1.6875e+03
ff3 =
    ' the shear stress on each bolt is    '
shearg2 =
   59.6906
ff21 =
    ' the total on the bolts is   '
loadtotal3 =
   1.4211e+04
```

```
ff121 =
    'the load on each pin is  '
loadoneachp3g =
   2.3684e+03
ff3211 =
    ' the shear stress on each pin is    '
shearga1 =
   30.1594
```