

ADAMA SCIENCE AND TECHNOLOGY UNIVERSITY

School of Mechanical, Chemical and Materials Engineering



Design and Analysis of Flywheel Equipped Cycle Rickshaw

A thesis submitted to Adama Science and Technology University in partial fulfillment of the requirements for the award of the degree of

Master of Science

In

Automotive Engineering

By

Fikru Tadele

Advisor: Mr. N. Ramesh Babu

Mechanical System and Vehicle Engineering program

February, 2018

Adama, Ethiopia

ADAMA SCIENCE AND TECHNOLOGY UNIVERSITY

SCHOOL OF MECHANICAL, CHEMICAL AND MATERIALS ENGINEERING

Mechanical System and Vehicle Engineering program



Design and Analysis of Flywheel Equipped Cycle Rickshaw

By: - Fikru Tadele

APPROVED BY THE EXAMINERS BOARD

----- Chairman, program Graduate Committee	----- Signature	----- Date
----- Advisor	----- Signature	----- Date
----- Internal Examiner	----- Signature	----- Date
----- External Examiner	----- Signature	----- Date

CANDIDATE'S DECLARATION

I hereby declare that the work which is being presented in the thesis entitled “Design and Analysis of Flywheel Equipped Cycle Rickshaw” in partial fulfillment of the requirements for the award of the degree of Master of Science in Automotive Engineering is an authentic record of my own work carried out from February 2016 to February 2018 under the supervision of N. RAMESH BABU Department of Mechanical and Vehicle Engineering, Adama Science and Technology University, Ethiopia.

The matter embodied in this thesis has not been submitted by me for the award of any other degree or diploma. All relevant resources of information used in this thesis have been duly acknowledged.

Name	Signature	Date
Fikru Tadele	-----	2018-01-16-GC

Student

This is to certify that the above statement made by the candidate is correct to the best of my knowledge and belief. This thesis has been submitted for examination with my approval.

Name	Signature	Date
N. Ramesh Babu	-----	-----

Advisor

Dedication

To my beloved family and friends for motivation. To whoever provided help and contribute.

ACKNOWLEDGEMENT

All praises to the Almighty God because he gives to us peace, health, strength to accomplish our task successfully.

It is my great pleasure to acknowledge Mr. Ramesh Babu, (Associate professor) thesis advisor, for his scientific, technical guidance and support throughout this thesis, without his continuous corrections and constructive comments this project wouldn't have not been successful.

Also, I would like to extend my sincere appreciation to Adama Science and Technology University, Department of Mechanical System and Vehicle Engineering for their educational guidance and support.

Last but not least, I would like to express the utmost gratitude to my family and friends (special Naol Desalign) for their, understanding, support, encouragement and helping me to survive all the stress and complete this study successfully.

TABLE OF CONTENTS

Contents

ACKNOWLEDGEMENT	i
LIST OF FIGURES	vi
LIST OF TABLES.....	ix
ACRONYMS AND ABBREVIATIONS	xi
<i>ABSTRACT</i>	x
CHAPTER ONE: INTRODUCTION.....	1
1.1 Background of the Study.....	1
1.2. Statement of problem	4
1.3. Objectives.....	5
1.3.1. General objective	5
1.3.2. Specific objectives	5
1.4. Scope of the project.....	5
1.5. Limitations of the Study	5
1.6. Significance of the study	6
1.7. Thesis outline	6
CHAPTER TWO: LITERATURE SURVEY	7
2.1. Human power	7
2.2. Power output of the human body	7
2.3 Origin of cycle Rickshaw	8
2.4. Improvement Attempts.....	9
2.4.1 The Ox-trike	9
2.4.2 Tajmahal Cycle Taxi Improvement Project.....	10
2.4.3 The Indonesian Cycle Rickshaw-Becak	11
2.5. Kinetic energy recovery system	12
2.6. Ergonomics.....	13
2.7. Anthropometry	14
2.7.1. Human size percentile	14
2.8. Classification of cycle rickshaw.....	15
2.8.1 Cycle rickshaw layout decision	15

CHAPTER THREE: METHODOLOGY AND DESIGN SPECIFICATION	16
3.1. Project Methodology	16
3.2. Finite element analysis	16
3.3. Finite element software (ANSYS)	17
3.3.1. General steps to solving problem in ANSYS	18
3.4. Flow chart for design.....	19
3.5. Design specifications.....	20
CHAPTER FOUR: DETAIL DESIGN OF FLYWHEEL EQUIPPED CYCLE RICKSHAW ...	21
4.1. Frame design.	21
4.1.1. Frame section.....	21
4.1.2. Anthropometric study and Frame geometry analysis	22
4.1.3. CAD Modeling of the frame.....	24
4.1.4. Frame Analysis:.....	25
4.2. Finite Element Analysis of frame.....	27
4.3. Passenger seat design	35
4.3.1. Fit parameters	35
4.3.2. CATIA modeling of the passenger seat concept	36
4.4. Finite element analysis of passenger seat frame	36
4.5. Steering system of cycle rickshaw	39
4.5.1. Design requirements	39
4.5.2. Steering fork design.....	39
4.5.3. Steering geometry.....	40
4.6. Finite element analysis of front fork of cycle rickshaw.	42
4.7. Handlebar selection	48
4.8. Other steering parts	49
4.9 Drive train	50
4.9.1 Drivetrain part selection	51
4.9.2. Drive hub design for cycle rickshaw	55
4.9.3. Finite element analysis	56
4.9.4. Differential in cycle rickshaw.....	59
4.9.5. Design of Bevel Set	61
4.9.6. Splines for side bevel gears	63

4.9.7. Finite element analyses of side bevel gear	64
4.9.8. Cycle rickshaw rear axle shaft.....	68
4.9.9. Rear half axle shaft design calculation	69
4.9.10. Differential planet gear shaft design.....	73
4.9.11. Differential casing	75
4.9.12. Finite element analysis of differential casing assembly	76
4.9.13. Rear wheel hub shell.....	81
4.9.14. Finite element analysis of hub shell	82
4.10. Kinetic energy recovery system (KERS) design.....	85
4.10.1. KERS in cycle rickshaw	85
4.10.2. Working principle of cycle rickshaw KERS	86
4.10.3. KERS design requirement	87
4.10.4. Components of KERS of the cycle rickshaw	87
4.10.5. Design of flywheel.....	89
4.10.6. Stored energy on the flywheel at each gear ratio.....	90
4.10.6. Clutch.....	91
4.10.7. Axle	94
4.10.8. Chain.....	96
4.10.9. Other Accessories	96
4.11. Cycle Rickshaw Brake	98
4.11.1. Brake system requirements.....	98
4.11.2. Design Alternatives	99
4.12. Power needed for a ride cycle rickshaw	100
CHAPTER FIVE: RESULT AND DISCUSSION	103
5.1 Frame static structural analysis result	103
5.2. Passenger seat frame analysis result.....	104
5.3. Front steering link analysis result.....	105
5.4. Drive hub analysis result	106
5.5. Differential gear analysis result.....	106
5.6. Differential casing assembly analysis result	107
5.7. Wheel hub analysis result.....	108
5.8 kinetic energy recovery system result	108

CHAPTER SIX: SUMMARY, CONCLUSIONS AND RECOMMENDATIONS.....	109
6.1 Summary and Conclusion	109
6.2. Recommendation.....	110
6.3. Future works.....	110
Reference	111
Appendix A: Anthropometrics Data	115
Appendix B: Detail calculation of cycle rickshaw dynamics	117
Appendix C: Detail calculation for chain and sprocket selection.....	122
Appendix D: Terminology and proportion of miter bevel gear.....	132
Appendix E: Detail and assembly drawing of cycle rickshaw	134

LIST OF FIGURES

Figure 2. 1 Jinrikisha (rickshaw).....	9
Figure 2. 2.The ox-trike.....	10
Figure 2. 3.Taj Mahal Cycle rickshaw	11
Figure 2. 4.The Indonesian becak.....	12
Figure 2.5 The relative sizes of different percentile humans.....	14
Figure 3.1. Flow chart.....	19
Figure 4. 1 Frame Sections type.....	21
Figure 4. 2. Human body parts dimension needed to design cycle rickshaw.....	22
Figure 4. 3.Bicycle frame geometry	23
Figure 4. 4. Flywheel equipped cycle rickshaw frame model concepts.....	24
Figure 4. 5 Simplified representation of cycle rickshaw when negotiating road curve.....	25
Figure 4. 6 A simplified representation of the vehicle under dynamic braking deceleration.....	26
Figure 4. 7 Fly wheel equipped cycle rickshaw frame.....	27
Figure 4. 8 a), Course b), medium, c), fine, d), 10mm element size e) 3 mm element size meshed new cycle rickshaw frame model.....	30
Figure 4. 9 Loading and boundary condition for static start up.....	31
Figure 4. 10. Static start up simulation result, (a) Total deformation, (b) Von-Mises Stress, and (c) Factor of safety.....	32
Figure 4. 11 Steady state pedaling loading and boundary condition.....	33
Figure 4. 12 Stand pedaling simulation result, (a) Total deformation, (b) Von-Mises Stress, and (c) Factor of safety.....	34
Figure 4. 13 Different passenger seat frame model concepts.....	36
Figure 4.14 Meshed passenger seat structure model.....	37
Figure 4.15 Passenger seat frame loading and Boundary condition.....	37
Figure 4.16 Passenger seat (a) Total deformation, (b) Von-Mises Stress, and (c)Factor of safety.....	38
Figure 4. 17 Rigid Fork.....	40
Figure 4.18 Cycle rickshaw or bicycle steering geometry.....	41
Figure 4.19 Front link assembled model.....	43
Figure 4.20 Meshed front link model.....	43

Figure 4.21 Free body diagram of front wheel under front brake action.....	44
Figure 4.22 Font fork loading and boundary condition.....	46
Figure 4. 23 Total deformation (a) Total deformation, (b) Von-Mises stress on bolt, brace and fork leg and (d)Factor of safety.....	48
Figure 4. 24 Selected riser handlebar.....	49
Figure 4. 25 Headset.....	49
Figure 4. 26 Thread less stem.....	50
Figure 4. 27 (a) crank arm, (b) chain wheel.....	51
Figure 4. 28 Square taper bottom bracket.....	52
Figure 4. 29 Five speed Freewheel.....	52
Figure 4. 30 Front and rear derailleurs.....	53
Figure 4. 31 Gear shifter.....	54
Figure 4. 32 Standard foot pedal styles.....	54
Figure 4. 33 Drive hub assembly.....	55
Figure 4. 34 Drive hub model.....	56
Figure 4. 35 a) Meshed model and b) boundary and loading condition of hub body.....	57
Figure 4.36 Drive hub simulation (a) Total deformation, (b) Von-Mises Stress, and (c)Factor of safety.....	58
Figure 4. 37 Assembled and exploded view of flywheel equipped cycle rickshaw differential...60	60
Figure 4. 38 Straight miter gears.....	61
Figure 4. 39 Bevel-gear tooth forces.....	63
Figure 4. 40 a) General form of spline (b) internal spline.....	64
Figure 4. 41 a) Differential gear model and b) Meshed model.....	65
Figure 4. 42 Loading and boundary condition.....	66
Figure 4. 43 Differential gear simulation result, a) total deformation, b) Von-Mises stress, c) factor of safety.....	67
Figure 4. 44 Semi-floating rear axles.....	68
Figure 4. 45 Automobile Semi-floating axle free body diagram.....	69
Figure 4. 46 Free body diagram of new cycle rickshaw half rear axle.....	70
Figure. 4.47 Differential planet gear shaft free body diagram.....	74
Figure 4. 48 External retaining ring.....	75

Figure 4. 49 (a) isometric, (b) side, and (c) top view of differential casing assembly.....	76
Figure 4. 50 Meshed model of differential casing assembly.....	77
Figure 4. 51 Loading and boundary condition.....	78
Figure 4. 52 Differential casing assembly simulation result, a) Total deformation, b) Von-Mises stress, c) factor of safety.....	79
Figure 4. 53 Final drive assembly exploded view.....	80
Figure 4. 54 Flanged bearing.....	80
Figure 4. 55 a), Radial ball and b), sintered bronze flanged cylindrical bushings bearing.....	81
Figure 4. 56 a) Hub shell model, b) Meshed model of hub shell.....	83
Figure 4. 57 Boundary and Loading condition.....	84
Figure 4. 58 a) Total deformation, b) Von-Mises stress, c) factor of safety.....	85
Figure 4. 59 KERS of cycle rickshaw.....	86
Figure 4. 60 larger and smaller sprockets of KERS.....	87
Figure 4. 61 Flywheel.....	88
Figure 4. 62 Flywheel speed at different gear ratio versus stored kinetic energy graph.....	91
Figure 4. 63 Spiral jaw clutch.....	92
Figure 4. 64 Friction clutch free body diagram and its model.....	93
Figure 4. 65 Free body diagram of KERS axle.....	94
Figure 4. 66 KERS shaft.....	95
Figure 4. 67 Internal retaining ring.....	96
Figure 4. 68 Clutch cable and lever.....	97
Figure 4. 69 Compression and Tension spring.....	97
Figure 4. 70 a) Rim, b) drum, and c) disk brake.....	100
Figure 4. 71 Disc brakes selected for the braking system.....	100
Figure 4.72 Road resistance powers versus speed graph.....	102

LIST OF TABLES

Table 4. 1 Upright bicycle frame size	23
Table 4. 2 Frame Materials property and composition	28
Table 4. 3 Seat anthropometric measurement	35
Table 4. 4 Steering system specifications	39
Table 4. 5 Adhesion coefficient on different road condtions [40].....	45
Table 4. 6 Riser handle bar specification [42]	48
Table 4. 7 Specification of head set [42].....	49
Table 4. 8 Stem Specification [42].....	50
Table 4. 9 Specification of front axle [42]	50
Table 4. 10 Crankset Specifications [42]	51
Table 4. 11 BB Specifications [42]	52
Table 4. 12 Cassette Specifications [42]	52
Table 4. 13 Derailleur Specifications [42]	53
Table 4. 14 Shifter Specifications [42]	54
Table 4. 15 Pedals Specifications [42].....	54
Table 4. 16 Miter bevel gear size calculation.....	61
Table 4. 17 Acting loads new cycle rickshaw miter bevel gear tooth.....	63
Table 4. 18 Formulas for SAE straight spline [45].	64
Table 4. 19 Half axle shaft load calculation.....	72
Table 4. 20 Axle shaft properties [44].....	72
Table 4. 21 Extetnal retaining ring specification [47]	75
Table 4. 22 Mechanical property of differential assembly.....	77
Table 4. 23 Flanged bearing specifactions.....	80
Table 4. 24 Drive hub Radial ball bearing specifactions [48].....	81
Table 4. 25 Differential casing sintered bronze flanged cylindrical bushings bearing specifactionsn [48]	81
Table 4. 26 Hub shell major dimension	82
Table 4. 27 Flywheel speed at different gear ratio and stored kinetic energy.....	91

Table 4. 28 Properties of materials commonly used for lining of friction surfaces. [41]	93
Table 4. 29 KERS angular contact ball bearing specifactions[48]	96
Table 4. 30 Internal retaining ring [47]	97
Table 4. 31 Stainless steel compression spring specification [51].	98
Table 4. 32 Stainless steel tension spring specification [50].	98
Table 4. 33 Advantages and disadvantages of brake options	99
Table 4. 34 Total resistance power	102
Table 5. 1 . Effect of the meshing on the no. of nodes, no. of elements and results	103
Table 5. 2 Frame simulation result.....	104
Table 5. 3 Seat simulation result	105
Table 5. 4 Front steering link simulation result	105
Table 5. 5 Simulation result of hub body	106
Table 5. 6 Simulation result of miter bevel gear	107
Table 5. 7 Simulation result of differential casing assembly	107
Table 5. 8 Simulation result of hub shell	108

ACRONYMS AND ABBREVIATIONS

$\vec{a}$	Acceleration		
A_a	Addendum		
C_D	Drag coefficient	KERS	Kinetic energy recovery system
D_d	Deddendum		
K_s	Service factor	l	length, wheel base, offset distance
n	factor of safety		
A	Area	M	Bending moment
AISI	American Iron and steel Institute	m	mass
ANSYS	Analysis system	p	power, Chain pitch
ASTU	Adama science and technology university	r	Radius, rake,
		t	Thickness, trail,
BB	Bottom bracket	T	Torque, teeth number
b	Distance. Face width	V	Velocity
$C.G$	Center of gravity.	$V.R$	Velocity ratio
d	Diameter	W	Weight
F	Force	y	Distance from the neutral axis
FE	Finite element	α	Head angle.
g	Gravitational acceleration	Γ	pitch angle
GR	Gear Ratio	θ	Angle
h	height	μ	Coefficients friction
HPV	Human power vehicle	σ	Stress
I	Moment of inertia	τ	Shear stress
J	Polar moment of the inertia	φ	Back cone Angle
2D	Two dimensional	ψ	Pressure Angle
		ω	Angular velocity
		ρ	Density
		3D	Three dimensional

ABSTRACT

Recent awareness in fossil fuel depletion due to large consumption and environment degradation due to fossil fuel combustion emission has generated interest in ecofriendly transportation system in both developed and developing regions of the world. Government and private sectors are encouraging innovative development and use of energy efficient vehicles for transportation of people and goods. This thesis focuses on the design of efficient human powered vehicle (or flywheel equipped cycle rickshaw) which can harvest kinetic energy lost during braking or slow down for later use. This done by using a mechanism incorporated with flywheel and clutch system. A flywheel can temporarily store the kinetic energy from the cycle rickshaw when the rider needs to slow down or riding downhill. The operating mechanism of this vehicle similar to upright bicycle except KERS. To comprehensive this project the study conducted by reviewing different related previous works. For the design of this vehicle, or its parts, and system like frame, passenger seat, front fork, differential parts, drive hub, rear axle, hub shell and kinetic energy recovery system ergonomic or anthropometry data had been used. After ergonomic or anthropometry data collected geometry of those parts, and system are determined and its model developed by using CATIA software. In the next phase, static structural FE analysis are performed on the basis of static and dynamic load, and material section design requirement point of view. That is taking in to consideration material availability, material cost, machinability, able to assemble and disassemble. For FE analysis ANSYS 15 and 18.1 is used to find out total deformation, von-mises stress, and factor of safety. And last the rest parts of this vehicle are selected from standard bicycle parts. The 3D and 2D production drawing were prepared on CATIA V5. And required power to ride cycle rickshaw was analyzed for asphalt level road to investigate whether designed cycle rickshaw can be propelled by human power or not. Generally, from FEA and other analysis optimum result was obtained.

Keywords: *Cycle rickshaw, Human powered, Flywheel, Anthropometrics, Ergonomics, KERS .*

CHAPTER ONE: INTRODUCTION

1.1 Background of the Study

History of human powered transportation goes back as far as 3500 BC when wheels were used to pull carts in ancient civilizations. Advent of steam engine and internal combustion engine in the 17th century propelled development of modern transportation system on the wheels. Availability of inexpensive source of energy gave rise to mass transportation system in the following centuries. In recent years, rise of energy cost, awareness on depleting fossil fuel reserve, and concern for environment has generated awareness in developing eco-friendly and sustainable technology in transportation across the globe. Currently there are more than 800 million cars and truck on the road worldwide, burning in excess of 260 billion gallons of fuel each year releasing over 5 trillion pounds of CO₂ in the atmosphere[1]. Due to accelerating population and economic growth in the developing regions of the world, these numbers are increasing rapidly. Concern for depletion of fossil fuel reserve, environmental pollution and global warming underscores the need to stop or reverse this trend through more efficient, eco-friendly and sustainable transportation system in the near future. Meanwhile, the bicycle or the human powered vehicle industry, which currently generates no pollution [2]. Human powered vehicles or non-motorized transport, such as bicycle, tricycle, cycles rickshaws etc. are some of the purest forms of transportation.

Among human powered cycle rickshaw is valid for this study, it is a three wheeled tricycle design, pedaled by a human driver in the front, and with a back seat in the rear for passengers or for conveying goods. They are capable of carrying a driver and 2 to 3 passengers or freight loads of up to 250 kg at speeds of 5 to 12 km/hr over distances of up to 40 km. It is widespread in developing countries, where cycle rickshaws cater for a wide range of mobility needs. Common forms include cycle-rickshaw (Bangladesh, India), becak (Indonesia), cyclos (Vietnam, Cambodia), samlor (Thailand), saika (Myanmar), bicitaxi (Columbia, Cuba) and ecologico (Mexico) [3].

The potential benefit of cycle rickshaws and its popularity ranges from the flexibility, easy availability and lower cost. Cycle rickshaws are available for door to door services and are very flexible serving neighborhoods where road width is too narrow to accommodate larger modes of transport. They are ideal for short and medium length trips. cycle rickshaws are the most efficient

means of public transit over short distances in cities. cycle rickshaws occupy less road and parking space. Investment and operating costs are low and hardly need any foreign exchange. They also do not consume any fuels that means cycle rickshaw do not contribute to air and noise pollution, are environmentally sound and contribute least to roadway damage. They provide earning opportunities for a multitude of low income urban residents having no alternative means of survival. Cycle rickshaws play a further role in supplementing the congested public transport services [4].

The popularity and significance of cycle rickshaws also gradually increasing in several developed countries of Europe and parts of North America over the last 10-15 years. This re-emergence of cycle rickshaws in the form of pedicels is primarily attributable to their flexibility and contribution to the urban dynamic; their eco-sustainability, and their ability to operate where use of motorized vehicles is restricted. In particular, in cities such as Berlin, London, New York and Vancouver, pedicabs are making the transition from 'novelty' to 'serious' transport mode. In these countries the riders of the pedicab are also have full dignity and in many cases well educated and also acts tour guide. Students also ply these on part time basis to earn income to support their studies [3] [5].

There are many types of cycle rickshaws all over the developing world. Predominantly, they can be categorized as the ones with sitting cart behind the driver, low sitting cart in front of driver and a side sitting cart. In India and Bangladesh (high rickshaw usage), the rickshaws are with the sitting cart behind. The traditional rickshaws are made with a front bicycle frame, angle iron chassis, wooden rear body, steel seat and mass produced bicycle components and other decorative components [4].

Generally, there are different design improvements done on specific areas that are targeted upon to reduce weight, improve efficiency, improve strength, reduce flex, enhance safety and increase comfort. Many steps were taken to improve the traditional cycle rickshaws to make them more efficient. First, the improvements were made with strong tubular frame and a wide comfortable seat with woven seat suspension – its height was kept low for easy access to women clad in traditional clothes and for children and the elderly who are the most common users. The low center of gravity for easy pulling and less effort by the multispeed gear system was provided. and safety features have been enhanced as well. There have been many experiments with the rear differential, multispeed gear system, and disk brake. Still, FE analysis is not done yet and cycle rickshaw is not

able to harvest the kinetic energy lost in process of braking or slowing down. Capturing and storing this kinetic energy for later use would vastly improve efficiency of cycle rickshaw.

The technique of capturing and storing kinetic energy of the vehicles for later use are called The Kinetic Energy Recovery System (KERS). It is a system which first stores the kinetic energy into the form of useful energy and recovers back as per requirement. In other words, KERS is a collection of parts which stores some of the kinetic energy of a vehicle under deceleration and then releases this stored energy back into the drive train of the vehicle. Kinetic energy recovery systems (KERS) store energy when the vehicle is braking and return it when accelerating. It is used for providing a power boost to vehicle [6].

For the driver, it is like having two power sources at his disposal, one of the power sources is the engine while the other is the stored kinetic energy. Almost complete energy is wasted during braking because kinetic energy is mostly converted into heat energy or sometimes sound energy that is dissipated into the environment. KERS enables Vehicles to harness some of this kinetic energy and in doing so will assist in braking. It provides braking and boosting both to the vehicle. By using a proper mechanism, this stored energy is converted back into kinetic energy giving the vehicle extra boost of power [7].

There are two basic types of KERS systems i.e. Electrical and Mechanical. The main difference between them is in the way they convert the energy and how that energy is stored within the vehicle. Battery-based electric KERS Bareilly two systems require a number of energy conversions each with corresponding efficiency losses. On reapplication of the energy to the driveline, the global energy conversion efficiency is 31–34%. The mechanical KERS system storing energy mechanically in a rotating fly wheel eliminates the various energy conversions and provides a global energy conversion efficiency exceeding 70%, more than twice the efficiency of an electric system. This design of KERS cycle rickshaw was motivated by a desire to build a flywheel energy storage unit to improve the efficiency [8].

Cycle rickshaws are not common transportation in Ethiopia. But, as developing country Ethiopia facing the different problems that transport shortage in urban area, unemployment of graduate, less educated and poor rural citizens who looking for work. Like those developed and developing country mechanically efficient cycle rickshaw can play its own role as part of solution.

This thesis goal was to design human powered flywheel equipped cycle rickshaw that would be an aesthetically attractive, sustainable, and efficient commuter vehicle. Design lay out of cycle rickshaw is a delta-style with rear-wheels drive for a single rider and two passengers.

One of the most important requirements is that the vehicle must have an optimum strength, weight and low price. In order to achieve this goal, it is necessary to use the optimal material and suitable frame to ensure the required quality with the lowest cost and simple construction. Finally, to represent 3D model of new cycle rickshaw and it's component CATIA V5 R20 software are used and ANSYS 15 or 18.1 used as well for FE analysis.

1.2. Statement of problem

Transport is an important component of economic activity in all countries so in Ethiopia. It provides access to jobs, education, health care, agricultural and other services. Also transportation is the backbone for development of civilization [9][10]. It is related with economy that is utilized for movement of people and goods from one place to another. Various means of transportation can be self-propelled or manually driven. Every normal human being is using either one or the other form of transportation facilities available. Among transportation means petroleum fuel powered road transport is the nearest and flexible to people which allows even to operate from door-to-door over short distance at the most competitive prices. However, most road transportation that internal combustion engine based propulsion it uses petroleum fuel and it causes environmental impacts, hurt our country economy due to foreign exchange and its cost increase from time to time. Moreover, in this time transportation of both goods and humans increased rapidly with regard of population and economic growth. Increased amount of transportation means increased amounts of vehicles on the roads and increased amounts of kilometers driven. This development has not been possible without increased amounts of fuel used, the consequence of which is inevitably increased amount of CO₂ emissions; this dependence on vehicles has a negative impact on our global environment due to the large-scale consumption of fossil fuels and greenhouse gas emissions. Additional, petroleum fuel is limited natural resource that means it can deplete in the future [2]. Also in our country there are localized issues of meeting transportation need of vast masses of population that cannot afford a personal vehicle and also due to the fact that many roads are narrower to accommodate cars and buses, not to mention the fact that all roads, lanes and streets cannot be connected directly with public transportation systems and there is always a need for a

feeder system for the mass transportation systems and door to door transport services. This project is a way to design efficient human powered vehicle that potentially will serve as a partial solution to those problems.

1.3. Objectives

1.3.1. General objective

This thesis aim is to design flywheel equipped cycle rickshaw based on ergonomic principles, which would be more comfortable and efficient for the rider as well for the passengers.

1.3.2. Specific objectives

- To design ergonomic, rigid and possible light weight frame.
- To design front steering link.
- To design differential system to cycle rickshaw.
- To modify drive hub.
- To modify wheel hub shell.
- To design kinetic energy recovery system and.
- To perform FEA by ANSYS software to designed and modified pars of flywheel equipped cycle rickshaw.

1.4. Scope of the project

The scope of this thesis work is to design kinetic energy storing cycle rickshaw that during braking or downhill climbing for later use or to boost the speed of cycle rickshaw. The design includes the frame, passenger seat, front steering link, power splitting system or differential and KER system. The scope of this thesis also includes modeling of each component of cycle rickshaw using CATIA V5R20 software and deigned 3D model components import to finite element analysis software (ANSYS 15.07/ or 18.1 software) to carry out static structure (stress and deformation) analysis.

1.5. Limitations of the Study

The main limitation of this project was resource and computer capacity. The simulations have been run on a laptop with a 2.4GHz CPU formed by three cores with 4 GB of RAM memory. Because of this limitation, the analysis simulations needed more time to mesh the model to fine mesh in order to get more accurate result. This limitation also forced the meshes to be as optimized as possible which is to be compatible.

Non-availability of licensed ANSYS software is another big limitation, lack of experience and training in the software and compatibility of CATIA V5R20 and ANSYS also another problem. Additional software ANSYS Space claim had to be learned to solve this compatibility problem. Due to that it took long time for designing and analysis.

1.6. Significance of the study

If this vehicle (flywheel equipped cycle rickshaw) manufactured and being operational it can reduce environment degradation, noise and air pollution. Because unlike internal combustion engine propelled vehicle, flywheel equipped cycle rickshaw driven by human muscle power which is eco-friend, quiet and pollution free. And it can contribute to income generation and employment for educated or less educated unemployed youths, wellbeing of their families and to the country's economic development itself.

1.7. Thesis outline

This thesis organized in six chapters. Chapter one begins with an introduction and background of the research. This chapter also contains the problems statement, the objectives, scope, limitation and significances. Chapter 2. This portion is where the researcher gained additional knowledge of this study. presents literature review on relevant topics and works done in the field. Chapter 3. This section expresses all the processes, tools, and approaches that were used by the researcher in obtaining the objectives or goals of this study. Also, describes specification of flywheel equipped cycle rickshaw. Chapter 4. Show detail design of flywheel equipped cycle rickshaw components. It includes CATIA model of each designed and modified components. Also, includes ANSYS FEA of the frame, steering link, passenger seat, drive hub, differential, and wheel hub shell. At the end of this chapter required power to drive flywheel equipped cycle rickshaw theoretically determined. In chapter 5 overall result of FE simulation were discussed. In last conclusions including recommendation or future works were described in chapter 6.

CHAPTER TWO: LITERATURE SURVEY

2.1. Human power

Surprisingly, the basic principles of the operation of the human can be compared with the operation of the internal combustion engine. In both cases, the input to the system is chemical energy and oxygen. In the case of the human body, the chemical energy is taken in the form of nutrition (food), whereas in the combustion engine, the corresponding input energy is fuel. In both cases, oxygen for the process is taken from the ambient air.

There are chemical similarities regarding the input energy of both cases. Human nutrition can be divided into three main groups: carbohydrates, fats and proteins. The chemical elements present at these nutritional compounds are mostly carbon, hydrogen and oxygen, out of which carbon and hydrogen can be found also from hydrocarbon fuels.

In the energy conversion process of the human body and the combustion engine, the end result in both cases is the same: mechanical work, heat and carbon dioxide. In the combustion engine, the main difference being that in the human body, the "fuel burning" process is slower and takes place at lower temperatures.

The only wanted output from a combustion engine is mechanical work, but the side-products, heat losses and formation of carbon dioxide, cannot be avoided. However, in the case of the human body, also some heat generation is needed to keep the body vital and functional [11].

2.2. Power output of the human body

The human body produces, as already mentioned both heat and mechanical work. In terms of using the power to move a vehicle, naturally the mechanical power is the one that matters. The heat generation is another factor that needs consideration in human powered vehicle design to keep the ambient conditions in the vehicle pleasant for the driver. However, let's at this point concentrate on the mechanical power.

The human body works at a certain power level even at rest. The heat production is roughly 100 W when the person is not working. If the person is working and producing, for example, 100 W of mechanical power, there is a higher amount of heat generated at the same time. This phenomenon, typical to the human body, equals the case of most combustion engines: they both produce more heat than mechanical power, meaning that the efficiency is below 50 %.

A healthy trained human being can produce relatively high amount of power for a short period of time. However, the power delivering capability lowers quite rapidly because of the fatigue and exhaustion of the body parts. When designing a vehicle, the continuous power output should be used as the basis for the engineering work instead of the peak power.

The actual amount of power available depends, of course, very much of the particular individual. The physical condition and health of the person involved has a great influence of the power that he or she is capable of producing.

A vehicle intended to be moved by the power from the human legs is a very good way of utilizing human power, since the most powerful muscles of the human body are located in the legs.

The peak values are over 1200 W, but the level goes rapidly down along with time. For them, power over 300 W-450 W is possible for several hours [11-13].

2.3 Origin of cycle Rickshaw

The modern day cycle rickshaw evolved from the Jinrikisha (literally: "man powered car") introduced in Yokohama Japan by Baptist Missionary Jonathan Goble in 1874.

The explosion of the Jinrikisha or "rickshaw", as a means of economical transportation is credited to Akiha Daisuke who made several improvements upon the original model. While there are vague references (illustrations and art) to a similar human drawn vehicle operating in France in the mid-seventeenth century, the concept never really took hold in Europe. It is thought that a combination of poor timing and a lack of technology were principal reasons for its absence.

The Jinrikisha replaced the Kago, a litter carried on poles on the shoulders of two men, as a more efficient and convenient form of transportation. By 1874 the human pulled rickshaw was exported throughout Asia and their numbers were growing every year. At the height of their popularity near the end of the nineteenth century, there were close to 200,000 rickshaws in Japan. The first cycle rickshaws began to appear in Singapore in the late 1920's and Thailand in the early 1930's these were called "Trishaws" based on their tricycle design. The trishaws became most popular during WW II as a means of providing efficient transportation while dealing with fuel shortages during the war [14-16].

Just as the Jinrikisha replaced the Kago, the cycle-rickshaw began to replace its human-pulled counterpart. By the mid-twentieth century, most of the traditional Jinrikisha had been forced out of

business either by government bans, competition from the cycle-rickshaws or other more modern forms of rapid transportation.

The cycle-rickshaw can be found in many countries with some basic consistency of design features. Known by many different names, they are all variations of the modern day cycle-rickshaw. Tony Wheeler and Richard l'Anson provide an excellent history of the rickshaw in text and pictures in their book "Chasing Rickshaws". "Whether it's called a trishaw, sidecar, pedicab, cyclo or becak, it's the true Asian taxicab" [14].



Figure 2. 1 Jinrikisha (rickshaw) [14].

2.4. Improvement Attempts

2.4.1 The Ox-trike

The first cycle rickshaw improvement project was launched in the early 1970's. In 1973 Stuart Wilson, an engineering lecturer at Oxford University, designed a differential for use in Indian rickshaws. The aim was to overcome the problem that only one of the rear wheels was driven. The following year a "Mark-2" version was produced which was completely re-designed rickshaw. This was named 'Ox-trike'. The main features of the Ox-trike were a 3-speed Sturmey-Archer gear (which allowed three gearings smaller, stronger wheels; chassis made of box section steel sheet; powerful foot brake and the differential as shown in Fig.2.1. The plan was to dispatch the vehicle in kit form for local assembly. Then as the Ox-trike became more widely used, kits would be limited to complex components which couldn't be locally made (e.g. 3-speed free wheel, etc.). Ox-trike

was such a radically different design. Many of its components were unable in third world countries. Local rickshaw makers would have had to learn a completely new design [17-18].



Figure 2. 2 The oxtrike [17].

2.4.2 Tajmahal Cycle Taxi Improvement Project

The Taj Mahal Cycle Taxi Improvement Project was initiated by the Institute for Transportation and Development Policy (ITDP) based on a request from the Municipality of Agra, the Indian Government. Many individuals in India had been working for years on possible improvements on the cycle rickshaw. In March of 1998, ITDP's human powered vehicle engineer arrived in New Delhi and decided to develop three prototypes. The first would be a modest improvement of the existing cycle rickshaw operating in Delhi within very tight cost parameters. The new vehicle should cost no more than 25% more than the cost of the existing vehicle, or no more than \$150.

The second prototype would be a luxury version of the cycle rickshaw operating in Agra, sticking more or less to the traditional design, but creating an integral frame, a lighter weight vehicle, with superior components. This vehicle would retain the classic hand crafted features of the traditional vehicle but be targeted to a higher end tourist market.

The third vehicle would be an essentially experimental vehicle which would be the best cycle rickshaw that can be made from the perspective of speed, stability, and style, but using all Indian components, without concern for the cost. The purpose of this third vehicle was to demonstrate the technical possibilities of human powered vehicle technology, which can be made to travel up to the speed of a normal motor vehicle.

New Delhi made an attempt to improve the cycle rickshaw, with the objective to reduce the strain on the rickshaw puller through a multiple gearing system [19].

The carriage has also been made lightweight as shown in Fig.2.3.



Figure 2. 3 Taj Mahal Cycle rickshaw [19].

2.4.3 The Indonesian Cycle Rickshaw-Becak

Using the technology that helped to improve the Indian rickshaw, Indonesian drivers look forward to a becak that is much easier to use. The traditional Indonesian becak and the Indian cycle rickshaw are somewhat different. In the Indian vehicle, the passengers sit behind the driver, facing forward. In the becak, the passengers face forward, sitting in front of the driver. Both are very heavy, have no gears, and are not designed with consideration for the ergonomics of the driver.

An Indonesian team from Gadjah Mada University (GMU) and the Municipality of Yogyakarta requested support for its own rickshaw modernization program. They wanted to reduce dependence on motor transport while preserving a place for the becak in traditional Javanese culture.

The Indonesian becak (Fig.2.4), by contrast, is entirely made in small shops, using only mass-produced wheels and some components from a normal bicycle. The frame is made out of sturdy pipe, and the seat is made out of wood, steel drum, and steel filigree. The iron and wood chair and chassis are so strong that the vehicle lasts more than ten years, with replacement only of the wheels, tires and brakes. The traditional becak is much more comfortable than a traditional cycle rickshaw, but it is much heavier, around 100kg, and very hard to steer.



Figure 2. 4 The Indonesian becak [20].

The final prototype reduced the weight of the traditional vehicle by more than 30 percent, to 65 kg. This was done by replacing the angle-iron chassis and traditional bicycle frame with a single, integral tubular frame designed specifically for rickshaws, and by replacing the heavy wooden seat with a light, tubular seat and permanent canopy. While a gear system was developed, it was not popular. Ultimately, the modern vehicle succeeded commercially not because it was lighter but because it was more comfortable for the passengers and cost less to make than the traditional vehicle

2.5. Kinetic energy recovery system

KERS means Kinetic Energy Recovery System and it refers to the mechanisms that recover the energy that would normally be lost when reducing speed of vehicle. The energy is stored in a mechanical form and retransmitted to the wheel in order to help the acceleration. Electric vehicles and hybrid have a similar system called Regenerative Brake which restores the energy in the batteries. The device recovers the kinetic energy that is present in the waste heat created by the car's braking process. It stores that energy and converts it into power that can be called upon to boost acceleration.

There are principally two types of system - battery (electrical) and flywheel (mechanical). Electrical systems use a motor-generator incorporated in the car's transmission which converts mechanical energy into electrical energy and vice versa. Once the energy has been harnessed, it is stored in a battery and released when required.

Mechanical systems capture braking energy and use it to turn a small flywheel which can spin at up to 80,000 rpm. When extra power is required, the flywheel is connected to the car's rear wheels. In contrast to an electrical KERS, the mechanical energy doesn't change state and is therefore more efficient [21-22].

Sudipta Saha, Abhik Bose, G. Sai Tejesh, S.P. Srikanth(2013) [23] investigate the performance of a flywheel can be attributed to three factors, i.e., material strength, geometry (cross-section) and rotational speed. While material strength directly determines kinetic energy level that could be produced safely combined (coupled) with rotor speed, this study solely focuses on exploring the effects of flywheel geometry on its energy storage/deliver capability per unit mass, further defined as Specific Energy. Proposed Computer aided analysis and optimization procedure results show that smart design of flywheel geometry could both have a significant effect on the Specific Energy performance and reduce the operational loads exerted on the Shaft/bearings due to reduced mass at high rotational speeds. This paper specifically studies the most common five different geometries (i.e., straight/concave or convex shaped 2D).

M.lavakumar, R .prasanna srinivas (2013) [24] This paper involved the design and analysis of flywheel to minimize the fluctuation in torque, the flywheel is subjected to a constant rpm. The objective of present work is to design and optimize the flywheel for the best material. The flywheel is modeled with solid 95 (3-D element), the modeled analyses using free mesh. The FEM mesh is refined subject to convergence criteria. Preconditioned conjugate gradient method is adopted during the solution and for deflections. Von-mises stress for both materials (mild steel and mild steel alloy) are compared, the best material is suggested for manufacture of flywheel.

2.6. Ergonomics

Ergonomics is the science of designing user interaction with equipment and workplaces to fit the user and it is study of designing equipment and device that fit the human body, its movement, and its cognitive abilities.

Ergonomics (or human factor) is the scientific discipline concerned with the understanding of interaction among humans and other elements of a system, and the profession that applies theory, principles, data and methods to design in order to optimize human well-being and overall system performance.

Ergonomics is employed to fulfill the two goals of the health and productivity. It is relevant in the design of such things as safe furniture and easy to use interfaces to machines and equipment. Proper ergonomic design is necessary to prevent repetitive strain injuries, which can develop over time and can lead to long-term disability.

2.7. Anthropometry

Anthropometry is the science that measure the range of the body size in a population. An early tool of physical anthropology, it has been used for identification, for the purposes of understanding human physical variation, in paleoanthropology and in various attempts to correlate physical with racial and psychological traits. Anthropometry involves the systematic measurement of the physical properties of the human body, primarily dimensional descriptors of body size and shape[25].

Today, anthropometry plays an important role in industrial design, clothing design, ergonomics and architecture where statistical data about the distribution of body dimensions in the population are used to optimize products [26].

2.7.1. Human size percentile

Since people come in a variety of sizes, and their bodies are not built in the same proportion, “design for an average” (50th percentile) will not work. For each body segment to be fit, the dimensions that are critical for the design must be determined. A minimal or a maximal value, or a range may be critical. The goal of product design is to accommodate as many users as possible.

A number of countries in the world have sampled sufficient numbers of their respective population to have developed a national anthropometric profile and presented in tabular format or annotated diagrams of human figures. The primary dimensions measured are of bone, muscle, and adipose tissue. This data is used in human factors/ergonomics applications in order to ensure that designs and standards are realistic [27].

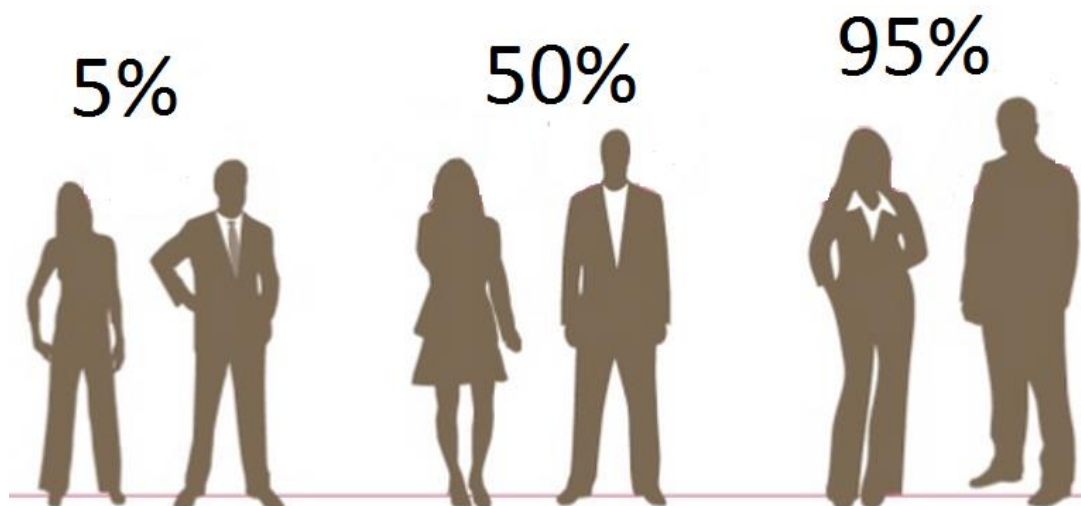


Figure 2.5 The relative sizes of different percentile humans.

2.8. Classification of cycle rickshaw

Cycles rickshaw classified in to two based on the layout of the wheels, such as delta and tadpole [5]

- I. **Delta layout** (fig. 2.2. Ox-trike or fig 2.3 Taj Mahal Cycle rickshaw) with two wheels at the back and one steered wheel at the front; Out of the two rear wheels, one or both may be driven.
- II. **Tadpole layout** (Fig. 2.4. Indonesian becak) with one wheel at the back, usually driven and two wheels at the front that are steered.

2.8.1 Cycle rickshaw layout decision

The delta layout cycle rickshaw is valid for this work based on safety, stability and available space for KERS. On delta layout the passenger sits behind the rider, this important for the following reasons.

- There is no passenger or pillars that block rider visibility.
- Location of the center of gravity is far away from front axle which means cycle rickshaw is stable during hard braking.
- It required less effort for steering.
- Free available place for KERS mount beneath passenger seat.

While on the tadpole layout passenger seat in front of rider, this lead to

- Passenger or pillars of passenger seat compartment can block rider visibility; this can expose the vehicle to traffic accidents.
- Location of center of gravity near to front axle, this can cause cycle rickshaw flip over during hard braking due weight shifting from rear axle to front.
- It required more effort for steering due to presence of passenger weight on the front axle.
- Space limit for KERS.

CHAPTER THREE: METHODOLOGY AND DESIGN SPECIFICATION

3.1. Project Methodology

In order to complete this thesis project, there were several steps which had been followed through. First this thesis started with literature review study. Various sources like journal, books, article, others and previous research works on bicycle and tricycle or cycle rickshaw were referred in order to have comprehensive understanding of the subject under research and to review other similar research works.

After related research review was done on the area the frame configuration has been selected. Then the basic or overall dimension of the frame has been determined from the anthropometric data of the 95% population size, and bicycle frame size that from upright bicycle manufacturing manual data. The other components like, front link, drive axle, differential, and KERS dimension has been obtained from anthropometric data, design analysis and existing bicycle parts. After all, needed dimension obtained 3D CAD modeling of the frames and components of cycle rickshaw generated in CATIA V5R20 and saved in CATIA format or IGIS format. That saved model imported to ANSYS 15.07/or 18.1 to perform finite element analysis. In addition, that ANSYS space claim used for cleaning and repairing the model created in CATIA software in order to increase mesh refinement and result accuracy. Also, keyShot 7 software had been used for rendering finalized cycle rickshaw model. Some components of bicycle which used in cycle rickshaw are selected from bicycle parts like steering components (handlebar, stem, headset, and handle grip), drivetrain components (chain, crankset, pedals, sprockets, wheels, front axle, bottom bracket, derailleur and gear shifter) and brake components (brake, levers, cable). Also, the other standard accessory, seat material, pedal, roof cover etc. has been selected from based on purpose. Finally, required power to ride cycle rickshaw had been performed its resistance loads (road load) such as air drag and rolling resistance.

3.2. Finite element analysis

In the design of many modern devices and equipment, the finite element method has become a tool for the many successful products that we have come to use daily. In this study, the analysis of a flywheel equipped cycle rickshaw frame and its other components is carried out by finite element method. Historically, intuition and trial-and-error in physical prototyping testing processes have played important roles in coming out with the evolution of a machine components.

However, using such a physical trial-and-error design procedure is costly and time consuming, and has its limitations, especially when new materials are introduced and when new applications or demands are placed on the structure. Hence, there is the need to use the finite element tool to help the designer come up with reliable properties in the design to meet the demands expected by the consumers. Using the finite element method to perform a virtual prototyping instead of a physical prototyping can save lots of cost, time and effort.

There will be number of factors to be considered when it comes to designing a flywheel equipped cycle rickshaw frame and its other components. For example, factors like the weight of the frame, the maximum load on the frame, the impact toughness of the frame, and so on. In a finite element analysis, one would require information on the material properties, the boundary conditions and the loading conditions. The loading conditions on a cycle rickshaw can be extremely complex. In this study we consider a loading condition of a static start-up is applied to the flywheel equipped cycle rickshaw [28].

3.3. Finite element software (ANSYS)

American Dr. John Swanson founded ANSYS. Inc. in 1970 with a vision to commercialize the concept of computer simulated engineering, establishing himself as one of the pioneers of Finite Element Analysis (FEA). ANSYS publishes engineering analysis software across a range of disciplines including finite element analysis, structural analysis, computational fluid dynamics, explicit and implicit methods, and heat transfer. It enables us to solve complex structural engineering problems and make better, faster design decisions. With the finite element analysis (FEA) tools available in the suite, we can customize and automate solutions for our structural mechanic's problems and parameterize them to analyze multiple design scenarios. We can connect easily to other physics analysis tools for even greater fidelity. ANSYS structural analysis software is used throughout the industry to enable engineers to optimize their product designs and reduce the costs of physical testing.

ANSYS provides a cost-effective way to explore the performance of products or processes in virtual environment. This type of product development is virtual prototyping. With virtual prototyping techniques, users can iterate various scenarios to optimize the product long before the manufacturing is started. The enables a reduction in the level of the risk, and in the cost of infective design.

ANSYS also can work by integration with other CAD software Connections are referred to as Geometry Interfaces. With direct interfaces to all major computer-aided design (CAD) systems, support of additional readers and translators, and an integrated geometry modeler exclusively focused on analysis, ANSYS offers the most comprehensive geometry-handling solutions for engineering simulation in an integrated environment. The currently supported CAD Connections are: Autodesk, CATIA V4, V5 and V6, Creo Elements/Direct Modeling, Creo Parametric, Solid Edge, SolidWorks.

3.3.1. General steps to solving problem in ANSYS

Like solving any problem analytically, it needs to define (a) physical model, (b), material properties, (c) meshing model, (d) boundary condition and (e)solution.

a) Build geometry

Construct a two or three dimensional representation of the object to be modeled and tested using the work plane coordinates system within ANSYS.

b) Define material properties

Now that the part exists, define a library of the necessary materials that composed the object (or project) being modeled. This includes thermal and mechanical properties.

c) Generate mesh

At this point ANSYS understands the makeup of the part. Now define how the modeled system should be broken down into finite pieces.

d) Applying loads

Once the system is fully designed, the last task is to burden the system with constraints, such as physical loading or boundary conditions.

e) Obtain solution

This is actual a step, because ANSYS need to understand within what state (steady, state, transient etc.) the problem must be solved [28].

3.4. Flow chart for design

The flow diagram shown below describes the method followed in the overall design and analysis of the new cycle rickshaw components on the base of ergonomics, strength, material and configuration of the wheel.

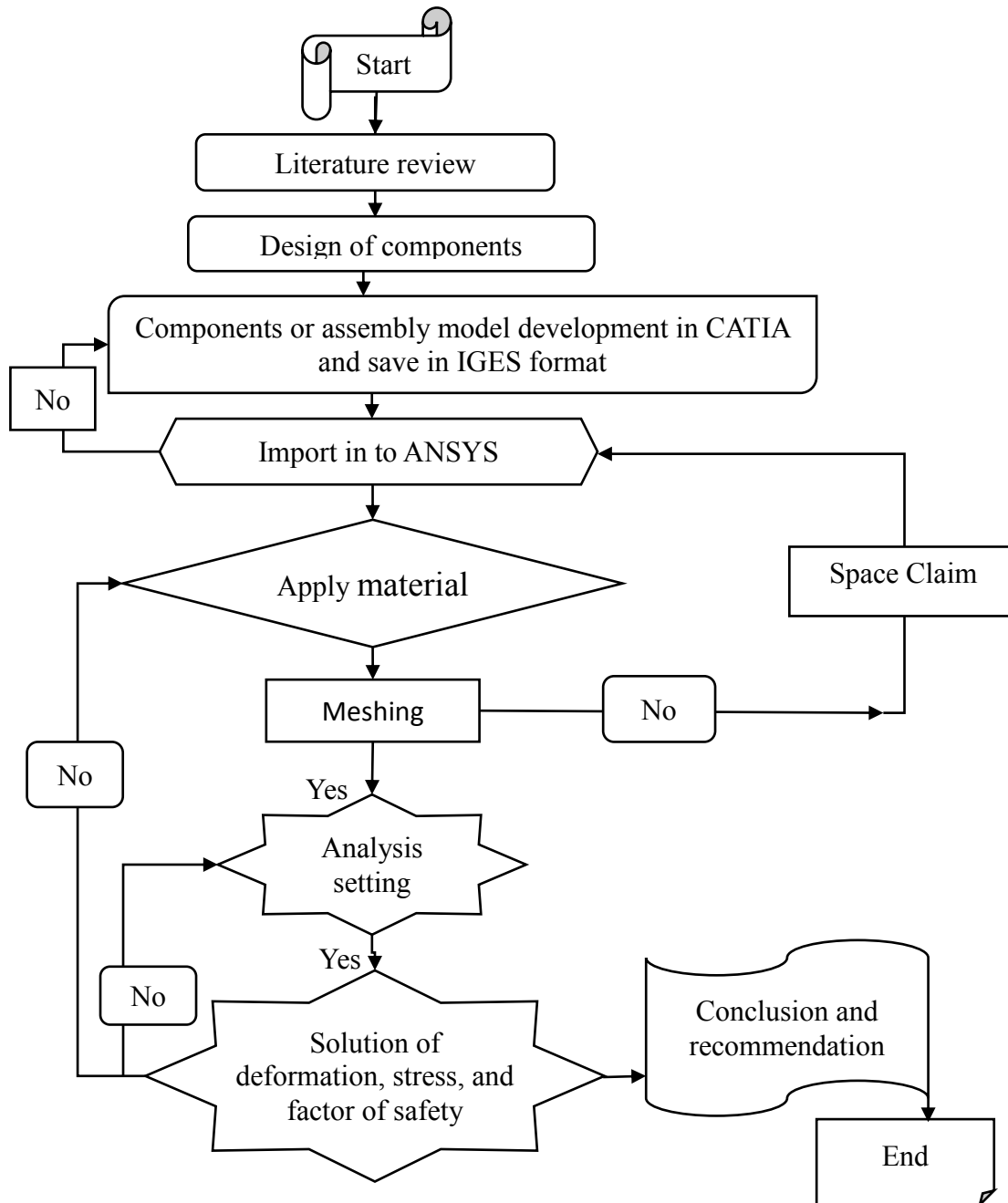


Figure 3.1 Flow chart

3.5. Design specifications

The following specification of this study cycle rickshaw considered based on rider strength, comfort, efficient performance and reliability to the vehicle.

- ❖ Maximum load: 240kg (rider plus two passengers each 80kg).
- ❖ Wheelbase: up to 1.95 m.
- ❖ The new cycle rickshaw operating mechanism must be like bicycle except KERS.
- ❖ The frame of the new cycle rickshaw must accommodate 5% - 95% size population.
- ❖ The maximum speed of the vehicle not exceeding 30km/hr.
- ❖ The cycle rickshaw will be steered through the bicycle's handlebars at 1:1 steering ratio will be maintained.
- ❖ 610 mm (24 inches) both rear and front wheels.
- ❖ This new cycle rickshaw compatible with existing parts.

Safety

- ❖ The cycle rickshaw passenger seat must include footrest, backrest and side hand rest support and safety for the passenger while the new cycle rickshaw is in motion.
- ❖ Protection devices around moving parts of the device will be used to protect the passenger from any injuries (particularly additional wheels and chain) if the said devices are within reach of the passenger.
- ❖ Armrests or similar side supports will be incorporated into the passenger seating of the to ensure minimal lateral motion of the passenger during turning.
- ❖ The device must not contain any sharp edges that could harm the passenger.

CHAPTER FOUR: DETAIL DESIGN OF FLYWHEEL EQUIPPED CYCLE RICKSHAW

The mechanical design of the flywheel equipped cycle rickshaw consists of the following structure and components frame, steering fork link, passenger seat, drive hub, differential, drive axle, wheel hub shell and KERS.

4.1. Frame design.

A cycle rickshaw frame is the main supporting structure to which all other components are attached. It supports load of passengers, rider, and other mechanical components to deal with static and dynamic loads, without undue deflection. Therefore, the frame should be strong enough to withstand the rigorous force that will act on it but at the same time it should be as light in weight as possible to increase efficiency [29]. the design of flywheel equipped cycle rickshaw frame is an assurance between rigidity, weight and volume, without forgetting final cost have to be considered the load resistance and fatigue, the load limit on the merging points, the building and assembly. In this thesis only takes in to accounts static structural analysis.

4.1.1. Frame section

Generally, frame of a vehicle made from the following section such as channel, box and tubular section as shown in fig. 4.1.

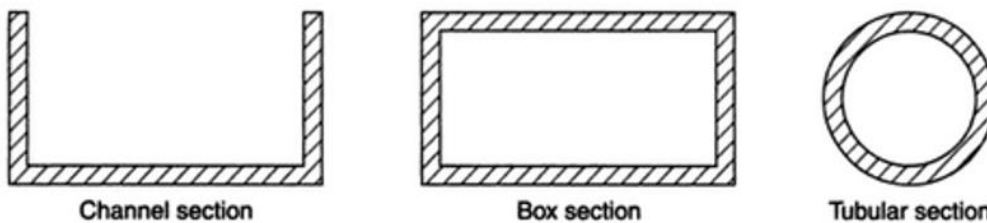


Figure 4.1 Frame Sections type

Channel section used in long member of the frame since it has good resistance to bending. Box section used in short members of the frame and it has good resistance to bending and torsion. While tubular section is used in the frame which subjected to torsion. Therefore, box and tubular section is selected for this thesis work base on load carrying character criteria [29].

4.1.2. Anthropometric study and Frame geometry analysis

A cycle rickshaw is a transport machine used by and driven by human being and many of its dimensions are conditioned by the morphology of the human body. So it is important to see what the standard dimensions of the human beings are, in order to design comfortable frame to any size of user. To achieve this larger size to takes into account that is 95% of the population. In order to determine overall length and width of cycle rickshaw frame ten dimension of the human body parts are enough. Figures 4.2. shows human body parts dimension needed during cycle rickshaw design. Finally, the information has been taken from international data on anthropometry, developed by Hans W. Jürgens Ivar and A. Aunt in 1998 [30], in their book Ethiopian population anthropometric data are available. Detail tabulate data found in appendix A.

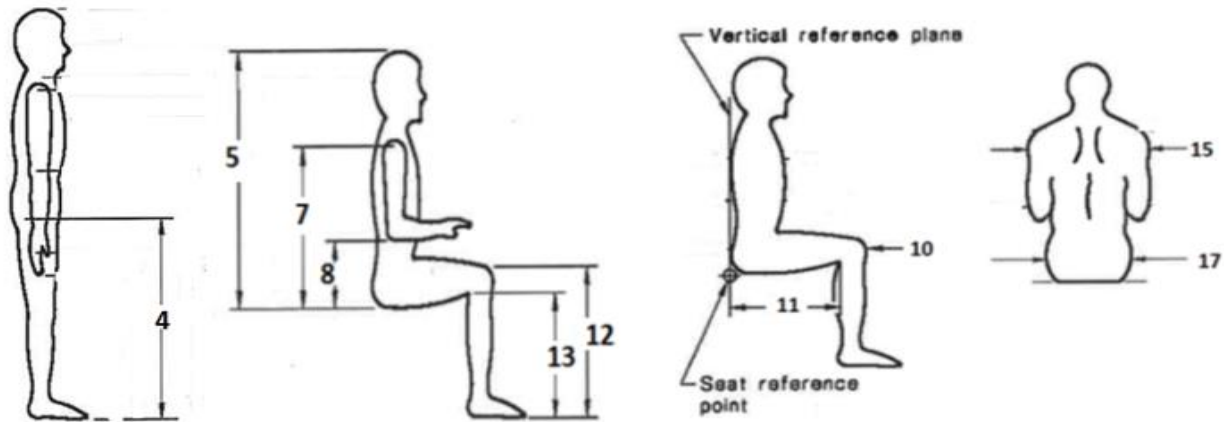


Figure 4.2 Human body parts dimension needed to design cycle rickshaw [30].

Anthropometrics is basic parameters to design comfortable frame but this only is not enough to determine every component size that means experimentally determined component size such as head tube length and angle, seat tube length and angle, reach size, stack size, and crank length are needed especially to rider part frame. We can get such data from upright bicycle frame builder company based on rider inside leg size.

Bicycle manufacturers set and provide bicycle frames in a variety of sizes with the top of the head tube moving up and away from the crank center as the size of the rider increases, so theoretically, one could use the manufacturer's guidelines for a suitable size based on rider height and then fine tune the position using the adjustability of seat posts, and stems etc. This because, it is practically impossible that only one bicycle or cycle rickshaw frame could be functional and satisfy all size of

the population without creating from a certain discomfort to severe difficulty in management for those people which are in the limits. To solve this problem frame will be equipped with an adjustable moving tube which permits the displacement of the seat post and stem [31]. For this thesis work upright bicycle frame parameters which is seat tube length and angle, head tube length and angle, stack, reach, BB shell size, crank arm size and handler bar had been selected based on 50% of the adult population size from road bikes on sale in the UK today table 4.1

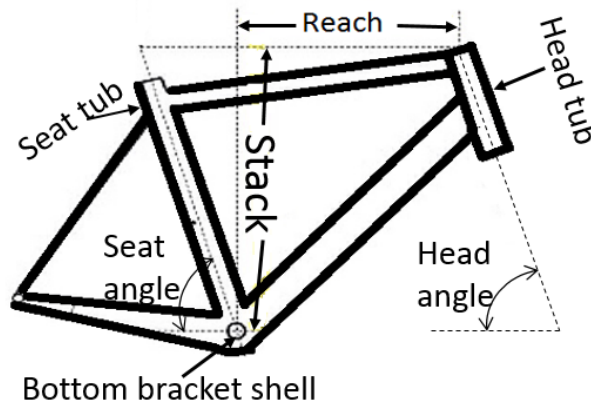


Figure 4.3 Bicycle frame geometry

Table 4. 1 Upright bicycle frame size

Size (cm)	48	50	52	54	56	58
Height (cm)	152-160	160-168	165-173	170-178	175-183	178-188
Inseam (cm)	70-74	74-77.7	76.3-80	78.6-82.3	81-84.6	82.3-87
Seat tube length (mm)	435-445	455-480	485-500	500-520	515-565	540-580
Horizontal top tube (mm)	505-520	510-530	520-540	535-550	540-565	560-585
Seat tube angle (degree)	74-75	73.5-74.5	73.25-74	73.0-73.5	73.0-73.5	73.0
Head tube length (mm)	100-120	110-140	120-155	120-175	140-190	160-225
Reach (mm)	355-370	360-375	375-380	380-390	385-400	390-410
Stack (mm)	500-515	505-535	510-550	520-565	550-590	570-620
Crank Length (mm)	165	165	165-170	170	172.5	172.5
Handlebar width (mm)	380	400	420	420	420	440
BB shell width (mm)	68 or 73	68 or 73	68 or 73	68 or 73	68 or 73	68 or 73

The height and inseam of the rider is 1690mm and 816mm respectively then select the correct size of bike frame from table that is 54cm.

Dimension for passenger carriage frame

Rear total track width = number of passenger * hip breath of passenger + space for side support.

From anthropometric 95th% data been used for design. The cycle rickshaw passenger compartment carrying capacity is two passengers.

$$\text{Rear total track width} = 2 \times 410\text{mm} + 50\text{mm} = 870\text{mm}$$

Length of passenger compartment = Buttock-knee length + assume little free space of passage foot rest and rear support.

$$\text{Length of passenger compartment} = 660\text{mm} + 100\text{mm} = 760\text{mm} \approx 800\text{mm}$$

4.1.3. CAD Modeling of the frame.

The four CAD model concepts of the frame was developed based on anthropometrics and selected bicycle frame size analyzed data. The frame CAD models concepts was developed on CATIA V5. Two section type (i.e. box and tabular) has been used to develop CAD models and with standard section size of 35mm diameter and 2.5 mm thick for head tube, 30mm diameter tube with 1.5mm thickness, for box section 1.5 mm thick and (30x30) mm, and (35 x30) mm was used.

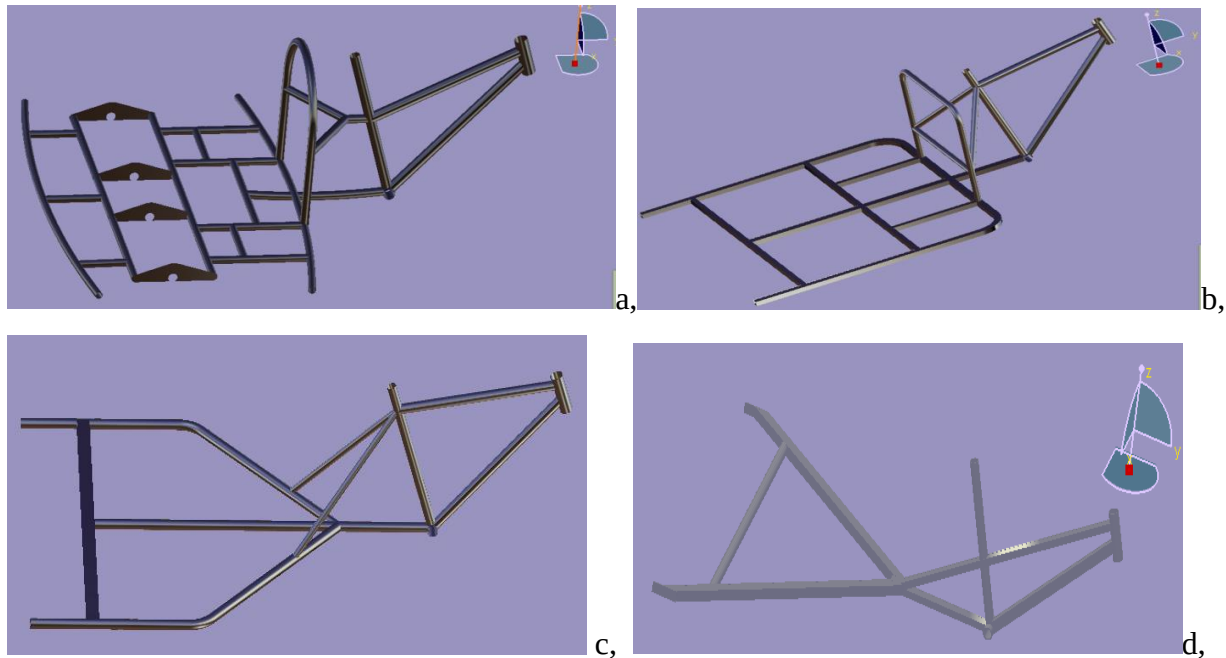


Figure 4.4 Flywheel equipped cycle rickshaw frame model concepts,

From four developed frame model concept model (d) concept has been selected based on its simple to construction, less welded parts, compatible free space at the center of passenger carriage frame for purpose of mounting kinetic energy recovery system and less mass when it compared with other three model concept. The mass of each model concept is, for (a) 22.7kg, (b) 19.3kg, (c) 15.4kg and (d) 11.7kg, this obtained from CATIA after select frame that is steel. The CATIA software automatically calculate mass frame using material density and frame volume.

4.1.4. Frame Analysis:

In this work frame analysis consider tipping point scenario which occurs under dynamic condition of the vehicle. During operation of vehicle, rider control vehicle as situation like accelerating, negotiating road curve and decelerating (braking). But these controlling situations create dynamically weight shifting condition from original C.G location and concentrated on other sides. This situation induces additional load on the frame parts. To design safe frame, we have considered the tipping point of cycle rickshaw. In order to determine the tipping point, the vehicle can be represented as a simple wireframe model. A representation of the vehicle is shown in in figure 4.5. below:

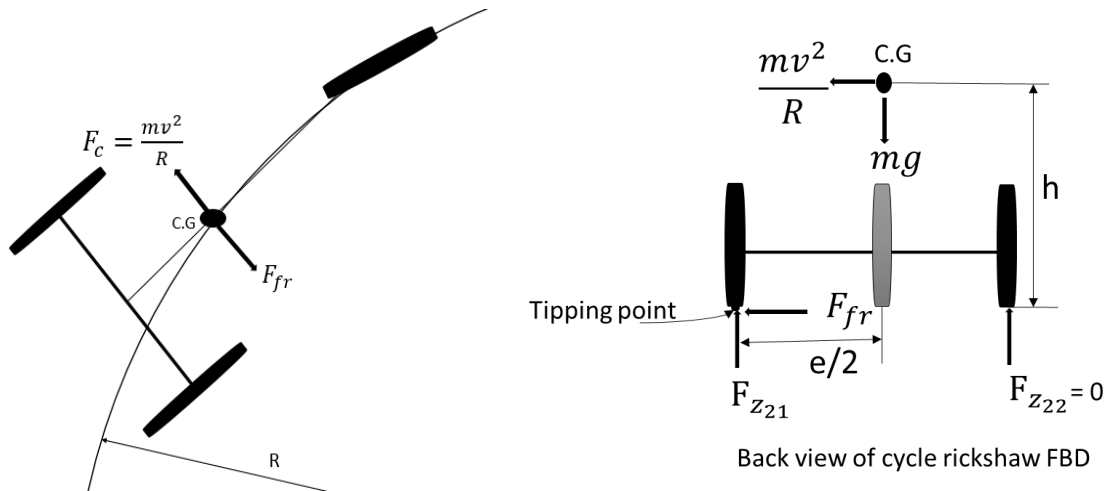


Figure 4. 5 Simplified representation of cycle rickshaw when negotiating road curve.

The physics of a turning, three-wheeled vehicle were derived as shown in Appendix B. By summing the moments due to centripetal acceleration about an imaginary tilt axis, one can arrive at the equation below to determine at what velocity and radius of turn the vehicle will tip at.

$$\frac{v^2}{Rg} = \frac{e(l-a)}{2h\sqrt{(e/2)^2 + l^2}} \dots \dots \dots (4.1)$$

Ideally, this desired value would be made as large as possible. This can be achieved by decreasing the height of the center of gravity in addition to decreasing the horizontal distance of the center of gravity from the front wheel axle.

Weight Transfer Analysis:

Another important aspect to consider in dynamic vehicle frame design is the amount of weight transfer during braking. Too much forward weight transfer can cause the vehicle to flip over the front axle. A simplified representation of the vehicle under braking deceleration is shown in figure 4. 6.

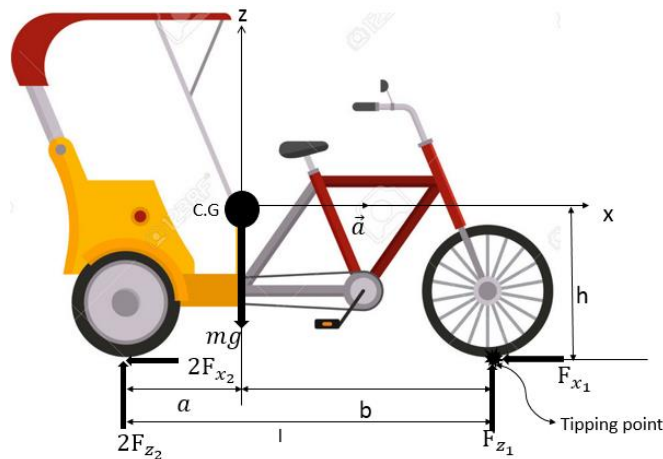


Figure 4.6 A simplified representation of the vehicle under dynamic braking deceleration. From this sketch the change in weight over the rear wheel can be solved using moments.

$$F_{z1} = mga/l + m\vec{a}h/l \quad \text{and} \quad F_{z2} = \frac{1}{2}mgb/l - \frac{1}{2}m\vec{a}h/\dots \dots \dots (4.2)$$

The derivation for this equation can be found in the Appendix B. By decreasing the height of the center of gravity and increasing the distance of the center of gravity from the front axle the weight transfer can be reduced. Notice that increasing the distance from the front axle improves braking performance but is detrimental to tilting during cornering [32].

Generally, dynamic force is about 0.6 to 0.8 times of static load for more factor of safety [33]. Hence, for both static and dynamic condition twice of static load has been taken for thesis work.

4.2. Finite Element Analysis of frame

Finite element analysis is important to do a numerical simulation in order to find out how cycle rickshaw frame react when the rider, passengers and other mechanical system subject to weight load and dynamic load. In order to perform finite element analysis, FEA modeling process requires three types of input data such as geometry, material properties, and loading. After modeling the analysis of frame is done in ANSYS 15.07 workbench, for that purpose frame model preparing the in CATIA then it is imported to ANSYS workbench, the file is imported from CATIA by file>import>IGS. To carry out the analysis three scenarios are considering like sit pedaling, stand pedaling and braking

a). Frame model

CAD model of fly wheel equipped cycle rickshaw frame has been prepared in CATIA V5 as shown below in figure 4.7. The dimensions were obtained from anthropometrics, bicycle frame size and other parts. The thickness size of each section of the frame is considered relative to steel bicycle and standard parts. The size of section used for modeling final frame is head tube (160mm length, 35mm diameter and 2.5mm thickness), for top, down, seat stay and cross tubes (square tube 30x30 (mm) and 1.5mm thickness), for chain stay and passenger carriage tubes (rectangular tube 30x35(mm) and 1.5mm thickness), seat tube (30mm diameter, 450mm length and 1mm thickness), BB shell (68mm width, 42mm outer diameter and 35 inner diameter), rear axle support plate (100x180 (mm) plate and 4 mm thickness), KERS support tube (20mm outer diameter and 1.5 mm thickness) for further and detail dimension see appendix D.

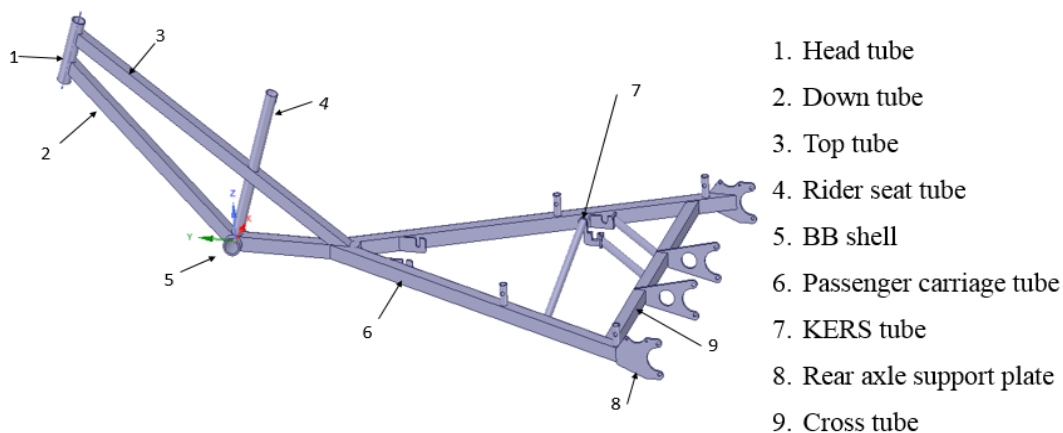


Figure 4.7 Fly wheel equipped cycle rickshaw frame

b). Frame Material

Material selection for human powered vehicle frame applications plays very important role in providing the desired strength, endurance, safety, efficient performance and reliability to the vehicle. The frame incorporates major decisions for both materials and design. Some basic requirements for the material selection of the frame are: light weight, high strength, availability, manufacturability and cost. With these design constraints in mind, material properties of common materials utilized in human powered vehicle, including Carbon Steel, Titanium and Aluminum.

Most common carbon steel is AISI 4130 is steel alloy it is a low-cost, strong, easy to manufacture, weld able and available material and is a common choice to form frames and light vehicle chassis. If optimizing performance and weight is interest, Titanium is an excellent material to incorporate into cycle frames; however, it is difficult to manufacture and is expensive, which outweighs the benefits of its rigid frame structure. Similar to Titanium, Aluminum is lightweight, expensive and is relatively easy to machine but is difficult to weld [34].

The selection of material for cycle rickshaw frame is done by detailed study of properties of material regarding strength, weight, manufacturability, cost and availability results found that steel alloy AISI 4340. But prefer to use AISI 4130 over the rest carbon steel because of its higher yield strength and high strength to weight ratio and corrosion resistant.

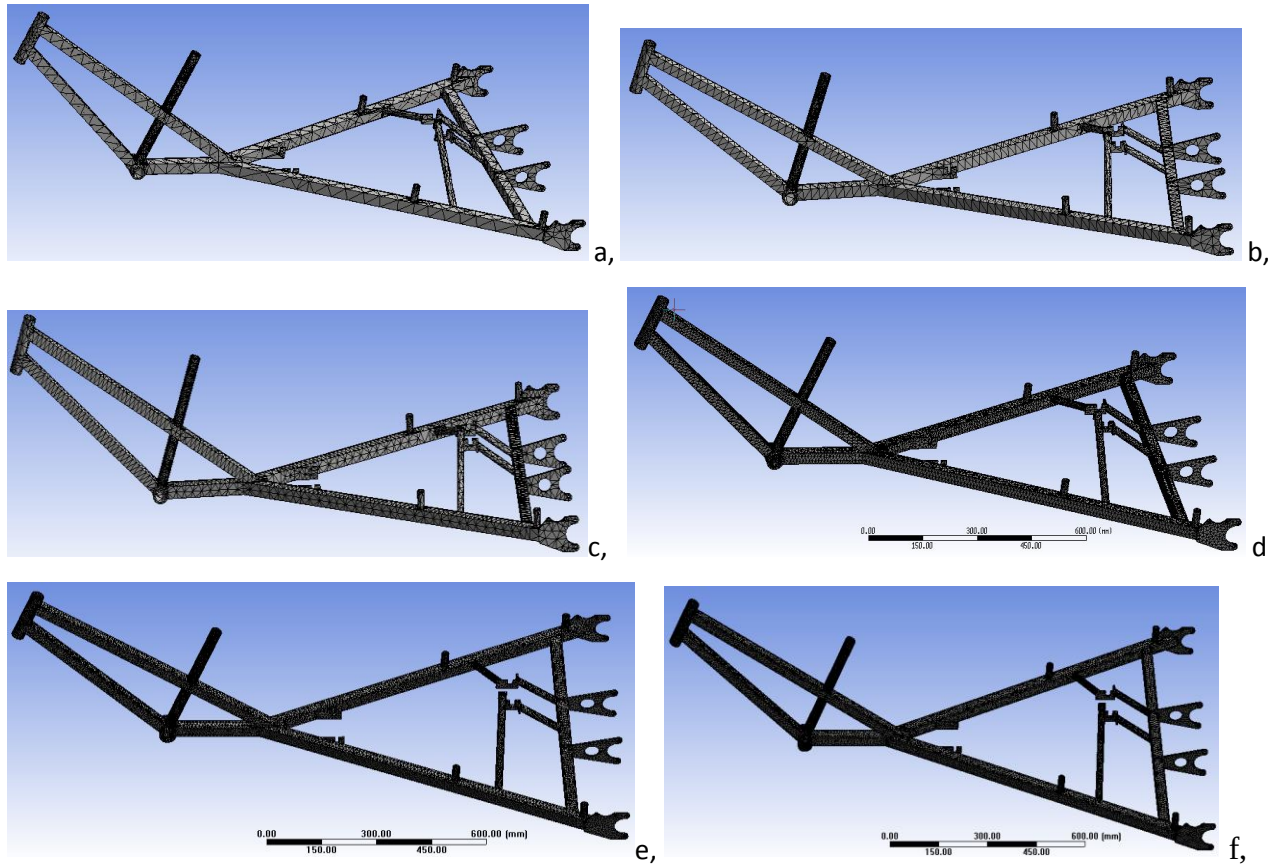
Table 4. 2 Frame Materials property and composition

	AISI 1018	AISI 1020	AISI 4130	AISI 4340.
Density (kg/m^3) $\times 10^3$	7.8	7.7	7.85	7.85
Young's Modulus (GPa)	210	210	205	200
UTS (MPa)	400	394	670	1255
YTS (MPa)	220	294	435	1165
Carbon, C	0.14 - 0.20 %	0.17 - 0.230 %	0.28 - 0.33%	0.370 - 0.430 %
Iron, Fe	98.81 - 99.26 %	99.08 - 99.53 %	97.03 –98.2%	95.19 - 96.33 %
Manganese, Mn	0.60 - 0.90 %	0.30 - 0.60 %	0.40 – 0.60%	0.600 - 0.800 %
Phosphorous, P	≤ 0.040	≤ 0.040 %	$<0.035\%$	<0.035
Sulphur, S	≤ 0.050 %	≤ 0.050 %	$<0.04\%$	<0.04
Chromium, Cr	0%	0%	0.80 – 1.10%	0.700- 0.900%
Molybdenum, Mo	0%	0%	0.15 – 0.25%	0.200 - 0.300%
Silicon, Si	0%	0%	0.15 – 0.30%	0.150 - 0.300%

c). Meshed frame model

Meshing is one of the most important parts of the finite element analysis (FEA). Fine, medium and coarse elements have been studied in the meshing methods. While effect of the element size has also been studied, three different element sizes have been considered to their effect on the number of nodes and number of elements.

Meshing for this cycle rickshaw frame model was done using the automated ANSYS meshing refinement feature. That with fine relevance center, fine smoothing and with 3mm element size, these parameters controlled on ANSYS global mesh control features. Finally, the frame meshed to course 30759 and 15427, medium 36477 and 18235, fine 43861 and 22091, 10mm element mesh 140998 and 70503, 5mm element mesh 179611 and 89645, 4mm element mesh 245719 and 123146, and 3mm element mesh 430832 nodes and 217457 nodes and elements numbers respectively. The meshed model is shown in figure 4.9.



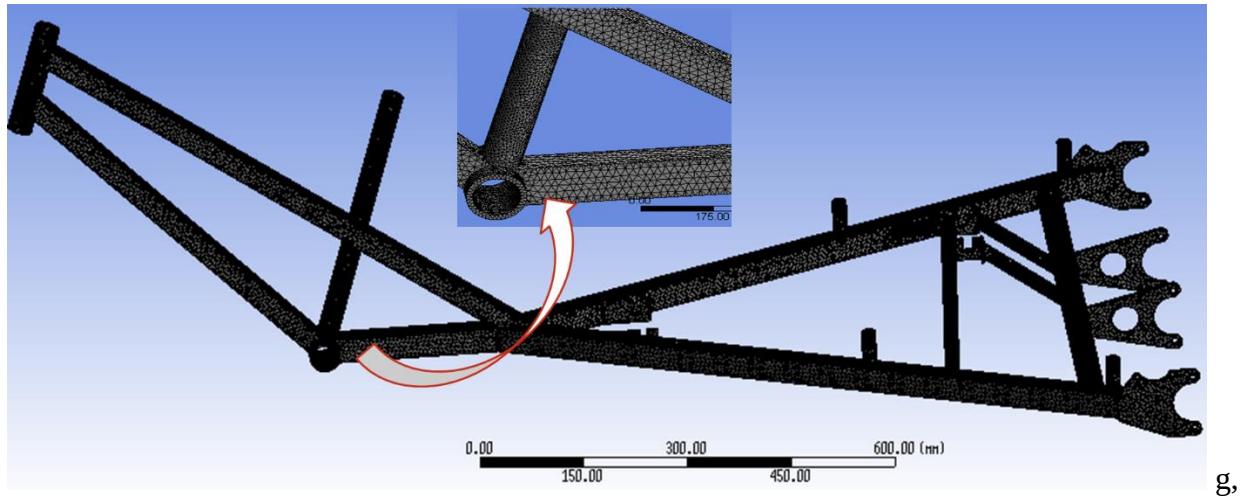


Figure 4.8 a), Course b), medium, c), fine, d), 10mm element size, e) 5 mm element size f) 4 mm element size and g) 3mm element size meshed new cycle rickshaw frame model

d). Boundary condition or loading

Finally, the externally acting loads must be specified in terms of magnitude, direction, and point of application on the frame. Loading information is derived from experiment and analytical deduction. This information is then used to specify forces acting at endpoints (nodes) of key elements such as carriage tubes, bottom bracket bearing, head tube, seat post, and KERS support tube and rear axle support plates. And the model fixed in all degrees of freedom near the supports; fixed support are applied at bottom side of head tube and rear wheel axle support plates. It shown in figure 4.9.

Loading

Correct loading information is absolutely vital in producing reliable results. In this study, a major effort was made to define a realistic and self-consistent set of load conditions. The loading conditions are static start up and steady state pedaling. For those scenario analysis simulation done in similar way except rider loading.

I. Static start up

An 70kg rider is applied maximum effort to bottom bracket to accelerate cycle rickshaw with 160kg passenger on seat mount ,8kg flywheel on mount and other mechanical components from stand position. For this scenario the rider is out of the saddle. The cycle rickshaw frame is in vertical equilibrium with head tube (at A) and rear drive axle assembling plate hole were fixed (at A but for

multi face selection ANSYS work bench label only one point). We find that loads generated in this case are very similar to those encountered in hill climbing.

The following figure 4.9. shows that the loading and boundary condition on a cycle rickshaw frame, this load is applied according to the design of frame, Also the boundary condition is applied at fixed position of a model which is consider as a supports, the load is applied considering weight shift during operating condition of cycle rickshaw it taken twice of static load on each load application point.

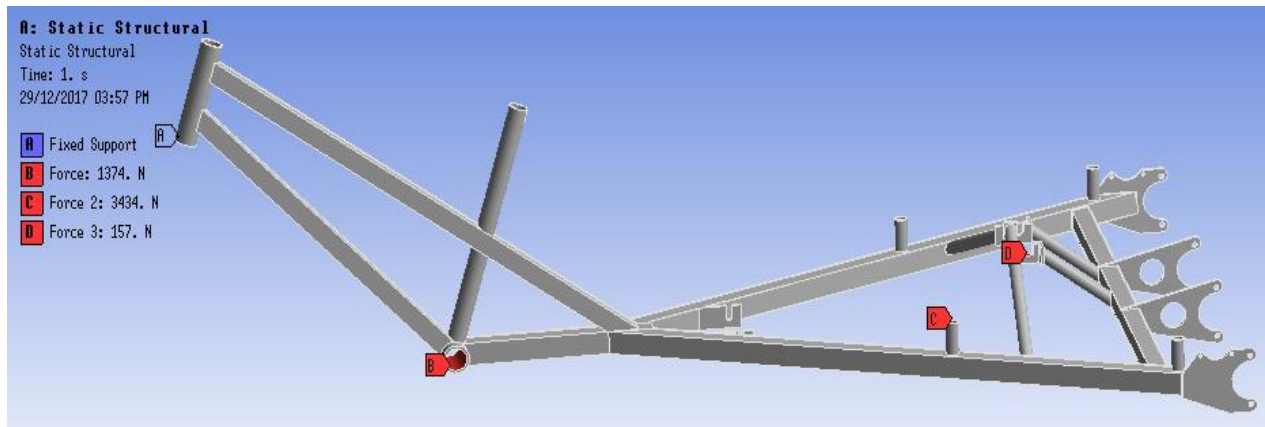


Figure 4.9 Loading and boundary condition for static start up

Obtained results after applying load for static start up condition

The following Figure 4.10 shows the result of (a) total deflection, (b) equivalent (von-mises) stress and (c) factor of safety, for static start up condition in the frame when the load of 1374N (at B), 3434N (at C) and 157N (at D) is applied for material AISI 4130. The red color shows the maximum simulation value of those parameters and blue color shows minimum simulation value of those parameters generated. But for factor of safety blue color shows maximum value and red color shows minimum value.

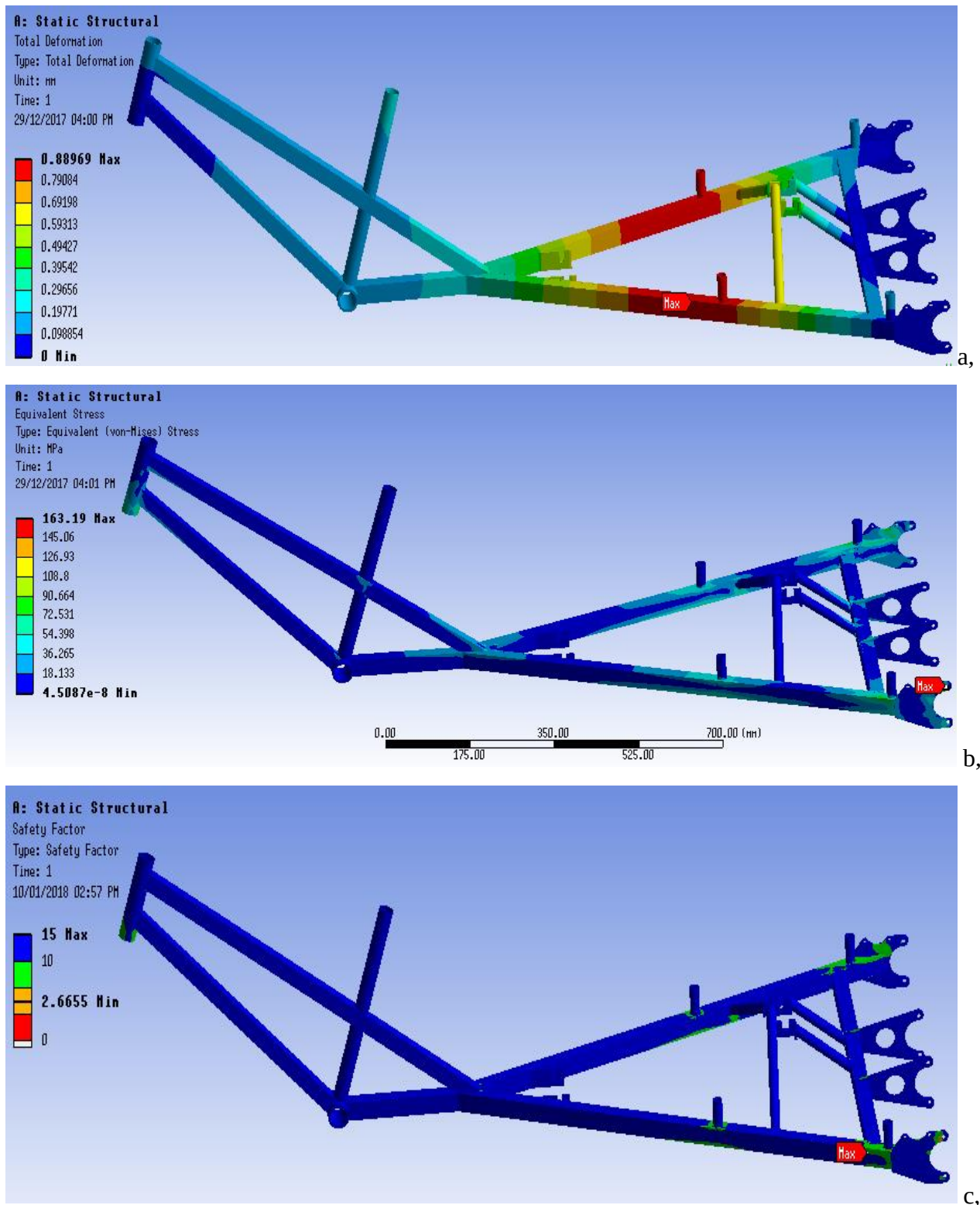


Figure 4.10. Static start up simulation result, (a) Total deformation, (b) Von-Mises Stress, and (c) Factor of safety

II. Steady state pedaling

This presents the most complex array of applied loads for any case under investigation. A 70kg cyclist, seated on a 60kg cycle rickshaw, is generating .54 horsepower (400 watts) at a cadence of 100 rpm. The front sprocket has 53T, the rear has 17T. In this study, the pedal load analysis was applied to the "worst case" situation: when the cranks are positioned at the maximum loading angle, approximately parallel with the ground.

From total load of rider 60% load (824N) considered as applied on seat tube and 40% load (550N) applied on pedal or BB bearing. But other loads not changed it is the same as case one. Those all load applied to negative Z axis.

Figure 4.11 shows load and boundary application on frame for steady state pedaling condition

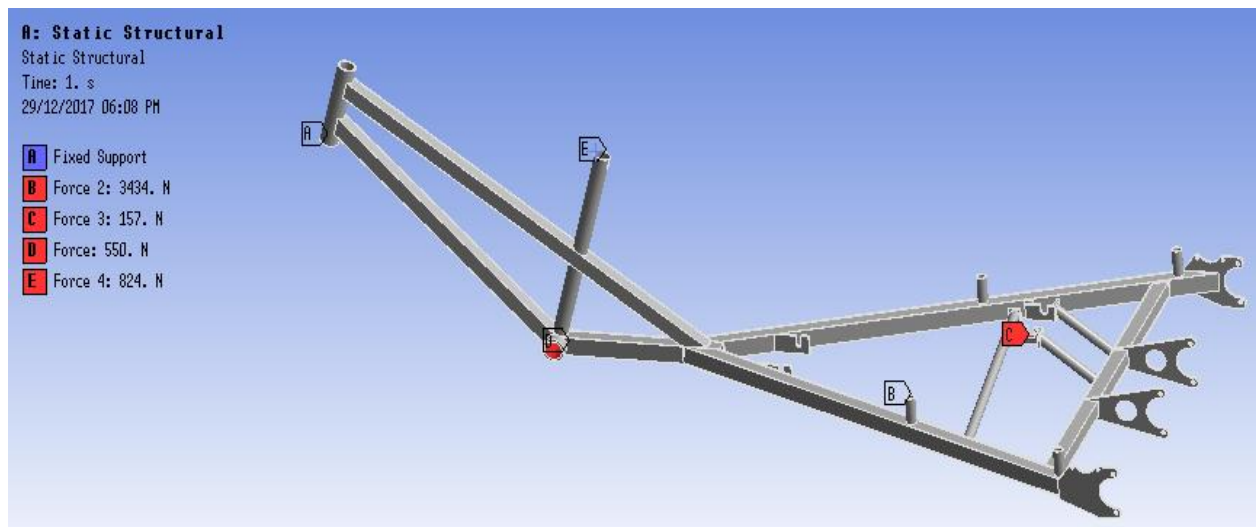


Figure 4.11. Steady state pedaling loading and boundary condition

Obtaining results after applying load for steady pedaling condition

The following Figure 4.12 shows the result of (a) total deflection, (b) equivalent (von-mises) stress and (c) factor safety for steady state pedaling condition in the frame when the load of 824N (, 550N, 3434N and 157N is applied for material AISI 4130. The red color shows the Maximum simulation value of those parameters and blue color shows minimum simulation value of those parameters generated. But for factor of safety blue color shows maximum value and red color shows minimum value.

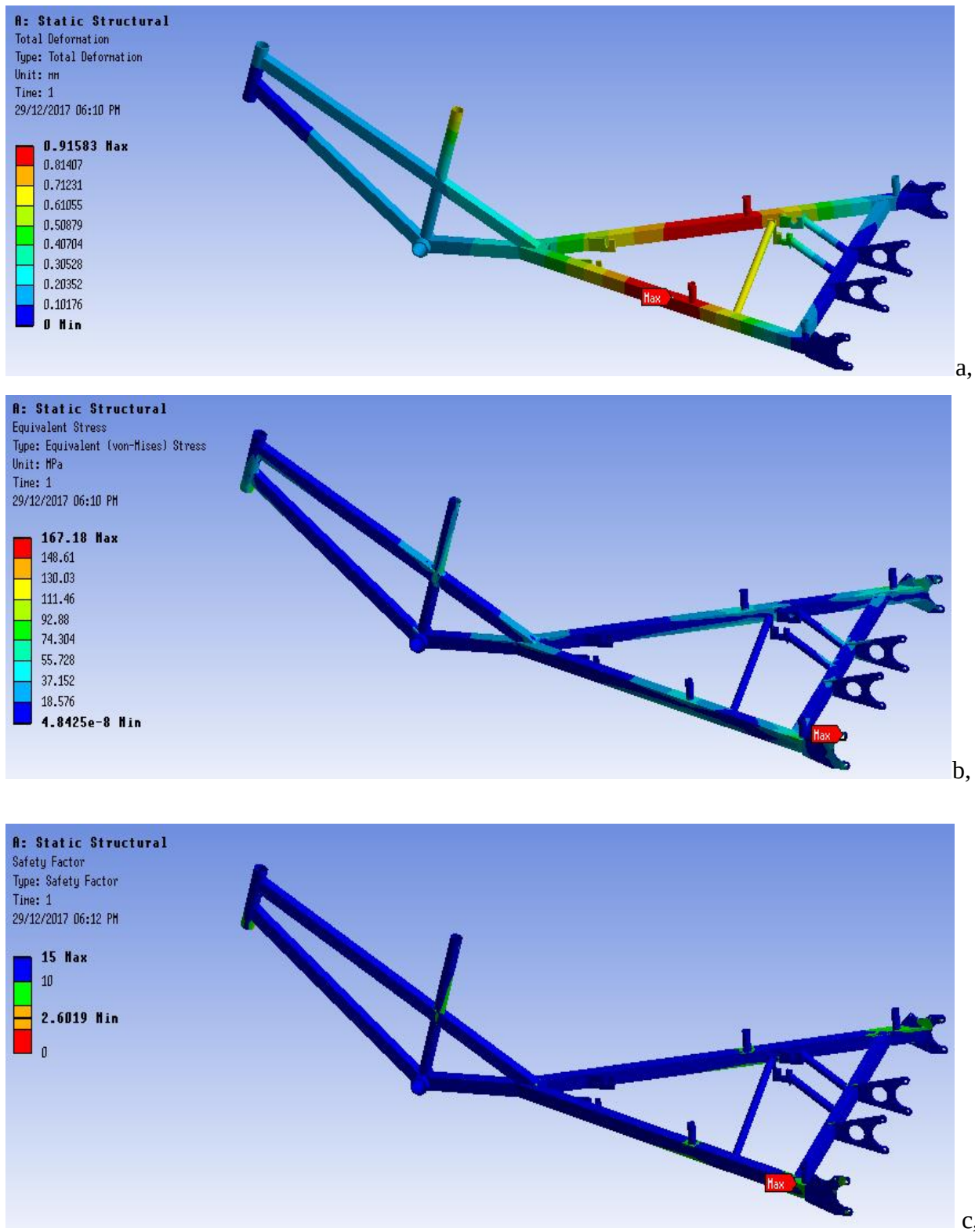


Figure 4.12 Stand pedaling simulation result, (a) Total deformation, (b) Equivalent Stress (Von-Mises Stress), and (c) Factor of safety

4.3. Passenger seat design

4.3.1. Fit parameters

The principle that the seat should fit the sitter is the most universally employed concept in seating ergonomics. If a chair is to be used by only one sitter, careful measurements of that person's body will yield appropriate dimensional specifications for the seat. However, in the cycle rickshaw passenger seat, where a single seat must accommodate a large percentage of the population, knowledge of population anthropometry is required.

The constraints on Fit parameter design values are usually imposed by the desire to accommodate a sufficient range of the population on one anthropometric measure. A widely used design criterion is that the seat should accommodate the members of the population who lie between the 5th-percentile and 95th-percentile people values on some anthropometric measure of interest.

Table 4. 3 Seat anthropometric measurement

	Body part	Measurement (mm)
7	Shoulder height	623
15	Hip breath	360
16	Lower leg length	444
23	Seat depth	499

For ergonomic design, the popliteal height is the distance from the underside of the foot to the underside of the thigh at the knees. It is sometimes called the "Lower leg length". For men, the median popliteal height is 444mm. The popliteal height, after adjusting for heels, clothing and other issues is used to determine the height of the chair seat. Mass produced chairs are typically 350 to 432mm high. The buttock popliteal length is the horizontal distance from the back most part of the buttocks to the back of the lower leg. This anthropometric measurement is used to determine the seat depth. Mass produced chairs are typically 360 to 432mm deep take 400mm. Backrest height requirements are affected by geometric constraints imposed by dealing with head restraints for protection in rear impacts. Within these constraints, there is only a small range of backrest heights that can be specified. The backrest should extend 410 to 550 mm [35]. Based on ergonomic data the passenger seat arrangement is shown in figure 4.2.

4.3.2. CATIA modeling of the passenger seat concept

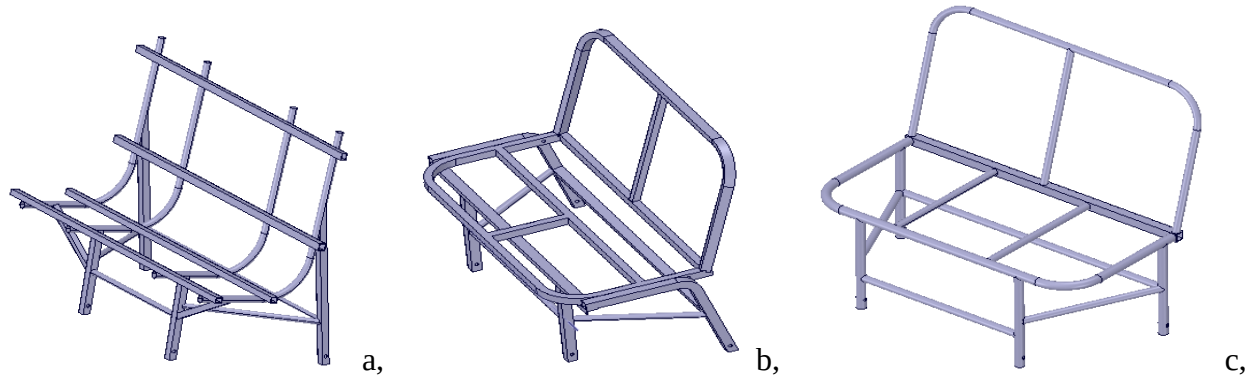


Figure 4.13. Different passenger seat frame model concepts.

The concept c has been selected for this work, the main reason is simplicity and lightness its mass is 7.7kg when it compared with concept a (15kg) and b (13.47kg). These mass values obtained from CATIA software when material employed it calculate automatically.

4.4. Finite element analysis of passenger seat frame

A cycle rickshaw passenger seat frame was designed to carry the three medium (95% population) or two large (95% population) size passengers and their loads. For carrying out the finite element analysis of passenger seat frame has been modeled according to the anthropometric ergonomic design dimension using CAD software CATIA V5. To generate seat structure 2mm thick box section of (25x20) mm, 20 mm (for support tube) and 25mm (leg and seat tube) circular tube with 2 mm thickness was used.

a). Material for seat frame

The lower part of the passenger seat must be strong as it bears all the weight of the passenger. The weight of material should be considered for better performance of vehicle. Available material is cast iron, carbon steel, steel alloy and aluminum. Further discussion already done on frame section, on the bases of cost, manufacturability and strength AISI 1018 has been selected see table 4.3 for mechanical properties.

b) Meshed seat structure model

To mesh passenger seat structure model global mesh control was used with 3mm element size in order to get more accurate results. The mesh done on ANSYS with 360456 nodes and 180316 elements. The figure 4.14 shows meshed model of passenger seat frame.

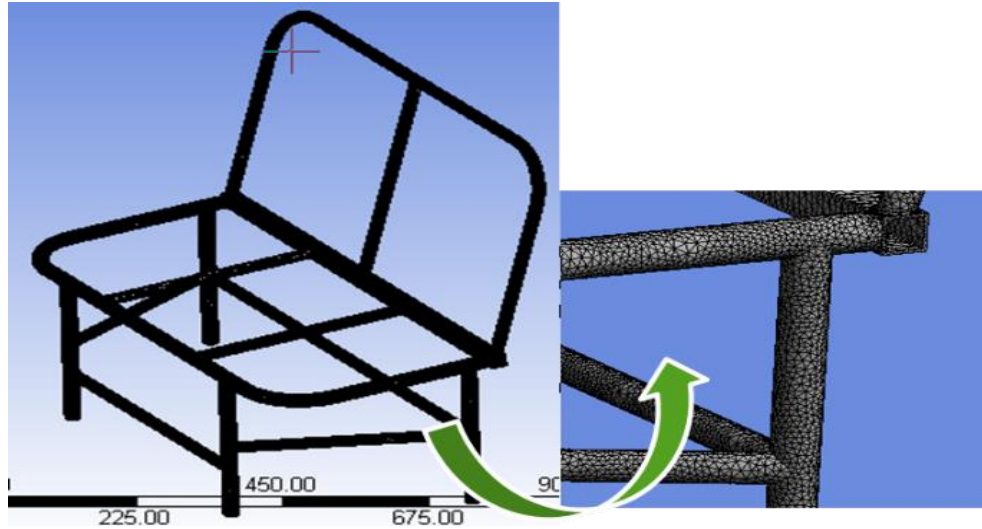


Figure 4.14. Meshed passenger seat structure model

c). Boundary and Loading condition

To perform FEA analysis, the seat frame fixed at leg bottom face and mounting bolt hole. The total load applied on passenger seat is 160kg or 1570N is total weight of the passenger which applied on the top seat tube faces (or at point B). Also there is a load which applied on the back rest due to slope of road. That load is mass time acceleration of gravity times road slope angle. The maximum slope for motor vehicle road 5° [40]. So load calculates as $160kg \times 9.81m/m^2 \times \sin 5^{\circ} = 137N$. The figure 4.15 shows boundary and loading condition of passenger seat frame.

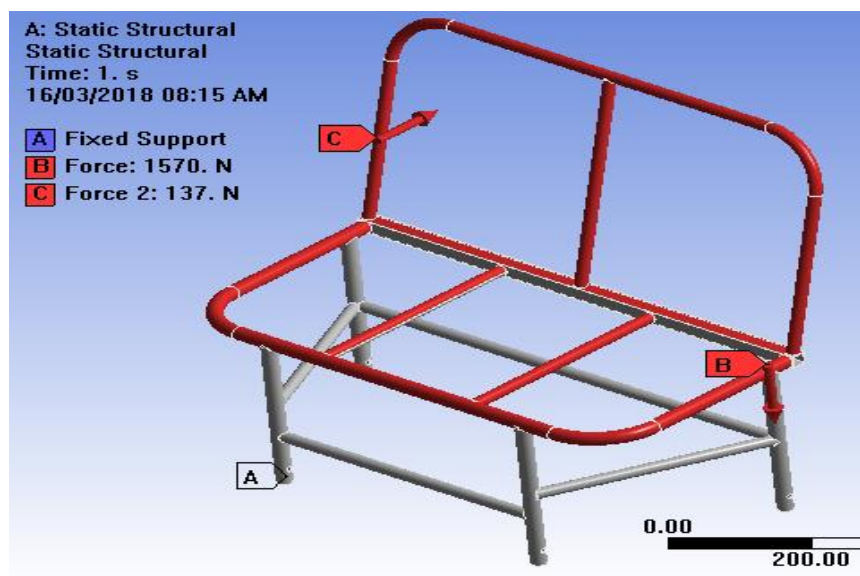


Figure 4.15. Passenger seat frame loading and Boundary condition

Passenger seat frame analysis result

the following figures 4.16 shows the result of (a) total deflection, (b) equivalent (von-mises) stress and (c) factor safety in the frame when the load of 1570N is applied on passenger seat frame. The red color shows the Maximum simulation value of those parameters and blue color shows minimum simulation value of those parameters generated. But for factor of safety blue color shows maximum value and red color shows minimum value.

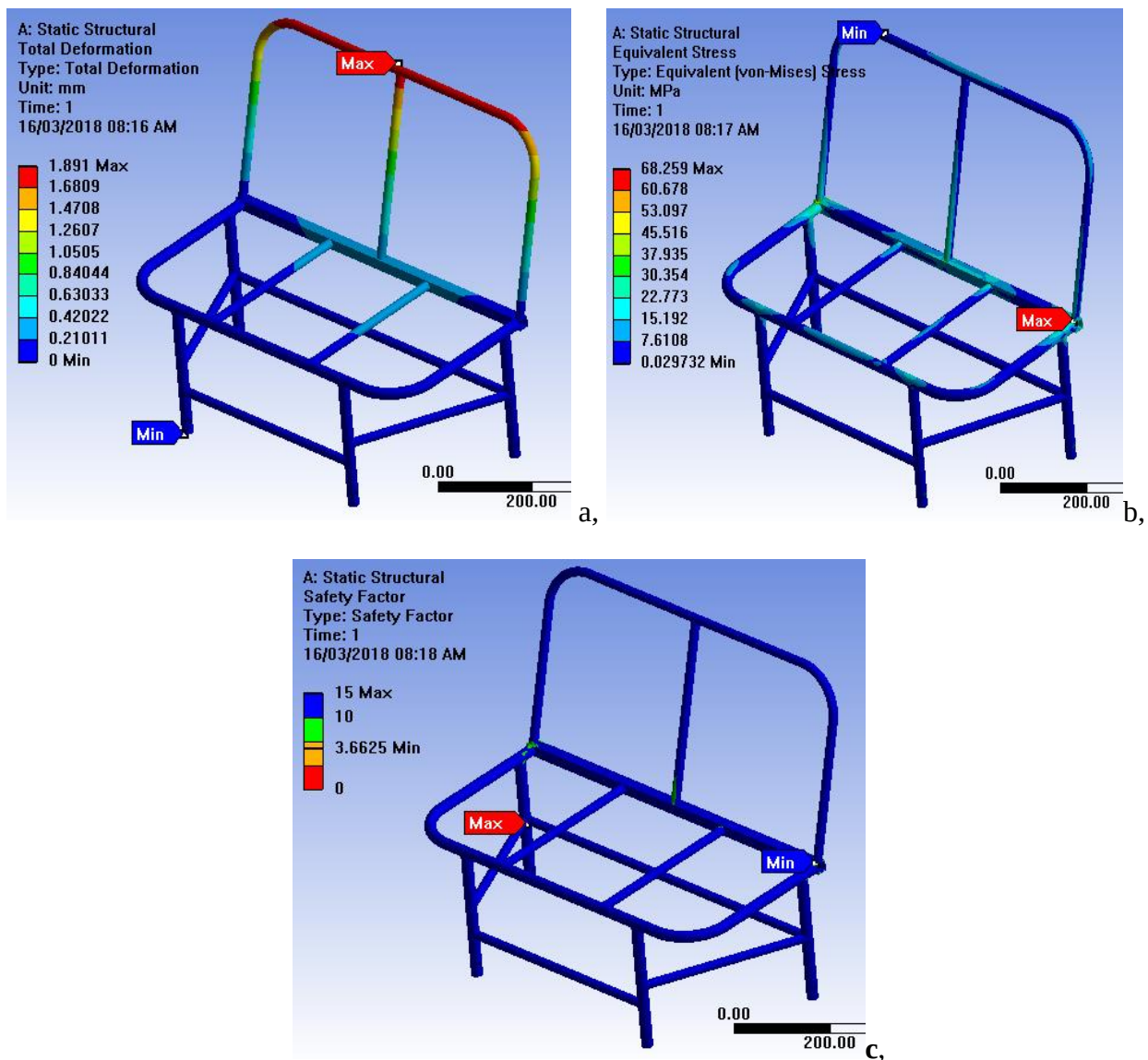


Figure 4.16. Passenger seat (a) Total deformation, (b) Von-Mises Stress, and (c) Factor of safety

4.5. Steering system of cycle rickshaw

The function of the steering system was to provide reliable and responsive direction control of the cycle rickshaw. The new cycle rickshaw steering is the same as bicycle. The only difference is bicycle steering system design to one person, but new cycle rickshaw designed to carry three person including rider. That means new cycle rickshaw steering fork require modification to withstand three persons load without fail. This achieved by modifying existing fork or selecting stronger material. The steering system of a new cycle rickshaw consists of the fork legs, headset, stem, handlebars, bottom attachment and handlebar grips. To meet the goals of practicality and simplicity, the steering was designed to be constructed by a semi-skilled laborer.

4.5.1. Design requirements

It is required that the user be able to maintain full control of the cycle rickshaw under standard operating conditions. This means that the cycle rickshaw will turn freely, move forward in a straight line, and provide responsive handling at all operating speeds. The steering mechanism must be strong and easily accessible to the user at all times. The turning circle of the cycle rickshaw must allow for riding on roadways without issue. The steering and frame design must interact such that the cycle rickshaw will not overbalance if the user makes a turn, or is forced to avoid an obstruction at speed. Furthermore, the steering had to be stable in a straight line with no steering wobble. The vehicle also had to turn smoothly and in control at high speeds, but was also be able to make tight turns at low speed. It wanted the steering to be intuitive for the driver, and not interfere with the other subsystems in the design in a negative manner.

Table 4. 4 Steering system specifications

Steering ratio	1:1
Turning angle	$\leq 90^{\circ}$
Rake offset distance	40mm

4.5.2. Steering fork design

There are three types of a forks design used in bicycle or delta configuration trike steering system such rigid fork design, telescopic fork design and triple fork design.

Rigid forks are the simplest forks on the market. They are called rigid because they are on solid unit and contain no moving pieces. Due to their design rigid forks are simplest and least expensive

kind of that can be produced. The rigid fork commonly produced from cromoly steel. This is considered a high strength and somewhat exotic material, but due to the simple design of the rigid fork it is inexpensive. While the rigid fork is the most widely produced bicycle fork it does not qualify as a competitor for this project due to steerer section and its threaded part easily damaged and it need entire frequent replacement.



Figure 4. 17 Rigid Fork [36].

Telescoping fork design, a trip to any bicycle or motorcycle shop will show that the telescoping suspension fork is the only commercially available suspension fork on the market. This is because they are providing good performance for rider's right out of the box. However, there are some major downfalls of the telescoping fork design. For instance, damaged fork tubes will generally disrupt the suspension function of the fork. One damaged component may also require that the entire fork be replaced. Also it is complex to construct because its integrated suspension system makes more expensive and add weight, and more suitable for high speed bike or motor bike and special purpose bike that is mountain bicycle. While **Triple fork design** is the constructed from three tube and two support brace it provides rigid and strong system to hold fork legs. It simple to construct and allow large trail or rake without bending fork legs. The offset distance for rake done the on the support brace and fork leg dropout. It also simple to maintenance which means if one component damaged we not throw entire system it need to replaced only damaged component. Based on above criteria triple fork selected for this thesis work [36].

4.5.3. Steering geometry

Cycle rickshaw or bicycle steering geometry deals with how to head angles, rake, trail and tube lengths and how those variables will affect the handling and riding qualities of a bike. Today it knows that the careful balance of the head tube angle and the fork rake will give a measurement called trail [37].

I. Fork rake and Trail

The fork rake (or offset) is the perpendicular distance from the steering axis to the center of the front wheel. Fork rake must be carefully matched to the head tube angle to produce a bike with desirable handling characteristics. Generally, forks with more rake are considered better for faster steering because of their ability to absorb more road shock. Forks with less rake are considered better for slower steering because of the more sensitive feel of the road they afford. For bicycle or tricycle, the rake distance value ranges between 40 to 60mm, while trail is the *horizontal* distance from where the front wheel touches the ground to where the steering axis intersects the ground. The comfort range of trail is 50 to 70mm. Beyond these limits in either direction would be considered less desirable. The diagram which follows shows in more detail how to find trail: I take 40mm rake size for this thesis work, if we take beyond this value trail value is out of comfort ranges even with larger size radius of wheel [38].

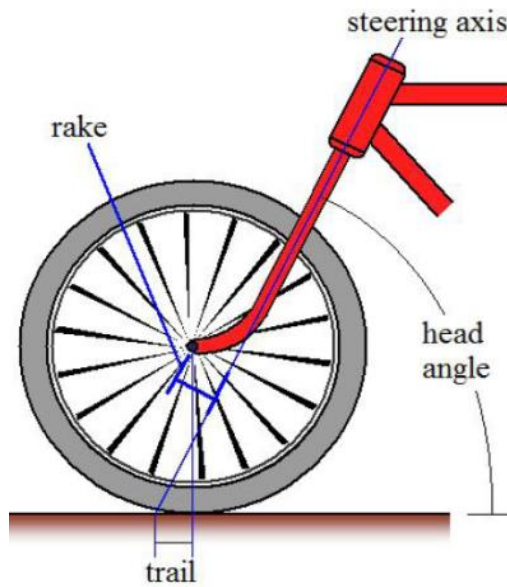


Figure 4.18 Cycle rickshaw or bicycle steering geometry [38]

The following formulae will give the trail if rake, head angle, and wheel radius are known:

$$t = \frac{R \cos \alpha - r}{\sin \alpha} \dots \dots \dots (4.3)$$

$$r = R \cos (\alpha) - t \sin \alpha \dots \dots \dots (4.4)$$

Where: - R is wheel radius, r is rake, t is trail, α is head angle.

II. Size of fork leg

Length of fork leg determine from head tube angle and stack length of frame that because it forms right angle, us Pythagoras theorem.

$$L_F = \frac{\text{stack length}}{\sin 73^0} \dots\dots\dots 4.5$$

$$= \frac{520\text{mm}}{\sin 73^0} = 544\text{mm}.$$

Where, L_F is fork length.

But total length of the fork is greater than 544mm because head set space and fork brace thickness must be included. And, circular section with 25mm has been chosen for the fork leg.

III. Brace and it geometry

Brace is component of cycle rickshaw steering system which supports two fork legs and steered tube to assemble with main frame head tube. There two brace one in bottom end of the head tube and the other is at the top end of head tube.

The size of the brace determined based on front wheel hub width and steering geometry of cycle rickshaw that is rake distance or offset distance.

IV. Steerer tube

The steer tub size determined from standard head set size and head tube length plus head stem mount size.

4.6. Finite element analysis of front fork of cycle rickshaw.

a). Assembly model cycle rickshaw front fork

From a functionality point of view, the front fork assembly of cycle rickshaw front bicycles are quite similar. Figure 4.19. illustrates a typical fork assembly. The assembly consists of a steerer tube, the fork legs tube, two fork brace and a pair of front axle dropout attachments. The steerer is a circular tube section with precision machined areas to provide the bearing seats for the upper and lower headset bearings. The steerer tube and the fork legs held in place on brace by a bolt and nut.

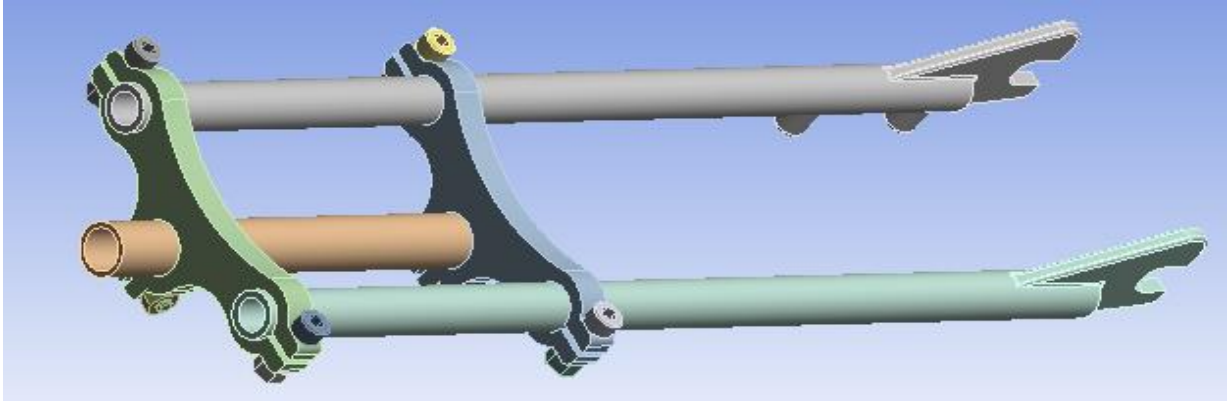


Figure 4.19 Front link assembled model

b). Material of front fork

To minimize the weight of front, link the same material as frame is used that AISI 4130 (see mechanical properties of material in table 4.3) for fork legs, steerer tube and brace. Strong alloy bolt M8 selected of minimum yield tensile strength 1100MPa and ultimate tensile strength 1200MPa. The reason to select such strong bolt is factor of safety.

c). Meshed model of front link

To mesh front link ANSYS global mesh control, refinement and hex dominant mesh was used in order to get more accurate results. Front link mesh in 249198 nodes and 133079 elements. The figure 4.21 shows meshed model of front link.

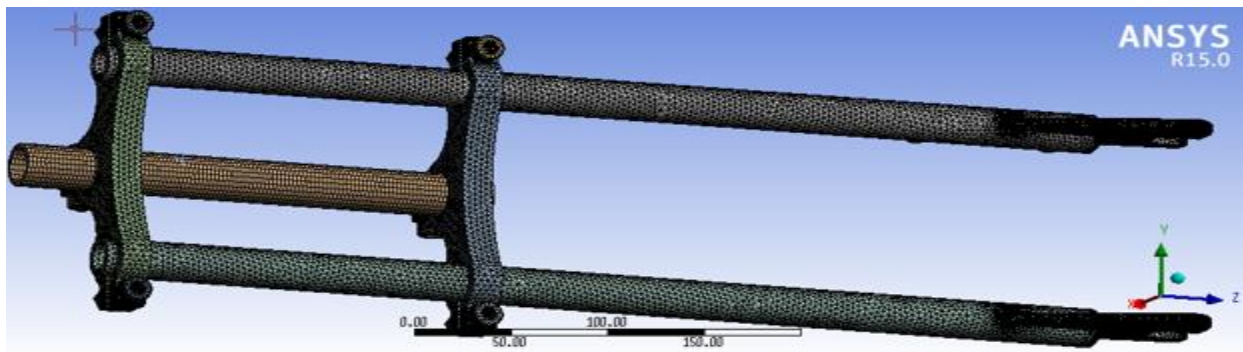


Figure 4.20 Meshed front link model.

d). Boundary condition and loading

The steerer tube threaten as support because brake force acting on bottom attachment of frock leg. The critical loading scenario for the front fork structure is a front hard braking. This loading cases

introduce a horizontal load component applied at the front axle attachment. This introduces a maximum bending load in the fork legs area of the structure. There are other loading scenarios such as frontal impact, vertical impact, side-rim, handle-bar torsion, etc., however, panic braking cases generally control the design [39].

These impact loads do not occur frequently enough to be considered fatigue loads but, nevertheless, are a threat to the life of the fork structure. The fork structure must be designed with adequate load carrying capacity to handle these loads without yielding or buckling the fork structure. For this project hard braking scenario was considered which will cause dive due to all weight transferred to front wheel and extra force creation. Under hard braking there are two separate set of forces to compress the forks and cause dive. The first is due to load transfer F_t we calculate a component F_{s1} in line with fork legs:

$$F_{s1} = F_t \cos \theta \dots\dots\dots 4.5$$

The second is due to the braking force F_b , in this case the fork component F_{s2} is:

$$F_{s2} = F_b \sin \theta \dots\dots\dots 4.6$$

From equation 4.3 and 4.4 the total extra fork legs force F_s is:

$$F_s = F_{s1} + F_{s2} = F_t \cos \theta + F_b \sin \theta \dots\dots\dots 4.7$$

For a sport bike F_t typically taken half of F_b and rake angle is complement of head tube angle (73°) that is 17° , the extra force become:

$$F_s = F_b (0.5 \cos 17^\circ + \sin 17^\circ) = 0.77 F_b \dots\dots\dots 4.8$$

This the extra vertical force from load transfer cause bending fork legs.

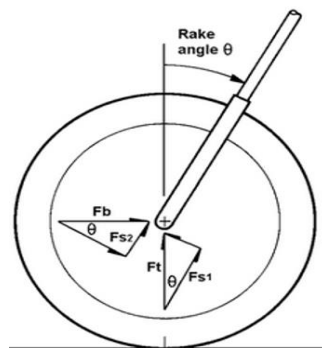


Figure 4.21 Free body diagram of front wheel under front brake action

Maximum brake force calculation

The maximum brake force calculates as total weight of vehicle times maximum deceleration of the vehicle.

Maximum deceleration

The deceleration or braking cannot be increased infinitely. A limit is set by theoretically achievable deceleration by acceleration due to gravity (g) at $g = 9.81 \text{ m/s}^2$. Another limit is the friction value between the tire and the road surface which is expressed by the adhesion coefficient (μ). Usually (there are expectations) deceleration is greatest when the wheels do not lock during the braking process, that is they just continue turn. Any increase in the braking force would thus not automatically achieve greater braking performance but would instead result in locking of the wheels and thus a loss of the wheels and a loss of steer ability, and cause the vehicle to swerve.

Depending on the road surface and its condition at the time (dry, wet), a certain adhesion coefficient is obtained which determines the maximum achievable braking deceleration. It is computed as follows:

Max. Attainable deceleration = gravity x adhesion coefficient

$$\tilde{a}_{max.} \left(\frac{\text{m}}{\text{s}^2} \right) = g \times \mu [40] \dots \dots \dots 4.9$$

Table 4. 5 Adhesion coefficient on different road conditions [40]

Road surface	Dry μ	Wet μ	
		clean	Slippery
Concrete, granite	0.7	0.6	0.4
Asphalt	0.5	0.4	0.3
Snow	0.2	0.1	0.1

Dry asphalt adhesion coefficient is considered for this project

$$\tilde{a}_{max.} = 9.81 \frac{\text{m}}{\text{s}^2} \times 0.5 = 4.9 \text{ m/s}^2$$

The braking force is mass of vehicle (350kg) times max. deceleration

$$F_b = 320 \text{ kg} \times 4.9 \text{ m/s}^2 = 1568 \text{ N}$$

Extra transferred load is 0.77 time of brake force that is 1207N.

These both forces applied to the fork attachment the static analysis.

Initial tension force in a bolt

Bolts are designed on the basis of direct tensile stress with a large factor of safety in order to account for the indeterminate stresses. The initial tension in a bolt, based on experiments, may be found by the relation [41].

$$T_b = 1420 d N \dots\dots\dots 4.10$$

Where:- T_b =Initial tension in a bolt, and d = Nominal diameter of bolt, in mm. $d= 8\text{mm}$.

$T_b = 1420 \times 8 = 11360\text{N}$ this tension force used to tighten each bolt to assemble front fork.

The following picture shows loading and boundary condition of the fort fork.

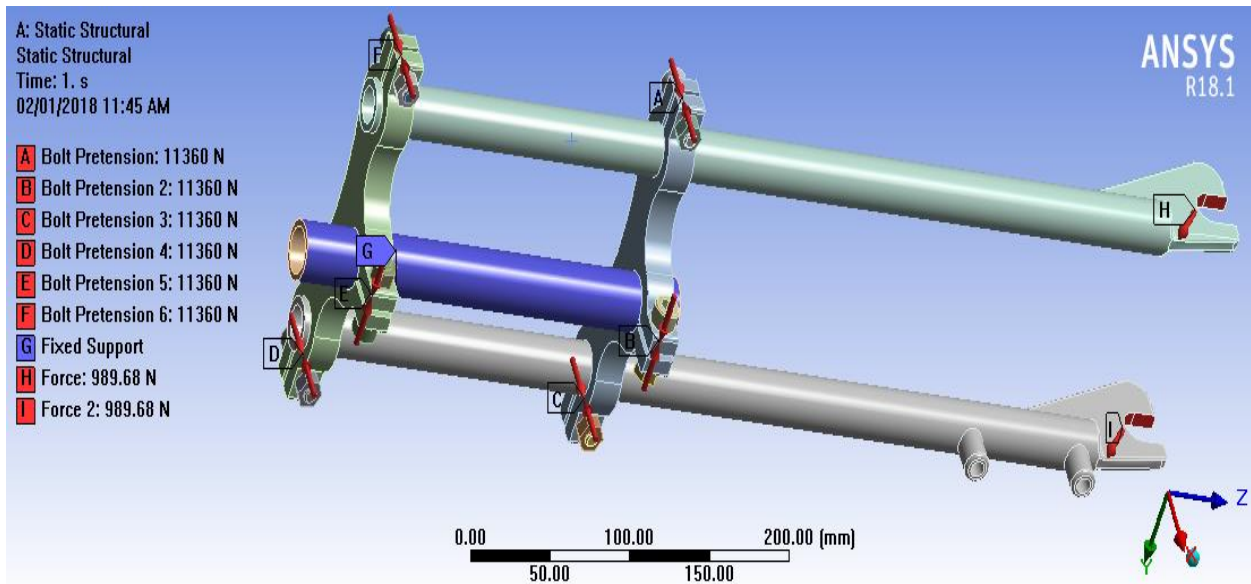
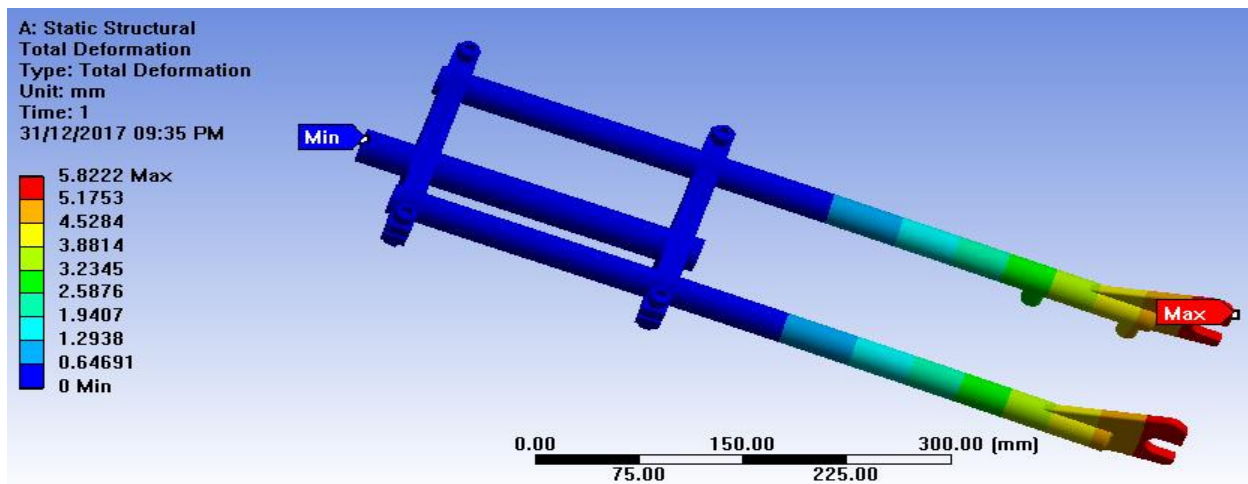


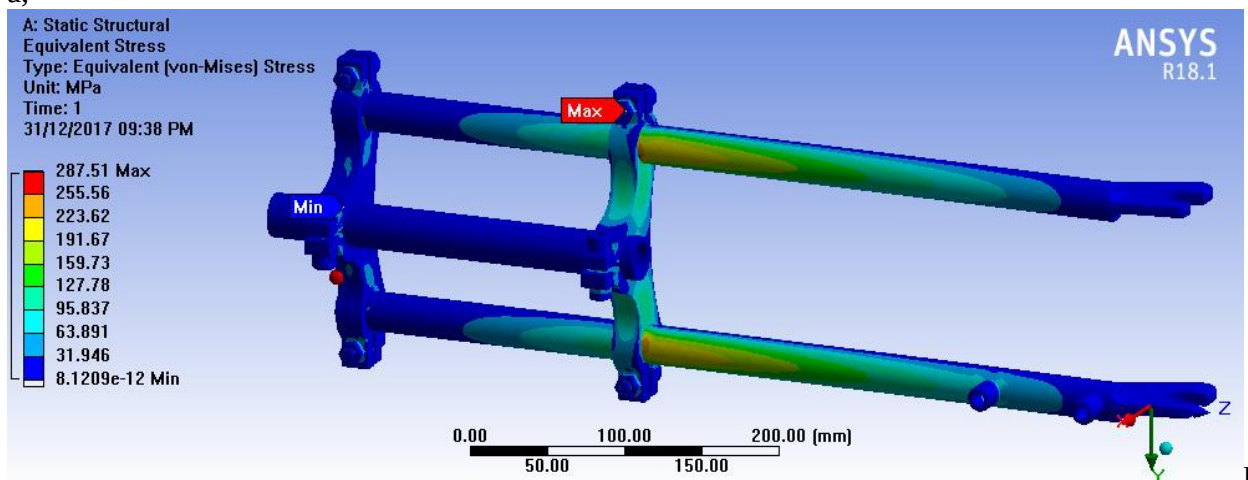
Figure 4.2 Font fork loading and boundary condition

Front fork analysis result

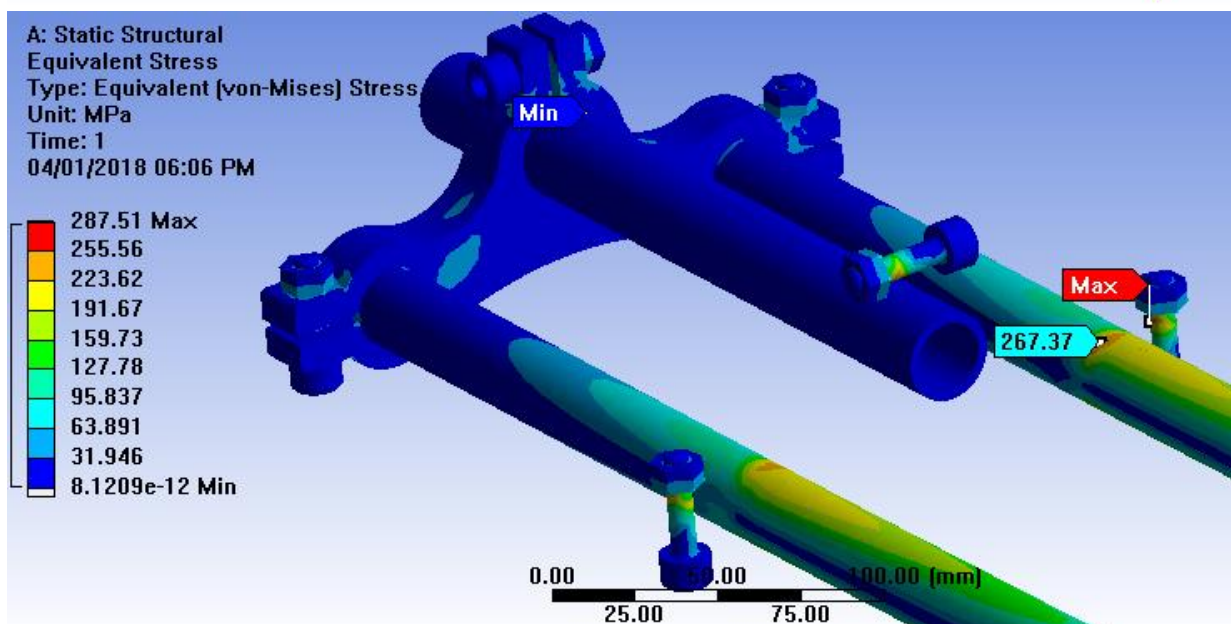
the following figures 4.24 shows the result of (a) total deflection, (b) von-mises stress and (c) factor safety in the front fork when the load of 11360N on each bolt, 1568N X-axis and 1207N in -z-axis on bottom attachment of fork leg applied. The red color shows the Maximum simulation value of those parameters and blue color shows minimum simulation value of those parameters generated. But for factor of safety blue color shows maximum value and red color shows minimum value

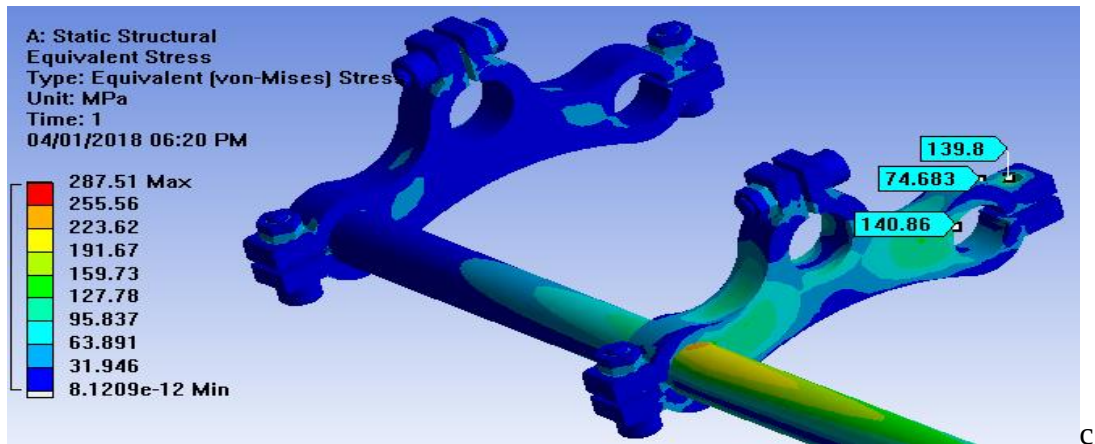


a,

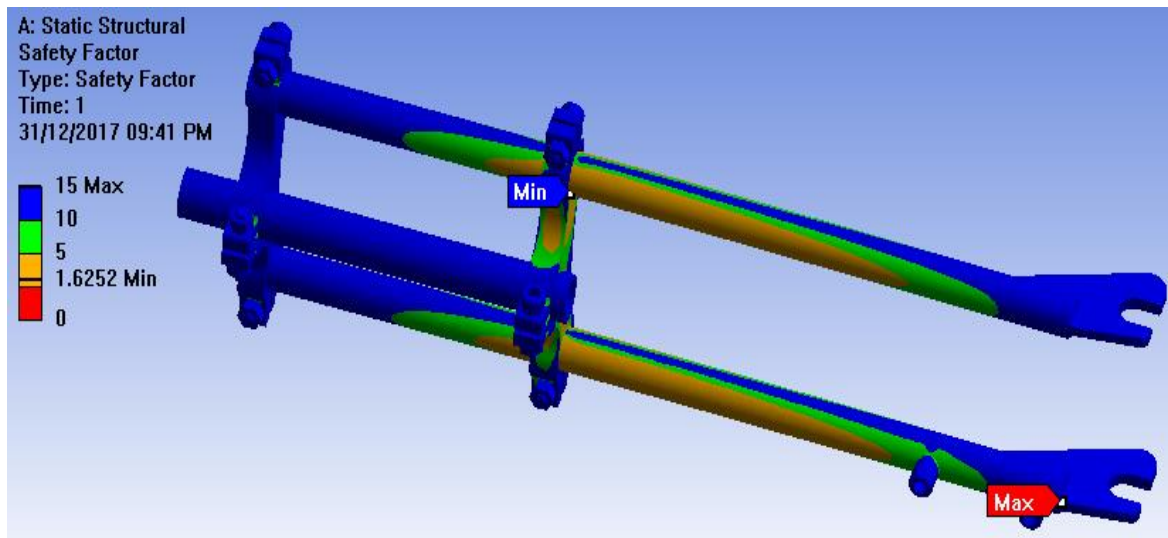


b,





c,



d,

Figure 4. 23 Total deformation (a) Total deformation, (b) von-mises stress on bolt, brace and fork leg and (d)Factor of safety

4.7. Handlebar selection

Handlebar is one of steering mechanism link and it is equivalent of a steering wheel automobile. Besides steering, handlebars also often support a portion of the rider's weight, depending on their riding position, and provide a convenient mounting place for brake levers, shift levers, bells, etc. Handlebars are attached to a bike's stem which in turn attaches to the fork.

Table 4. 6 Riser handle bar specification [42]

Weight	497g	Clamp diameter	25.4mm
Material	Chrome-plated steel	Rise height	100mm
Sweep angle	15°	Length	600mm



Figure 4. 24 Selected riser handlebar [42]

4.8. Other steering parts

I. Headset

The headset of a cycle rickshaw is actually the set of rounded components that provide the interface between the cycle rickshaw steerer and the head tube within the frame. In other words, they are pieces within the front of the cycle rickshaw that allow the cyclist to turn and balance. The bearing system that ensures smooth movement of the fork and front wheel. The headset needs to be compatible with the frame and fork. Broadly speaking there are two types of headset: threaded and integrated. A threaded headset is installed on a fork with a threaded steering column; an integrated headset is installed where there are no threads on the steering column.

Table 4. 7 Specification of head set [42]



Material	Steel alloy
Stack height	70.3mm (including lock ring)
Diameter inner and outer	28 and 30mm

Figure 4. 25 Headset [42]

II. Stem

The stem is the component on a cycle rickshaw that connects the handlebars to the steerer tube of the cycle rickshaw fork. There are two types of stems design such as threaded (quill) and threadless. Threaded stems have a quill that slides into the fork steerer and is tightened by a bolt and wedge. While thread less stems, on the other hand, clamp directly to the steerer tube and are tightened by one or more bolts that bind the stem to the fork. Stems are defined by their Intended use, Length, Diameter and Rise.

Table 4. 8 Stem Specification [42]

Weight	450g
Material	Forged alloy
Rise	25°
Steerer diameter	230mm or 25.4mm using the included shim
Clamp diameter	25.4mm
Clamp height	50mm
Length	90mm



Figure 4. 26 Thread less stem [42]

III. Front axle and drive hub axle

The front axle of new cycle rickshaw carry the weight of front wheel also takes the load come from the uneven road. It also with stand under hard braking weight transferred that means strong enough to total weight of vehicle (350kg or 3433.5N). Based on this steel alloy heavy duty industrial tricycle axle with cartage bearing has been selected.

Table 4. 9 Specification of front axle [42]

	front axle	Hub axle
Material	Steel alloy	Steel alloy
Length	170mm	135mm
Diameter	15mm	10mm

4.9 Drive train

The drivetrain of cycle rickshaw is the system which transfers the human power of the rider to the drive wheels to propel vehicle. Drivetrain systems use a mechanism to convert speed and torque via gear ratios to grant a larger speed or torque range with minimal effort. The cycle rickshaw uses a chain-driven system, similar to that found on a regular bicycle except modified hub body, differential, rear axle, KERS and rear wheel hub shell. There rest part (pedal, Crankset, bottom

bracket, Chain, cassette or (multiple sprocket), Derailleur, sprocket and gear shifter) is similar with upright bicycle part so it can be bought (or selected) from bicycle part store based on standard size used in cycle rickshaw.

4.9.1 Drivetrain part selection

a) Cycle rickshaw Crankset

The crankset is the primary interface between the rider and the cycle rickshaw. It is the component of a cycle rickshaw drivetrain that converts the reciprocating motion of the rider's legs into rotational motion used to drive the chain which in turn drives the rear wheel. It consists of the drive side assembly, which includes the crank arm, spider, spindle, and chain rings, and the non-drive side crank arm. It is connected to the rider by the pedals, to the cycle rickshaw frame by the bottom bracket, and to the rear sprocket or cassette via the chain.

Table 4. 10 Crankset Specifications [42]

Crank arm	170mm crank arm length
	BCD 130 mm
	5 arm spider
	Square bottom brackets
	Material steel
Chain wheel	52 teeth
	BCD 130 mm
	Blade thickness 2 mm
	Material steel



Figure 4. 27 (a) crank arm, (b) chain wheel [42]

b) Bottom bracket

The bottom bracket (abbreviated BB) is a set of bearings that attaches to either side of a cycle rickshaw frame's bottom bracket shell. The bottom bracket is essentially the interface for the crankset and the frame, and provides a tight fit that does not allow the crankset to wobble as it turns. The bearings inside of the bottom bracket help the cranks spin smoothly and efficiently inside the frame. Bottom brackets come as either threaded or press fit types.

Table 4. 11 BB Specifications [42]

Bottom Bracket	Sealed cartridge bearings
	Square taper spindle
	BB Shell Width: 68 mm
	BB Thread Type: English
	Square Taper



Figure 4. 28 Square taper bottom bracket [42]

c) **Cassette**

Cassette is the set of multiple sprockets that attaches to the drive hub. A Cassette works with a rear derailleur to provide multiple gear ratios to the rider. Cassette come in two varieties, freewheels or cassettes, of which cassettes are a newer development. Although cassettes and freewheels perform the same function and look almost the same when installed, they have important mechanical differences and are not interchangeable. Selecting cassette (or multiple gears) for new cycle rickshaw is important because make easy during start up and to obtain maximum speed.

Table 4. 12 Cassette Specifications [42]

Cassette	Compatibility Shimano 7-speed
	Size 13-28T: 13-15-17-19-24-28
	BB Shell Width: 68 mm
	ISO 1.375in*24ipt
	Material Steel



Figure 4. 1 Five speed Freewheel [42]

d) **Gear shifter selection**

The selection of gear shifting mechanism for cycle rickshaw requires integrations with braking and KERS. They must work together to control the cycle rickshaw.

e) Derailleur

Cycle rickshaw have variable-ratio transmission system, it consisting of a chain, multiple sprockets of different sizes, and a mechanism to move the chain from one sprocket to another. The mechanism that drives the gear-changing process on cycle rickshaw is called derailleur.

Cycle rickshaw has rear derailleurs typically consist of a moveable chain-guide that is operated remotely by a Bowden cable attached to a shifter mounted on the handlebar. When a rider operates the lever while pedaling, the change in cable tension moves the chain-guide from side to side, "derailing" the chain onto different sprockets. The rear derailleur serves double duty: moving the chain between rear sprockets and taking up chain slack caused by moving to a smaller sprocket at the rear. In order to accomplish this second task, it is positioned in the path of the bottom, slack portion of chain. Sometimes the rear-derailleurs are re-purposed as chain tensioners for single-speed bicycles that cannot adjust chain tension by a different method. It also needs to accommodate large differences in chaining size: from as many as 52 teeth to as few as 28 teeth.

Table 4. 13 Derailleur Specifications [42]

Derailleur	Compatibility: 1x7	X-
	ACTUATION	
	Size 13-28T: 13-15-17-19-24-28	
	Cage Sizes :medium cage	
	Pulleys: 12-tooth pulley wheels	
	Material aluminum, and steel	



Figure 4. 30 Front and rear derailleurs [42]

f) Gear shifter and selection

A cycle rickshaw gear shifter is a component used to control the gearing mechanisms and select the desired gear ratio. It operates by derailleur mechanism. The control is operated by moving a cable that connects the shifter to the gear mechanism. Shifters are mounted on the handlebars.

There two type of shifter such grip shifter and trigger shifter. The grip shifter has no levers, the gear shifted by twisting a section of the grip forward or backward, depending on whether you want a

harder or easier gear. While trigger shifter has two levers for each hand one lever moves the chain up through the sprockets and the other one moves the chain down. On one hand, the top lever makes the gears harder, and on the opposite hand the top shifter makes the gears easier.

Table 4. 14 Shifter Specifications [42]

Shifter	Cassette Compatibility: 7 Speed
	Model :7 Speed Twist
	Clamp Material: Aluminum
	Shift Cable: Stainless



Figure 4. 31. Gear shifter [42]

g) Pedal and section

The pedals are an important element of the transmission since they are in direct contact with rider's feet and they permit to control pedaling power. Foot pedals on most bikes and trikes come in one of three distinct forms. The first is a simple, rugged platform on which the foot presses. These pedals are typically textured or rigid in some manner in order to keep the user's shoe (and thus the user's foot) firmly on the pedal throughout its full range of rotation. Second, there are pedals that are designed to clip directly to specially manufactured shoes. These allow for much more user freedom of movement and power output by keeping the foot firmly mounted to the pedal. Finally, there are pedals that strap the foot in place. These pedals work similarly to the clip pedals, but do not require special footwear. However, they are larger, heavier, and provide less of a hold on the shoe.

Table 4. 15 Pedals Specifications [42]

Pedals	Size: 118.3 x 112.6 mm
	Model :plat
	Thread: 9/16"
	Material Aluminum



Figure 4. 32 Standard foot pedal styles [42]

4.9.2. Drive hub design for cycle rickshaw

Drive hub of new cycle rickshaw is a mechanical system used to transmit power from crankset to rear differential. It includes some type of a mechanism to convert speed and torque via gear ratios. That mechanism is multiple sprocket set and final drive single sprocket. The working principle of flywheel equipped cycle rickshaw drive hub is the same as bicycle. The only difference from bicycle drive hub is new cycle rickshaw drive hub shell has no flanged part and it assembled separately from wheel this because of cycle rickshaw rear wheels is far away from center of chain wheel, the distance between chain wheel to rear axle is long, there are also differential and KER system mount on middle of rear axle this make difficult to install multiple sprocket and sprocket shifting system. To overcome these difficulty drive hub of new cycle rickshaw, need modification. That is the flange part removed and replace by both end threaded. Hub body end thread purpose is to assemble standard cassette and final drive sprocket. The used thread cut method is right hand thread on drive side for driving cassette and left hand thread on the driven side for final drive sprocket. These thread methods cut use in order to secure or lock during riding.

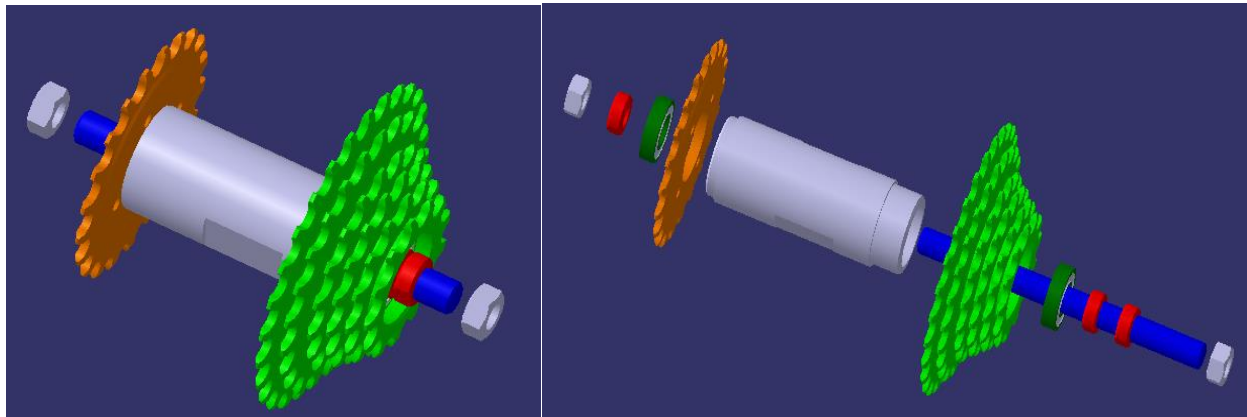


Figure 4. 33 Drive hub assembly

Design requirement

- Must be compatible to standard cassette (free wheel) and finale drive sprocket assembly.
- Must transfer driving torque to rear sprocket without failure.
- Strong enough to with stand driving torsional load.
- Light weight and Inexpensive
- Must fit on main frame mount.

4.9.3. Finite element analysis

The drive hub is a component which assembled with multiple driving sprocket for purpose of manipulation of torque during operation. That driving torque induce torsion load on the drive hub to find out deformation and stresses the finite element analysis method has been used.

Material for drive

Selecting material for drive hub is important by considering operating load type, weight, strength manufacturability and cost. Keeping these constraints in mind inexpensive cold drawn shaft material AISI1015 has been selected with mechanical property of ultimate tensile strength 390Mpa, yield strength 250Mpa, young modulus 205Gpa and poisson ratio 0.3.

Model

The model of drive hub generated in CATIA V5 by using the relevant dimension taken from cassette, crankset and rear axle sprocket location, and bearing. That is six speed free wheel of width 32mm and 34.9mm inner diameter with thread diameter is 32 mm and 1.375*24ipt., ball bearing (6901) 6mm width and 28mm outer diameter, and length 105mm. figure 4.35 shows drive hub model.

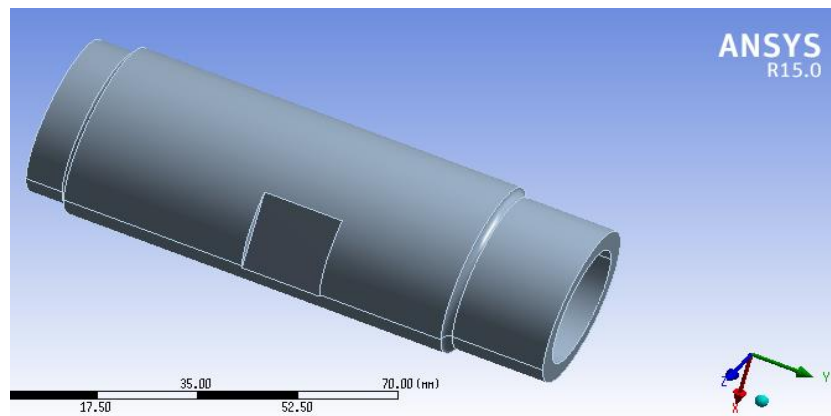


Figure 4. 34 Drive hub model

Mesh

Fine mesh generated on ANSYS workbench using global mesh control with element size 1mm and the model was meshed to 251194 nodes and 170545 elements. The figure 4.35, (a) shows mesh drive model.

Boundary and loading condition

The drive hub is an intermediate power transmitting component so it is subjected to driving torsional load, for 80kg rider 63Nm applied to drive hub, this obtain from calculation of sprockets, see appendix C. However, all rider may not equal with body weight some may less or higher let assume twice of that torque in order to design safe drive hub that is 126Nm applied at both ends. Finally, fixed support was applied at other face (A). The figure 4.36 (b) shown loading boundary condition of drive hub.

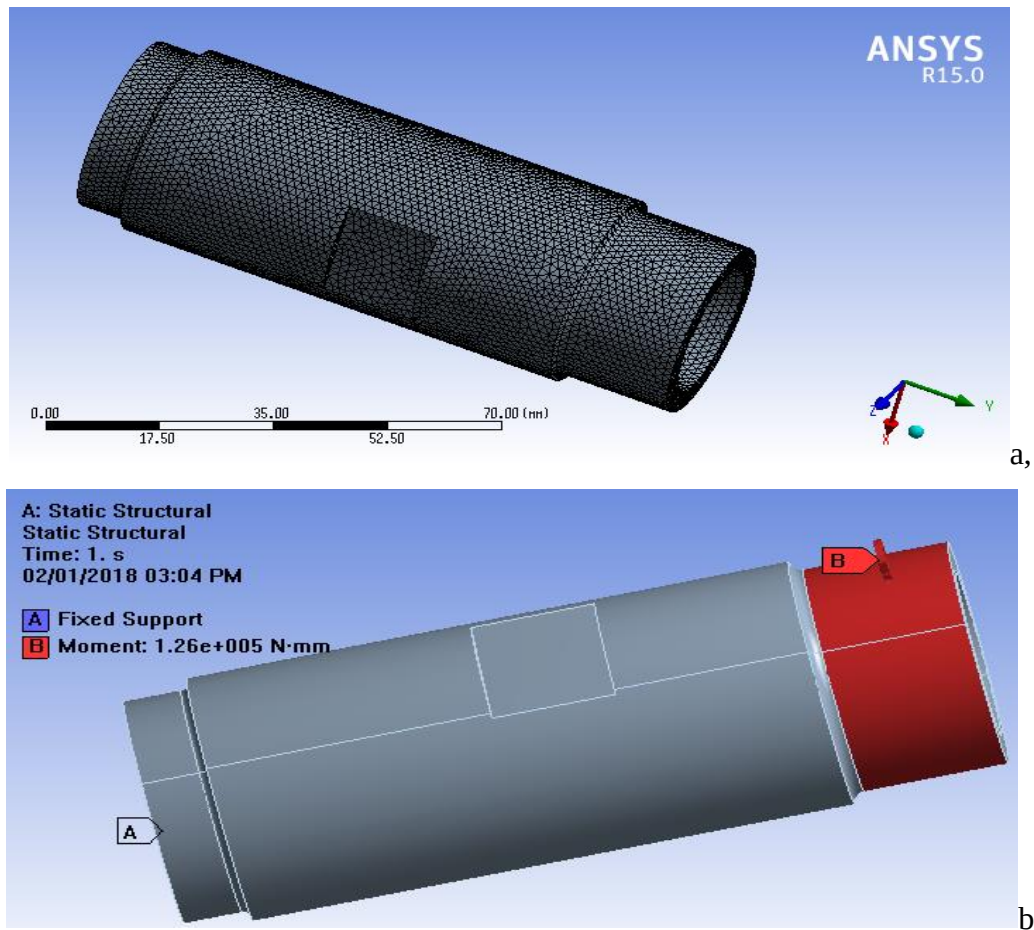
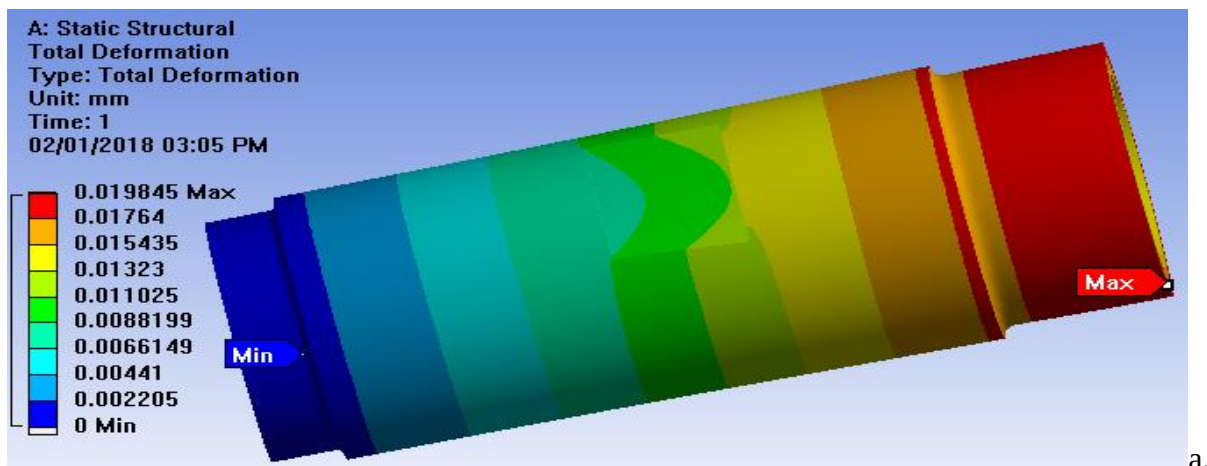


Figure 4. 35 a) Meshed model and b) boundary and loading condition of hub body

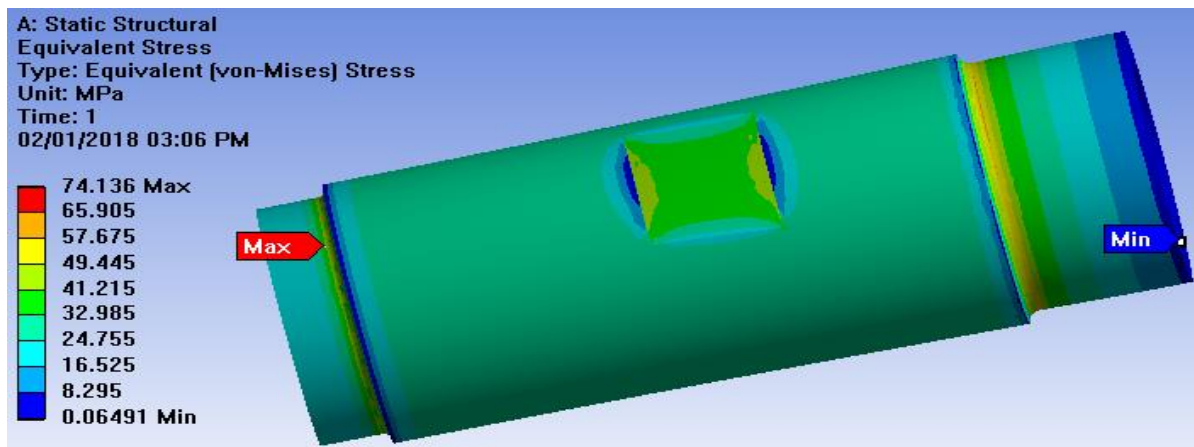
Drive hub analysis result

The following figures 4.36 shows the result of (a) total deflection, (b) von-mises stress and (c) factor safety in the drive when the torsion load of 126Nm is applied to the drive side face. The red color shows the Maximum simulation value of those parameters and blue color shows minimum

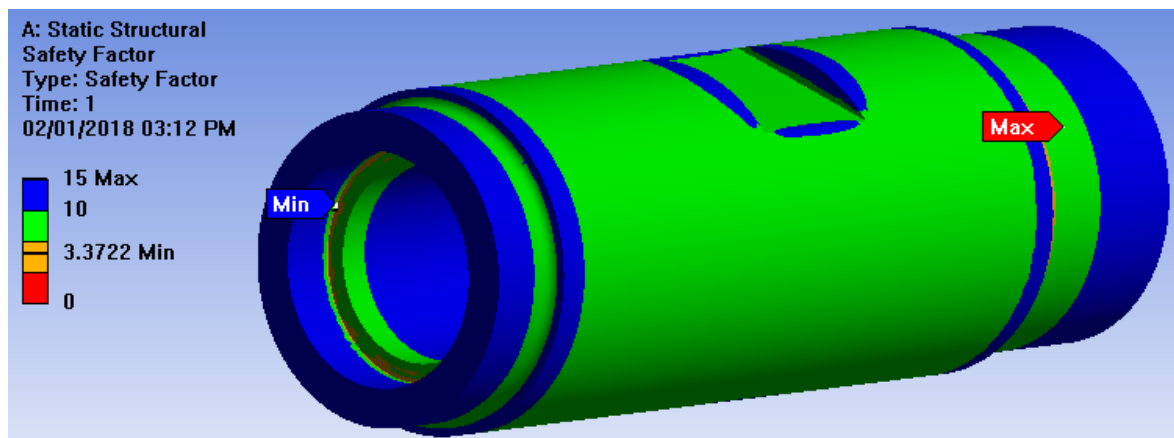
simulation value of those parameters generated. But for factor of safety blue color shows maximum value and red color shows minimum value



a,



b,



c,

Figure 4.36 Drive hub simulation (a) Total deformation, (b) Von-Mises Stress, and (c)Factor of safety

4.9.4. Differential in cycle rickshaw

Differential is the another main component of the power train. The main function of cycle rickshaw differential is, to transmit driving torque to driving axle, to provide gear reduction in order to increase torque, to split torque between two wheel and to allow wheel to turn at different speed when cycle rickshaw negotiating road curve. The differential consists of a set of bevel gears, sprockets, shaft and chain to connects the crankset with the rear axles.

Differential is used when a vehicle takes a turn, the outer wheel on a longer radius than the inner wheel. The outer wheel turns faster than the inner wheel that is when there is a relative movement between the two rear wheels. If the two rear wheels are rigidly fixed to a rear axle the inner wheel will slip which cause rapid tire wear, steering difficulties (handling instability) and poor load holding that because the torque transmitted to both the rear wheels is equal in any condition [43]. So the both rear wheels will run at same speed while taking a turn which causes those problem

4.9.4.1. Type of differential

There are three type of differential such as Conventional, Non-slip or self-locking type, and Double reduction type. Conventional type differential delivers the same driving torque to each rear wheel. It is used light vehicle but if vehicle operating in sand, mud, and snow environment this type differential not effective because the one loss traction. When one tire loses traction, the opposing tire also experiences a torque reduction. In the worst case, vehicle is stuck with one wheel freely spinning while the tire with better traction can't deliver enough torque to budge the vehicle. While Non-slip or self-locking type differential overcome this drawback. It construction is similar to that of conventional type differential. But, two set of clutch plates are provided additionally. Also, the ends of planet shafts are loose in notches provided on the differential cage. Finally, double reduction type differential provides further speed reduction by additional gear. This type of differential is used in heavy duty automobiles which require larger reduction between engine and wheels. [44]

For this thesis work conventional type differential design has been selected based on lightness of vehicle, operating environments is urban and short distance commuter. Conventional type differential simple to design, light in weight, inexpensive and there no complicated controlling system comparing to Non-slip or self-locking type differential. [43,44]

Design requirement

1. Must turn inner and outer wheel at different speed when cycle rickshaw negotiating turn.
2. Must with stand maximum driving load without frailer.
3. Light weight.
4. feasible and inexpensive
5. Aesthetically pleasing

4.9.4.2. Components of cycle rickshaw differential

The cycle rickshaw differential consists sprocket, set of bevel gears, differential casing, bush bearing and assembling bolts. Unlike car differential cycle rickshaw not consist pinion and crown bevel gears. That because cycle rickshaw is not use propeller shaft to transmit driving power to differential assembly instead that it uses chain and sprocket drive.

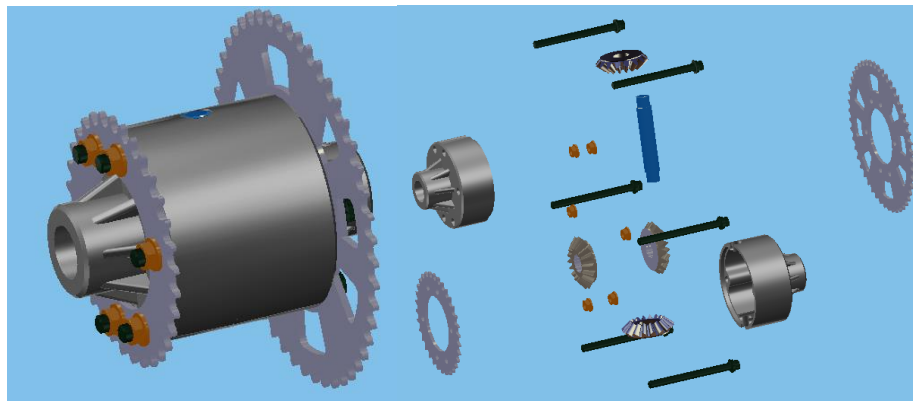


Figure 4. 2 Assembled and exploded view of flywheel equipped cycle rickshaw differential

a). Sprocket

Sprockets are used to transmit power from crankset to differential through chain. Two sprockets must be used. The gear ratio here taken in to account. One sprocket with higher number of teeth is to be selected for KERS and another one having lesser number of teeth for direct drive. The differential small sprocket connected by chain with smaller sprocket installed on drive hub. This is to ensure that to provide final gear reduction which in turn to increase wheel torque.

B). Set of bevel gears.

Bevel gears are gears where the axes of the two shafts intersect and the tooth-bearing faces of the gears themselves are conically shaped. Bevel gears are most often mounted on shafts that are 90

degrees apart, but can be designed to work at other angles as well. The pitch surface of bevel gears is a cone. The most common differential bevel set gears are straight bevel gears which have conical pitch surface and teeth are straight and tapering towards apex.

For this work Miter gears has been used, which is type of straight bevel gears that have equal numbers of teeth. The shafts are positioned at right angles from each other, and the gears have matching pitch surfaces and angles, with a conically shaped pitch surface. Miter gears are useful for transmitting rotational motion at a 90-degree angle with a 1:1 ratio [43].



Figure 4. 38 Straight miter gears

4.9.5. Design of Bevel Set

Input Parameter, power: 1.393 kW, Speed: 121.8 rpm, Torque: 218 Nm, reduction (or final gear) Ratio: 1.524:1 this data available at appendix C. Table C.5.

Constraints for Bevel Set

Miter Gear Pitch angle $\theta_p = 45^\circ$

Back cone Angle (ϕ) = $90^\circ - \theta_p = 45^\circ$

Pressure Angle (ψ) = 20°

Module (m) = 4 mm

Velocity Ratio (G) = 1 (1:1)

a). Sizing of Bevel Set

Table 4. 16 Miter bevel gear size calculation

Addendum	$A_a = 1M$	$A_a = 1 * 4 = 4mm$
Deddndum	$D_d = 1.25M$	$D_d = 1.25 * 4 = 5mm$
Total depth	$H = A_a + D_d$	$H = (4 + 5)mm = 9mm$
Pitch diameter	$D = Z * M$	$D = 16 * 4mm = 64mm$
Cone distance	$OA = \frac{D/2}{\sin\theta_p}$	$OA = \frac{64/2}{\sin 45^\circ} = 45.255mm$
Addendum angle	$\theta_a = \tan^{-1}\left(\frac{A_a}{OA}\right)$	$\theta_a = \tan^{-1}\left(\frac{4}{45.255}\right) = 5.051^\circ$

Deddndum angle	$\theta_d = \tan^{-1}\left(\frac{D_d}{OA}\right)$	$\theta_d = \tan^{-1}\left(\frac{5}{45.255}\right) = 6.305^\circ$
Face angel	$\theta_F = \theta_p + \theta_d$	$\theta_F = 45^\circ + 5.051^\circ = 50.051^\circ$
Root angle	$\theta_R = \theta_p + \theta_d$	$\theta_F = 45^\circ - 6.055^\circ = 38.945$
Outside diameter	$D_o = D + 2 A_a \cos \theta_p$	$D_o = (64 + 2 * 4 * \cos 45^\circ)mm = 69.66mm$
Inner diameter	$D_i = D - 2 D_d \cos \theta_p$	$D_i = (64 - 2 * 5 * \cos 45^\circ)mm = 56.93mm$
Total depth	$H = A_a + D_d$	$H = (4 + 5)mm = 5mm$
Clearance	$= 0.2M$	$= 0.2 * 4mm = 0.8mm$
Working depth	$= 2M$	$= 2 * 4mm = 8mm$
Thickness of tooth	$= 1.5708M$	$= 1.5708 * 4mm = 6.283mm$
Face width	$b = \frac{OA}{3}$	$b = \frac{45.255mm}{3} = 16mm$
Fillet radius at root	$= 0.4M$	$= 0.4 * 4mm = 1.6mm$
Mean radius	$R_m = \frac{D}{2} - \frac{b \sin \theta_p}{2}$	$R_m = \left(\frac{64}{2} - \frac{16 \sin 45^\circ}{2}\right)mm = 37.66mm$
Minimum number of teeth	$Z_{min.} = \frac{2A_a \cos \theta_p}{M(\cos(\psi))^2}$	$Z_{min.} = \frac{2 * 4 * \cos 45^\circ}{4(\cos(20^\circ))^2} = 12.08 \approx 13 \text{ teeth}$ Since $Z_{actual} > Z_{min.}$ it is permissible

b). Force analysis of bevel gear

In determining shaft and bearing loads for bevel-gear applications, the usual practice is to use the tangential or transmitted load that would occur if all the forces were concentrated at the midpoint of the tooth. While the actual resultant occurs somewhere between the midpoint and the large end of the tooth, there is only a small error in making this assumption. For the transmitted load, this gives

$$F_T = \frac{T}{R_m} \dots\dots\dots 4.10$$

where F_T is Tangential Force, T is Torque acting on Bevel Set and R_m is pitch radius at the midpoint of the tooth for the gear under consideration.

The forces acting at the center of the tooth are shown in Fig. The resultant force F has three components: a tangential force F_T , a radial force F_R , and F_A an axial force. From the trigonometry of the figure,

$$F_R = F_T \tan \psi \cos \theta_p \dots\dots\dots 4.11$$

$$F_A = F_T \tan \psi \sin \theta_p \dots\dots\dots 4.12$$

The three forces F_T , F_R , and F_A are at right angles to each other and can be used to determine the bearing loads by using the methods of statics.[45]

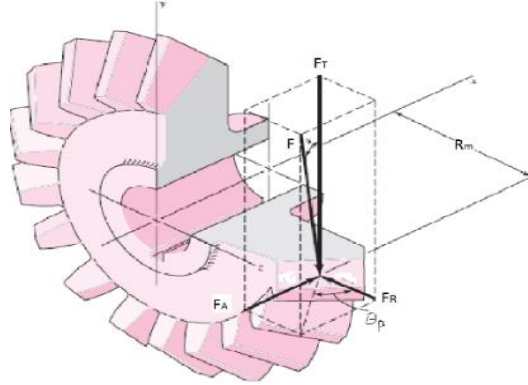


Figure 4. 39 Bevel-gear tooth forces. [45]

Torque acting on Bevel Set (T) = Nm =218000Nmm

Speed of rotation of Bevel Set = 107.3 rpm

Table 4. 17 Acting loads new cycle rickshaw miter bevel gear tooth

Tangential Force (F_T)	$F_T = \frac{T}{R_m}$	$F_T = \frac{218000Nmm}{37.66mm} = 5108N$
Radial Force on Pinion (F_R)	$F_R = F_T * \tan\psi * \cos\theta_p$	$F_R = 5108N * \tan 20^\circ * \cos 45^\circ = 1315N$ For miter gear radial and axial force is equal. $F_R = F_A$

4.9.6. Splines for side bevel gears

The side bevel gears of cycle rickshaw are assembled to rear half axles in order to transmit driving torque or power to the road wheel. The mechanism used to assemble must tolerate uniform torque or power flow between them (side gears and axle shafts), and give easy access to disassemble for maintenance or service purpose. The splines are best option to accomplish this task. A spline can be described as a series of axial keys machined into a shaft, with corresponding grooves machined into the bore of the mating part (gear, sheave, sprocket, and so on): see Figure 4.49). The splines perform the same function as a key in transmitting torque from the shaft to the mating element. The advantages of splines over keys are many. Because usually four or more splines are used, as compared with one or two keys, a more uniform transfer of the torque and a lower loading on a given part of the shaft/hub interface result. The splines are integral with the shaft, so no relative motion can occur as between a key and the shaft. Splines are accurately machined to provide a controlled fit between the mating internal and external splines. The surface of the spline is often

hardened to resist wear and to facilitate its use in applications in which axial motion of the mating element is desired [46].

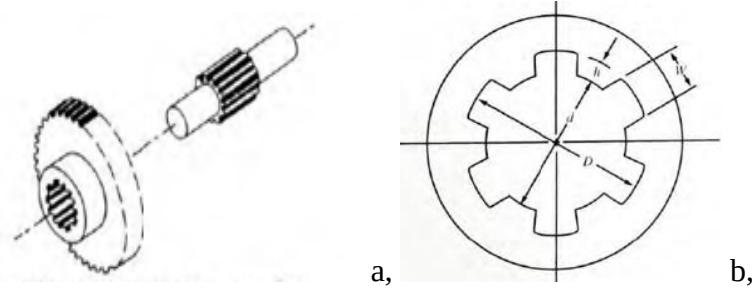


Figure 4. 40 a) General form of spline (b) internal spline [46]

Table 4. 18 Formulas for SAE straight spline [45].

No. of splines	w	A: permanent fit		B: to slide without load		C: to slide under load	
		h	d	h	d	h	d
4	0.241D	0.075D	0.850D	0.125D	0.750D		
6	0.25D	0.05D	0.900D	0.075D	0.850D	0.100D	0.800D
10	0.156D	0.045D	0.910D	0.070D	0.860D	0.095D	0.810D
16	0.098D	0.045D	0.910D	0.070D	0.860D	0.095D	0.810D

Length of Splines. Common designs use spline lengths from $0.75D$ to $1.25D$, where D shaft diameter. Take $l = 0.75D = 18.75mm$

4.9.7. Finite element analyses of side bevel gear

The set of miter bevel gears are used in flywheel equipped cycle rickshaw differential to transmit the power. During this phase, they encounter high stress on the gears teeth at the point of contact. To find out the characteristics of gear model due induce load at contact the finite element analysis done on ANSYS workbench 15.

a). Material for bevel gear set

The gear material should have high tensile and endurance strength to prevent failure against static load and dynamic load respectively. They also have low coefficient of friction and good manufacturability.

Alloy Steel material has been selected for bevel gears set with Case hardened 20MnCr5 is used for manufacturing of gear. It is a special alloy steel with 0.17-0.22% of Carbon. Case-hardening process offers a means of enhancing the strength and wear properties of parts made from relatively inexpensive easily worked materials. The treatments confer a high-hardness wear resistant surface also improve fatigue strength. As bevel gears are small in size and torque transmitted is comparatively high so to withstand case hardening process is needed. So strength and resistance to stress are the criteria for the selection of material. 20MnCr5 also provides good wearing resistance [39]. The mechanical property of this material is of ultimate tensile strength 814Mpa, Yield tensile strength 703Mpa, Young's Modulus 206Gpa, Poisson ratio 0.29, and density 7850 Kg/m³.

b). Modelling of gear

The model of miter bevel gear has generated in CATIA V5 by using table 4.16 and 4.18. Miter bevel gear size calculation data and spline size. The figure 4.42 (a) illustrates the model of the bevel gear.

c). Mesh

The bevel gear has been meshed in ANSYS workbench 15 by using hex dominant mesh refinement method with 1mm element size. This method applied because the gear geometry has many curved face which cause reduce the mesh quality by other method. The figure 4.42. (b) shows meshed model of miter bevel gear with 268580 nodes and 73459 elements.

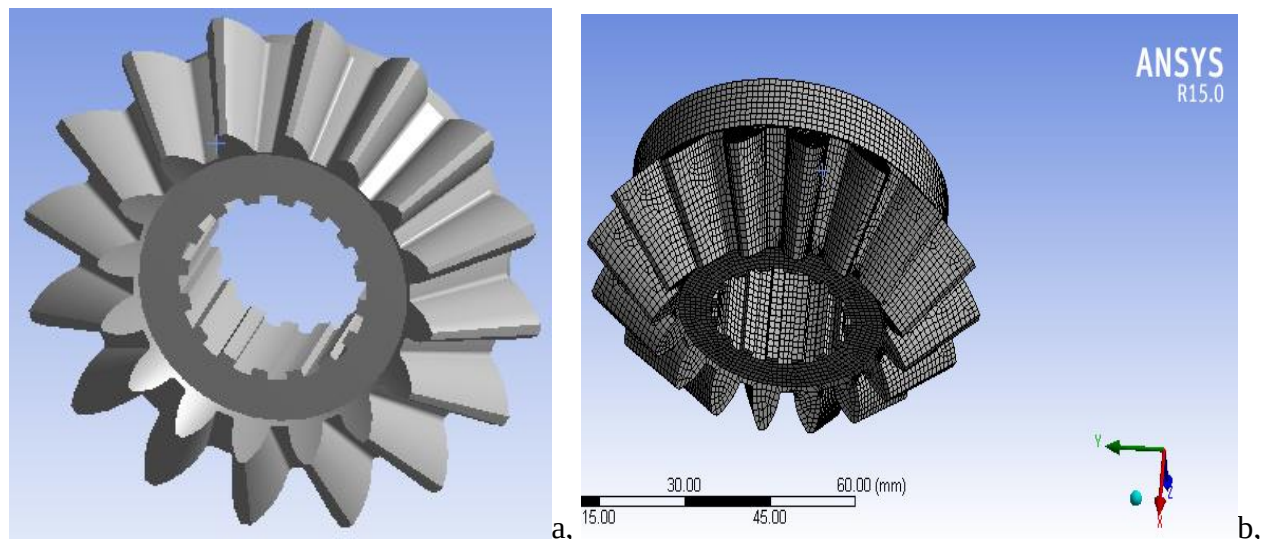


Figure 4. 41 a) Differential gear model and b) Meshed model

d). Loading and boundary conditions

A load has been applied on one of the teeth face which in contact with other bevel gear teeth, while a fixed support has been applied at the inner splined faces. The gear assembled with axle shaft considered as rigidly secured with shaft to transmit power to road wheels without slip. The load applied on single already determined in section (b). force analysis of bevel gear. The bevel gear tooth was loaded in the following manner; axial force (1315N) in negative x- axis, radial force (1315N) positive y-axis and tangential force (5108N) in the negative z- axis. Shown below figure 4.43.

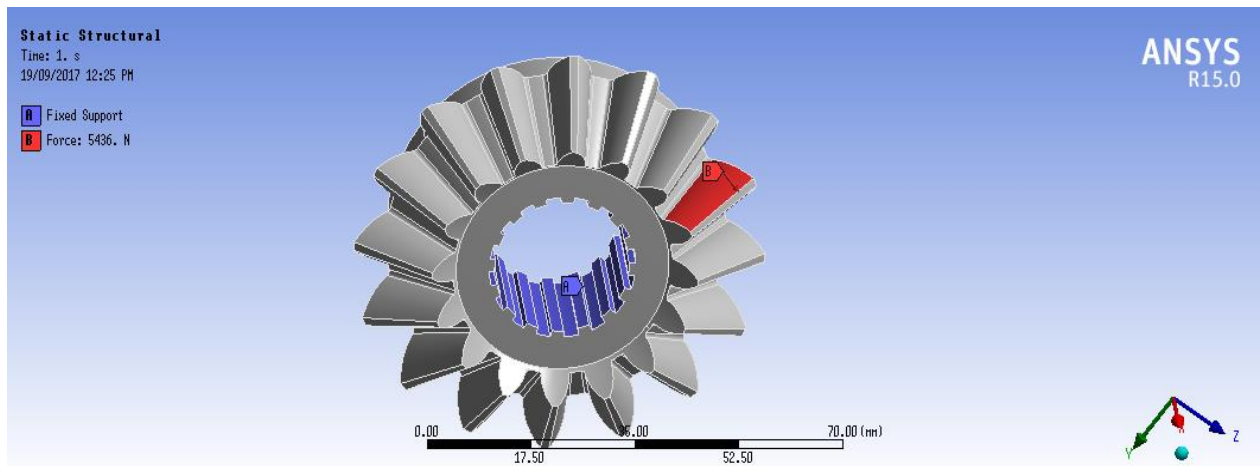


Figure 4. 42 Loading and boundry condition

Differential gear analysis result

The following figures 4.44. shows the result of (a) total deflection, (b) von-mises stress and (c) factor safety in the differential gear when the load on applied single teeth face. The red color shows the Maximum simulation value of those parameters and blue color shows minimum simulation value of those parameters generated. But for factor of safety blue color shows maximum value and red color shows minimum value

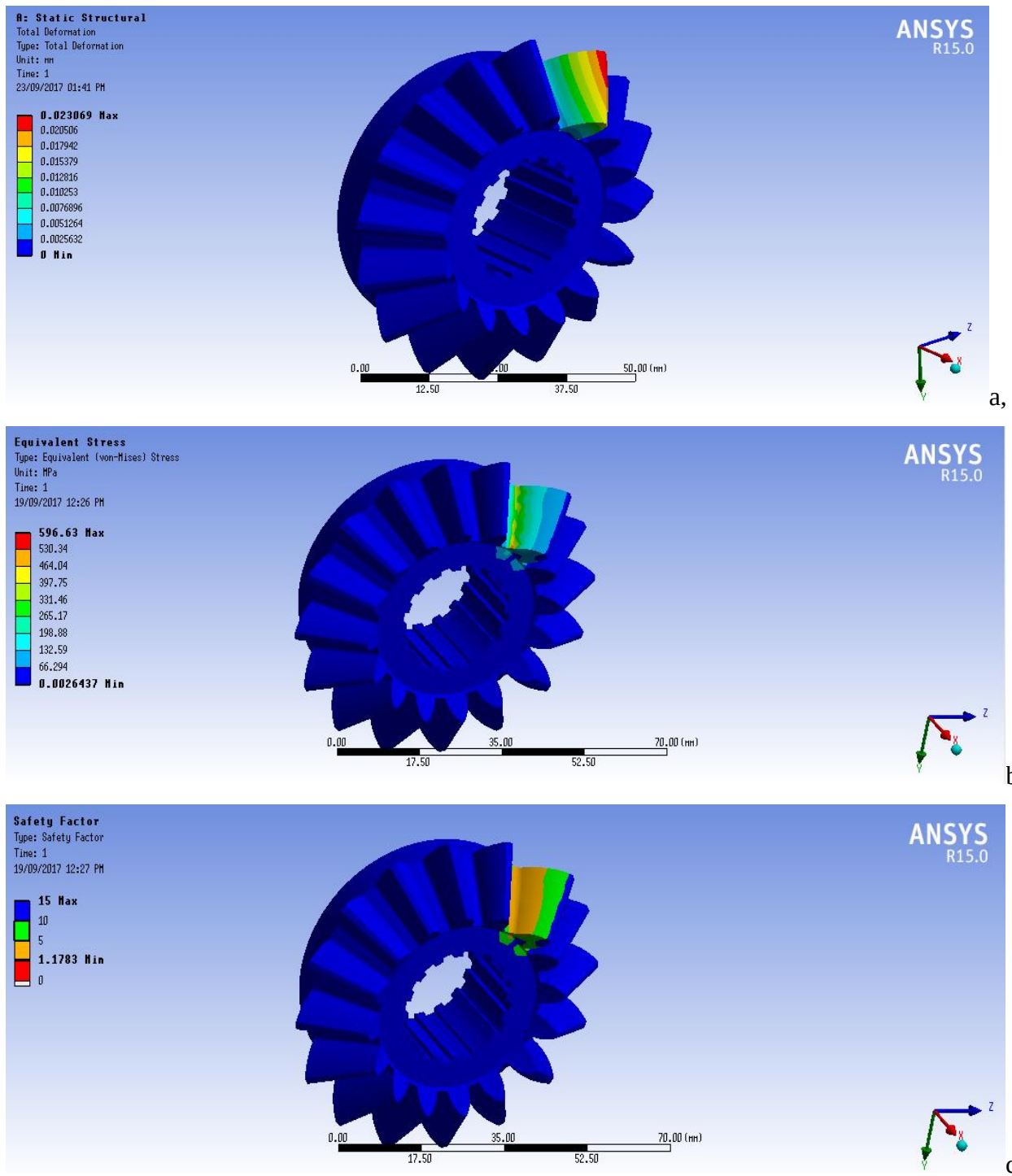


Figure 4. 43 Differential gear simulation result, a) total deformation, b) Von-Mises stress, c) factor of safety

4.9.8. Cycle rickshaw rear axle shaft

Axle shaft of flywheel equipped cycle rickshaw is the last member of power train. It lies between the driving wheels and the differential gear and transmit power from the differential to the driving wheels. It consists of two half shaft connected to the differential gear, one for one wheel. The inner end of each half shaft connected to the sun gear of the differential and the outer end to the wheel. The axle shafts mainly perform the function of carries the total weight of the cycle rickshaw including pay load, and convey driving power to the road-wheels. [44]

a). Types of axles

Automobile axle shafts are divided into three main groups depending on the stresses to which the shaft is subjected such as Semi-floating, three-quarter floating, and fully floating. However, the last two is not valid to this thesis work because they used in medium and heavy truck motor vehicle respectively [44].

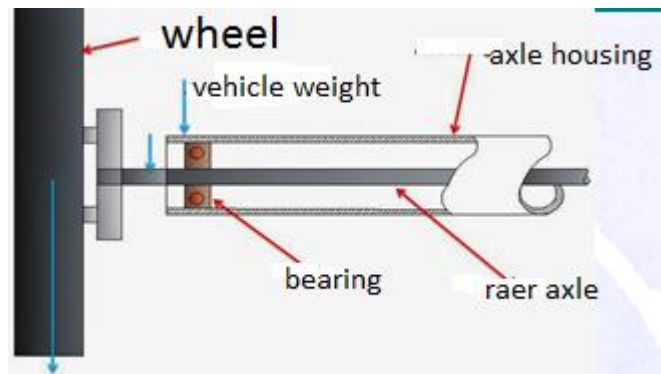


Figure 4. 44 Semi-floating rear axles [44]

Semi-floating half axle

The road-wheel is attached to the axle hub, which is an extension of the axle half-shaft. A single bearing inside the tubular axle-casing supports the outer end of the shaft. The inner end of the shaft is splined and supported by the final-drive unit, which itself is mounted on bearings within the axle casing (figure 4.46). The semi-floating axle along with its overhanging hub is subjected to the driving torque as well as to both vertical and horizontal loads. The vertical load produces a shearing force, and the distance between the wheel and the suspension-spring seat on the axle causes a bending moment, the reaction of which is shared between the axle bearing and the final-drive-unit bearings. The horizontal load due to tilting of the vehicle, cornering centrifugal force, or side wind gives rise to both side-thrust and a bending moment. This bending moment may add

to the vertical bending moment or may oppose it, depending on the direction of application of the side-force [44].

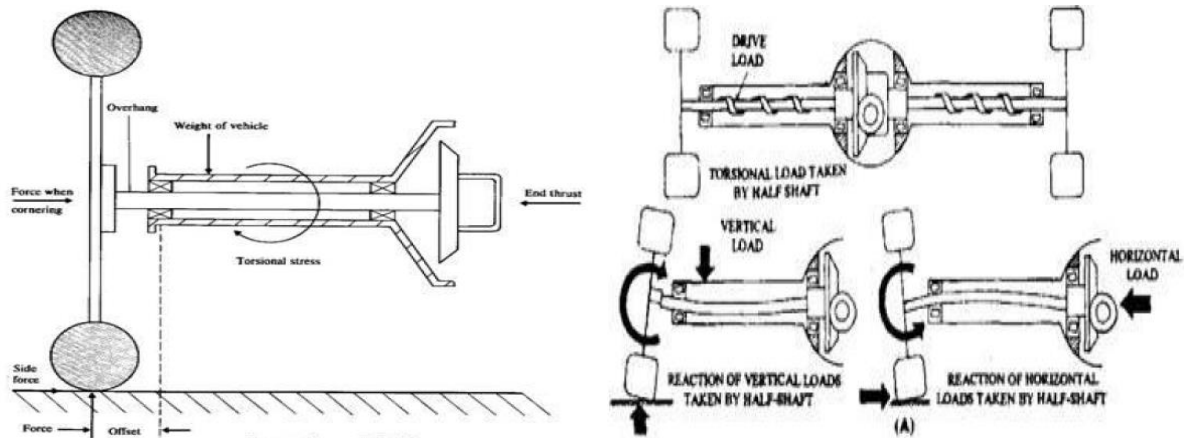


Figure 4.45 Automobile Semi-floating axle free body diagram [44]

b). Rear axle shaft design requirement

- Shaft must withstand different loading stress due to driving and braking torque, shear and bending stresses due to the weight of the vehicle, tensile and compressive stresses due to cornering forces.
- Must fit and transmit power from differential sun gears to road wheels.
- Machine able and Inexpensive
- Esthetical pleasing.

4.9.9. Rear half axle shaft design calculation

During operation of cycle rickshaw, the shafts are subjected to both bending and torsion. Hence, the diameter, of the shaft should vary with the bending moment along the length. Accordingly, the diameter is minimum near the differential end where the shaft is subjected to nearly total torsion is maximum at the outboard bearing, where bending moment is maximum. The maximum stress in shaft occurs when the wheel's slip or lock due respectively to a sudden application of power or braking on dry, hard pavement.

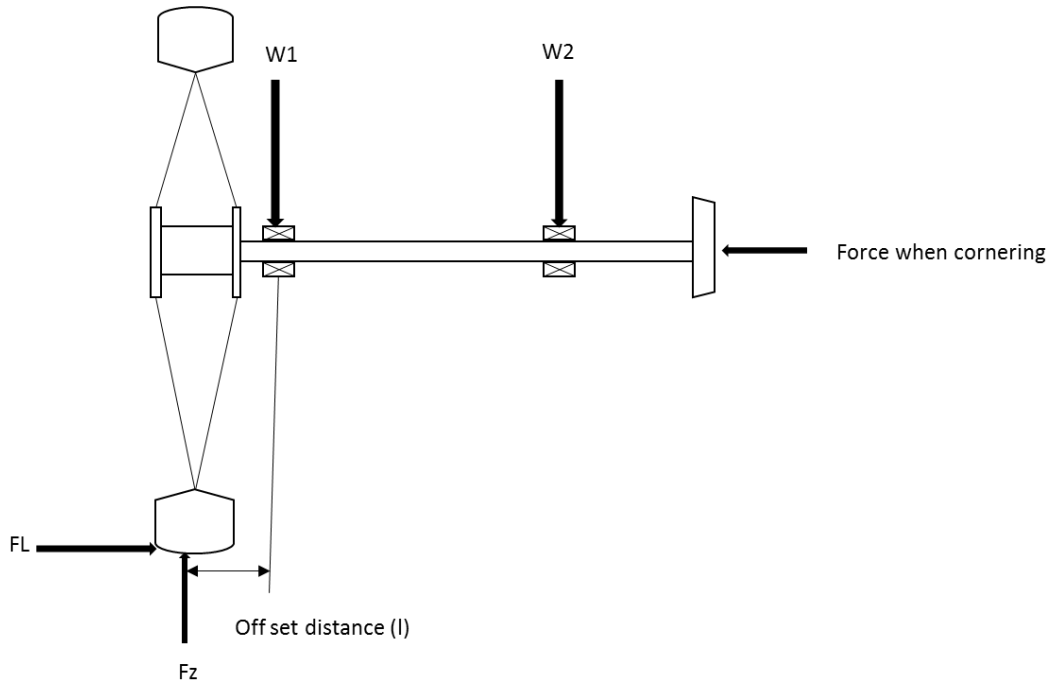


Figure 4.46 Free body diagram of new cycle rickshaw half rear axle

The frictional force under these conditions

$$F_f = \mu * W \dots\dots\dots 4.13$$

where F_f is frictional force, μ is frictional coefficient between tire and the it value is 0.6 for dry and hard pavement, and $W = w1 + w2$ is load supported on the wheel under fully loaded condition.

The friction force F_f produce the torque during braking or skidding and given by

$$T_f = \mu * W * r_w \dots\dots\dots 4.14$$

and a bending moment at the center of the outboard bearing, given by

$$M_f = \mu * W * l \dots\dots\dots 4.15$$

where r_w is effective radius of tricycle tire and l is offset distance from center of outboard bearing to center of wheel hub.

The load W produces a bending moment on the axle the center of the outboard bearing , given by

$$M_v = W * l \dots\dots\dots 4.16$$

Again the driving torque on each wheel, given by $T_d = 96.2. Nm$ already find. But driving torque should be less than the torque due to frictional force. Thus, there are two bending moments on the shaft at the outboard bearing, a vertical moment, M_v and a horizontal moment, M_f . From equation 4.13 and 4.14 the resultant of the these bending moment in the shaft given by. [44]

$$M_b = \sqrt{M_v^2 + M_f^2} \dots\dots\dots 4.17$$

And the resultant bending stress in the shaft, given by

$$\sigma_b = \frac{M_b * y}{I} \dots\dots\dots 4.18$$

Where y is the distance from the neutral axis to the outer –most fiber of the shaft

For solid circular shaft

$$y = \frac{d}{2} \dots\dots\dots 4.19$$

Where: - d is shaft diameter

$$I = \frac{\pi d^4}{64} \dots\dots\dots 4.20$$

Where: - I is the moment of inertia of the section, for solid circular shaft .

Combining equation 4.18, 4.19 and 4.20 we get

$$\sigma_b = \frac{32 * M_b}{\pi d^3} \dots\dots\dots 4.21$$

The shear stress in the shaft is given by,

$$\tau_s = \frac{T_f * y}{J} \dots\dots\dots 4.22$$

where, J is polar moment of the inertia of the shaft section.

$$J = \frac{\pi d^4}{32} \dots\dots\dots 4.23$$

Combining equation 4.19 ,4.22 and 4.23 we get

$$\tau_s = \frac{16 * T_f}{\pi d^3} \dots\dots\dots 4.24$$

using the formula for combined stress, the maximum tensile (or compressive) stress in the shaft

given by,

$$\sigma = \frac{1}{2} \sigma_b \pm \frac{1}{2} \sqrt{\sigma_b^2 + 4 \tau_s^2}, \dots\dots\dots 4.25$$

Combining equation 4.14 ,4.15,4.16,4.12,4.23 and 4.2 we get

$$\sigma = \frac{16w}{\pi d^3} [(l \sqrt{\mu^2 + 1} \pm \sqrt{\mu^2 (r_w^2 + l^2)} + l^2] \dots\dots\dots 4.26$$

(negative sign for compressive) and the maximum shear stress in the shaft is given by,

$$\tau_{max.} = \frac{1}{2} \sqrt{\sigma_b^2 + 4 \tau_s^2}, \dots\dots\dots 4.27$$

Assumption

- High speed cornering plus acceleration, that is worst scenario. Tipping point at one rear wheels this means when normal force become zero at front and other rear side wheel which all weight transferred to one rear wheel during high speed cornering plus acceleration. So total weight on each rear wheel is 310kg.
- $g = 9.81\text{m/s}^2$ is gravitational acceleration.

$$W = m \times g = 310\text{kg} \times 9.81\text{m/s}^2 = 3041\text{N}$$

$$l = 0.05\text{m}$$

$$r_w = 0.305\text{m}$$

Table 4. 19 Half axle shaft load calculation

Frictional force	$F_f = \mu * W$	$F_f = 0.6 * 3041\text{N} = 1825\text{N}$
Torque due to friction force	$T_f = \mu * W * r$	$T_f = 1825\text{N} * 0.305\text{m} = 556\text{m}$ $T_f > T_d(109.3\text{Nm})$
horizontal bending moment	$M_f = \mu * W * l$	$M_f = 0.6 * 3041\text{N} * 0.05\text{m} = 91\text{Nm}$
Vertical bending moment	$M_v = W * l$	$M_v = 3041\text{N} * 0.05\text{m} = 152\text{Nm}$
Resultant bending moment	$M_b = \sqrt{M_v^2 + M_f^2}$	$M_b = \sqrt{152^2 + 91^2} = 177\text{Nm}$

Material for axle shaft

There are a wide variety of material used for shaft. The machine shaft were originally made from steel and alloy steel because to with stand high stresses caused from static and dynamic loads. There is basic requirement that shaft material should have such as, high strength, good machinability, low notch sensitivity factor, good heat treatment properties and high wear resistant properties[36]. Many shafts are made from low carbon, cold-drawn or hot-rolled steel, such as ANSI 1020-1050 steels. The cost of the material and its processing must be weighed against the need for smaller shaft diameters. When warranted, typical alloy steels for heat treatment include ANSI 1340-50, 3140-50, 4140, 4340, 5140, and 8650. When a shaft of high strength is required, then an alloy steel such as nickel, nickel-chromium or chrome-vanadium steel is used. [45]

ANSI 1030 is steel alloy is chosen for shaft based on its inexpensiveness, for surface hardening include carburizing grades suitable. And the axle shaft factor of safety = 1.5- 2.5 take $n_f = 1.5$

Table 4. 20 Axle shaft properties [44]

ANSI no.	Processing	$\sigma_{ut}, in\ Mpa$	$\sigma_y, in\ Mpa$
1015	HR	340	190
	CR	390	320
1018	HR	400	220
	CR	440	370
1020	HR	380	210
	CR	470	390

In case of ductile materials *e.g.* mild steel, where the yield point is clearly defined, the factor of safety is based upon the yield point stress. In such cases, [44].

$$n_f = \frac{\sigma_y}{\sigma_b} \dots \dots \dots 4.28$$

Where: n_f is factor of safety, σ_y is yield strength of material, and σ_b is design stress.

Design stress is equal to combined stress

$$\sigma_b = \frac{\sigma_{yield}}{n_f} = \frac{16w}{\pi d^3} [(l\sqrt{\mu^2 + 1} \pm \sqrt{\mu^2(r_w^2 + l^2) + l^2}) + l^2], \text{ then diameter of half shaft is}$$

$$d = \sqrt[3]{\frac{\frac{16w}{\pi} [(l\sqrt{\mu^2 + 1} \pm \sqrt{\mu^2(r_w^2 + l^2) + l^2}) + l^2] \times n_f}{\sigma_{yield}}} \dots \dots \dots 4.29$$

Then by substituting parameter value

$$d = \sqrt[3]{\frac{16 \times 3041 / \pi [(0.05\sqrt{0.6^2 + 1} \pm \sqrt{0.6^2(0.305^2 + 0.05^2) + 0.05^2}) + 0.05^2] \times 1.5}{440 \times 10^6}}$$

0.02363m or 23.63mm take 25mm.

4.9.10. Differential planet gear shaft design.

The main function of differential planet gear shaft is to transmit driving power on differential casing to differential planet gears and then those gears transmit to differential side gears which in turn to drive wheels.

The differential planet gears shaft predominantly subjected to bending load due to driving torque of rider and KERS. But it is not subjected to torsion load.

Therefor to design differential planet gear shaft equation 4.21 and 4.28 is sufficient. The shaft supported section by differential casing is 12mm on each end, next 6mm by each bush bearing and 28mm by each planet gears. The application point for load middle had considered.

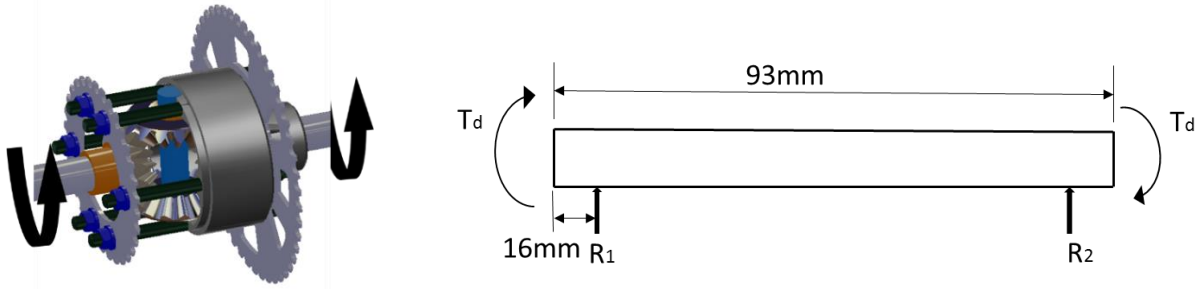


Figure. 4.47 Differential planet gear shaft free body diagram

Where: T_d is driving moment or torque in (Nm), and R is reaction force or tangential load of differential bevel gears.

The magnitude of driving torque is 109Nm applied at both ends. AISI 1018 cold drawn material had selected from table 4.20 to the shaft. For factor of safety magnitude of two had considered.

Using equation 4.21 and 4.28 it can have obtained diameter of differential shaft.

$$d = \sqrt[3]{\frac{32Mn}{\pi\sigma_{yield}}} = \sqrt[3]{\frac{32 \times 109 \times 2}{\pi \times 370 \times 10^6}}$$

$$= 0.018007\text{m or } 18.007\text{mm} \approx 18\text{mm}$$

a). Axle shaft and sun gear securing component selection.

When cycle rickshaw takes road turn the lateral force is developed on the wheels. This force transferred to wheel hub creates axial movement of rear axle shaft. In order to prevent this axial movement of axle shaft (loosening from sun gear) retaining ring is used. Retaining rings are placed on the axle shaft grooves to prevent the axial movement of a machine element. Figure 4.47 shows the two varies styles of rings available. The different designs allow for either internal or extremal mounting of the ring. The amount of axial thrust load capacity and the shoulder height provided by the different ring styles also vary. The manufacturer's catalog provides data on the capacity and installation data for the rings. From ANSI DIN 471 metric retaining ring catalog internal retaining ring has been select for 25mm axle shaft.

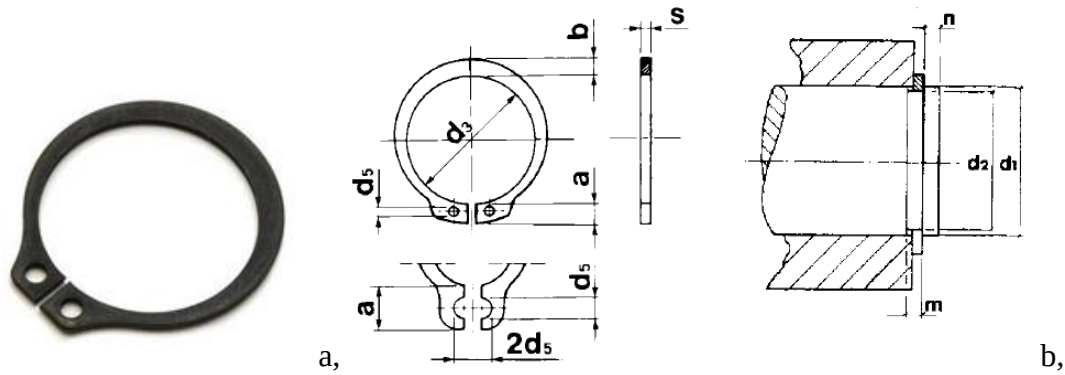


Figure 4.48. External retaining ring [47].

Table 4. 21 Extetrnal retaining ring specification [47]

Nominal dimension in (mm)						
d_1	s	d_3	A	b	d_5	d_5
15	1	13.8	3.6	2.2	1.7	14.3
25	1.5	23.2	4.4	3	2	23.9

4.9.11. Differential casing

The new cycle rickshaw differential casing is the shelled metal component in which a differential gears and certain part of axles are sealed on it in order to protect from dust, external debris and to keep lubrication from leak. This differential casing is rotate and transfer power to the set of bevel gears. The casing is powered by two sprockets, one sprocket get power from direct driving and the other or bigger sprocket from KERS. The set of bevel gears assembled on casing control power flow and speed between two half rear axles. The design of this differential casing idea is the same as automobile deferential casing but in the place of crown gear sprocket have been used. The strength of the gear box casing is important since dynamic load induced during operation.

a). Design requirement of differential casing

- It must withstand dynamic load.
- It must transfer power to set of bevel gears.
- Must fit with set of bevel gears assemble and rear axles.
- Must be light in weight.
- Machinability.

- Must be circular because it rotating with axle unless it creates unnecessary vibration.

b). Differential casing geometry

The cycle rickshaw differential casing geometry dimension considered from assembled bevel gears set, bearing, rear axle, and assembling bolts sizes. After dimension determined 3D model of differential casing is generated in CATIA v5.

4.9.12. Finite element analysis of differential casing assembly

Static structural analysis is done to fine find out how the differential casing reacts to externally applied driving loads and bolt pretension load. The finite element done for assembled model. The following step is used to carry out finite analysis in ANSYS 15.

a). Model of differential casing assemble

The two half differential casing assembled with two sprockets in each side by six M10x1bolts. The figures 4.48 shows (a)isometric, (b) side, and (c) top view of differential casing assembly.

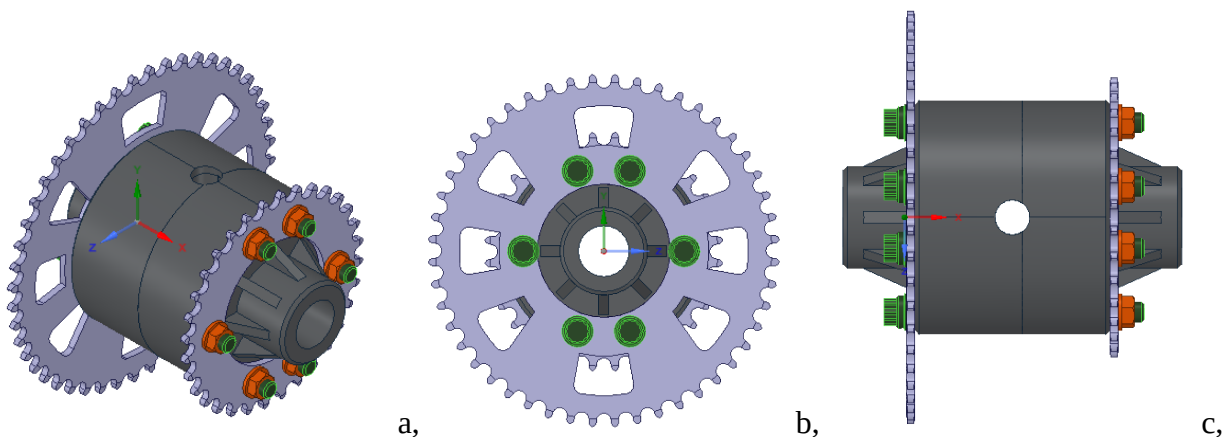


Figure 4.49 (a) isometric, (b) side, and (c) top view of differential casing assembly

b). Meshed model

The multi zone mesh is used for this finite element analysis based on the model geometry features and number of components assembly. The assembled differential casing and sprockets and bolts meshed to 174269nodes and 102227 elements.

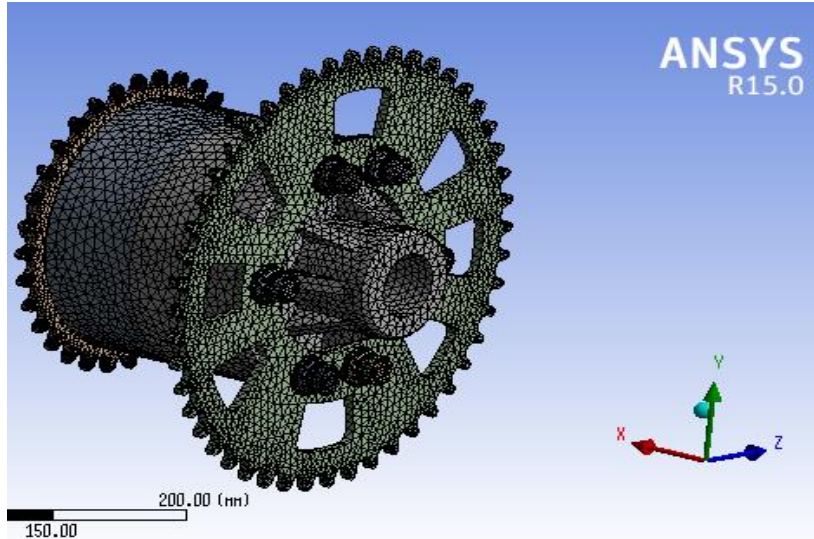


Figure 4.50. Meshed model of differential casing assembly

c). Material

The material for differential casing must be strong and light. Differential Casing fabricated by casting method due its complex geometric shape, it's weight may heavier if it used steel. But more weight is not required human powered vehicles because it reduces the performance of rider. Based on this fact aluminum alloy (Al T6) is selected for differential casing. For assembling low or medium carbon steel flanged M10 bolt has been used. And selected sprocket material is structural steel.

Table 4. 22 Mechanical property of differential assembly

Material	σ_{ut} (Mpa)	σ_y (Mpa)	ρ (kg/m ³)	Poisson ratio	Young module(Gpa)
aluminum alloy (Al T6)	324	169	2800	0.33	70
low or medium carbon steel bolt	520	420	7850	0.29	206
Sprocket (AISI4130)	670	435	7850	0.29	210

d). Boundary and loading condition

The load is applied in the form of moment. The moment of 109Nm applied on the two faces of two sprockets. And pretension load applied on each bolt of magnitude of 14200N. The frictionless support is applied on the bore of casing. fixed support places a normal constraint on a differential shaft hole and axle shaft hole. Which allow model to rotational direction.

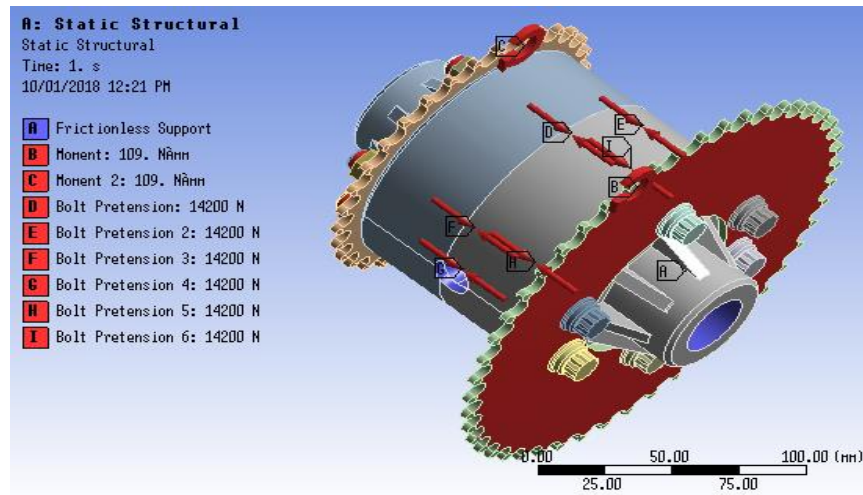
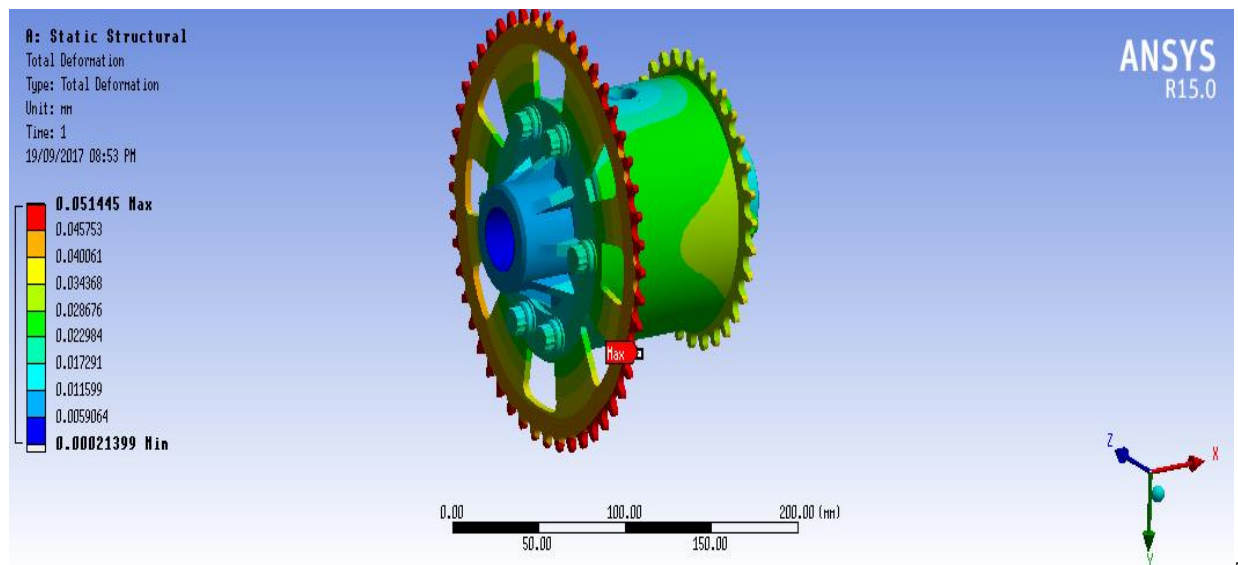


Figure 4.51 Loading and boundary condition

Differential casing assembly analysis result

The following figures 4.52. shows the result of (a) total deflection, (b) von-mises stress and (c) factor safety in the differential gear when the load on applied single teeth face. The red color shows the Maximum simulation value of those parameters and blue color shows minimum simulation result



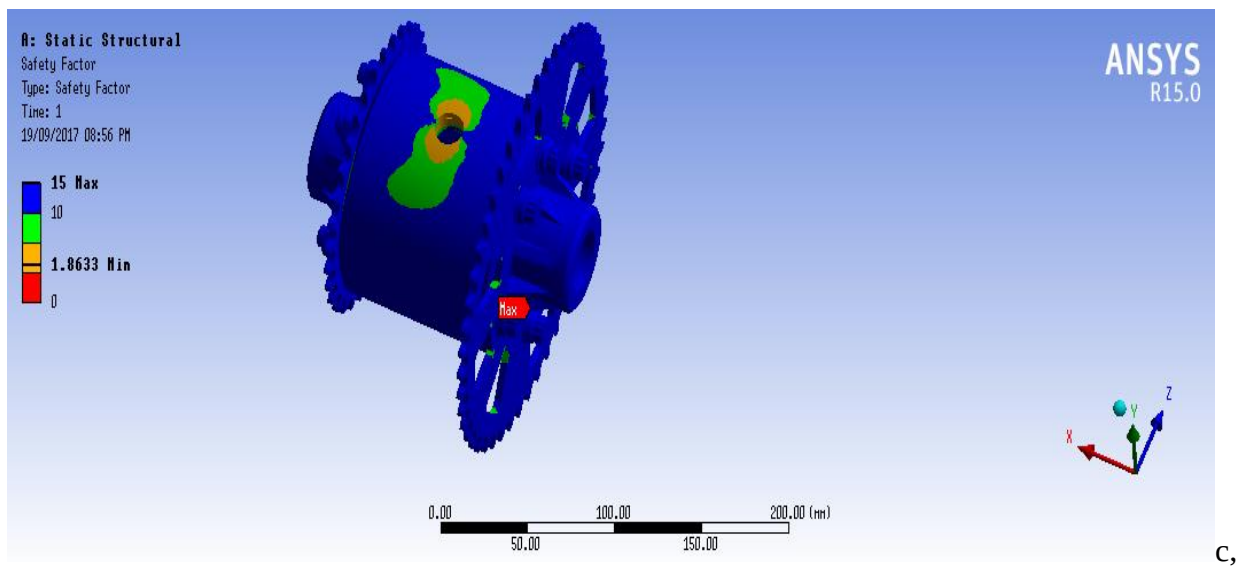
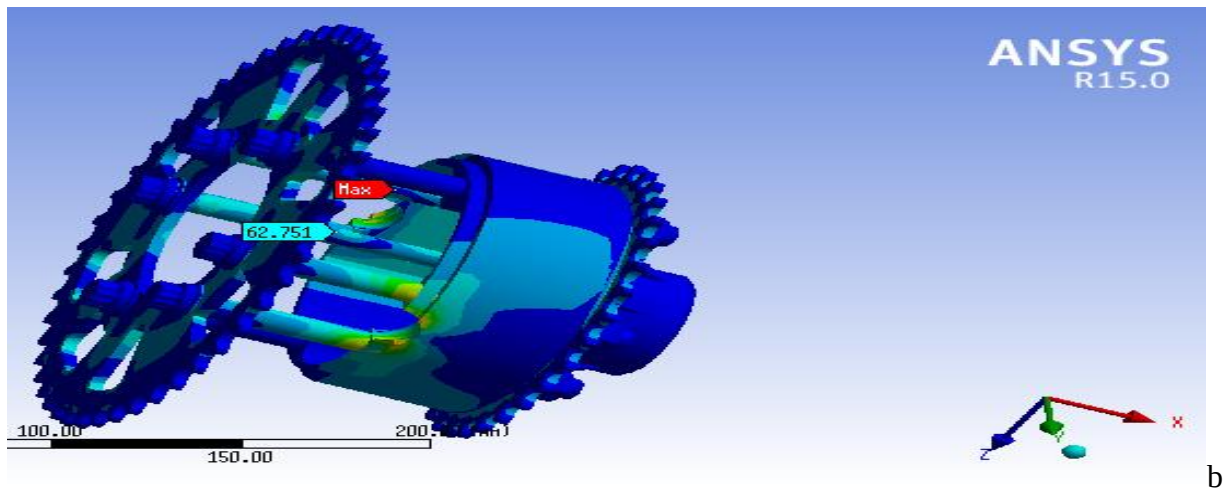
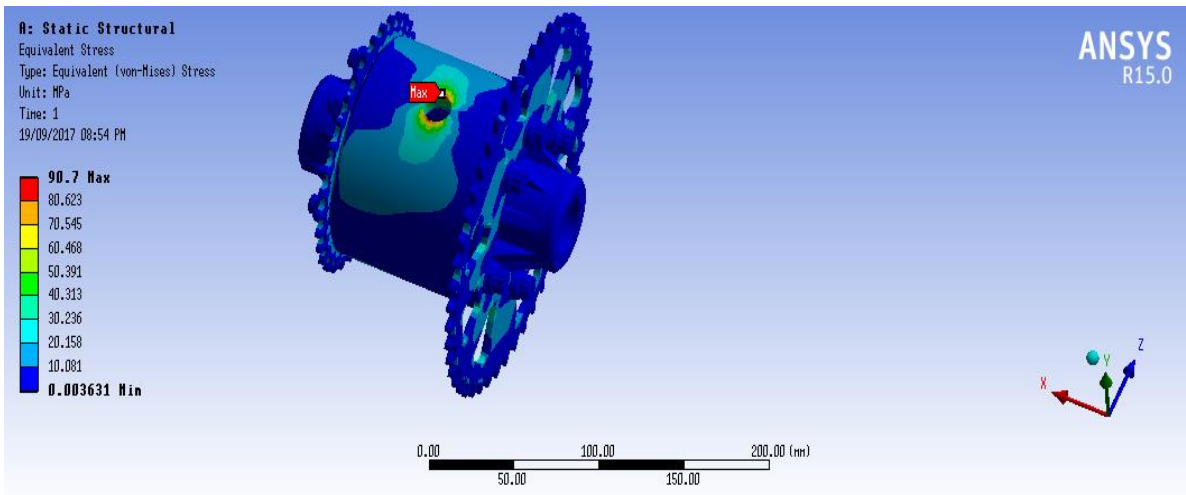


Figure 4.52 Differential casing assembly simulation result, a) Total deformation, b) Von-Mises stress, c) factor of safety.

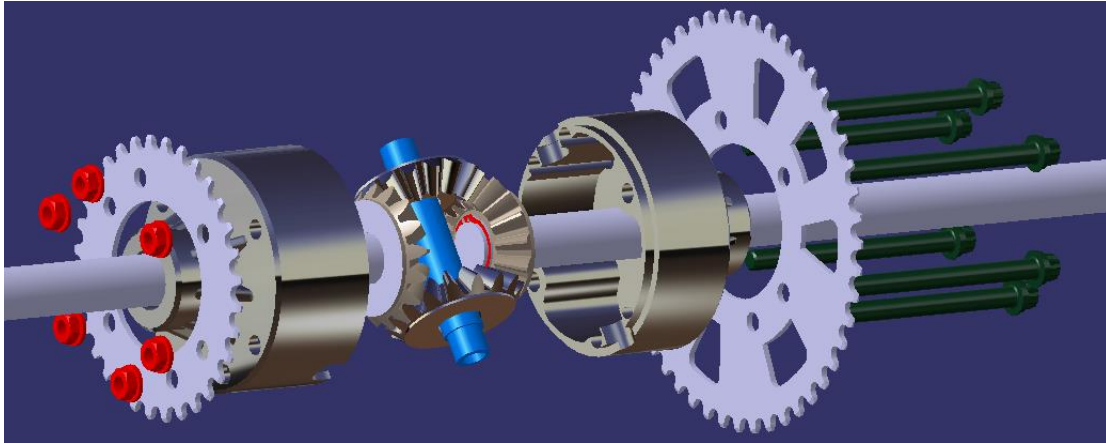


Figure 4. 53 Final drive assembly exploded view

Bearing selection for axle shaft

The purpose of a bearing is to support a load while permitting relative motion between two elements machine. This project vehicle rear axle shaft assembled to frame support plate in order to supporting total weight of cycle rickshaw and transmitting power to road wheel. To accomplish those job, it need bearing. For this thesis work four rhombus flanged units pressed steel housing eccentric locking collar type bearing has been selected from known NTN product for 25mm axle shaft. The features of selected bearing are, it has locking pin to prevent axial movement of axle shaft, both axial and radial load carrying capacity is compatible for this thesis vehicle as shown in table 4.

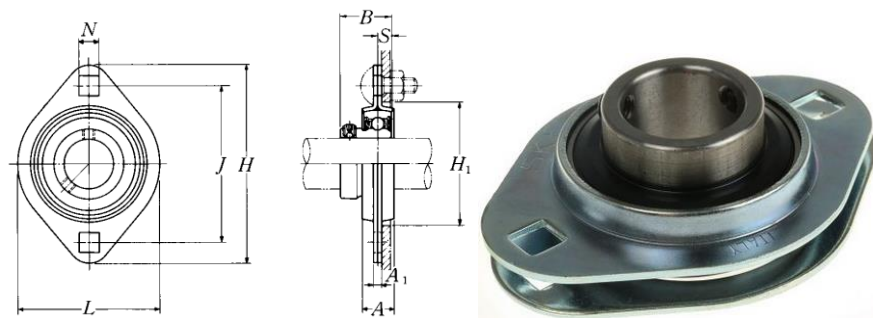


Figure 4. 54 Flanged bearing [48]

Table 4. 23 Flanged bearing specifactions

Shaft dia.	Bearing no.	Nominal dimension (mm)									Bolt size	Max. load (N) recommended	
		H	J	A1	N	A	L	B	S	H1		radial	axial
25	ASPFL205	95	76	4	9	18	71	27	7.5	60	M8	4000	2000

Radial ball bearing for drive hub.

Table 4. 24 Drive hub Radial ball bearing specifactions [48]

Bore (d) mm	Bearing no.	OD (D) (mm)	Width W (mm)	Basic load rating (N)	
				Dynamic	Static
10	6000	26	8	4600	2000

Bearing for differential casing and axle shaft required special when cycle rickshaw negotiating cornering there is relative motion them for this purpose sintered bronze flanged cylindrical bushings selected.

Table 4. 25 Differential casing sintered bronze flanged cylindrical bushings bearing specifactionsn [48]

d (mm)	D (mm)	B (mm)	D1 (mm)	B1(mm)	Bearing no.
18	24	12	30	3	PSMF 182412 A51
25	32	20	39	3.5	PSMF 253220 A51

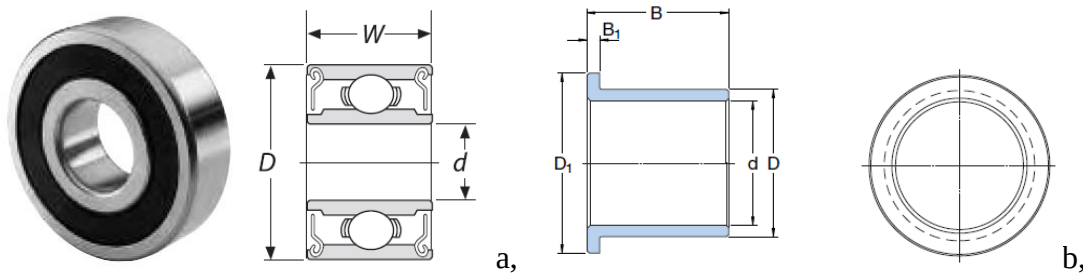


Figure 4. 55. a), Radial ball and b), sintered bronze flanged cylindrical bushings bearing [48].

4.9.13. Rear wheel hub shell

A wheel hub shell is the center part of a cycle rickshaw wheel which attached with rim by spokes. For rear wheel driven cycle rickshaw the hub shell attached to driving axle through key or spline to transmit drive torque to the road wheels in order to propel the cycle rickshaw on the road. The hub shell typically has two machined or forged metal flanges to which spokes can be attached. Each flange has holes or slots to which spokes are affixed. It is wheel structure where all static and dynamic loads of cycle rickshaw transfer to the wheel-tire through the spokes. The hub handles tremendous static and dynamic loads created by the road, the passenger, and rider.

a). Design requirement of hub shell

- It must withstand at static and dynamic load.
- It must transfer power to tire
- Must fit with drive shaft and standard spoke.
- Must compatible with rim.
- Must light and strong.

b). Hub shell geometry

The main reason to modify or design new hub shell is there no compatible existing hub shell for this thesis cycle rickshaw. The new designed hub shell most geometrical features and dimensions is the same as standard except splined parts. Modern bicycle or tricycle have adopted standard axle: the front are generally 80-100mm wide fork spacing, road wheel with freewheel (rear wheel hubs) generally have a 140mm wide. Which allows clearance to mount freewheel, a brake disc on the hub or to decrease the wheel for more durable wheel. However, for this thesis vehicle freewheel is not assembled on hub shell. The thickness of flange taken based on spoke bent size.

Table 4. 26 Hub shell major dimension

			Dimension in mm
1	Hub inner diameter		25
2	Flange	Outer diameter	68
		Thickness	3.5
		Spoke center diameter	60
		Spoke diameter and number	2.6 and (36 spokes)
3	Disc brake flange	Outer diameter	52
		Brake bolt center	44
		M6	6
4	Hub shell width		80

4.9.14. Finite element analysis of hub shell

The finite element of analysis of hub shell was carried in the following steps:

a). Material

Inexpensive carbon steel AISI 1018 has been selected hub shell. The mechanical property of this material is available in table 4.28.

b). Meshing

After the geometry has been generated in CATIA V5 and imported to ANSYS workbench, the 3D model is then meshed in ANSYS workbench design modular. In order to get a better result, locally finer meshing applied in the region which is suspected to have the highest stress that is around the sheet metal. Finally, the model has 354215 numbers of Nodes and 230931 numbers of Elements. The figure is shown.

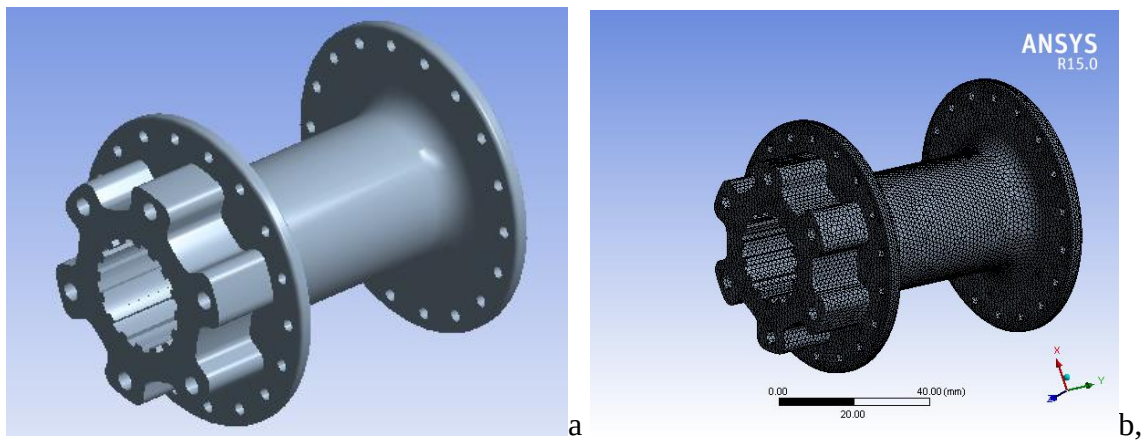


Figure 4. 56 (a) Hub shell model (b) Meshed model of hub shell

c). Boundary and Loading condition

The hub shell is the part of wheel that means, it subjected to both static and dynamic loading. That loads of cycle rickshaw frame weight, passenger weight, rider weight, acceleration, braking, and cornering forces are applied. For this thesis work high speed cornering plus acceleration, and only rear hard braking (lock skidding) condition was considered that is worst scenario. That is the same as axle shaft design consideration. Fixed support are applied to splined hub that because there no relative motion between hub shell and axle shaft. The static load (320kg or 3434N) half load applied to each flange spoke hole radially out ward direction and for hard baking or skidding 628Nm torque applied to disc brake assembly face and bolt hole in the negative y-axis.

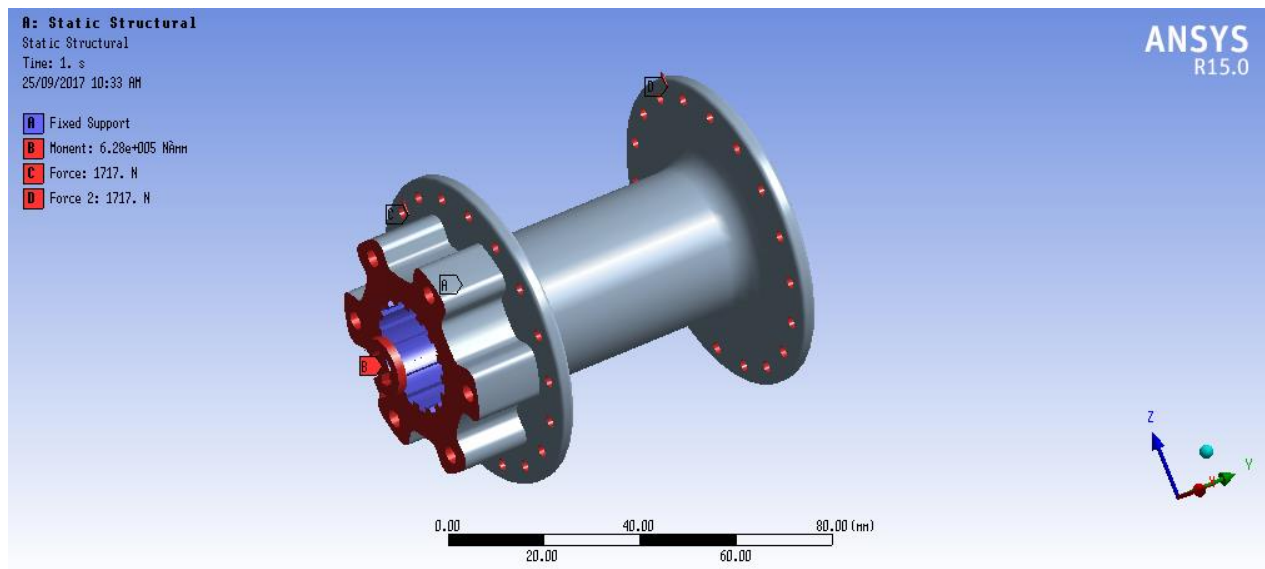
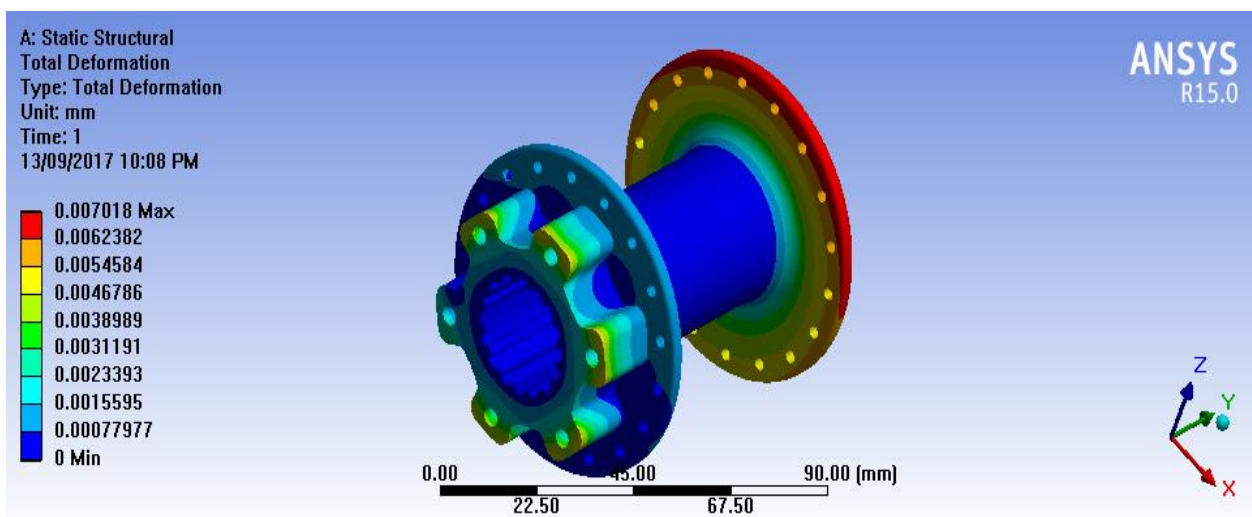


Figure 4. 57 Boundary and Loading condition

wheel hub shell analysis result

The following figures 4.58. shows the result of (a) total deflection, (b) von-mises stress and (c) factor safety in the wheel hub shell when the load on applied single teeth face. The red color shows the Maximum simulation value of those parameters and blue color shows minimum simulation



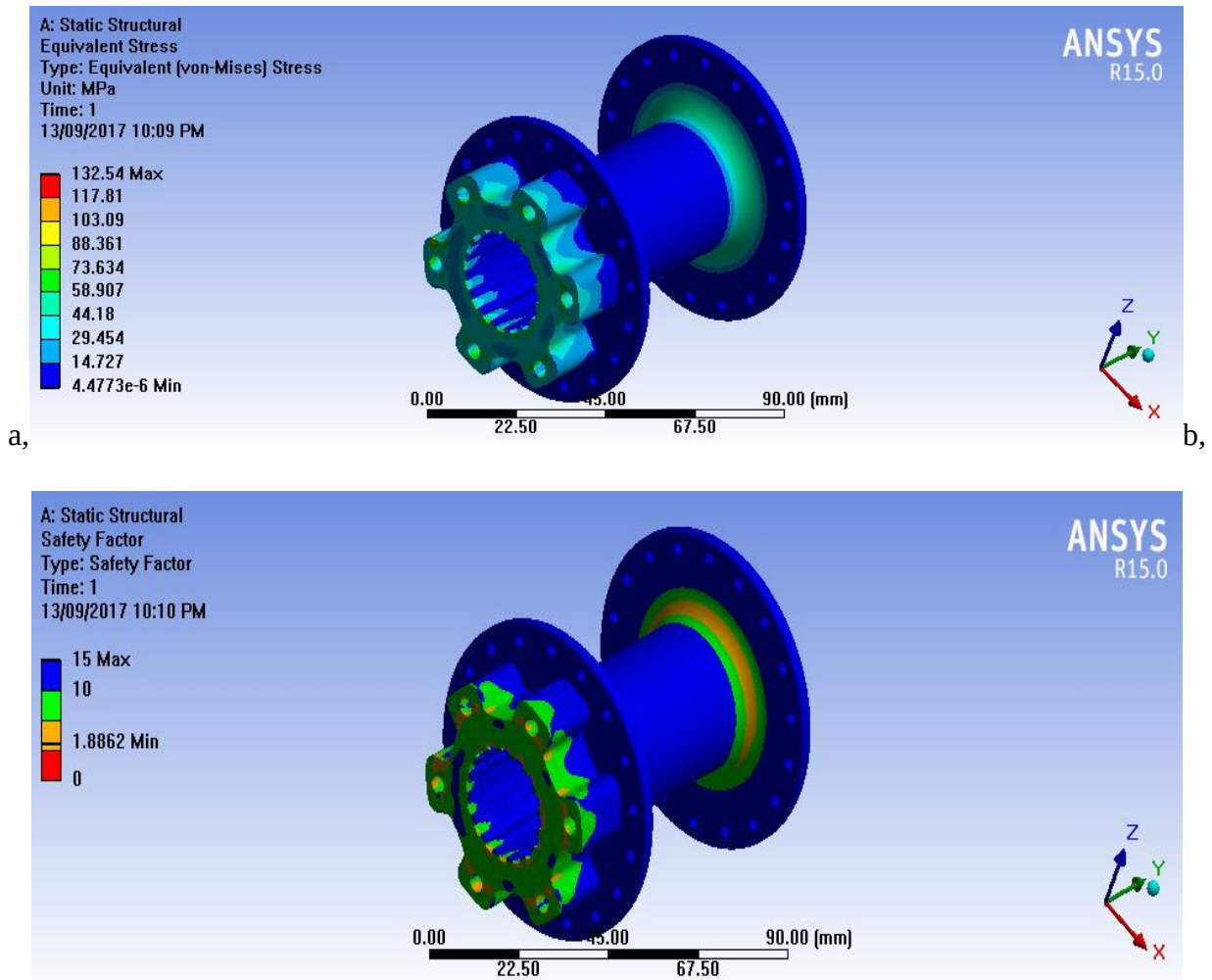


Figure 4. 58 a) Total deformation, b) Von-Mises stress, c) factor of safety

4.10. Kinetic energy recovery system (KERS) design

4.10.1. KERS in cycle rickshaw

In this project KERS used in a cycle rickshaw. It is a complete mechanical system in which flywheel is used for the temporary storage of energy, sprockets, chain and clutch mechanism is for transmitting power from rear wheel axle to flywheel and vice versa. During braking, energy is wasted because kinetic energy of the cycle rickshaw is mostly converted into heat energy or sometimes sound energy that is dissipated into the environment. But KERS converts this kinetic energy into mechanical energy by means of flywheel with assisting in braking simultaneously and then supplies it back to rear wheel axle providing boost or assisting in accelerating. [49] The purpose

of using KERS in cycle rickshaw is to reduce rider effort while accelerating in other word to increase efficiency of cycle rickshaw.

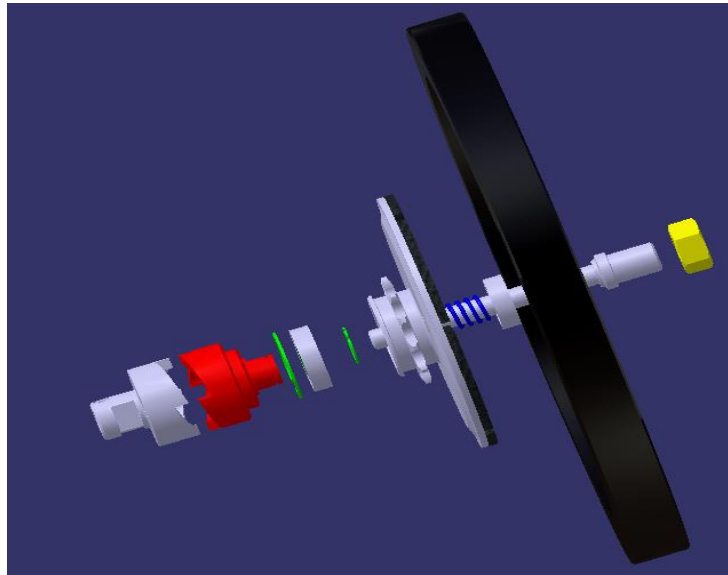


Figure 4.59 KERS of cycle rickshaw.

4.10.2. Working principle of cycle rickshaw KERS

The components used in the system are flywheel, clutch, sprockets, chain, Bearing, frame etc. Flywheel which is mounted on the frame is driven by the bigger differential sprocket through the chain and clutch mechanism. This is being achieved by using chain transmission at a specified gear ratio, final drive sprocket to clutch sprocket helps to increase the overall speed of flywheel. Now at a time when a speed reduction is required, clutch is applied which makes the contact between the clutch and flywheel. Then the flywheel starts rotating, also the speed of cycle rickshaw is decreased this because flywheel rotation consumes energy. Thus a regenerative braking system is achieved. On course energy is stored in flywheel. In case the brake has to be applied fully then after flywheel rotations clutch is disengaged and then the brake is applied. Now when we again ride the cycle rickshaw during which we would apply clutches at this time as final drive rotation is lesser compared to flywheel the energy gets transmitted from the flywheel to the final drive. Now also we can reduce the overall pedaling power required in course of overrides by having clutch fully engaged. We can reduce overall pedaling power. Also situation arises such as traffic jam, down climbing a hill where we do not intend to apply brake fully. For such cases we can apply KERS which would allow us to decelerate and allow us to boost acceleration after this during normal riding and distance that can be covered by pedaling can also improve.

4.10.3. KERS design requirement

There are requirements that need to be met to design a product that is both feasible and optimal. There are also some constraints, both geometric and engineering that also need to be satisfied. The following list describes the requirements and constraints store energy while braking (or decelerating) or downhill climbing

1. Return energy to start up
2. Must fit on a cycle rickshaw frame
3. Light weight
4. Inexpensive and affordable
5. Economical
6. Manufacturability
7. Should not obstruct normal riding.
8. Controlled release.

4.10.4. Components of KERS of the cycle rickshaw

There are different parts used in cycle rickshaw KERS for different purposes like as following.

I. Sprocket

Sprockets are used with chain to transmit power from rear axle to flywheel via clutch and vice versa. Two sprockets must be used. The gear ratio is to be taken in to account here. One sprocket with higher number of teeth is to be selected and another one having lesser number of teeth. The smaller sprocket is to be placed at the flywheel axle end and larger sprocket at rear wheel axle. This is to ensure that to provide maximum flywheel rotations so that energy storage increases.

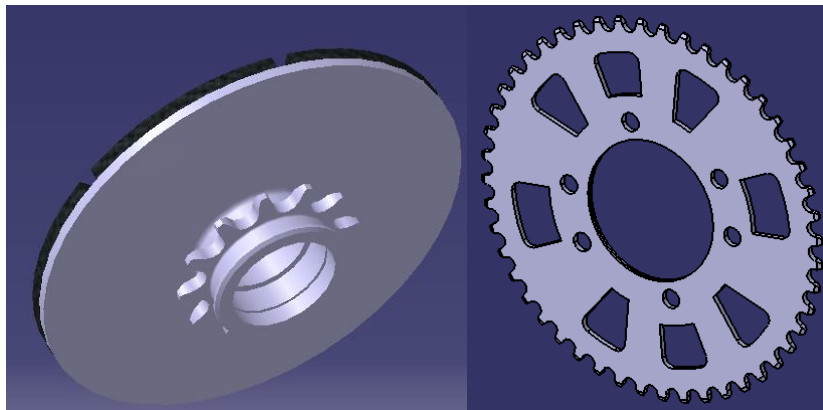


Figure 4. 60 larger and smaller sprocket of KERS

II. Calculation based on sprocket.

Larger sprocket with 48 teeth numbers and smaller sprocket with 12 teeth number are selected.

The gear ratio will be

$$\frac{T_1}{T_2} = \frac{48}{12} = 4:1 \dots\dots\dots 4.30$$

It simply means that it will rotate the flywheel with a speed = (4 times speed of rear wheel axle).

In a cycle rickshaw, the maximum speed is 30km/h, for 24inch tire the rotational speed is 260rpm.

Then

The speed of flywheel will be= $260\text{rpm} \times 4 = 1040\text{rpm}$

Using smaller sprocket on the flywheel end we can get more rotational speed or rpm and so that more energy can be stored because energy stored in flywheel is directly proportional to angular speed. Maximum energy storage in flywheel can be achieved by either increasing the number of the teeth of the larger sprocket or reducing the number of teeth on smaller sprocket.

III. Flywheel

A flywheel is an energy storage device that uses its significant moment of inertia to store energy by rotating. Flywheels have long been used to generate or maintain power and are most identified with the industrial age and the steam engine. In one sense it can be thought of as a rechargeable battery that store energy in the form of mechanical energy instead of electrochemical. [50]

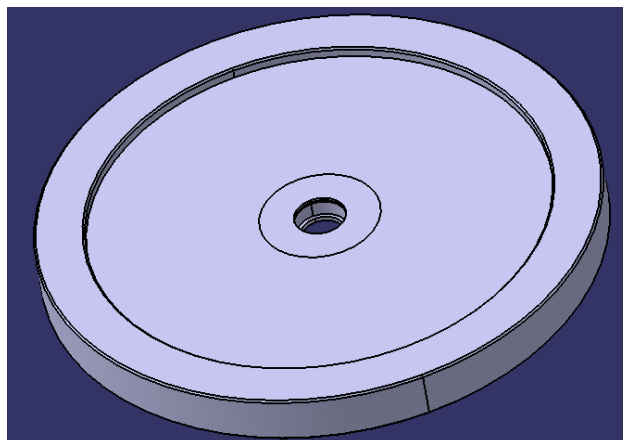


Figure 4.61 Flywheel

4.10.5. Design of flywheel

The equation for the energy stored in a flywheel reads as follows:

$$E = 1/2 I \omega^2 \dots\dots\dots 4.31$$

Where E is the energy (Joules); I is the inertia of the flywheel (kgm^2), and ω is the angular velocity (rad/sec) of the flywheel.

The equation for the inertia of a flywheel is:

$$I = \frac{1}{2} m(r_o^2 - r_i^2) \dots\dots\dots 4.32$$

Where m is the mass of the flywheel; r_i and r_o are the inner and outer radius of the flywheel respectively.

Combining equation (4.31) and (4.32) we get:

$$E = \frac{1}{4} m(r_o^2 - r_i^2) \omega^2 \dots\dots\dots 4.33$$

From equation 4.33, a flywheel's energy is proportional to its mass, and proportional to the square of its rotational speed or angular velocity. In other words, by doubling the mass, the energy stored is also doubled, and by doubling the speed, the energy stored is quadrupled. Thus by increasing the speed of the flywheel it will be possible to reduce the mass and size of it, to a level where its weight is insignificant while analyzing energy efficiency.

Critical stress

Another design consideration to take into account is the stress experienced by the flywheel. This is given by

$$\sigma_t = \rho r^2 \omega^2 \dots\dots\dots 4.34$$

Where σ_t is the tensile stress on the rim of the flywheel; ρ is the density of flywheel material, r is the outer radius of the flywheel and ω is the angular velocity of the rotating flywheel. Usually the key for material selection of a flywheel is the highest possible tensile strength over density. The flywheel can be fabricated using different materials based on the maximum rotational speed requirements and other design constraints. High speed flywheels for speeds above 30000 rpm are

usually composed of high strength carbon fiber. A large mass is not desired for high speed flywheels because extra mass means more energy will be needed to accelerate the vehicle. On the other hand, low speed flywheels with speed values below 20000 rpm, are generally made of steel or other metals for low cost [40]. For a cycle rickshaw flywheel, the size and speed are not likely to even approach a higher enough value to create a stress above the tensile strength of a material, so this consideration is not so important. The tensile strength of standard steel is 250 MPa so it is safe to use as a material of flywheel.

a) Mass of flywheel

Normally energy stored in the flywheel is directly proportional to the mass and radius. Hence increase in mass proves to improve the performance. But as we know that the maximum safe mass that can be used is limited due to frame properties and rider compatibility. And also after some extent the radius can't be increased and the energy storage thus seems to be limited to some particular extend. This is also because of the fact that the total running speed is being reduced due to mass. Energy storage capacity increases with increase in mass but limitation seems to be the speed driving the flywheel. And performance of system is directly linked with the energy stored. Optimum value lies between 5 and 8 kg for human powered vehicles. And radius cannot be more than 20 cm because of the structure of cycle rickshaw. For this thesis work 8kg and 15cm radius fly wheel has been taken.

$$m = \rho v \dots\dots\dots 4.35$$

$$v = 2\pi t(r_o^2 - r_i^2) \dots\dots\dots 4.36$$

$$t = \frac{m}{2\pi\rho(r_o^2 - r_i^2)} \dots\dots\dots 4.37$$

Where :- v is volume and t is thickness

4.10.6. Stored energy on the flywheel at each gear ratio

The stored energy obtain from the equation $= \frac{1}{4}m(r_o^2 - r_i^2)\omega^2$, speed of axle (differential) at each gear. at each gear ratio appendix C. Table C.6 and gear ratio KERS that is four.

Table 4. 27 Flywheel speed at different gear ratio and stored kinetic energy

Differential sprocket speed at each gear.		Flywheel speed (rad/s)		Stored kinetic energy (J o Nm) or $E = \frac{1}{4}m(r_o^2 - r_i^2)\omega^2$ For m=8kg, $r_o = 140mm$, $r_i = 13mm$
rpm	rad/s	Rpm	rad/s	
261	27.3	1044	109.3	464.03
227.4	23.8	909.6	95.2	352.24
200.7	21.0	802.8	84.0	274.38
179.6	18.8	718.4	75.2	219.72
142.2	14.9	238.1	59.5	137.74
121.8	12.7	204.1	51.0	101.05

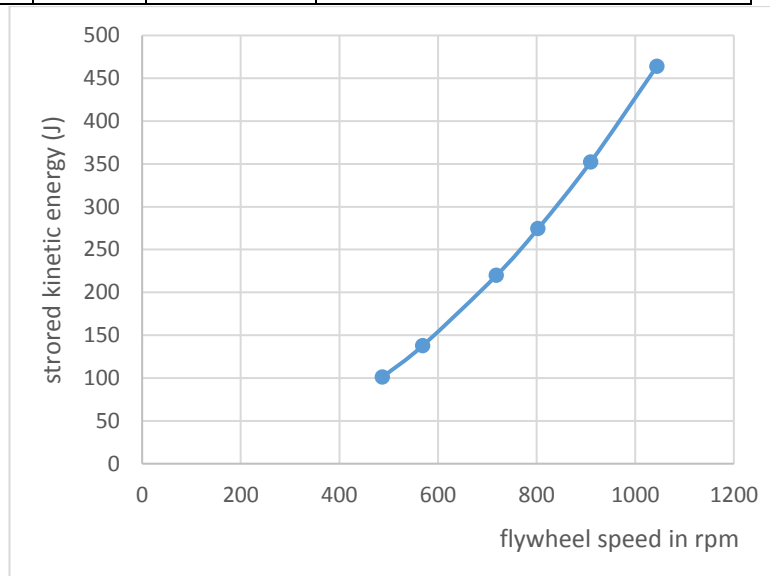


Figure 4. 62 Flywheel speed at different gear ratio versus stored kinetic energy graph

4.10.6. Clutch

A clutch is a device used to connect or disconnect a driven component from the prime mover of the system. Here for this thesis work a clutch has to be provided so as to control the power delivery and release from the KERS. This can be achieved by providing a clutch plate that is linearly moved to and from by applying a lever mechanism incorporated with a spring assembly for providing return mechanism. Linear clutch movements have to be made possible. For this purpose, spiral jaw clutch

can be used. One end of each rod was variably cut. This variable length is female part of another. One part of this is fixed near the frame side. This can be achieved by bolting the part. Another part is made rotatory. This part can be rotated by applying force on it from lever via cable. This moves only partially over fixed one and firstly this is hold in position by aspiring arrangement. A friction disc is used with a solid plate to rotate flywheel [41].

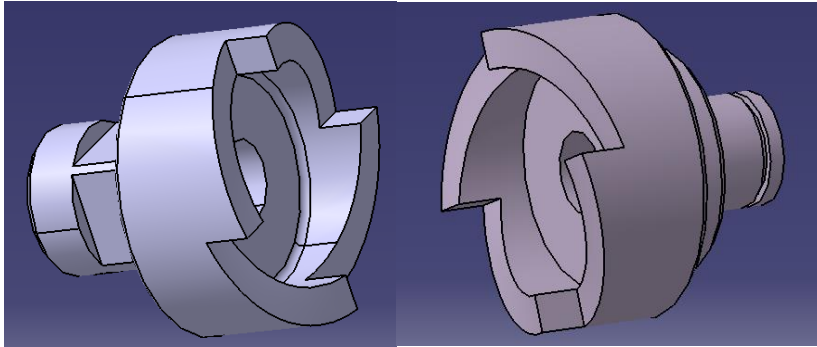


Figure 4. 63 Spiral jaw clutch

b) Friction Clutch design

A friction clutch has its principal application in the transmission of power of shafts and machines which must be started and stopped frequently. Its application is also found in cases in which power is to be delivered to machines partially or fully loaded. The force of friction is used to start the driven shaft from rest and gradually brings it up to the proper speed without excessive slipping of the friction surfaces.

Requirement

- The contact surfaces should develop a frictional force that may pick up flywheel.
- The heat of friction should be rapidly dissipated.
- The surfaces should be backed by a material stiff enough to ensure a reasonably uniform distribution of pressure.

Material for Friction Surfaces

The materials commonly used for lining of friction surfaces and their important properties are shown in the following table.

Table 4. 28 Properties of materials commonly used for lining of friction surfaces. [41]

Friction surface materials	Operating condition	Coefficient of friction	Maximum operating temperature (°C)	Maximum pressure (N/mm ²)
Cast iron on steel	dry	0.15 – 0.20	250 - 300	0.25 – 0.4
steel on steel	In oil	0.06	250 - 300	0.6 – 0.8
Pressed abestos on steel	dry	0.3	150 - 250	0.2 – 0.3
Powder metal on steel	dry	0.4	550	0.3

c) Design calculation

There are two theories to design friction clutch such as uniform pressure, and a uniform axial wear. In case of a new clutch, the intensity of pressure is approximately uniform, but in an old clutch, the uniform wear theory is more approximate. The uniform pressure theory gives a higher friction torque than the uniform wear theory. Therefore, in case of friction clutches, uniform wear should be considered. [40]

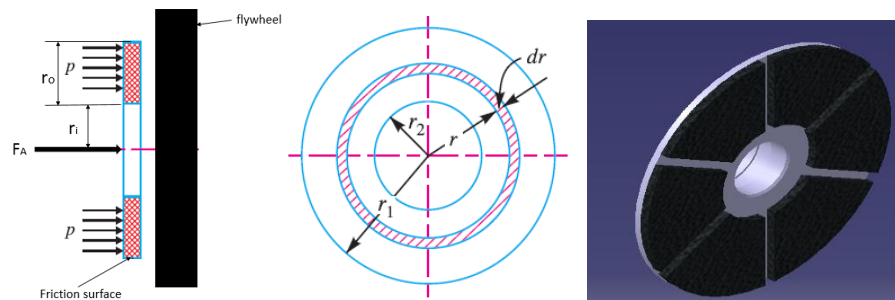


Figure 4.64 Friction clutch free body diagram and its model

Forces on a disc clutch.

Consider an elementary ring of radius r and thickness dr as shown in Fig.

We know that area of the contact surface or friction surface

$$A = 2\pi r dr \dots\dots\dots 4.38$$

Normal or axial force on the ring,

$$\delta F_A = \text{Pressure} \times \text{Area} = p \times 2\pi r dr \dots\dots\dots 4.39$$

and the frictional force on the ring acting tangentially at radius r ,

$$F_f = \mu \times \delta F_A = \mu p \times 2\pi r dr \dots\dots\dots 4.40$$

Frictional torque acting on the ring,

$$T_f = F_f \times r = \mu p \times 2\pi r dr \times r = 2\pi \mu p r^2 dr \dots\dots\dots 4.41$$

When the pressure is uniformly distributed over the entire area of the friction face

$$P = \frac{F_A}{r_o^2 - r_i^2} \dots\dots\dots 4.42$$

Substituting equation 4.42 in 4.41 then integration found

$$T_f = \frac{2}{3} \mu F_A \left[\frac{r_o^3 - r_i^3}{r_o^2 - r_i^2} \right] \dots\dots\dots 4.43$$

Where: -T, Torque transmitted by the clutch, p = Intensity of axial pressure with which the contact surfaces are held together, r_o and r_i is external and internal radii of friction faces, r mean radius of the friction face, and μ coefficient of friction.

4.10.7. Axle

The axle has to be made so as to carry the flywheel and clutch units. The flywheel can be inserted after bearing is added to it and if variable diameter is provided on axle within mid-point the flywheel can be made to be inserted from one end and it automatically locks in the middle of the axle over which it rotates. Also the clutch units sequentially clutch plate and the fixed and moving rods along with its mechanism can be mounted over the axle. The provision for axle placement is provided in the modified frame. The axle should withstand the forces coming to play. In fabrication of axle, first of all the diameter of the shaft is reduced uniformly throughout the length. Step is formed to restrict the linear movement of flywheel. Provisions are also made at both ends to lock it with KERS frame.

I. Design of Axle

The axle subjected to bending load due to weight of the flywheel, clutch and driving system

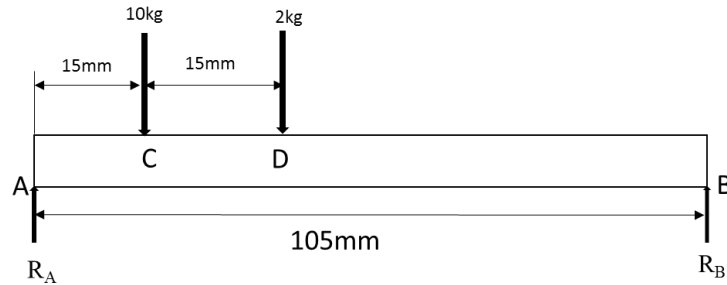


Figure 4. 65 Free body diagram of KERS axle

Balancing forces in vertical direction, by multiply by gravitational acceleration ($9.81m/S^2$)

$$R_A + R_B = 10kg \times 9.81m/S^2 + 2kg \times 9.81m/S^2$$

$$R_A + R_B = 118 N \dots\dots\dots 4.44$$

Where R_A and R_B are the Reaction Forces at point A and B respectively.

Taking moment at point A, $R_B \times 105mm = 98.1N \times 15mm + 19.62N \times 30mm$

$$R_B = 19.6N$$

From eq. (4.38), $R_A = 98.1N$

From diagram we can say that maximum bending moment occurs at point C. So,

$$M = 98.1N \times 0.015m = Nm$$

Consider the dynamic factor and factor of safety 2 for each to axle because is support rotate flywheel

From equation 4.21 and 4.29

Material for axle AISI1015 selected from table 4.28 yield strength of material is 190Mpa.

$$d = \sqrt[3]{\frac{32 \times 1.5 \times 2 \times 2}{\pi \times 190 \times 10^6}} = 0.0873 \text{ m or } 8.73\text{mm take } 10\text{mm}$$

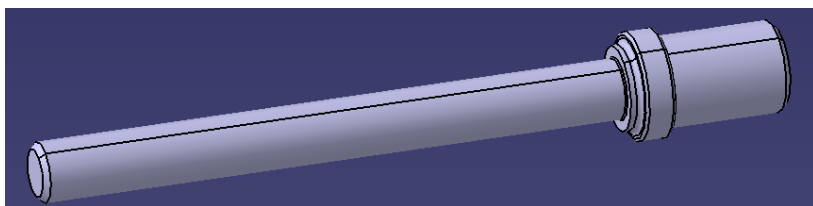


Figure 4. 66 KERS shaft.

4.10.8. Chain

The chains are made up of rigid links which are hinged together in order to provide the necessary flexibility for warping around the driving and driven wheel. The wheels have projecting teeth and fit into the corresponding recesses in the links of the chain. The wheels and the chain are thus constrained to move together without slipping and ensure perfect velocity ratio. Chain mechanism is generally used to avoid slipping. The chains are mostly used to transmit motion and power from one shaft to another, when the distance between the centers of the shaft. Chain selected in appendix C

4.10.9. Other Accessories

I. Bearings

Bearings are placed on the axle with flywheel and clutch plate. It is provided for the free rotation of the flywheel. Two angular contact bearing selected

Table 4. 29 KERS angular contact ball bearing specifactions[48]

Bore (d) mm	Bearing no.	OD (D) (mm)	Width W (mm)	Basic load rating (N)	
				Dynamic	Static
10	6200	26	8	4600	2000
15	6203	32	8	5500	3200

II. Retainer ring

Retainer ring requered in KERS to secure moveable spirial clutch and bearing, sprocket and bearing and flywheel and bearing.

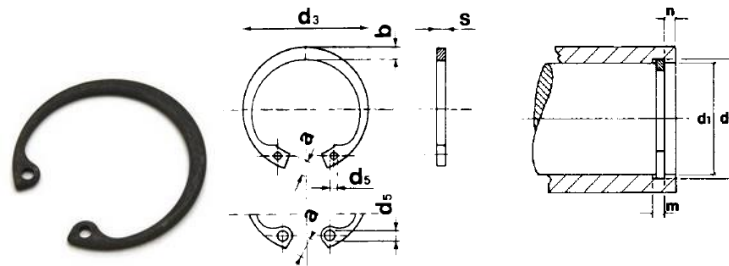


Figure 4.67 Internal retaining ring [47]

Table 4. 30 Internal retaining ring [47]

Nominal dimension in mm						
d_1	s	d_3	a	b	d_5	d_5
28	1.2	30.1	4.8	2.9	2	29.4
32	1.2	34.4	5.4	3.2	2.5	33.7

II. Clutch Cable and Lever

These are the essential components of clutch mechanism. Pressure is applied through the clutch cable by lever to engage the clutch with flywheel. [41]



Figure 4. 68 Clutch cable and lever [41]

III. Springs

This is also an important part of clutch mechanism. Two springs are used in it. Both springs are used for return mechanism, springs are stretched when clutch is engaged and when pressure is removed then springs pull back clutch plate into its initial position. Figure 4.68- spring

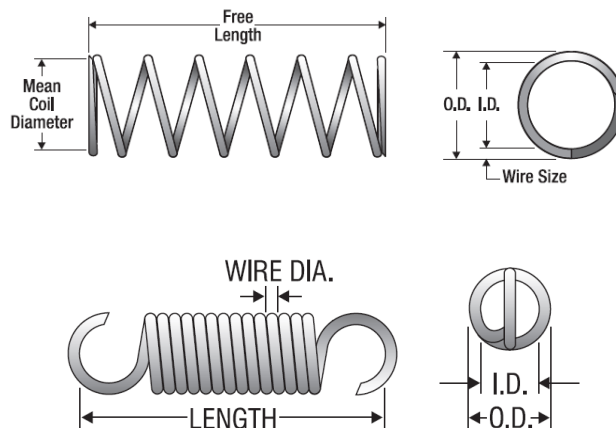


Figure 4.69 Compression and Tension spring [51]

Table 4. 31 Stainless steel compression spring specification [51].

OD (mm)	Stock No.	Free length (mm)	ID(m m)	Rate N/mm	Deflection (mm)	Suggested maximum load (N)	Solid length (mm)	Wire diameter (mm)
14.27	3990	15.9	11.3	6.8	8.1	55	7.7	2

Table 4. 32 Stainless steel tension spring specification [50].

OD (mm)	Stock No.	length h (mm)	Rate N/m m	Suggested maximum deflection(mm)	Initial tension (N)	Suggested maximum load (N)	Wire diameter (mm)
15.06	432	66.7	8.7	20	60	218	2.44

4.11. Cycle Rickshaw Brake

A brake is a device by means of which artificial frictional resistance is applied to a moving machine member, in order to retard or stop the motion of a machine. In the process of performing this function, the brake absorbs kinetic energy of the moving vehicle. The energy absorbed by brakes is dissipated in the form of heat in to environment [41].

4.11.1. Brake system requirements

- ❖ The brakes must be strong enough to stop the cycle rickshaw with in a minimum distance in an emergency.
- ❖ The rider must have proper control over the cycle rickshaw during braking
- ❖ The brakes must have good anti-fade characteristics
- ❖ The brakes should have good anti-wear properties.

The cycle rickshaw brake system will have to provide adequate braking for user-controlled brakes will involve the operator manually controlling the brakes while the cycle rickshaw is in motion.

The brakes will need to accommodate a cycle rickshaw trike traveling at 30km/h while carrying a load of 3434N.

4.11.2. Design Alternatives

When the cycle rickshaw is in operation, a brake system, operated by the rider is required to slow and stop the vehicle. The chosen system will involve one different component. The first component is a user operated hand braking mechanism. For this hand braking mechanism, a brake handle will be mounted to steering handlebar. These brakes will require pressure from the operator to provide adequate power to stop the wheels. In addition to having an operable brake for stopping the vehicle while moving, a parking brake will be required to keep the cycle rickshaw stopped while not in operation. The type of brakes in consideration are listed in Table 4.34 the type of parking brake will be dependent of what brake type is chosen.

Brake types that were considered are rim brakes, drum brakes, and disk brakes. Simple advantages and disadvantages of the brakes are listed below in Table 4.33

Table 4. 33 Advantages and disadvantages of brake options

Brake type	Advantage	Disadvantage
Rim brake	• Lightweight • Low cost • Simple operation	• High maintenance • Poor performance in wet condition • Need for additional parking brake
Drum brake	• Weather resistant • Good stopping power • Very low maintenance	• Risk of over heating • heavy
Disc brake	• good heat dissipation • easy alterations • great on variety of terrain • phenomenal stopping power	• High cost



Figure 4.70 a) Rim, b) drum, and c) disk brake

Three Avid BB7 mechanical disk brakes were used for the vehicle. An image of these brakes, which are attached to the front wheels, is shown in Figure 4.71. The brakes were linked together and actuated by a single lever which allowed the force to be distributed evenly. The disc brakes were used instead of typical friction bicycle brakes (v-brakes) for a number of reasons.



Figure 4. 71 Disc brakes selected for the braking system [42]

4.12. Power needed for a ride cycle rickshaw

The driving resistance of a human powered vehicle consists of rolling resistance, air resistance and grade resistance. The total resistance (road load) is the sum of these three resistance on a slope road such as

Air drag

The air drag or resistance is the force exerted by the air that opposes the motion of a vehicle passing through it or the force which opposes the forward movement of a cycle rickshaw is aerodynamic drag in which air rubs against the exterior vehicle surfaces and forms disturbances about the body, thereby retarding forward movement.

The power P_D needed to overcome air drag or resistance

$$P_D = \frac{\rho_a}{2} C_D A V^3 \dots\dots\dots 4.45$$

where V is velocity in m/s. ρ_a = density of air, $\rho_a \approx 1.18 \text{ kg/m}^3$ for local air. C_D = drag coefficient and A = frontal area. below 25mph in drag force is small. Let assume 0.8

Rolling resistance

A turning tire on the ground generates a longitudinal force called rolling resistance. The force is opposite to the direction of motion and is proportional to the normal force on the tire print.

The power P_R for overcoming the tires' rolling resistances is given by:

$$P_R = \mu_r mgV \dots\dots\dots 4.46$$

Where P_R is rolling resistance power, μ_r is coefficients of rolling resistance, m is total mass including vehicle, rider and passenger, g is gravity, nominally 9.81 m/s^2 and v is vehicle speed.

Rolling resistance coefficient μ_r is due to the deformation work on the tire structure near its contact area and micro slippage of the tire tread on the road when the tire is rolling on it.

Measurements on a roller-mechanism give low-speed coefficients of 0.003 to 0.005 take average value 0.004. [40].

Climbing power

Driving vehicle uphill creates the slope (gradient) resistance power.

The vertical climbing power P_S on slope is given by

$$P_S = Vmg \sin\theta_s \dots\dots\dots 4.47$$

Total resistance Power

Total resistance power is the sum of all resistance power, given by

$$P_{Tr} = P_D + P_R + P_S \dots\dots\dots 4.48$$

The total resistance power is equal the power required to ride flywheel equipped cycle rickshaw. [39].

The total required power equation 4. done on standard level road that means slope power is zero because $\theta_s = 0$ for different speed.

Table 4. 34 Total resistance power

v(km/hr)	P_d (w)	P_r (w)	P_R (W)
0	0	0	0
3	0.31	41.20	41.51
6	2.5	82.40	84.90
9	8.44	123.60	132.04
12	20	164.80	184.80
15	39.06	206.01	245.07
18	67.5	247.21	314.71
21	107.18	288.41	395.60
24	160	329.61	489.61
27	227.81	370.81	598.63
30	312.5	412.02	724.52

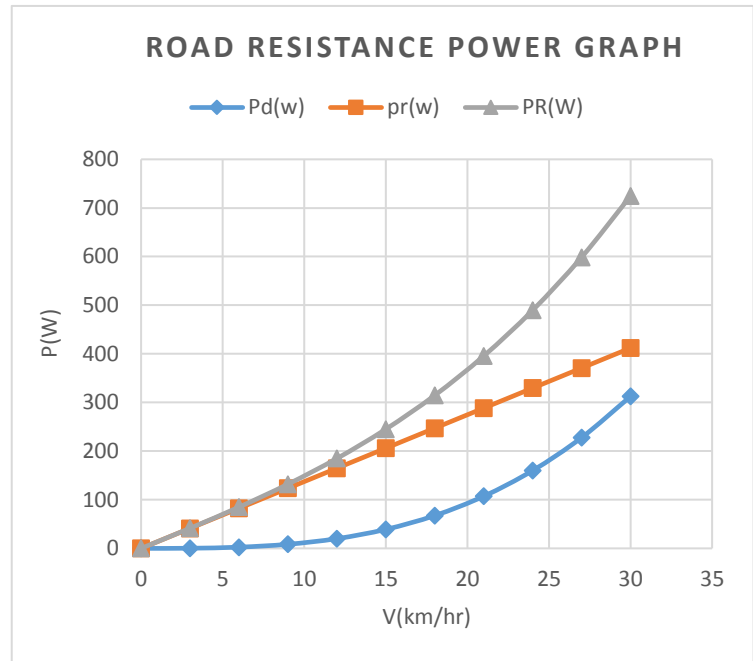


Figure 4.72 Road resistance powers versus speed graph

Result and discussion

The 80kg rider can generate the peak power of 1225W at 100rpm pedaling for 170mm crank arm length of crank set. But the rider sustained with this power generation only for short time (minutes). As human power vehicle study the maximum power that healthy rider can generate for several hours' ranges from 300W to 400W (or 500W depends on rider strength). Based on this fact flywheel equipped cycle rickshaw can be operative by human power. And from table 4.35 and total resistance power graph we can predict that the vehicle can be operated for several hours at 20km/hr by considering 90% efficiency of drive line.

CHAPTER FIVE: RESULT AND DISCUSSION

In this section several numerical simulations present, in order to assess the behavior of the cycle rickshaw frame and its components subjected to various loading conditions.

Deflection, von-mises stress, safety factor etc. are some tools which are used as a resulting tool to show the behavior of the cycle rickshaw frame and its components to applied load conditions.

5.1 Frame static structural analysis result

Table 5. 1 Effect of the meshing on the no. of nodes, no. of elements and results

Method	Nodes	Elements	Von-mises stress (Mpa)
Coarse meshing	30759	15427	95.03
Medium meshing	36477	18235	97.62
Fine meshing	43861	22091	99.10
Meshing of element size 10mm	140998	70503	131.58
Meshing of element size 5mm	179611	89645	138.67
Meshing of element size 4mm	245719	123146	147.74
Meshing of element size 3.5mm	312754	157140	159.32
Meshing of element size 3mm	430832	217457	163.19

Table 5.1 shows the effect of mesh on FE analysis result, which means with more node and element number the accuracy of result increases. The above table data obtained from FE analysis on ANSYS for static startup condition for the same load and boundary condition. But, as shown in table von-mises stress value is different for each meshing. Generally, for this thesis work maximum possible node and element number were used to get better results.

As seen in figure 4.10 and 4.12 in section 4.1.4 static structural analysis of FE for cycle rickshaw frame done on ANSYS workbench for static start up and steady state pedaling condition. From this analysis the result of total deformation, equivalent (von-mises) stress and factor of safety obtained and tabulated below in table 5.2.

Table 5. 2 Frame simulation result

Result	Scenario			
	Static start up		Steady state pedaling	
	Maximum	Minimum	Maximum	Minimum
Total deformation (mm)	0.88	0	0.91	0
Von-Mises Stress (MPa)	163.19	0.0027	167.77	0.00168
Factor of safety	15	2.67	15	2.60

Deformation

From analysis result the total deformation under applied load of rider, passenger, other mechanical system Weight and operational dynamic load the maximum total deformation obtained around front leg mount of passenger seat and magnitude of 0.88mm for static startup and 0.91mm for steady pedaling condition. For both condition maximum total deformation value is less 1mm this indicates cycle rickshaw frame stiffness is feasible under application of those loads.

Von-mises stress

The simulation results for both static startup and steady pedaling condition revealed that the maximum stress or von-mises stress generated on the frame is of the magnitude of 163.19MPa and 167.77Mpa reactively which is less frame material yield strength. these stress found on rear axle mount plate.

Also, The Factor of safety of the frame for static startup and steady pedaling condition on analysis was found to be of 2.67 and 2.6 reactively which means it can be considered practically feasible.

5.2. Passenger seat frame analysis result

As seen in figure 4.17 in section 4.2.3 static structural analysis of FE for passenger seat frame done on ANSYS workbench 15 for static start up and steady state pedaling condition. From this analysis the result of total deformation, equivalent (von-mises) stress and factor of safety obtained and tabulated below in table 5.3.

Table 5. 3 Seat simulation result

Result	Maximum	Minimum
Total deformation (mm)	1.89	0
Equivalent Stress(Von-Mises Stress) (MPa)	68.25	0
Factor of safety	15	3.66

The location of maximum Von Mises stress and minimum factor of safety are just near the seat tube and at the joining portion of back rest tube. The maximum Von-mises stress is about 68.25 MPa and the maximum deflection is about 1.89 mm. The total deformation value of seat frame indicates it is stiff to applied load and minimum factor of safety is 3.66 safe.

5.3. Front steering link analysis result

As seen in figure 4.24 in section 4.3.2.2. static structural analysis of FE for front steering link done on ANSYS workbench 15 for passenger weight load. From this analysis the result of total deformation, equivalent (von-mises) stress and factor of safety obtained and tabulated below in table 5.4.

Table 5. 4 Front steering link simulation result

Result		Maximum	Minimum
Total deformation (mm)		5.82	0
Von-Mises Stress)(MPa)	Bolt	287.51	0
	Fork leg	267.37	0
	Brace	140.86	0
Factor of safety		15	1.62

The maximum deformation found on fork leg blow bottom brace of the magnitude of 5.82mm. The force simulation on the CAD model of the front link revealed that the maximum value of stress (von-mises) generated on bolt, fork leg and brace was of the magnitude of 287.51Mpa, 267.37Mpa and 140.86Mpa reactively which is less than the yield stress (strength) of the models materials.

The Factor of safety of the bolt for front hard braking on analysis was found to be of 3.81 (or 1100Mpa/287.51) which means it can be considered practically feasible. And factor of safety for fork leg on analysis was found to be of 1.62 (435Mpa/ 267.37Mpa) which are considered safe for practical purposes. And factor safety for brace 3.08 (435Mpa/140.86Mpa).

5.4. Drive hub analysis result

As seen in figure 4.37 in section 4.4.2.1. static structural analysis of FE for drive hub done on ANSYS workbench 15 for torsional load condition. From this analysis the result of total deformation, equivalent (von-mises) stress and factor of safety obtained and tabulated below in table 5.5.

Table 5. 5 Simulation result of hub body

Result	Maximum	Minimum
Total deformation (mm)	0.022	0
von mesis stress (MPa)	74.14	0.526
Factor of safety	15	3.37

The total deformation obtained from FEA for drive hub under driving torque of 80kg rider applied on 170mm crank arm of magnitude of 0.022mm. This value indicates drive hub is stiff to applied load. Also, driving torque induce maximum or von mesis stress of magnitude of 74.14Mpa which is less than yield strength of hub material. Therefore, the minimum of factor of safety is 3.37 which means the hub is safe under driving load application.

5.5. Differential gear analysis result.

As seen in figure 4.44 in section 4.4.5.2. static structural analysis of FE for single differential bevel gear was done on ANSYS workbench 15 for driving load from rider and KERS. From this analysis the result of total deformation, equivalent (von-mises) stress and factor of safety obtained and tabulated below in table 5.6.

Table 5. 6 Simulation result of miter bevel gear

Result	Maximum	Minimum
Total deformation (mm)	0.023	0
von mesis stress(MPa)	596.63	0.0026
Maximum shear stress(MPa)	305.6	0.00149
Factor of safety	15	1.17

The total deformation obtained from FEA for single differential gear under driving torque of twice of 80kg rider applied on 170mm crank arm of magnitude of 0.023mm. This value indicates differential gear teeth is stiff to applied load. Also, driving torque induce maximum or von mesis stress of magnitude of 596.63Mpa which is less than yield strength of gear material. Therefore, the minimum of factor of safety is 1.17 which means the gear teeth is safe under driving load application. But safety factor value close to one, however, the it is acceptable because differential split total torque into equal to set of bevel gears and KERS torque is less the torque generated by rider this also increase factor of safety of gear teeth. Which mean the factor of safety is twice of 1.17 i.e. 2.34 it is safe.

5.6. Differential casing assembly analysis result

As seen in figure 4.52 in section 4.4.9.1. static structural analysis of FE for differential casing was done on ANSYS workbench 15 for driving load from rider and KERS. From this analysis the result of total deformation, equivalent (von-mises) stress and factor of safety obtained and tabulated below in table 5.7.

Table 5. 7 Simulation result of differential casing assembly

Result	Maximum	minimum
Total deformation (mm)	0.051	0.00021
Equivalent(von mesis) stress(Mpa)	90.7	0.0036
Factor of safety	15	1.86

The total deformation obtained from FEA for differential casing assembly under driving torque and bolt pretension load application of magnitude of 0.0051mm on KERS sprocket. This value indicates sprocket is stiff to applied load. Also, driving torque induce maximum or von mesis stress of magnitude of 90.7Mpa which is less than yield strength of differential casing material. Therefore, the minimum of factor of safety is 1.86 which means the differential casing is safe under driving load application. And for sprocket and bolt minimum factor of safety greater than 1.86.

5.7. Wheel hub analysis result

As seen in figure 4.58 in section 4.4.10.1. static structural analysis of FE of wheel hub shell done on ANSYS workbench 15 for wheel lock condtion. From this analysis the result of total deformation, equivalent (von-mises) stress and factor of safety obtained and tabulated below in table 5.8.

Table 5. 8 Simulation result of hub shell

Result	maximum	minimum
Total deformation (mm)	0.007	0
Equivalent (von-mises) stress(MPa)	132.54	4.47×10^{-6}
Maximum shear stress(MPa)	76.46	2.49×10^{-6}
Factor of safety	15	1.88

After solving the model, output results in table 5.8 shows total displacement and stresses of the rear wheel hub shell maximum total deformation of magnitude of 0.007mm on outer side flange. Von-mesis stress, and minimum factor of softy are found as 132.54Mpa, and 1.88 respectively on brake disc mount face. The obtained results that the stresses induced in materials are within their allowable limits. The design considers largest load on a hub shell is created by the tension of the spokes, static and dynamic load experienced during operation.

5.8 kinetic energy recovery system result

In section 4.5.10 analytical result of maximum kinetic energy stored in flywheel presented in table 4.33. These indicates maximum energy restored in flywheel directly increase speed under limit of mass and radius of flywheel. Increasing speed of flywheel increase size of driving sprocket. The

maximum value obtained at top speed of cycle rickshaw of magnitude of 464J. This energy can assist rider to boost its speed.

CHAPTER SIX: SUMMARY, CONCLUSIONS AND RECOMMENDATIONS

6.1 Summary and Conclusion

The purpose of this thesis was to design a safe, ergonomic, efficient human powered vehicle. The flywheel equipped cycle rickshaw was designed to move a three persons including rider in most energy efficient manner. That by utilizing kinetic energy lost during braking (or decelerating) motion is captured, stored in a flywheel and later utilized to boost vehicle acceleration.

Some parts for flywheel equipped cycle rickshaw which were selected are similar to up right bicycle. The analysis performed on modified parts. Both existing and newly modified parts are modeled in CATIA V5R20 and imported to ANSYS 15 workbench in the form of IGES for static structural analysis. The modified parts of flywheel equipped cycle rickshaws are main frame with KERS mount, steering link, drive hub body, differential, rear axle, wheel hub shell and KERS.

The frame, seat and roof of the flywheel equipped cycle rickshaw were designed on the basis of anthropometrics, this makes vehicle comfortable for both passenger and rider. Also integrated frame design increases stiffness and reduces the weight of the frame which in turn increases the performance of the vehicle. The frame was made of 35mm circular head tube 30x30 square and 30x35 rectangular AISI4130 steel tube with steering link constructed from 25mm AISI4130 steel tubing that successfully underwent a series of ANSYS static structural tests, it was able to support a 6868N vertical load (both static and dynamic loads). Disc brakes were attached to each wheel and could be activated with a single brake lever to bring the cycle rickshaw to an unassisted. flywheel equipped cycle rickshaw utilized a double line chain drivetrain system linked from a 52 tooth front chain ring and pedal system along the length of the trike to a 6-speed cassette and KERS.

Two idlers protect and guide the chain along the bottom of the frame, keeping it at a distance greater than 200mm from the pavement and maintaining the chain tightness. The maximum flywheel equipped cycle rickshaw speed was 30km/h. though that would increase with the addition of a larger front chain ring.

From the results of FEA, it is apparent that the stresses induced in the cycle rickshaw frame and its components least and the factor of safety is also well above the limit. Also the Von Mises stresses are less than ultimate strength for the material. Thus the design of cycle rickshaw frame and its components is sturdy. The use of ANSYS software makes the process of calculation fast and several iterations are permissible to arrive at the best possible results. The results are relevant provided the assumptions and boundary conditions are perfect.

6.2. Recommendation

As recommendation I put here my feeling to Ethiopian investors, researchers, motor vehicle maintenance and body builder companies, and government to invest their money and knowledge on the sector of human power vehicles to develop eco-friend vehicle, create job opportunity for idle peoples and solve urban transport problems partly, and minimize the loss of our currency.

6.3. Future works

- ✓ For frame, front link and rear axle the impact and modal analysis should be performed in future.
- ✓ If frame analysis done for slope road it will help to determine how the frame behave under that condition to the applied load.
- ✓ A parking brake not considered, addition of it which would help to provide stability when parking the cycle rickshaw on an incline road.
- ✓ Suspension system is part of need improvement.
- ✓ Passenger and rider roof cover strength analysis should be performed.
- ✓ Manufacturing process of all designed and modified components should be performed.
- ✓ KERS got a wide scope in engineering field to minimize the energy loss, but instability caused by flywheel rotation was not studied in thesis.
- ✓ If continuously variable transmission implemented to KER system which would prove in drastic improvement in energy storage.
- ✓ If electric power assistance system coupled to this cycle rickshaws, making the entire system more viable as an alternative transportation method to car.
- ✓ Prototype and test should be performed.
- ✓ If computational fluid dynamics (CFD) analysis is performed for this vehicle it will help to determine actual drag coefficient.

Reference

- [1] Alamgir A. Choudhury, Jorge Rodriguez, Jian Peng He, Ryan Kamm, Saleem Hason Hinton, and Brandon De Youn, "Human powered energy efficient vehicle design", ASEE Annual Conference and Exposition, Conference Proceedings, 2012. (In USA).
- [2] Benjamin Koch, "Pedicab with Fluid Power Regenerative Braking", university of Minnesota, 2010. (In USA).
- [3] Rahman, Mamun Muntasir, D'Este, Glen and Bunker Jonathan M., Non-motorised public transport, Australasian Transport Research Forum, 2010. (In Australasian).
- [4] Tiwari, Geetam and Arora, Anvita and Jain, Himani, "Bicycling in Asia", Interface for Cycling Expertise, (I-ce), Netherlands, 2008.
- [5] Das, Mr.Amarendra Kumar," Design development of an indigenous cycle rickshaw", Ph.D. Thesis, India institute of technology Guwahati, 2008.
- [6] Kumar, Ranjan. "Regenerative Brake: To Harness the Kinetic Energy of Braking." *Journal of Emerging Technologies and Innovative Research*. Vol. 2. No. 1 (February-2015). JETIR, 2015.
- [7] Menon, Sreevalsan S., M. S. Sooraj, Sanjay Mohan, Rino Disney, and Suneeth Sukumaran. "Design and analysis of kinetic energy recovery system in bicycles." *International Journal of Innovative Research in Science, Engineering and Technology* 2013 (2015).
- [8] Cibulka, J. "Kinetic energy recovery system by means of flywheel energy storage." *Advanced Engineering* 3.1 (2009): 27-38.
- [9] Greene, David L., and Michael Wegener. "Sustainable transport." *Journal of Transport Geography* 5.3 (1997): 177-190.

- [10] Black, John A., Antonio Paez, and Putu A. Suthanaya. "Sustainable urban transportation: performance indicators and some analytical approaches." *Journal of urban planning and development* 128.4 (2002): 184-209.
- [11] Ikonen, Markku, "Human power - About the Possibilities to Power Vehicles with Human Muscles," LLP Erasmus Intensive Program: Terrassa, Spain 2011.
- [12] Colin Austin, Miles Graugnard, Max Herrmannsfeldt, Dane Kornasiewicz, and Leif Kjos, "A human powered vehicle", Santa Clara University (USA, California), 2013.
- [13] Schmidlin, Dennis. "Human powered vehicle and drive system." U.S. Patent 5,280,936, issued January 25, 1994.
- [14] Wheeler, Tony, and Richard I'Anson. "*Chasing rickshaws*" Lonely Planet, 1998.
- [15] Steele, M. William. "Rickshaws in South Asia: Introduction to the Special Section." *Transfers* 3, no. 3 (2013): 56-61.
- [16] Replogle, Michael. "Bicycles and cycle-rickshaws in Asian cities: issues and strategies." *Transportation Research Record* (1992): 76-76.
- [17] Wilson, S. S. "Oxtrike." *Appropriate Technology* 3, no. 4 (1977): 21-22.
- [18] Wilson, David Gordon, John Furber, Lawrence M. Halls, and Lauren Howard. "Understanding Pedal Power." (1987).
- [19] Hook, W. "Taj Mahal cycle taxi improvement project: an NGO-private sector partnership." Conference on Public-Private Partnerships in the Transport Sector, Jakarta, Indonesia, June. 1998.
- [20] Azuma, Yoshifumi. *Urban peasants: beca drivers in Jakarta*. Pustaka Sinar Harapan, 2003.
- [21] Chibuka J. "kinetic energy recovery system by means of flywheel energy storage" *Advanced engineering* vol.3, I, pp, 27-38,1998.
- [22] Iba Husain "Electric and hybrid vehicles design fundamental" CRC press, Tylor and Francis Group, USA, 2012.
- [23] Sudipta, S. *Computer aided design & analysis on flywheel for greater efficiency IJAERS/Vol. I. Issue II/299-301 International Journal of Mechanical Engineering and Research*, ISSN 0973-4562 Vol. 5 No. 1, 2015.
- [24] Lavakumar, M., and R. Prasanna Srinivas. "Design and analysis of light weight Motor vehicle flywheel." *International Journal of Computer Trends and Technology (IJCTT)* 4, no. 7 (2013).

- [25] Karwowski, Waldemar. "Ergonomics and human factors: the paradigms for science, engineering, design, technology and management of human-compatible systems." *Ergonomics* 48, no. 5 (2005): 436-463.
- [26] Beshah, Birhanu, Berhane Belay, Shimelis Tilahun Betscha Tizazu, and Amare Matebu. "Anthropometric data of Bahirdar Citys adult men for clothing design." *International Journal of Vocational and Technical Education* 6, no. 5 (2014): 51-57.
- [27] Man, Normal, and Cranfield Man. "Anthropometry and Workspace Design." (2002).
- [28] Moaveni, Saeed. *Finite Element Analysis Theory and Application with ANSYS, fourth edition*. Pearson Education India, 2011.
- [29] Rajput, R. K. *A text book of Automobile Engineering*. Firewall Media, 2007.
- [30] Jürgens, Hans W., Ivar A. Aune, Ursula Pieper, Zulassung Biozide Bundesstelle Chemikalien, and Bundesrepublik Deutschland. *International Data on Anthropometrie: english edition of the research report Internationaler anthropometrischer Datenatlas*. International Labour Office, 1990.
- [31] <http://bikedynamics.co.uk/bikesizingbd.htm>
- [32] [Reza N. Jazar. Vehicle dynamics: theory and application. Spring Street, New York 2008.](#)
- [33] Rakesh Ramakrishna. 'super mileage vehicle with its structural analysis' Department Of Mechanical Engineering, GIT, Belgaum, Jan 21, 2016
- [34] Maestrelli, L., and A. Falsini. "Bicycle frame optimization by means of an advanced gradient method algorithm." 2nd European HTC Strasbourg 2008 (2008): 06.
- [35] Diebschlag, Wilfried, Florian Heidinger, Bernhard Kurz, and Roland Heiberger. *Recommendation for ergonomic and climatic physiological vehicle seat design*. No. 880055. SAE Technical Paper, 1988.
- [36] Lindenberger, ryan. "girder mountain bike fork." PhD diss., University of Cincinnati, 2012.
- [37] Paterek, Tim. *The Paterek Manual for bicycle framebuilders*. Framebuilders' Guild, 1985.
- [38] Bhanage, Amol, K. Padmanabhan, and Rupesh Sundge. "Static and Fatigue Simulation of Telescopic Fork Suspension System Used For Motorcycle." *International Journal of Applied Engineering Research* 10, no. 71: 2015

- [39] D Croccolo*, M De Agostinis, and N Vincenzi” An analytical approach to the structural design and optimization of motorbike forks ” University of Bologna, Bologna, Italy, 16 June 2011.
- [40] Gillespie, Thomas D. "Vehicle Dynamics." *Warren dale* (1997).
- [41] Gupta, J. K., and R. S. Khurmi. "A Textbook of Machine Design." S. Chand Publications, 14th Revised Edition (2005): 1138-1160.
- [42] www.nashbar.com
- [43] Heisler, Heinz. *Advanced vehicle technology*. Elsevier, 2002.
- [44] Rajput, R. K. *A text book of Automobile Engineering*. Firewall Media, 2007.
- [45] Shigley, Joseph Edward. *Shigley's mechanical engineering design*. Tata McGraw-Hill Education, 2011
- [46] Mott, Robert L. *Machine Elements in Mechanical Design* fourth edition, Columbus, Ohio 2004.
- [47] <https://ametric.com/Images/document/RetainingRings-Metric.pdf>
- [48] <http://www.bl.bearings.com>
- [49] Ludlum, Kevin. "Optimizing Flywheel Design for use as a Kinetic Energy Recovery System for a Bicycle." *Senior Theses, Pomona College, Claremont, California* (2013).
- [50] Menon, Sreevalsan S., M. S. Sooraj, Sanjay Mohan, Rino Disney, and Suneeth Sukumaran. "Design and analysis of kinetic energy recovery system in bicycles." *International Journal of Innovative Research in Science, Engineering and Technology* 2013 (2015)
- [51] http://www.centuryspring.com/media/wysiwyg/Century/csc_catalog_web.pdf
- [52] Lee, C., C. Chung, and J. Sun. "Uniform Design and Explicit Dynamics Finite Element Analysis in Improving Permanent Deformation of an On-Road Bicycle Frame Undergoing the Drop-Frame Impact Test." In *International Conference of Electrical, Automation and Mechanical Engineering (EAME 2015)*, pp. 688-691. 2015.
- [53] Lessard, Larry B., James A. Nemes, and Patrick L. Lizotte. "Utilization of FEA in the design of composite bicycle frames." *Composites* 26, no. 1 (1995): 72-74.
- [54] Tayadea, Bharati A., and T. R. Deshmukhb. "A study on structural health of bicycle frame using Finite Element Analysis." *lateral* 2, no. 4 (2015).

- [55] Kurlapkar, Rohan R., M. M. Mirza, and V. M. Naik. "Design and Static Structural Analysis of Bevel Gear."
- [56] Gupta, Mr Rajeev, and Mr GVR Seshagiri Rao. "Analysis of Mountain Bike Frame By FEM."
- [57] Kumar, Vivek, and Prakash Verma. "FINITE ELEMENT ANALYSIS OF PEDAL POWER HUB DYNAMO."
- [58] Ramamurti, V., T. Sukumar, S. Mithun, N. Prabhakar, and P. V. Hudson. "Design analysis of Hub, Rim and Drum in Brake Assembly." *Mechanical Engineering Research* 3, no. 1 (2013): 170.
- [59] Callister, William D., and David G. Rethwisch. *Materials science and engineering*. Vol. 5. NY: John Wiley & Sons, 2011.

Appendix A: Anthropometrics Data

Table A. 1 Anthropometric Length Data of Body Segments (Standard Human), all dimension in mm [30].

Name		5 th %	50 th %	95 th %
Stature	1	1510	1690	1810
Eye height	2	1420	1603	1750
shoulder height	3	1260	1424	1570
Inside leg height (inseam)	4	709	816	890
Sitting height	5	790	905	985
Eye height (sitting)	6	680	790	860
shoulder height (sitting)	7	510	623	695
Elbow height	8	190	243	280
Thigh clearance	9	112	146	170
Seat depth	10	430	499	560
Buttock-knee length	11	543	604	664
Knee height	12	460	530	602
Lower leg length	13	380	444	495

Shoulder breath(biacromial)	14	365	400	430
Shoulder breath bi-deltoid	15	395	474	485
Elbow to elbow breath	16	390	478	540
Hip breath	17	333	360	410

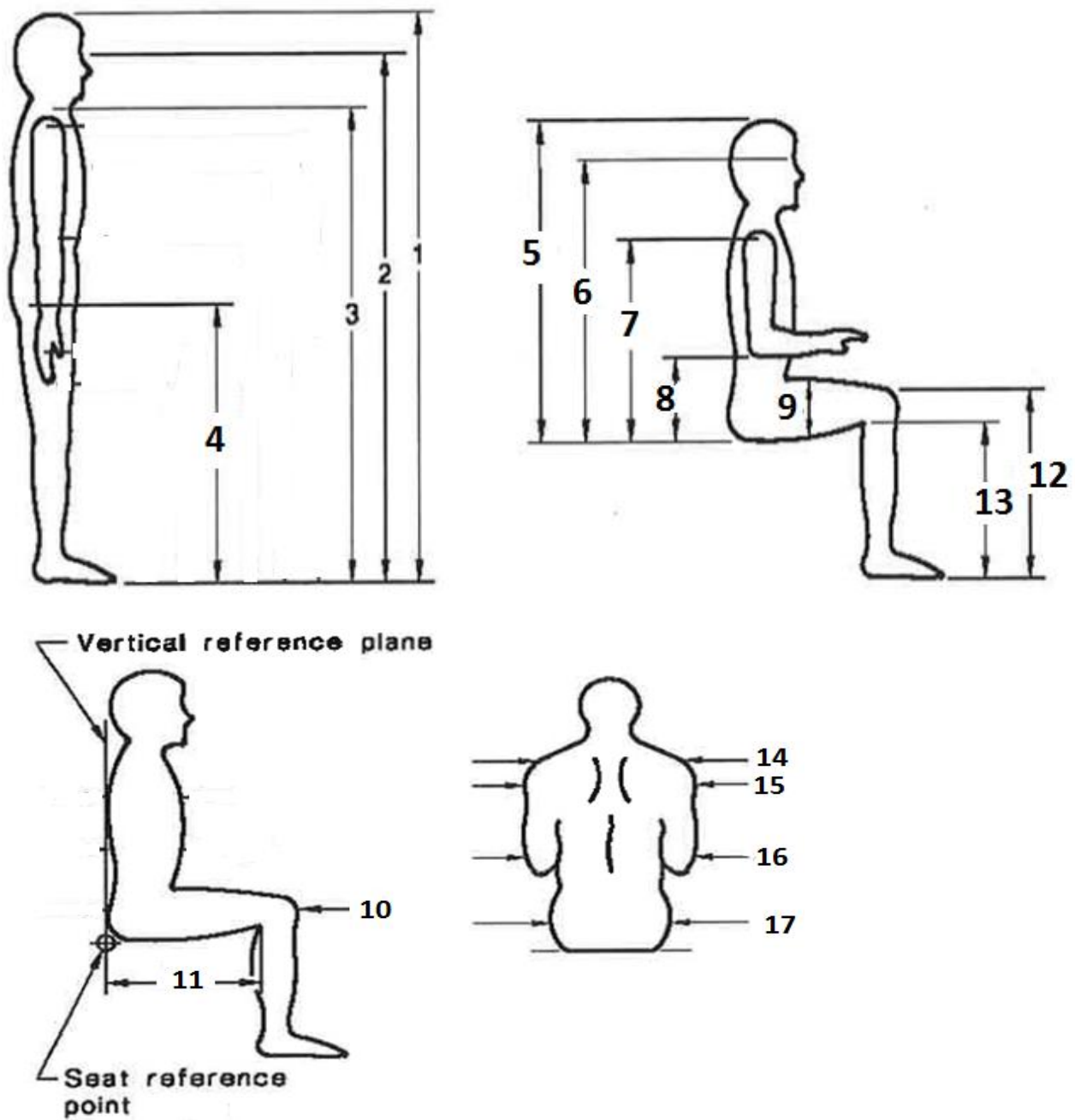


Figure A. 1 Human Body Segments anthropometric [30]

Appendix B: Detail calculation of cycle rickshaw dynamics

Weight transfer when cycle rickshaw Accelerating on a Level Road

The vehicle is considered as a rigid body that moves along a horizontal road. The force at the tire print of each tire may be decomposed to a normal (F_z) and a longitudinal force(F_x). The equations of motion for the accelerating cycle rickshaw come from Newton's equation in x-direction and two static equilibrium equations. See accelerating cycle rickshaw free body diagram figure B.1.

$$\sum F_x = m\vec{a} \dots\dots\dots \text{B. 1}$$

$$\sum F_z = 0 \dots\dots\dots \text{B. 2}$$

$$\sum M_y = 0 \dots\dots\dots \text{B. 3}$$

Expanding the equations of motion produces three equations for four unknowns F_x , F_{z_1} ,

F_{z_2} . Take M_y counter clockwise direction.

$$2F_x = m\vec{a} \dots\dots\dots \text{B. 4}$$

$$F_{z_1} + 2F_{z_2} - mg = 0 \dots\dots\dots \text{B. 5}$$

$$F_{z_1}b - 2F_{z_2}a + 2F_xh = 0 \dots\dots\dots \text{B. 6}$$

Where:- $2F_x$ is driving force on two rear wheels, F_{z_1} is normal force on front wheels, $2F_{z_2}$ is normal force on two rear wheels, m is mass of cycle rickshaw, $\vec{a}$ is acceleration of cycle rickshaw, a is distance from rear wheel center to C.G., b is distance from front wheel center to C.G., l is wheel base, g is gravitational acceleration and C.G is center of gravity.

However, it is possible to eliminate $2F_x$ by Substitute equation B.4 in to B.6, the we can obtain following equation

$$F_{z_1}a - 2F_{z_2}b + m\vec{a}h = 0 \dots\dots\dots \text{B.7}$$

and solve for the normal force F_{z_2} from B.7 equation

$$F_{z_2} = \frac{1}{2}F_{z_1}b/a + \frac{1}{2}m\vec{a}h/a \dots\dots\dots \text{B.8}$$

Now solve for the normal forces F_{z_1} and F_{z_2} by using equations B.5 and B.8, but $a + b = l$

$$F_{z_1} = mga/l - m\vec{a}h/l \dots\dots\dots \text{B.9}$$

$$F_{z_2} = \frac{1}{2}mgb/l + \frac{1}{2}m\vec{a}h/l \dots\dots\dots \text{B.10}$$

The equations B.9 and B.10 contain static and dynamic parts.

The static parts are weight distribution for a stationary cycle rickshaw and depend on the horizontal position of the mass center or (C.G).

$$(F_{z_1})_{static} = mga/l$$

$$(F_{z_2})_{static} = \frac{1}{2}mgb/l$$

However, the dynamic parts indicate the weight distribution according to horizontal acceleration, and depend on the vertical position of the mass center.

$$(F_{z_1})_{dynamic} = -m\ddot{a}h/l$$

$$(F_{z_2})_{dynamic} = \frac{1}{2}m\ddot{a}h/l$$

When accelerating $\ddot{a} > 0$, the normal forces under the front wheel are less than the static load, and under the rear wheels are more than the static load. Because $\frac{1}{2}m\ddot{a}h/l$ dynamic load transferred to each rear wheels.

Acceleration tipping condition is occurring when all front wheel weight transferred to rear wheel that means $(F_{z_1} = 0)$ also this means static and dynamic load equal on the front wheel tire $(0 = mgb/l - m\ddot{a}h/l)$ the equals $(mga/l = m\ddot{a}h/l)$.

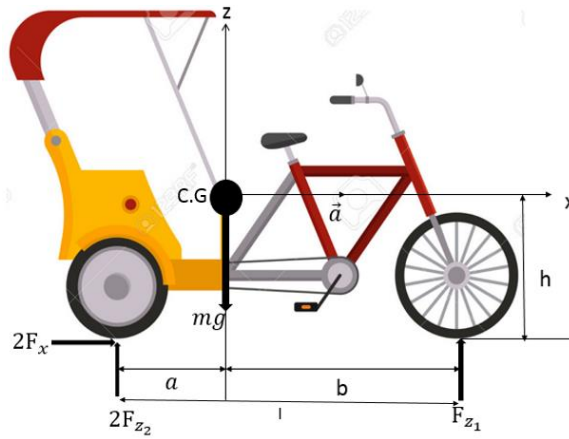


Figure B. 1 The figure shown above displays the accelerating weight transfer calculations and equations used in design analysis of a human powered Vehicle.

Braking weight transfer calculation

During braking cycle rickshaw is decelerating to final speed so as we know deceleration means negative acceleration.

Equilibrium equations.

$$\sum F_x = m(-\vec{a}) \dots\dots\dots \text{B. 11}$$

$$\sum F_z = 0 \dots\dots\dots \text{B. 12}$$

$$\sum M_y = 0 \dots\dots\dots \text{B.13}$$

Expanding the equations of motion produces three equations for four unknowns F_x , F_{z_1} ,

F_{z_2} . Take M_y counter clockwise direction.

$$-F_{x_1} - 2F_{x_2} = m(-\vec{a}) \dots\dots\dots \text{B.14}$$

$$F_{z_1} + 2F_{z_2} - mg = 0 \dots\dots\dots \text{B.15}$$

$$F_{z_1}b - 2F_{z_2}a + (-F_{x_1} - 2F_{x_2})h = 0 \dots\dots\dots \text{B.16}$$

Where: $-F_{x_1}$ front wheel brake force, $2F_{x_2}$ brake force on two rear wheels.

However, it is possible to eliminate $(-F_{x_1} - 2F_{x_2})$ by Substitute equation B.14 in to B.16, then we can obtain following equation

$$F_{z_1}a - 2F_{z_2}b - m\vec{a}h = 0 \dots\dots\dots \text{B.17}$$

and solve for the normal force F_{z_2} from A.7 equation

$$F_{z_2} = \frac{1}{2}F_{z_1}b/a - \frac{1}{2}m\vec{a}h/a \dots\dots\dots \text{B.18}$$

Now solve for the normal forces F_{z_1} and F_{z_2} by using equations A.5 and A.8, but $a + b = l$

$$F_{z_1} = mga/l + m\vec{a}h/l \dots\dots\dots \text{B.19}$$

$$F_{z_2} = \frac{1}{2}mgb/l - \frac{1}{2}m\vec{a}h/l \dots\dots\dots \text{B.20}$$

The equations B.19 and B.20 contain static and dynamic parts.

The static parts are weight distribution for a stationary cycle rickshaw and depend on the horizontal position of the mass center.

$$(F_{z_1})_{static} = mga/l$$

$$(F_{z_2})_{static} = \frac{1}{2}mgb/l$$

However, the dynamic parts indicate the weight distribution according to horizontal deceleration, and depend on the vertical position of the mass center.

$$(F_{z_1})_{dynamic} = m\vec{a}h/l$$

$$(F_{z_2})_{dynamic} = -\frac{1}{2}m\vec{a}h/l$$

When decelerating $\vec{a} < 0$, the normal forces under the rear wheels are less than the static load, and under the front wheel are more than the static load. Because $m\vec{a}h/l$ dynamic load transferred to front wheel.

Braking tipping condition is occurring when all real wheel weight transferred to front wheel that means ($F_{z_2} = 0$) also this means static and dynamic load equal on the real wheel tires ($0 = \frac{1}{2}mgb/l - \frac{1}{2}m\vec{a}h/l$) the equals ($\frac{1}{2}mgb/l = \frac{1}{2}m\vec{a}h/l$).

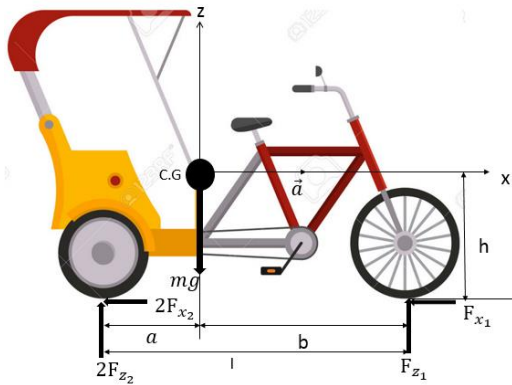


Figure B. 2 The figure shown above displays the braking weight transfer calculations and equations used in design analysis of a human powered Vehicle.

Side weight transfer under cycle rickshaw take road curve

When a cycle rickshaw moves on a curved path, a centrifugal force acts on the cycle rickshaw in outward direction through the center of gravity of the vehicle (Fig B.3)

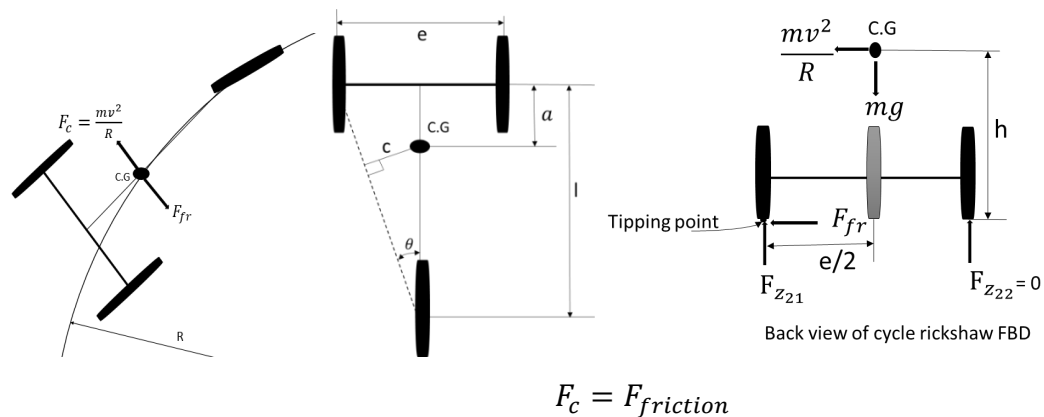


Figure B. 3 The figure shown above displays the tipping point calculations and equations for design analysis.

$$\sin\theta = \frac{c}{l-a} = \frac{e/2}{\sqrt{(e/2)^2+l^2}} \text{ solve for } c \Rightarrow c = \frac{e/2(l-a)}{\sqrt{(e/2)^2+l^2}} \dots\dots\dots \text{B.21}$$

when tipping condition occur all weight from inner wheel transfer to outer wheel which means normal force on inner wheel is zero.

Take moment at tip point in counter clockwise direction.

$$\sum M_{tip} = 0, \frac{mv^2}{R}h - mgc = 0 \dots\dots\dots \text{B.22}$$

Substituting equations B.21 in to B.22 then we have

$$\frac{mv^2}{R}h = \frac{e/2(l-a)}{\sqrt{(e/2)^2+l^2}}mg \text{ and cancel m by m from both side then we get}$$

$$\frac{v^2}{R}h = \frac{e/2(l-a)}{\sqrt{(e/2)^2+l^2}}g \quad \text{or} \quad \frac{v^2}{g} = \frac{e/2(l-a)}{h\sqrt{(e/2)^2+l^2}} \dots\dots\dots \text{B.23}$$

Appendix C: Detail calculation for chain and sprocket selection

C.1. Maximum cranking Torque and Power

C.1.1 Maximum cranking Torque

Torque is measure of a force's tendency to produce rotation. Force is applied to pedals. If randomly trike riders are standing on the pedal and the pedal is parallel to the flat road or crank at 90^0 , then the rider exerting weight force downward on the pedal. Really riders cannot apply their full load into the pedals at all times and the average force will be less, but for design purpose we have to assume maximum loading scenario. The torque is then applied to the chainring which pulls the chain. Therefore, determining torque, chain tension load and power on crank used to design or select appropriate sprocket and chain.

Table C. 1 New cycle rickshaw torque and power calculation input

Mass of rider (kg)	80kg
Crank length (m)	0.17
Gravitational acceleration (m/s^2)	9.81
pedaling rate (rpm)	100

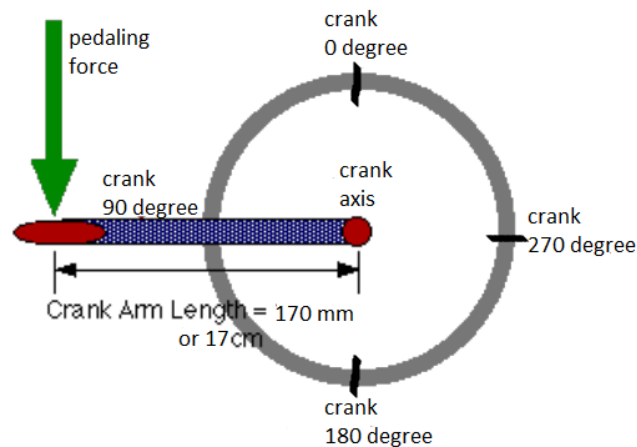


Figure C. 1 Free body diagram of cranking

Torque of crank = pedaling force * crank arm length * sine of crank angle

Maximum torque of crank (T) = weight of rider(w_r) * crank arm length(l_c) * sine 90^0

Or $T = w_r * l_c$ C.1

Maximum torque of crank = $80\text{kg} * 9.81 \frac{\text{m}^2}{\text{s}} * 0.17\text{m} * \sin 90^\circ = 133\text{Nm}$ for single leg in Half cycle

The work done is the weight of the cyclist times twice the crank length since the crank length represents the radius of the circle through which the pedal moves. Remind them that once they obtain a value, they must multiply this by two since both legs are used in one complete cycle that is 234Nm.

C.1. Maximum cranking Power

Power is rate of doing a work. To determine the power generated by trike rider it need to know the torque and maximum cadence (or pedaling rate).

In cycling, cadence (or pedaling rate) is the number of revolutions of the crank per minute; this is the rate at which a cyclist is pedaling/turning the pedals. Cadence is directly proportional to wheel speed, but is a distinct measurement and changes with gearing—which determines the ratio of crank rpm to wheel rpm. The maximum cadence ranges between 100rpm to 120rpm. Using maximum pedaling rate that is racer 100rpm.

power (p) = torque * angular velocity of pedaling

$$p = \frac{2\pi \times T(\text{Nm}) \times N_p(\text{rpm})}{60} \dots\dots\dots \text{c.2}$$

$$= \frac{2\pi \times 133 \times 100}{60} = 1393\text{W or } 1.393\text{KW}$$

where: -p is power

T is torque

N_p is pedaling angular velocity.

C.2 Roller Chain terms

Pitch of chain: - It is the distance between the hinge Centre of a link and the corresponding hinge center of the adjacent link, as shown in Fig. It is usually denoted by p.

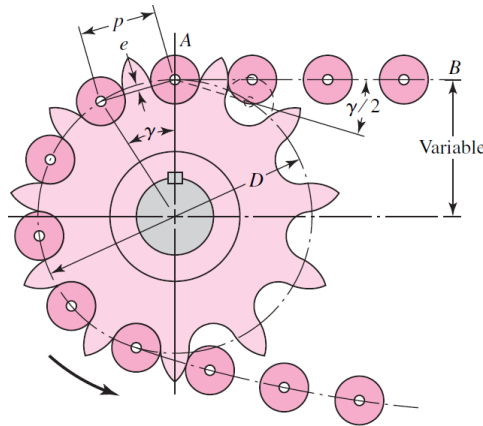


Figure C. 2 Engagement of a chain and sprocket.

Pitch circle diameter of chain sprocket: -It is the diameter of the circle on which the hinge centers of the chain lie, when the chain is wrapped round a sprocket

Denoting the chain pitch by p , the pitch angle by γ , and the pitch diameter of the sprocket by D , from the trigonometry of the figure we see

$$\sin \frac{\gamma}{2} = \frac{p/2}{D/2} \text{ or } D = \frac{p}{\sin(\frac{\gamma}{2})} \dots\dots\dots \text{C.3}$$

Since $\gamma = 360^\circ/T$, where T is the number of sprocket teeth, Eq. (a) can be written

$$D = \frac{p}{\sin(180^\circ/T)} \dots\dots\dots \text{C.4}$$

The angle $\gamma/2$, through which the link swings as it enters contact, is called the angle of articulation.

The sprocket outside diameter (D_o), for satisfactory operation is given by

$$D_o = D + 0.8 d_1 \dots\dots\dots \text{C.5}$$

where d_1 = Diameter of the chain roller.

C.2.1. Velocity Ratio of Chain Drives

The velocity ratio (V.R) of a chain drive is given by

$$V.R = \frac{N_1}{N_2} = \frac{T_2}{T_1} \dots\dots\dots \text{C.6}$$

Where:- N_1 = Speed of rotation of chain wheel or large sprocket in rpm, N_2 = Speed of rotation of small sprocket on cassette in rpm., T_1 = Number of teeth on the chain wheel or large sprocket, and T_2 = Number of teeth on the small sprocket on cassette sprocket.

The maximum speed of vehicle is 30km/hr. or 8.33m/s, the diameter of the tire is 24 inch or 610mm or *effective tire radiud* (R_w) = $\frac{610}{2} = 305mm$ or 0.305m.

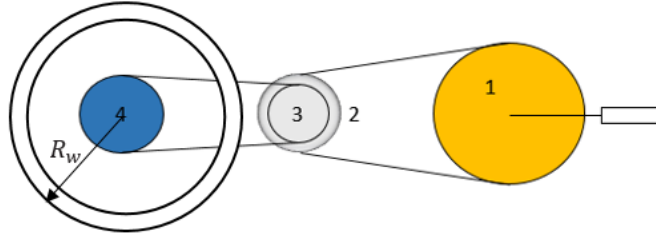


Figure C. 3 Cycle rickshaw drive configuration

The angular velocity in rear sprocket (4) is obtained as

$$N = \frac{60V}{2\pi R_w} \dots\dots\dots C.7$$

Where: - v is velocity (m/s) and N(rpm)

$$N_4 = \frac{60 \times 8.33/0.305}{2\pi} = 261rpm$$

the rear wheel and rear sprocket (4) are mounted in the same the shaft so they rotate with the same speed.

The velocity ratio

$$V.R = \frac{N_2}{N_1} \times \frac{N_4}{N_3} = \frac{T_1}{T_2} \times \frac{T_3}{T_4} \dots\dots\dots C.8$$

the angular velocity of sprocket 2 and 3 are equal or $N_2 = N_3$ because they mount in the same shaft then the above equation reduced to

$$V.R = \frac{N_4}{N_1} = \frac{T_1}{T_2} \times \frac{T_3}{T_4} \dots\dots\dots C.9$$

$N_1 = 100rpm$ that is maximum driver pedaling rate.

$$V.R = \frac{261rpm}{100rpm} = 2.61 = \frac{T_1}{T_2} \times \frac{T_3}{T_4}$$

There are four unknown sprocket teeth number, three obtained from standard bike parts for sprocket 1 or chainring size vary from as few as 20 teeth to as many as 72 take $T_1 = 52$ and for multi-gear bike or cassette the smallest teeth number is $T_2 = 13$ this is most cassette available with smallest teeth.

Standard tooth for differential sprocket or sprocket 4 can be selected based on differential casing size 32 tooth sprocket has been selected.

$$2.61 = \frac{52}{13} \times \frac{T_3}{32} = T_3 = 20.88 \text{ take } 21$$

C.2.3. Roller chain section calculation

Design power(P_D) = Rated power(P) $\times$ Service factor (K_s)

$$P_D = P \times K_s \dots\dots\dots \text{C.10}$$

The service factor (K_s) is the product of various factors K_1 , K_2 and K_3 . The values of these factors are taken as follows:

$$K_s = K_1 \times K_2 \times K_3 \dots\dots\dots \text{C.11}$$

Table C. 2 Chain service factors

service factor	Condition	Value
Load factor (K_1)	Constant load	0.8
	Variable load with mild shock	1.25
	Heavy shock loads	1.5
Lubrication factor (K_2)	Continuous lubrication	0.8
	Drop lubrication	1
	Periodic lubrication	1.5
Rating factor (K_3)	8 hours per day	1
	16 hours per day	1.25
	Continuous service	1.5

Taken value for this thesis work $K_1 = 1.5$, $K_2 = 1.5$, $K_3 = 1$, $K_s = 1.5 \times 1.5 \times 1 = 2.25$

Design power from equation Eq.C.10

$$P_D = 1.393KW \times 2.25 = 3.13KW$$

Now let select chain for rated power (1.393KW) and 261rpm from table C.3 below

Table C. 3 Power rating (in kw) of simple roller chain.

Speed of smaller sprocket or pinion (rpm)	Power (kw)				
	06B	08B	10B	12B	16B
100	0.25	0.64	1.18	2.01	4.83
200	0.47	1.18	2.19	3.75	8.94
300	0.61	1.70	3.15	5.43	13.06
500	1.09	2.72	5.01	8.53	20.57
700	1.48	3.66	6.71	11.63	27.73
1000	2.03	5.09	8.97	15.65	34.89

From the table C.3 08B roller chain is compatible for thesis cycle rickshaw design. From table C.3, we can find that corresponding to a pinion speed of 300-500rpm. the power transmitted for chain No. 8 is 1.7 kW per strand. Therefore, a chain No. 8 with two strands can be used to transmit the required power.

For the 08B chain we find characteristics of roller chains like pitches, roller diameter, Width between inner plates and Breaking load from table C.4.

Table C. 4 Characteristics of roller chains according to IS: 2403-1991

ISO chain number	Pitch (p) mm	Roller diameter (d_1) mm maximum	Width between inner plates (b_1) mm maximum	Transverse pitch (P_1) mm	Breaking load (KN) minimum		
					Simple	duplex	Triplex
05B	8.00	5.00	3.00	5.64	4.4	7.8	11.1
06B	9.525	6.35	5.72	10.24	8.9	16.9	24.9
08B	12.70	8.51	7.75	13.92	17.8	31.1	44.5
10B	15.875	10.16	9.65	16.59	22.2	44.5	66.7
12B	19.05	12.07	11.68	19.46	28.9	57.8	86.7

16B	25.4	15.88	17.02	31.88	42.3	84.5	126.8
20B	31.75	19.05	19.56	36.45	64.5	129	193.5
24B	38.10	25.40	25.40	48.36	97.9	195.7	293.6
28B	44.45	27.94	30.99	59.56	129	258	387
32B	50.80	29.21	30.99	68.55	169	338	507.10
40B	63.50	39.37	38.10	72.29	262.4	524.9	787.3
48B	76.20	48.26	45.72	91.21	400.3	800.7	1201

Pitch, $p = 12.7$ mm, roller diameter $d_1 = 8.51$ mm, width between inner plats $b_1 = 7.75$ mm,
Breaking load, $WB = 17.8$ kN

C.2.4. sprocket Pitch circle and outer diameter calculation

using equation C. 3, 4 and chain 08 characteristics data sprocket Pitch circle and outer diameter is computed in table C.4.

Table C. 5 Sprocket Pitch circle and outer diameter

sprocket	Teeth No.	D $= \frac{p}{\sin(180^\circ/T,)}$	$D_0 = D + 0.8d_1$
1	52	210.3	217.1
2	13	53.07	59.88
3	15	61.08	67.89
4	17	69.11	75.92
5	19	77.16	83.97
6	21	85.21	92.02
7	24	97.3	104.1
8	28	113.4	120.2
9	32	129.57	136.38

C.2.5. Maximum Chain Tension load

During cycle rickshaw riding the power transmitted through chain. The power subject chain to tension load, the load obtained by dividing maximum crank torque to pitch radius of chainring sprocket 105.15mm (or 0.10515m)

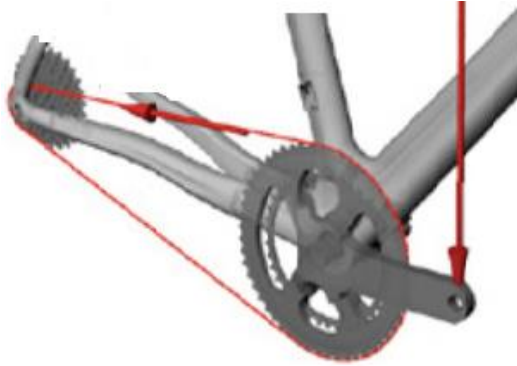


Figure C. 4 Free body diagram of chain tension

$$\text{Chain tension load} = \frac{133Nm}{0.10515m} = 1265N$$

Gear ratio study

In order to corroborate that the chain wheel and cassette are correct for our trike, it has made a gear ratio study. It is the relation between the rate at which the rider pedals and the rate at which the drive wheel turns. It is calculated by the number of teeth of the chain wheel and the number of teeth of the sprocket. As cycle rickshaw has 7 sprockets in the cassette and one chain wheel, there are 6 (6 x 1) different gear ratios.

Final drive gear ratio

first let calculate final drive gear ratio using equation C.6 final drive sprocket teeth number is 21 teeth sprocket on drive hub and 32 teeth sprocket on differential casing.

$$\frac{21}{32} = 0.656 \text{ or } \frac{32}{21} = 1.524$$

that mean reduces the differential speed at each sprocket by 0.656 or increase toque of the differential speed at each sprocket by 1.524.

The torque at each cassette obtain by dividing maximum crank torque to gear ratio. And differential torque obtained by multiplying cassette torque by finial gear ratio (1.524).

Distance covered by one turn of chain wheel calculated by multiplying perimeter of a circumference (pi times tire diameter) of tire (24 inch), gear ratio and final gear ratio. Finally, differential speed calculated by multiplying maximum pedal rpm, gear ratio and final gear ratio.

Table C. 6 Cycle rickshaw gear ratio, torque, speed data

gear	52	Gear ratio $= \frac{52}{\text{cassette sprocket}}$	Torque (Nm) at each cassette sprocket	Torque (Nm) on differential at each sprocket	Differential rpm at each gear.
6 th	13	4	33.25	50.67	261
5 th	15	3.467	38.36	58.46	227.4
4 th	17	3.059	43.48	66.26	200.7
3 rd	19	2.737	48.59	74.05	179.6
2 nd	24	2.167	61.37	93.53	142.2
1 st	28	1.857	71.62	109.15	121.8

Length of Chain and Centre Distance

An open chain drive system connecting the two sprockets is shown in Fig. 21.3

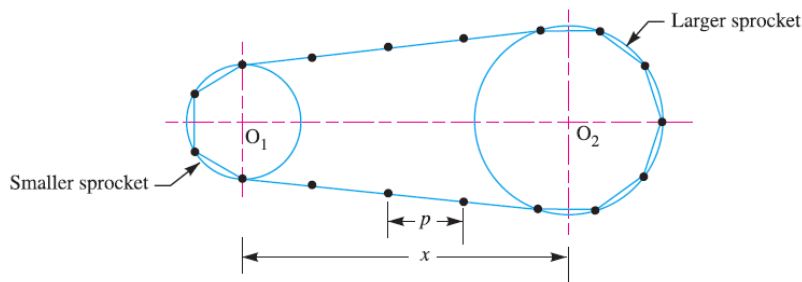


Figure C.5 Length of chain.

The length of the chain (L) must be equal to the product of the number of chain links (K) and the pitch of the chain (p).

$$L = Kp \dots\dots\dots \text{C.12}$$

The number of chain links may be obtained from the following expression, *i.e.*

$$K = \frac{T_1 + T_2}{2} + \frac{2x}{p} + \left[\frac{T_2 - T_1}{2\pi} \right]^2 \frac{p}{x} \dots\dots\dots C.13$$

Where: - T_1 = Number of teeth on the smaller sprocket, T_2 = Number of teeth on the larger sprocket, p = Pitch of the chain, and x = Centre distance.

The center distance obtained for this work from direct measurement from one sprockets to other sprocket center from assembled cycle rickshaw model in CATIA

Table C.7 Length of chain calculation

	Sprocket	X(mm)	k	L(mm)
Crankset to drive hub	$T_1 = 28$	460	112.80	1432 plus dueler length
	$T_2 = 52$			
Drive hub to differential	$T_1 = 21$	766	142.28	1807
	$T_2 = 32$			
Differential to KERS	$T_1 = 12$	298	78.32	994
	$T_2 = 48$			

Appendix D: Terminology and proportion of miter bevel gear

Pitch circle. It is an imaginary circle which by pure rolling action, would give the same motion as the actual gear.

Pitch angle: -It is the angle subtended by pitch line with the axis of the shaft. It is denoted by ‘ θ_p ’

Cone center: - is the apex point of the pitch cone or the point of intersection of the axis of the two mating gear. It denoted by ‘O’.

Back cone angle: -

Pitch diameter: - It is the diameter of the pitch cone. It is denoted by ‘D’.

$$D = ZM \dots\dots\dots D.1$$

wher Z is gear teeth number and M is module unit in mm.

Module. It is the ratio of the pitch circle diameter in millimeters to the number of teeth. It is usually denoted by *M*. Mathematically, Module,

$$M = D / Z \dots\dots\dots D.2$$

Cone distance: - It is the length of the pitch cone element. It is also called as a pitch cone radius. It is denoted by ‘OA’. Mathematically, cone distance or pitch cone radius,

$$OA = \frac{\text{pitch radius}}{\sin \theta_p} = \frac{D/2}{\sin \theta_p} \dots\dots\dots D.3$$

Addendum. It is the radial distance of a tooth from the pitch circle to the top of the tooth. It is denoted by ‘ A_a ’ Mathematically, addendum

$$A_a = 1M \dots\dots\dots D.4$$

Addendum angle. It is the angle subtended by the addendum of the tooth at the cone center. It is denoted by ‘ θ_a ’ Mathematically, addendum angle,

$$\theta_a = \tan^{-1} \frac{A_a}{OA} \dots\dots\dots D.5$$

Dedendum. It is the radial distance of a tooth from the pitch circle to the bottom of the tooth. It is denoted by ‘ D_a ’ Mathematically, addendum

$$D_d = 1.25M \dots\dots\dots D.6$$

Dedendum angle. It is the angle subtended by the dedendum of the tooth at the cone centre. It is denoted by ' θ_d '. Mathematically, dedendum angle,

$$\theta_d = \tan^{-1} \frac{D_d}{OA} \dots\dots\dots D.7$$

Face angle. It is the angle subtended by the face of the tooth at the cone centre. It is denoted by ' α '. The face angle is equal to the pitch angle *plus* addendum angle ($\alpha = \theta_a + \theta_p$).

Root angle. It is the angle subtended by the root of the tooth at the cone centre. It is denoted by ' θ_R '. It is equal to the pitch angle *minus* dedendum angle ($\theta_R = \theta_p - \theta_d$).

Outside or addendum cone diameter. It is the maximum diameter of the teeth of the gear. It is equal to the diameter of the blank from which the gear can be cut. Mathematically, outside diameter,

$$D_o = D + 2 A_a \cos \theta_p \dots\dots\dots D.8$$

Inside or dedendum cone diameter. The inside or the dedendum cone diameter is given by

$$D_i = D - 2 D_d \cos \theta_p \dots\dots\dots D.9$$

Tooth thickness. It is the width of the tooth measured along the pitch circle. Mathematically, thickness of tooth = $1.5708 M$.

Working depth. It is radial distance from the addendum circle to the clearance circle. It is equal to the sum of the addendum of the two meshing gears. Mathematically, working depth = $2 M$

Clearance. It is the radial distance from the top of the tooth to the bottom of the tooth, in a meshing gear. A circle passing through the top of the meshing gear is known as *clearance circle*. Mathematically, Clearance = $0.2 M$

Total depth. It is the radial distance between the addendum and the dedendum circle of a gear. It is equal to the sum of the addendum and dedendum. It is denoted by 'H' Mathematically, total depth given

$$H = A_a + D_d \dots\dots\dots D.10$$

Appendix E: Detail and assembly drawing of cycle rickshaw



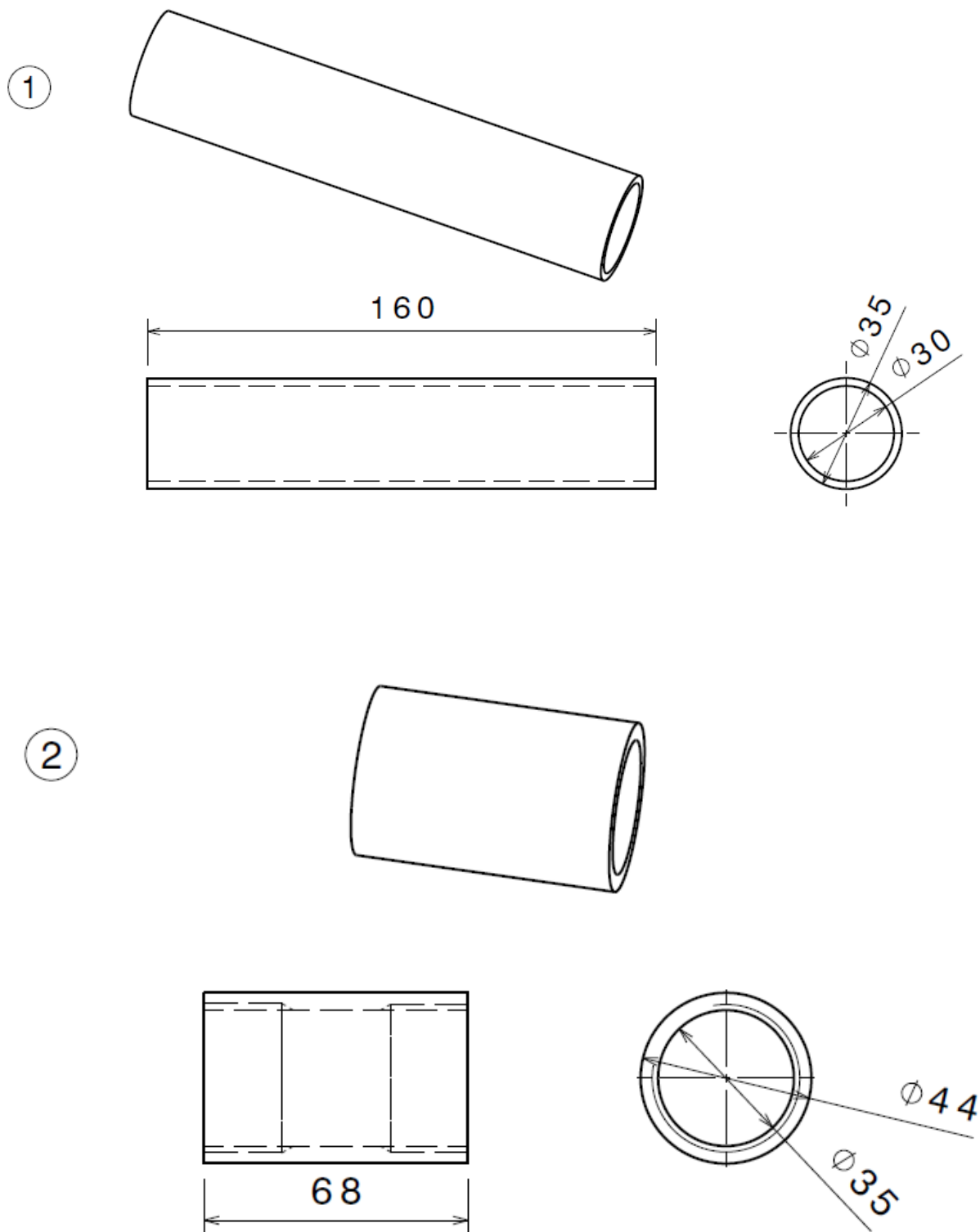


Figure E.1 (1) Head tube and (2) BB shell

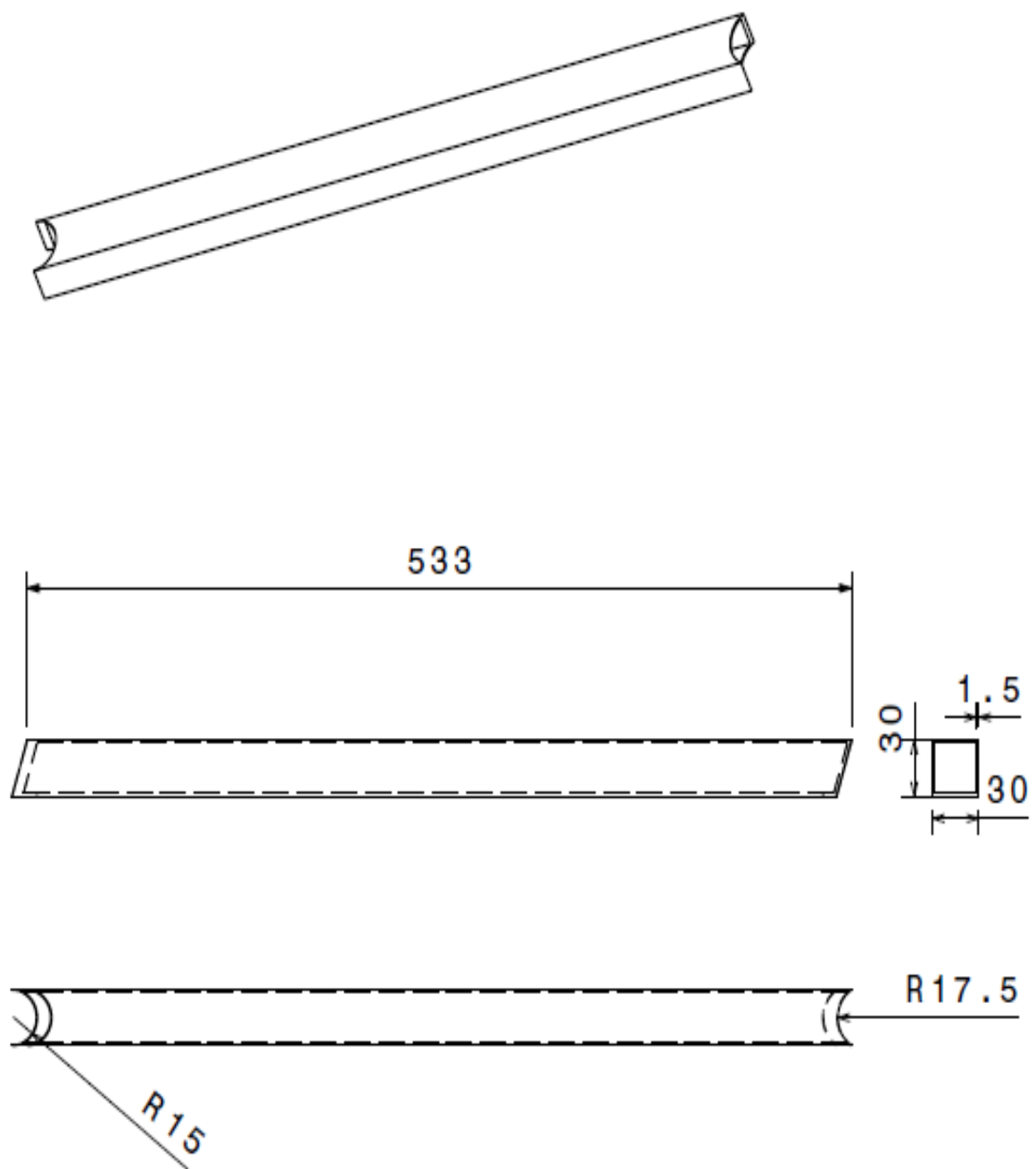


Figure E.2 Top tube

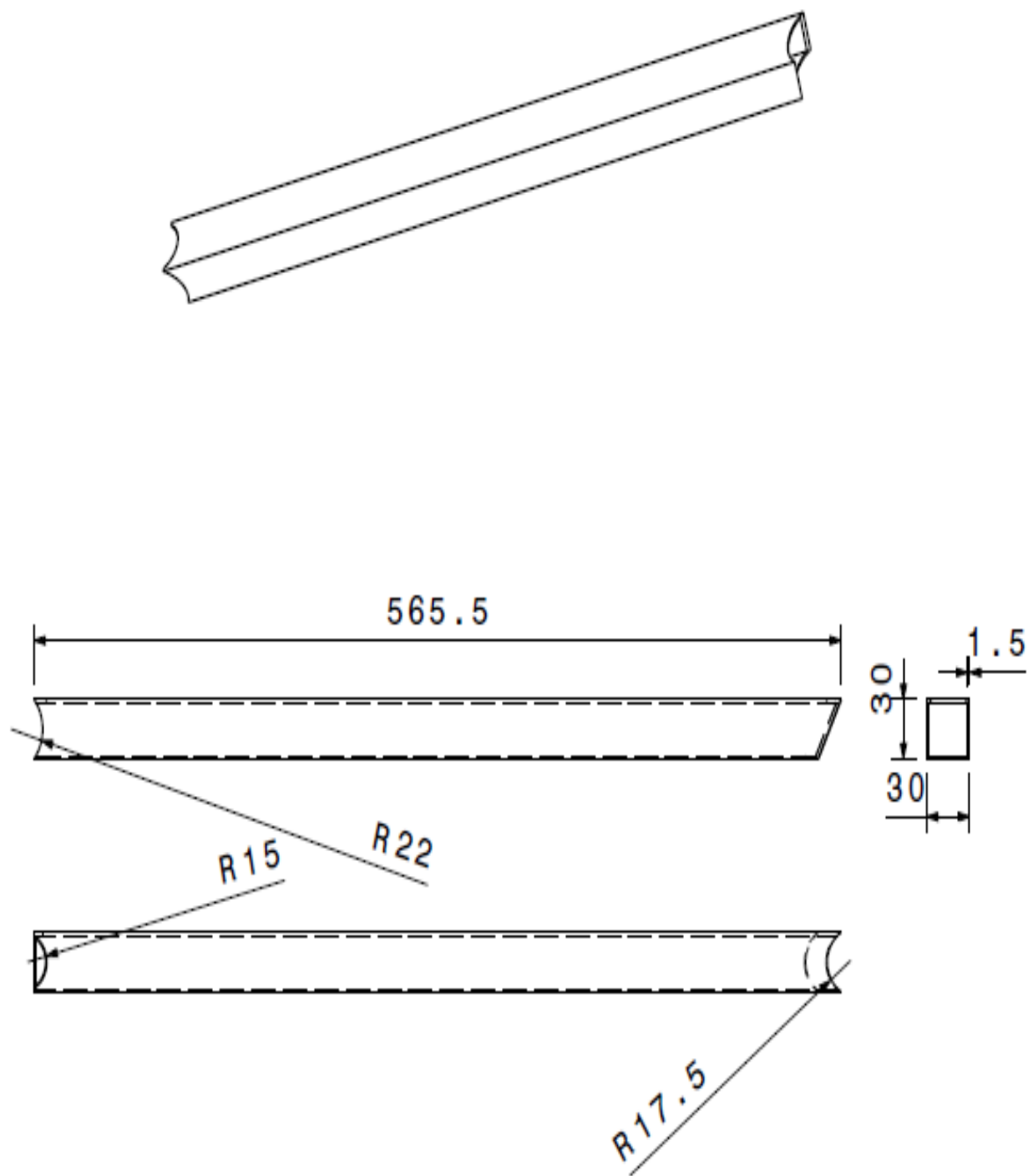


Figure E.3 Down tube

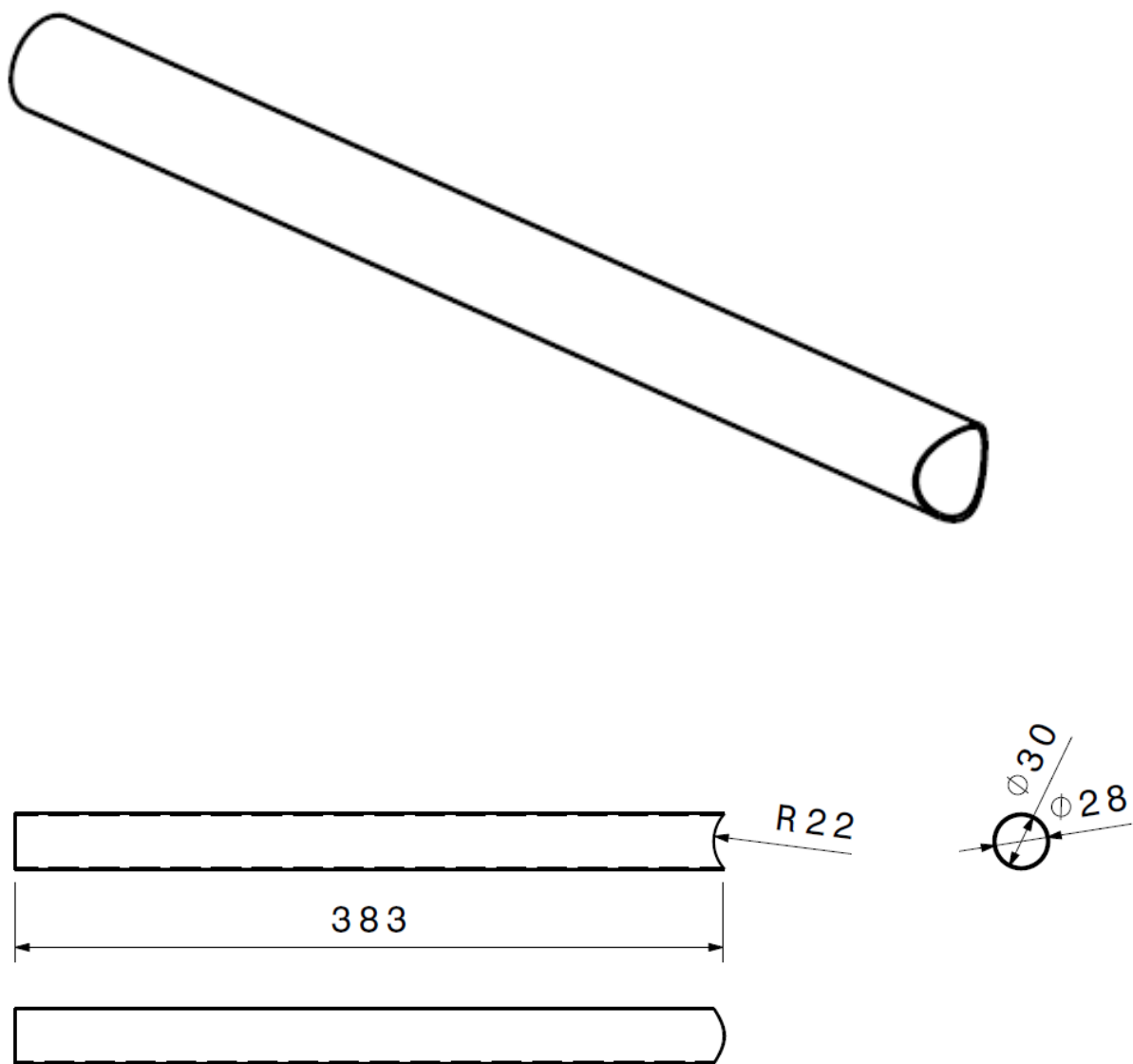


Figure E.4 Rider seat tube

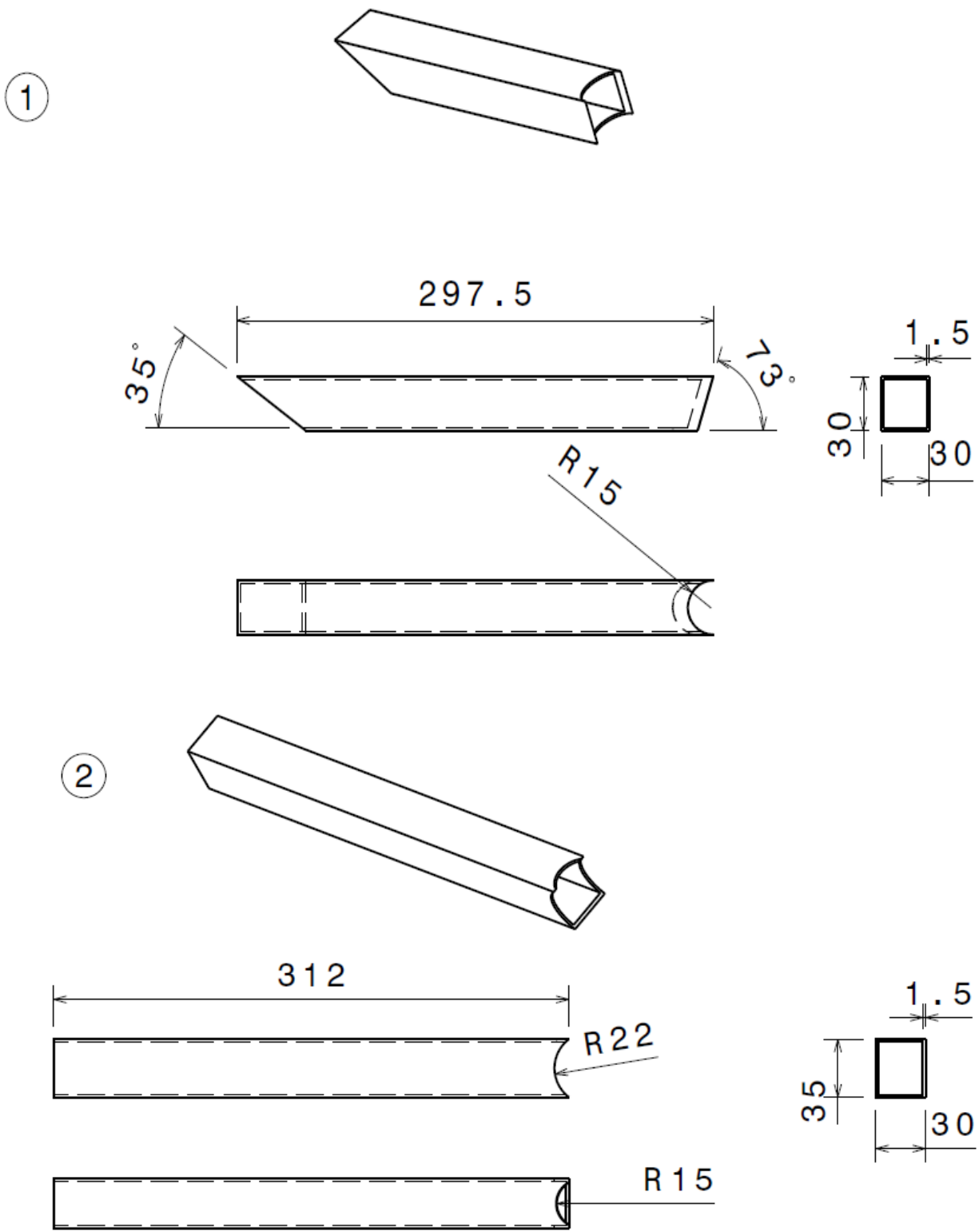
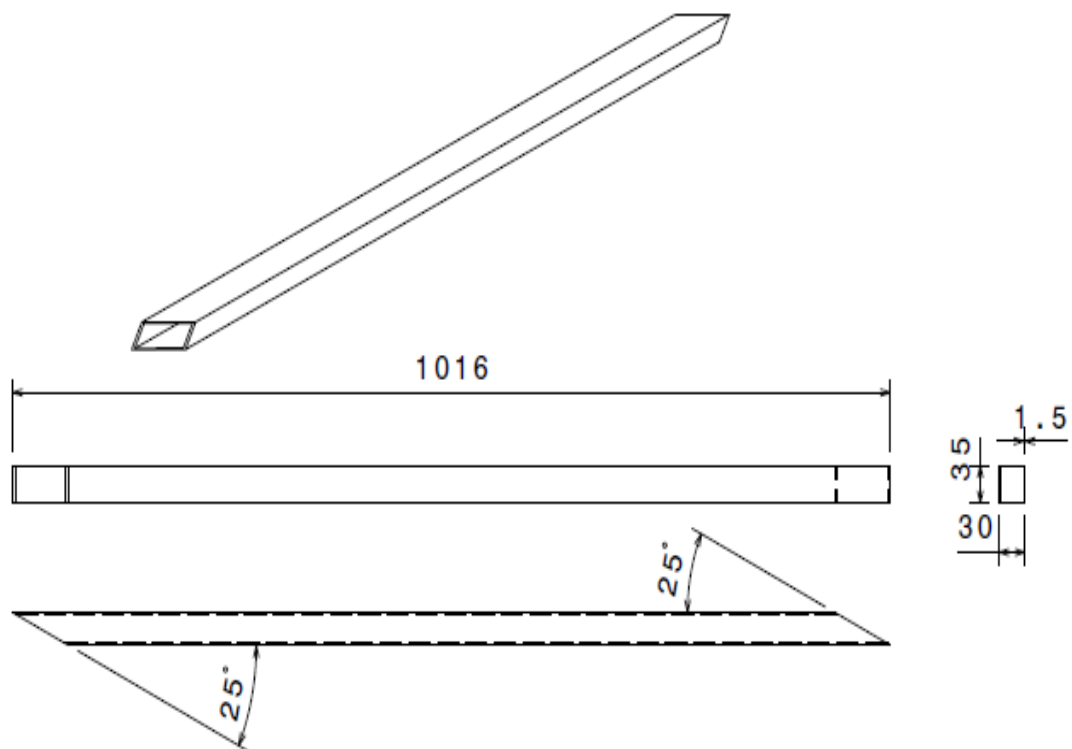


Figure E.5 (1) Seat and (2) Chain stay tube

1



2

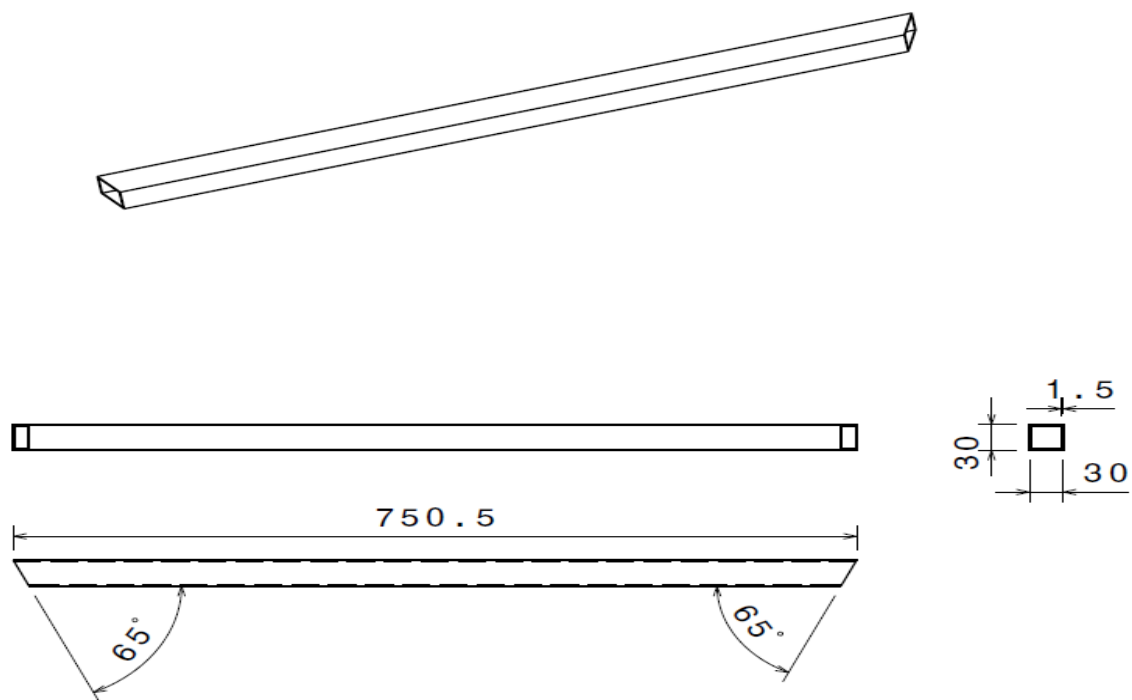


Figure E.6 1) passenger carriage tube and 2) cross tube

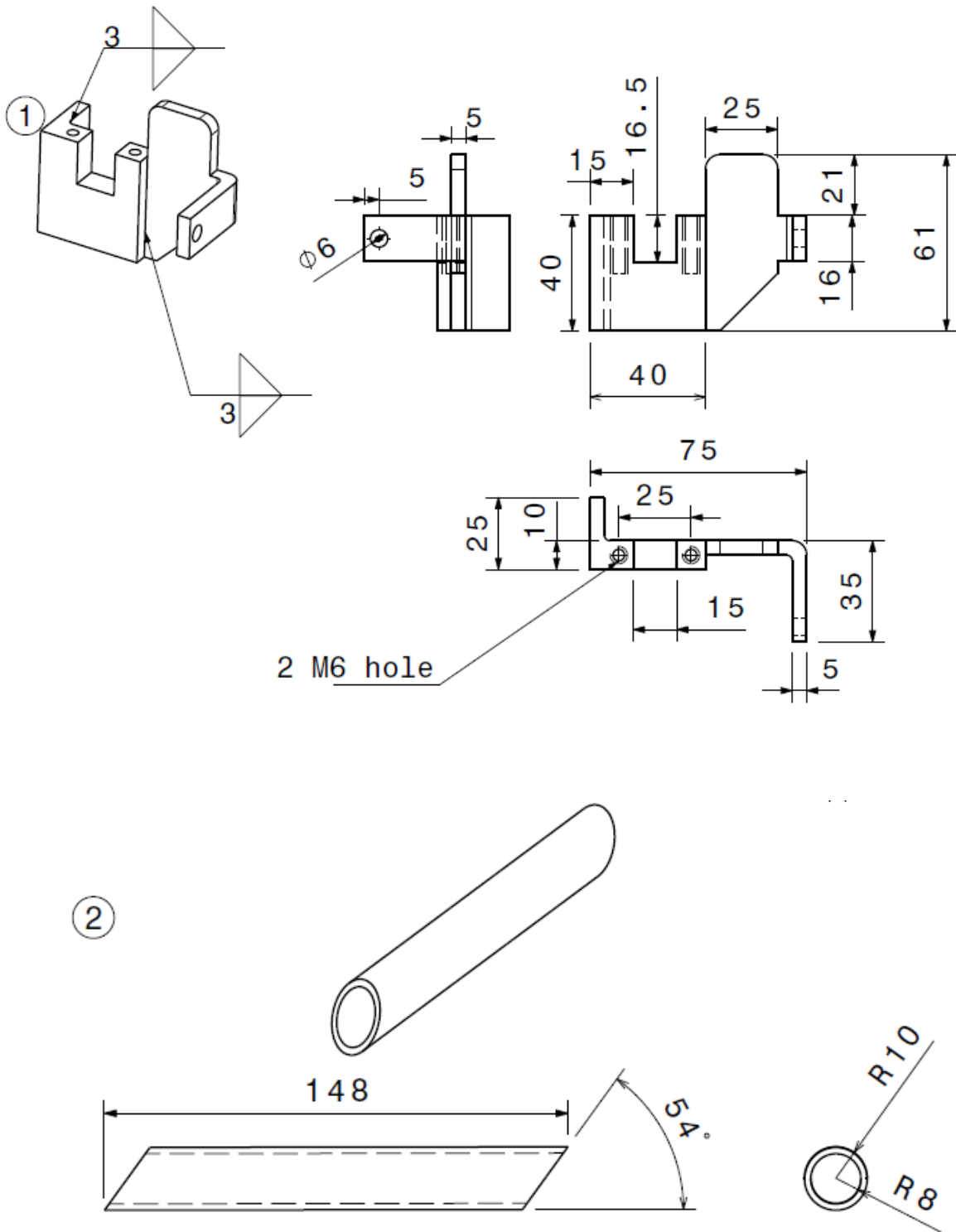


Figure E.7 1) Left KERS mount plate and 2) Rear KERS support tube

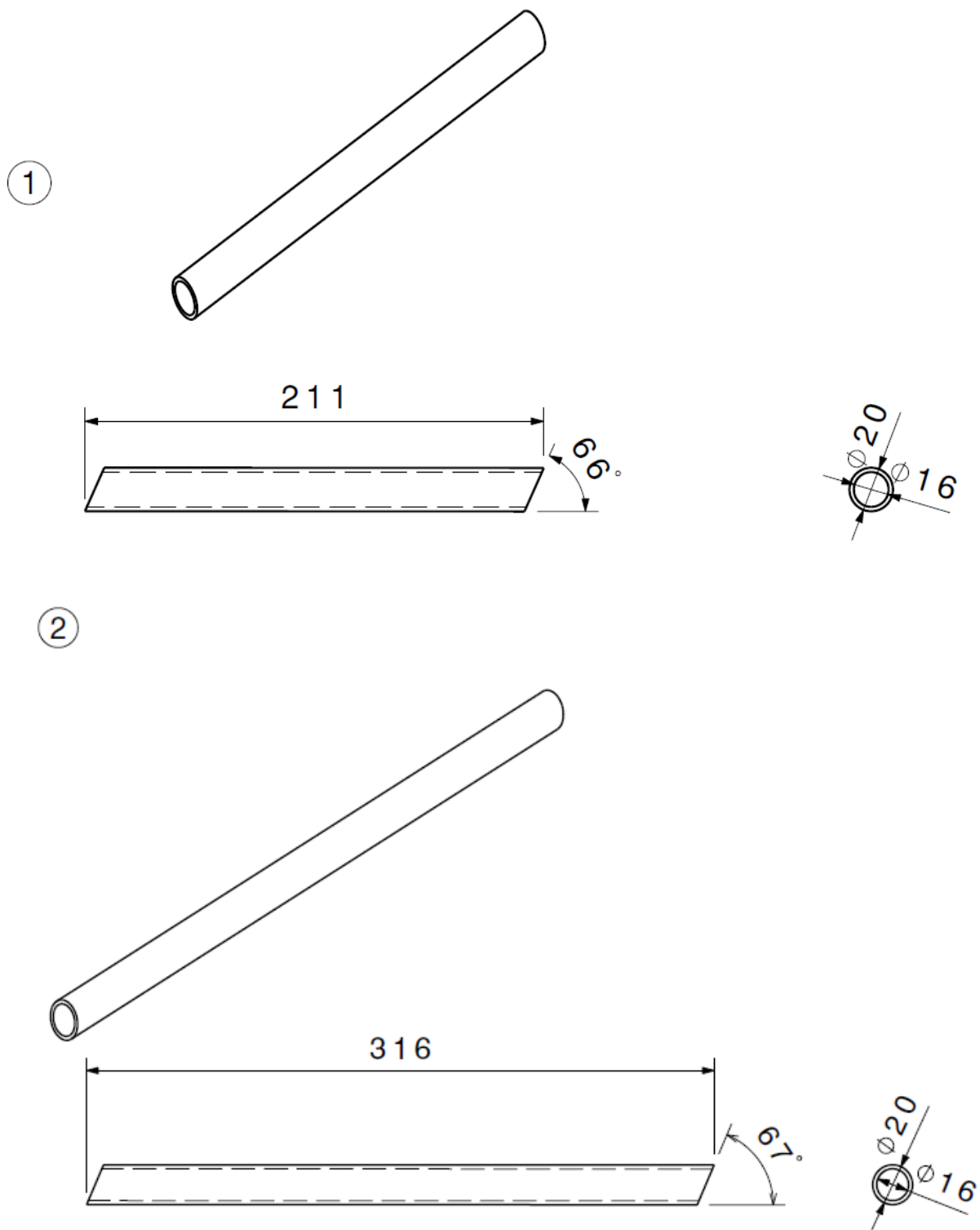


Figure E.8 front KERS support tube (1) right and (2) left side

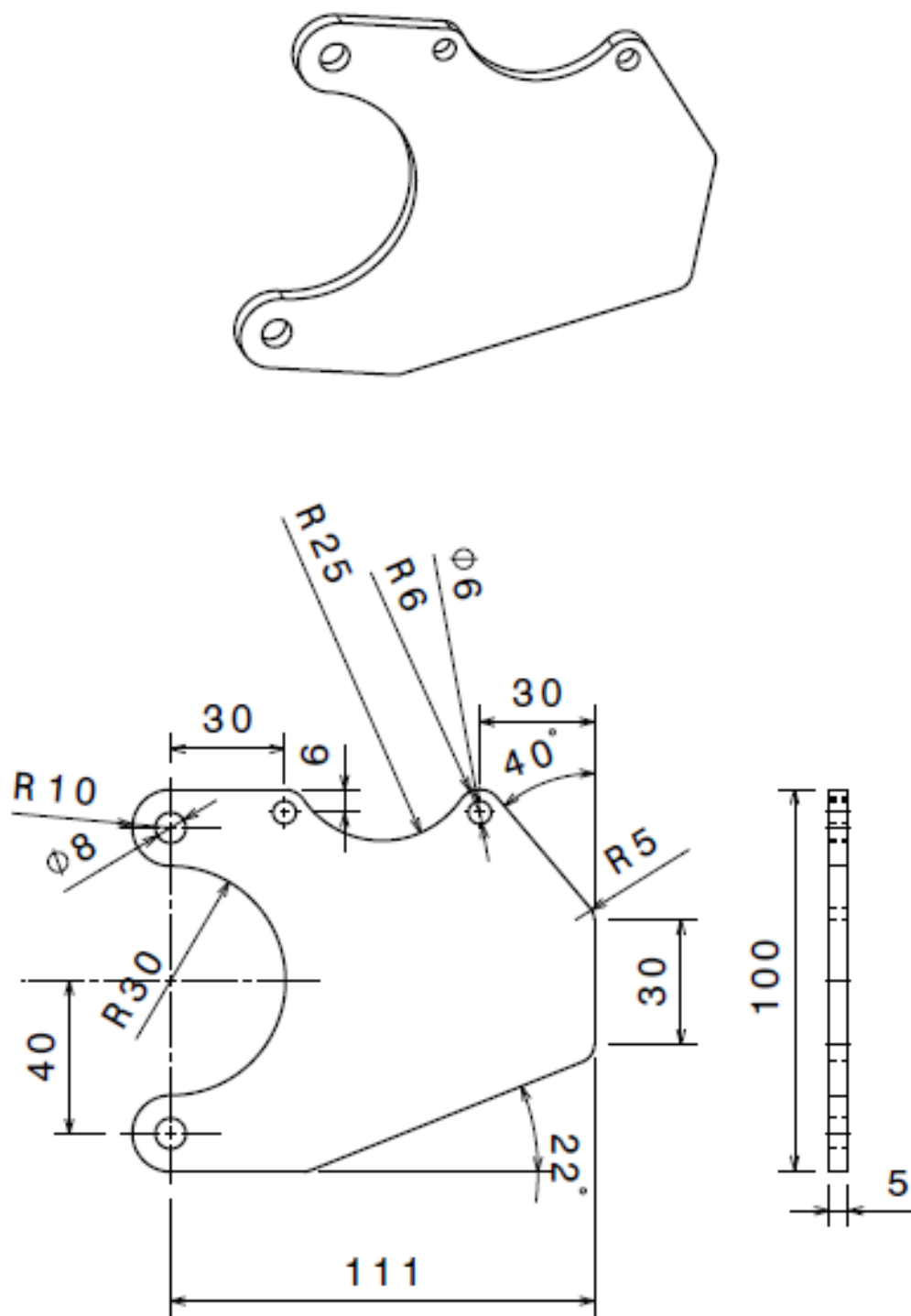


Figure E.9 Rear axle support end plate

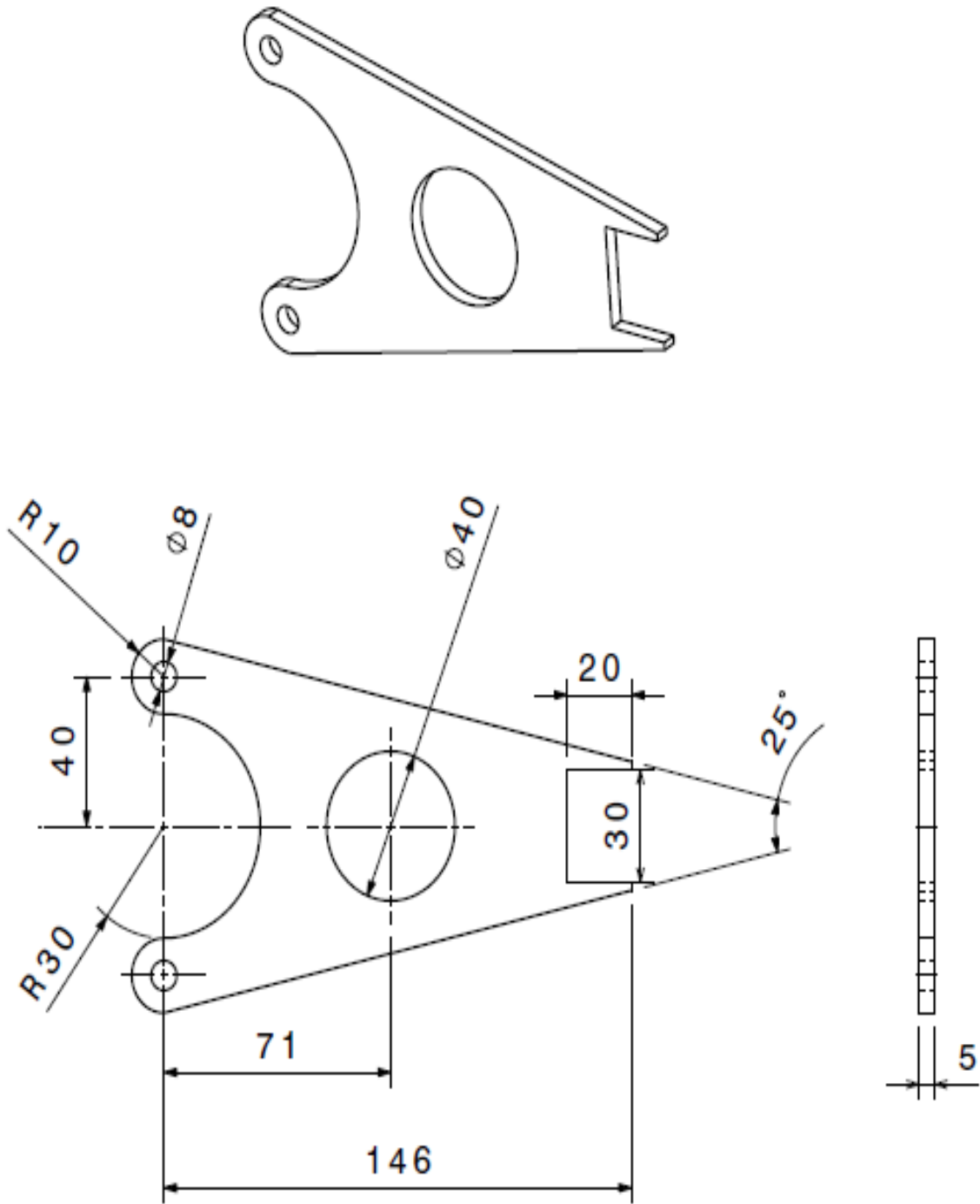
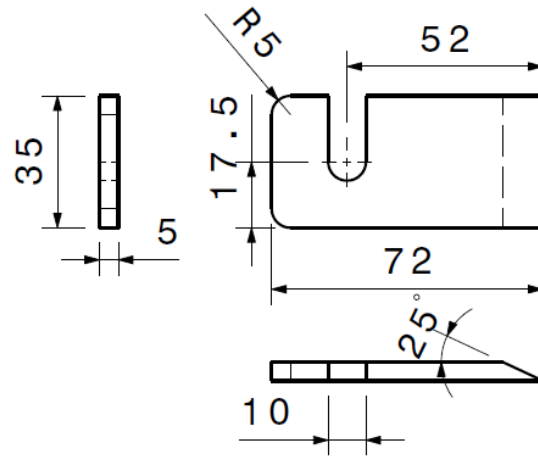
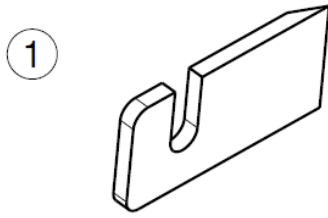


Figure E.10 Rear axle support middle plate



②

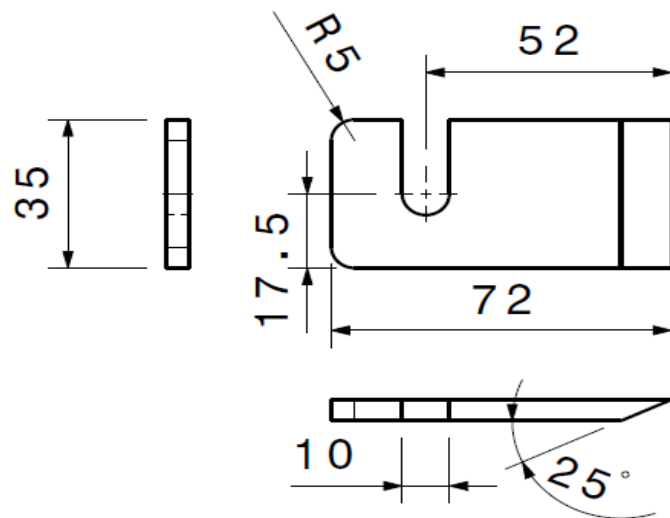
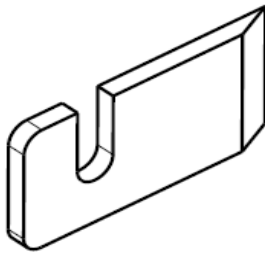


Figure E.11 (1) Left and (2) Right side drive hub support plate

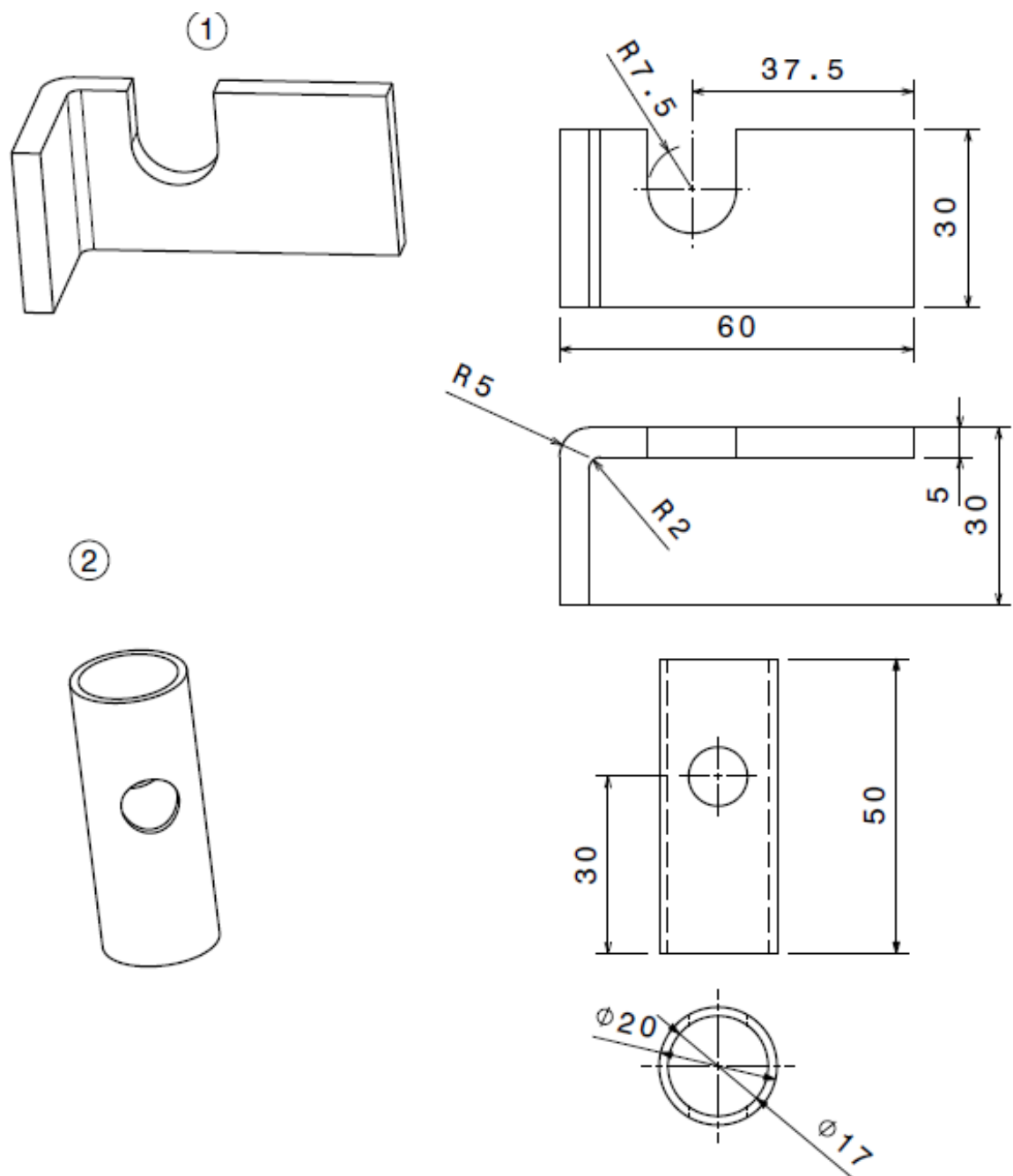
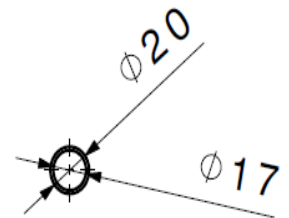
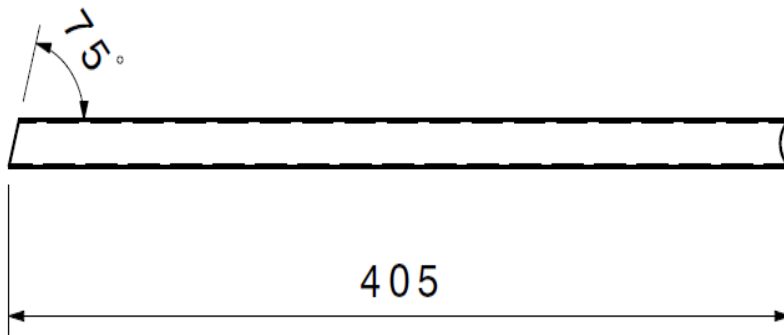
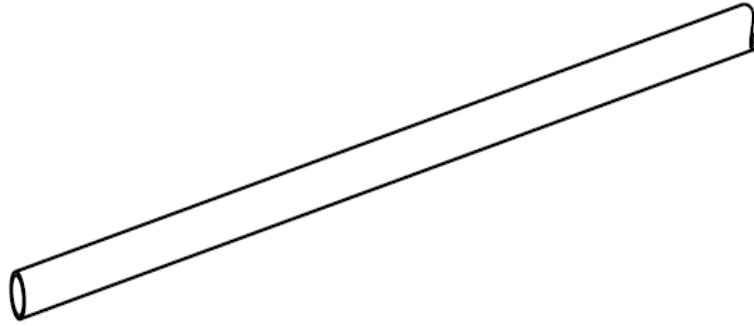


Figure E.12 1) Right side KERS mount plate and 2) Passenger seat mount tube

①



②

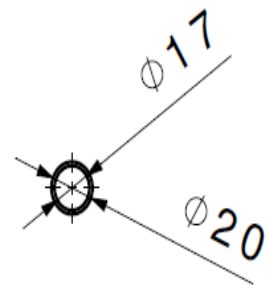
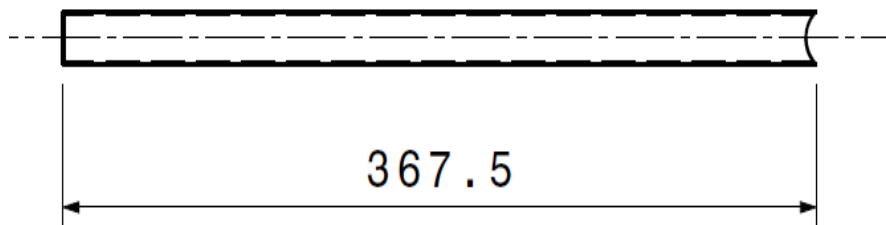
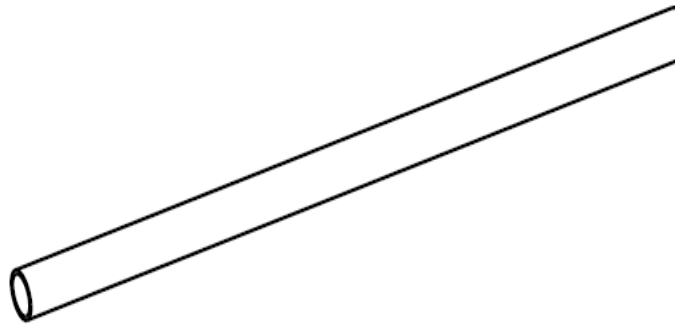


Figure E. 14 (1) Back rest and (2) Seat middle tube

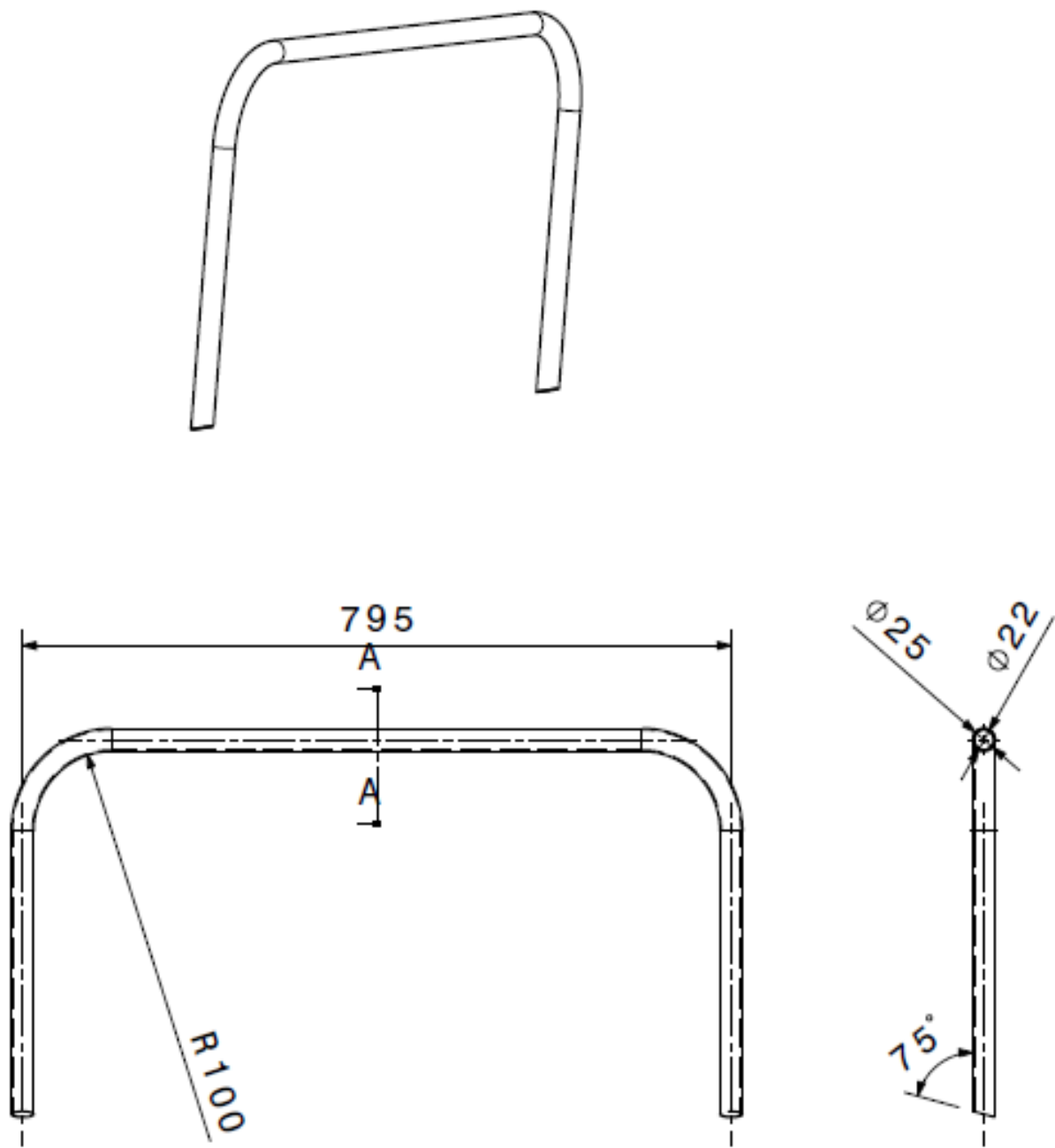
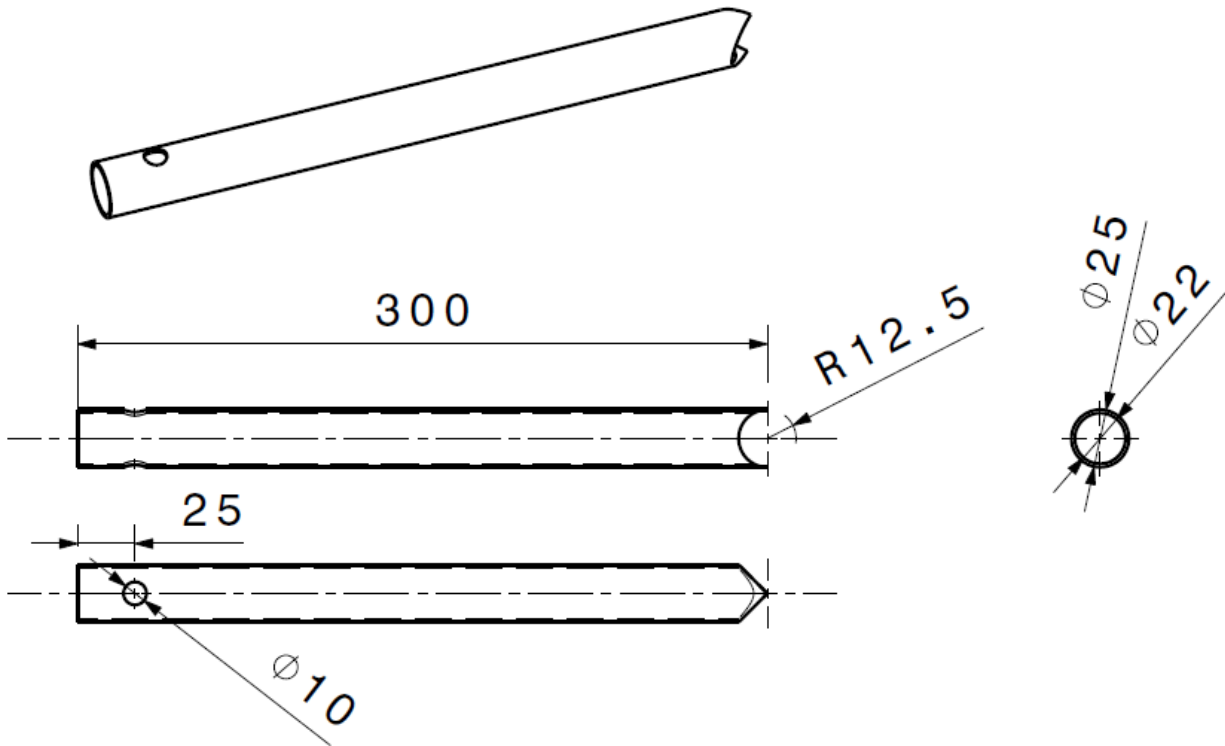


Figure E.15 Back rest tube

1



2

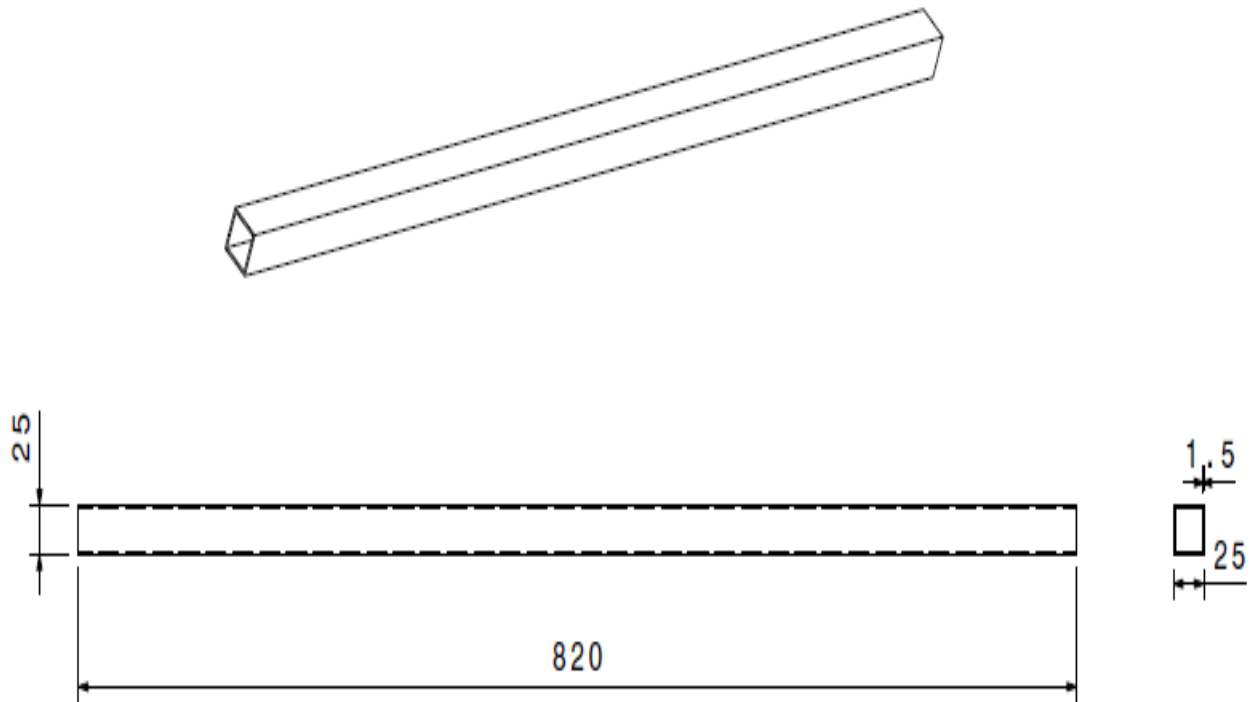


Figure E.16 passenger seat (1) leg and (2) back tube

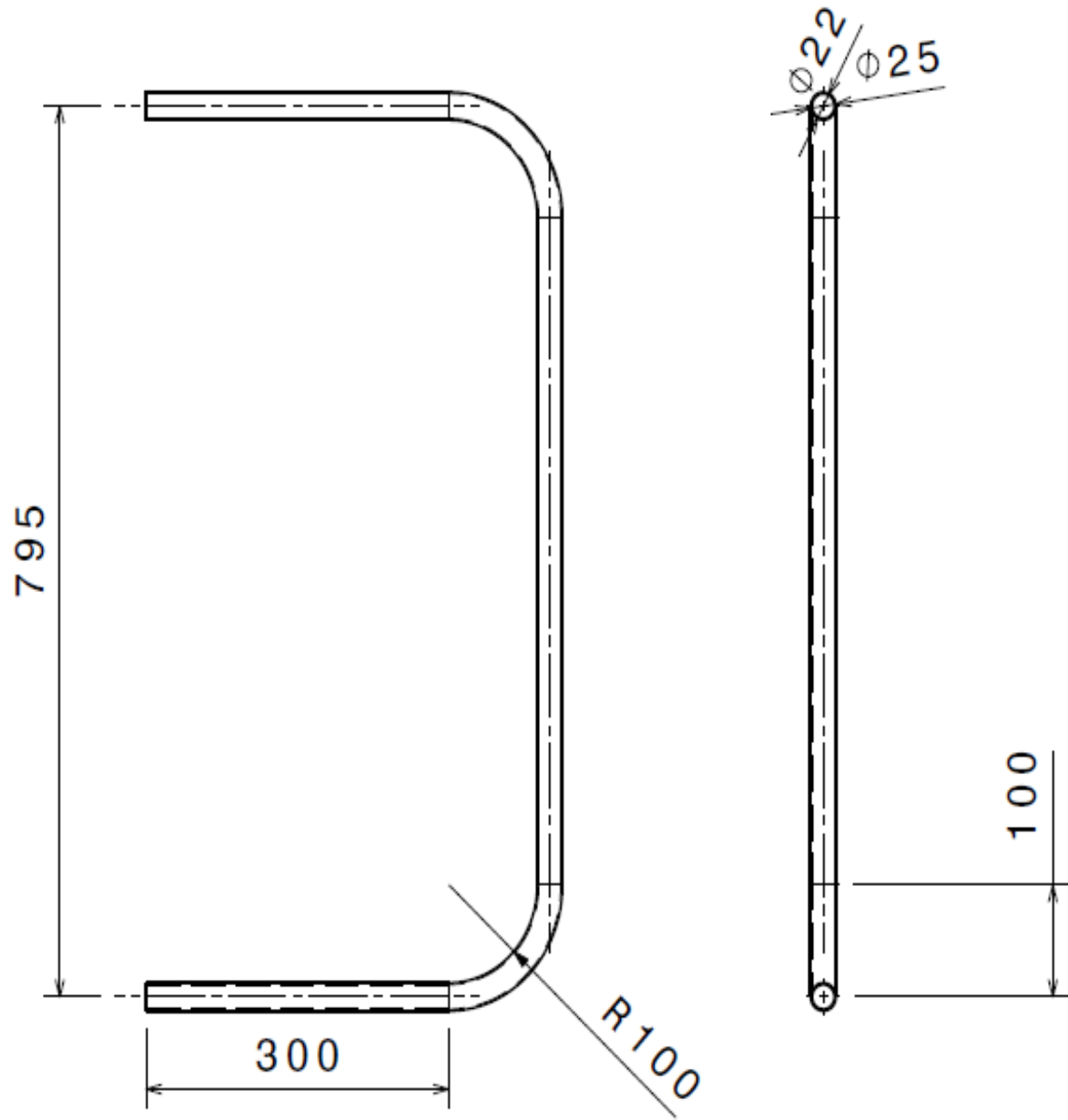


Figure E.17 Bottom seat tube

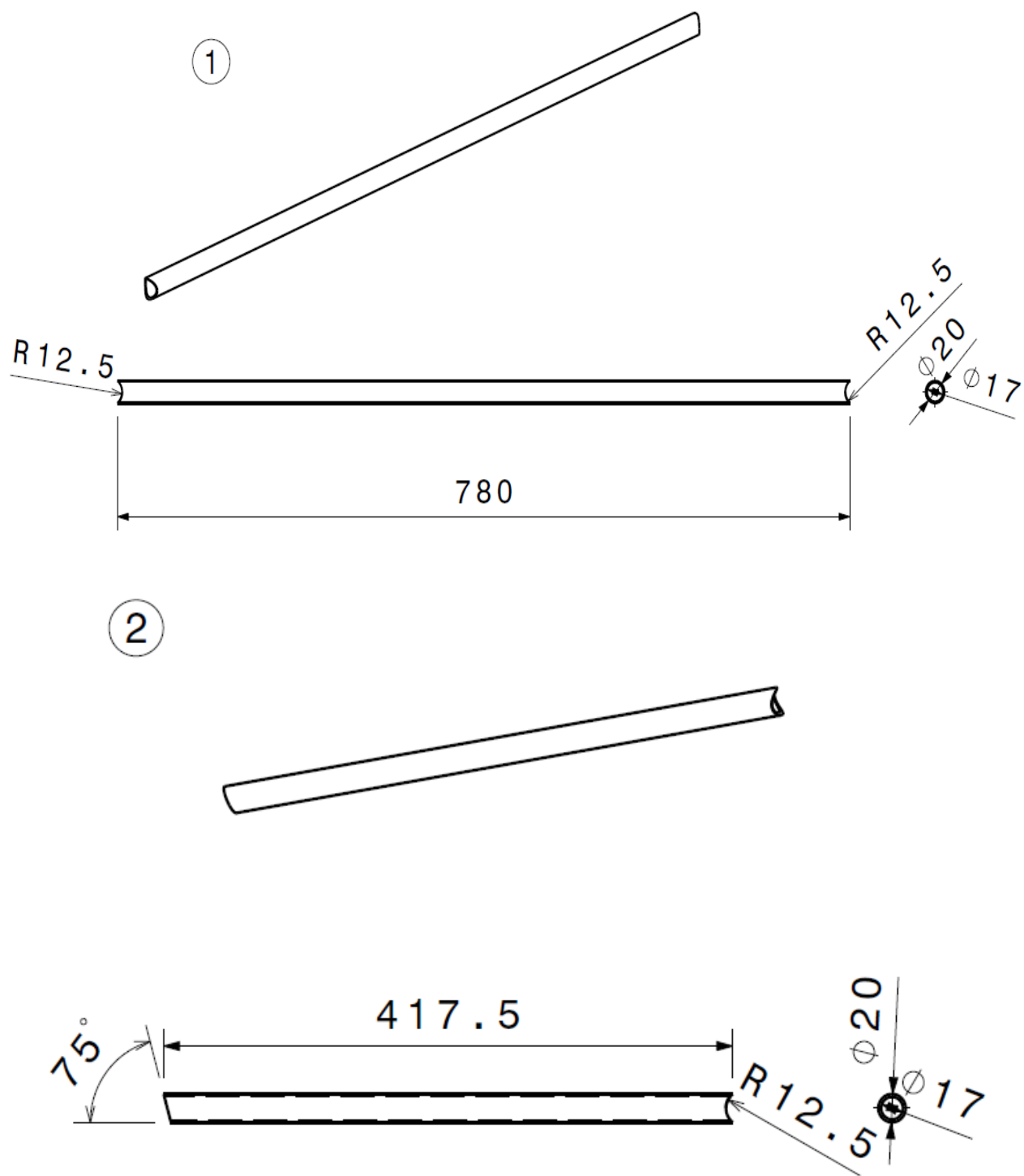


Figure E. 18 (1) Rear side support and (2) Back rest middle tube

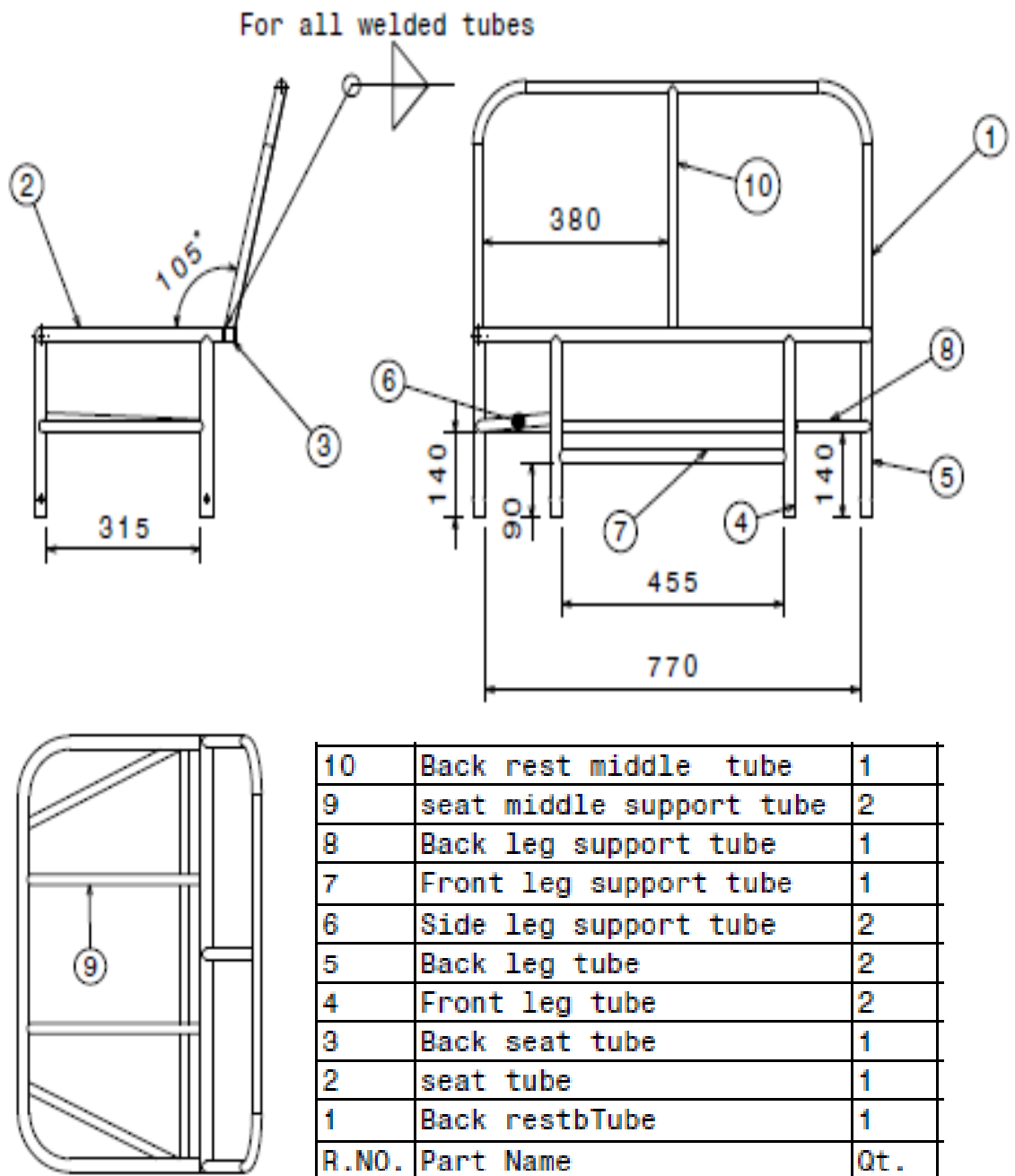


Figure E.19 Passenger seat side, front and top view assembly drawing

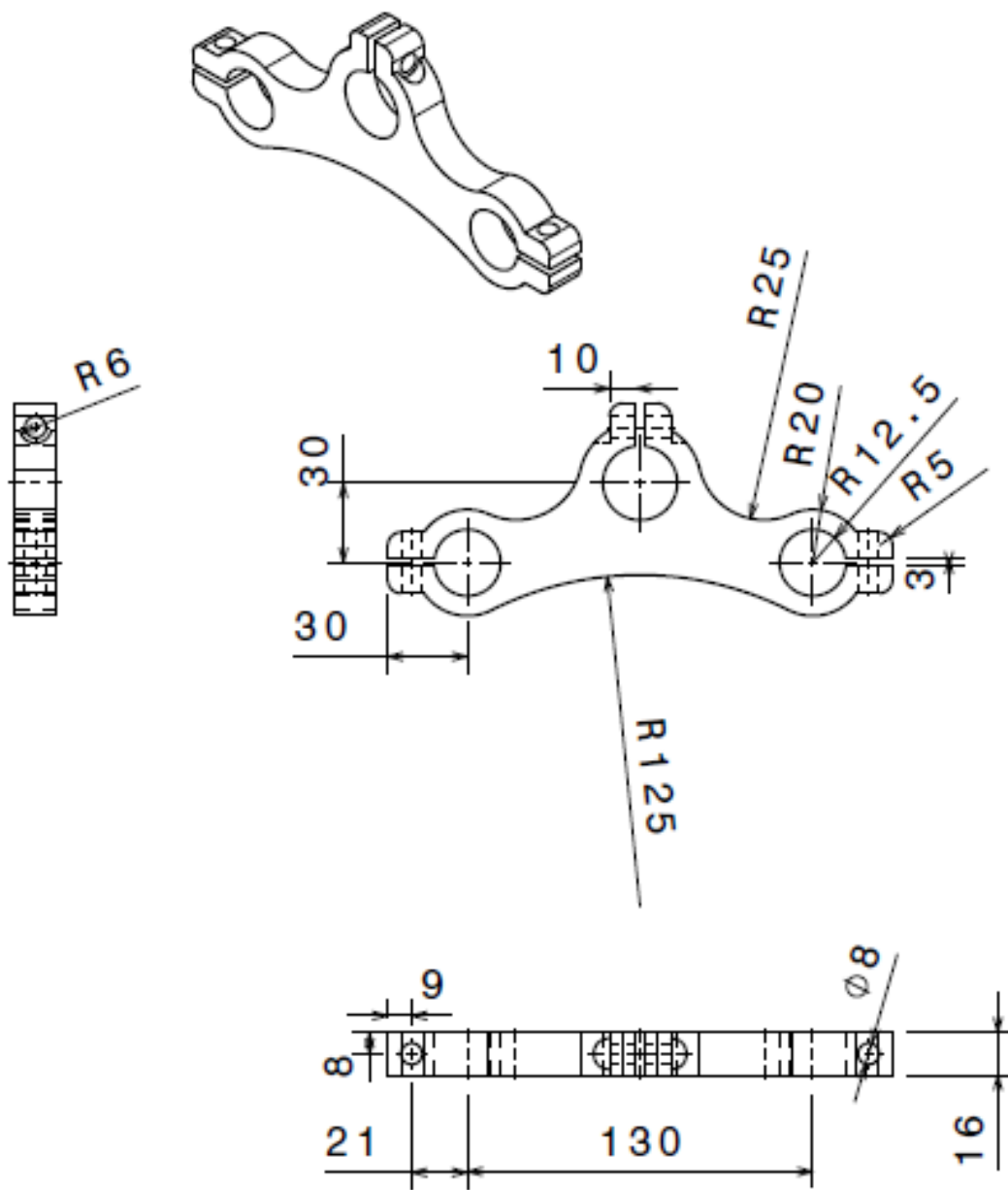


Figure E. 20 Brace

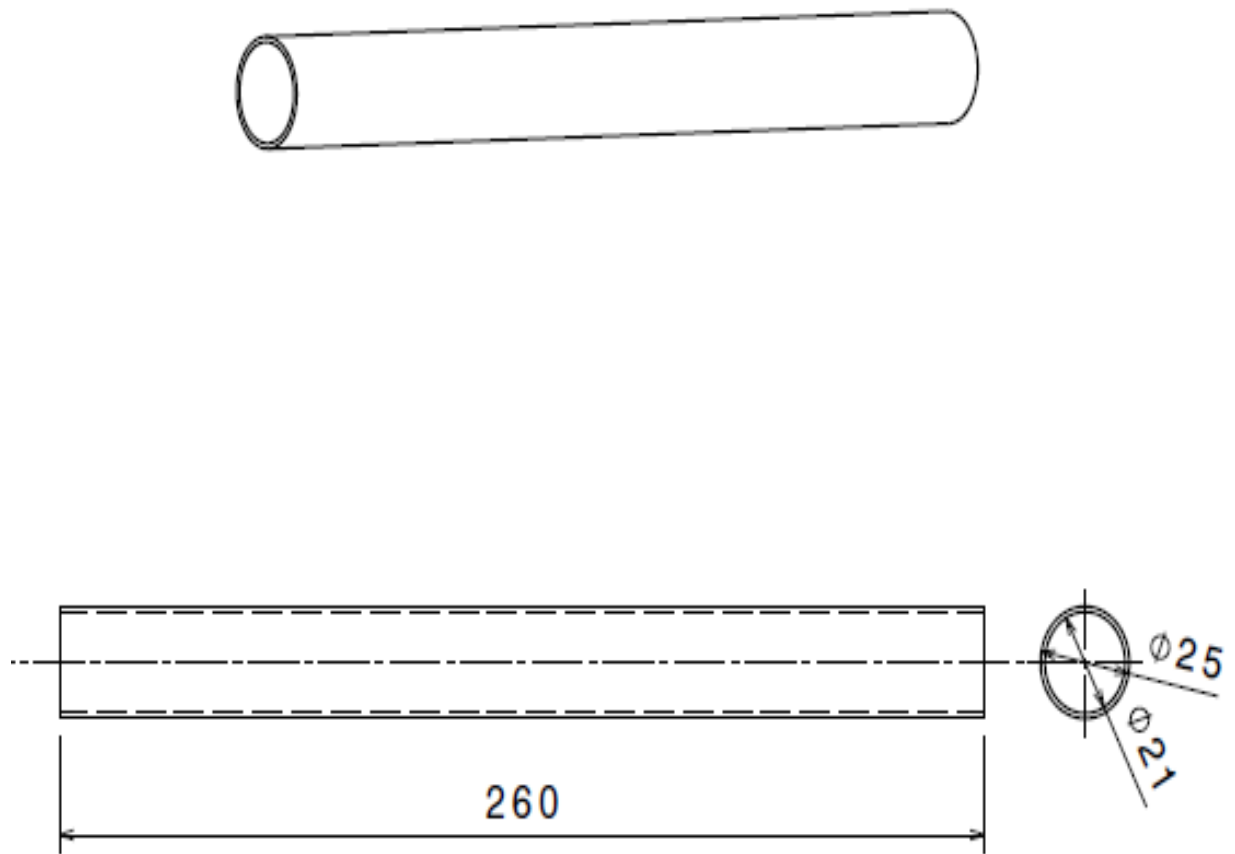


Figure E. 21 steerer tube

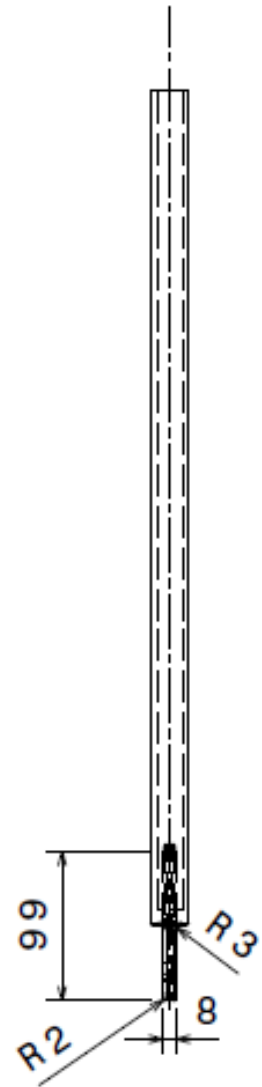
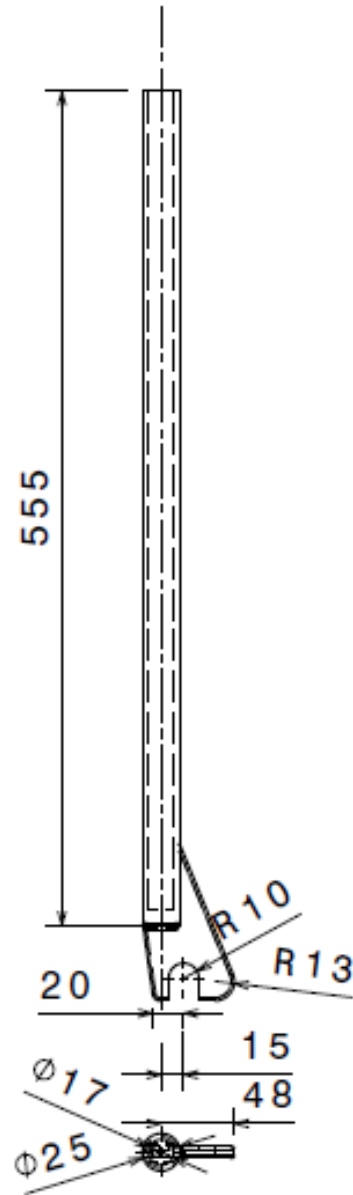


Figure E. 22 Fork leg

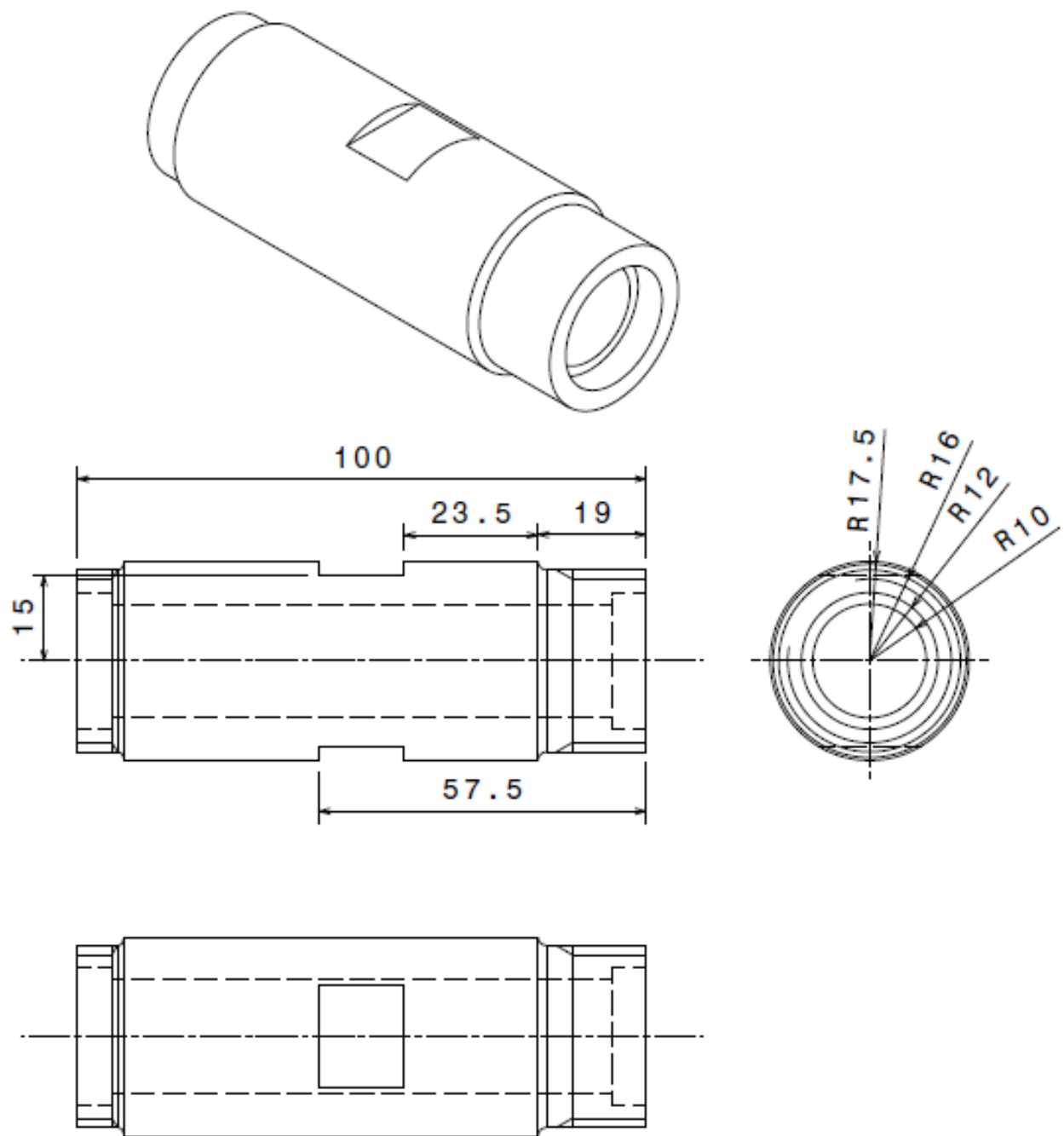


Figure E. 24 Drive hub

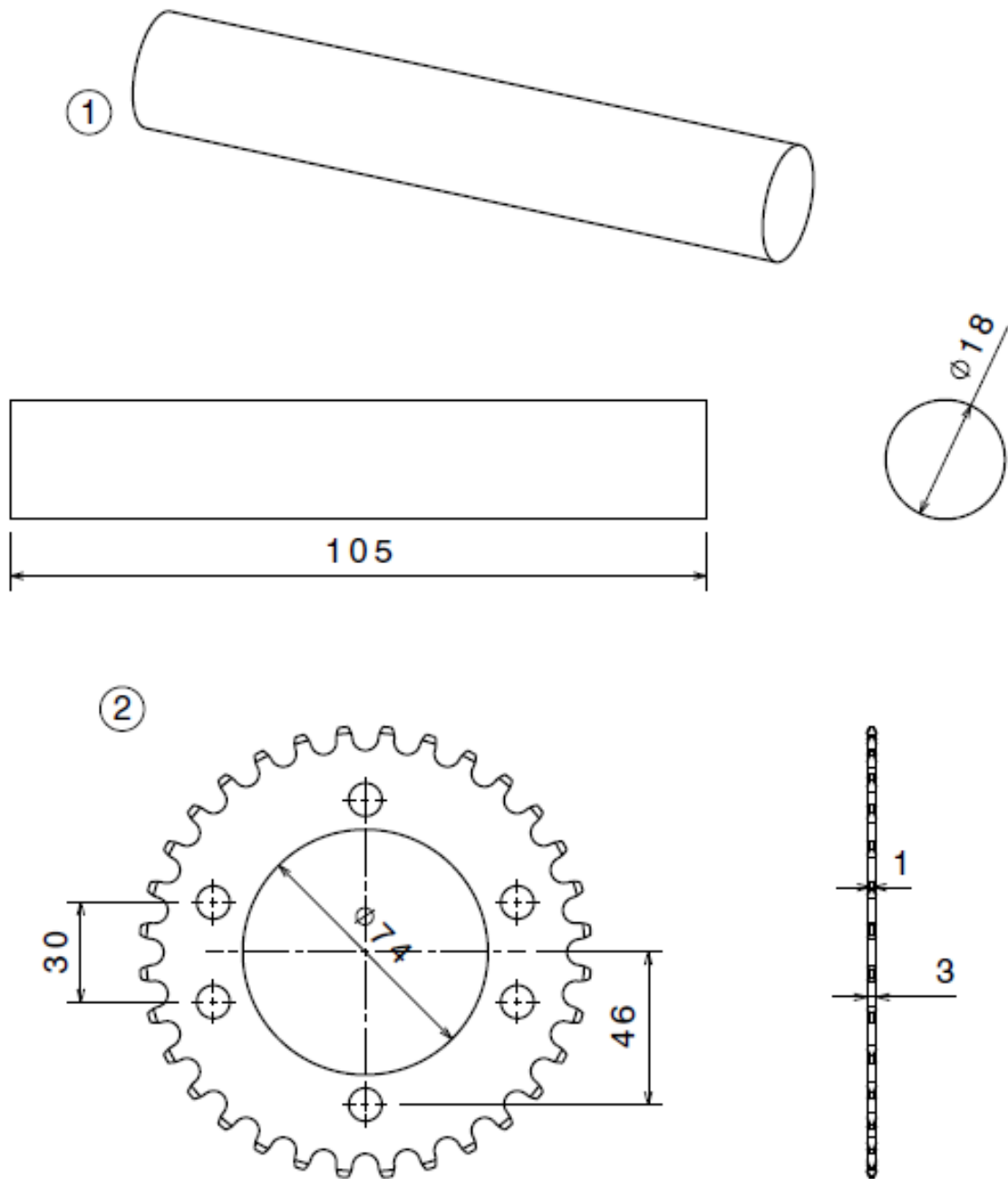


Figure E. 25 (1) Differential SHAFR and (2) Differential sprocket

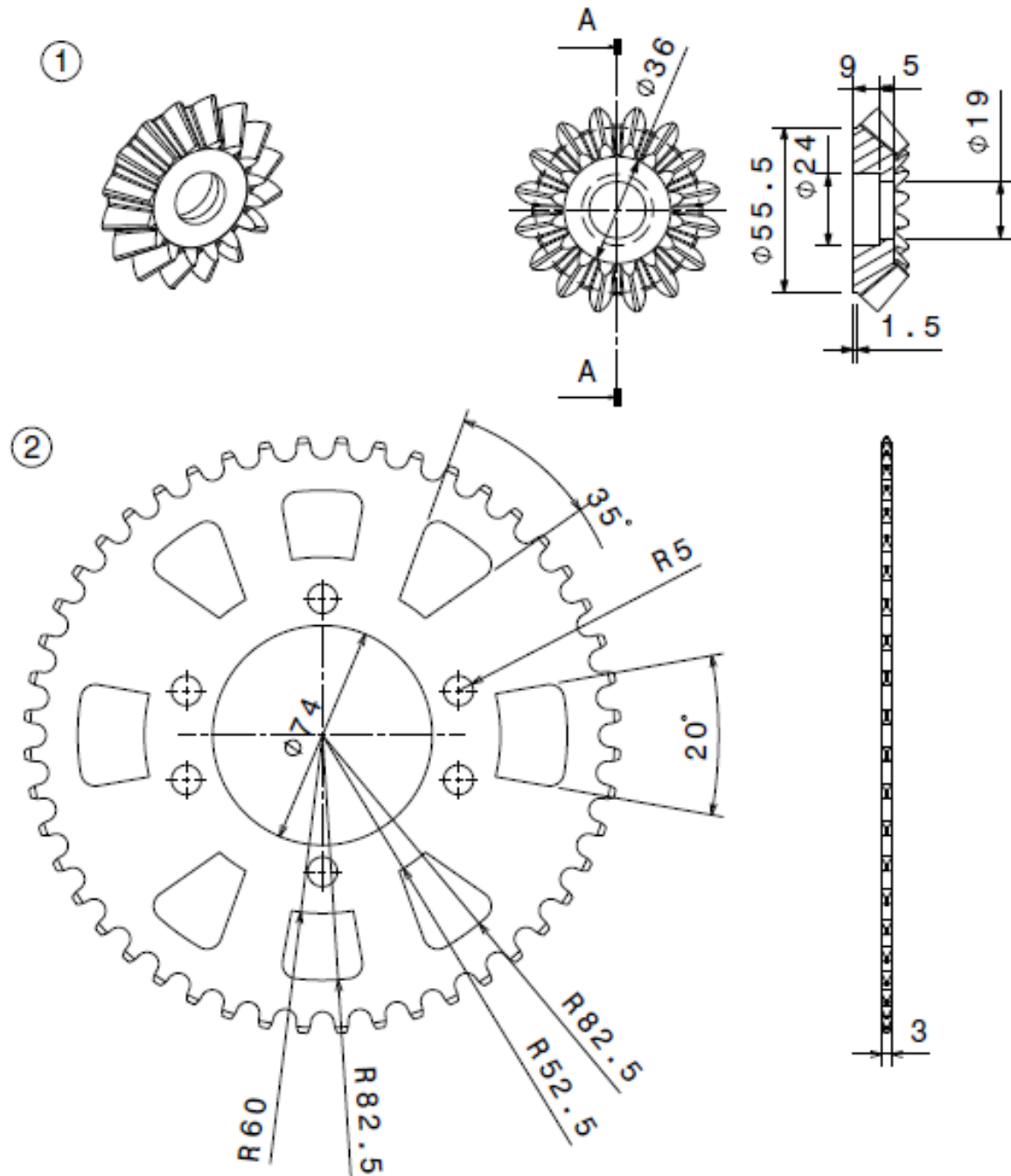


Figure E. 26 (1) Differential planet gear and (2) Differential KERS sprocket

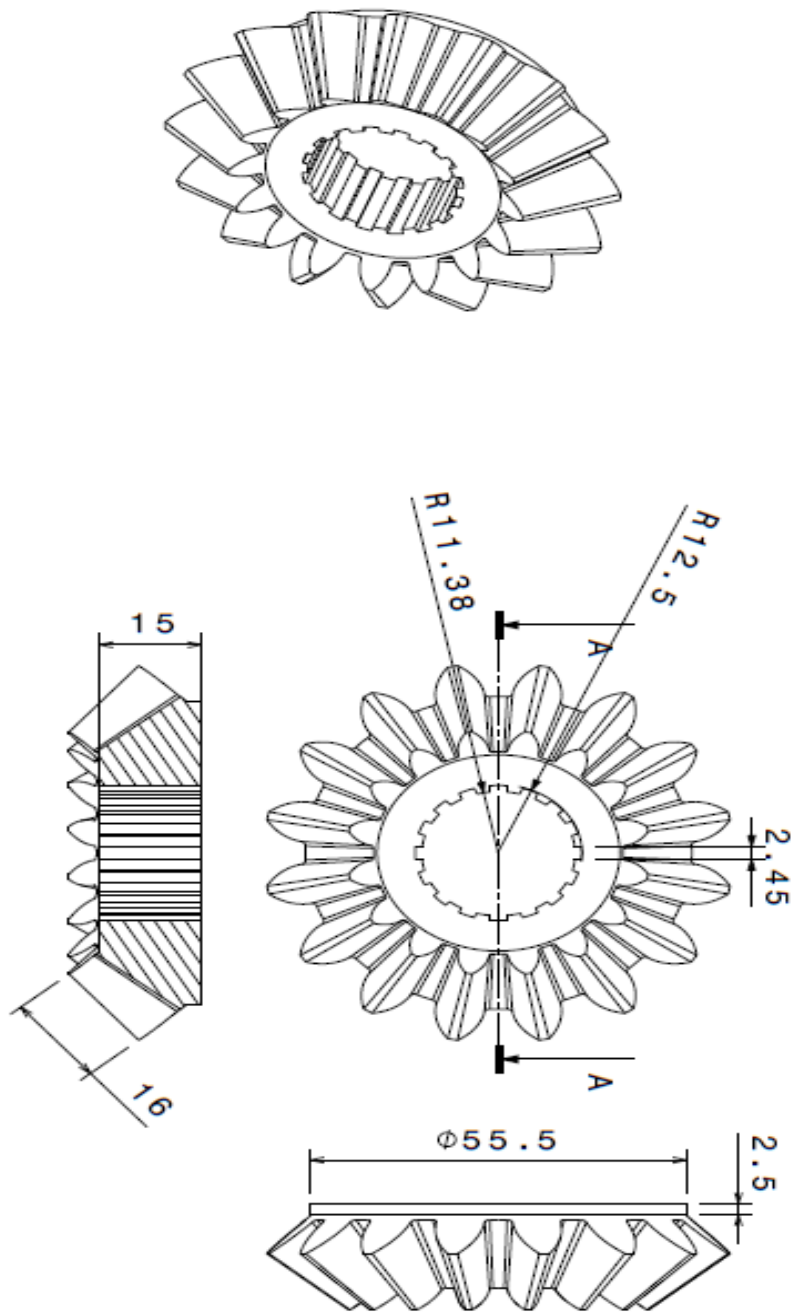


Figure E. 27 Differential side gear

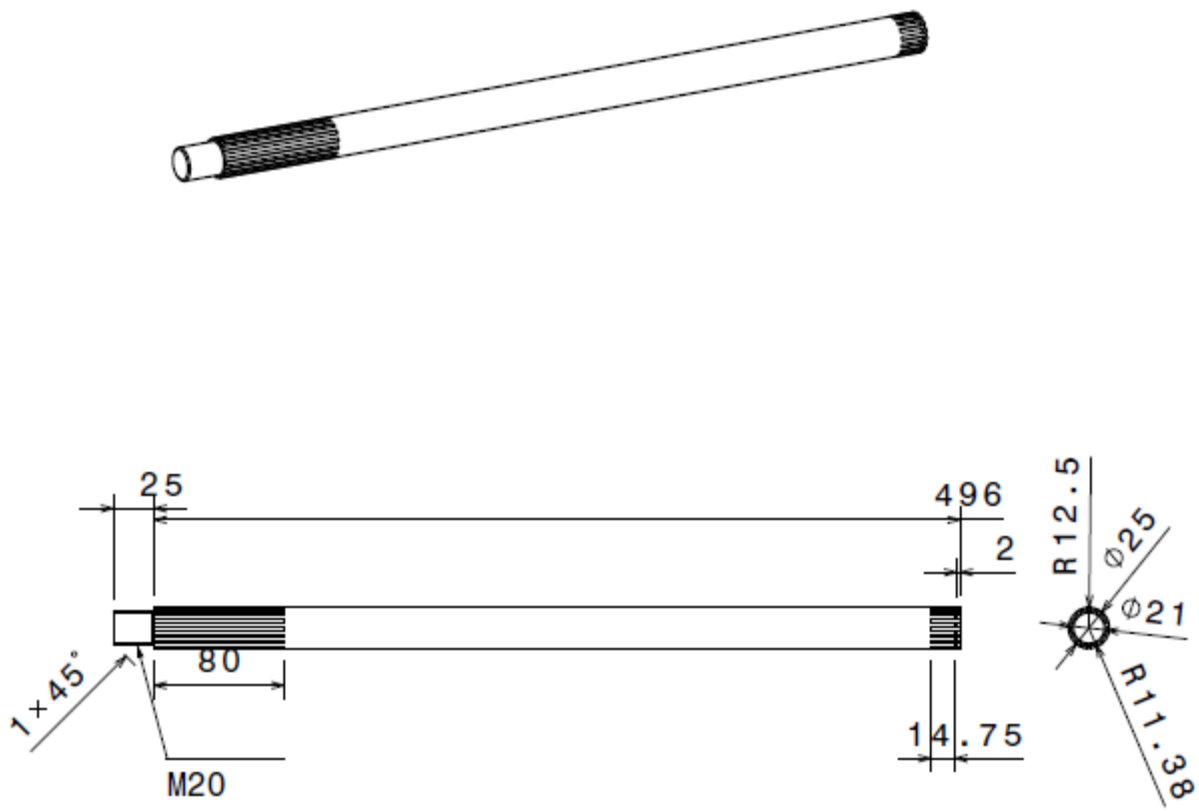


Figure E. 28 Axle shaft

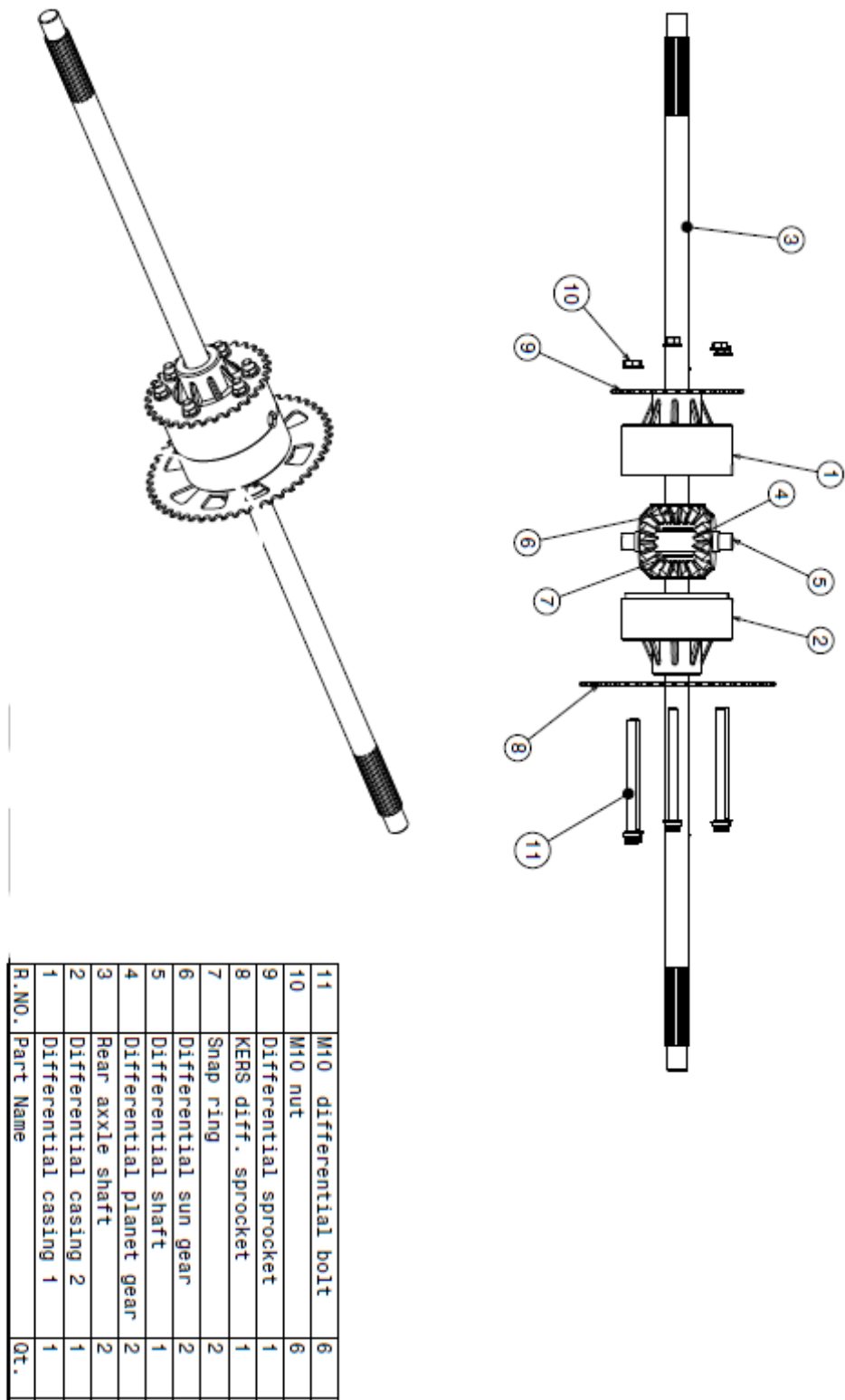


Figure E. 29 Final drive assembly

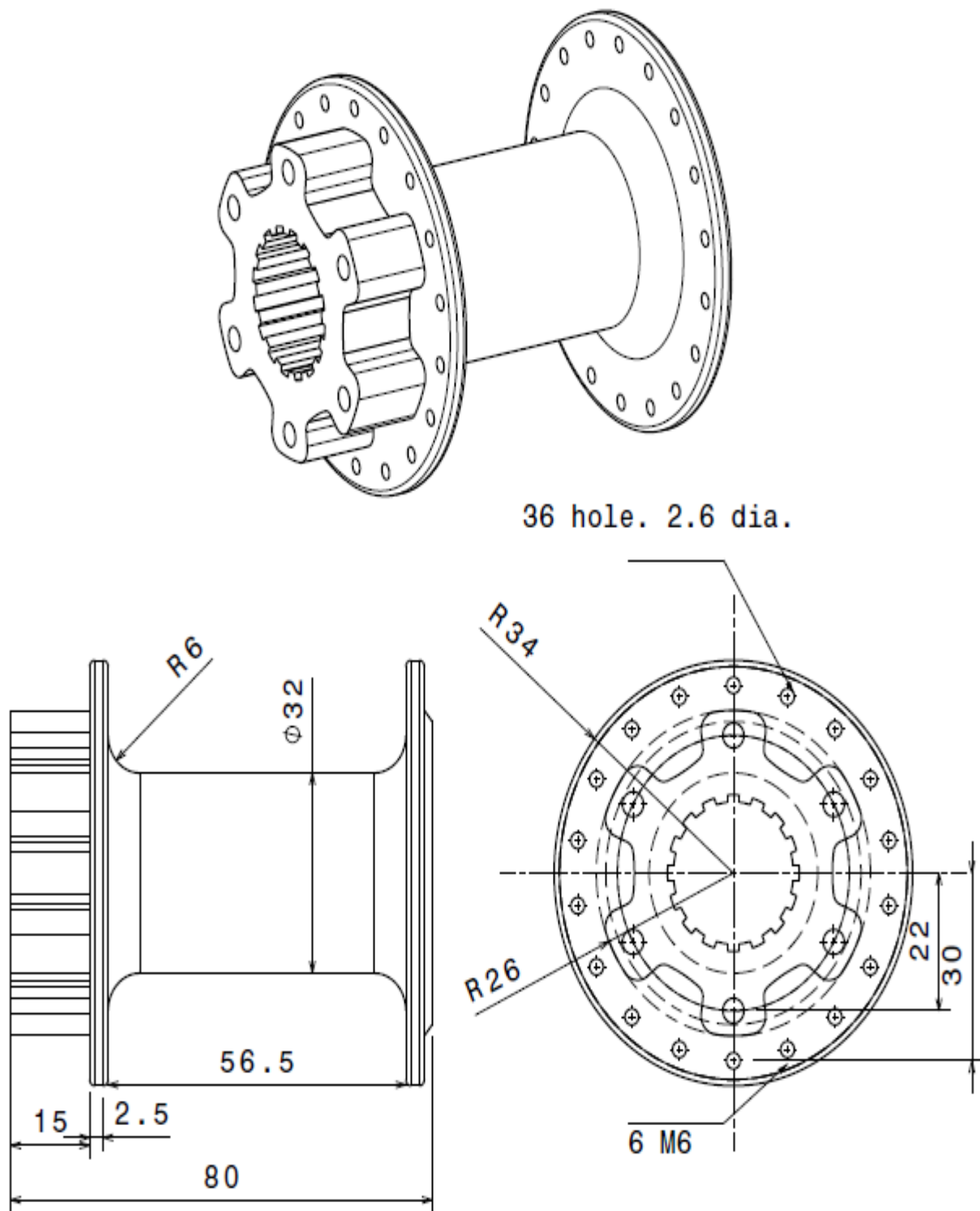


Figure E. 30 Rear wheel hub shell

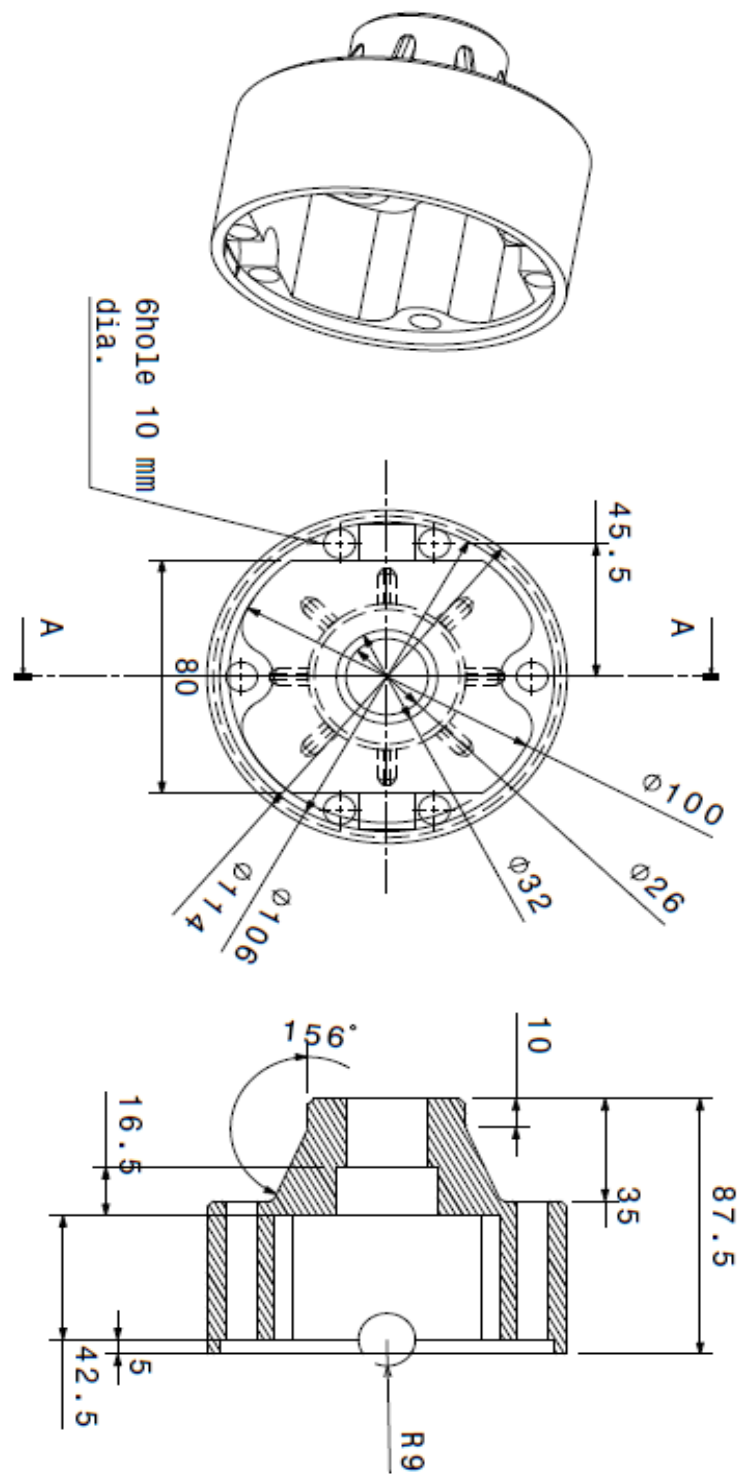


Figure E. 31 Differential casing 1

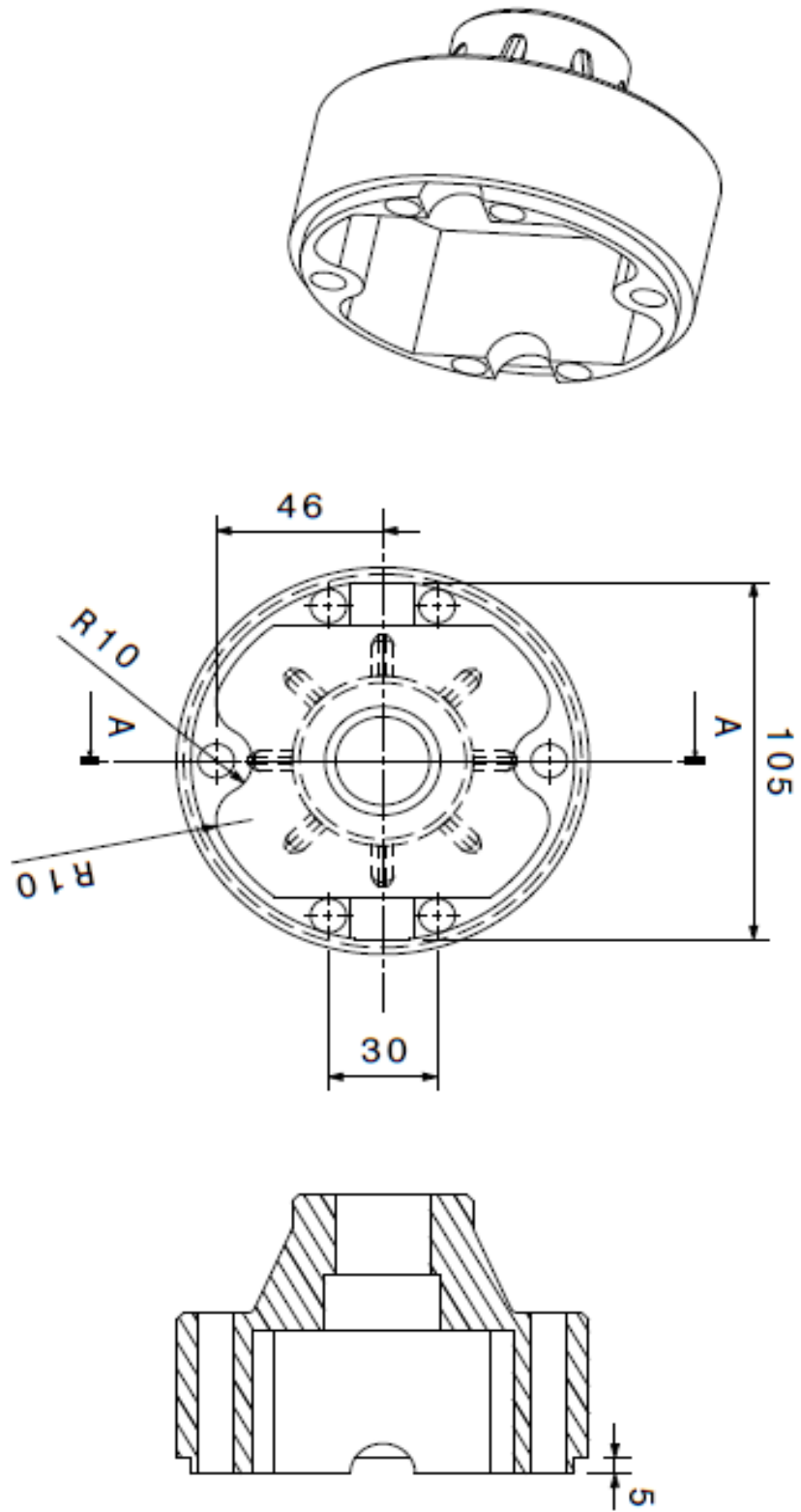


Figure E. 32 Differential casing 2

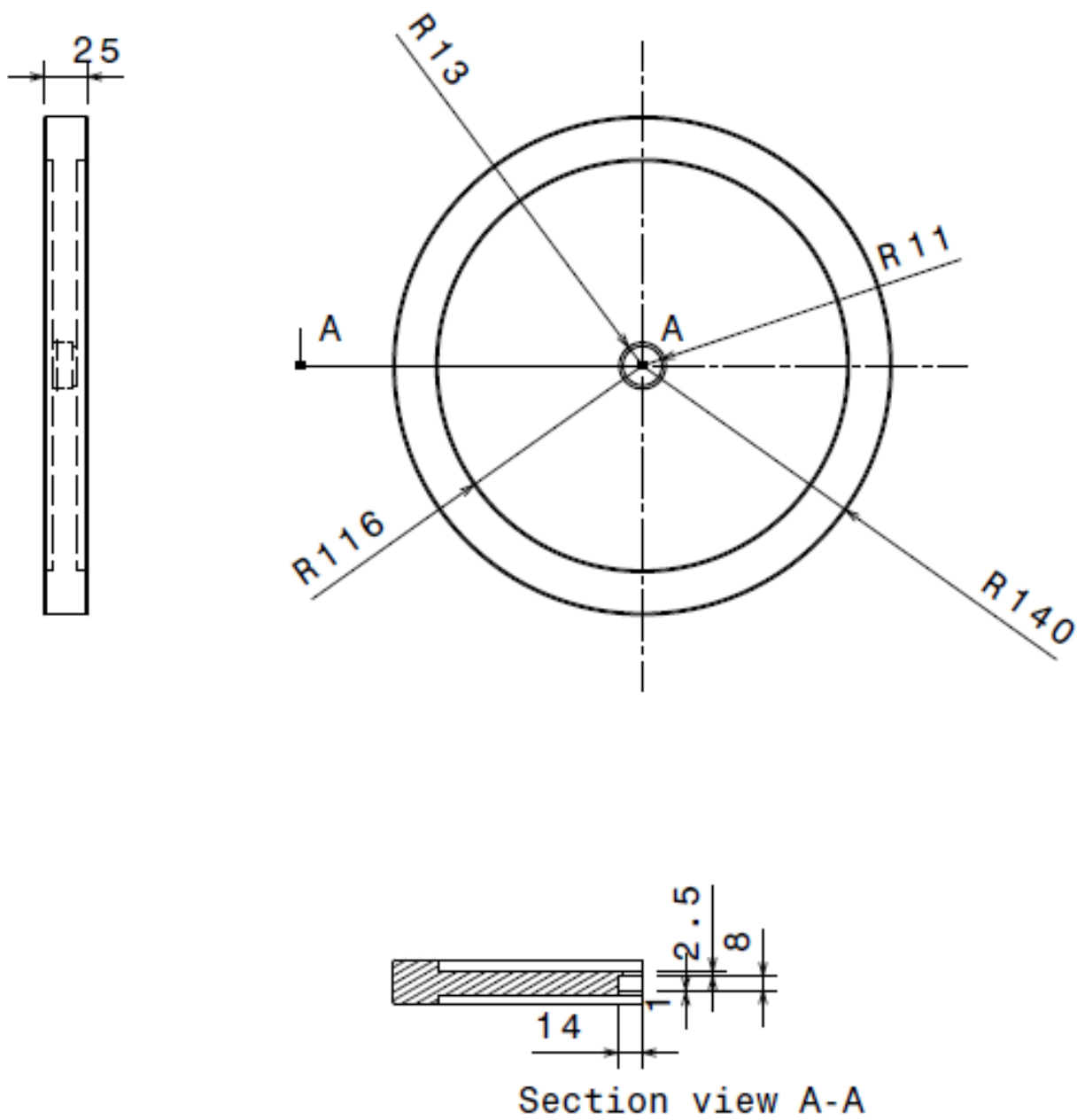


Figure E. 33 Flywheel

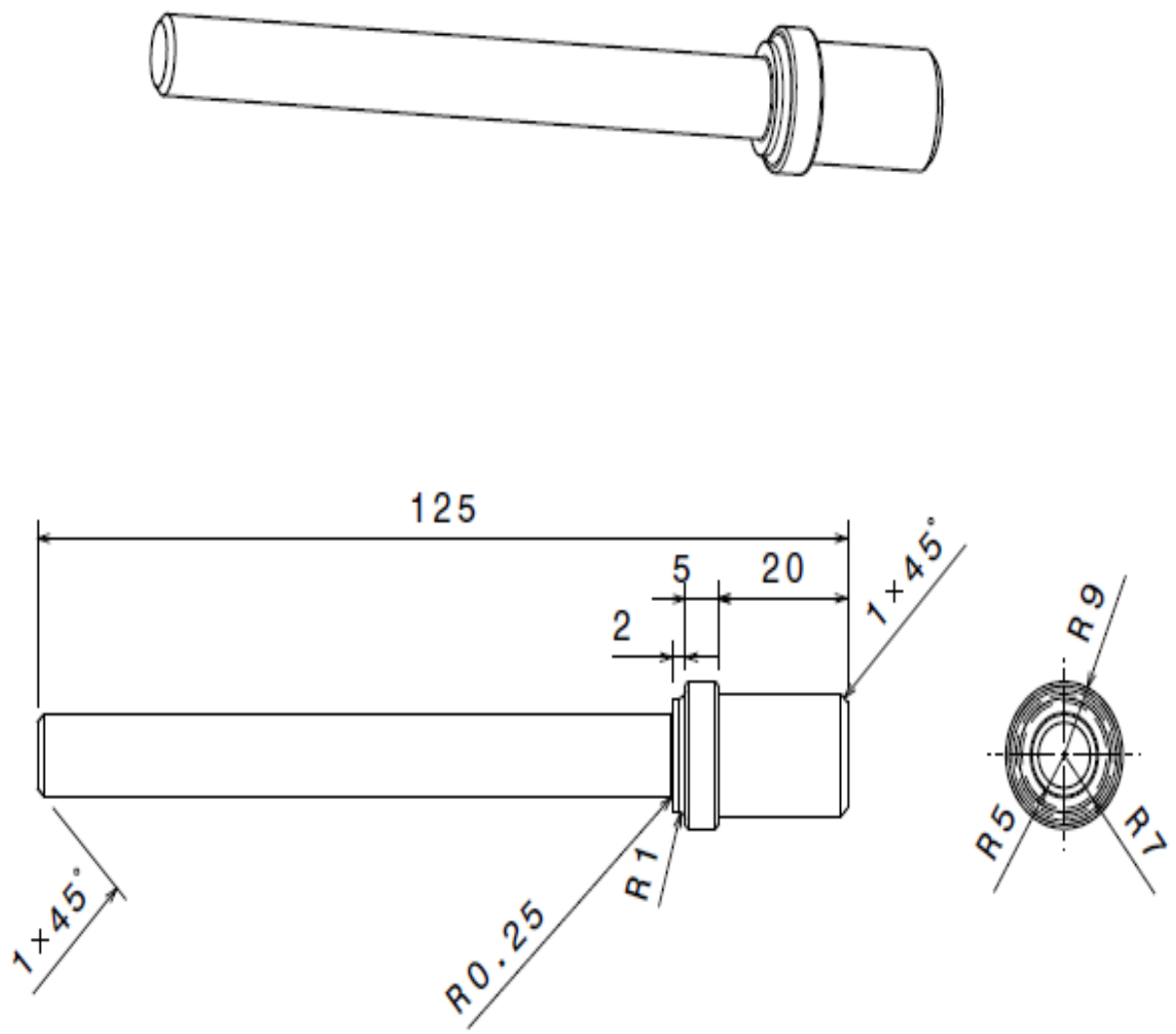


Figure E. 34 KERS shaft

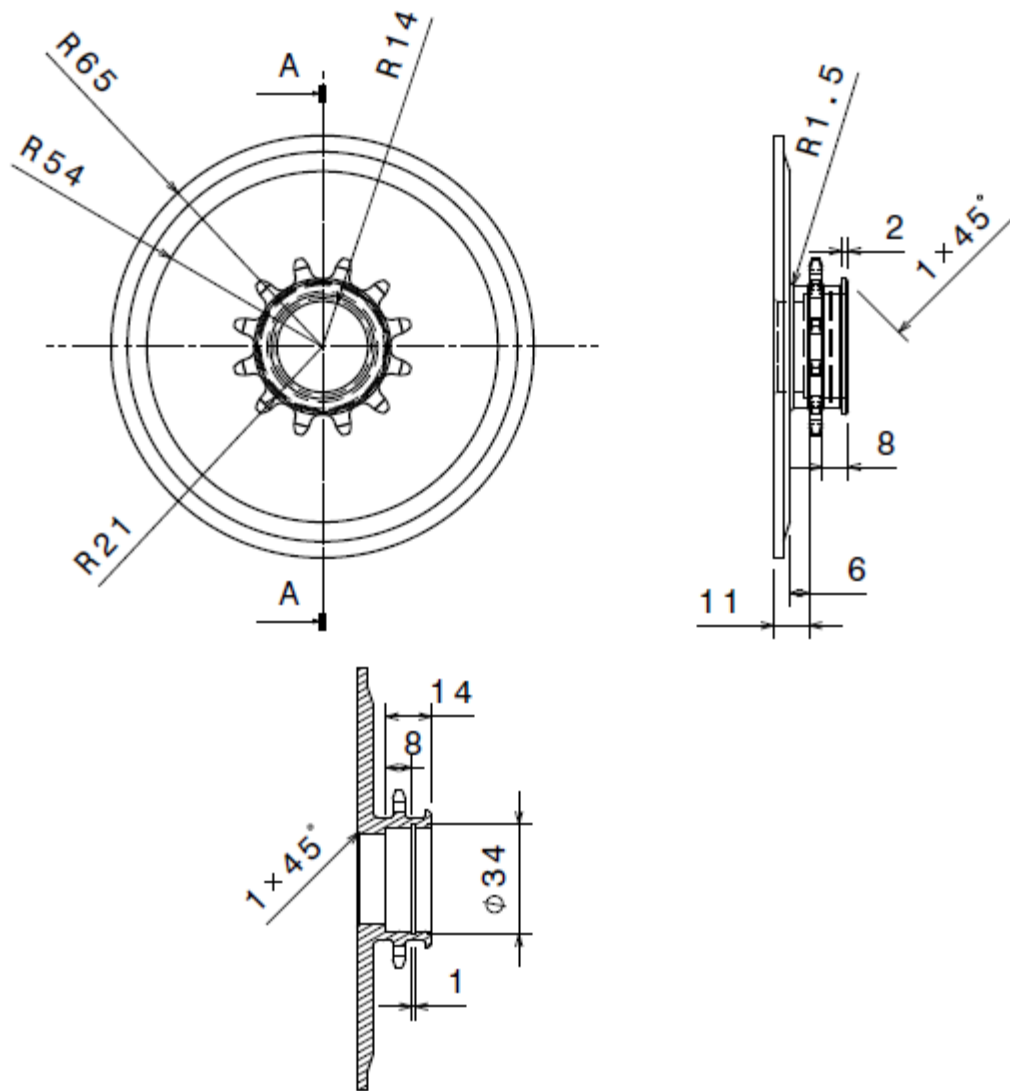


Figure E. 35 KERS sprocket

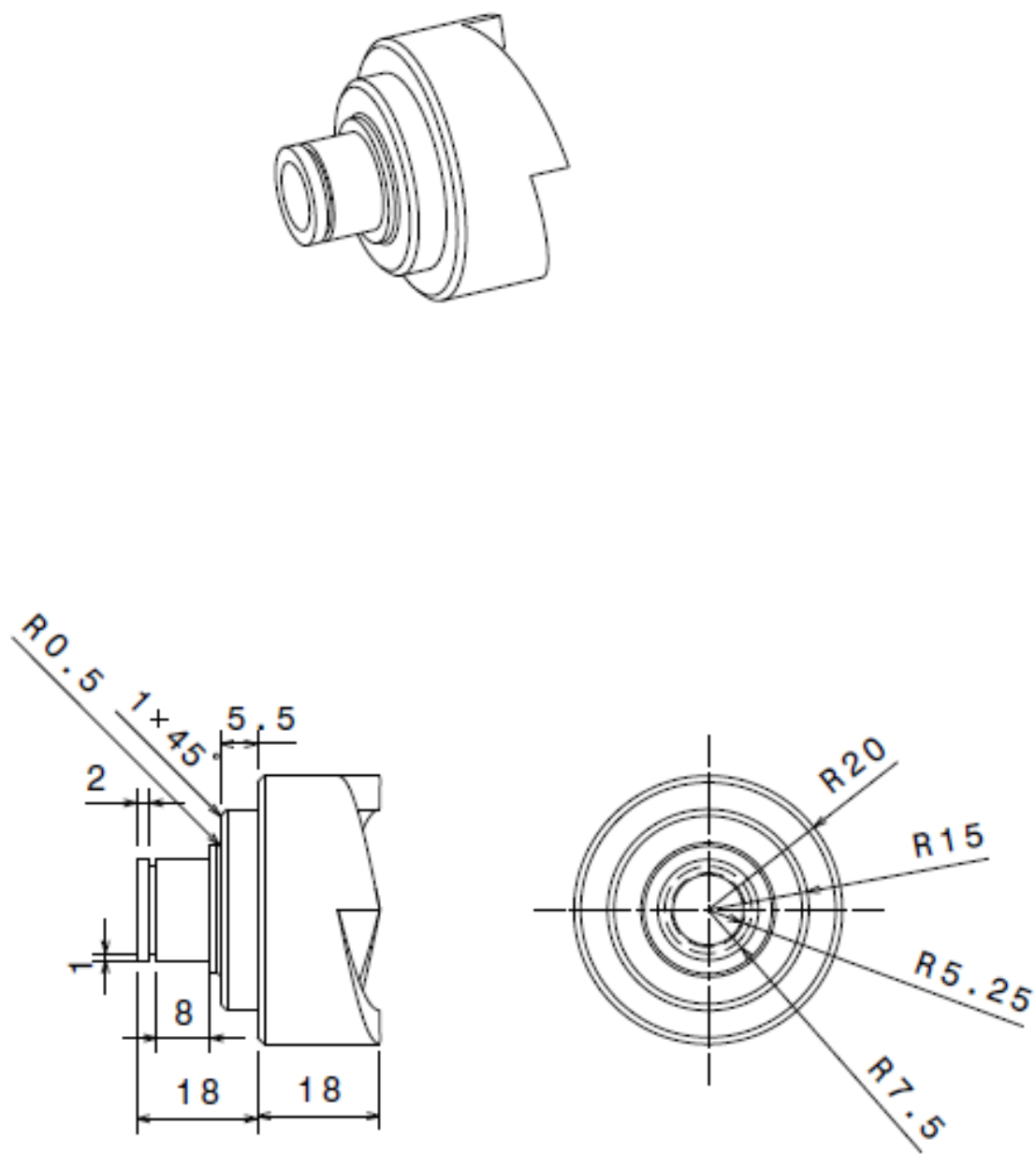


Figure E. 36 KERS spiral jaw clutch 2

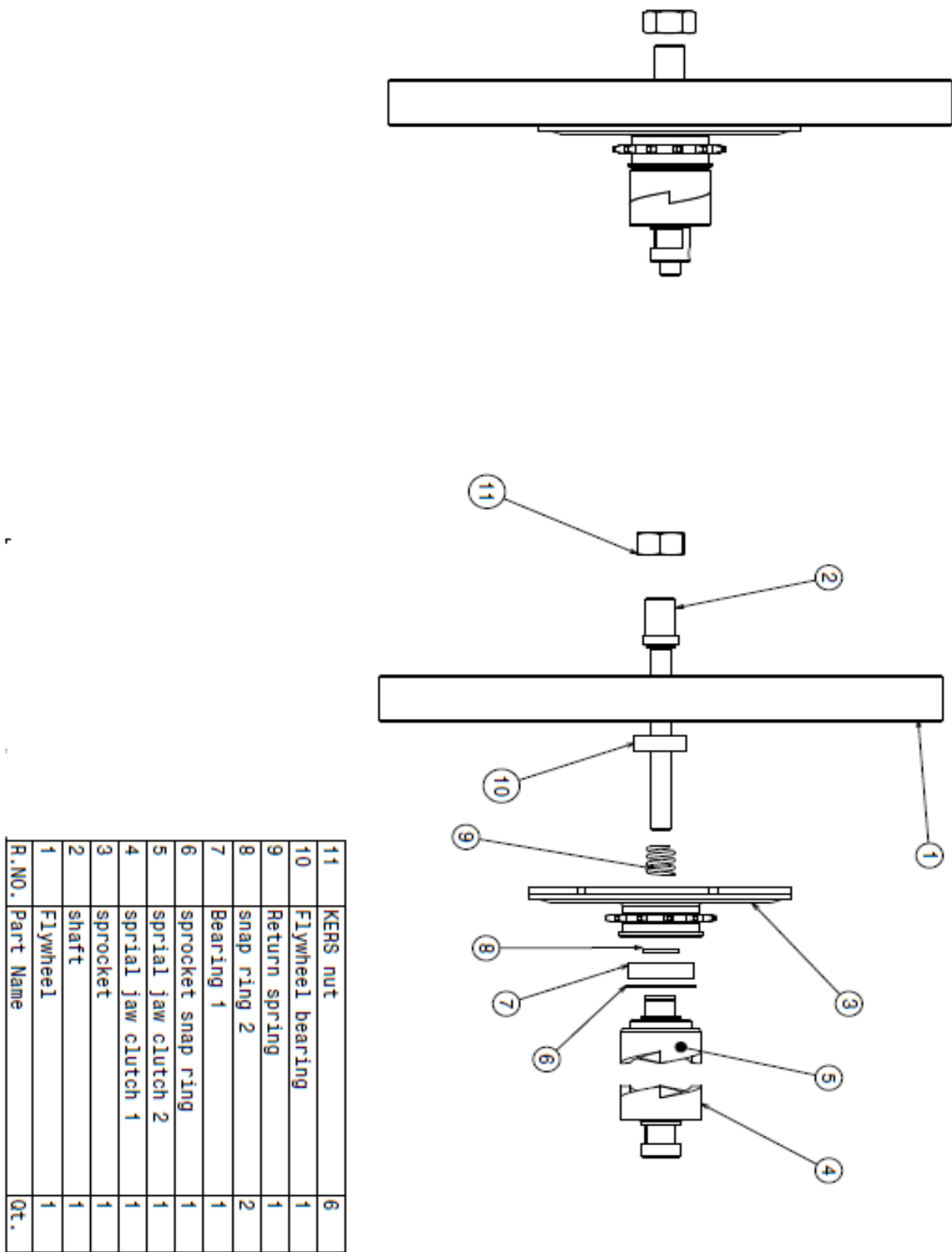


Figure E. 36 KERS assembly