DESIGNING A MECHANISM OF LIFTING SUSPENSION IN WHEELED ARMOURED VEHICLES

TEKERLEKLİ ZIRHLI ARAÇLAR İÇİN SÜSPANSİYON GERİ ÇEKME MEKANİZMASI

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AHMET ÇAĞKAN ÇEVİK

ABSTRACT

DESIGNING A MECHANISM OF LIFTING SUSPENSION IN WHEELED ARMOURED VEHICLES

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Master of Science, Department of Mechanical Engineering
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Swimming for armored amphibious vehicles is particularly important for both attack/defense sustainability and crew life during operation. The wheels of the amphibious vehicle due to the buoyancy force in the water form as an obstacle during the movement of the vehicle in the water according to the Archimedes Principle. This is especially the case for vehicle-carrying amphibious vehicles, vibration, trembling etc. There is a risk of immersing the vehicle and the vehicles it carries on it. Therefore, designs had been made in the past by removing the suspension arms and reducing the area forming the resistance, but different solutions have been sought as the problem cannot be solved. As a solution, the rack and pinion gear design mechanism used such in steering systems has been tested to withstand the high force values from the wheels and suspension elements. This mechanism design approach was used for the first time for such systems exposed to force of this magnitude. The analysis procedures solved by adhering to some pioneer gearing standards were solved by the ANSYS Workbench with non-linear calculations. It is proved that the mechanism works by applying Kinematic Analysis solutions with ADAMS program. In terms of the development of the design, 1/1 scale vehicle in FLUENT program, the system that lifts the wheels and the design of the wheels in which the design is not used by dissolving them together with the system in the water environment. It has been seen that analytical solutions and analysis solutions overlap with the proposed solution.

Keywords : Armoured Vehicles, Suspension System, Mechanism, Gear Design

ÖZET

TEKERLEKLİ ZIRHLI ARAÇLAR İÇİN SÜSPANSİYON GERİ ÇEKME MEKANİZMASI

Ahmet Çağkan ÇEVİK

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Suda yüzen araçlar için yüzme işlemi özellikle operasyon sırasında hem saldırı/savunma sürdürülebilirliği hem de mürettebat hayatı için önem arz etmektedir. Suda operasyon sırasında kaldırma kuvvetinden dolayı yüzen aracın tekerlekleri aşağı inerek aracın suda hareketi sırasında engel oluşturmaktadır. Bu durum özellikle araç taşıyan amfibik araçlar için titreşim, silkeleme vb. durumlardan dolayı aracı ve üzerinde taşıdığı araçları batırma riski ortaya çıkmaktadır. Bu yüzden geçmişte süspansiyon kollarının kaldırılarak direnç oluşturan alanın azaltıldığı tasarımlar yapılmıştır, fakat sorun giderilemediği için farklı çözüm yolları aranmıştır. Çözüm olarak yönlendirme sistemlerinde kullanılan kremayer ve pinyon dişli tasarım mekanizması kullanılarak tekerleklerden ve yürüyen akşamlardan gelen yüksek kuvvet değerlerine bağlı dayanımı test edilmiştir. Bu mekanizma tasarımı yaklaşımı, bu büyüklükteki kuvvete maruz kalan sistemler için ilk defa kullanılmıştır. Belli dişli standardlarına bağlı kalarak çözülen analitik işlemler, ANSYS Workbench ile 3 boyutlu doğrusal olmayan hesaplamalar yöntemi ile çözdürülmüştür. ADAMS programı ile Kinematik Analiz çözümleri uygulanarak mekanizmanın çalıştığı ispatlanmıştır. Tasarımın gelişimi açısından FLUENT programında 1/1 ölçekli araç, hem tekerlekleri kaldıran sistem hem de tasarımın kullanılmadığı tekerleklerin su ortamında indiği sistem ile birlikte çözdürülerek karşılaştırma yapılmıştır. Önerilen çözümde iyileştirme ispatlanarak, analitik çözümler ile analiz çözümlerinin birbiriyle örtüştüğü görülmüştür.

Anahtar Kelimeler : Zırhlı Araçlar, Süspansiyon Sistemleri, Mekanizma, Dişli Tasarım

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SYMBOLS AND ABBREVIATIONS

Symbols

σ	Stress
τ	Shear Stress
σ_b	Bending Stress
F_t	Tangential Force
Y	Lewis Form Factor
K_v	Velocity Factor
J	Bending Stress Geometry Factor
m	Module
K_O	Overload Factor
K_m	Load Distribution Factor
K_S	Size Factor
K_r	Reliability Factor
K_T	Temperature Factor
s	Factor of Safety
P_c	Maximum Value of Contact Stress
E	Modulus of Elasticity
μ	Poisson's Ratio
d_p	Pitch Diameter
C_P	Elastic Coefficient
K_B	Rim-Thickness Factor
Y_J	Geometry Factor for Bending Strength
ϵ_r	Contact Ratio
Z_R	Surface Condition Factor

Z_I	Geometry Factor for Pitting Resistance
σ_{all}	Allowable Bending Stress
Y_N	Stress Cycle Factor for Bending Stress
K_R	Reliability Factor
S_c	Allowable Contact Stress
Z_N	Stress Cycle Life Factor
Z_W	Hardness Ratio Factor for Pitting Resistance
S_H	Safety Factor
S_F	Safety Factor
H_B	Brinell Hardness
HRC	Rockwell Hardness C Class
C_ψ	Helical Overlap Factor
K_f	Fatigue Stress Concentration Factor
I	Surface Strength Geometry Factor
C_ψ	Helical Overlap Factor
K_f	Fatigue Stress Concentration Factor
m_B	Backup Ratio
z	Number of Teeth

Abbreviations

AGMA	American Gear Manufacturer Association
ANSI	American National Standards Institute
HRC	Rockwell Hardness C Class

1. INTRODUCTION

From the last thirty years, Turkey has been facing with terrorist attacks, expensive technological sales unwillingly, financial breakdown strikes etc. Those kind of unwanted and non-logical strikes unfortunately harmful both national economy, social life and country survivability. Those stated gaps of security provides and supports the national security vehicle for Police and Armed Forces to make more powerful, sustainable and local production. Exclusively, in Turkey where is a land on three sides surrounded by seas has to be more brave for develop and advance navy and land forces' vehicles. For instance about a combined vehicle which would be used by navy and land forces were declared by Turkish government in terms of attack scenario of go ashore. Moreover, there are plenty vehicle design and those had been implemented by several defence companies in Turkey. Most of them is named and designed as amphibious vehicle. Sadly, gathering two different types of vehicle demand into one is utterly arduous and proned to be generating problem for both sides. The most important problem is being balancing the vehicle in the sea is relatively difficult situation. That stated problems may be destructive for the operation achievement and can make loss of time and on the top of it may come up to a loss of crew members or soldiers.

From out of the box, amphibious vehicles can make difference on battlefield while using for both in land and water. It can be planned to strike from offshore to coast or the other way. In order to achieve those kind of crucial attacks, it must be had some specialized equipment on. As underline from the beginning, below the waist the rear side of the vehicle have water jets and orientation plate is a good example for special component owned by amphibious vehicles. Whilst fording the vehicle before swimming operation or after swimming operation, center of gravity and geometry of the vehicle plays tremendous role such as while in swimming.

In that study, an amphibious vehicle balance problem is tried to be solved while doing both operations; swimming and transportation. Due to unwanted obstructions from vehicle itself can affect the balance of the vehicle, then the consequence may be sinking. In order to find the solution for that problem and increase the effectiveness of both navy and land forces, the most optimum design and solution is discussed. In that mentioned design solution has been fortified by using analysis programs and analytical solutions with the help of some general gear design standards. As problem definition, while in swimming operation the wheels and suspension systems come down due to its weight and it affects on water currency. That directly reaches negatively to balance of the vehicle. As a solution, for controlling and directing the wheel with steering system components rack&pinion has been deployed for lifting up and down for suspension systems arms and vehicle’s rims. The robust characteristics of the rack with pinion is extremely suitable and sufficient to use in that manner.

That study contains great improvement for national security. Because gathering information for armoured vehicles has drastically hard issue in order to indicate proper answer. Other types of solutions came up with huge cost and non-local components usage after consideration. For that development, both localization ratio for problem solving has been achieved and known mechanism for different solution was used for such a contrasting design.

2. METHODS AND STANDARDS OF STUDY

In that study, especially for gear some standards are being used for exact results. ISO, AGMA are widely used in researches. Especially AGMA is being used in that study. AGMA plays significant role when considering about gear design from the beginning.

2.1. Literature Study

The earliest trial to understand stress on gears was originated and revealt firstly by W. Lewis in 1892 for spur gears. This attempt was wrapped for try to find root of a gear tooth stress. Lewis based his analysis that the gear tooth can be demonstrated as cantilever beam such as used in construction site structure mentality. There is two important point according to him. One is the root fillet of the gear and the other is perpendicular to tangential arc side of the gear. [1]

After it was realized that tooth arc that occur root fillet can cause stress concentration effect. By Dolan and Broghamer used the photoelastic models of gear teeth so that Lewis formula has an ability to cover real terms in order to find stress concentration factor. With the help of these visualization the stress and its concentration effects proved to be based on high bending stresses. Those mentioned visualization techniques were used for design a spur gears to observe most critical point on gear. First stress measurement with photoelastically and analytically according to finite element method analysis programs and integral methods by Winter and Hirt. In terms of popularity of the gear design by experiments about bending stress analysis the American Gear Manufacturers Association (AGMA) published their one of the most important standard regarding gears generally based on Lewis' original equation. The standard was established in 1982 and in modern mechanical world, if any gear part is being likely to used by engineers then AGMA standard must be examined and used. Once the world changing FEA has been revealed, the first attempts about spur gears was modelled from 2D in terms of simplifiying the calculation. As well as using the spur gears for prior analysis, the only resolution has been creating mesh then calculating bending stress. Comparing to analytical method, FEA has cost effective, competent and parametric. After the calculation of accurate result for bending stress, engineers were stubborn to generate these into a design that must be involved in safety factor. Generated calculation by FEA for contact analysis has been inputted due to its high importance of deformation. The first trial was done by Hertz Equation before to find deformable two cylindrical bodies. The expanded works are being held by researchers. Once cluster technology become irresistable to use, the mesh sizes evolves to nano size in order to find more accurate and significant numbers.[2-9]

First ANSYS and Lewis Equation comparison by static structural analysis of gear tooth has been examined by Pravin B. Sonawane, P.G. Damle. In their study, between two contacted cylinders have been obtained from Hertz and Lewis equation. With the help of FEM analysis, the whole body stiffness owing to rotation of gear pairs by bending deflection, shearing displacement and contact deformation. In order to examine their study, usage of both analytical solutions and FEM approach shows the way of calculation. [10]

As per understand various size of modules with different materials using AGMA by S. Prabhakaran and S. Ramachandra showed the FEM usage in order to evaluate the module with materials and they compare their analytical results with AGMA Standards. [11]

The effect of gear tooth root radius has been examined according to find involute spur gear with usage of any static loading. The research in order to consider the world's most usable and rotatable product which is gear in terms of behaviour of load. According to Jani, Snehal; Shah, Jinesh, gear failure can be examined from tooth bending strength. In that study focused on helical pair gears in order to gain an information about its contact and bending stresses. Same evaluation or approach is valid for spur gears or rack as well. [12-14]

According to the large amount of parametric factor are used by researchers and engineers in AGMA Standard. Their work is related to eliminate them within systematically for design procedure steps. In that way, they try to increase the stress value unnecessarily. [15-16]

2.2. Machine Elements

According to classical mechanical formulas of stress is the result of internal forces or the forces of external effects of one to another. Those mentioned internal forces are caused while load is applying to the part and internally reaction forces would appear. The most common load affect on an object longitudinally. Other types would include as axially, torsional loaded, twisting, bending etc. In order to examine the force reaction on load that causes stress. For understading that, examiner can look inside the part.

As long as cutting through the element can show the result of the load. In every cases, destructive testing methods will not be a choice. Therefore, cutting through action must be done by hypothetically based on create a diagram of determine internal/external force effects.

From the perspective of loading must represent the stress due to some area of the object. As stated previously, hypothetically cutting the object are going to reveal cross sectional area. In such cases, stress is internal distribution of forces within a body. The distribution may be uniform or not. And it depends of the type of external force naturally. In order to calculate those stress, force must be measured by engineering tools because cross sectional area like Figure 2.1; calculations are extremely easy to find and according to normal stress formula only unknown is left as force.

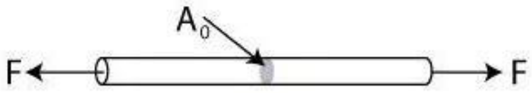


Figure 2.1 - Load with Area

<u>Types of Stresses</u>

Normal Stress; is stress that acts perpendicular to the body. It is given in formula (1.1).

$$\text{Stress } \sigma = \frac{Force}{Cross\ Sectional\ Area} = \frac{F}{A} \qquad (1.1)$$

- Tensile Stress; is one of the most commonly subjected stress in modern engineering problems and circumstances in Figure 2.2. It is the stress that acts on two opposite forces subjected to a body. Instinctively, tensile stress lean to increase the length of the body and decrease the area of the portion. Because without any external effect, nothing change the volume of the body. Application of the force is as same as normal stress which is acted on perpendicular. If the angle changes, tensile strain involves in a problem.

Figure 2.2 - Tensile Stress Illustration

- Compressive Stress; is commonly encountered stress as well as tensile. In comparison with the tensile is that only difference comes from the force applying. Compressive forces are approaching each other in Figure 2.3. Thus, compressive stress lean to decrease the length of the body and increase the area of the body.

Figure 2.3 - Compressive Stress lllustration

- Shear Stress; is another stress type but the difference is coming from the opposite forces apply as eccentricaly in Figure 2.4. Basically, forces apply on the object and it is parallel to the object's cross sectional area as given in (1.2). As a result the object shape is deformed. Practically, the shear stress may not be distributed uniformly.

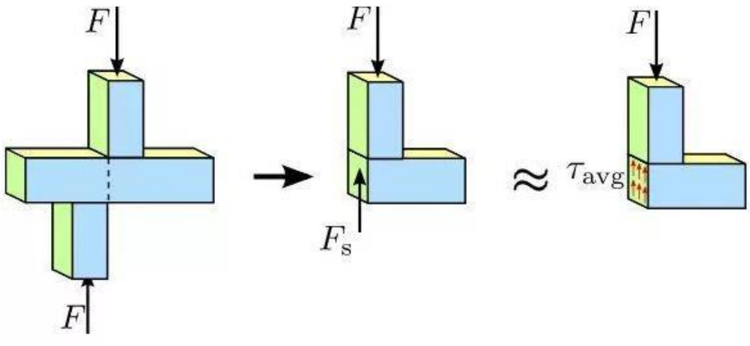


Figure 2.4 - Shear Stress Illustration

$$\tau = \frac{Force}{Shear\ Area} = \frac{F}{A} \tag{1.2}$$

- Bending Stress; is more spesific than the normal stress which is caused in a body subjected to load. When a load is applied perpendicular in Figure 2.5 to the length of a beam in some particular points that causes object to a bend. In order to determine bending stress in a beam, the classical formula (1.3) which is below is used. Bending stress vary from plastic bending to complex bending.

$$\sigma_b = \frac{(Calculated\ Bending\ Moment).(Vertical\ Distance\ from\ Neutral\ Axis)}{Moment\ of\ Inertia\ Around\ the\ Neutral\ Axis} = \frac{M.y}{I} \tag{1.3}$$

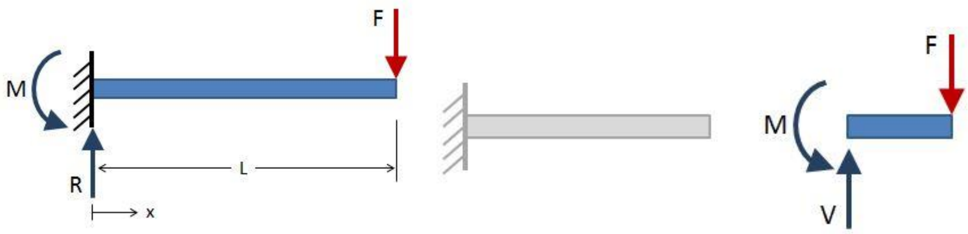


Figure 2.5 - Bending Moment with Load

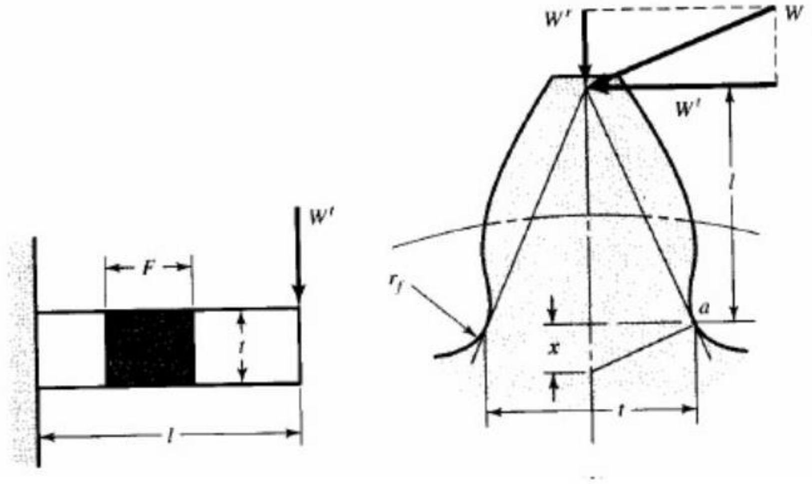


Figure 2.6 - Bending Stress on a Single Teeth of Gear

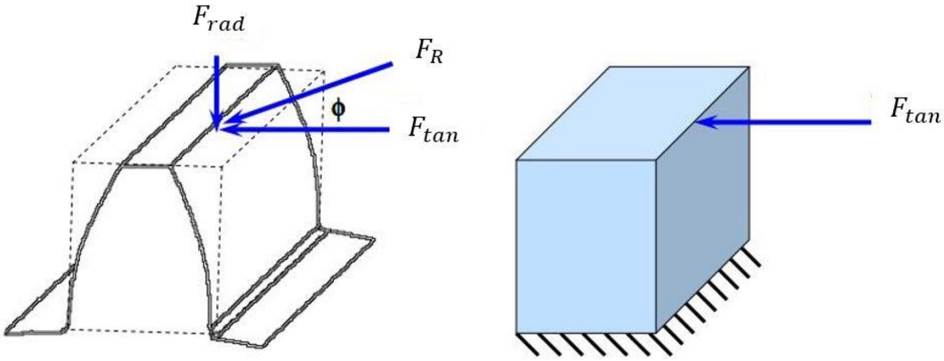


Figure 2.7 - Distribution of Loads on Sinle Teeth

In the gear design, surface strength and tooth root (fillet) strength are the essential for one of the main reason regarding to gear failures in various types of gear pairs. Therefore, analysing and making calculations are widely used in order to find optimal design, gear ratios etc. Stress analysis plays key role of minimize failure. Especially, while considering to find best design for pairs two stress types must be found which are bending stress and contact stress.

The classical and most commonly used approach to find gear stress determination is that consideration of the gear with its tooth as cantilever beam Figure 2.6-2.7. With the aid of that method, analytic calculations get very easy to implement. However, with the permission of the finite element method programs, exact predictions and computations of the stress can be found and help to compare with the analytic solutions.

First stress measurement with photoelastically in Figure 2.8 and analytically according to finite element method analysis programs and integral methods by Winter and Hirt. [4]

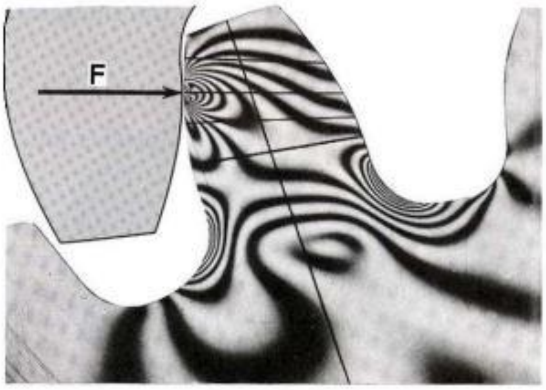


Figure 2.8 - Photoelastic Analysis of Two Gear

The photo shows that the gear contact point sustains in every point on arc curvature. One of the high stress was obtained in that exact point. While rolling sliding on top of them surface fatigue may appear. At point A and B, due to bending stress compressive and tensile stress are revealed. According to Winter and Hirt, compressive stress has a bigger magnitude due to the radially tooth force Ft. The greatest stress exist on the edge where the black shady lines are close to each other. In two gear teeth in contact illustrates that two different types of relatively great stress area has occured. At A point the tensile stress has been seen, on the other area at B point compressive stress has been examined due to Force. The important part is that compressive stress is bigger than the F Force. Because radially applied on part may maintain mentioned effect on teeth. It also might lead to fatigue failure. At C Point contact stress condition has occured at it cause that surface fatigue of the tooth.[3]

According to Lewis who thought that gear tooth can be considered as cantilever beam; [10]

- Only static force at the tip of the single tooth in static condition can be handled.

- Other force generator which called radial component must be negligible to implement the calculations valuable.
- In terms of average load usage, distribution of load uniformly must be generated to the one force and it must be act on full face width.
- Forces may allow friction on tooth sliding that must be negligible.
- Stress concentration tooth radius must be negligible.

The reason why those approach and negligible portions are being used for constant face parabola the teeth profile geometry is hard to find and implement solutions properly. Lewis had been derived the equation from General Moment Equation. The reason why he used that step is that while touching and pressing on teeth would move like beam, and in engineering world its effective movement has a name which is called cantilever.

Lewis' essential approach of spesific formula has one directional way of thinking; tip of the tooth. In his day, even though manufactured best gear teeth may not be accurate like today. Due to unleveled gear tooth side surface, most commonly one sinle tooth carry all the loads and that means load is not carried with other tooth. Once it carries all the load with one single teeth, it may occur and handle the greatest stress while rolling over each other. That's why Lewis focused on those incident as much as he would have faced in his relatively technological life.

$$\sigma = \frac{M.I}{c} = \frac{6.F_t, l}{b.t^2} \tag{1.4}$$

While considering the real tangential force, due to moment calculation only logical move to calculate stress is to find and use real tangential force in formula (1.4). Gear-Force combination is related with pyhtogorean theorem. Radial force almost affects on tooth radius edge, therefore it can be taken negligible. In Lewis equation, maximum stress occurs in tooth end.

2.3. Formulation of Lewis Equation

In Figure 2.9 illustrates as the general gear force illustrations are;

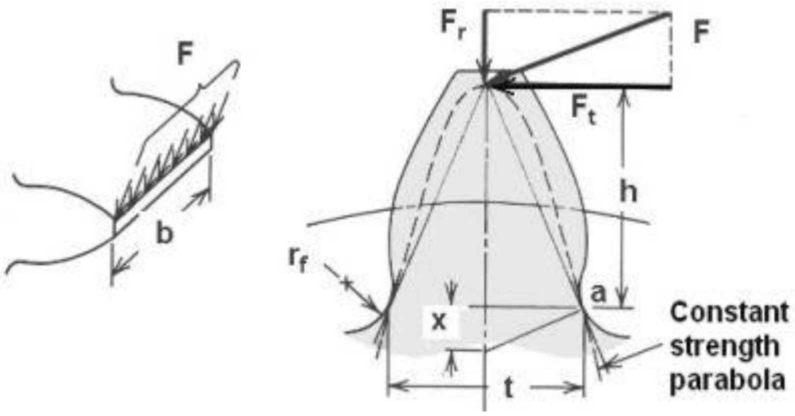


Figure 2.9 - Display of Single Teeth as Cantilever Beam

At point A; acccording to classical formula of bending stress is that

$$\sigma_b = \frac{M.y}{I} = \frac{6.F_t.h}{b.t^2} \tag{1.5}$$

With the same triangle in Figure 2.8,

$$\frac{t/2}{x} = \frac{h}{t/2} \rightarrow \frac{t^2}{h} = 4x \tag{1.6}$$

Combining the two equations

$$\sigma = \frac{6.F_t}{4.b.x} \tag{1.7}$$

$$y = \frac{2x}{3p} \tag{1.8}$$

In that equation, Y is defined as Lewis form factor.

$$\sigma = \frac{F_t}{b.p.y} \tag{1.9}$$

In SI units, gear are widely use in standard modules. Hence by substituting,

$$p = \pi.m \tag{1.10}$$

So, we get

$$\sigma = \frac{F_t}{b.\pi.y.m} \tag{1.11}$$

In order to use Lewis form factor graphic,

$$Y = \pi.y \tag{1.12}$$

$$\sigma = \frac{F_t}{b.Y.m} \tag{1.13}$$

Hence, standard Lewis Equation which has been derived from (1.5) to (1.13) for tooth bending stress was found in that way. In Table 2.1, Y or y can be thought as tooth shape factor and it also vary with the number of teeth in the gear.

Table 2.1 - Lewis Form Factor

Number of teeth	Ø = 20° a = 0.8m* b = m	Ø = 20° a = m b = 1.25m	Ø = 25° a = m b = 1.25m	Ø = 25° a = m b = 1.35m+
12	0.335 12	0.229 60	0.276 77	0.254 73
13	0.348 27	0.243 17	0.292 81	0.271 77
14	0.359 85	0.255 30	0.307 17	0.287 11
15	0.370 13	0.266 22	0.320 09	0.301 00
16	0.379 31	0.276 10	0.331 78	0.133 63
17	0.387 57	0.285 08	0.342 40	0.325 17
18	0.395 02	0.293 27	0.352 10	0.335 74
19	0.401 79	0.300 78	0.360 99	0.345 46
20	0.407 97	0.307 69	0.369 16	0.354 44
21	0.413 63	0.314 06	0.376 71	0.362 76
22	0.418 83	0.319 97	0.383 70	0.370 48
24	0.428 06	0.330 56	0.396 24	0.384 39
26	0.436 01	0.339 79	0.407 17	0.396 57
28	0.442 94	0.347 90	0.416 78	0.407 33
30	0.449 02	0.355 10	0.425 30	0.416 91
34	0.459 20	0.367 31	0.439 76	0.433 23
38	0.467 40	0.377 27	0.451 56	0.446 63
45	0.478 46	0.390 93	0.467 74	0.465 11
50	0.484 58	0.398 60	0.476 81	0.475 55
60	0.493 91	0.410 47	0.490 86	0.491 77
75	0.503 45	0.422 83	0.505 46	0.508 77
100	0.513 21	0.435 74	0.520 71	0.526 65
150	0.523 21	0.449 30	0.536 68	0.545 56
300	0.533 48	0.463 64	0.553 51	0.565 70
Rack	0.544 06	0.478 97	0.571 39	0.587 39

* Stub teeth + Large fillet

2.3.1. Spur Gear – Modified Lewis Equation

The Modified Lewis Equation is;

$$\sigma = \frac{F_t}{K_v'.b.Y.m} \tag{1.14}$$

Where K_v' is known as Velocity Factor and it can be found in (1.14) . V is for velocity (m/s). [15]

There ise three different type of gear manufacturing process for selection of Velocity Factor.

- Used for cut or milled teeth of gears in (1.15) which is not manufactured carefully in Figure 2.10.

$$K_v' = \frac{6}{6+V} \tag{1.15}$$

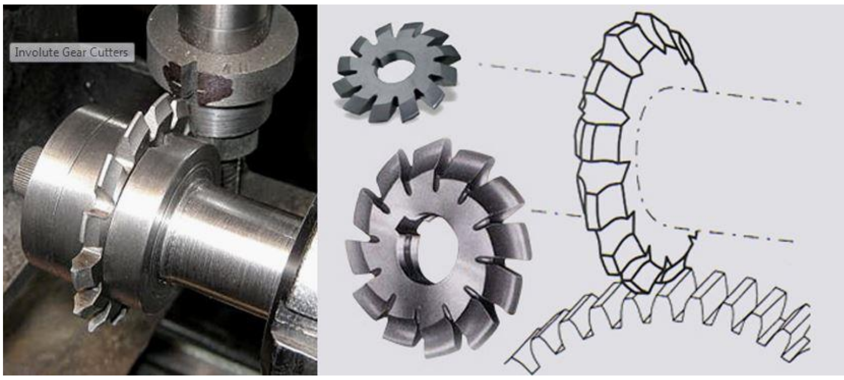


Figure 2.10 - Cut and Milled Teeth Manufacturing

- Used for hobbed and shaped gears in Figure 2.11-2.12 which has been formulated in (1.16).

$$K_v{}' = \frac{50}{50 + (200.V)^{1/2}} \tag{1.16}$$

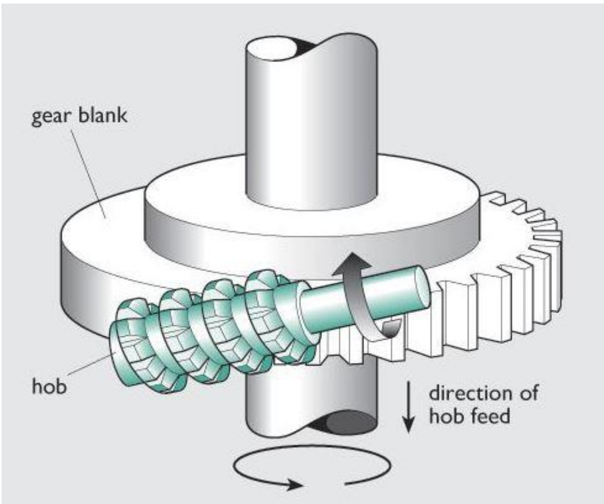


Figure 2.11 - Hobbed and Milled Teeth Manufacturing

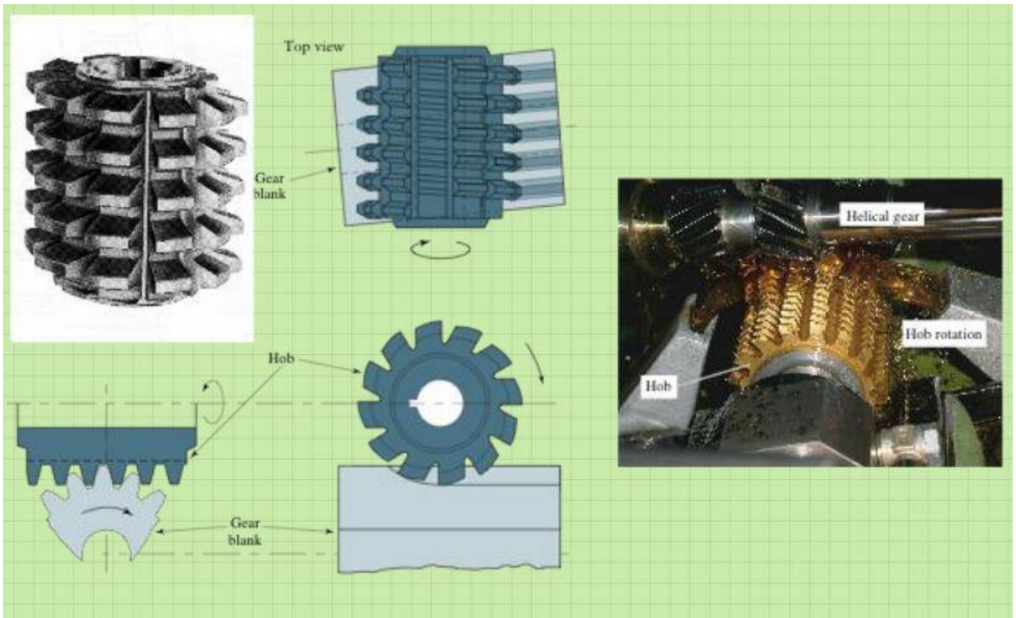


Figure 2.12 - Hobbed and Milled Teeth Manufacturing

- Used for high precision shaved and ground teeth in Figure 2.13 which has been formulated in (1.17)

$$K_v' = \left[\frac{78}{78 + (200.V)^{1/2}}\right]^{1/2} \quad (1.17)$$

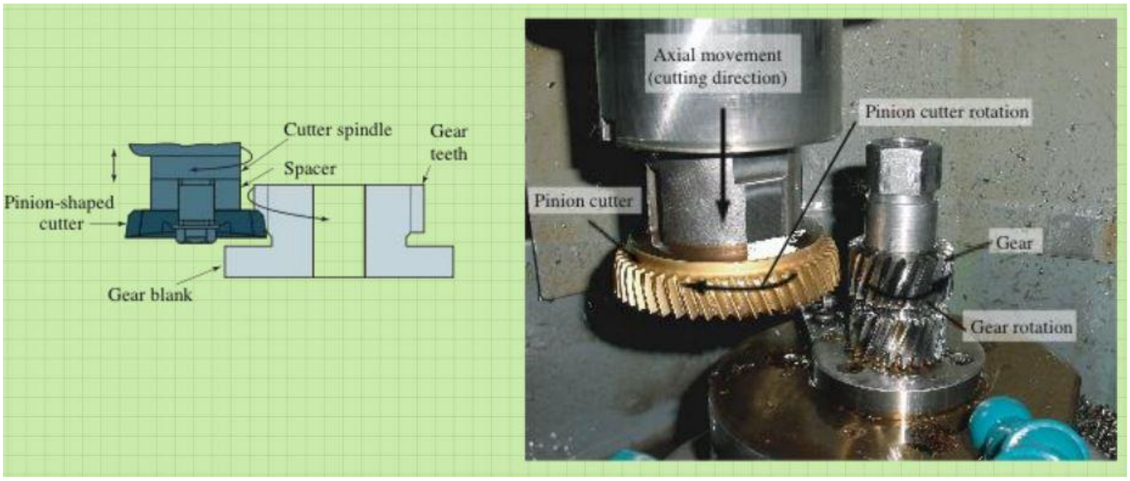


Figure 2.13 - High Precision Shaved and Ground Teeth Manufacturing

2.4. Hertzian Stress Theory

To understand the surface-contact definition properly, we must examine Hertz Equation. Actually, first of all assumption of examining two gear pairs in contact was calculated by [14] and then it evolves to Hertzian Stresses.

The general definiton of Hertz Stresses appears when one part with radial surfaces contact with a plate which can be called as rack so its radius is to be infinite or two different curved surfaces are in contact with their rigid center. For example, if two cylinders were in contact, the contact region would be in a line, however if mentioning about two different spheres then the contact region would be a point. Nonetheless, all of the theorotical assumptions are not appeared in real world by reason of elastic deformation was resulted. That stated stresses are called as Hertzian Stresses as in (1.18).

In equation shows that the contact stresses between two cylindrical parts in Figure 2.14.

$$P_c = \frac{2.F}{\pi.b.l} \tag{1.18}$$

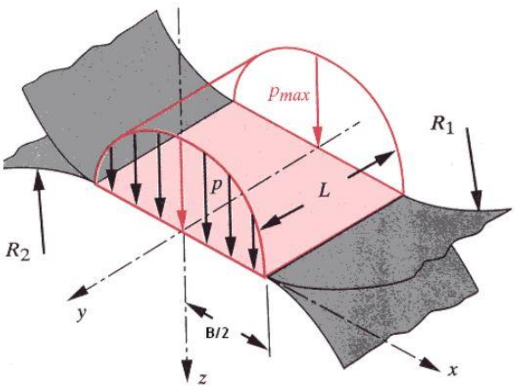


Figure 2.14 - Load Distribution on Line of Action

$$b = \sqrt{\left[\frac{2P.\left(\frac{1-\mu_1^2}{E_1}+\frac{1-\mu_2^2}{E_2}\right)}{\pi.l.\left(\frac{1}{d_1}+\frac{1}{d_2}\right)}\right]} \tag{1.19}$$

Half width of the deformation can be expressed as above equation (1.19).

- P_c= maximum value of contact stress (N/mm^2)
- F= force pressing the two cylinders together (N)
- b = half width of deformation (mm)
- l = axial length of cylinders (mm)
- d_1, d_2 = diameters of two cylinders (mm)
- E_1, E_2 = modulus of elasticity of two cylinder materials (N/mm^2)
- μ_1, μ_2= Poisson's ratio of the two cylinder materials (unitless)

Substituting the both sides and squaring both values that we get a new equation. In that equation P_{max} which derived from (1.20) can be written as σ_c.

$$\sigma_c{}^2 = \frac{1}{\pi}.\left(\frac{F}{l}\right).\left[\frac{\left\{\frac{1}{r_1}+\frac{1}{r_2}\right\}}{\left\{\frac{1-\mu_1^2}{E_1}+\frac{1-\mu_2^2}{E_2}\right\}}\right] \tag{1.20}$$

For cylinders, the contact region appears as rectengulat shape while in contact. If one of the cylindrical gear is infinite gear or as known as rack then it must be considered that r_1 goes to infinite as well. Therefore $\frac{1}{r_1}$ must be zero as limitation.

The assumption of calculation result derived according to other formulas. In those equation, pure bending of relatively short beam and elliptic distribution as tooth contact are not calculated properly and they have some misalignment for output approach due to elastic compression. Some research and evaluation shows that the size and shape of the gear or tooth must be inserted into equation what they may have supported. As discussed previously, environmental conditions, design requeirements and safety levels plays very important role on contact line.

Wear fatigue occurs in the nearist point of pitch line. So in hertzian theory the radii of the curvature in (1.21) and (1.22) of the tooth profile is that

$$r_1 = \frac{d_p sin\phi}{2} \tag{1.21}$$

$$r_2 = \frac{d_g sin\phi}{2} \tag{1.22}$$

- ϕ = Pressure angle
- d_p, d_g = pitch diameters of the pinion and gear

Especially, while combining the mechanical properties of gear and pinion; AGMA defines and serves a new formula related to gear pair simplification called Elastic Coefficient as (1.23).

$$C_P = \left[\frac{1}{\pi.\left\{\frac{1-\vartheta_P^2}{E_P} + \frac{1-\vartheta_G^2}{E_G}\right\}}\right]^{1/2} \tag{1.23}$$

With the adding of Velocity Factor (K_v) can be written as;

$$\sigma_c = -C_P.\left[\frac{K_v.F_t}{b.cos\phi}.\left(\frac{1}{r_1} + \frac{1}{r_2}\right)\right]^{1/2} \tag{1.24}$$

Where the sign is negative due to σ_c is compressive stress as (1.24).

Besides the addition of K_v Velocity factor, so many uniformity was occured on gear system. In terms of finding endurance limits or estimated life of the gear system, S-N curves must be known and calculated by Dudley. The data shows that AGMA Standard Steel Grade 1 and Steel Grade 2 as illustrated in Figure 2.15. [15].

The Grade 1 has property of good quality but non-uniformly translated and Grade 2 is best manufactured quality. L_1 and L_{10} are reliability factor of tooth.

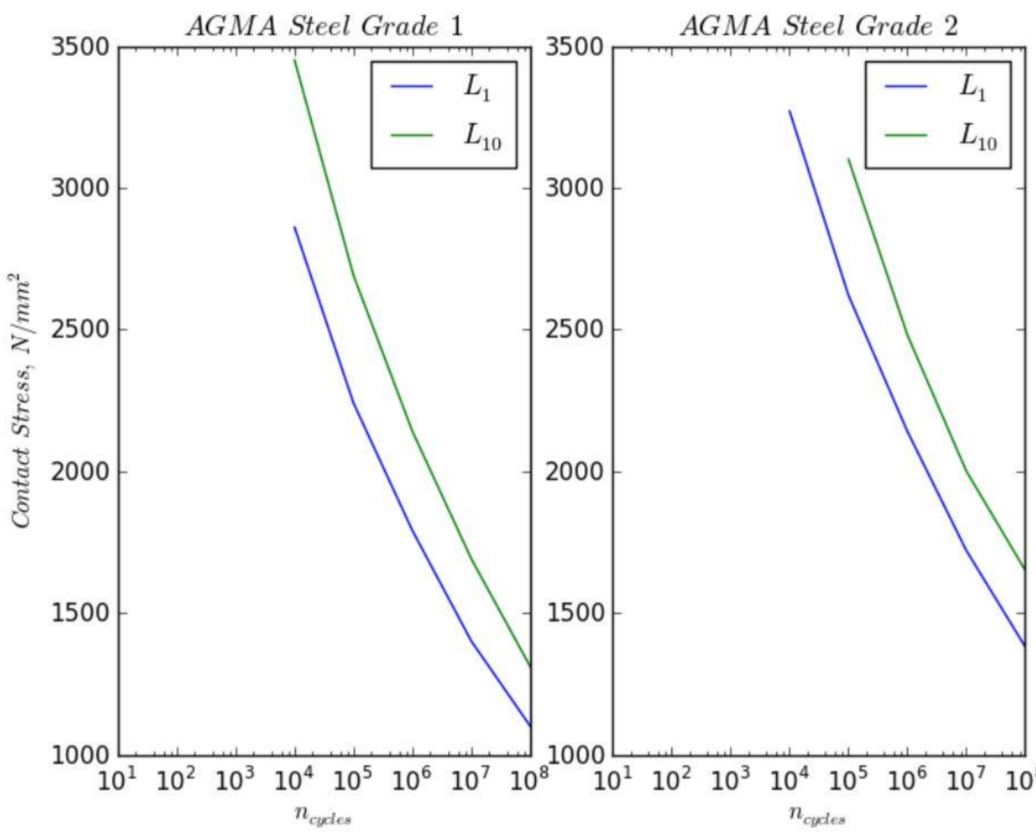


Figure 2.15 - Reliability Factor Comparison by Steel Grade

2.5. Formulation of AGMA Tooth Bending Stress Equation

American Gear Manufacturing Association (AGMA) used Lewis formulation based with the additional factors in (1.25) and parameters which is considered as significant by AGMA.

$$\sigma = \frac{F_t}{b.J.m}.K_v.K_O.K_M \tag{1.25}$$

J is called Spur Gear Geometry Factor which includes Lewis Form Factor with a fatigue stress concentration.

AGMA's modification has some reasons. Due to photoelasticity image, the major amount of the stress gather in tangential force area and fillet area. So using that unknown as parameter is logical in some way.

$$J = \frac{Y}{K_f} \tag{1.26}$$

Fatigue Stress Concentration Factor can be found by using AGMA J Factor table as (1.26). From the evaluation of the gear pair, it is relatively easy to obtain necessary unknown of J in Table 2.2.

Table 2.2 - Fatigue Stress Concentration Factor

Number of teeth	Number of teeth in mating gear							
	1	17	25	35	50	85	300	1000
18	0.244 86	0.324 04	0.332 14	0.338 40	0.344 04	0.350 50	0.355 94	0.361 12
19	0.247 94	0.330 29	0.338 78	0.345 37	0.351 34	0.358 22	0.364 05	0.369 63
20	0.250 72	0.336 00	0.344 85	0.351 76	0.358 04	0.365 32	0.371 51	0.377 49
21	0.253 23	0.341 24	0.350 44	0.357 64	0.364 22	0.371 86	0.378 41	0.384 75
22	0.255 52	0.346 07	0.355 59	0.363 06	0.369 92	0.377 92	0.384 79	0.391 48
24	0.259 51	0.354 68	0.364 77	0.372 75	0.380 12	0.388 77	0.396 26	0.403 60
26	0.262 89	0.362 11	0.372 72	0.381 15	0.388 97	0.398 21	0.406 25	0.414 18
28	0.265 80	0.368 60	0.379 67	0.388 51	0.396 73	0.406 50	0.415 04	0.423 51
30	0.268 31	0.374 62	0.385 80	0.395 00	0.403 59	0.413 83	0.422 83	0.431 79
34	0.272 47	0.383 94	0.396 71	0.405 94	0.415 17	0.426 24	0.436 04	0.445 86
38	0.275 75	0.391 70	0.404 46	0.414 80	0.424 56	0.436 33	0.446 80	0.457 35
45	0.280 13	0.402 23	0.415 79	0.426 85	0.437 35	0.450 10	0.461 52	0.473 10
50	0.282 52	0.408 08	0.422 08	0.435 55	0.444 48	0.457 78	0.469 75	0.481 93
60	0.286 13	0.417 02	0.431 73	0.443 83	0.455 42	0.469 60	0.482 43	0.495 57
75	0.289 79	0.426 20	0.441 63	0.454 40	0.466 68	0.481 79	0.495 54	0.509 70
100	0.293 53	0.435 61	0.451 80	0.465 27	0.478 27	0.494 37	0.509 09	0.524 35
150	0.297 38	0.445 30	0.462 26	0.476 45	0.490 23	0.507 36	0.523 12	0.539 54
300	0.301 41	0.455 26	0.473 04	0.487 98	0.502 56	0.520 78	0.537 65	0.555 33
Rack	0.305 71	0.465 54	0.484 15	0.499 88	0.515 29	0.534 67	0.552 72	0.571 73

Other parameters such as Velocity Factor, Overload Factor and Load Distribution Factor can be calculated as below;

- In order to use K_v;

There ise three different type of gear manufacturing process for selection of Velocity Factor.

Used for cut or milled teeth of gears which is not manufactured carefully in (1.27)

$$K_v{}' = \frac{6}{6+V} \tag{1.27}$$

Used for hobbed and shaped gears (1.28).

$$K_v{}' = \frac{50}{50+(200.V)^{1/2}} \tag{1.28}$$

Used for high precision shaved and ground teeth (1.29)

$$K_v{}' = \left[\frac{78}{78+(200.V)^{1/2}}\right]^{1/2} \tag{1.29}$$

$K_V{}'$ has some graphic explanation in terms of velocity with cutting provedures. That demonstration shows the areas that fit for finding the unknown parameters.

In order to use K_O;

Overload factor in Table 2.3 which matches with the degree of non-uniformity while driving or rolling on.

Table 2.3 - Overload Factor

	Driven Machinery		
Source of Power	Uniform	Moderate Shock	Heavy Shock
Uniform	1.00	1.25	1.75
Light Shock	1.25	1.50	2.00
Moderate Shock	1.50	1.75	2.25

In order to use Table 2.4 for finding K_m;

Table 2.4 - Load Distribution Factor

	Face Width (mm)			
Characteristic of Support	0-50	150	225	400-...
Accurate mountings, small bearing clearances, minimum deflection, precision gears	1.3	1.4	1.5	1.8
Less rigid mountings, less accurate gears, contact across the full face	1.6	1.7	1.8	2.2
Accuracy and mounting such that less than full-face contact exists	Over 2.2	Over 2.2	Over 2.2	Over 2.2

A procedure has been derived by American National Standards Institute-American Gear Manufacturers Association, in order to calculate contact stresses in spur and helical gears. The main cause for using AGMA is that most of the examples had been covered by stated standard as well as being used it and almost all of the literature study has been endured by the researchers. In terms of gear standard exploration, the most countable and pioneering ways always rely on AGMA standards. [17-18]

Two basic formulation are used in AGMA methodology. One is for and called Bending Stress as in (1.30) and the other one is for and called Contact Stress. It is varying due to SI and US standards however we'll be using SI units for understanding the Metric systems. [21]

$$\sigma = F_t . K_v . K_o . K_s . \frac{1}{b.m} . \frac{K_m . K_B}{Y_J} \tag{1.30}$$

- F_t is the tangential transmitted load (N)
- K_o is the overload factor
- K_v is the dynamic factor
- K_s is the size factor
- b is the face width of the narrower member, in (mm)
- K_m is the load-distribution factor
- K_B is the rim-thickness factor
- Y_J is the geometry factor for bending strength (which includes root fillet stress-concentration factor)
- m is the transverse metric module

Many of the parameters can be estimated due to some special testing proofs or AGMA Standard Tables [21]

Before investigating or trying to calculate output of the system by designer who must consider and examine from the obtainable to unknown parameters such us below;

- Size of teeth
- Translational Load Magnitude
- Overload Condition
- Load Distribution

- Dynamic Behaviour of Load and System
- Root Fillet Dimension
- Rough Geometry like Pitch diameter, base diameter, face width
- Module
- Rim Support of the tooth

Other AGMA stress is contact Stress in (1.31) as known as Pitting Resistance;

$$\sigma_c = C_P.\sqrt{F_t.K_v.K_o.K_s.\frac{K_H}{d_p.b}.\frac{Z_R}{Z_I}} \tag{1.31}$$

- C_P is an elastic coefficient (N/mm2)
- Z_R is the surface condition factor
- d_p is the pitch diameter of the pinion (mm)
- Z_J is the geometry factor for pitting resistance

When the AGMA stresses has been calculated, it should be compared with Allowable Stress Numbers formulated in (1.32) and (1.33).

Allowable Bending Stress is;

$$\sigma_{all} = \frac{S_t}{S_F}.\frac{Y_N}{K_T.K_R} \tag{1.32}$$

- S_t is the allowable bending stress (N/mm^2)
- Y_N is the stress cycle factor for bending stress
- K_T are the temperature factors
- K_R are the reliability factors
- S_F is the AGMA factor of safety, a stress ratio

Allowable Contact Stress is;

$$\sigma_{all,c} = \frac{S_c}{S_H} \cdot \frac{Z_N . Z_W}{K_T . K_R} \qquad (1.33)$$

- S_c is the allowable contact stress, (N/mm2)
- Z_N is the stress cycle life factor
- Z_W are the hardness ratio factors for pitting resistance
- K_T are the temperature factors
- K_R are the reliability factors
- S_H is the AGMA factor of safety, a stress ratio

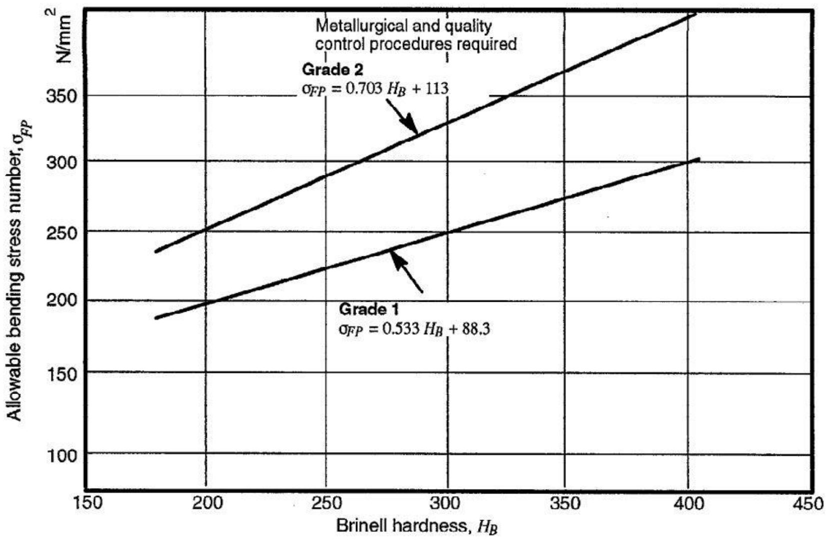


Figure 2.16 - Allowable Bending Stress Number by Steel Grade for Brinell Core Hardness

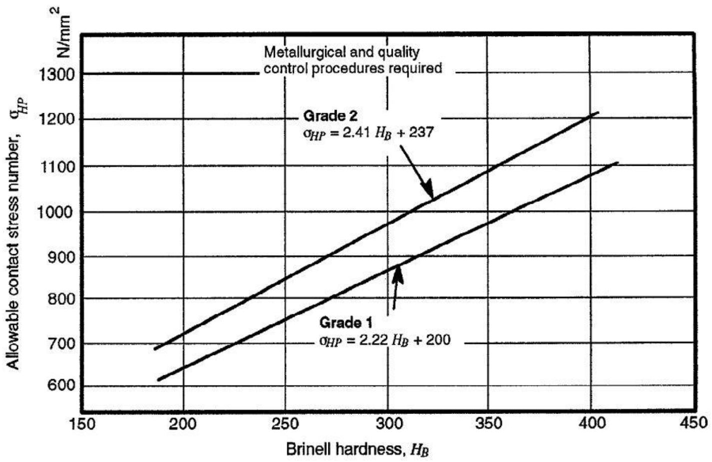


Figure 2.17 - Allowable Contact Stress Number by Steel Grade for Brinell Core Hardness

According tro evaluation of difference between case and core hardness of material Figure 2.16 and Figure 2.17, distractive testing might be offered but cutting the specimen through the axis, then measuring hardness. However due to the Jominy in Figure 2.18 quenching test structure, for some special material there is a plotted table by material. This is the most commonly used method for determinining the hardenability by Jominy and Boegehold. The details are in ASTM A 255-10 [22-23].

Essentially, the test is being used in choosing the sufficient combination of alloy steel and together with heat treatment in order to reduce thermal stress, unwanted fatigue or distortion. In the test we will use the graph of hardness change due to distance by material.

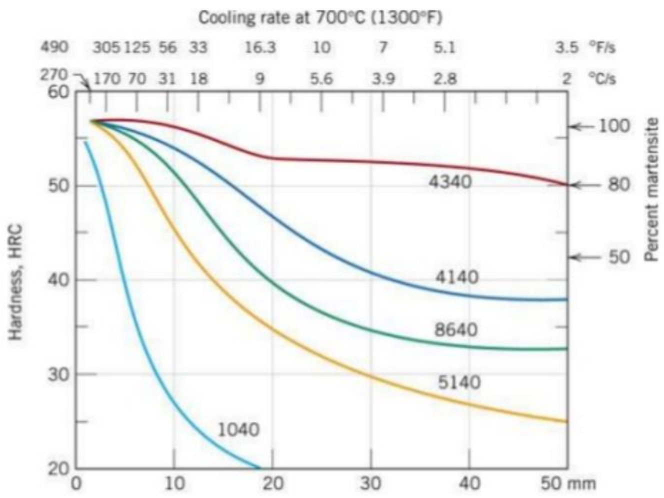


Figure 2.18 - Core Hardness Evaluation through Dimension by Material

Elastic Coefficient

Elastic coefficient which forulated in (1.34) is an factor that can be used in order to add material properties for both gear and pinion together. By doing this, output error for different behaviour of materials has been prevented and can be provided good acknowledgement about tangential force itself. Including Young Modulus of elasticity and Poisson's coefficient plays magnificent role in formulation.

$$C_P = \sqrt{\frac{1}{\pi.\left[\left(\frac{1-\nu_1^2}{E_1}\right)+\left(\frac{1-\nu_2^2}{E_2}\right)\right]}} \tag{1.34}$$

If the system designer uses the same material as in (1.35) then the expression simplifies that;

$$C_P = \sqrt{\frac{1}{2\pi.\left(\frac{1-\nu^2}{E}\right)}} \tag{1.35}$$

In AGMA 2001/D04 has a Table 2.5 of different materials C_P based on pinion material and Modulus of Elasticity. [17]

Table 2.5 - Elastic Coefficient

Elastic Coefficient C_p (Z_E), $\sqrt{\text{psi}}$ ($\sqrt{\text{MPa}}$) *Source:* AGMA 218.01

		Gear Material and Modulus of Elasticity E_G, lbf/in² (MPa)*					
Pinion Material	**Pinion Modulus of Elasticity E_p psi (MPa)***	**Steel 30×10^6 (2×10^5)**	**Malleable Iron 25×10^6 (1.7×10^5)**	**Nodular Iron 24×10^6 (1.7×10^5)**	**Cast Iron 22×10^6 (1.5×10^5)**	**Aluminum Bronze 17.5×10^6 (1.2×10^5)**	**Tin Bronze 16×10^6 (1.1×10^5)**
Steel	30×10^6 (2×10^5)	2300 (191)	2180 (181)	2160 (179)	2100 (174)	1950 (162)	1900 (158)
Malleable iron	25×10^6 (1.7×10^5)	2180 (181)	2090 (174)	2070 (172)	2020 (168)	1900 (158)	1850 (154)
Nodular iron	24×10^6 (1.7×10^5)	2160 (179)	2070 (172)	2050 (170)	2000 (166)	1880 (156)	1830 (152)
Cast iron	22×10^6 (1.5×10^5)	2100 (174)	2020 (168)	2000 (166)	1960 (163)	1850 (154)	1800 (149)
Aluminum bronze	17.5×10^6 (1.2×10^5)	1950 (162)	1900 (158)	1880 (156)	1850 (154)	1750 (145)	1700 (141)
Tin bronze	16×10^6 (1.1×10^5)	1900 (158)	1850 (154)	1830 (152)	1800 (149)	1700 (141)	1650 (137)

Surface Condition Factor

The surface condition factor Cf is used only for in pitting resistance equation. According to AGMA, indicating a procedure to understand and add the formula to Surface Condition has no standard at all. Instead, if any detrimental effect are not valid then the value can be taken as unity.

Moreover, the surface condition factor depends on related bullets.

- Residual Stress
- Plasticity region which is hardening.
- Manufacturing the surface quality and method such as cutting, lapping, shotpening, shaving, hobbing etc.

Load Distribution Factor

The load distribution factor which has written as (1.36) is combination of several factor which includes surface manufacturing factors to summation of other factors.

Especially, that factor focused on line of contact area due to nonuniformity effect on stress equations. In ideal world, base diameter of the pinion and opponent gear midspan must be aligned without any potential slope when additional load is applied on. Unfortunately, owing to various effects it is not doable in real world. Therefore, the following formulation and principle can be used and expressed by AGMA. In order to use the load distribution factor, some conditions and restrictions must be visible.

- Face width divided by pitch diameter ratio $\frac{b}{d} \leq 2$
- Gear elements mounting as assembly between one or several bearings.
- Face width up to 1.000mm
- Special contact point or line according to full width action on the narrowest point.

$$K_m = C_{mf} = 1 + C_{mc}(C_{pf}.C_{pm} + C_{ma}.C_e) \quad (1.36)$$

- C_{mf}: Face Load Distribution Factor
- C_{mc}: Surface crowning or uncrowning condition
- C_{pf}: Face width evaluation cases including pitch diameter (1.37)
- C_{pm}: Straddle-mounting cases according to uneveness.
- C_{ma}: Gear manufacturing empirical constants (1.38) and can be found from Figure 2.21
- C_e: Either mounting gear pairs as in or with assembly or other different conditions.

So the conditions and constants in Figure 2.19 are being held in following way.

- $C_{mc} = \begin{cases} 1, & for\ uncrowned\ teeth \\ 0.8, & for\ crowned\ teeth \end{cases}$

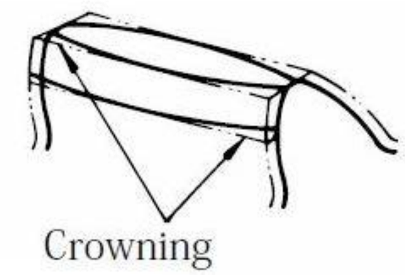


Figure 2.19 - Crowned Teeth Feature

$$C_{pf} = \begin{cases} \frac{b}{10.d} - 0.025, & b \leq 25.4mm \\ \frac{b}{10.d} - 0.0375 + 0.0125.b, & 25.4mm < b \leq 431.8mm \\ \frac{b}{10.d} - 0.1109 + 0.0207.b - 0.000228.b^2, & 431.8mm < b \leq 1016mm \end{cases} \quad (1.37)$$

But if F/(10d) < 0.05, F/(10d) = 0.05 is used which has been related with Figure 2.20.

- $C_{pm} = \begin{cases} 1, \; for\ straddle - mounted\ pinion\ with\ \frac{S_1}{S} < 0.175 \\ 1.1, \; for\ straddle - mounted\ pinion\ with\ \frac{S_1}{S} \geq 0.175 \end{cases}$

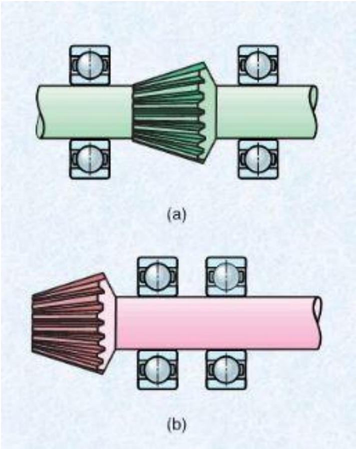


Figure 2.20 - a) straddle b) Overhang mounting (non-straddle)

Unknowns are given in Table 2.6:

$$C_{ma} = A + B.b + C.b^2 \tag{1.38}$$

Table 2.6 - Gear manufacturing empirical constants

Condition	A	B	C
Open gearing	0.247	0.0167	$-0.765(10^{-4})$
Commercial, enclosed units	0.127	0.0158	$-0.930(10^{-4})$
Precision, enclosed units	0.0675	0.0128	$-0.926(10^{-4})$
Extraprecision enclosed gear units	0.00360	0.0102	$-0.822(10^{-4})$

$$C_e = \begin{cases} 0{,}8, & for\ gearing\ adjusted\ at\ assembly \\ 1, & for\ all\ other\ conditions \end{cases}$$

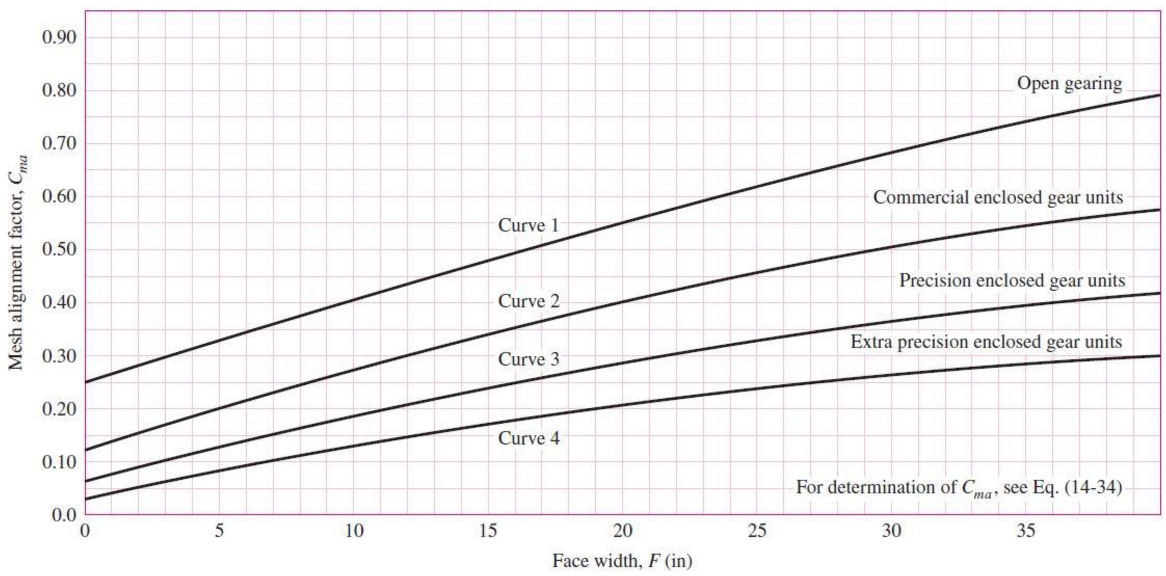


Figure 2.21 - Mesh Alignment Factor by Face Width

According to AGMA Standard, velocity factor which is one of the most important factor must be found if sufficient application environment has been earned. Briefly, velocity factor are used to account for inaccurant conditions especially for the manufacturing. The factor have other name called Transmission factor. There are some effects regarding to transmission error.

- Inaccurate tooth manufacturing as profile which leads to have runout, spacing, crowning etc.
- Permanent deformation effects such as wear, pitting etc.
- Unwanted vibration sequence on the tooth whilst running
- Magnitude change during movement
- Rough tooth surface's effect on friction value.
- While rotating unbalanced parts may harm other parts or components.

The factor has been identified from 1,00 to 2,25 as range due to load stage of mentioned application from uniform shock to heavy shock. On the other hand, gives related example definitions and launched same application ranges start from 1,00 to 2,75. [19,24]

$$K_v = \left(\frac{A + \sqrt{200.V}}{A}\right)^B \tag{1.39}$$

Which has parameters like A and B are respectively.

$$A=50+56.(1-B)$$
$$B=0,25(12-Q_v)2/3 \tag{1.40}$$

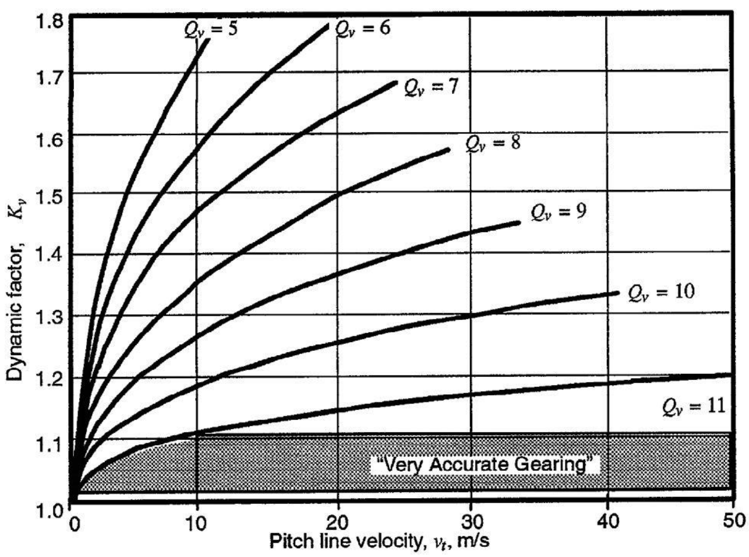


Figure 2.22 - Dynamic Factor According to Pitch Line Velocity

Dynamic factor has a formulation as (1.39) and (1.40) with different parameters related to gear manufacturing tooth accuracy which has given in Figure 2.22. In that way, features of the individual teeth has to be comparatively perfect and smooth, thus this makes maintained stress has no unwanted conditions.

Size Factor

As briefly and roughly described, K_s is factor that it follows non-linearity of material properties solely due to general dimensional size.

- Tooth size
- Diameter of part

- Ratio of tooth size to diameter of part
- Face width
- Area of stress pattern
- Ratio of case depth to tooth size
- Hardenability and heat treatment

In different standard, there are some known size factor which has been using. However due to some detrimental size effect with regards to gear teeth, some specialized formulation has to be used. In such unknown cases, AGMA suggested to use size factor only for if it is greater than unity. Which means that unless the detrimental size effect is occured, use unity.

$$K_s = \frac{1}{K_b} = 1{,}192.\left(\frac{F.\sqrt{Y}}{P}\right)^{0{,}0535} \tag{1.41}$$

This formulation in (1.41) is linked with Lewis formulation and its bending structure. Once it has been discussed with the manufacturer that teeth bending structure will be more different than the usual ones, then it won't be 1 at all. Diversely, It has been suggested that almost all of the manufacturer, K_s can be taken as 1.

<u>Bending Stress Geometry Factor (Y_J)</u>

AGMA suggest to use Y_J in (1.42) which has owned modified numerical value of the Lewis form factor Y. The formulation related by as fatigue stress concentration factor K_f and tooth load sharing factor m_n. Mentioned formulation can be used both spur and helical gears.

$$Y_J = \frac{Y.C_\psi}{K_f.m_n} \tag{1.42}$$

- C_ψ is only for helical overlap factor.

As understood from the equation, bending stress geometry factor is dimensionless factor. Again there are some features that may impressed J.

- Stress concentration. (which is as same as Lewis Equation.)
- Worst case load taking.
- Shape of the tooth.

As known from the first experimental test about photoelastic investigation. The unknown K_f is calculated. The load sharing factor is equal to face width divided by the minimum overall length of line of contact. For spur gears $m_n = 1$.

On the other hand, AGMA prepared a Figure 2.23 in terms of obtain Y_J for 20 degree pressure angle spur gears and full depth teeth. The reason is that generally, 20 degree pressure angle is widely using in the industry and manufacturer prefers to operate due to casting fixed cost.

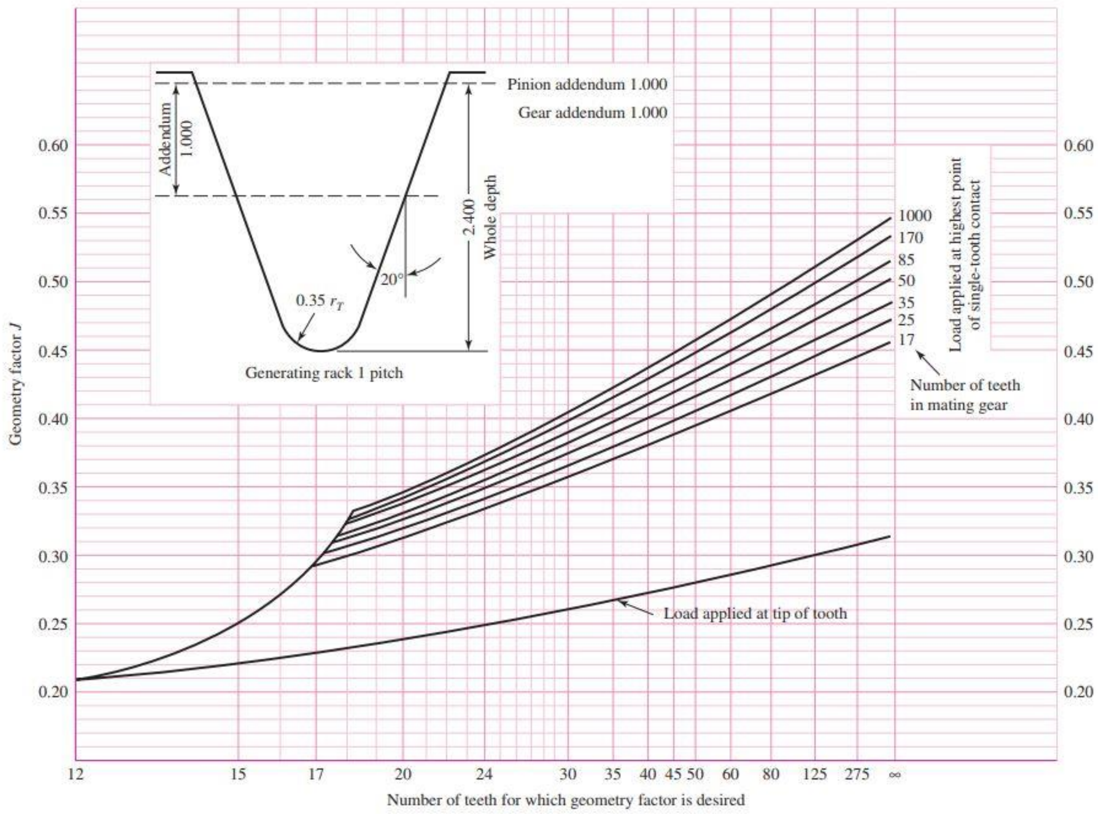


Figure 2.23 - Geometry Factor by Number of Teeth

As seen from above; while rack&pinion systems, number of teeth in mating gear can be handled as 1000 or infinite.

Actual J calculation is stated from Kuang research. [25]

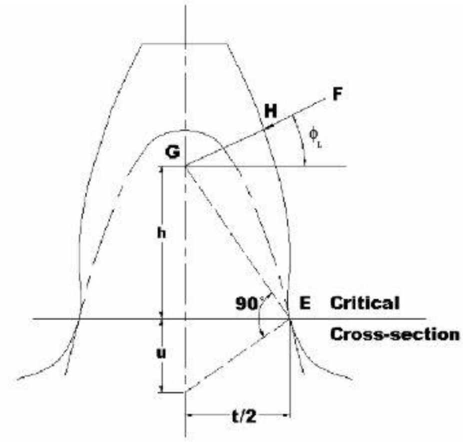


Figure 2.24 - Tangential Force Effect on Single Teeth

$$Y = \frac{1}{\frac{m.\cos\theta_L}{\cos\theta_x}.\left(\frac{1.5}{u} - \frac{\tan\theta_L}{t}\right)} \tag{1.43}$$

Where in (1.43) and (1.44) formulations; m is module, θ_L is loading angle, R_{bi} is Radius of individual base circle and b is the tooth width in Figure 2.24.

$$\theta_x = \cos^{-1}\left(\frac{R_{b1} + R_{b2}}{C}\right) \tag{1.44}$$

<u>Surface Strength Geometry Factor (Pitting Resistance) (Z_I)</u>

By AGMA Z_I is called pitting-resistance geometry factor and it is developed from the Equation in (1.45), (1.46) and (1.48) respectively.

$$\frac{1}{r_1} + \frac{1}{r_2} = \frac{2}{\sin\theta}.\left(\frac{1}{d_p} + \frac{1}{d_g}\right) \tag{1.45}$$

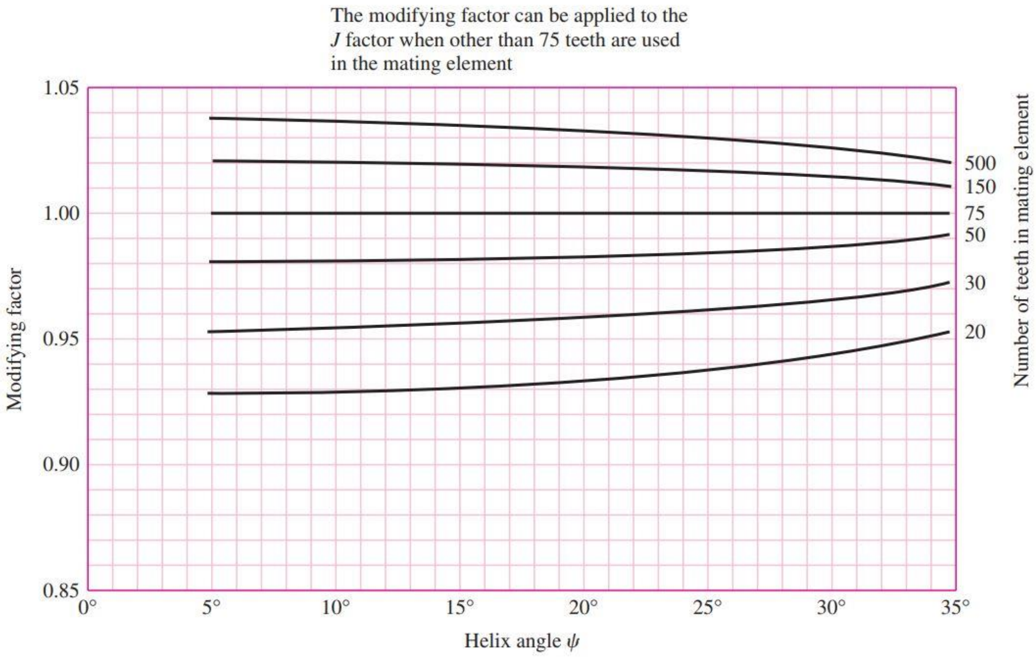


Figure 2.25 - Modifying Factor by Helix Angle

$$\frac{1}{r_1} + \frac{1}{r_2} = \frac{2}{d_p sin\theta} . \left(\frac{m_G + 1}{m_G} \right) \tag{1.46}$$

If it is substitute for the sum of reciprocals in contact stress equation then it will be as in (1.47);

$$\sigma_c = C_p . \left[\frac{K_V . F_t}{d_P . b} . \frac{1}{\frac{cos\phi_t . sin\phi_t}{2} . \frac{m_G + 1}{m_G}} \right]^{1/2} \tag{1.47}$$

Load sharing factor m_N is added in Figure 2.25, then the below equation has obtained for both spur and helix gears. To make underline $m_N = 1$ is for spur gears.

$$Z_I = \begin{cases} \dfrac{cos\phi_t . sin\phi_t}{2. m_N} . \dfrac{m_G}{m_G + 1}, & Internal\ Gears \\ \dfrac{cos\phi_t . sin\phi_t}{2. m_N} . \dfrac{m_G}{m_G - 1}, & External\ Gears \end{cases} \tag{1.48}$$

Especially for rack&pinion pairs, there is one Radius an one infinite tooth number. Diversely, $\frac{m_G}{m_G+1}$ can be counted as 1.

<u>Hardness Ratio Factor</u> (Z_W)

While pairing the gear and pinion couples, another consideration is related with heat treatment and part's hardening besides material's mechanical properties itself. Broadly, the pinion has smaller number of teeth than rack or gear. Along these lines, pinion has subjected to more teeth and contact point while rolling. If both pinion and gear has through hardening properties on them uniformly then any unwanted effect has been occured. Contrarily, if material hardening properties are not mismatched from each other, formerly a small effect can be arised. The difference example can be surface-hardened pinion coupling with through-hardened gear. The Hardness ratio factor which's written (1.49) can be written as;

$$Z_W = 1.0 + A'. (m_G - 1.0) \tag{1.49}$$

Where

$$A' = 8.98. (10^{-3}). \left(\frac{H_{BP}}{H_{BG}}\right) - 8.29(10^{-3}). 1.2 \leq \frac{H_{BP}}{H_{BG}} \leq 1.7 \tag{1.50}$$

H_{BP} and H_{BG} are Brinell Hardness of the pinion and gear. m_G is speed ratio which is covered in previous examples as in (1.50) and (1.51).

$$\frac{H_{BP}}{H_{BG}} < 1.2, \quad A' = 0 \tag{1.51}$$

$$\frac{H_{BP}}{H_{BG}} > 1.7, \quad A' = 0.00698$$

For pinion with surface-hardened has a hardening around HRC 48 (Rockwell C) or greater is to worked with through-hardened gear around 180-420 Brinell then another formulation as (1.52) would be added due to match a relationship between Rockwell and Brinell.

$$Z_W = 1.0 + B'.(450 - H_{BG}) \qquad (1.52)$$

Where B' can be found from (1.53)

$$B' = 0.00075 \exp(-0.28448. f_P) \qquad (1.53)$$

Where f_P is asserted as root-mean-square roughness R_a in μ mm in Figure 2.26 and Figure 2.27.

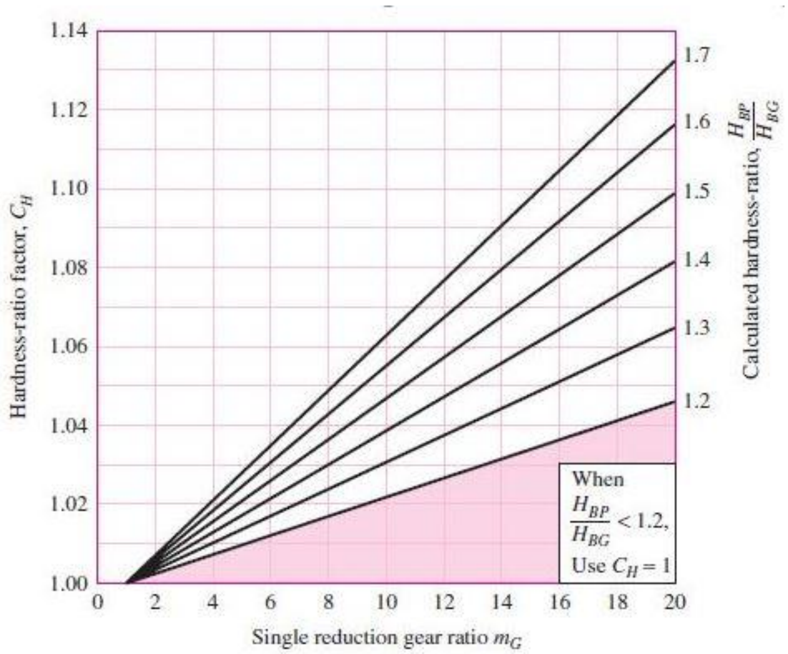


Figure 2.26 - Single Reduction Gear Ratio by Hardness-ratio Factor

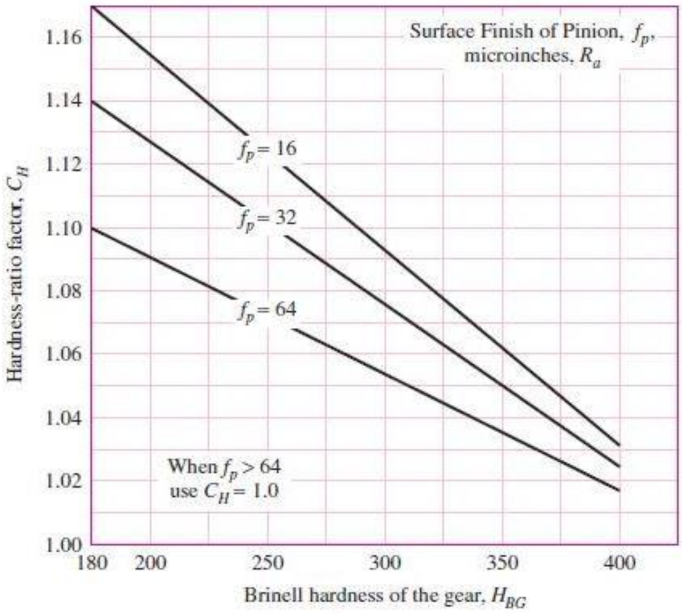


Figure 2.27 - Hardness Ratio Factor by Single Reduction Gear Ratio and Brinell Hardness

<u>Stress Cycle Factors Y_N</u>

Until now, all of the tables and figures are for general standard life of 10^7 cycles. Stress cycle factor are being used for other than 10^7 cycles. Due to usage of different properties of factor seems to find more accurate calculation values by testing procedures. What if different life cycle estimation would be put on by designer? At that time, mating gears can be used for different factors such as $(Y_N)_P$ and $(Y_N)_G$ in Figure 2.28.

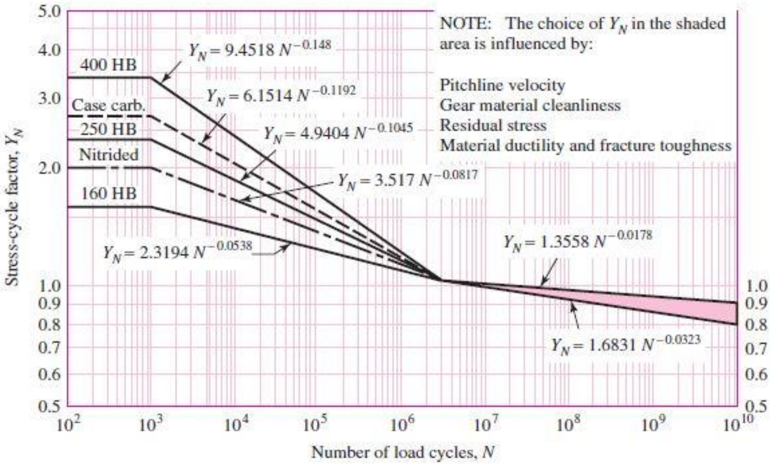


Figure 2.28 - Stress Cycle Factor By Number of Load Cycles

Repeatedly applied bending strength stress-cycle factor Y_N in Figure 2.29

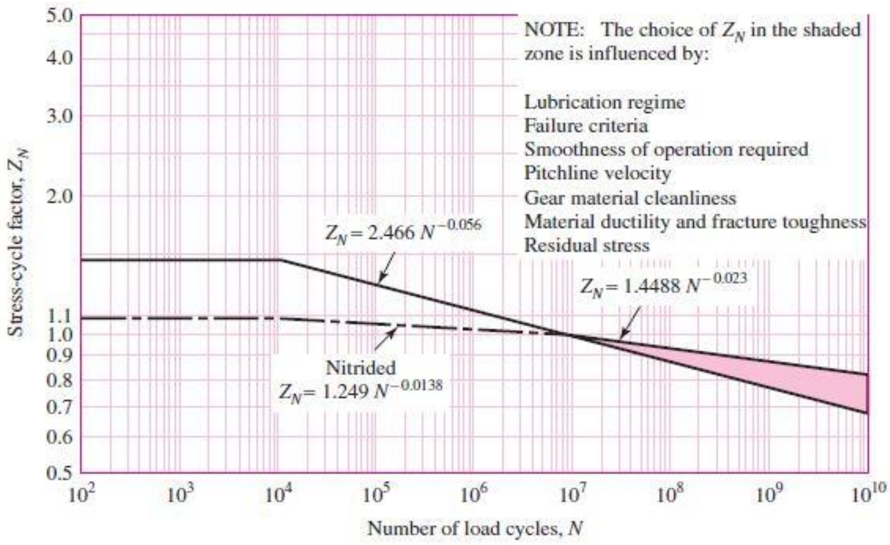


Figure 2.29 - Stress Cycle Factor By Number of Load Cycles

<u>Reliability Factor K_R</u>

By US Navy, for some using gear pairs they made a factor which based on reliability around 99 percent in Table 2.7 then they put into an equation as multiplier. AGMA accepted that approach due to achieve robust design purpose. In Reliability factor, load is not object in requirements.

Table 2.7 - Reliability Factor

Reliability	K_R (Y_Z)
0.9999	1.50
0.999	1.25
0.99	1.00
0.90	0.85
0.50	0.70

Roughly Reliability factor depends on statictical operation management regarding material fatigue characteristics from (1.54). Once usage operations are selected, then usage scenario of the system has to be considered from reliability point of view.

$$K_R = \begin{cases} 0.658 - 0.0759\ln(1-R), & 0.5 < R < 0.99 \\ 0.50 - 0.109\ln(1-R), & 0.99 \leq R \leq 0.9999 \end{cases} \tag{1.54}$$

As shown in above, K_R scale is extremely great. Unless, the table would be used, then formulation becomes sense in terms of finding the output properly. In order to use that nonlinearity must be considered. From analytical approach, least-square regression is being used by AGMA.

Temperature Factor K_T

While planned to use gear in particular design, temperature based knowledge is definetely the most easiest part in estimation process. It can be thought as digitally. If temperature environment is up to 120°C then K_T=1 for oil or gear-blank designs.

Only heat exchanger or finned designs may be accelarete the temperature to unwanted values. In that way, interpolation can be used as a starting point from 1, it will be greater than unity.

Rim Thickness Factor K_B

For specific round shaped gears, rim thickness is significant. The reason is that while gathering torque and consolidate to single point or line, bending fatigue failure region might be intesify throughout the pad of the gear which means thickness of gear might have stress concentration rather than tooth fillet. In that manner, adding K_B related to (1.56) is suggested by AGMA in Figure 2.30. If processing with thin thickness gear, the factor must be added to the formulation.

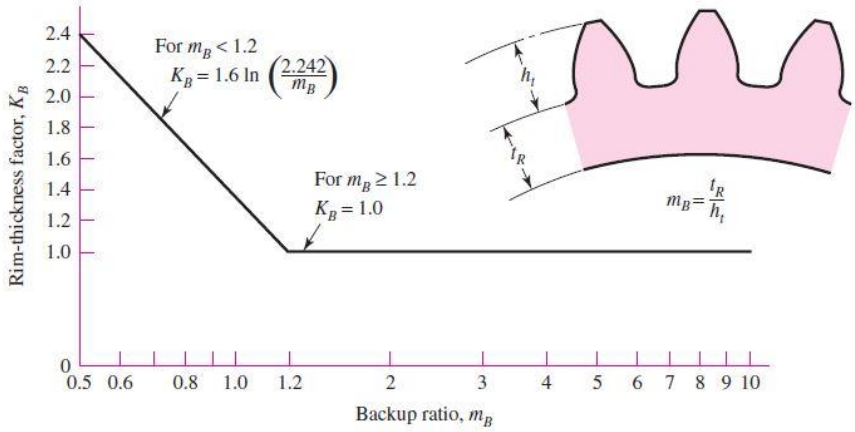


Figure 2.30 - Rim-Thickness Factor by Backup Ratio

Backup ratio (1.55) plays important role in terms of evaluating the output source as value.

$$m_B = \frac{t_R}{h_t} \tag{1.55}$$

t_R: Rim thickness on tooth

h_t: tooth height

$$K_B = \begin{cases} 1.6\ln\dfrac{2.242}{m_B}, & m_B < 1.2 \\ 1, & m_B \geq 1.2 \end{cases} \tag{1.56}$$

<u>Safety Factor S_FAnd S_H</u>

According to standards of and is relied on safety factor regarding bending fatigue failure S_F is in (1.57) and pitting failure S_H is in (1.58) [17, 18]

- Safety Factor of Bending Fatigue Failure:

$$S_F = \frac{S_F.Y_N/(K_T K_R)}{\sigma} = \frac{Fully\ Corrected\ Bending\ Strength}{Bending\ Stress} \tag{1.57}$$

- Safety Factor of Pitting Resistance Failure:

$$S_H = \frac{S_C . Z_N . Z_W / (K_T K_R)}{\sigma_C} = \frac{Fully\ Corrected\ Contact\ Strength}{Contact\ Stress} \qquad (1.58)$$

In AGMA Standard 2001-D04 [17] has been indicated Safety Factor S_F and S_H in order to prevent bending fatigue failure and contact fatigue failure.

To understand the evaluating calculated stress from FEA program with calculated stress from analytical solutions, there are some regulation for both bending stress and pitting resistance stress' results.

For S_F, while comparing with S_H even either for linear tooth array or helical array; ${S_H}^2$ must be evaluated and ${S_H}^3$ must be evaluated for only crowned teeth pattern. The reason for trying to compare the results is that establish the loss of threat prevention or obtain the circumstances as per cycle of time. That mentioned evaluation is not must to implement as calculation with real life or will not be unavoidable end. This formulation or evaluation only shows that it may appear during the performance and if any potential damage or fatigue may arrive, it might be a cause of it.

For using the same material type system designs, there is only one result fort he system; one is for S_F and S_H with bending strength and contact strength (pitting resistance.) With that usage, evaluation the results go very easily for that system.

Those calculations are strength divided by stress definition. Therefore, triggering effect is to be Transmitted (tangential) Load. However about surface load direction is nonlinear at that time.

2.6. Contact Ratio

Contact Ratio is one of the main step whilst the consideration of the behave of teeth. It also defines the working conditions and it may maintain the noise, strength, rotation, gear oscillation and etc. Standard gears contact ratio is varying from 1.2 to 1.8. In order to gain higher contact ratio above 1.8, the designer must change the standard manufacturing instruments and change them with modified ones. Unfortunately, it affects the cost efficiency in bad way even though higher contact ratio can reduce the noise. In general usage, the maximum gear tooth contact is the best choice solely with minimum deflection yet standart gear contact ratio between 1.2 and 1.8. [19, 20]

In order to find contact ratio, the formula of the spur gear with rack is more different than two pinions in Figure 2.31. The essential point of finding the contact ratio is that length of the general contact is to divided by base pitch. If the contact ratio is equal to 2 or more, the identifiation of the contact ratio becomes high contact ratio. In that manner, above 2 could have keep in touch with two teeth. Increasing the contact ratio may reduce the noise of the gear pair. In the contrary from the straight gear, gears with spiral teeth has better contact ratio. However, it must be evaluated by the gear engineer whether the increased cost of building helical gear is worthy or not. The equation of the contact ratio is varied from the gear pair types. In order to find gear to gear spur pair the equation is written as 1.59. [21]

$$C.R. = \varepsilon_a = \frac{\sqrt{\left(\frac{d_{a1}}{2}\right)^2 - \left(\frac{d_{b1}}{2}\right)^2} + \sqrt{\left(\frac{d_{a2}}{2}\right)^2 - \left(\frac{d_{b2}}{2}\right)^2} - a_x sin a_w}{\pi . m. cos\alpha} \qquad (1.59)$$

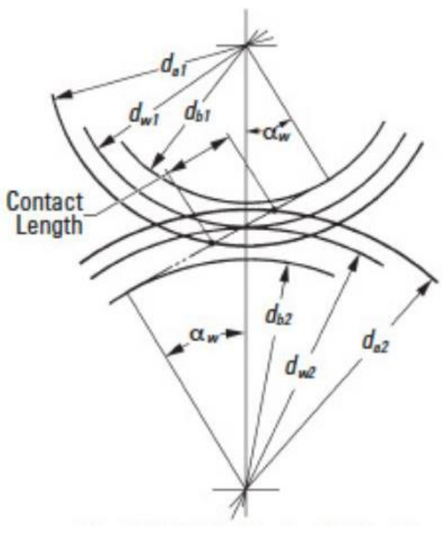


Figure 2.31 - Pinion Pair Circles

- $d_{a1} = d_{b1}$: Outer Diameter of First Pinion
- $d_{a2} = d_{b2}$: Outer Diameter of Second Pinion
- a_x: Pressure Angle
- m: Module

Unfortunately, in rack and pinion there is no second radius. Rack has defined as infinite teeth. So, it gets another formulation which is written in (1.60). [22]

$$C.R. = \varepsilon_a = \frac{\sqrt{\left(\frac{d_{a1}}{2}\right)^2 - \left(\frac{d_{b1}}{2}\right)^2} + \frac{h_{a2} - x_{1m}}{sina} - \frac{d_1}{2} sina}{\pi . m . cos\alpha} \tag{1.60}$$

- $d_{a1} = d_{b1}$: Outer Diameter of Pinion
- a_x: Pressure Angle
- m: Module
- h_{a2}: Addendum of Rack
- x_1: Rack Shift Coefficient

For Contact ratio, the more it closes to 2, the more cost effectiveness has been achieved. On the other hand, changing the pressure angle and adding more tooth depth (addendum) can be required and suggested for better contact ratio however those two items requires special tool design tools and cutters. For the companies which has not worked on specifically on gear design then increasing cost with stated suggestions may be negligible. Therefore, finding the more closed contact ratio number to two is most desired output.

In modern world 3-Dimensional design plays key role for finding almost everything on machine elements and assemblies. With the help of those tools, the length of the general contact can be found as Figure 2.32.

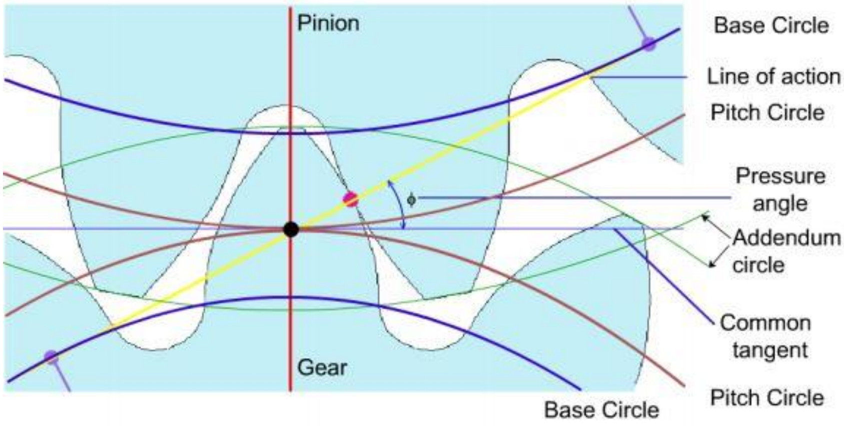


Figure 2.32 - Length of Contact Line

2.7. Suspension Elements

The purpose of the vehicle suspension is to increase the dynamic friction between the tires and the road surface in order to compansate stability of steering, good handling and the comfort of the people. In this article, we will explore the operation of car suspensions, their evolution over the years and their design for the future. [28]

If a road feature is flat enough and without defects, no suspension would be necessary. However the roads are unfortunately not flat. Even freshly tarmac has some drastic imperfections that that may create unexpected forces on tire of the vehicle. According to Newton's laws, all forces must be magnitude and direction as coordinates. A shock may directly affect impress on wheels owing to road surface. The size of the wheel, the rim size, weight of the vehicle or environmental effects depends on hiting a huge bump or a small dot. In any case, the wheel of the car undergoes a vertical acceleration when it exceeds an imperfection.

2.7.1.Springs

From outside as non-technical approach, assumption of spring role is only valid for pre-indication of vehicle height. Apart from cosmetic view, spring has huge ability to compansate the roughness effect surface from the road. Moreover, it can help to preserve any additional weight on vehicles such as baggages, passengers etc. There are three main spring types used throughout the years.

2.7.1.1. Coil Springs

Normal coil springs are shown in Figure 2.33 are widely used in so many application for push-pull mechanism like inside pen to heavy armoured vehicles. The logic behind is that absorbation. The name is defined as spiral geometry around aligned axis. Due to compressive and tension behaviour, these kind of springs serve to absorb wheel's movement. Usage of coil springs are very common in modern vehicles.

Figure 2.33 - Coil Spring

2.7.1.2. Leaf Spring

For only motion absorber, leaf springs which are shown in Figure 2.34 are conceivably the oldest spring system. That semi-elliptic system is occured from form of a arc–shaped plates. The end of the arcs are mostly used by mounting holes. The general shape of the leaf spring components illustrated as triangle shape. The plates are deployed from smaller size which are stacked bottom to bigger size. Those several layers are applicable in order to demonstrate movement of damper. The other mounting point of leaf spring is in the middle where bolted to vehicle axle. Due to its mounting simplicity, it is very common in assemblies. In todays world, leaf springs are used in heavy duty vehicles.

Figure 2.34 - Leaf Spring

2.7.1.3. Torsion Bars

Torsion bar definition which is shown in Figure 2.35 has not influence as spring however the originated name is torsion spring bar. So the mechanical meaning of the spring is granted from torsional stress in the field of a solid mechanics. Shear stress is owing to force which is taken from rough road condition. Once the force gets reduced, the negative torsional shear stress tend to behave reclaim movement. Torsional spring bars generally used by main battle tanks, infantry fighting vehicles and other combat vehicles. The application with tracked vehicles are satisfying and open a space underneath the hull. Accordingly, engineers are able to put more survivability items in that way.

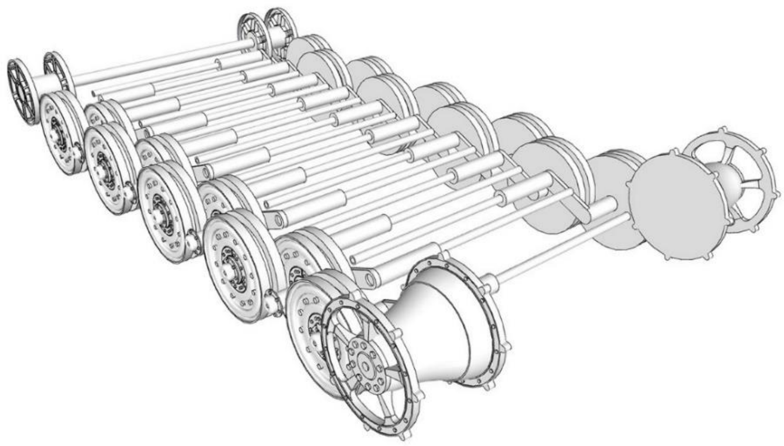


Figure 2.35 - Torsion Bar on Tracked Vehicle

2.7.1.4. Air Springs

By air pump or compressor send the air into textile rubber-case. The air filled around the pocket and the volume of it increased The design of the air spring which is shown in Figure 2.36 is made by longitutional so while the radius of the spring does not growing, only height is developing. In other words, the more air pressure increases, height of the suspension (ride height) escalates as well. With a simple entry of pneumatic valve and its sensor is very sufficient to use it. In comparison of technically simplicity of the design, the compressor and airbag failure is very common problem. So that making a good design with the calculations will be provide efficient output in terms of managing demands.

Figure 2.36 - Air Spring with Section View

2.8. Gear Elements

In the new design, Lifting suspension system by rack and pinion system in Figure 2.37 instead of lifting with steel rope directly from suspension's A-arms. The design elements are;

- Two rack
- Two pinion
- Friction Washers

The rack and pinion design was used due to mentioned advantages.

- Cheapness
- Stiff and Robust
- Most compact way to conversion between rotation to linear motion
- Easy control
- Motion feel

The critical and biggest disadventage is friction effect. But if the design was done in good way, friction and its loss will be diminishing eventually.

In that study; as pinion, spur gear is used. Spur gear is a cylindrical shaped gear that the most important distinctive property is the teeth it owned is parallel to the axis of cylinder. Among other types, it is the most common used spur gear in the world in order to transfer power and its manufacturing simplicity. The charcteristics of the spur gears are;

- Easy to manufacturing
- Only deal with tangential force, not any radial force generation at all.
- Easy to generate what it demands as output in terms of straightforward transmission.
- Due to equally distributed tooth make constant velocity.
- According to other types, due to its shape spur gears are reliable and compact.

As longutadional movement, spur rack is being used. Rack is linear shaped design gear which again same with spur gear parallel to the height of the rack. It has infinite radius. The tooth shaped is to be straight-lined.

- Easy to manufacturing
- While using, meshing with other gears are extremely easy.
- Translation from rotational to linear motion, the most commonly used part in the world.
- Without radial shape, it is suitable to mount with gears.

In gear design, load, width and module has required for parametric calculations. Almost all of the design engineer choose module (m) and calculate the face width from the equation from $3p<w<5p$. In order to find best iterative estimation there are some rules and regulations to be followed and those are provided from below;

- Estimation of average nodule while thinking about inputs.
- Choosing the best materials for mating gears.
- Choosing the best possible combination of heat treatment for surface hardening.
- Indicating the load and its movement for accelaration
- Determining the number of teeth.
- Determining the width of the gear
- Indicating the fatigue strentgh for selected materials.

As known as gear racks are exist to alter rotating movement into linear motion. A gear rack has straight lined teeth only in one surface of a square or rectengular shape operates with a pinion, which is a small cylindrical gear has same mesh with the gear rack. Generally, gear rack and pinion are collectively called “rack and pinion”.

In order to use these type of gears, there are many variations to use. Especially mechanical element wise, that transfer rotaryinto linear motion. Those type of motion exclusively operated for lifting mechanisms, vertical/horizontal movement, stopping the valid forces and ositioning the mechanisms.

Figure 2.37 - Rack and Pinion Couple

The natural characteristics of rack&pinion system service for steering systems. Steering shafts can be touched by a steering rack to transmit rotary motion (steering turn) convert into a linear motion through the rack. And it affects on wheel with the eccantrcity as mounted on wheel hub.

3. DESIGNING A SUSPENSION LIFTING-UP MECHANISM AND ANALYSIS

The amphibious vehicles have several advantages when goods of operations and country security subjects is thought. The usage of suspension systems are inevitably important for mobility, survivability and assault. In case of design inaccuracy, system would affect crew's life and process of assault.

Achieving the successful lifting the mechanism up would be a benefit for operation. That design can make better system balance while transfering heavy armoured vehicles and crews. So that, any potential or life loss would have been prevented from the beginning by that proposed desgin.

3.1. Definition of Problem

Problem can be described as for amphibious vehicle, there is another effects on vehicle which contains water and its physical effects. Vehicle has been thought in some kind of other world which density is different than reality. Therefore, while diving into water or sea or swimming forth or back will definetely face with different circumstances such as velocity change, waves, obstructions inside the water/sea. The most critical effect which may be underestimate is that currency effect while wheels get down in swimming operation.

While considering the vehicle is carrying crew, commander or any other vehicles. It may affect on vehicle and it can tremble without any reason. In order to prevent that effect, suspension systems must be lifted up by using a mechansim. Lifting such huge amount of volume can be dangerous and must be thought carefully.

Previous proposed designs contain the system which are being used for linear sliding rods and autoblockaged pairs components such as worm gear with pinion.

In that proposed design, the system are being tested by both gear standards and analytical calculations which may be available and suitable to use essentially for steering mechanism. There is not a single usage for lifting up or down for solely suspension motion. Such a heavy weight of wheel hubs and wheel have different motion curve than other knear system. With the linkage usage between rack&pinion and arm has cut the inaccurate levelling of motion.

3.2. Problem Datas

The type of vehicle is heavy and amphibious. Beside being amphibious, the vehicle is fortification for army and its mission is exploring the area as pioneer and transfer the heavy armoured vehicles by coast to coast. The place where it may not fording ashore, then several number of vehicle make a bridge in order to make a safe surrounding for transfer.

Due to the sea/water environmetnal effects, the mode shape of the vehicle may vary and tremble unevenly. It may harm the vehicle on it.

3.3. Problem Assumption and Restrictions

3.3.1. Assumptions

Assumptions can be found as Table 3.1.

Table 3.1 - Vehicle Properties for Proposed Design

Weight	36.000 kg
Crew	3
Length	13 m
Width	3.5 m
Height	4.1 m
Number of Axles	4
Suspension	Double Wishbone, Independent, Air Suspension
Max. Road Speed	50 km/h
Swimming (Loaded)	10 km/h (with 2 ea Pump Jets)

3.3.2. Restrictions

While design consederation in vehicle, some restrictions have been revealed. Due to swimming operation, the weight of the vehicle is extremely important. As well as weight, Center of gravity distribution is important.

To deploy adequate suspension components into necessary area, the chosen parts must be inserted properly in order not to spend most of the area. Above the rim and shock absorber there is very small area. For that demonstration the mounted dimension of rack&pinion gears has limited area to be drawn. At the result of summation, the gear diameter and the rack height must be smaller than 200 mm. Especially,pinion diameter restrictions came from that dimension, then module has been found subsequently.

With modern manufacturing machines, gear design's dimensional accuracy and cost are getting lower day by day. With the help of tolerance developing of the machines itself and metallurgical activity research gear design manufacturing with necessary demands are very easy. In that way, neither serial production nor costum design the cost are not that much for the proposed design. Apart from rack and pinion, other parts can be thought as fixed design component outputs so we can assume these as fixed cost and non changeable as well.

Prior motion starter in torquemotor in that proposed design. Without any change of currently used torqmotor will be both feasable and profitable for company wise. The torque of the torqmotor is 10.000 Nm. So it is very adequate for lifting up the system effectivelly.

In the first trial of the system in previous years, the test contrivance system had been testing for several revolution. In the end between 10 to 15 rpm had been selected and that value had been declared to the manufacturer of Hydrolic Motor supplier. The velocity has come from that demand and tests.

3.4. Design Implementation Steps

The most important part of the system is rack, pinion and linkage rod of the system. In order to validate necessary dimension, the restriction known circumstances has been considered one by one.

For velocity, revolution restriction was guided. Then minimum torque has been come up to equation.

The weight of the suspension systems part by part has been calculated and its Center of Gravity was taken from CATIA V6. The Newton law was used for generally weight for load on tooth. In order to find load, Accelaration from velocity and weight with gravitational accelaration was added to the equation.

3.4.1. Gear Module Selection

While considering integration of assemblies, most critical definitions are volume measurment and proper location. With stated restrictions, volume and its fitting as 110mm to 125mm is obligation in that design. Locating a cylindrical part is essential integration and the following mechanism parts are significant as well. So after evaluating the necessary diameter of pinion, the module selection is secondary critical parameter. Because with the help of outer diameter of pinion and module, other unknowns can easily be found. In that way, the rough integration and location of whole assembly may be put individually. Module measurement gauge was shown in Figure 3.1

Figure 3.1 - Module Measurement Gauge

In our system six different alternative module selection is drawn and compared according to shape, number of teeth, stress and contact ratio. The selection criteria must be overseen as overall.

- **Module: 2**

Number of Teeth: 57

Pitch Diameter: 114 mm

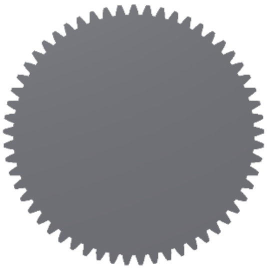


Figure 3.2 – Module 2 Design

- **Module: 3**

Number of Teeth: 38

Pitch Diameter: 114 mm

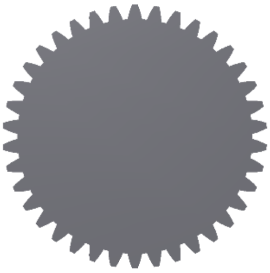


Figure 3.3 - Module 3 Design

- **Module: 4**

Number of Teeth: 29

Pitch Diameter: 116 mm

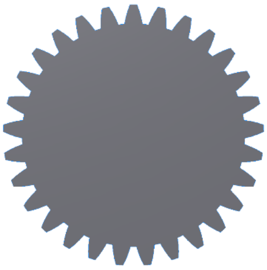


Figure 3.4 - Module 4 Design

- **Module:5**

Number of Teeth: 57

Pitch Diameter: 114 mm

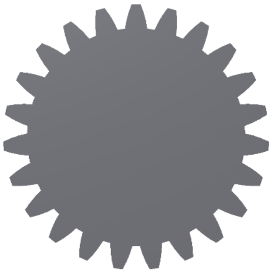


Figure 3.5 - Module 5 Design

- **Module: 8**

Number of Teeth: 23

Pitch Diameter: 115 mm

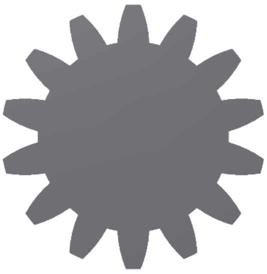


Figure 3.6 - Module 8 Design

- **Module: 10**

Number of Teeth: 14

Pitch Diameter: 112 mm

Figure 3.7 - Module 10 Design

There are some general acceptance about gear design. First of all, less than 25 number of teeth is not suggested for smooth movement.

The width of the teeth is proposed that between 6.m and 10.m. In that way, when the load's once applied the characteristics of the surface can be objected and met properly.

The stress evaluation and contact ratio are another selection procedures for gear. Those unknowns can be changed by module.

3.4.2. Gear Design

3.4.2.1. Pinion Design

Firstly, finding the most valuable parameters such as module and outer diameter has been selected. Once it has been done, the following step is to come with inner diameter, outer diameter, base circle diameter and pitch diameter.

In order to sketch pinion easily, we can use the force of the most advanced CAD program CATIA V6.

1. Generating Demanded Base Circle in Figure 3.8 for module 3 as example by the help of eight angled axis lines. The more axis lines are created, the more accuracy of gear arc has been implemented.

$$Outer\ Radius = R_O = R_P + m = 60mm \tag{1.61}$$

$$Pitch\ Radius = R_P = \frac{m.z}{2} = 57mm \tag{1.62}$$

$$Base\ Radius = R_b = R_P.\cos(\alpha) = R_P.\cos(20) = 53{,}56mm \tag{1.63}$$

$$Root\ Radius = R_r = R_P - 1{,}25.m = 53{,}25mm \tag{1.64}$$

$$Root\ Radius = m.(0{,}3) = 0{,}9mm \tag{1.65}$$

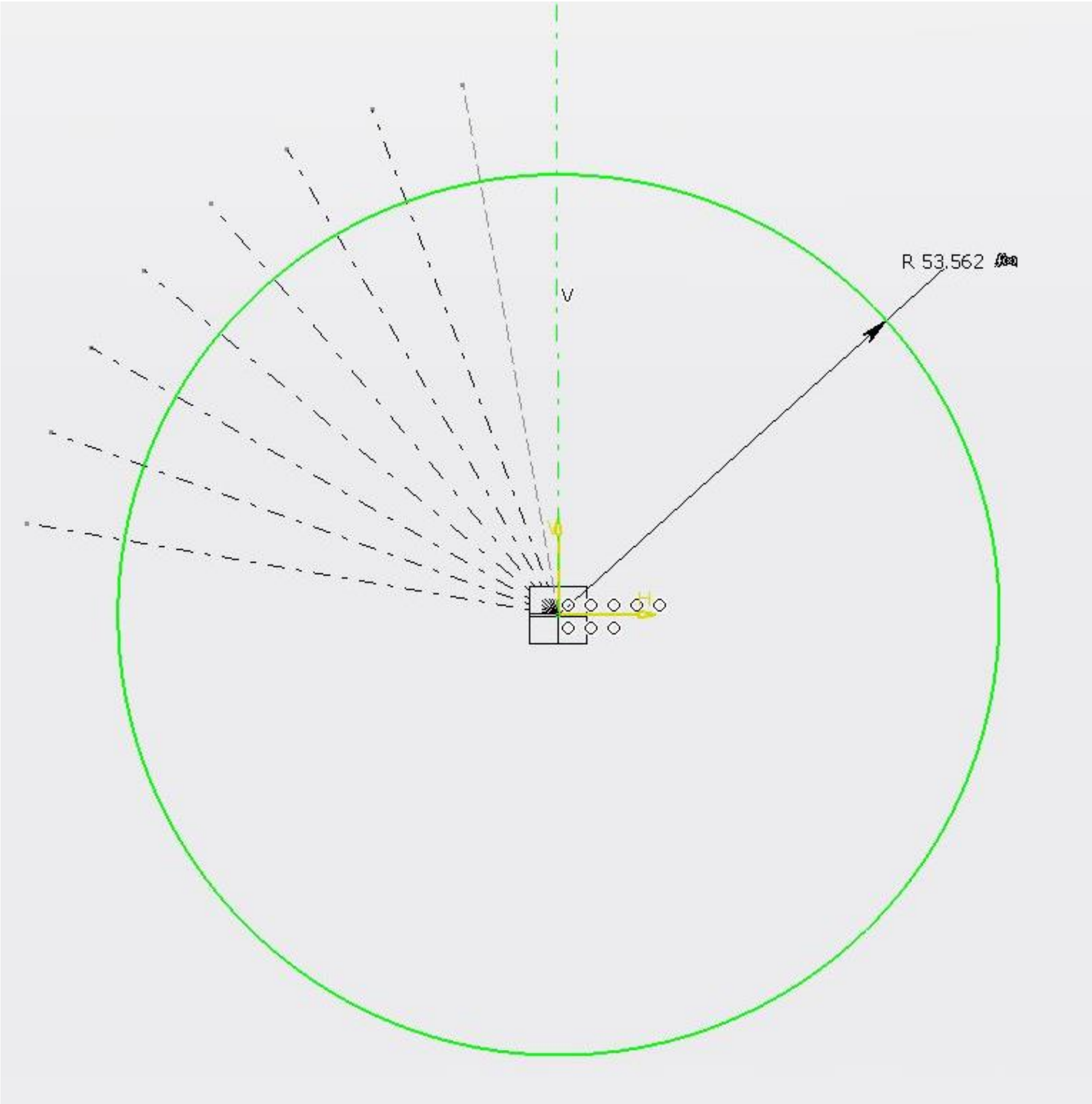


Figure 3.8 – Base Circle with Eight Involute Creation Points

All of the diameters has been found by following formulation.

2. All points have been bonded with axis and those must be in function of the first angle which is explained in Figure 3.9. Those kind of approach will help make the involute curve flexible for any change if needed.

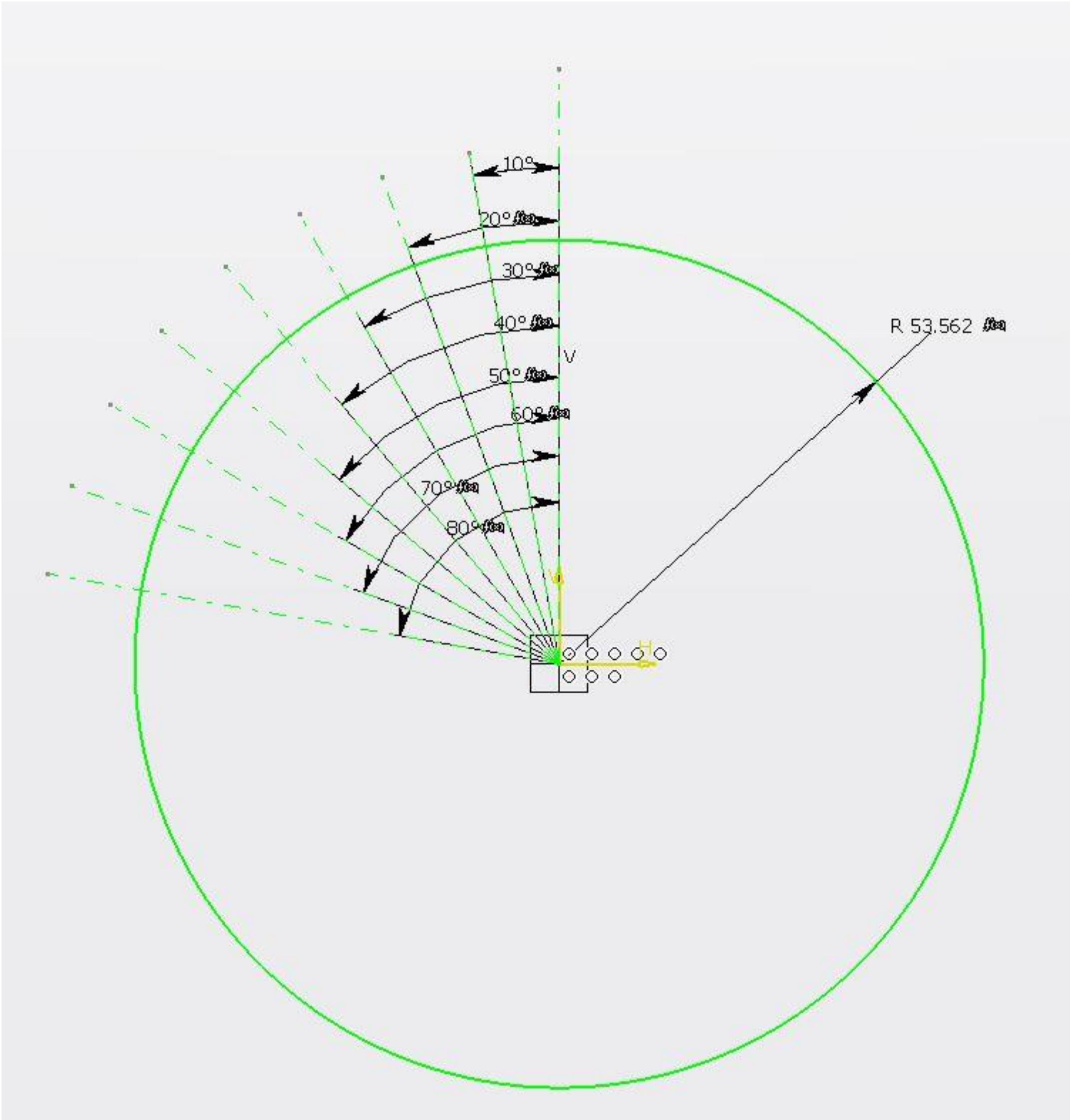


Figure 3.9 - Length Offset of Pinion

$$t = \pi . \mathrm{m} \tag{1.66}$$

$$t = \pi . 3 = 9.42$$

Length offset: $A = \frac{t}{4} = \frac{9.42}{4} = 2.35mm$

3. From the drawn axis of center in Figure 3.10 must be limited with perpendicular and tangency of the stated lines one by one. The important part is that the length of the line must be equal to the arc between the two lines forming the same angle which is formulated in Figure 3.11.

$$Length = \frac{R_b.(\gamma_i).\pi}{180deg} \tag{1.67}$$

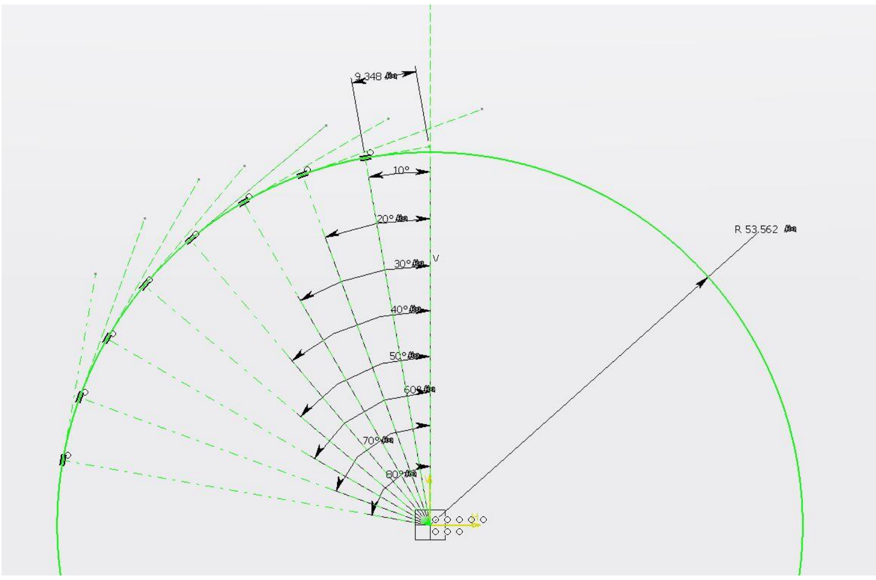


Figure 3.10 - Inserting Pressure Angle

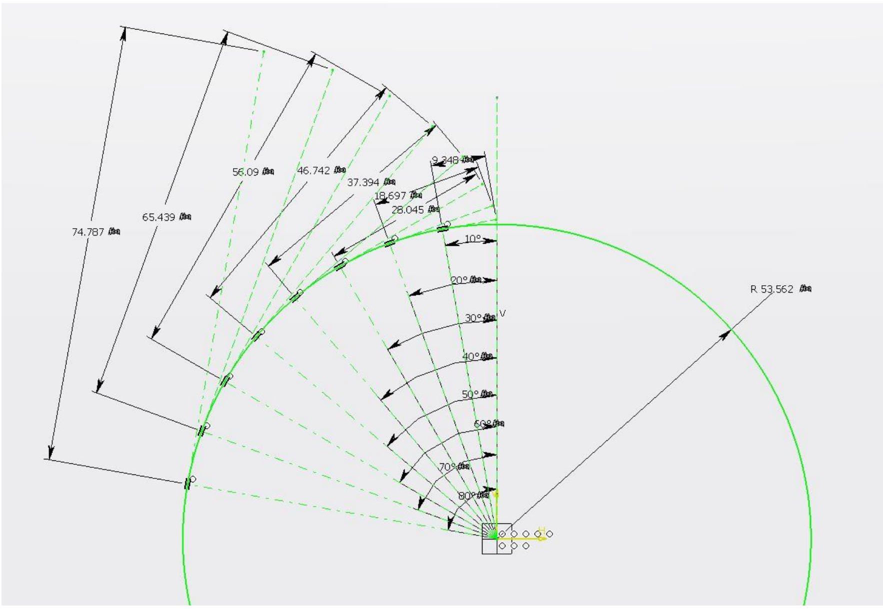


Figure 3.11 - Angle Between Teeth Arc and Axis

4. All the point from arc creation is to joined with spline (Figure 3.12) command in order to make involute shape.

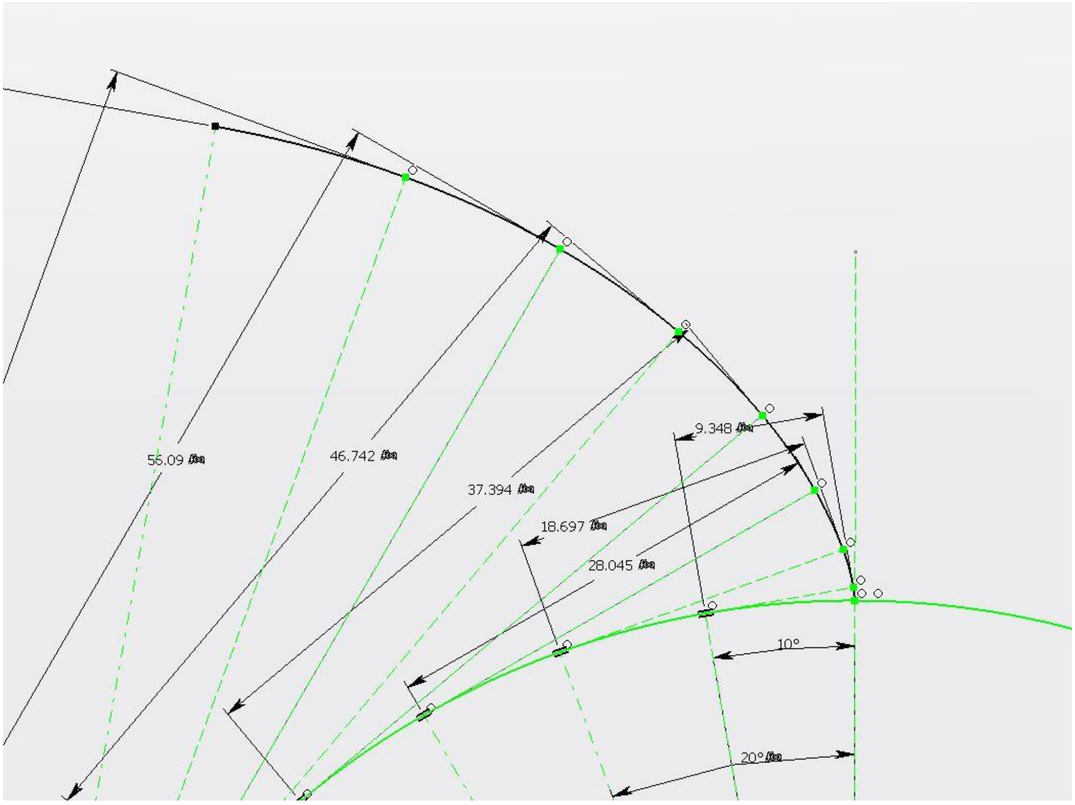


Figure 3.12 - Length of Each Points

5. Pitch diameter is made with Base diameter for make ordinary teeth. (Figure 3.13)

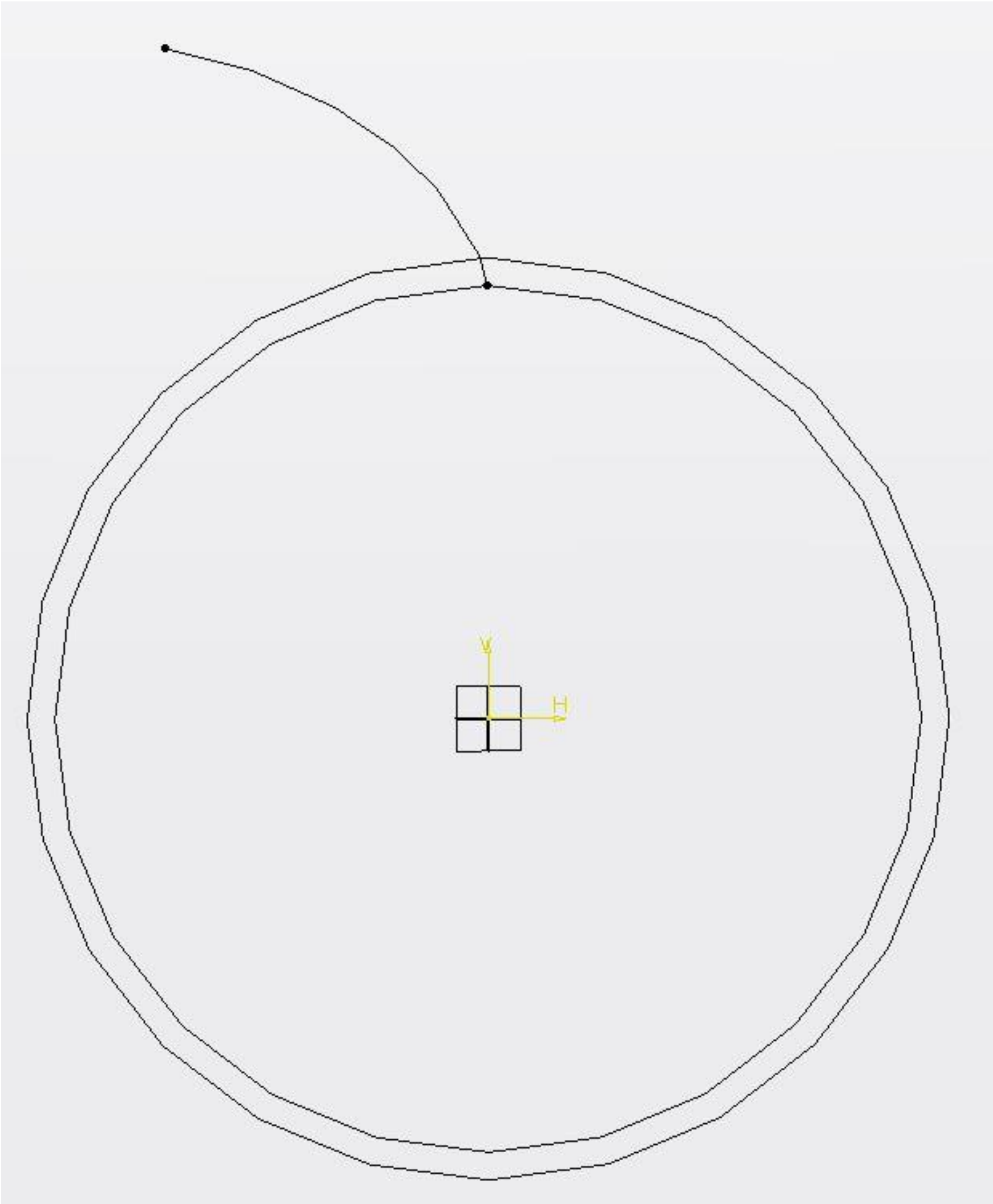


Figure 3.13 - Arc Illustration of Single Face of Teeth

6. Root circle and Outer circle which is black and dashed in Figure 3.14 have been created in order to see the root transition and end of the teeth.

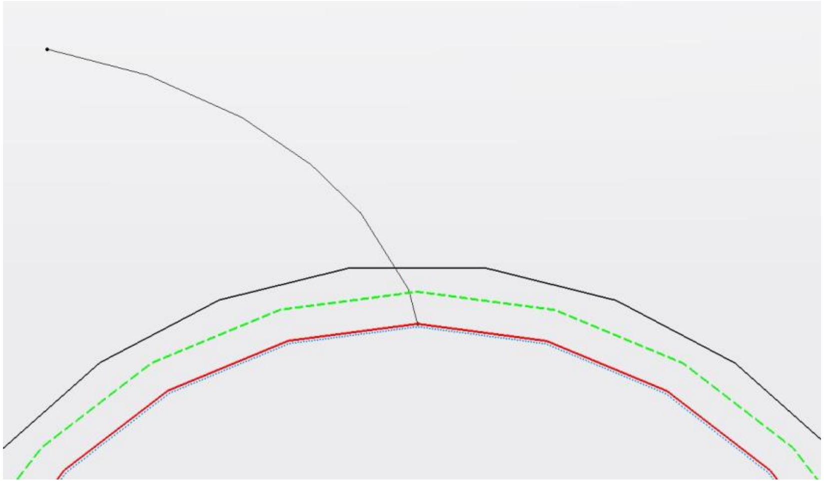


Figure 3.14 - Arc Illustration of Single Face of Teeth with Every Circle

All circles have been painted different color for better recognition.

7. Adding the corner for base and root circle which is drawn in Figure 3.15 with black dot.

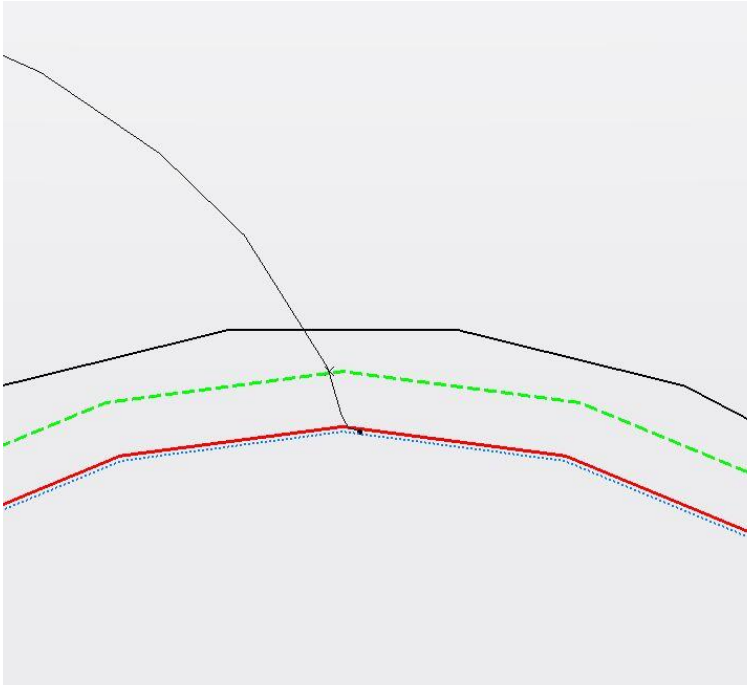


Figure 3.15 - Fillet Radius

8. For making the other face of teeth, the mirror axis must be created that is illustrated as square shaped n Figure 3.16.

$$Length\ of\ Mirror\ Axis = \frac{1.4}{z} \tag{1.68}$$

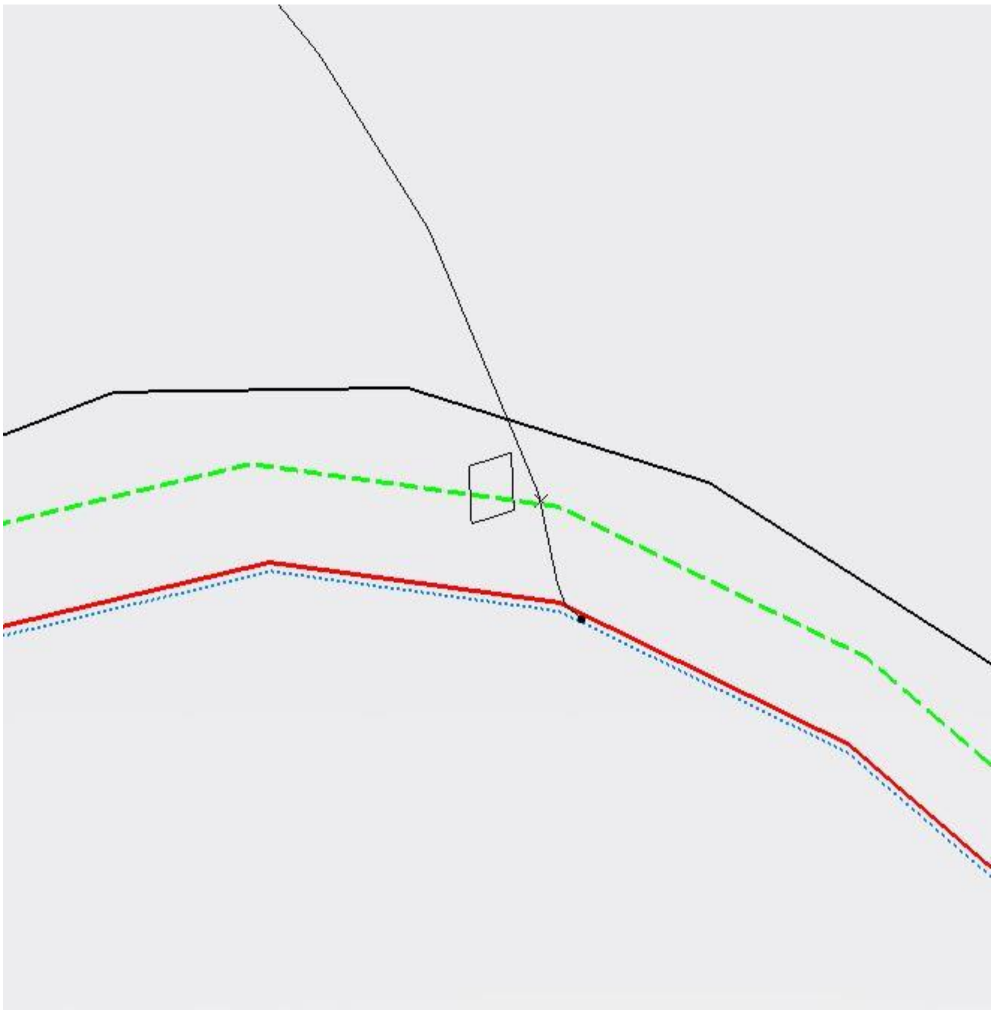


Figure 3.16 - Mirrored Axis

9. The arc was mirrored and trimmed. (Figure 3.17 and 3.18)

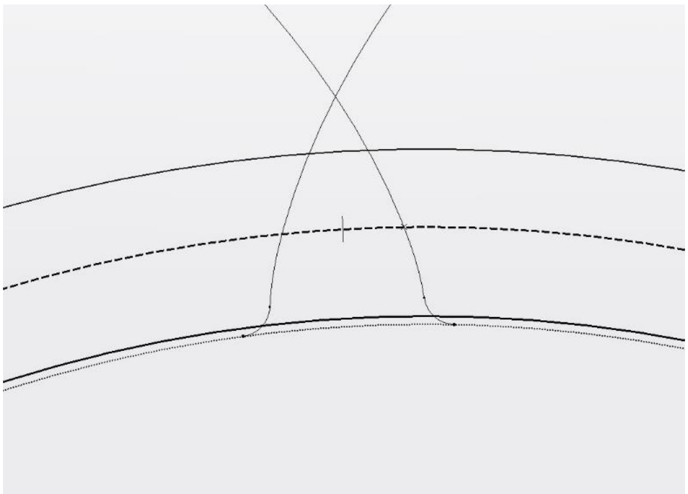


Figure 3.17 - Mirrored Face with Axis and Base Arc

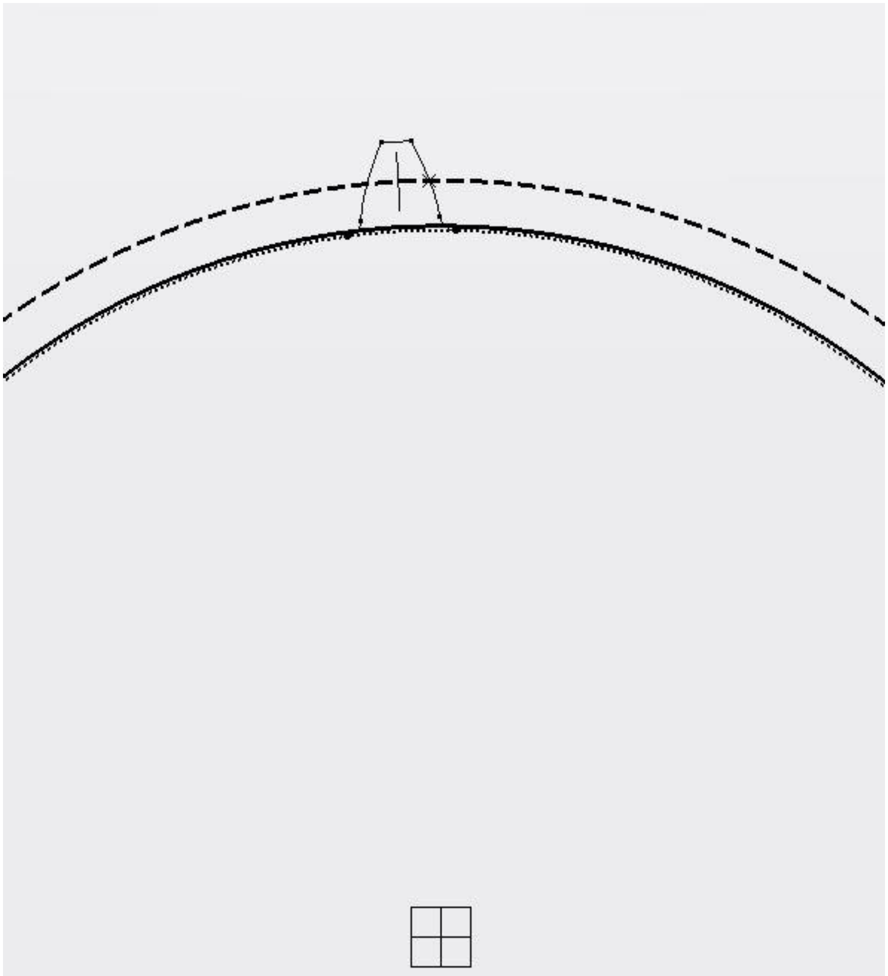


Figure 3.18 - Teeth Feature with Circles

10. After trimming and make one single teeth, it can be arrayed to number of teeth in blueish color in Figure 3.19.

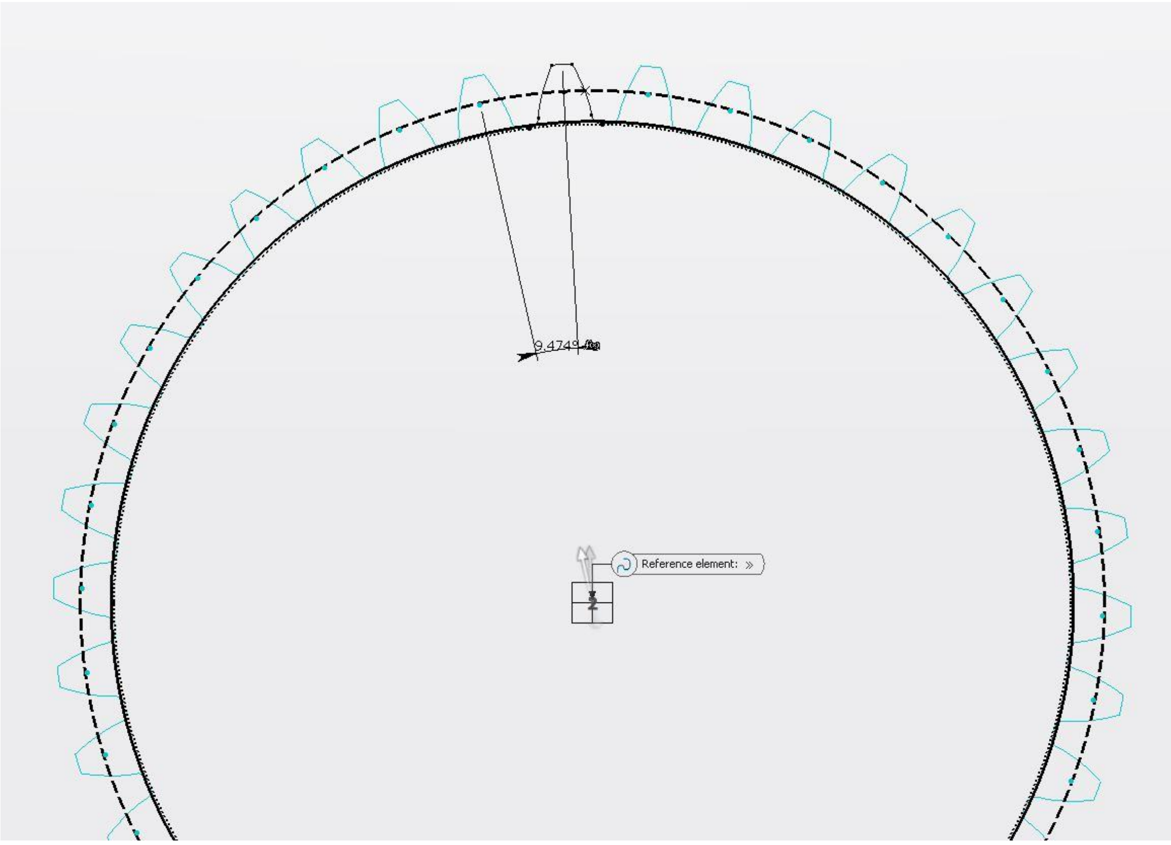


Figure 3.19 - Arrayed Tooth Features

The line from the center must be coincidenced on root diameter and it has to tangent on previously sketched. The already drawn center line with axis line crossing angle must be $\frac{180}{z} = \frac{180deg.2}{38} = 9{,}474°$ according to formulation of involute gear teeth sketching.

11. Then the number of the pinion can be inserted as Array command and make it 38 number of tooth at pinion. As normal behaviour due to geometric rules, the pinion tooth has been fulfilled itself as a round as sketched in Figure 3.20.

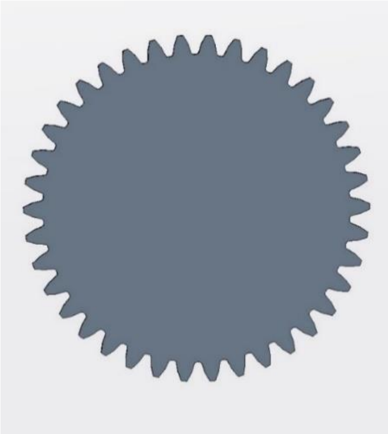


Figure 3.20 - General Feature of Pinion

12. Due to rotational movement of pinion, a key which has been designed and was inserted to the design with the standard of 8x7 key feature in Figure 3.21.

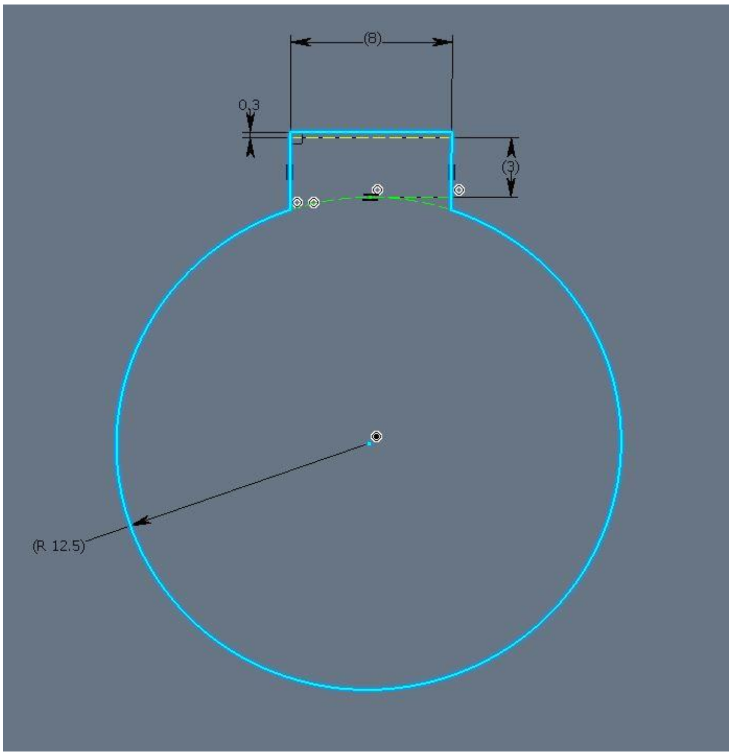


Figure 3.21 - 8x7 Key Feature On Pinion Axis

13. The general visualization of the pinion is;

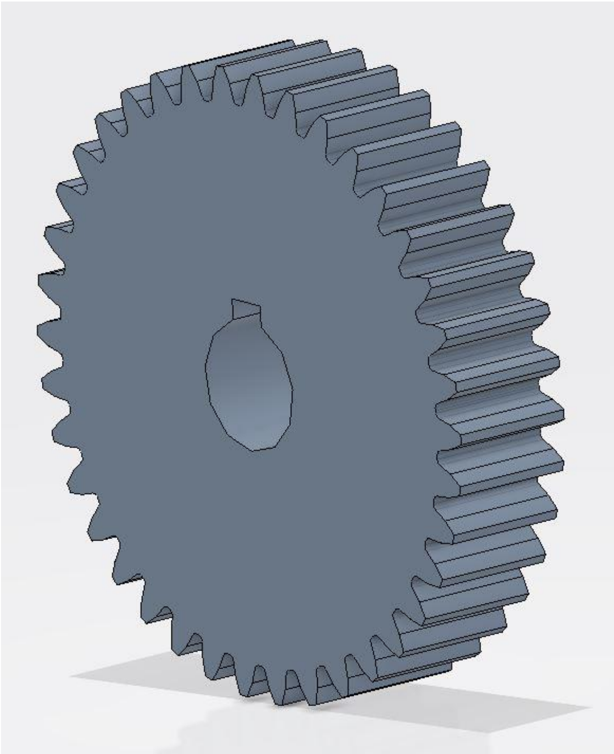


Figure 3.22 - General Display of Pinion with Key

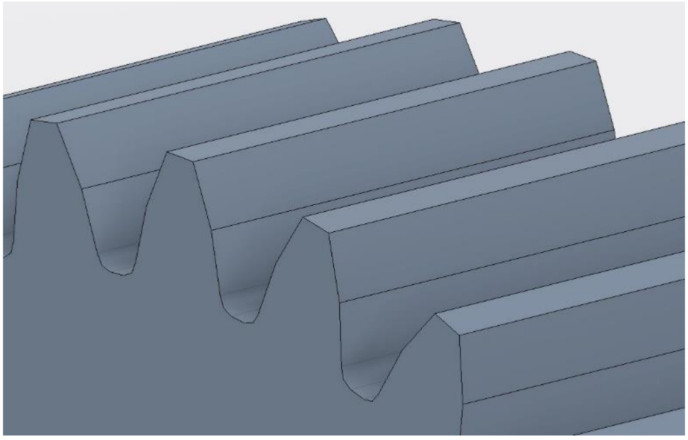


Figure 3.23 - General Display on Tooth

3.4.2.2. Rack Design

With the usage of module 3 as example for previous pinion design, rack has been drawing as in Figure 3.24. Due to occuring of the leveled mounting provision as mounting plate, the rack thickness is to be more than what it demanded which’s drafted as in Figure 3.25. It is not a problem while designing rack. The more it extruded, the more the strength of the part is robust.

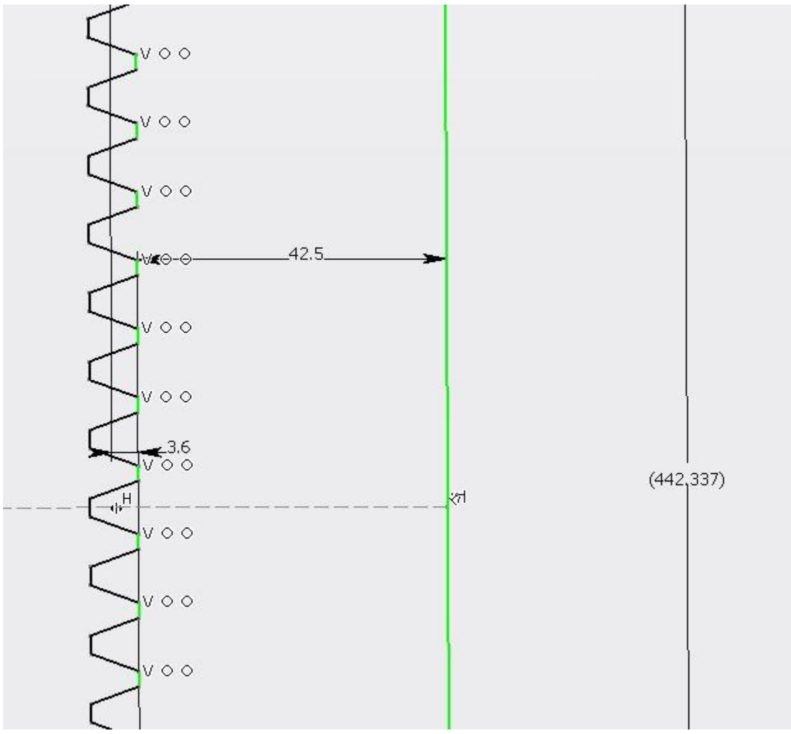


Figure 3.24 - Sketching Rack Design

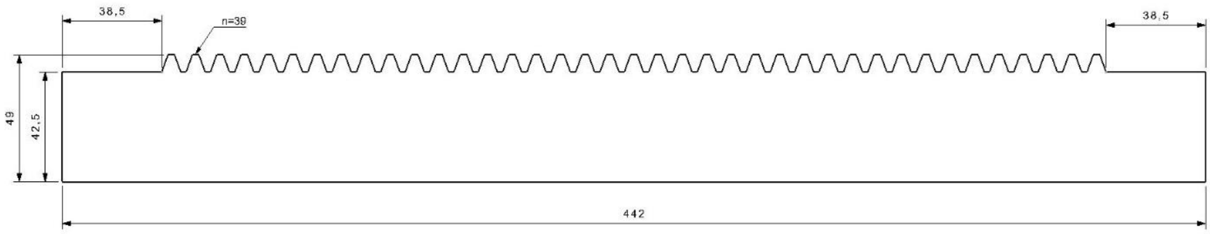


Figure 3.25 - General Dimensions of Rack

Figure 3.26 - Width of Rack

Then it extruded as 25 mm as same as pinion which's drawn in Figure 3.26. The reason why pinion and rack has the same depth is that first of all the design environment with dimensional restrictions force the design as it sketched and the other one is, in all of the literature studies, there are some design to be evaluated and almost all of the systems had been designed with same depth owing to ease to understand.

The general visualization of the rack is in Figure 3.27

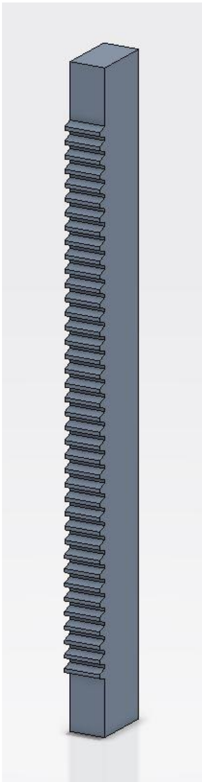


Figure 3.27 - General Display of Rack

3.4.2.3. Mounting The Rack And Pinion

In gear pairs, for better working accuracy and efficient output, line of action of the gears has to be aligned. Consideration of both gears are very simple and easy to thing because for every gear pitch diameter is drawn then it coincidinced on one single point which has demonstrated as in Figure 3.28. The line of action with the other regulations has been maintained in that way.

For rack&pinion, it is more easy. The reason is that, for one gear there are pitch diameter however owing to rack’s aligned shape as rectangular it has no pitch diameter at all. Hence, the pitch diameter of pinion becomes a line of contact for both rack and pinion.

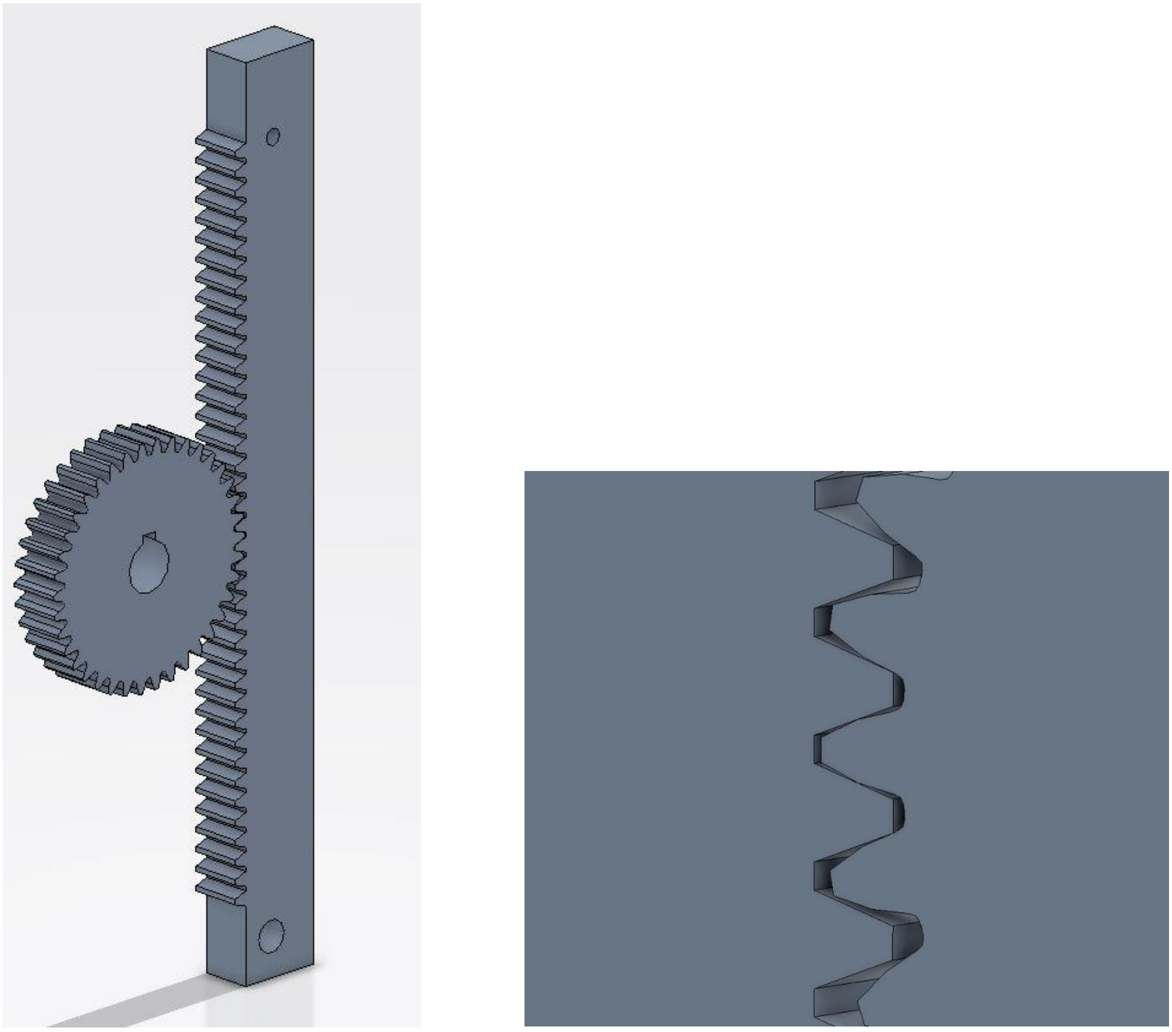


Figure 3.28 - Mounting of Rack and Pinion and Line of Contact

3.4.2.4. Gear Fatique Factors

3.4.2.4.1. Scoring

Also term of scuffing is a term of type of adhesive wear which damage directly on tooth surfaces. In the end, by interaction of single teeth load may cause a breakage of teeth and system as well. Without sufficient lubrication, metal to metal surfaces’ bond together and damaged teeth and lateral layer. Micro breakages may transfer one to another by rotation or movement of the gears. With the movement small particles can scratch just shown in Figure 3.29 and corrupt the surface of teeth. Sliding movement and high contact points are sole logic for that damages. One of the greatest prevention method of damage is lubrication layer. It would protect metal to metal contact. However, lubrication is not one and only prevention, with lubricaion scratches may be occured in meanwhile. [29]

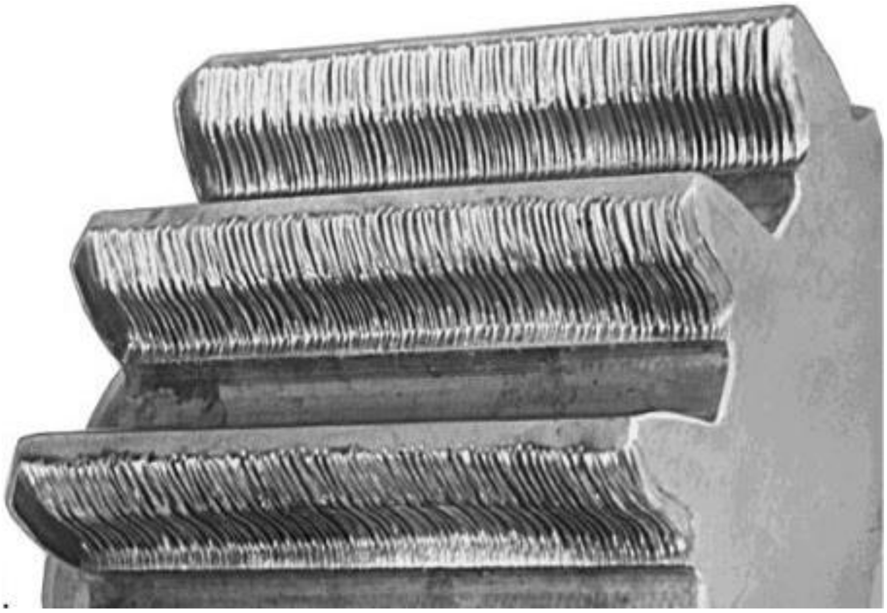


Figure 3.29 - Scoring Effect

3.4.2.4.2 Pitting

Pitting is one of the most common gear fatigue reason. Pitting is a surface damage from cylindrical contact stresses affect directly through with lubrication or non-lubrication film. Material in the fatigue region gets removed and pits evolved which has been obtained as in Figure 3.30. The unfortunate pit evolve to stress concentration and may occur fracture of teeth with high impact surface load. In fact, it also happens on cams, eccentric working components and other moving mechanisms which contains sliding and rolling contact. Again, the pitting launched with small cracks and propagates with repeated conditions. Ultimately, stated crack grows and reflects on tooth surface damage until unstable conditions. [30]

Large deflection of pitting may arise huge volume of vibration and noise. This type of failure happens in both through-hardened and surface-hardened gears.

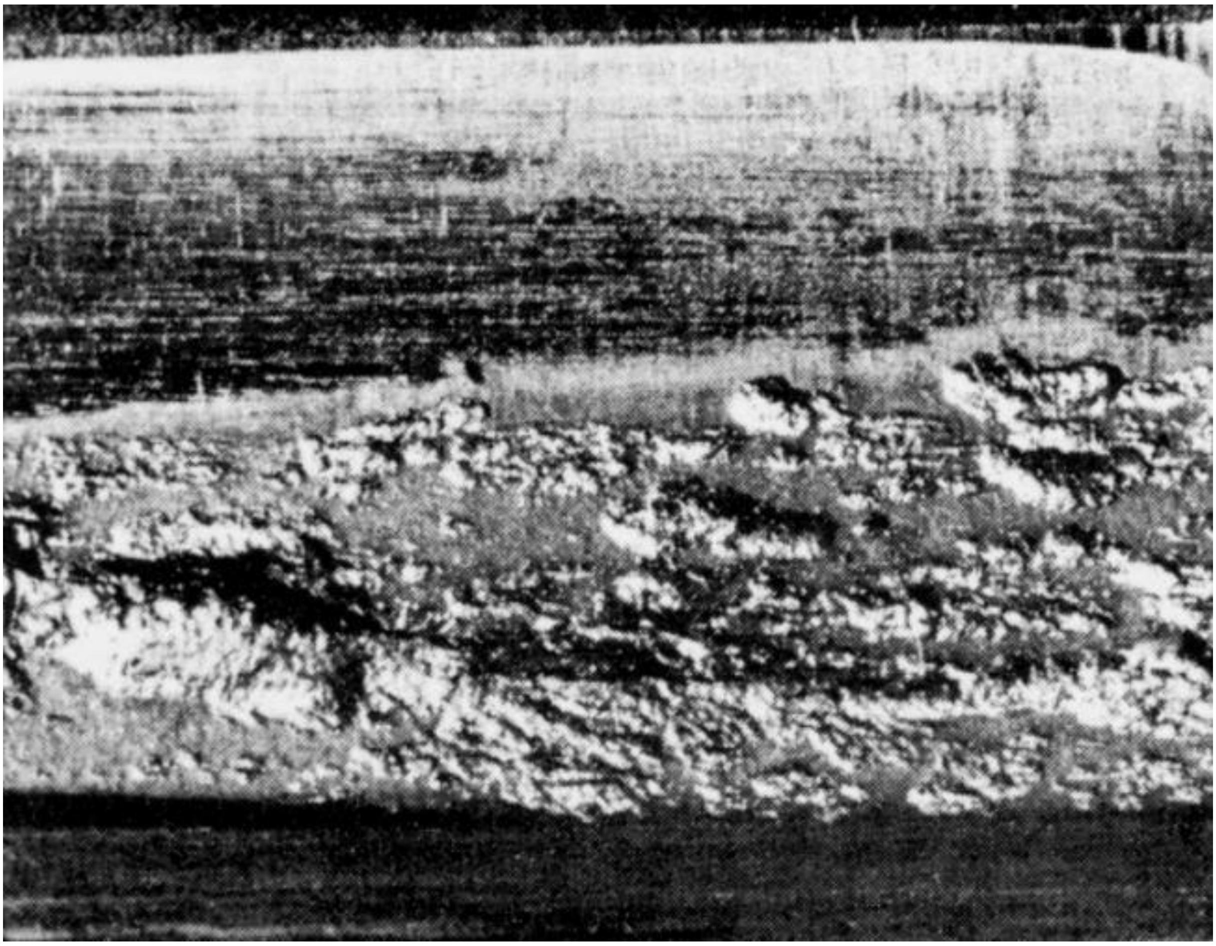


Figure 3.30 - Pitting Effect

3.4.2.4.3. Wear

As per general definition of wear means that continuous loss of material from the surface of a body. Due to mechanical connection and contacts, that kind of tooth damage where the layer of the surface removes one by one but uniformly. According to loss of layer quantity, the teeth gets lighter,weaker and thinner.

The causes of the gear tooth wear is chemical interaction, lack of sufficient lubrication and ingress of abrasive particles.

Generally wear has classified as;

- Adhesive
- Abrasive
- Chemical

Adhasive wear is very hard to see. After million cycles, only visualized adhasive wear can be detectable. Once considering about full load of several cycles gear thickness runs out. [31]

Abrasive wear has some noticable wear in comparison with adhasive. Because due to environmental conditions like transportation machinery, tarmac machines, building constructions and muddy surfaces can affect on wear. The most important point is that mentiones wear occurs gear pairs without housing. The housing brackets slightly reduce the effect of outside roles and decrease dust or other particles in order to prevent wear.

After interaction of environmental conditions, thinned tooth is to be defined as fracture of gear system and become totally failure.

Chemical wear is very understandable due to its own definition. Due to interaction of chemical effect may happen with excessive lubrication which might be acidic properties.

3.4.3. System Design

In general usage; suspension system, Wheel hub, lifting mechanism and mounting plates are shown together in the following Figure 3.31 an 3.32. With the essential usage of the system rely on the continious motion while riding. Every part is must be inserted in that system.

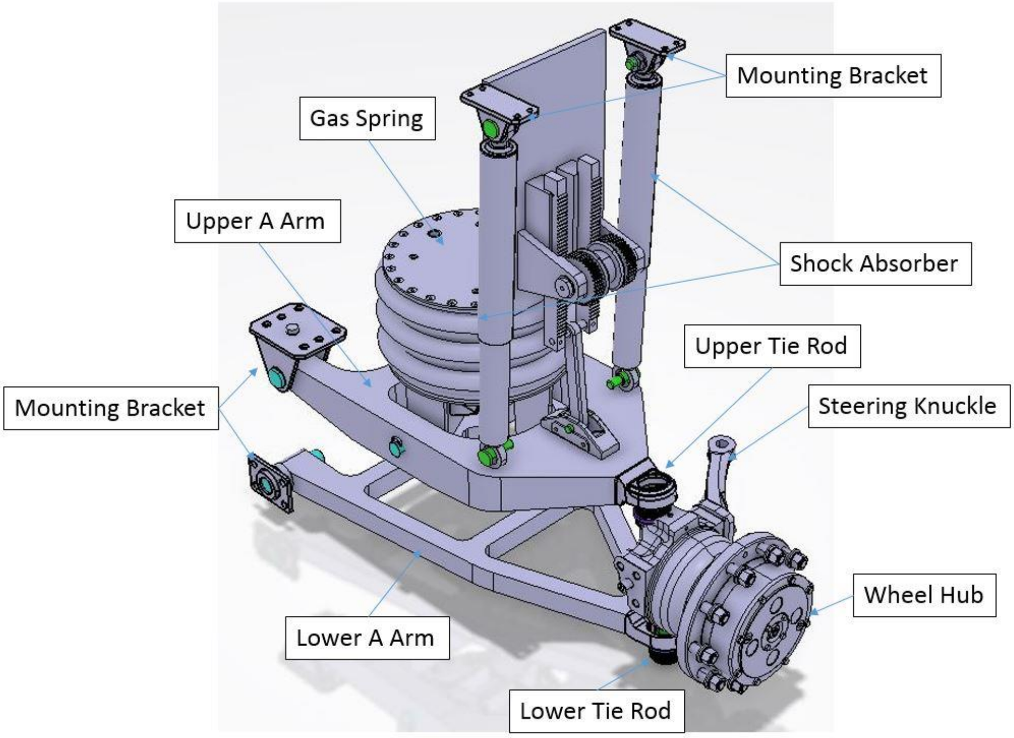


Figure 3.31 - Entire System Design

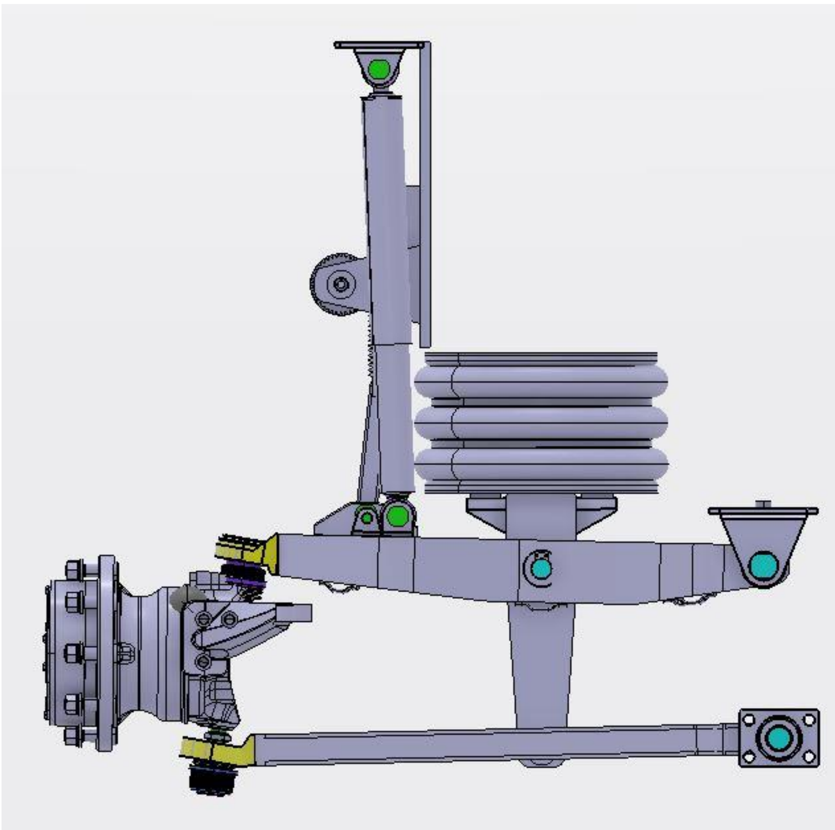


Figure 3.32 - Right Side of the System Design

The mechanism contains two rack and pinion, roller for rope and key usage of pin for motion in Figure 3.33 and 3.34. In that way, rotational movement transforms to linear movement by the help of the key pin. Then suspension system goes up in terms of demanded way which can be seen in Figure 3.35.

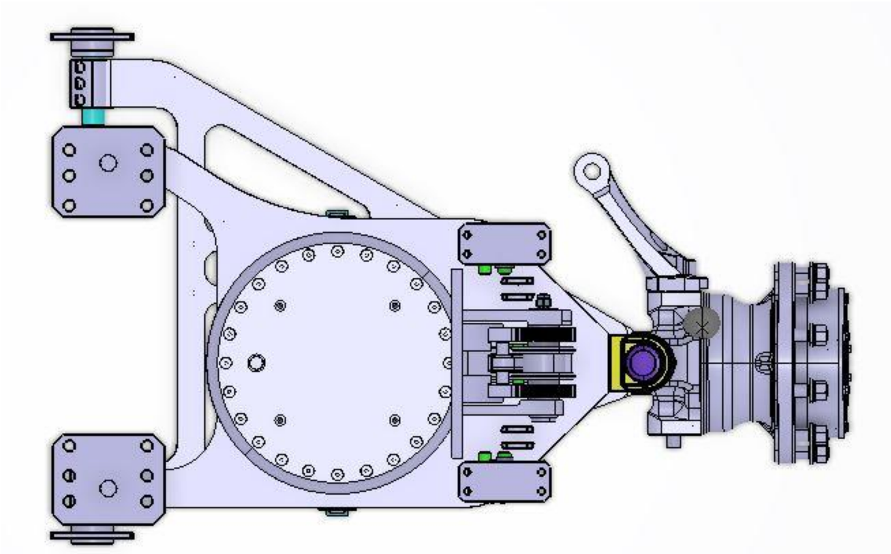


Figure 3.33 - Upper Side the System Design

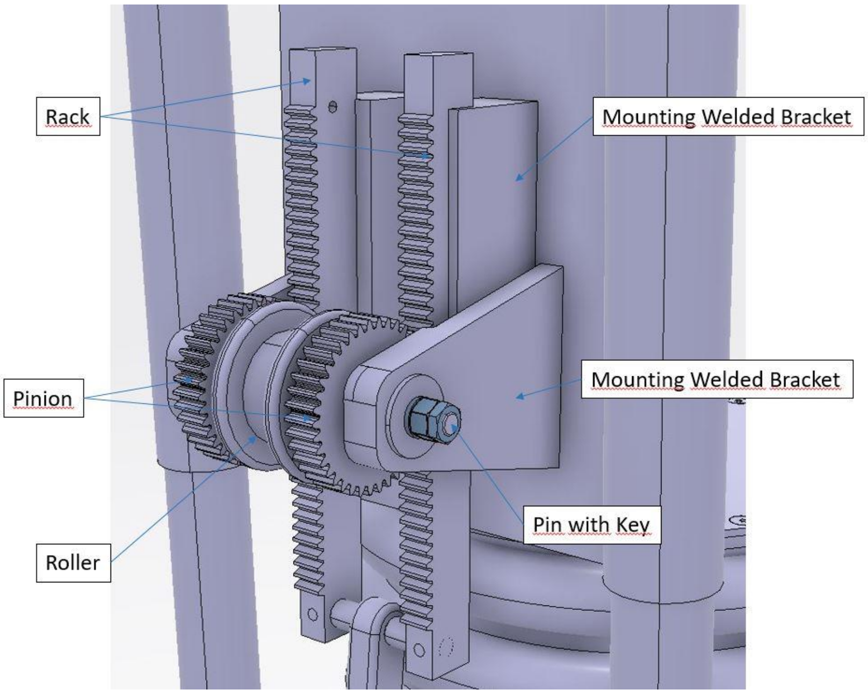


Figure 3.34 - Rack and Pinion with Roller

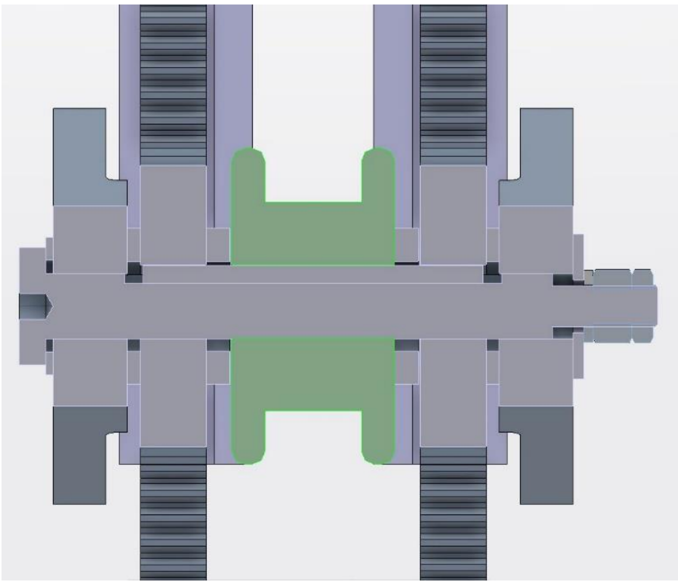


Figure 3.35 - General Mounting for Entire Gear Pair

In order to provide linear movement and prevent autobloackge of system, one additional linkage rod has been identified inside the system. That linkage helps the motion continiously variable and efficient without any potential stopper obstruction. The suspension elements four-bar rotating clash is prevented in terms of using mid-linkage part as shown in Figure 3.36.

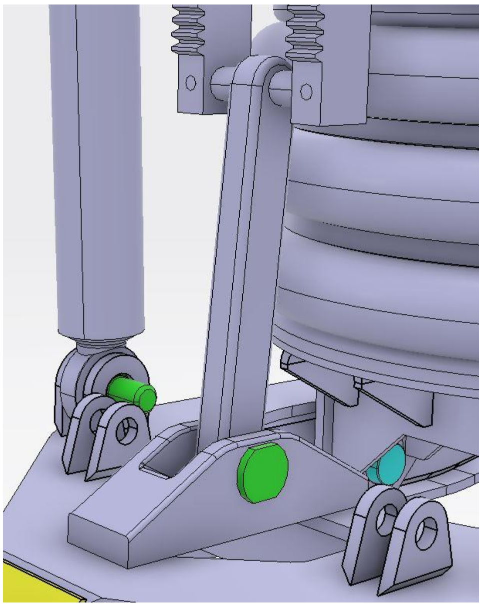


Figure 3.36 - Lifter Linkage Rod with Mounting on A-Arms

The motion seems to look alike slider-crack mechanism. Any rotational movement mistake or error can be overrun by mentioned mechanism in any case of unwanted condition such as rock, dirt etc.

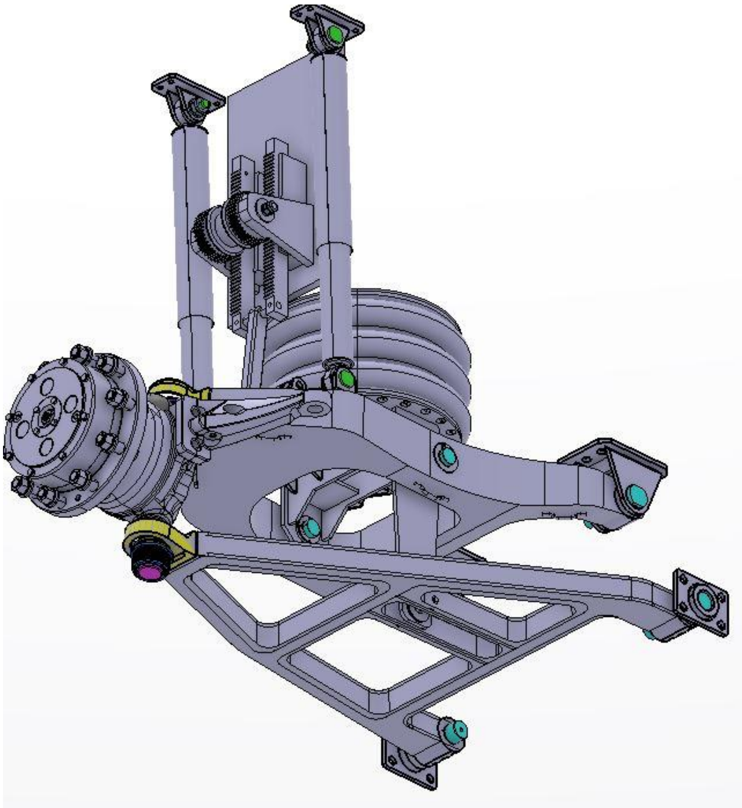


Figure 3.37 - Isometric View for System Design

Exploded View:

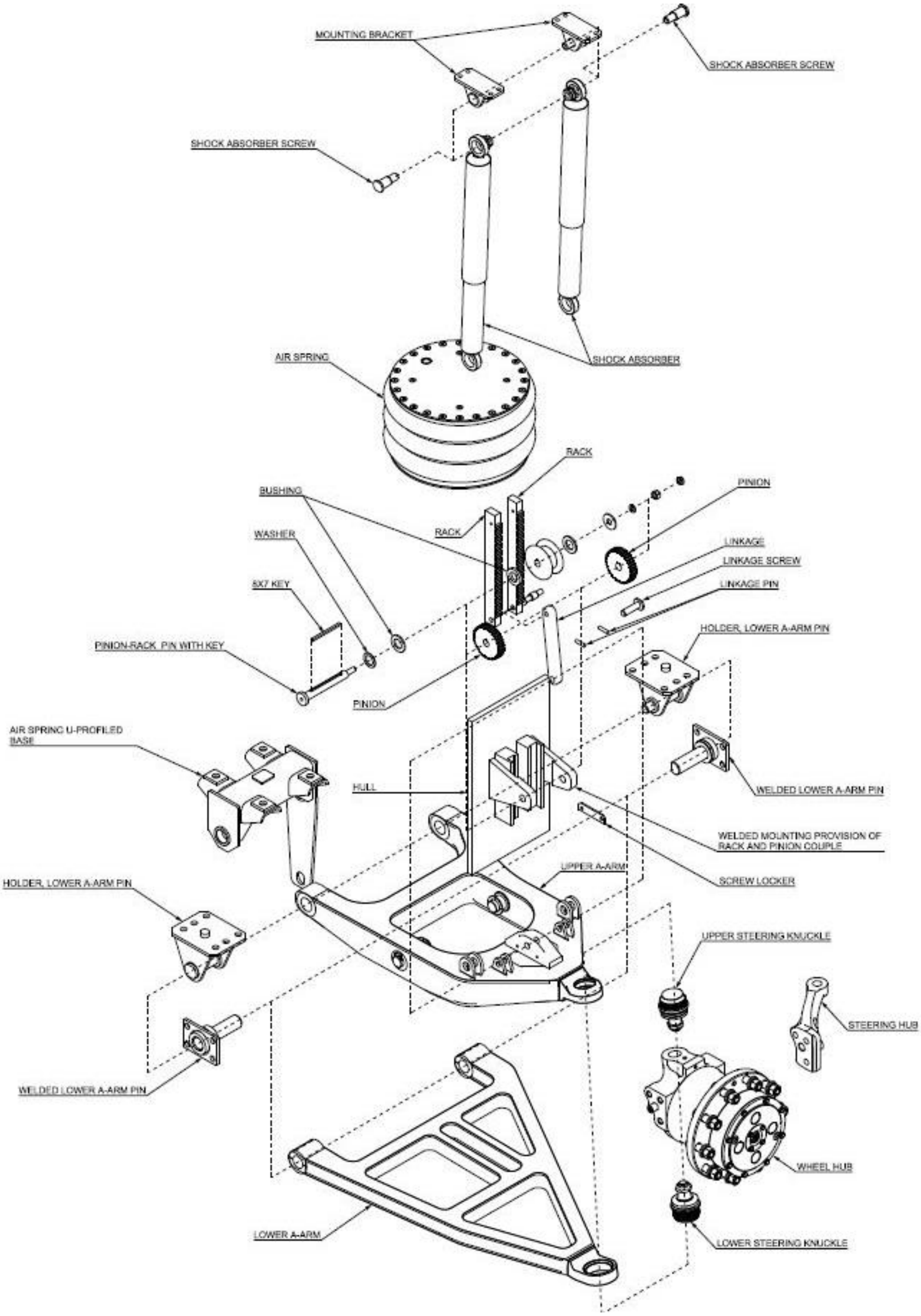


Figure 3.38 - Exploded View of the General System Design with its Components

Upper Side Of Design:

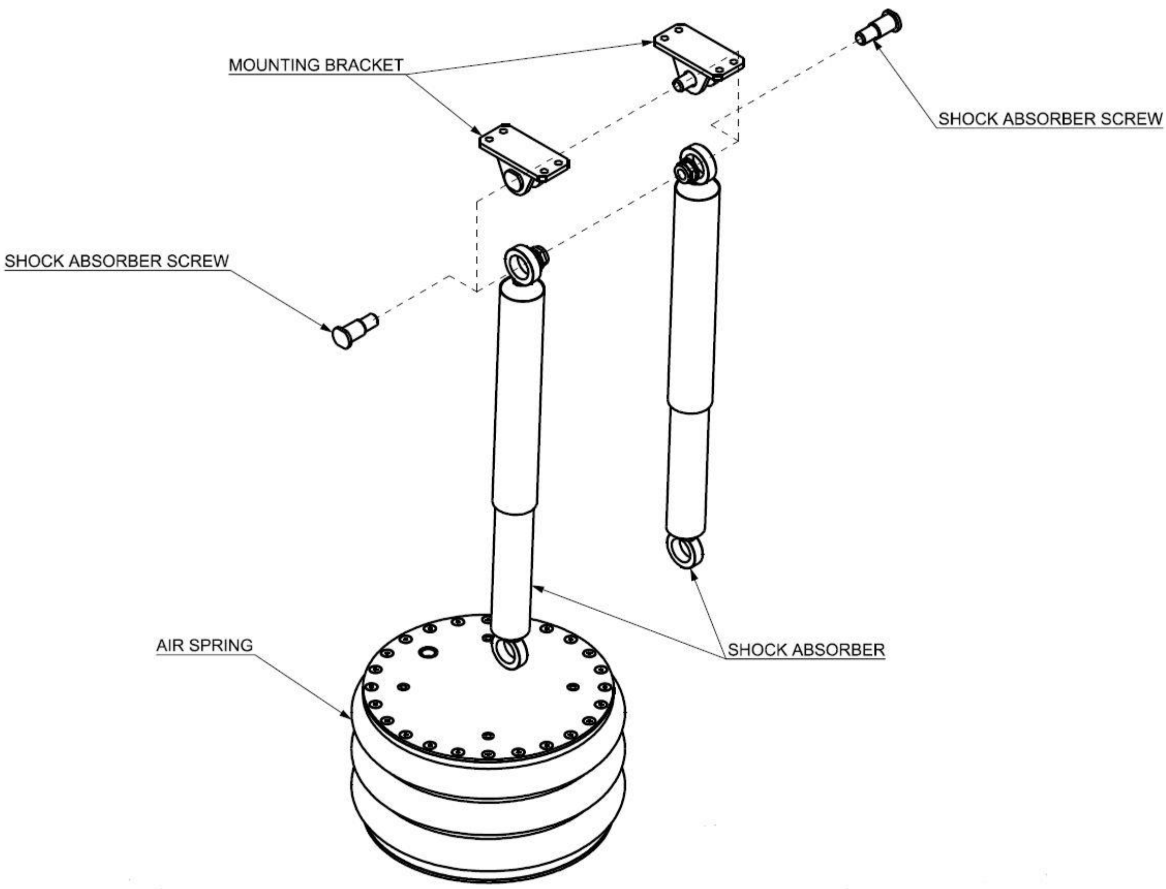


Figure 3.39 - Upper Side of Design

Mounting Brackets mounted to threaded blocks which is welded to vehicle hull with specific dimensions generally in Figure 3.38 and Figure 3.39. Mounting the ordinary suspension elements of vehicle which is shock absorber connected by screws. Unusual element which is air spring mounted on top of the Upper A-Arm.

Central Side of Design:

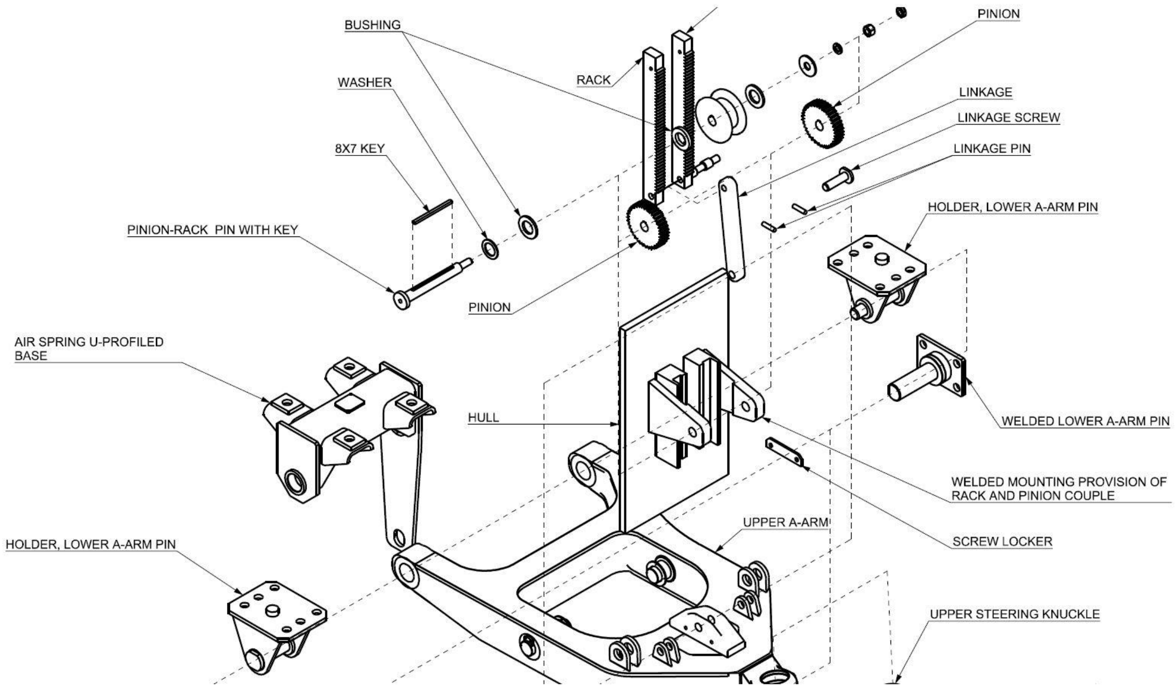


Figure 3.40 - Central Side of Design

Already mentioned two rack and two pinions have located on the welded bracket on vehicle hull. After sliding mission has been accomplished for the design, then the roller between two pinions have put. Inside the roller there is a 8x7 key rectengular hole in order to launch a torsional movement which is shown in Figure 3.40. With the assistance of pin with key mounted with bushing in order to make sliding/torsional movement without any dynamic friction and washer/rondelas for standart mounting rules.

Linkage with its pin helped the system releiving and positioning itself to any potential circumstances which may lead to clash and squeeze.

Holder of the A-arms mounted to brackets which is welded to vehicle hull and its pin screws are put for revolute joint movement.

Lower Side of Design:

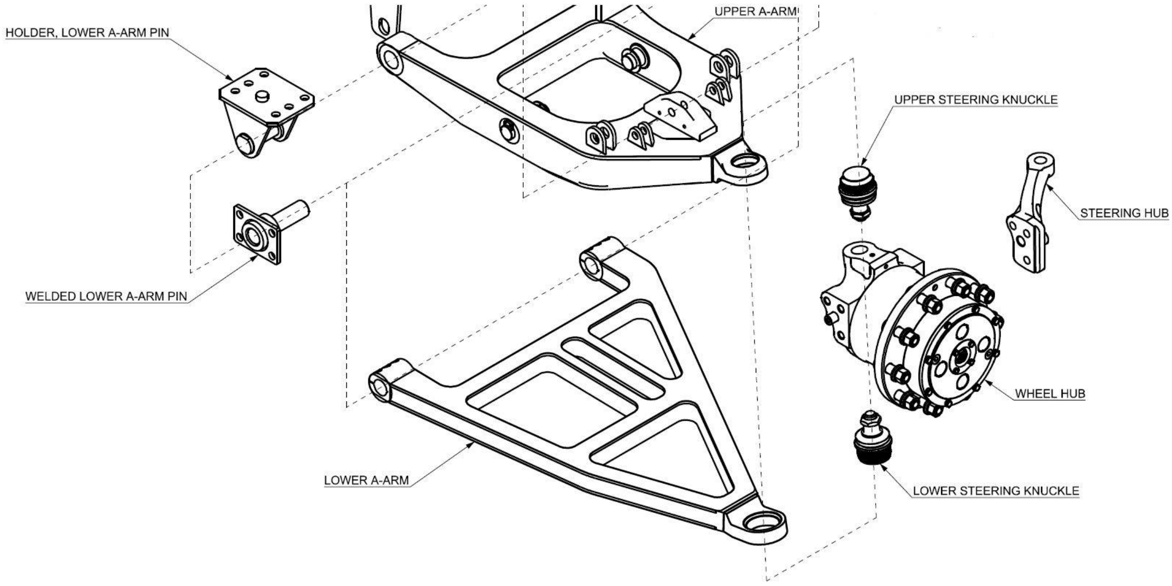


Figure 3.41 - Lower Side of Design

Between the double A-Arms, there must pinpointed a wheel hub and steering hub in order to steer the whole vehicle. As long as steering systems're located properly, there is not any unwanted conditions whilst driving and giving direction in operation. Upper and lower steering knuckle that shown in Figure 3.41 used only for connection between rods and wheels.

3.5. Stress Analysis by Analytic Methods

The study started with gain information about contact mechanics and gears as a general subject. In previous chapter as named as Theory Chapter, acknowledgement and findings are being explained. Theory chapter is also subjected as theorotical improvement step for that study.

The study of the literature was performed and it was ruled to study the norms concerning thoroughly. The intention to study equipment standards was to learn the theory and to go further behind the contact mechanics. The study also established a theoretical method that could be used when the computer program developed was used to be evaluated. Essentially, the research or design approach has crucial role in terms of finding mechanism solution throughout so many years. With that research about contact mechanic helped to find output of the system.

A kinematic model has been created and achieved afterwards. This kinematic method was tend to see the motion of the system either which can move or not. With the help of ADAMS, sliding motion an suspension arms' radial movement directly translate to the stress evaluation of the system. The stated method of kinematic has been enpowered by analytical methods and calculating ways fro the contact mechanics of gears.

Then inevitable comparison between analytical solutions and taken data by software were compared due to thererotical knowledge.

The final phase of the study will be evaluating the desicion maker by all stages such as manufacturing, mold&cavity, lead time, capability of manufacturer and deadline of the potential project. This situation should be underlined that such a design has never been made by any company.

In order to choose best module couple, the stress has to calculated by Lewis and AGMA Analytical Methods. Inside the equation some parameters can vary according to module change.

- Module
- Geometry Factor

Due to design restrictions, pitch diameter is changing very little. So the real difference is related to module number. To see comparison between modules, one selected module as example will be using in the equations and distribute same data to others.

3.5.1. Lewis Equation

$$\sigma = \frac{F_t}{b.Y.m} \tag{1.13}$$

- F_t: 5675N
- b: 25mm
- Y: 0,377 (38 pinion teeth)
- m: 3 (as example)
- V: 0,025m/s

$$\sigma = \frac{5675}{25.0,77.3}$$

$$\sigma = 200MPa$$

$$K_V{}' = \frac{50}{50 + (200.V)^{1/2}} \tag{1.16}$$

$$K_V{}' = \frac{50}{50 + (200.0,025)^{1/2}}$$

$$K_V{}' = 0,9572$$

$$\sigma = \frac{200\ MPa}{0,9572}$$

$$\sigma = 208,95\ \text{MPa}$$

3.5.2. AGMA Procedure For Finding Bending And Contact Stresses

A method has been derived to calculate bening and contact stresses in spur and helical gears using the American National Standards Institute-American Gear Manufacturers

ANSI/AGMA Standards 2001/D04 and AGMA Standards 908-B89 briefly described below. A more detailed description can be found in theory chapter. [17, 18]

The main reason for the use of the AGMA standard was based on the fact that it was used by most of them. References used in the study of the literature. The main elements of the system is rack and pinion couple.

In order to sketch those part, firstly module has to be defined. In general usage of other systems especially for steering mechanism etc. are being used as module 2-5. In some literature references, module 3-4 were used based on handling huge forces as same principle as lifting up the suspensions. For company wise in terms of supply chain system, availability and ease to find of the materials are extremely important.

In that example in order to show researchers best module selection, module 3 has been calculated. By leveling the system with better accuracy and prevent the slip away the component on themselves, two racks and two pinions are being used. In the design, racks are squeezed between mounting provision for mounting and pinion itself.

Compressing the rack has some advantage for the system.

- Always provide the line of contact with pinion.
- For leveling, ease to setting up the system.
- Usage of housing may effect on non-muddy environment and it makes sliding effortless.
- Cutting the load with software based program to prevent adjustment of the rack would be prevented by the system automatically.
- The most important mechanical behaviour of the system has not autoblockage system. For example worm gear can be used in that design as well. However, if it has been used, it blocked itself while suspension rising up and down.

In the mechanism, these are the given or calculated datas previously and can be used with Table 3.2.

- d_p: 114mm
- V: 0.025m/s
- t: 10s
- $total\ weight$: 578,5kg

Table 3.2 – Rack and Pinion Mechanical Properties

	Pinion	Rack
Material	AISI 4140	AISI 4140
Young Modulus (GPa)	210	210
Yield Strength (MPa)	1150	1150
Poisson's Ratio	0,3	0,3
Density (kg/m^3)	7850	7850

The Bending Stress for Pinion can be calculated from the equation of that equation.

$$\sigma = F_t.K_v.K_o.K_s.\frac{1}{b.m}.\frac{K_H.K_B}{Y_J} \tag{1.30}$$

In order to find solution. The factors and parameters will be taken step by step.

- F_t: $5675\ N$
- K_v: 1

In order to find K_v, manufacturing of the tooth must be machined the most perfect way. In that study, accuracy of pinion and its tooth has been considered as perfect and in good quality. Therefore, it has been selected in blue region. Moreover, it has been calculated that 0.025m/s is extremely low velocity according to design requirement and prior tests.

In the end of the setting up the Dynamic (Velocity) factor as in Figure 3.42, it has been taken as unity; 1.

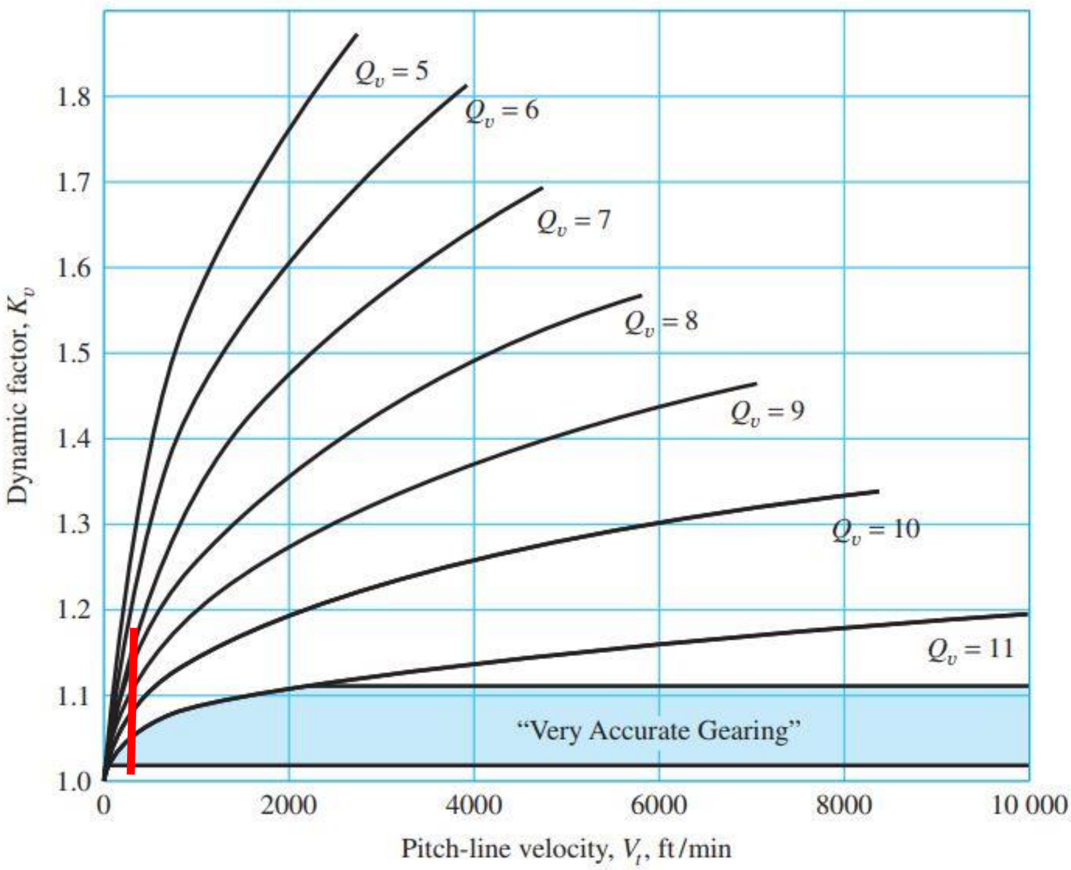


Figure 3.42 - Dynamic Factor by Pitch Line Velocity

- K_o: 1,5 in Table 3.3

Table 3.3 - Overload Factor Selection

	Driven Machinery		
Source of Power	Uniform	Moderate Shock	Heavy Shock
Uniform	1.00	1.25	1.75
Light Shock	1.25	1.50	2.00
Moderate Shock	1.50	1.75	2.25

In the design, during suspension rolling up and down through the road, mechanism always works yet without any start up force. So with that approach, from driven machinery side moderate shock may require because only exploding actions are defined as heavy shock. Since, in these situation, 500-600kg is relatively moderate shock among other systems such as steering systems etc. On the other side, torquemotor of the system is giving the demanded force in good curve. There is no much steep waves. That's why, light shock has been selected in that situation.

- K_s: 1

To AGMA, while standardization of every possible design; it has been tested and seen from researchers that neither the system design is lifting to much force with extreme shock or sliding the light hatches with very light force they see that taking K_s as unity or not makes not difference at all. In the end of the calculation procedure, K_s has to be taken as 1 by the goods of design wise in order to make a design as in Figure 3.43.

- b: 25mm

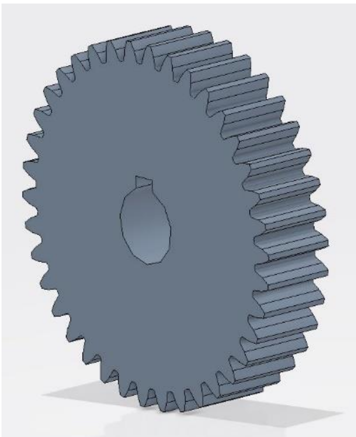


Figure 3.43 - General Pinion Design

- m: 3

For design, the most first selected item must be module.

- K_H

$$K_m = K_H = C_{mf} = 1 + C_{mc}(C_{pf}.C_{pm} + C_{ma}.C_e) \tag{1.36}$$

- $C_{mc} = \begin{cases} 1, & for\ uncrowned\ teeth \\ 0.8, & for\ crowned\ teeth \end{cases}$

C_{mc} is selected as 1 due to usage of uncrowned teeth.

$$C_{pf} = \begin{cases} \frac{b}{10.d} - 0.025, & b \leq 25.4mm \\ \frac{b}{10.d} - 0.0375 + 0.0125.b, & 25.4mm < b \leq 431.8mm \\ \frac{b}{10.d} - 0.1109 + 0.0207.b - 0.000228.b^2, & 431.8mm < b \leq 1016mm \end{cases} \quad (1.37)$$

$$\frac{b}{10.d} - 0.025, \qquad b \leq 25.4mm$$

That formulation can be used.

$$\frac{25mm}{10.114mm} - 0.025 = 0.02193 - 0.025 \rightarrow -0.00307$$

Which makes no sense for gain output of the system. So we can use the note at the end of the Load Distribution Factor section. [23]

İf F/(10d) < 0.05, F/(10d) = 0.05

So C_{pf} is 0.05

- $C_{pm} = \begin{cases} 1, & for\ straddle - mounted\ pinion\ with\ \frac{S_1}{S} < 0.175 \\ 1.1, & for\ straddle - mounted\ pinion\ with\ \frac{S_1}{S} \geq 0.175 \end{cases}$

C_{pm} is 1.0 due to rack&pinion coupled with two mounting provisions and it mounted very stable so it can be said the design is straddle. So, C_{pm} is 1.

C_{ma} = A + B.b + C.b^2 (1.38)

Table 3.4 - Gear Manufacturing Empirical Constants Selection

Condition	A	B	C
Open gearing	0.247	0.0167	−0.765(10^{-4})
Commercial, enclosed units	0.127	0.0158	−0.930(10^{-4})
Precision, enclosed units	0.0675	0.0128	−0.926(10^{-4})
Extraprecision enclosed gear units	0.00360	0.0102	−0.822(10^{-4})

With the selection of perfect accuracy manufacture teeth-pinion, extraprecision enclosed gear units can be selected in order to find C_{ma} as stated before such as Table 3.4.

$$C_{ma} = A + B.b + C.b^2 = 0.00360 + 0.0102.25mm + (-0.822.10^{-4}.25)$$

$$C_{ma} = 0.00360 + 0.0102.25mm + (-0.822.10^{-4}.25)$$

$$C_{ma} = 0.01372$$

C_e has been selected as 0.8

$$C_e = \begin{cases} 0.8, & for\ gearing\ adjusted\ at\ assembly \\ 1, & for\ all\ other\ conditions \end{cases}$$

Overall gathering the formulation for finding the C_{mf}

$$K_m = K_H = C_{mf} = 1 + 1(0.05.1 + 0.01372.0.8)$$

$$K_H = 1.03597$$

- K_B: 1

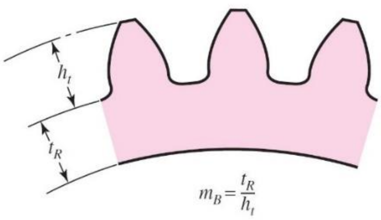


Figure 3.44 - Rim-Thickness Factor Figure

$$K_B = \begin{cases} 1.6\ln\dfrac{2.242}{m_B}, & m_B < 1.2 \\ 1, & m_B \geq 1.2 \end{cases}$$

m_B can be selected more than 1.2 due to the design which has been mentioned previously such as Figure 3.44.

- Y_J: 0.427

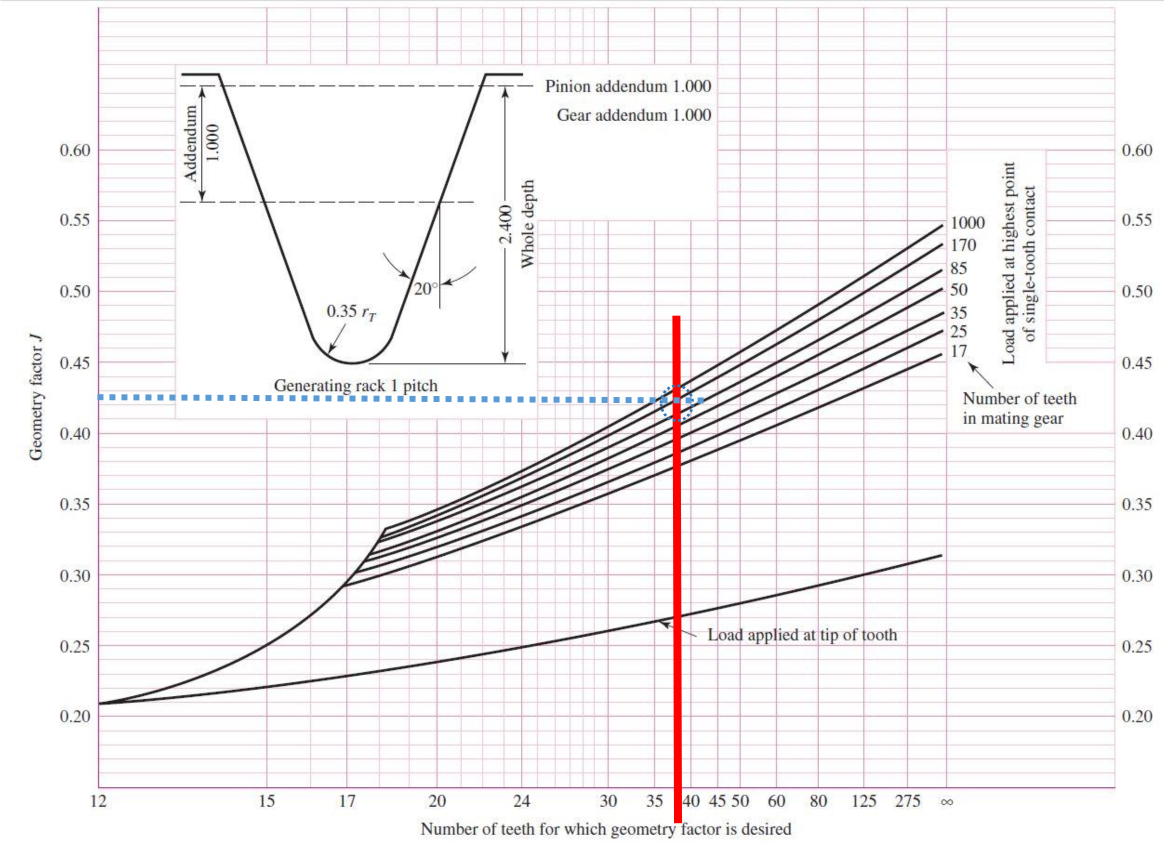


Figure 3.45 - Geometry Factor by Number of Teeth

Geometry Factor for pinion is very easy to determined. Number of teeth 38 was calculated. However the scheme is sketched to answer bosth second gear and rack as well. Once the study contains rack&pinion then infinite number of teeth must be selected. 1000 number of teeth is approximately as same as inifinite in the gear design world. So, J can be found between 0.40 and 0.45 due to Figure 3.45. Approximate interpolation will be 0.427.

So the general formulation of bending stress can be written with factors as well.

$$\sigma = F_t.K_v.K_o.K_s.\frac{1}{b.m}.\frac{K_H.K_B}{Y_J} \quad (1.30)$$

$$\sigma = 5675Nm.1{,}5.1{,}0.1{,}0.\frac{1}{25mm.3}.\frac{1{,}03597.1{,}0}{0{,}427}$$

$$\sigma = 275{,}37MPa$$

Pitting resistance stress can be calculated from the equation of that equation;

$$\sigma_c = C_P.\sqrt{F_t.K_v.K_o.K_s.\frac{K_H}{d_w.b}.\frac{Z_R}{Z_I}} \quad (1.31)$$

- F_t: 5675 N
- b: 25mm
- K_v: 1

Velocity factor is calculated and found in bending stress equation previously as unity.

- K_o: 1,5

Table 3.5 - Overload Factor Selection

	Driven Machinery		
Source of Power	Uniform	Moderate Shock	Heavy Shock
Uniform	1.00	1.25	1.75
Light Shock	1.25	$\boxed{1.50}$	2.00
Moderate Shock	1.50	1.75	2.25

Overload factor is not found by identified due to design requirement and restrictions in Table 3.5.

- K_s: 1

Size factor is taken as 1 as same as Velocity and Overload Factor.

- K_H: 1,03598

Load distribution factoris calculated previosly for bending stress. By taking the same load, the factor can be used again.

$$K_m = K_H = C_{mf} = 1 + 1(0.05.1 + 0.01372.0.8)$$

$$K_H = 1.03597$$

- d_w: 114mm which has been drawn in design stage stated as Figure 3.46.

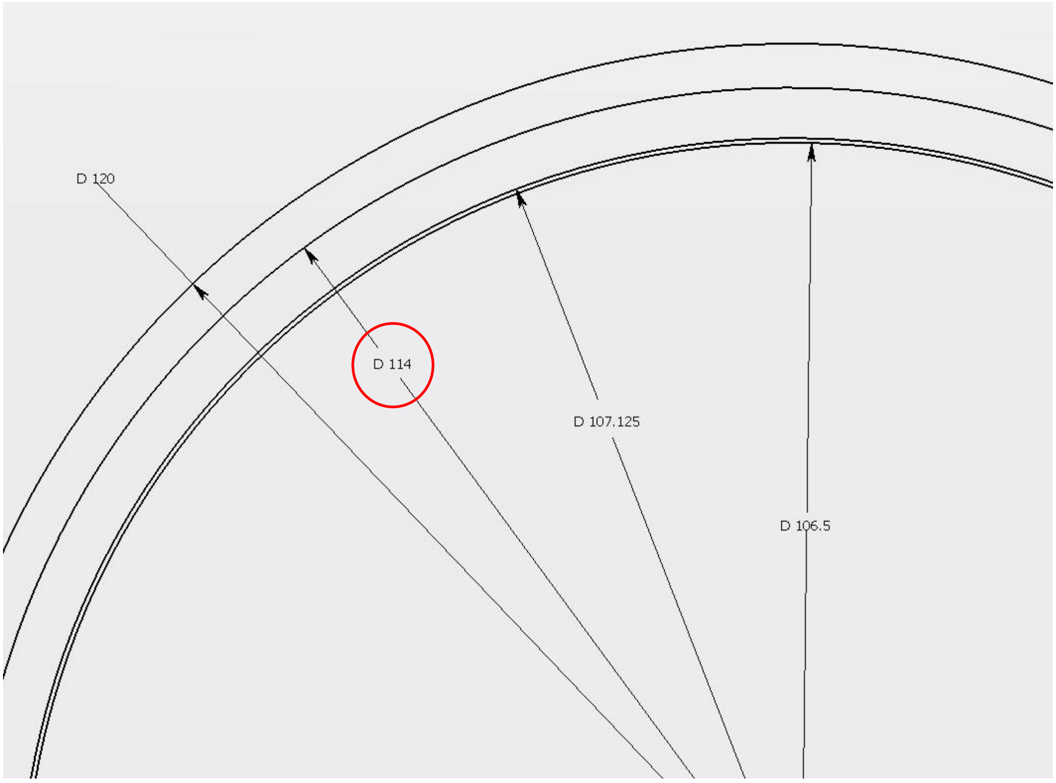


Figure 3.46 - Pitch Diameter

- Z_R: 1

Surface Condition Factor is to be 1 without any detrimental effect regarding pinion and rack.

- Z_I: 0,1606

Pitting resistance bending factor solely can be used for Pitting Resistance. In order to use, we need to pay attention about the note regarding m_N. For spur gear, m_N can be taken as 1.

Furthermore, in studied design the gear pairs are consisted of pinion and rack. So m_G can be taken as 1 as well. And another unknown transverse angle (ϕ_t) can be taken as 20° degree.

$$Z_I = \frac{cos\phi_t . sin\phi_t}{2. m_N} . \frac{m_G}{m_G + 1} \tag{1.48}$$

$$Z_I = \frac{cos20. sin20}{2.1} . 1$$
$$Z_I = 0{,}1606$$

- C_P: 191,7

$$C_P = \sqrt{\frac{1}{\pi . \left[\left(\frac{1 - \nu_1^2}{E_1}\right) + \left(\frac{1 - \nu_2^2}{E_2}\right)\right]}} \tag{1.34}$$

Simplification of the equation due to using the same type of material for both rack and pinion.

$$C_P = \sqrt{\frac{1}{2\pi . \left(\frac{1 - \nu^2}{E}\right)}} \tag{1.35}$$

For AISI 4140 steel has same mechanical properties.

Table 3.6 - AISI 4140 Elastic Modulus and Poisson's Ratio

Elastic modulus	190-210 GPa
Poisson's ratio	0.27-0.30

So the formulation can be written as;

$$C_P = \sqrt{\frac{1}{2\pi.\left(\frac{1-0{,}3^2}{210.000}\right)}}$$
$$C_P = 191{,}7$$

For all of the unknowns currently are found and ready to be put inside the general Pitting Resistance Contact Stress

$$\sigma_c = C_P.\sqrt{F_t.K_v.K_o.K_s.\frac{K_H}{d_w.b}.\frac{Z_R}{Z_I}} \quad (1.31)$$

$$\sigma_c = 191{,}7.\sqrt{5675N.1.1{,}5.1.\frac{1{,}03598}{114.25}.\frac{1}{0{,}1606}}$$

$$\sigma_c = 841{,}45MPa$$

However the founding number is not exactly contact stress, it is a contact stress number and its definition is special for only AGMA standards. In order to find the result is safe according to formulation in AGMA.

<u>Allowable Contact Stress Number</u>

$$\sigma_c \leq \frac{\sigma_{HP}}{FoS}.\frac{Z_N}{Y_\theta}.\frac{Z_W}{Y_Z} \quad (1.33)$$

$$\sigma_c \leq \frac{\sigma_{HP}}{1{,}15}.\frac{1}{1}.\frac{1}{1}$$

Finding σ_{HP} as allowable contact stress number is related to AGMA Standard Figure of through hardened AISI 4140 Steel can be read by its Brinell Hardness in Figure 3.47.

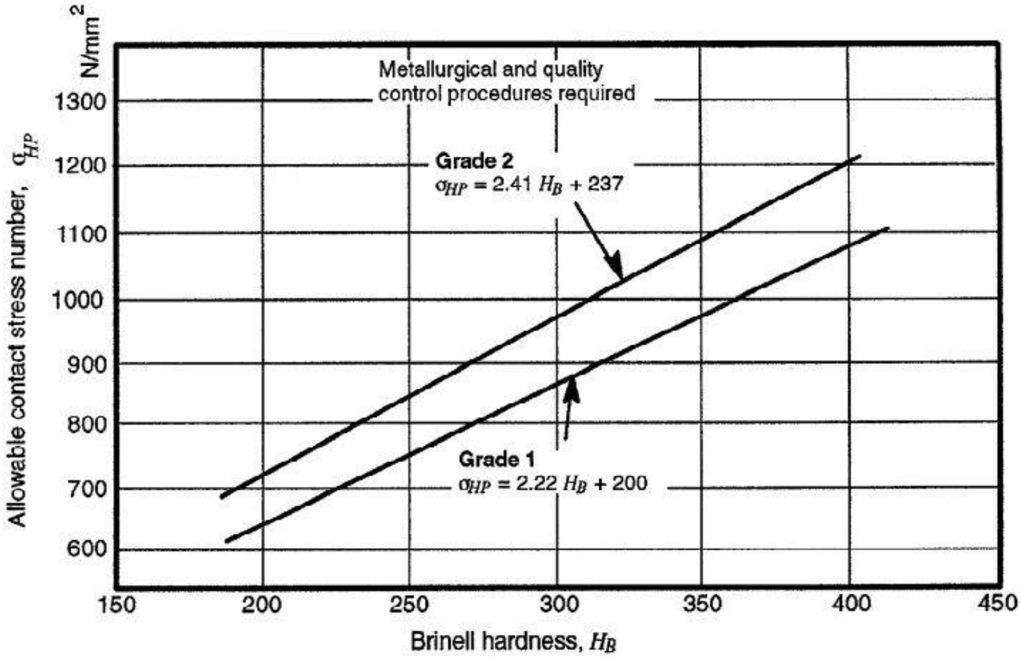


Figure 3.47 - Brinell Hardness According to Contact Stress Number

400 Brinell Hardness is very proper both for manufacturing and capability of 4140 Steel. Grade 2 is related to quality of the steel. It has been examines and discussed in AGMA [18] So;

$$\sigma_{HP} = 2{,}41H_B + 237$$
$$\sigma_{HP} = 1201MPa$$

Subsequently; the equation must be

$$\sigma_c \leq \frac{1201}{1{,}15}.\frac{1}{1}.\frac{1}{1}$$

$$841{,}45MPa \leq 1000MPa$$

So we can say that design specifications are safe.

Safety Factor Evaluation For Wear Resistance

- Safety Factor of Bending Fatigue Failure:

$$S_F = \frac{S_t . Y_N / (K_T K_R)}{\sigma} \tag{1.33}$$

Firstly, allowable bending stress numbers must be found as S_t. To AGMA, it can be found from interpolation and with heat treatment knowledge.

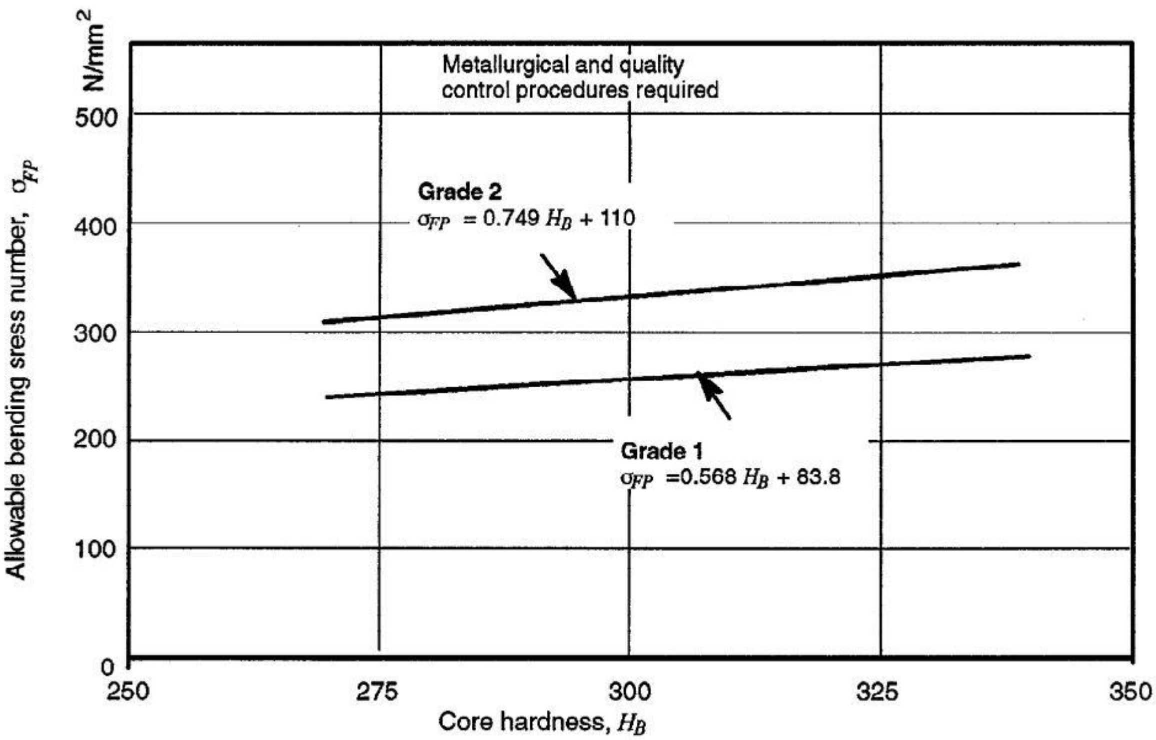


Figure 3.48 - Allowable Bending Stress Numbers for Through Hardened Steel Gears for AISI 4140

In order to find the neccessary bending stress numbers as in Figure 3.48, first it needed to find core hardness as Brinell Hardening and it put in the formulation as in MPa unit.

In that table, there is two different line on the plot. One is for Grade 1 and other is for Grade 2. In AGMA 2001 standard, Grades are being specified according to Steel heat treatment type which ar through hardened, flame or induction hardened, carburized and hardened and nitrided (through hardened). Allowable bending stress is differed from the selected material. In that study, AISI 4140 steel is selected and in terms of Tensile Strength from table Grade 2 can be chosen. AGMA 2001 Tables 7 thru 10 list cleanliness requirements for different steel heat treatments (through hardened, induction/flame hardened, carburized, nitrided). Grade 2 cleanliness is "AMS 2301 or ASTM A866, (no certification required)". Grade 3 cleanliness is "AMS 2301 or ASTM A866, certification required". Steel cleanliness mostly affects fatigue life and/or reliability rates. The allowable stress values listed in the standard are based on 10^7 cycles, 99% reliability, and unidirectional loading. There is also a notation that the Grade-2/Grade-3 cleanliness requirements only apply to the material around the gear teeth. [17, 37, 38]

In that study; due to cleanliness requirements Grade 2 was selected.

$S_t = 0{,}749.H_B + 110MPa$ (36HRC=330HB)

The estimated core hardness becomes 36HRC which is normal for behaviour of AISI 4140

$S_t = 0{,}749.330 + 110MPa$

$S_t = 357{,}17$MPa

For Y_N: Reliability 0,99 and 10^7 life cycle with upper part can be selected as max factor.

$Y_N = 1{,}3558.N^{-0{,}0178}$

$Y_N = 1{,}3558.10^{7.(-0{,}0178)}$

$Y_N = 1{,}3558.10^{7.(-0{,}0178)}$

$Y_N = 1{,}0176$

For K_T:

Temperatures up to 120°C, the factor must be unity.

For K_R:

Reliability factor changes according to below table.

Requirements of application	K_R [1)]
Fewer than one failure in 10 000	1.50
Fewer than one failure in 1000	1.25
Fewer than one failure in 100	1.00
Fewer than one failure in 10	0.85[2)]
Fewer than one failure in 2	0.70[2) 3)]

NOTES
1) Tooth breakage is sometimes considered a greater hazard than pitting. In such cases a greater value of K_R is selected for bending.
2) At this value plastic flow might occur rather than pitting.
3) From test data extrapolation.

Figure 3.49 - Reliability Factor

Then K_R must be 1.00 has been provided form Figure 3.49.

The equation will be;

$$S_F = \frac{S_t . Y_N / (K_T K_R)}{\sigma} \tag{1.57}$$

$$S_F = \frac{357{,}17 \,.\, 1{,}0176/(1.1)}{275{,}37}$$

$$S_F = 1{,}32$$

<u>Safety Factor of Pitting Resistance Failure:</u>

$$S_H = \frac{S_C . Z_N . C_H / (K_T K_R)}{\sigma_C} \tag{1.58}$$

For S_C: AGMA 2001[17] table can be used for the exact result.

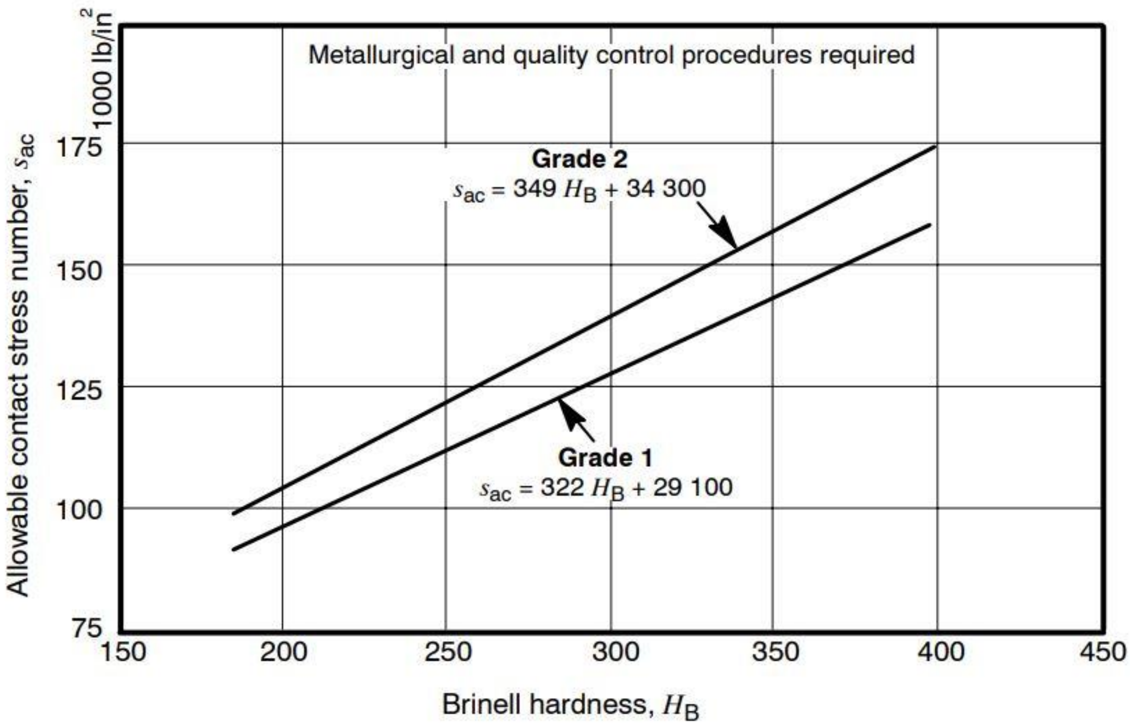


Figure 3.50 - Allowable Bending Stress Numbers for Through Hardened Steel Gears

From lb/in2 to MPa usage according to Brinell Hardening, the formulation becomes in Figure 3.50;

$$S_C = 2{,}41 H_B + 237 MPa$$

In the table, the maximum Brinell Hardening becomes 400, however in our system demanded HB must be 552. Therefore, for not violating the table 400HB can be used for the formulation's correct alignment.

$$S_C = 2{,}41 H_B + 237 MPa$$
$$S_C = 1201 MPa$$

As same as Bending Stress Safety Factor formulation; Z_N, C_H, K_T, K_R can be used as previously found.

$$S_H = \frac{1201.1{,}0176.1/(1.1)}{841{,}45}$$

$$S_H = 1{,}45$$

Identifying between $S_f = 1{,}32$ with $S_H{}^2 = 1{,}45^2 = 2{,}10$ so the threat in the pinion is not from wear.

As explained before, the design factor of safety has been defined as 1,15 in order to assume previous tests.

Other Modules are tabulated in Table 3.7.

Table 3.7 - Module Comparison

	m2	**m3**	**m4**	**m5**	**m8**	**m10**
Number of Teeth	57	38	29	23	14	11
Pitch Diameter	114	114	116	115	112	110
Contact Ratio	1,7766	1,7034	1,6459	1,5916	1,4623	1,3572
Bending Stress	375,26	275,37	220,47	190,67	191,71	Geometry Factor FAIL
Pitting Stress	841,45	841,45	834,17	837,79	1012,48	

Therefore due to select the best practice among the mentioned modules are being calculated as AGMA.

- As seen from tabulation and Figure 3.7 module 10 is imposiible to use in equation and it is neglected from table.
- As the general knowledge for gear is that less than 25 number of teeth is not suggested to use for better accuracy and distribution of load so module 5 and module 8 is eliminated.
- Module 2 has relatively higher stress value among others. Though the less contact ratio value selecting the other modules which are less stress value is logiccal to choose.

- Module 3 and module 4 is very close to each other. However, which module's contact ratio value are close to 2, then the selection criteria is completed. Module 3 is essential design constant from now on for our lifting design.

3.6. Kinematic Analysis

In this study, to see and understand the comprehensive analyzing steps ADAMS was chosen for the lifting up suspension mechanism. According to Vehicle Dynamics must be evaluated by 4th Generation Calculator Tools which are ADAMS, ABAQUS etc. Briefly, those kind of kinematic analysis manipulated by initial position of the system, final position of the system, velocity, accelaration and time dependent. CATIA models of the driving mechanism is impossible to translate directly to the ADAMS owing to capability of ADAMS. ADAMS is only related to motion and force change of general system. [30]

The most critical path of the system is that starting point of the kinematic and while in final point how the system pushes itself to the end. The nature of the motion in the end of the lifting up, the arms endure until the last teeth or before last teeth, so the force increase inevitably. Other significant portion of the building system is that using the hardpoints. [31]

The same motion approach is valid for lifting down the system as well. Within the contribution of the nature weight of the system, general force distribution delibaretely falls.

With the aid of use ADAMS, the potential problems such as clash, self-lockage, extreme force occurence can be seen. After that analysis, the general resistance of the stress must be evaluated according to both analytical calculations and FEA results.

Normal Location of the System is showed below. The natural usage of the ADAMS may encouraged engineers to simplify the designs more pure and less part quantity.

The inputs of the systems are represented in Figure 3.51 and Figure 3.52;

- Upper and Lower A-Arms

- Two Shock Absorbers
- Wheel hub
- Two Rack
- Combined Pinions with Roller
- Linkage between Racks and Upper A-Arm

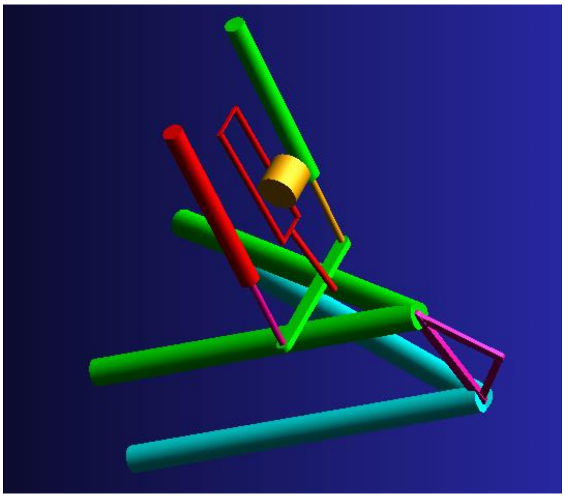


Figure 3.51 - Normal Status of Kinematic System

During the motion, some contacts and restrictions must be defined inside the ADAMS as properties. The reason of definition of these are ADAMS is checking both the components individually and locking possibility of the entire system.

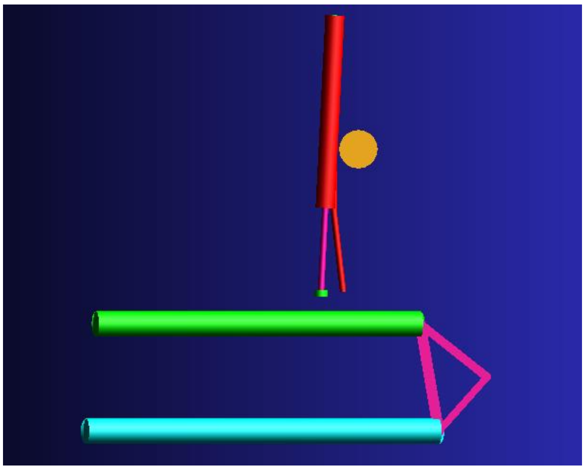


Figure 3.52 - Normal Status of Kinematic System

The accurate and true definition of contacts are to shown accountable results. Hence, stated restrictions are;

- Shock absorbers and air springs effective force get negligible due to previous tests.
- Shock absorbers motion has defined as translational contact.
- The two-end of the A-arms have defined as revolute contact.
- The contact between wheel hub and two A-arms motion have defined as spherical joint.
- Smaller diameter side of the shock absorbers have defined as bonded contact
- A linkage connection has defined as revolute joint
- Two rack has allowed to move only in Z-direction.
- The connection between combination of pinion and roller has put directly on their line of action which is related solely for pinion.
- The length allowance of racks has been limited between -200mm to +200mm in Z-direction.
- The whole swinging components such as wheel, wheel rim wheel hub, arms etc. has been deployed as point of mass of 578,5kg in its center of gravity.

3.6.1. Lifting Up the Whole System

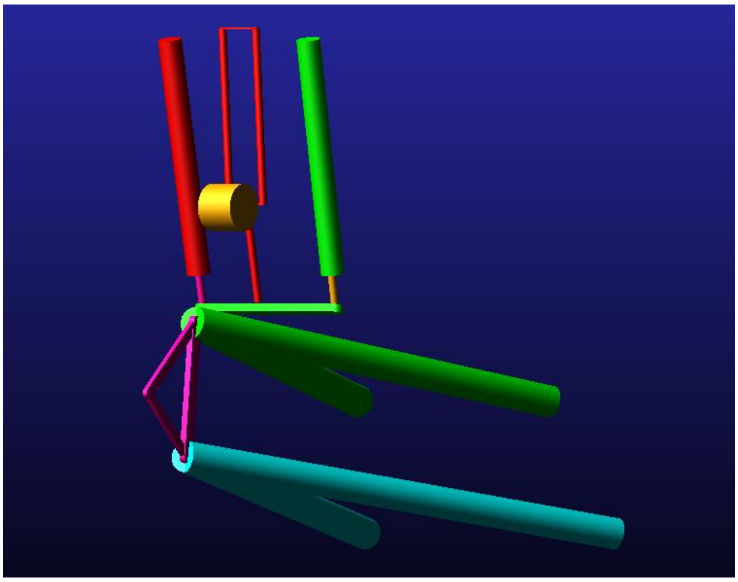


Figure 3.53 - Lifting Up Status of Kinematic System

As seen from Figure 3.53, seeing that motion end present the demanded motion has been achieved without any lockage literally. At the same time, that roller with pinion has turned until the rack dimensional translational reaches to the +200mm in Z-direction. A linkage has done its work while changing the angle due the nature of the suspension arms radii of curvature as shown in Figure 3.54.

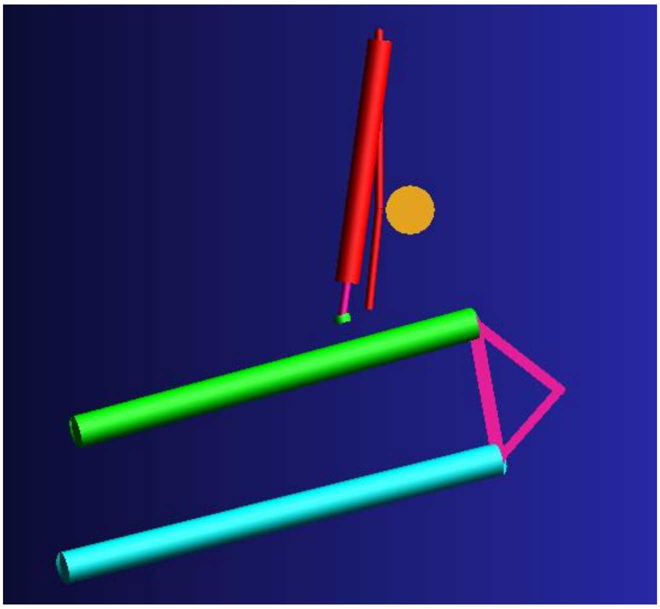


Figure 3.54 - Lifting Up Status of Kinematic System

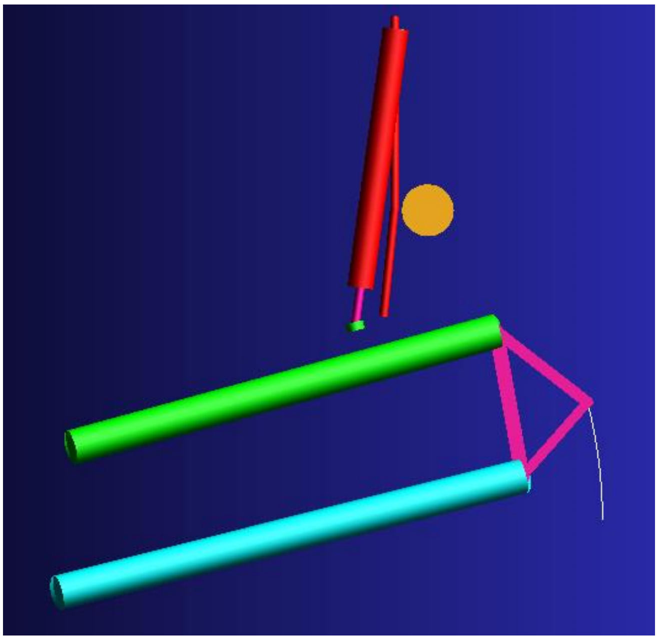


Figure 3.55 - Lifting Up Curvature of Kinematic System

The spline has been drawn by ADAMS from its natural location to whole lifting in Figure 3.55. As seen from above, the x-direction location change of the system is sufficient for wheel.

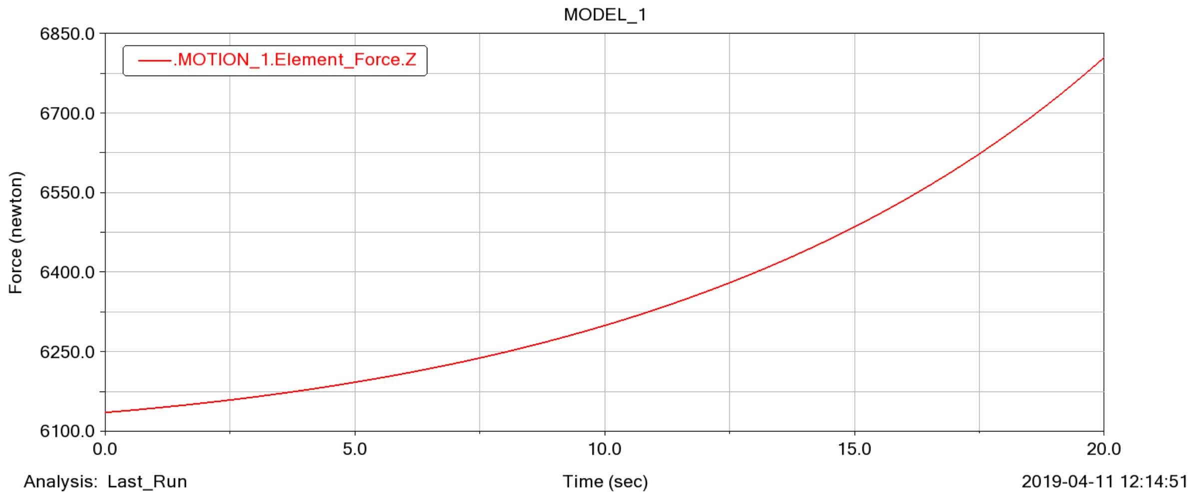


Figure 3.56 - Load Change on Single Teeth by Point of Mass

The motion has been calculated for twenty seconds in the system. It may vary in the future studies owing to requirements of the company and their clients.

The exponential trend shows in Figure 3.56 that approximately 6150 Newton increased to 6800 Newton. The graph shows that the end of the load may be faced in drastic situation therefore it may be considered as the maximum load of the system.

In previous analytical solutions of the weight estimation was 5675 N. So the comparison between the analytical solution with analysis solution according to loads are quite distinctive.

3.6.2. Lifting Down the Whole System

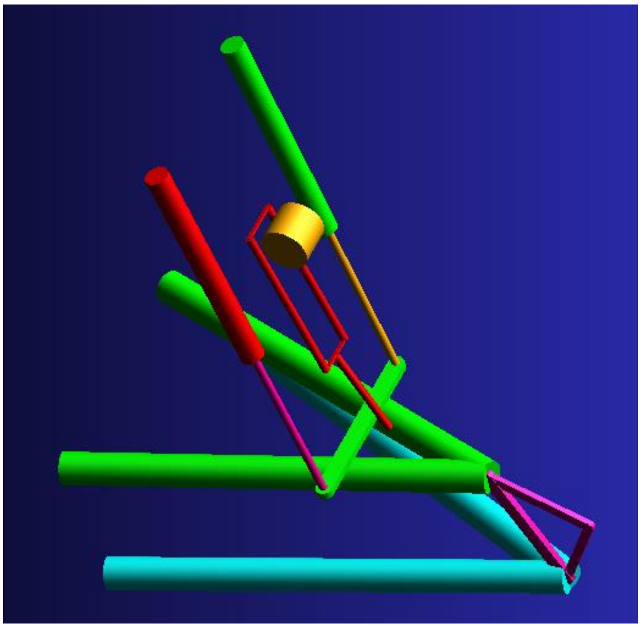


Figure 3.57 - Lifting Down Status of Kinematic System

As seen from the Figure 3.57 that lowering down the entire system to 200mm has been working as same as lifting up. The linkage worked as slider crank mechanism part. In Z-direction of -200mm is its minimum point of the system.

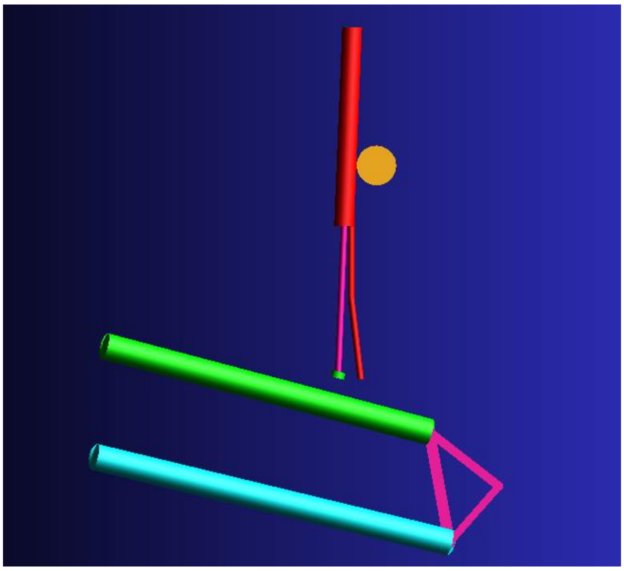


Figure 3.58 - Lifting Down Status of Kinematic System

According to advantage of the lowering down the system as in Figure 3.58 has used the whole systems weight.

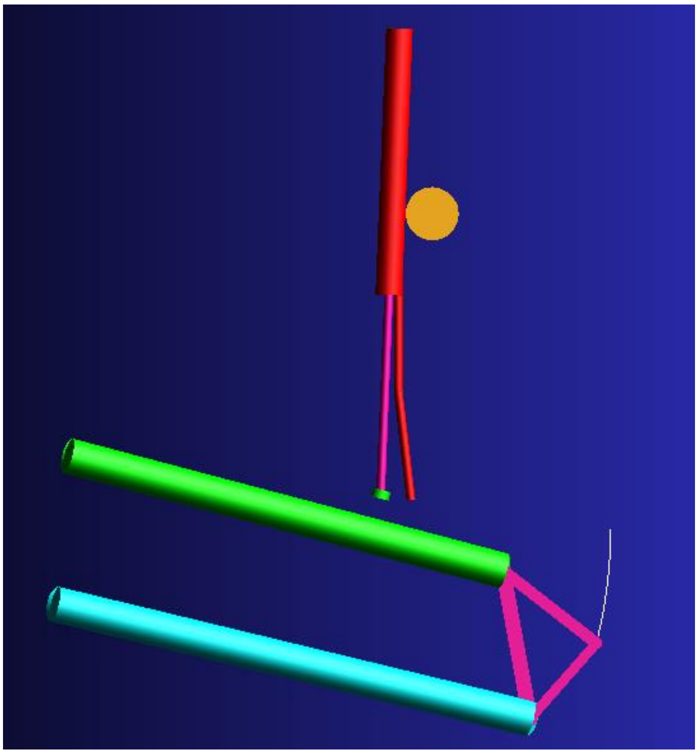


Figure 3.59 - Lifting Down Curvature of Kinematic System

The spline has been drawn by ADAMS from its natural location to whole lifting. As seen from Figure 3.59, the x-direction location change of the system is sufficient for wheel.

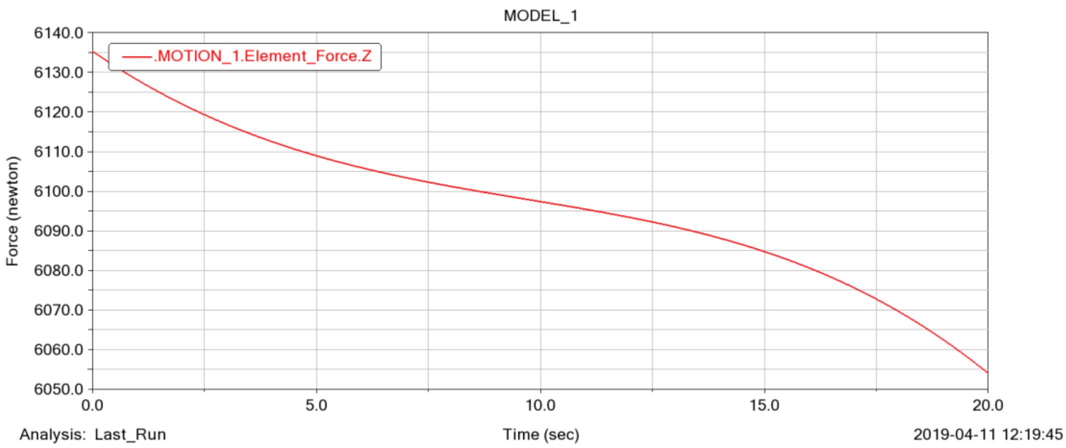


Figure 3.60 - Load Change on Single Teeth by Point of Mass

The usage of the whole weight did not change the load of the effect on pinion tooth at all. Starting from 6135 Newton has been changed to 6055 Newton in twenty seconds in Figure 3.60.

3.7. Stress Analysis by Finite Element Analysis

Hydraulic motor has not constant torque due to its working principle. According to its loading sensors, it can change the torque at that moment. Beside its starting torque value which is from zero, and it has proposed working torque scale as well. In the selected motor, 200 to 1000Nm torque selection is ready to use. According to proper analysing and interpolation process, 400Nm and 800Nm has been run by the point of mass. For the difference of kinematic analysis values, only rack and pinion z-directioned force has made upon them and it had been run as same as directly given torque.

In the design all of the models are taken from CATIA V6 3D program. For solid parts, necessary materials had been defined. As a formulation second order terms had been used. For the Hex dominant Method and sweep method had been used for the ease of the calculation.

For the better accuracy of the results, contact areas of the tooth between pinion and rack's meshes have been provided as fine mesh. [11]

Beside the identification of ANSYS properties on model, the general behaviour on design is reflected as deflection at middle of the bridge after the load application. For some node edges, errored stress values can be noticed. However, reading such changes one to another load is counted as negligible in these mentioned systems.

3.7.1. 400Nm Analysis of the system

After load application on the system, there are some deflection due to its point mass and torque. The displacement of the system is 0,12mm in the roller side in Figure 3.61.

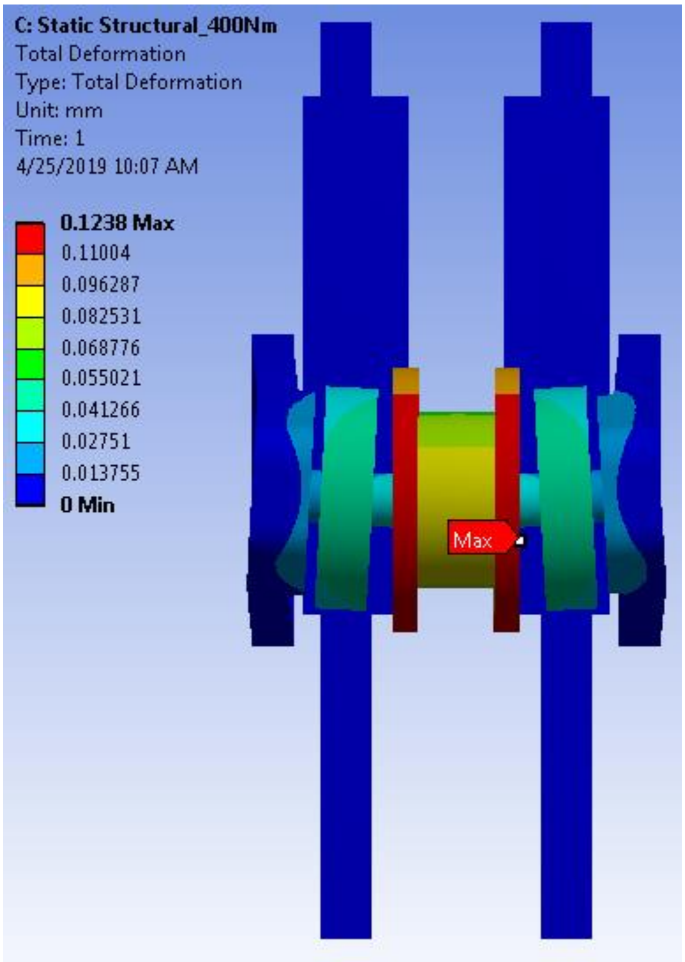


Figure 3.61 - Displacement of 400Nm Torque System

Two specific line is important for evaluating gear stresses. The first one is root fillet and the other one is line of action edge. As seen from the Figure 3.61 that average root stress becomes 170MPa and the line of action average stress becomes 220MPa. Those stress values are very low for both analytical and selected material results.

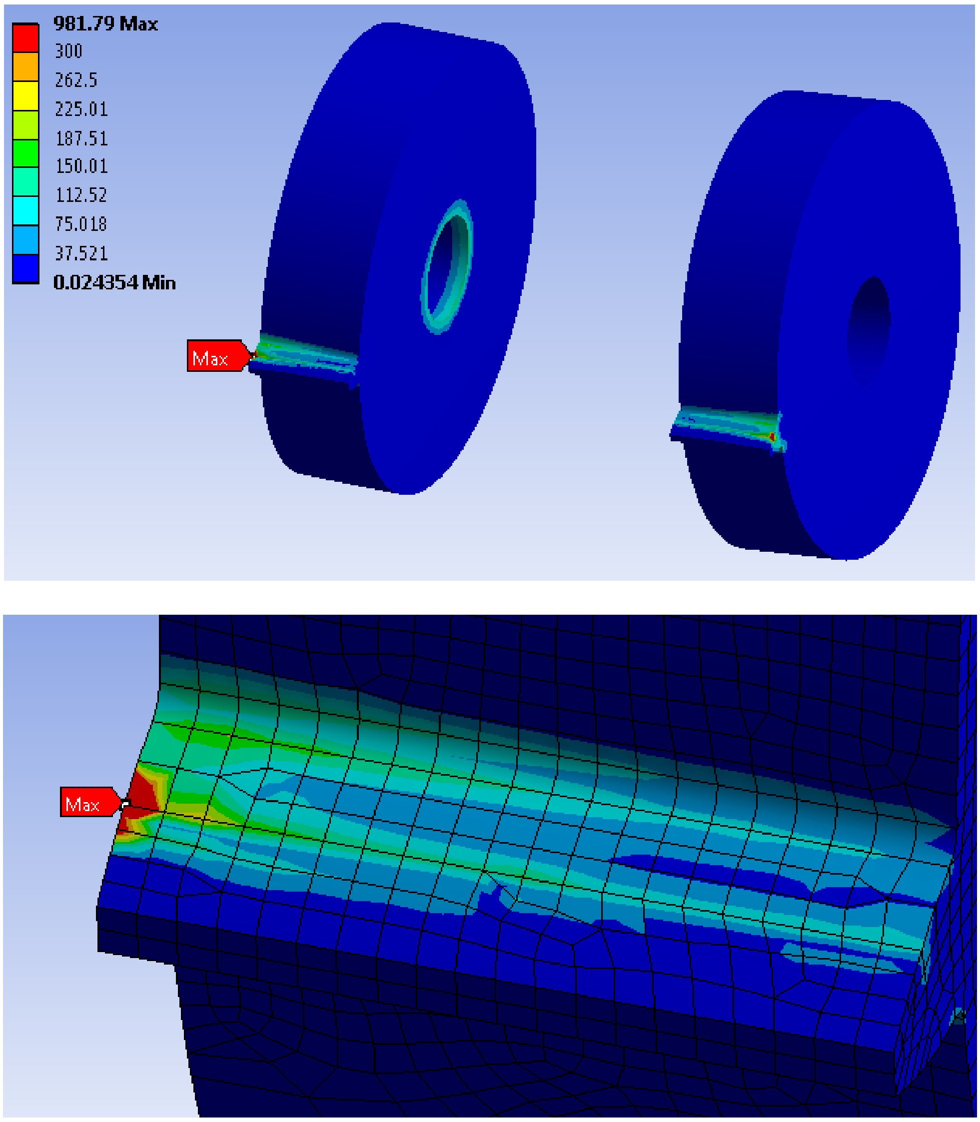


Figure 3.62 - Average Stress Distribution on Pinion Teeth

Average stress values are around 220MPa at line of contact area. Which means that around the notes 220MPa are result whether suitable or not.

For stress evaluation owing to deflection, the edge of the latest node has rise up to 950MPa. However on one single node there is approximately 700MPa difference. It needs to be merit. According to St. Vernant Principle; if the difference is getting bigger the source of the singularity is a way from the logiccal explanation so it makes the maximum difference “negligible”. Exaggeratedly, if the node size goes to infinite, the singularity goes squeeze itself on corner; in other word using additional area on them or making do not help to avoid that utopia. [30, 35, 36]

On the contrary, the rack stress has relatively lower than pinion has been proved in Figure 3.63. Because the shaped curvature of the rack is more linear shaped so the stresses of the rack is low.

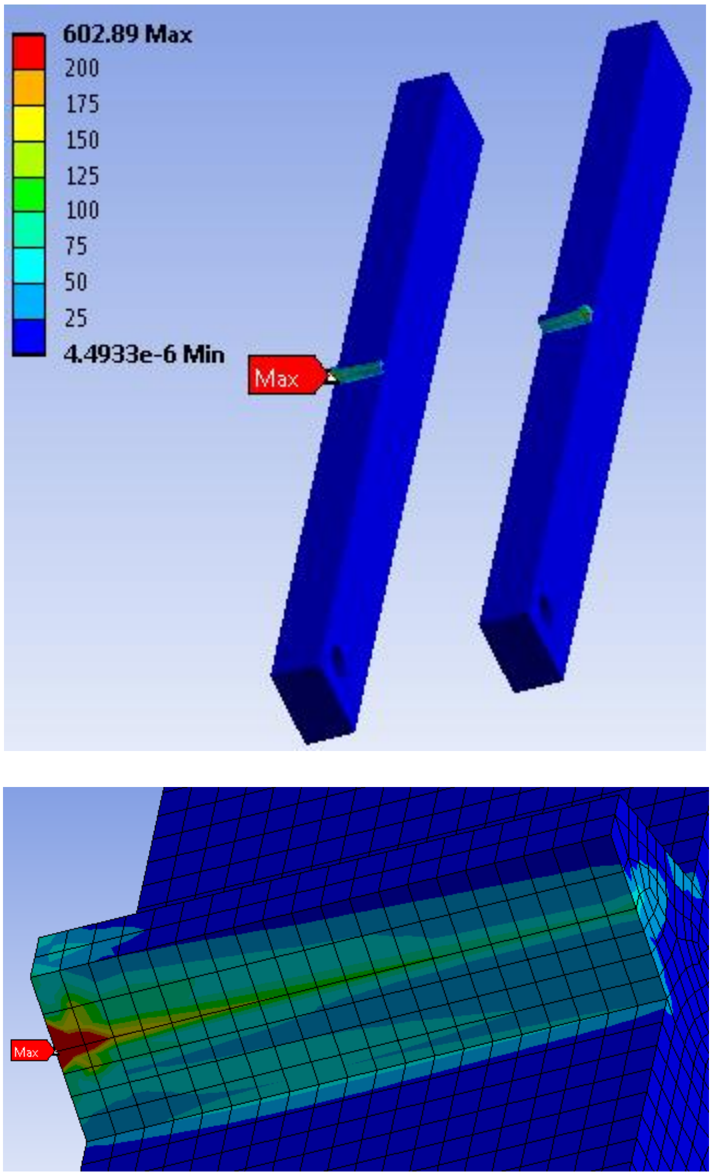


Figure 3.63 - Average Stress Distribution on Rack Teeth

Approximate stress distribution in root is 63MPa and on the line of contact approximate bending stress is 167MPa. It is very low whilst considering the pinion's stresses. So the importantancy mitigates itself for rack.

3.7.2. 800Nm Analysis of the System

Once torque value doubled the displacement of the gear doubled itself either. The displacement of the system is 0,24mm in the roller side again in Figure 3.64.

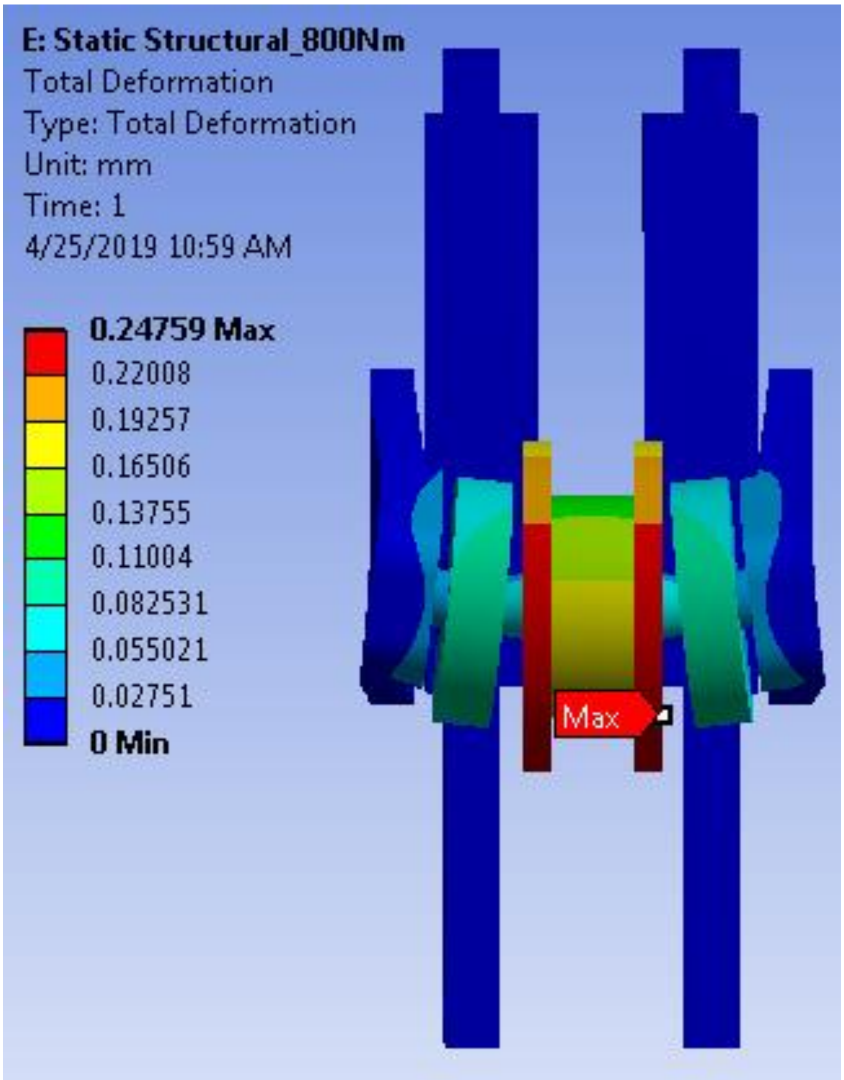


Figure 3.64 - Displacement of 800Nm Torque System

This time, as seen from the Figure 3.65 that average root stress becomes 330MPa and the line of action average stress becomes 450MPa. Those stress values are very low for both analytical and selected material results.

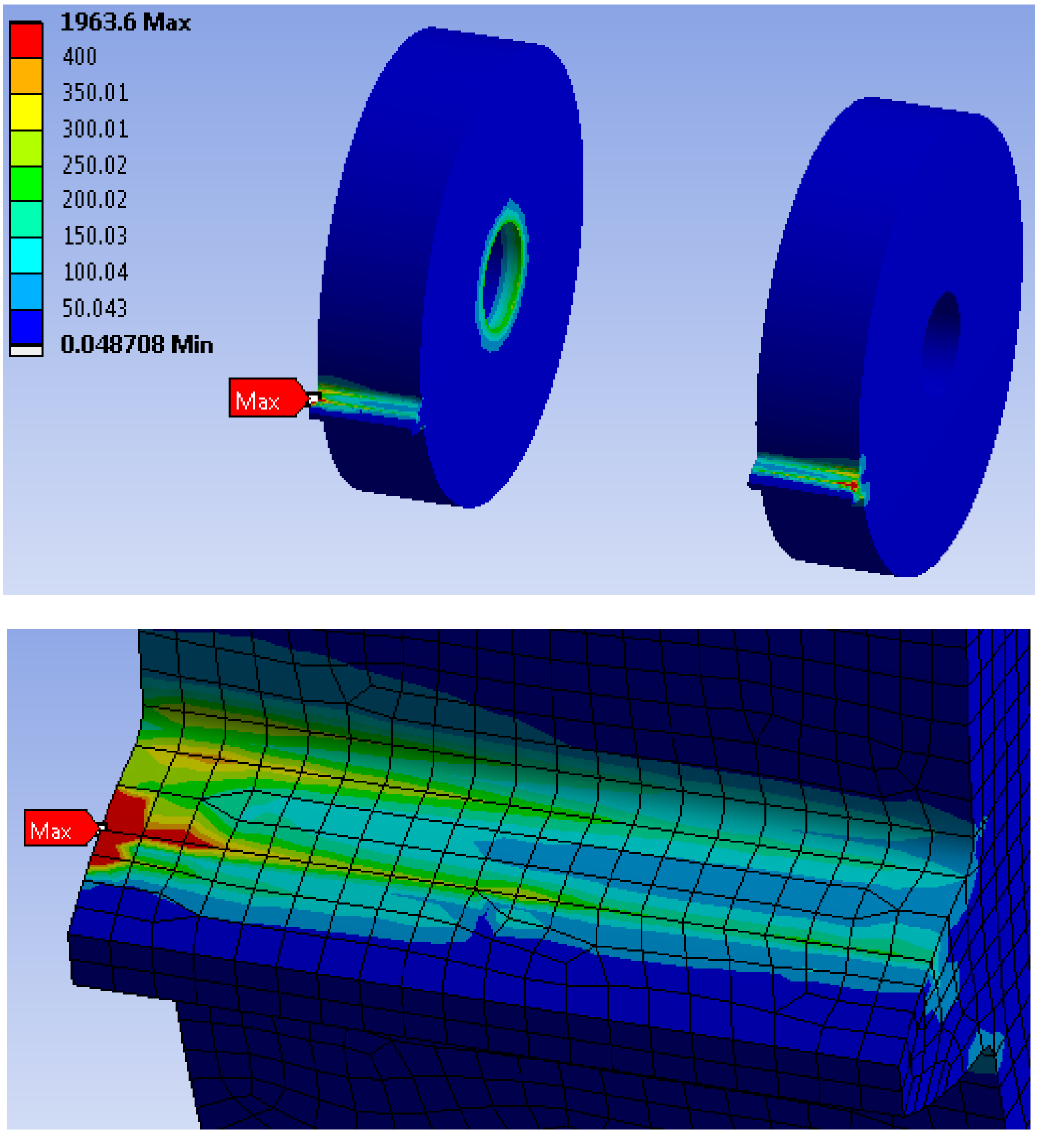


Figure 3.65 - Average Stress Distribution on Teeth

Average stress values are around 450MPa at line of contact area. Which means that around the notes 450MPa are result whether suitable or not.

It is the same principle with 400Nm singularity as 1963MPa. The rounding up the node size and changing add-on models can decrease the exaggeration.

On the contrary, the rack stress has relatively lower than pinion in Figure 3.66. Because the shaped curvature of the rack is more linear shaped so the stresses of the rack is low.

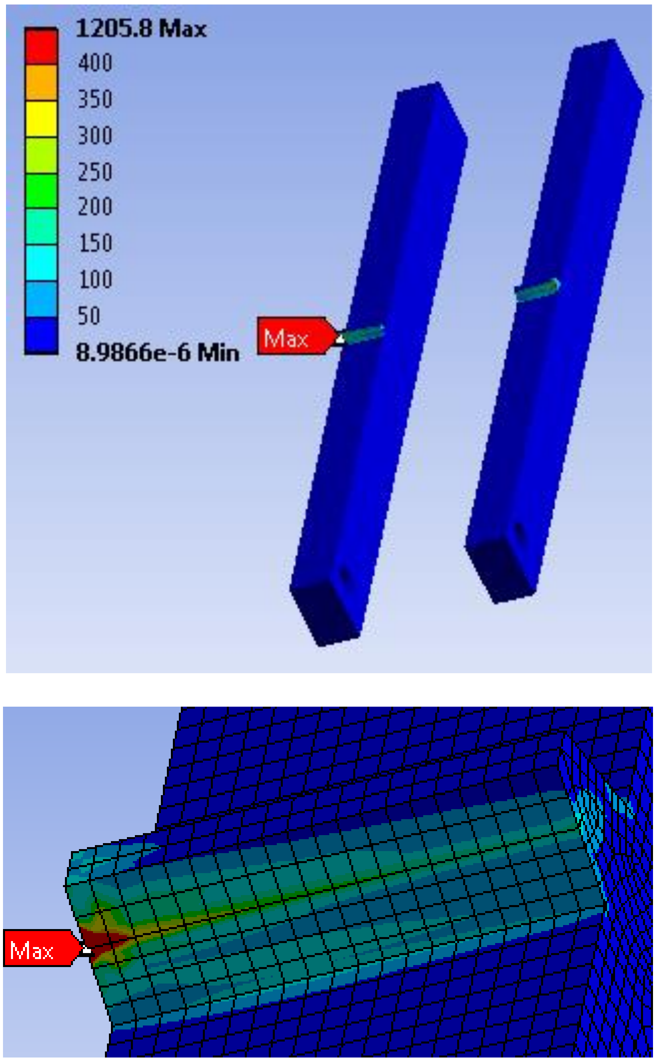


Figure 3.66 - Average Stress Distribution on Rack Teeth

Approximate stress distribution in root is 125MPa and on the line of contact approximate bending stress is 340MPa. It is very low whilst considering the pinion's stresses. It can be said that doubling the torque increases the stresses of the root and line of contact of rack almost double as well.

3.7.3. Kinematic Simulation Values on the System

In that stress analysis, only results had been used based on the ADAMS solver. Basically, z-direction force for both up and down motion has some values and the maximum values has been compromised for the evaluation of the system. In that anaysis, 6800Nm load value had been used for the system.

As shown in Figure 3.67, the displacement of only for the pinion is 0,02mm and the rack displacement is 0,008mm. From there the understanding is that whole system behaviour is reflecting on the undesired displacements and it may reflect the unwanted measurements on the mesh area.

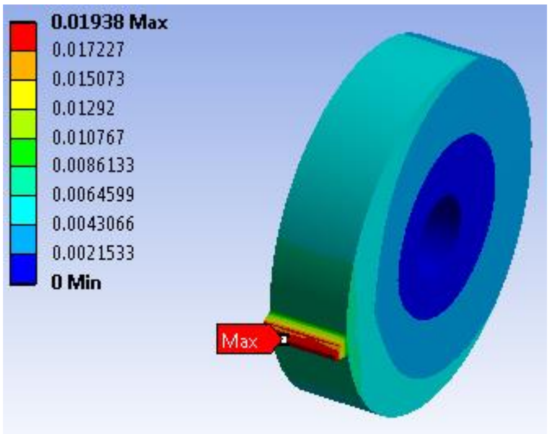


Figure 3.67 - Displacement of Kinematic Analysis

The stress evaluation of the pinion is measured as average for 262MPa in the root fillet region and 353MPa in the line of action region. At the value, the system torque working area must be between 400 to 800Nm. However, due to its varying principle the exact amount of the stress values can be interpolated by mentioned values just made average in Figure 3.68.

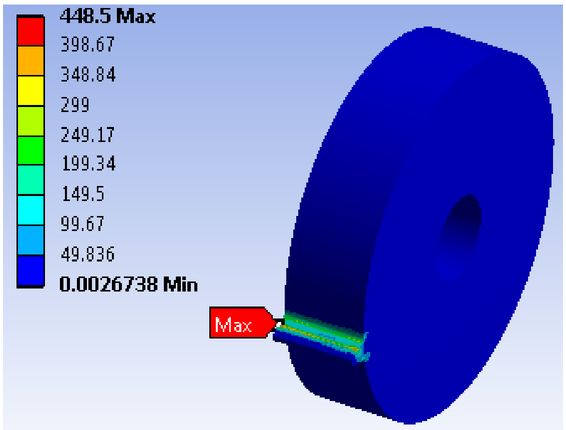


Figure 3.68 - Average Stress Distribution on Teeth

As rack side, due to its accurate shape the stress results are very low according to pinion. The rack is measured as average for 135MPa in the root fillet region and 265MPa in the line of action region in Figure 3.69.

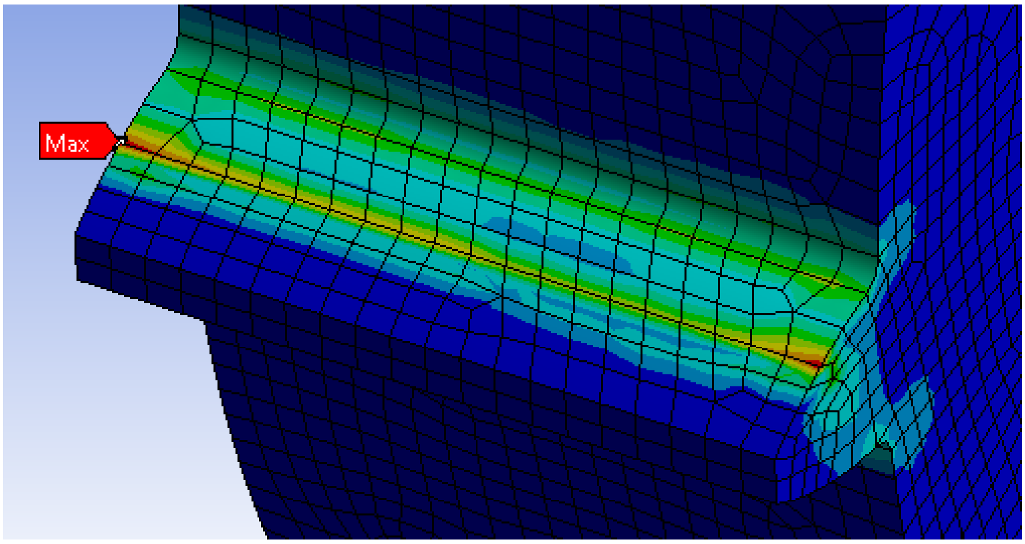


Figure 3.69 - Average Stress Distribution on Teeth

As explained before, the design factor of safety has been defined as 1,15 in order to assume previous tests.

Comparison between Torque of 400Nm, Torque of 800Nm, Analytical Solution and Kinematic solutions are being written on graph. The whole system change has been done between 400Nm and 800Nm, so average of 400Nm-800Nm of torque has been shown individually in Figure 3.70.

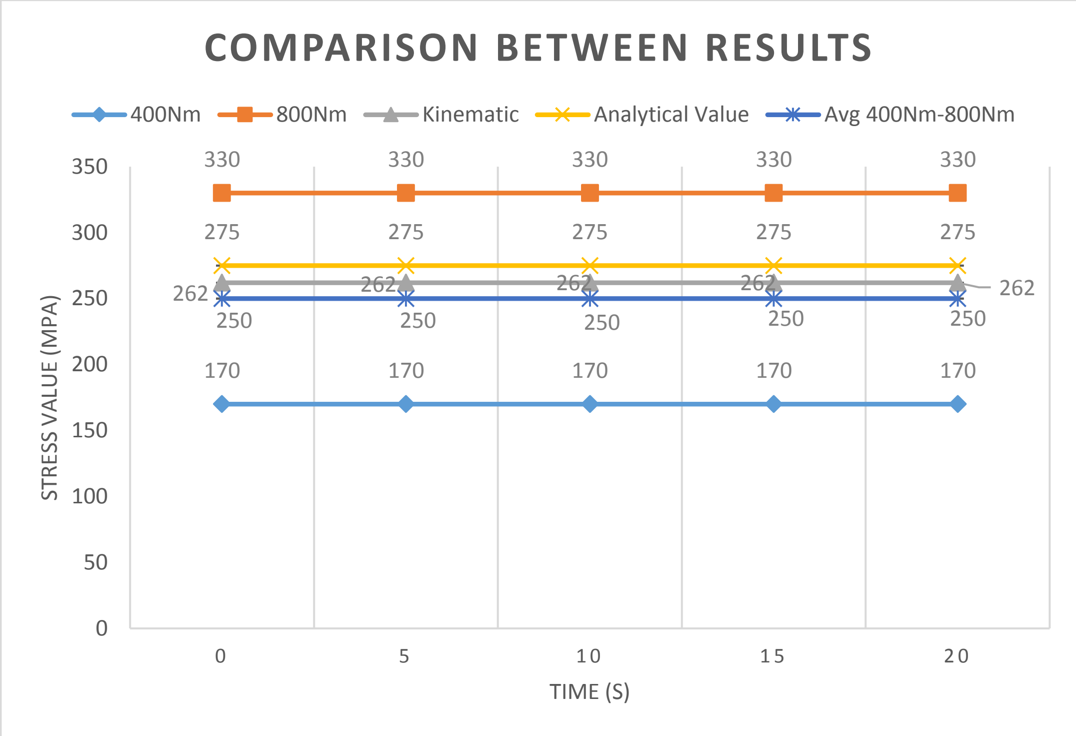


Figure 3.70 - Comparison Between Analytical Solution, 400Nm Solution, 800Nm Solution, Average of 400Nm-800Nm Solution and Kinematic Solution

3.8. Design Verification

Computational Fluid Dynamics (CFD) is essential a part design includes products that involve the liquids, gases and other types of flows. CFD is very good at solvinf turbulent , particulate elements and heat transfer. Optimal solving mechanism consists of choosing boundary conditions that make ensure how particulates move accurate. Due to background mathematical equations are hard, optimum result finding approach has been used in the use of required software programs. The most pioneer software program can be counted as ANSYS Fluent. With the help of several iterations, the add-on structure such as modelling, mesh, topology, material properties and contacts can be defined to program. According to modules, the necessary solver can calculate vibration, heat, optics etc.

In that study, FLUENT has been selected in order to swim the whole vehicle with the difference of lifting up and down the whole system roughly. The importance of the roughly modeled design has been come from the running time for solve the system. Solely, whole system has been done for seeing the difference between them.

In FLUENT, pressure based solver has been defined. The vehicle has a capability to swim around 1.1m/s according to specifications. The whole weight is 13.000 kg.

The whole calculations and modelling in FLUENT has been considered in that way. In simulation modelling, due to restrictions about workstation capability some inputs has been inserted as roughly. It helps to calculate in short time in order to see whether it is a valid soluion or not. Moreover, it has been illustrated in FLUENT as around the vehicle there is constant speed water currency and it makes effect in front the vehicle.

3.8.1. Lifting Up the Whole Wheels at Vehicle

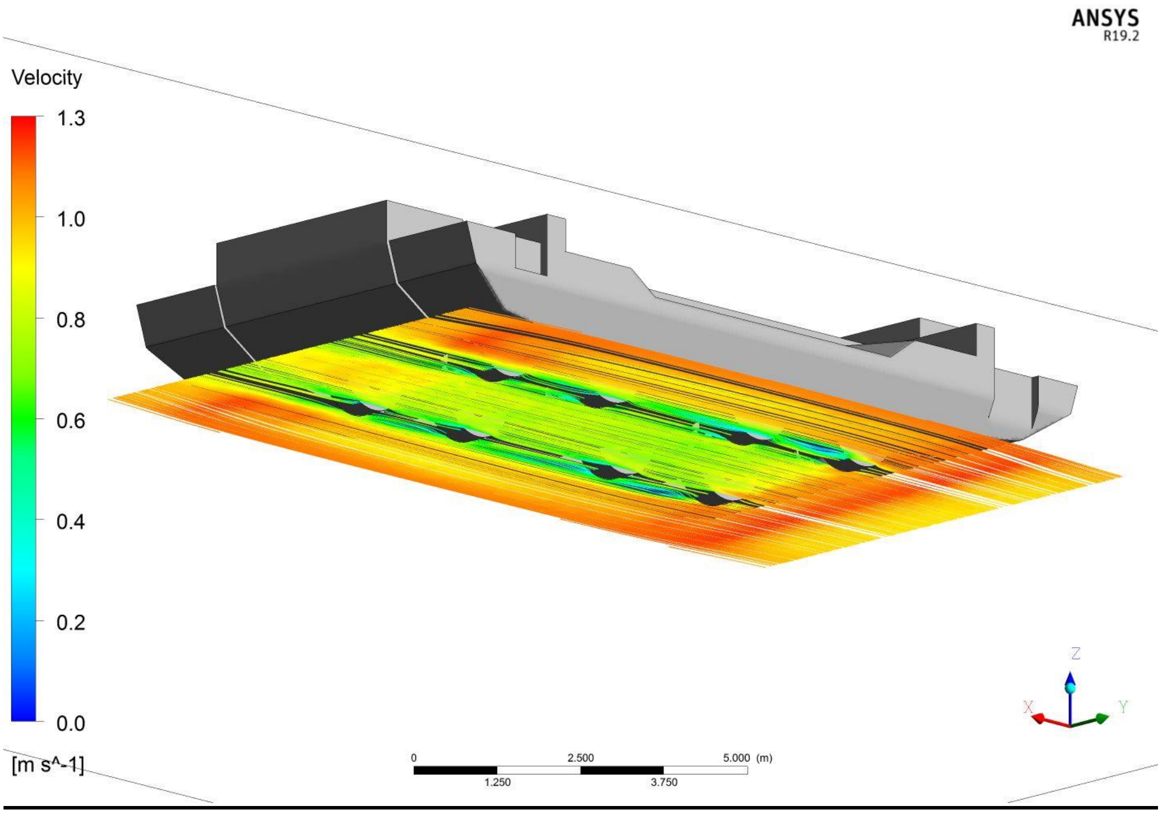


Figure 3.71 – Isometric Streamlines of Mechanism Lift Up

The streamline plane area has drawn in Figure 3.71, 150mm below the bottom of the vehicle. The reason of 150mm adjustment is that it is relatively best area to examine both lifting up force exposure and lifting down force exposure.

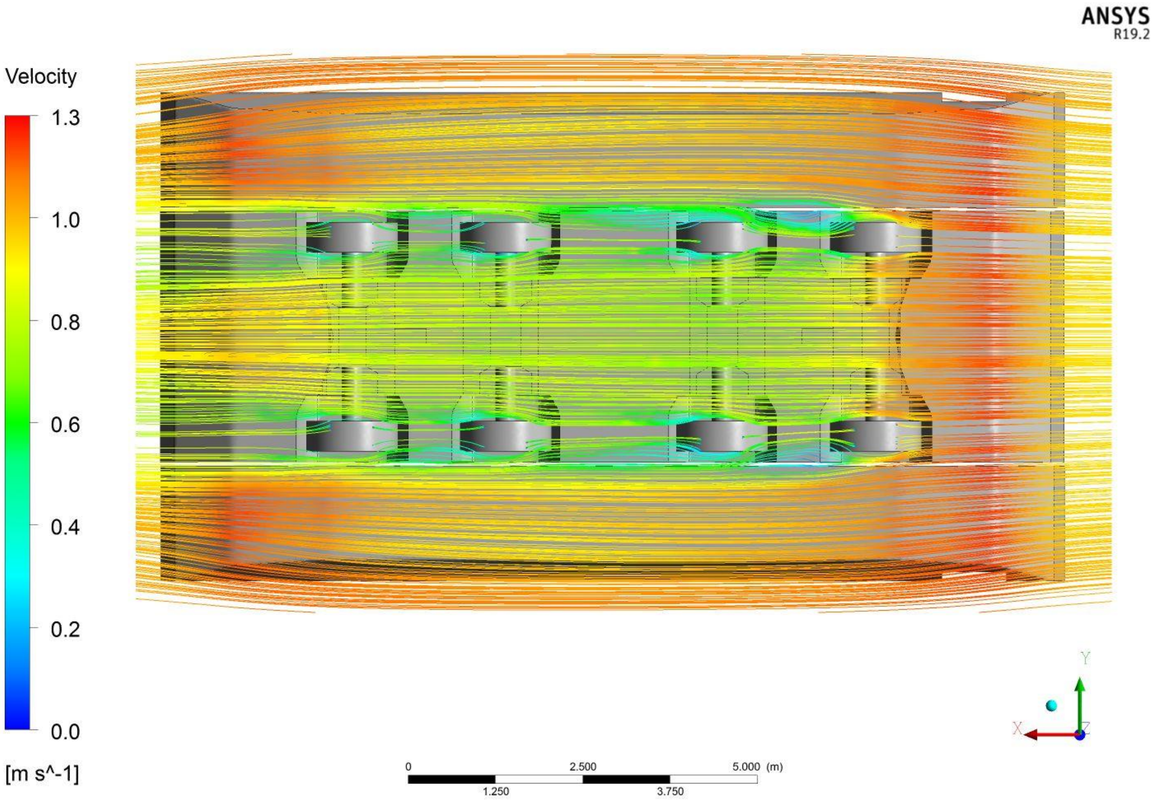


Figure 3.72 - Bottom Streamlines of Mechanism Lift Up

Velocity streamlines are changing from red color to yellow color from beginning to end. That demonstration in Figure 3.72 is very common when an obstruction is occured in front of the currency.

According to wheels that goes inside the water, pressure is important to examine. As seen from below, from the left side of the vehicle four wheels' pressure change can be seen.

3.8.2. Lifting Down the Whole Wheels at Vehicle

Different than lifting up, this time velocity increases to 1,26m/s. The obstruction area of wheel itself has effect to see that result.

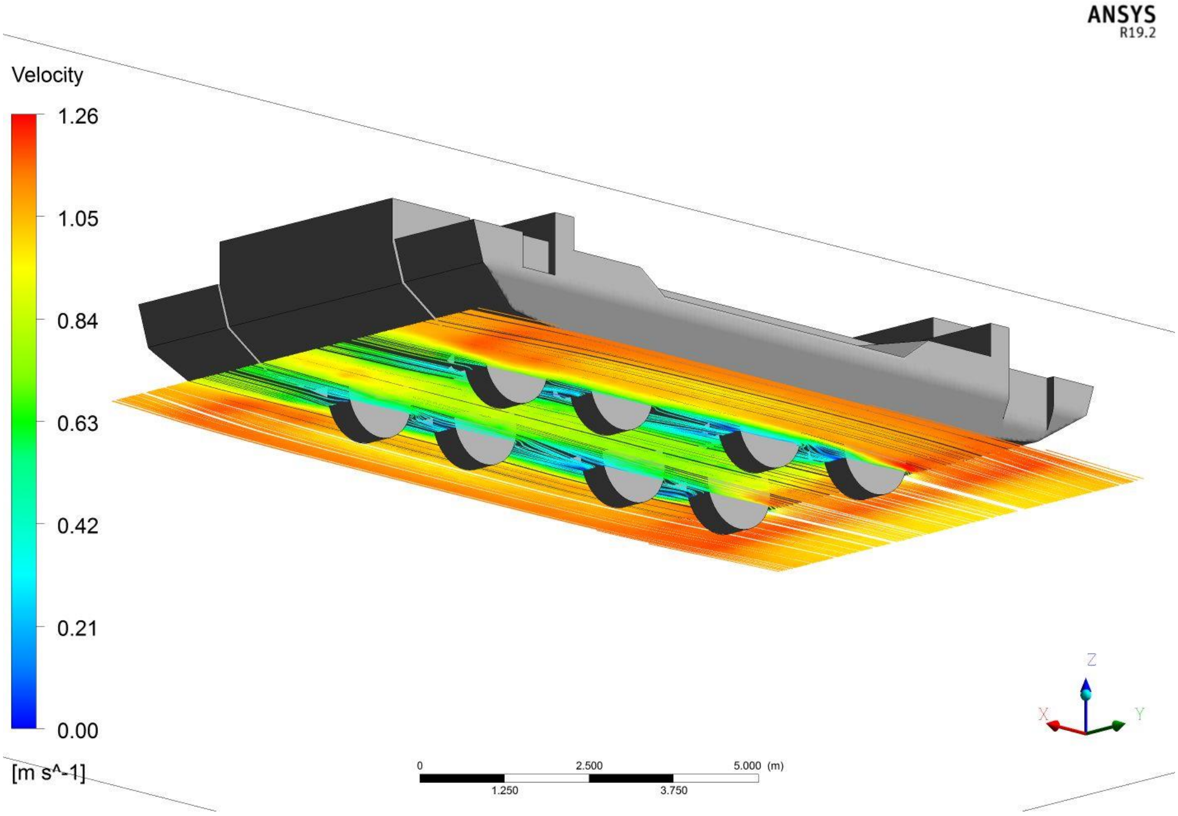


Figure 3.73 - Isometric Streamlines of Mechanism Lift Down

The streamline of the lift down mechanism has inevitably distinctive when fluctation has been seen in Figure 3.73 and Figure 3.74. The extreme change of currency shows that with the obstruction wheel by wheel vehicle body affected. For the bridge mission and other fortification objectives, the balance of the vehicle can tremble and lead up to undesired circumstances like roll over or sink.

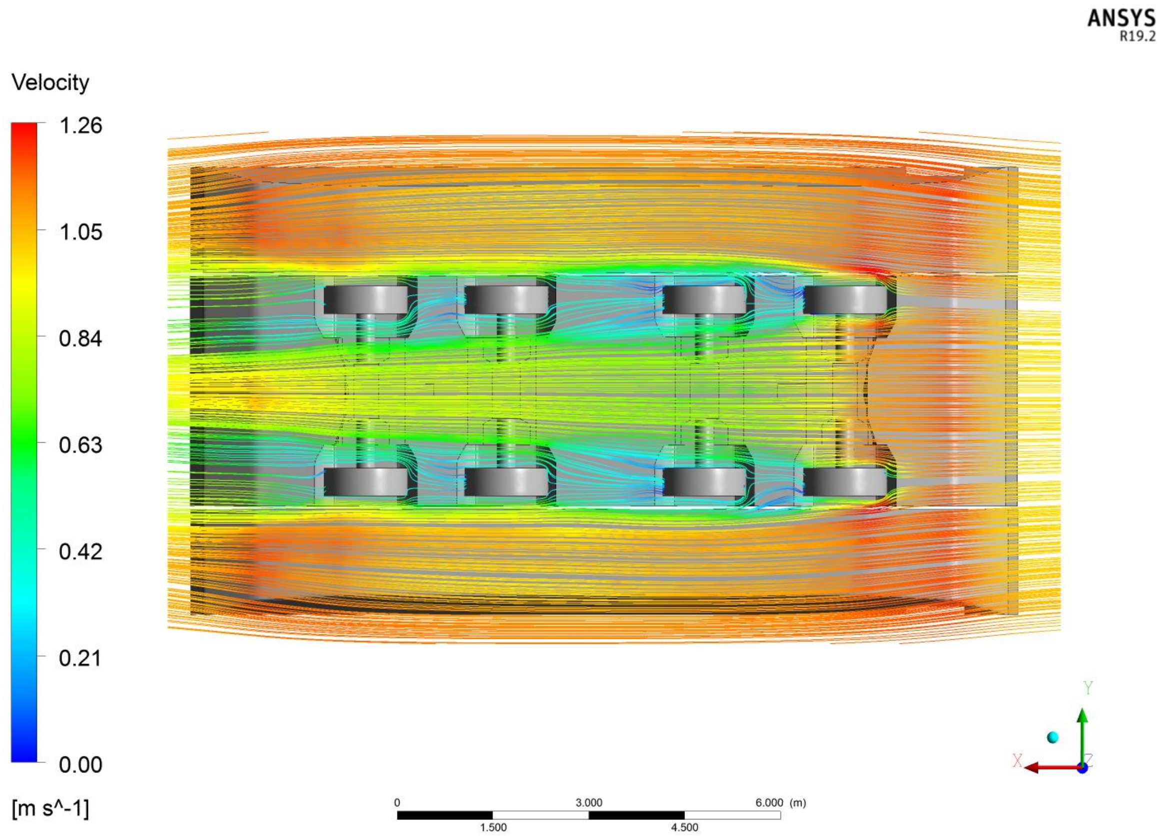


Figure 3.74 - Bottom Streamlines of Mechanism Lift Down

3.8.3. Comparison of Proposed Design Study

In order to understand the real effect of the current and proposed design, forces of X,Y and Z must be evaluated and put it in a compared graph.

Forces of X is major effect to understand the change between up and down. Because, 1.1m/s velocity of water currency is given. It is changed by the height with almost the same multiplier as 2,5 as defined in Figure 3.75. Because the effective force of the part is evaluated with density, height and gravitational acceleration. Only difference in here is height of the two systems.

The countervailing front vehicle has to overcome the maximum forces in any case. The following second and third wheels has almost same force distribution. On the other hand for the last wheel forces increased for both cases. According to flow characteristic, it must be in turbulence or transient area after linear flow. It can not be linear because in our ordinary world very few occurences have linear characteristics and such cases like swimming etc. must be objected as post linear flow case. The escalating trend of the graph may be vortex effect wheel by wheel and in the end on the last wheel it might broaden.

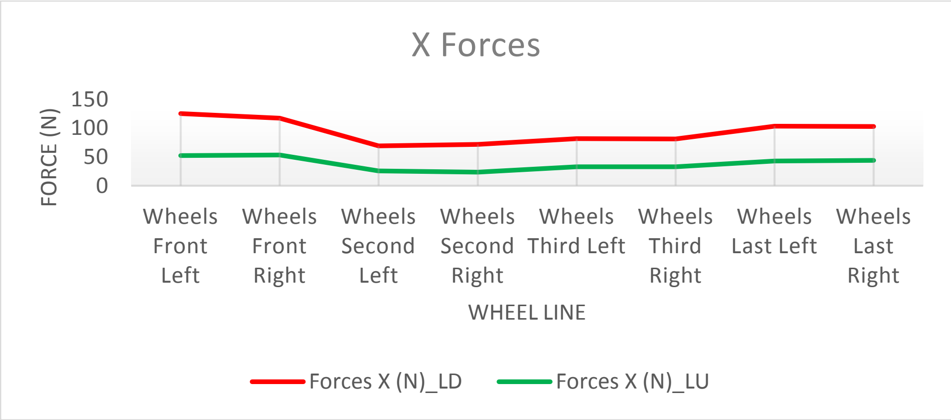


Figure 3.75 - X Forces Comparison of Wheels

Forces of Y has altering effect of side velocity changes. In that study, only estimaed values has been considered the currency effect from the nose area. Then the fluctations of the wheels have not that difference. Only difference comes from the right side of the vehicle has effect of negative value but on the contrary, left side of the wheels have the opposite value than right side. So it may alter the direction of the forces for left and right sides individually just demonstrated as in Figure 3.76.

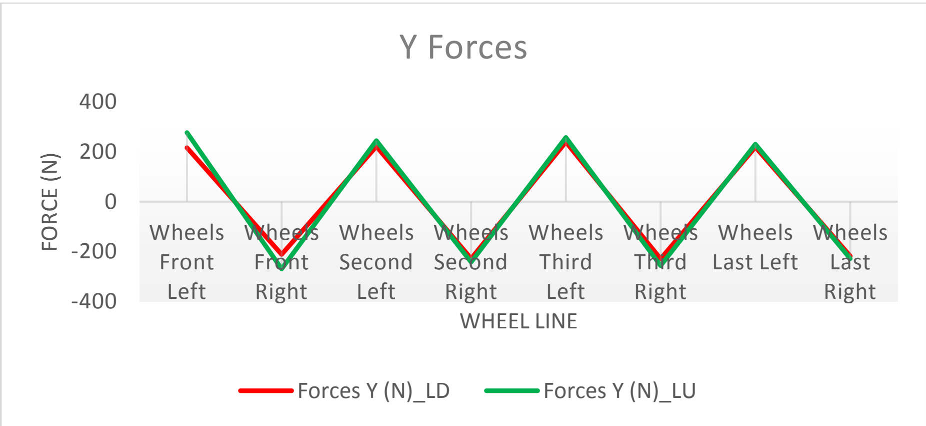


Figure 3.76 - Y Forces Comparison of Wheels

Forces of Z tends to be changed only the sinking height of the wheels itself as well as X. The only difference may be vary from the sinking volume of the wheel changes. So the volume of the lifting the mechanism up and down is stable for both wheels respectively which has been sketcehd in Figure 3.77. Therefore only portion has been changed due to height of the sinking area of the wheel.

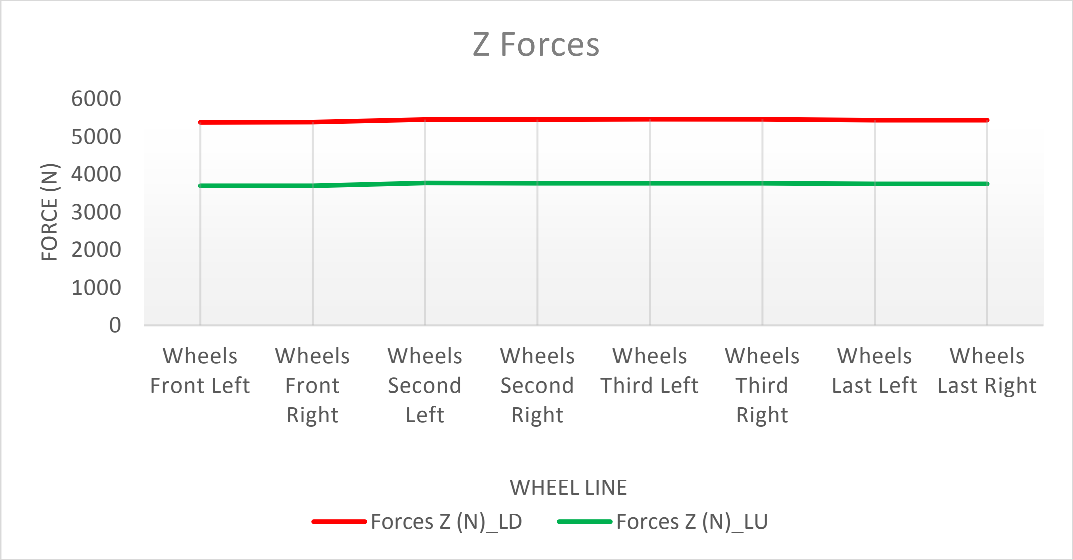
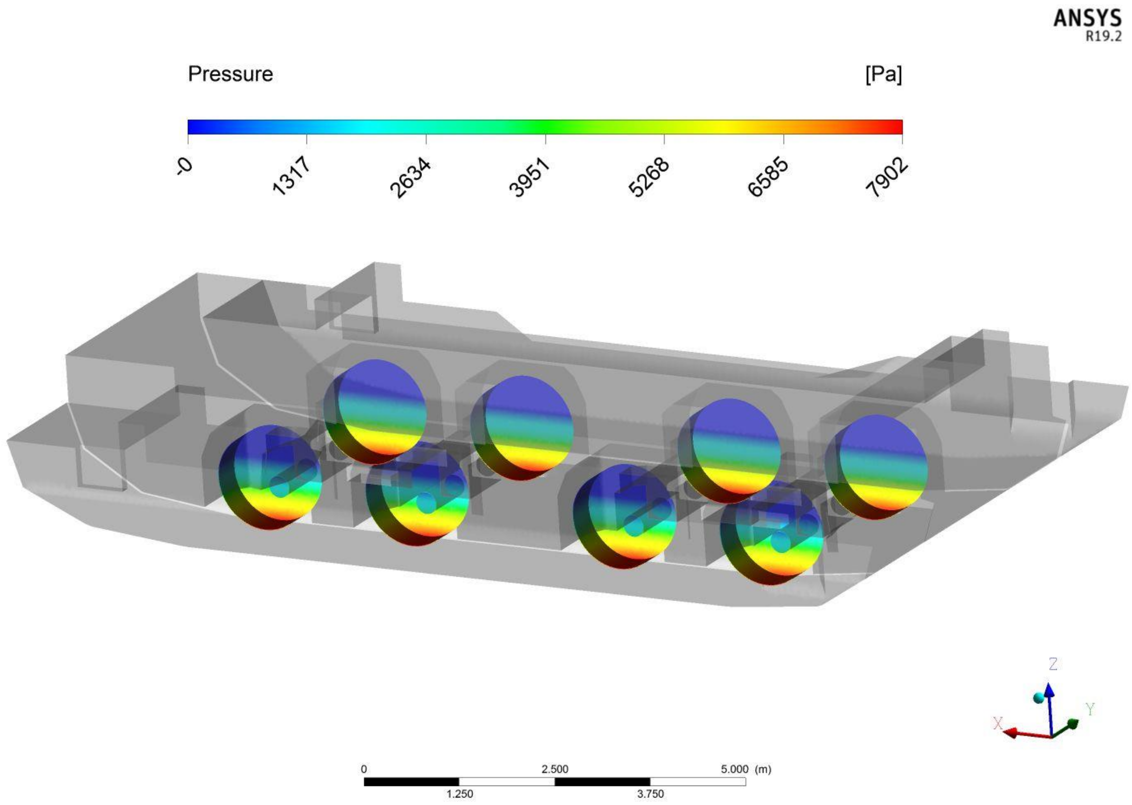


Figure 3.77 - Z Forces Comparison of Wheels

Briefly, the improvement of the system provides a force reducing. Hence, vehicle body goes through inside the water without any level-changing distraction or effect. In order to achieve the military mission, every parameter is very important.

For lifting up the system the X Forces are distributed relatively lower than lifting down the mechanism such as in Figure 3.78.



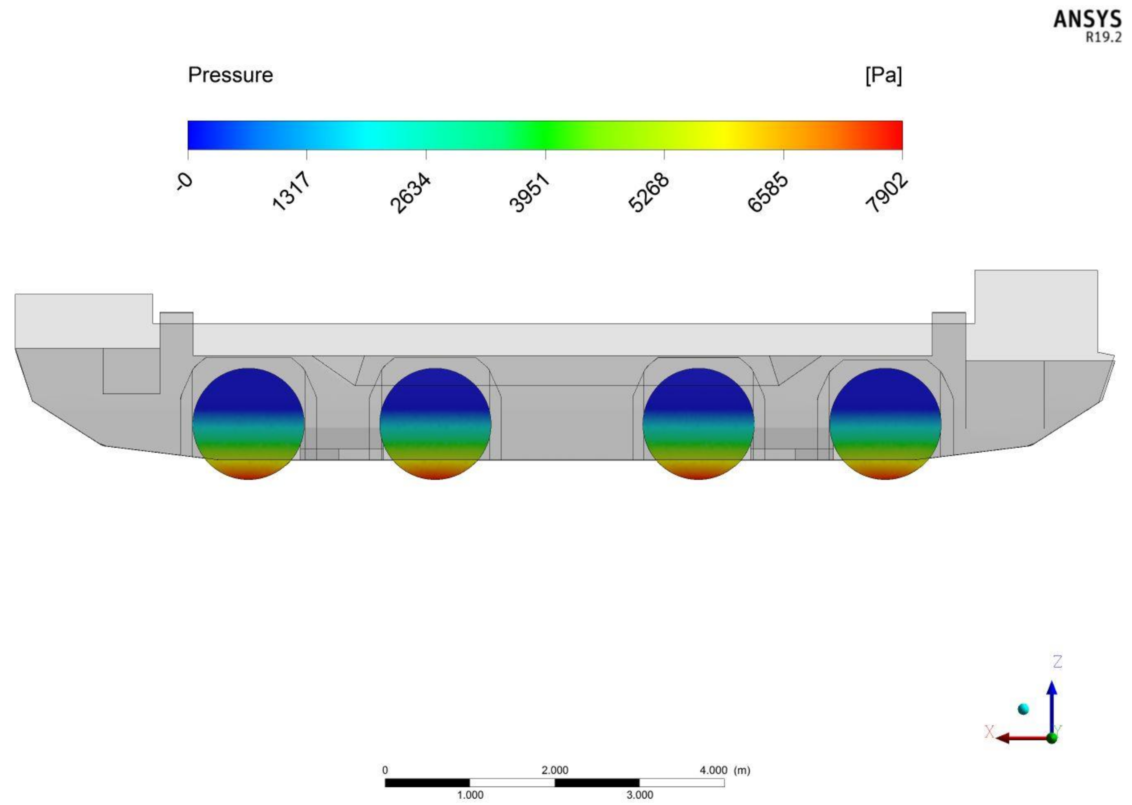


Figure 3.78 - Contours of Mechanism Lift Up

For lifting down the forces of X are illustrated as in Figure 3.79.

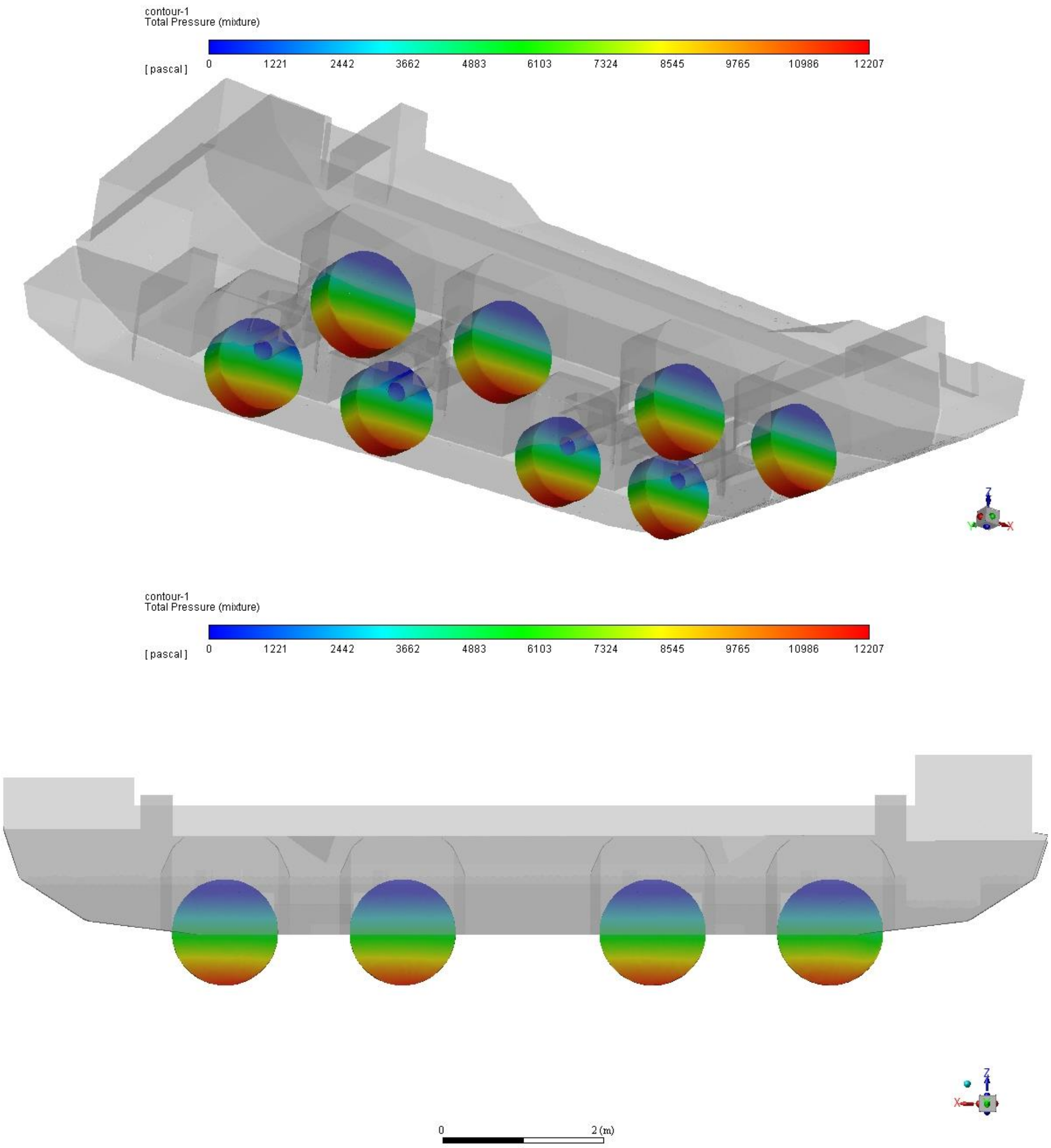


Figure 3.79 - Contours of Mechanism Lift Down

4. RESULTS

The provided results from the simulation of kinematics from ADAMS, structural analysis from ANSYS and swimming improvement by FLUENT showed that though fluctation of the hydraulic output the model used in CATIA V6 yielded same values of the bending stresses, especially at the average value of minimum and maximum torque outputs for the exact rack&pinion gear pairs. That mentioned improvements and related swimming simulations depends on the vehicle performance behaviour and naturally the achievement of the mission. In other words, it can be useful when any same mechanism can be considerable for amphibious vehicles.

In order to clear the untouched soil which controls by enemy lines may be full of mines or improvise explosive device (IED), the fortification vehicles are very important to eliminate stated threads. To avoid those potential attacks, a robust designed vehicles are inevitably crucial.

Firstly, to understand the goodness of the system efficiency and reliability, the analytical calculations are being done by useful in terms of compare the analysis solutions. The mathematical calculation approachs consist of the most common used Lewis Formula and AGMA Standards. The reason for the usage of Lewis and AGMA Standards are being referenced by so examiner and researcher.

Secondly, three different anaylsis steps are being implemented by ADAMS as kinematic wise, ANSYS as stress consideration wise and FLUENT as fluid dynamic wise. For ADAMS, the mechanism behaviour and success has been observed then on the grounds of accuracy the solutions of ANSYS, the stress analysis has been done according to minimum and maximum torque of the system. As design comparison with or without the lifting up mechanism, FLUENT pressure based solver has ben deployed in order to release the swimming simulation of vehicle.

The studies have shown that the developed design makes 2,5 time less unbalanced body behaviour than the untouched system. It gaines an impression by crew health, vehicle health and mission success. Using the common or easy manufactured parts can help to make more profit related to cost. Hence, due to cost effectiveness, the system design can be achieveable in terms of project angle.

The study would be used for the amphibious vehicle which has air spring without motion system or has no air spring at all. In modern military technology, multi-tasking vehicle are increasing their importance drastically. Tho solve such a solution for amphibious vehicle is very crucial.

As future study, in order to make better design, the bellow can be inserted in the system to prevent dirts or for better steering the system flat bearing can put inside the roller. Those will make tremendous difference however the difficulty of the mounting those parts without any hassles is hard task to do.

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