



ADDIS ABABA UNIVERSITY
ADDIS ABABA INSTITUTE OF TECHNOLOGY
SCHOOL OF MULTIDISCIPLINARY ENGINEERING
GRADUATE PROGRAM IN RAILWAY ENGINEERING

**STRUCTURAL OPTIMIZATION OF RAILWAYS FREIGHT
WAGON UNDER FRAME**

By

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in partial fulfillment of the requirements for the Degree of Masters of Science in
Mechanical Engineering**

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Abstract

Nowadays, the railway transport infrastructure in Ethiopia is one of the major issues in relation to the development of the country. To strengthen this infrastructure, related researches must be conducted at the beginning or during the construction of the sector.

This paper is mainly concerned on optimizing a railway flat wagon underframe structure that has been subjected to a multi objective optimization to improve stiffness to weight ratio.

In order to perform structural design optimization with a finite element method (FEM), a 3D model of the freight wagon underframe has been prepared by using a 3D modeling software CATIA V5R20 software. The discret FEM model has been imported in to ANSYS 14.5 workbench, and optimization has been done on the frame structure based on the initial parameters by using multi objective optimization techniques (ANSYS).

The result has been showed in the form of graphs which relates the input and output parameters with the design variable. After graphical interpretation, structural analysis is carried on by using ANSYS; just to check how far the input and output parameters are improved at the highest loading conditions.

Observing the optimization results and static simulation effects of the existing model at the highest loading conditions, it has been conclude that the center sill is a shell element (note a solid element) and the optimum plate thickness for the center sill is 23mm. The final product of the optimization process is a low weight underframe structure which reduces 18.02% of the overall weight of the wagon and 9.01% to 12.614% of reduction in power consumption. As a recommendation it is suggested that dynamic effects on the underframe should be done for safety and is recommended as future work.

Key Words: underframe, stiffness to weight ratio, multi objective design optimization, optimization algorithms.

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List of Abbreviations

AAiT	Addis Ababa institute of Technology
APDL	ANSYS Parametric Design Language
DOE	Design Optimization Experiment
EN	European Normalization/ standard
ERC	Ethiopian Railways Corporation
FEM	Finite Element Method
F_{\max}	Maximum operational load when the vehicle is stationary
F_v	Vertical load
g	gravitational acceleration
K_b	Bending stiffness
LP	Linear Programing
l_b	Lower boundary design variable
MDO	Multidisciplinary Design Optimization
MOO	Multi Objective Optimization
m_1	Design mass of vehicle body
m_2	Design mass of one bogie or running gear
m_3	Normal design pay load
m	Material mass of the under frame
NLP	None Linear Programing
RBF	Radial Basis Function Response Surface Model
S235	Structural steel grade 235

t_i	Thickness of the n^{th} plates
u_b	Upper boundary of the design variables
V	Material volume of the under frame
ρ	Density of structural steel
σ	Stress in the underframe
δ	Deflection of center sill

Statement of Original Authorship

I certify that the research work titled “Structural design optimization of railway freight wagon underframe” is my own work; and the work contained in this thesis has not been previously submitted to meet the requirements for an award at this or any other higher education institution. To the best of my knowledge and belief, the thesis contains no material previously published or written by another person except where due reference is made.

Name: Tadesse Yimer Signature: _____ Date: _____

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Chapter One

1. Introduction to Rail Vehicle Structural Design Optimization

1.1. Motivation

Today, weight and stiffness are possibly the most important parameters in railway vehicles underframe design considerations. When designing a wagon underframe or chassis for best performance, a high stiffness to weight ratio is often required [2]. Rigidity in bending and torsion, efficient load absorption and low weight are also key parameters to a strong underframe performance evaluation.

In railway vehicles structural weight has become gradually important as energy consumption standards have increased with increasing structural weight. During the conceptual design stage (when changes to the design are easiest to implement and have lower impact on overall project cost), structural weight and other characteristics related to it are usually unknown since detailed railway vehicle information is unavailable at this early stage [1].

The work done here focuses on the stiffness and weight of the underframe structure during the conceptual stage and seeks to develop an optimized preliminary rail freight wagon underframe structure that is used as an input for the detail design activities where every aspect of the freight wagon component design can be considered.

1.2. Background of the Study

1.2.1. Rail Freight Transport

Rail freight transportation industries play a major role in the economy of modern industrialized and developing countries. It accounts for a significant part of the final cost of products and represents an important component of the national expenditures of any country [28].

To satisfy the need for a high standard of living and economic success of the country, rail freight transportation is a vital component of the economy of all modern industrialized and some developing countries. Whether it is raw materials for manufacturing, fuel for electricity generation or different vehicles and machineries, long distance and medium distance bulk transport items and import-export commodities of the country depend on rail freight transport system.

Rail freight transport must achieve high performance levels in terms of economic efficiency and quality of service, and support production, trade, and consumption activities by ensuring the efficient movement and timely availability of raw materials and finished goods. Besides this rail freight has a series of advantages on freight transportation including being high capable, low cost, high efficient, and almost zero carbon emission [23].

1.2.2. Types of Freight Wagons

Freight railway transport is categorized in different types according to the types of goods being loaded or transported [11]. These are:

- Open wagons
- Covered wagons
- Temperature controlled wagon
- Two-axle flat wagons
- Composite flat wagons
- Bogie flat wagons
- Flat wagons with separate axles
- Wagons with opening roofs
- Special wagons
- Tank wagon

1.2.3. Rail Freight Vehicle Underframe

Underframe (chassis) is the backbone of a rail freight vehicle and is tasked at holding all the essential components of the rail vehicle. It should be designed to carry the payload, weights of other structural components mounted on it and instantaneous loads like traction and braking loads. The freight vehicle underframe has to withstand and give adequate performance under all these load conditions and its stiffness and natural frequencies vary according to the load distribution on it [11, 23, and 28]. The instantaneous requirement of

the underframe is strength, other performance requirements like frequency and bending are considered for underframe. To sustain various loads under different working conditions it should be strong in design. Moreover the underframe should be stiff and strong enough to resist severe twisting and bending moments to which it is subjected to.

The underframe a rail vehicle in general is composed of the Centre sill, upper side beam, bolster beam, end beam, large transverse beam, small transverse beam and steel longitudinal beam and floor [1, 23]. The long side members of the frame were called sole bars and the shorter end pieces were called headstocks. The bogies and brakes are fixed to the sole bars and the buffers and draw gear (couplings) are fixed to the headstocks.

The main underframe of a freight rail vehicle generally consists of four outer longitudinal member's sole bars with various reinforcing gusset plates and bars, and the two head stocks. The gusset plates protect the under frame against diagonal deflection and help in absorbing and distributing the buffing loads over different members. As already mentioned, the gusset plates and knees are provided at critical locations to impart additional strength to the joints. The whole structure is so designed that various loads are uniformly distributed and no single member has to bear excessive load than designed for.

Welding is generally used for joining the underframe members; but in earlier wagons, riveting has been also used for joining some members of the underframe. In the case of bogie wagons, the underframe has comparatively stronger cross members, known as bolsters, for fitting the upper center pivot casting, which rests on the bogie pivot.

1.3. Problem Statement

Design optimization of a railway vehicle underframe structure is an important aspect in the overall rail vehicle design process. The underframe structure is important to ensure that the weight of various components and loads applied during rail vehicle operation are sufficiently supported without too much considerable deflection. The underframe should be rigid and strong so that it can also resist shocks, twists, stresses and vibrations to which

it is subjected while the vehicle is moving on road as well as it is stationary. A vehicle structure should accomplish these two objectives while maintaining a low weight.

Structural optimization is a must duty within the rail vehicle industry, where the mass is usually minimized subject to a number of structural performance constraints. Structural weight optimization has also become increasingly important because energy consumption standards have increased with increasing structural weight.

This research has been done to accomplishing the following problems:

- How do we analyze the static response due to maximum loads on the freight wagon underframe? We can determine the static response of the freight wagon underframe by using finite element analysis software.
- Why do we optimize railway freight wagon underframe? Since we face a problem of material weight with the preliminary design, we have to compromise it by optimizing the frame structure and considering the effect of structural size (thickness) in the static analysis on the rail freight vehicle body.
- What do we optimize from the designed model of the freight wagon underframe to improve its weight? We can optimize the wagon underframe structural weight by using a multi objective optimization technique; by considering the center sill not as solid element, it is possible to improve the weight, factor of safety, and/ or other suspected structural properties.
- How other researchers are doing structural design optimization? By referring literatures, and different country (European and Chinese) standards as well to fill the gaps.

1.4. Objective of the Research

1.4.1. General Objective

General objective of the study is to find the set of parameters that minimizes weight of the wagon underframe without exceeding the allowable stress of structural steel (S235); that is

optimizing the underframe structural weight and bending stiffness of the existing Chinese standard model for Ethiopian Railways Corporation (ERC).

1.4.2. Specific Objectives

In conducting the study, the following specific objectives will address the duties that will be performed in order to achieve the general objective of the study.

Specific objectives of this study are:

- To model the flat wagon underframe structure for container transportation and analyze the structure using ANSYS and parametric study on the exact steels material (S235).
- To optimize or improve structural weight and bending stiffness of railways freight wagon underframe by changing plate thickness of the underframe structure.
- To analyze the static behavior of the optimized freight vehicle underframe.
- To develop a new freight wagon underframe plate thickness.

1.5. Scope and Limitation of the Research

1.5.1. Scope of the Research

Scope of the research includes the following;

- To undergo Multi Objective Optimization of a flat wagon underframe by taking center sill plate thickness as design parameter.
- To minimize weight of the underframe keeping bending stiffness and other parameters in their permissible range.
- To reduce material and energy consumption of the rail vehicle by minimizing the overall weight of the underframe.
- To have an optimized flat wagon underframe that is affordable in weight, stiffness, material utilization and energy consumption.

1.5.2. Limitations of the Research

Some of the limitation that I faced when conducting the research includes:

- Data: the required data has not been available easily; and I have been forced to undergo a physical measurement and to take several assumptions.
- Software: none traditional optimization process (trial and error) needs computer special optimization software. I have got this commercially available software problem and have been forced to use ANSYS workbench software for multi objective optimization.
- Computer capacity: design optimization and finite element analysis needs large capacity computers. I have been forced to wait patiently more than 3 days to get a response on a single parameter for each simulated result.
- Even though side bares, cross bares, head stock, center sill... are parts of flat wagon underframe, the analytical calculation is based on center sill geometrical structure which carries the majority of the loads that the underframe can handle.
- Modelling: the model for the center sill structure may not possibly have the same plate thickness throughout the entire structure. I have taken the center sill of same plate thickness with assumptions.

1.6. Organizing the Research

The research is organized in six parts.

Chapter 1 is the introductory Part as stated above which contains the general background, problem statement, scope and limitation, organizing the research. Chapter 2 is a literature review which has been investigated by different researchers and relevant to this thesis work. And it briefly describes the finite element method in structural design optimization of the railway freight wagon under frame. Chapter 3 shows the modeling and simulation process starting from physical to mathematical modeling using different modeling, simulation and Optimization software like CATIA (V5R20) and ANSYS14.5 workbench software. In Chapter 4, results and discussions of structural design optimization of the underframe using ANSYS has been discussed. Chapter 5 describes about structural static analysis done on modified underframe just to check the degree of optimality. Hopefully

it presents design optimization results and comparisons. In chapter 6 conclusion and recommendations including proposing future work of the thesis has been given.

1.7. Research Methodology

1.7.1. Data Collection

Data which are necessary for the study were collected from different areas. These are:

- By interviewing concerned bodies Ethiopian Railway Corporation (ERC), consulting advisors and discussing with friends.
- Searching for thesis, publishing's, newspapers, books, etc. on different web sites which are helpful for meeting the secondary objective and the primary objective as well.
- Collecting standard data from chines and European standards or other railway company Frame dimension standards.
- Collecting Data by physical measurement from METEC locomotive industry workshop where a physical model is practically exposed for investigation.

1.7.2. Data Organization

The data are organized using different tools like:-

- tables
- figures/graphs
- charts and others

1.7.3. Data Analysis

- Modeling the object using CATIA V5R20 software.
- The model is simulated with appropriate accuracy by considering the effect of welding, as well as bolted and riveted joints.
- Stress analysis is done by using Finite element analysis (ANSYS 14.5 workbench).

1.7.4. Data Source

The type of data that I have been utilized is the secondary data; and Data sources are from Chinese and European standards, Ethiopian Railways Corporation, and different countries wagon manufacturer standards.

1.7.5. Research Area

Static stress analysis and optimization is one of the most applicable research in railway engineering fields. In this research, Ethiopian Railways Corporation multi modal freight transport system line has been the research areas.

1.8. Significance and Beneficiary of the Research

The research on structural optimization of freight wagon underframe is significant to the future of our country. Now a day our country is opening an opportunity for manufacturing industries especially, metal factories which manufacture train body parts such as freight wagon underframes, side frames, floors and roofs (like METEC which contributes a lot). Significance of this research is to strengthen these manufacturing industries to save their material utilization by optimizing plate thickness. But the first and the most advantage of this research is to make our own future standards in designing and manufacturing a railway freight wagon structure in general and freight wagon underframe in particular.

As a research, the primary beneficiary of the study goes to Ethiopian Railway Corporation (ERC) future research center. Since there are a lot of studies to be done in this area, it will give a comprehensive starting point for more advanced researches on rail freight transportation design and manufacturing.

1.9. Thesis Conceptual Frame Work and Thesis Structure Flow Chart

1.9.1. Thesis Conceptual Frame Work

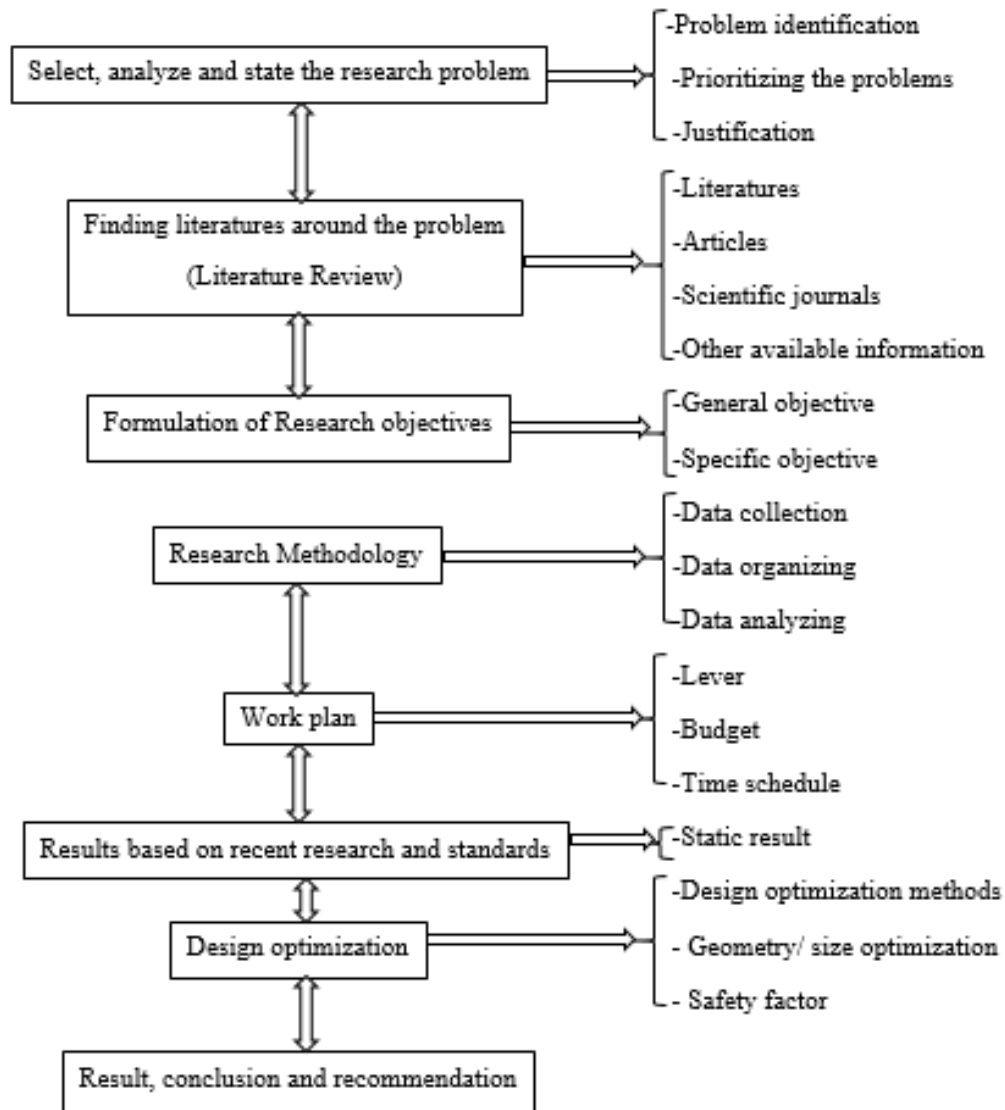


Figure 1-1: thesis conceptual frame work [21]

Chapter Two

2. Literature Review

2.1. Introduction

Optimization may be defined as the process of maximizing or minimizing a desired objective function while satisfying the prevailing constraints [3]. In every stage of design, construction and maintenance of engineering systems, engineers are forced to take certain technological and managerial decisions. The ultimate goal of all such decisions is either to minimize the effort required or maximize the desired benefit. Since either of these goals in any physical situation can be expressed as a function of certain design variables. It may also be defined as the process of finding the conditions that give the maximum or minimum value of a function [9].

Optimization is one of the oldest fields in mathematics and has found modern application in a variety of scientific and engineering disciplines [2]. Among some engineering disciplines rail vehicle structural design and optimization has been the focus of a number of previous works. These previous works were more preliminary in design analysis and optimization of automotive vehicle in general and rail vehicles in particular. They provided general information that forms foundation for the main part of this work. To effectively address the problem at hand, it is important to gain an in-depth understanding of the methods of design optimization and algorithms done related to ANSYS. The following are reviews of some of the previously conducted works related to rail vehicle underframe structural design optimization.

Prashant Kumar Srivastava, et al [2]: had done Structural Design Optimization of a bracket with Finite Element Analysis using ANSYS software. A mathematical model had been developed for the bracket based on its volume. It consists of objective function, design variable and constraints. They followed ANSYS optimization software procedure; and developed the optimum design of a bracket by minimizing the volume to accomplish the objective of minimizing the material weight.

In this paper, mathematical model is developed for the underframe weight optimization based on its plate thickness. It consists of objective function, design variable and constraints as of the researchers. ANSYS software is used to find the result in which the safe designing has been done by the help of the optimization algorithm. A series of the analysis were carried out searching for the minimum weight by changing its plate thickness each time.

Vijay Krishna, et al. 2009 [3]: had done Structural Beam Optimization by using ANSYS classic and radial basis function based response surface models. Response Surfaces method which is an effective and simple structural optimization method had been used for this work and had also showed how these Response Surfaces can then be utilized for further Multi Objective Optimization process. Here, the true or exact responses were calculated using ANSYS Classic while the codes for the simulation of Response Surface were programmed in MATLAB.

Once the responses were imported the Response Surface were generated using a RBF (Radial Basis Function Response Surface Model) approximation model. This RBF model was then optimized, and then another design point and the optimized points were then added to the DOE (Design Optimization Experiment). This process was carried out until the results had been converged. The center points and the coefficient points of each loading condition were then used for multi objective analysis.

In this paper; we have been utilized all the Response Surface, Response Surface Optimization and Six Sigma Analyses to undergo a multi objective optimization.

Rodrigo E. Castro, et al, 2002 [16]: had used Topological Optimization Techniques for linear isotropic structures subjected to static and self-weight loading conditions. Here, the researchers had described the optimal criteria and methods used for topological optimization of isotropic material under different loads and boundary conditions with the objective to reduce mass of an existing material and study the different shape obtained.

According to the researchers, APDL (ANSYS Parametric Design Language) had been employed for utilizing the topological optimization capabilities of commonly used finite element solver ANSYS. The main concept involved in shape optimization is mesh parameterization i.e. how can the coordinates of the grid points be related to a finite number of parameters and material properties.

In this paper, size optimization is applied and the problems deals with determining the outline, material volume and/or size of the underframe. In a ‘sizing’ problem mesh geometry is unchanged as the parameters that are changed.

On the other way besides reviewing, different aspects of optimization related to MDO (Multidisciplinary Design Optimization) are introduced in this chapter. Definition of design optimization, structural design optimization characteristics, optimization techniques and processes that are used during optimization and the relevant design parameters that are related to stiffness and weight of a rail freight wagon underframe or chassis has been discussed. The concept of multi objective optimization is also explained and a general optimization problem is defined.

2.2. Structural Design Optimization

To sustain various loads under different working conditions the structure should be strong and stiff enough to resist severe twisting and bending moments to which it is subjected to. Optimization seeks the selection of design variables to achieve within the limits (constraints) placed on the structural behavior, geometry, or other factors, its goal of optimality defined by the objective function for specified loading or environmental conditions [5].

Three types of structural optimization can be distinguished: size, shape, and topology optimization. In size optimization, the design variables represent some kind of structural property, e.g. sheet thickness in the different parts of a wagon. In shape optimization on the other hand, the design variables represent the shape of structural members. Topology

optimization is the most general form of structural optimization which is used to find where material should be placed to be most effective [7].

2.2.1. Optimization Algorithms

An optimization algorithm is a procedure which is executed iteratively by comparing various solutions till an optimum or a satisfactory solution is found [4]. This section discusses genetic algorithms for optimum design; and the basic idea of a genetic algorithm is to generate a new set of designs from the current set such that the average fitness of the objective function is improved [3, 7]. The process is continued until a stopping criterion is satisfied or the number of iterations exceeds a specified limit.

Three genetic operators are used to accomplish this task: reproduction, crossover, and mutation [3]. The genetic algorithms use only the function values in the search process to make progress toward a solution without regard to how the functions are evaluated. Continuity or differentiability of the problem functions is neither required nor used in calculations of the algorithms. Therefore, the algorithms are very general and can be applied to all kinds of problems discrete, continuous, and non-differentiable. In addition, the methods determine global optimum solutions as opposed to the local solutions determined by a derivative based optimization algorithm. The methods are easy to use and program since they do not require the use of gradients of cost or constraint functions.

With the advent of computers, optimization has become a part of computer aided design activities. There are two distinct types of optimization algorithms widely used today [8].

(a) Deterministic Algorithms

They use specific rules for moving one solution to other. These algorithms are in use to suite some times and have been successfully applied for many engineering design

(b) Stochastic Algorithms.

The stochastic algorithms are in nature with probabilistic transition rules. These are gaining popularity due to certain properties which deterministic algorithms do not have. The optimization algorithms are also classified into a number of groups as:

2.2.1.1. Single Variable Optimization Algorithms

These algorithms can be used to solve minimization of problems of the following type:

$$\text{Find } x = \{x_1, x_2, x_3, \dots, x_n\} \text{ which minimizes } F(x) \quad (2.1)$$

In the above equation F represents the objective function to be minimized and x_1 to x_n represents parameters that are varied through the optimization process. Optimization algorithms will vary the independent parameters in order to achieve the desired goal.

Again, these types of algorithms are classified into two categories;

- i. Direct methods
- ii. Gradient based methods

Direct methods do not use any derivative information of the objective function [4]; only objective function values are used to guide the search process. However, gradient based methods use derivative information (first and/ or second order) to guide the search process.

Although engineering optimization problems usually contain more than one variable, single variable optimization algorithms are mainly used as unidirectional search methods in multivariable optimization algorithms.

2.2.1.2. Multi Objective (Variable) Optimization Algorithms

These algorithms demonstrate how the search for the optimum point progresses in multiple dimensions. Depending on whether the gradient information is used or not used, these algorithms are also classified into direct and gradient based techniques. There are many practical applications where the designer may want to optimize two or more objective functions simultaneously. These are called multi objective, multi criteria, or

vector optimization problems; which is referred as multi objective optimization problems [7].

Optimizing a single function simply means determining a set of stationary points, identifying a local maximum or minimum, and possibly finding the global optimum. In contrast, the process of determining a solution for a multi objective optimization problem is slightly more complex and less definite than that of a single objective problem.

The predominant solution concept in defining solutions for multi objective optimization problems is that of Pareto optimality. A key characteristic of multi objective optimization methods is the nature of the solutions that they provide. Some methods always yield Pareto optimal solutions but may skip certain points in the Pareto optimal set. Other methods are able to capture all of the points in the Pareto optimal set, but may also provide non-Pareto optimal points. The former quality is beneficial when we are interested in using a method to obtain just one solution point. The latter quality is useful when the complete Pareto optimal set needs to be generated.

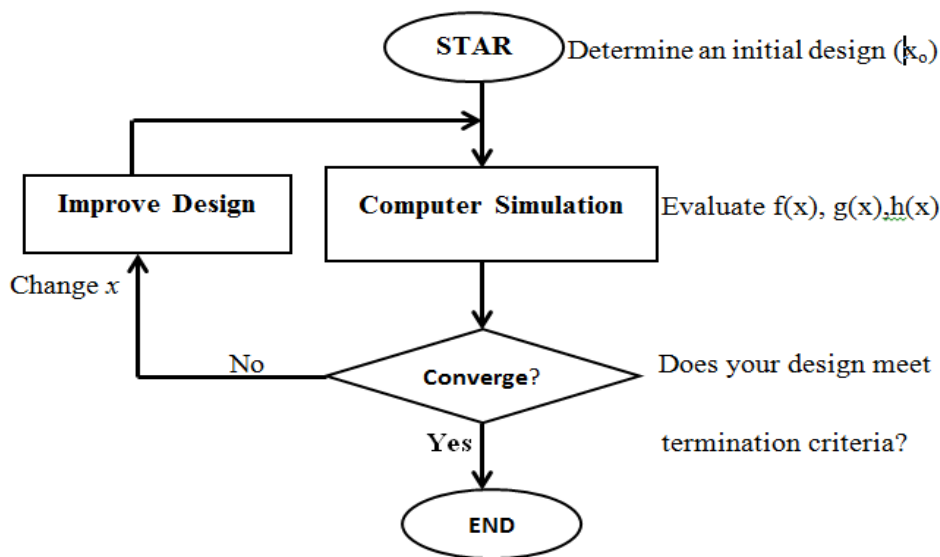


Figure 2 1: Algorithms for structural design optimization using ANSYS [22]

2.3. Optimization Problem

An optimization problem is a problem in which certain parameters (design variables) needed to be determined to achieve the best measurable performance (objective function) under given constraints [6].

The problem is a linear programming (LP) problem if the objective and constraint functions are linear functions of the design variables, and a non-linear programming (NLP) problem if the objective function or any of the constraint functions are non-linear. The formulation in Equation (1.2) also allows for maximization problems as $\max f(x)$ can be replaced by $\min (-f(x))$.

2.3.1. Single Objective Optimization Problem

Single objective optimization problems are relatively simply solved through dedicated methods, challenges still exist when more conflicting objectives are considered. For example, one of the engineering tasks where optimization is successfully used is weight reduction, since it represents one important measure to take in order to improve the engineering systems' performance. However, weight reduction usually implies the reduction of other performance criteria such as the stiffness and strength properties or the material cost. That means, an increased performance in one objective leads to a decreased performance for the others [1].

A single objective optimization problem has one objective function that is to be minimized and can be expressed as follow:

$$\text{Minimize: } f(x)$$

$$\text{Subject to: } g_j(x) \leq 0, \quad j= 1, 2, 3 \dots m$$

$$h_k(x) = 0, \quad k= 1, 2, 3, \dots, n$$

$$\text{and } (x_i)_l \leq x_i \leq (x_i)_u \quad (2.2)$$

Where $f(x)$ is the objective function, x_i is a design variable and $g_j(x)$ and h_k are the

constraints, $(x_i)_l$ and $(x_i)_u$ are lower and upper bounds of the design variable x_i . The purpose of such an optimization algorithm is to find a solution x , for which the function $f(x)$ is minimum.

2.3.2. Multi Objective Optimization Problem

The optimization problem defined in Equation (2.2) is a single objective optimization problem; it has one objective function that is to be minimized.

When solving multi objective optimization (MOO) problems, also called multi criteria optimization problems, two or more objective functions are simultaneously being optimized. An optimization problem containing n objective functions is formulated as:

$$\begin{aligned} &\text{Minimize: } f_1(x), f_2(x) \dots f_L(x) \\ &\text{Subject to: } g_j(x) \leq 0, \quad j= 1, 2, 3 \dots m \\ &\quad \quad \quad h_k(x) = 0, \quad k= 1, 2, 3, \dots, n \\ &\quad \quad \quad \text{and } (x_i)_l \leq x_i \leq (x_i)_u \end{aligned} \quad (2.3)$$

In the above equation x is the variables to be adjusted through optimization, $f_1, f_2 \dots f_L$ are multi objective functions, x is a design variable and $g_j(x)$ and h_k are the constraints, $(x_i)_l$ and $(x_i)_u$ are lower and upper bounds of the design variable x_i . The purpose of such an optimization algorithm is to find a solution x , for which the functions $f_1(x), f_2(x) \dots f_L(x)$ are to be minimum.

The easiest way to solve a multi objective optimization problem is to convert it into a single objective optimization problem [2]. The first procedure is to minimize one of the objective functions, specially the most important one, and to treat all the others functions as constraints. The second approach is to create one single objective function as a combination of the original multi objective functions. Weight coefficients can then be used to reflect the relative importance of the original objective functions [7].

The disadvantage of these methods is that one single optimum is found. If the designer wants to modify the relative importance of the objective functions in the survey, the optimization process must be performed once again [9]. An alternative is to find a number of Pareto optimal solutions. A point is Pareto optimal if there is no other feasible point yielding a lower value of one objective function without increasing the value of at least one other objective function [2]. The designer may have a set of points to choose among, and the trade-off between the different objective functions can be performed after the optimization process has been carried out. Pareto optimal solutions can be found using evolutionary algorithms; and the subject of multi objective optimization is in general and Pareto optimal solutions is in particular.

2.4. Formulation of the Optimization Problem

Problem formulation is usually the most difficult part of the optimization process. [2, 5, 7] It is the selection of design variables, constraints, objectives, and models of the disciplines. This section describes the process of transforming the design parameters of a selected system and/or subsystem into an optimum design problem. The formulation of an optimum design problem involves translating a descriptive statement of it into a well-defined mathematical statement.

There are five steps used in formulation procedure for design optimization problems: problem description; data and information collection; definition of design variables; optimization criterion; and formulation of constraints [3]. The formulation process begins by developing a descriptive statement for the problem. To develop a mathematical formulation for the problem, we need to gather information on material properties, performance requirements, resource limits, cost of raw materials, and so forth.

The next step in the formulation process is to identify a set of variables that describe the system, called the design variables. There can be many possible designs for a system, and some are better than the others. The problem is how one compare designs and select one as better than another. For this, one must have a measure that associates a number with

each design. The final step in the formulation process is to identify all constraints and develop expressions for them.

The majority of engineering problems contain a constrained optimization formulation, that is, the task of minimizing (or maximizing) a vector of objective functions (performance indices) subject to different types of constraints. Objective functions and constraints must be clearly expressed as functions (g,h) of the design variables x .

A simple optimal design is achieved by comparing a few (limited up to ten) alternative solutions created by using previous problem knowledge [7]. In this method feasibility of each design solution is first examined. Thereafter an estimate of underlying objective (weight, stiffness, etc.) of each solution is compared and best solution is implemented.

It is clear that, design parameters vary from product to product and different techniques are used in different problems. That means, it is impossible to apply single formulation procedure for all engineering design problems. But, we usually follow some common outline steps of figure 2.1. which is involved in an optimal design formulation.

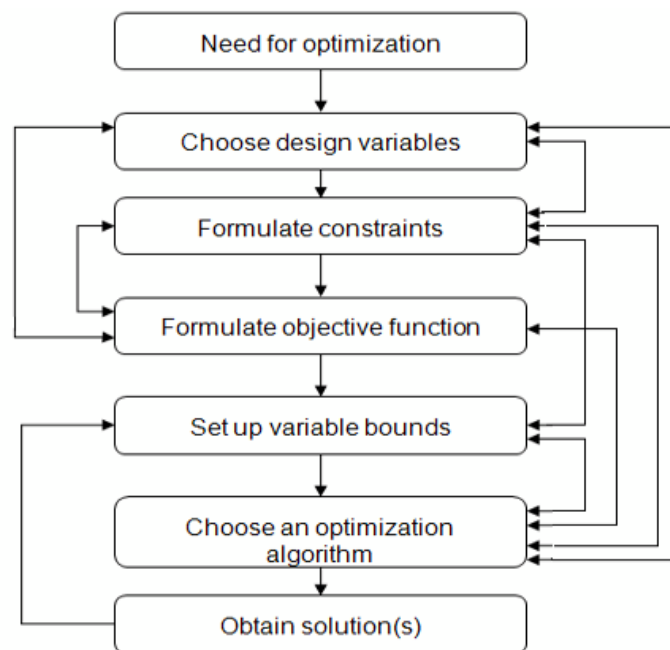


Figure 2 2: design optimization outlines [5]

2.4.1. Design Variables

A design variable is a specification that is controllable from the point of view of the designer [2, 9]. For instance, the thickness of a structural member can be considered as a design variable; the material properties that the structure is made of may be another design variables.

The formulation of an optimization problem begins with identifying the underlying design variables, which are mostly varied during the optimization process. A design problem usually involves many design parameters, of which some are highly sensitive to the proper working of the design. These parameters are called design variables. Other (not so important) design parameters usually remain fixed or vary in relation to the design variables [2, 11].

Design variables are sometimes bounded, that is, they often have maximum and minimum values. They can be continuous or discrete, meaning that they can take any value, or only certain discrete values, between the upper and lower limits. Design points that satisfy all constraints are feasible, while all other design points are unfeasible. An unconstrained optimization problem lacks constraints, as opposed to a constrained optimization problem [3].

The first thumb rule of the formulation of an optimization problem is to choose as few design variables as possible [4]. The outcome of that optimization procedure may indicate whether to include more design variables in a revised formulation or to replace some previously considered design variables with new design variables.

Type of design variables:

- continuous variables
- integer programming (discrete variables)
- mixed variables

2.4.2. Constraints

Constraints are conditions that must be satisfied in order for the design to be possible [2, 3]. The constraints represent some functional relationships among the design variables and other design parameters satisfying certain physical conditions and resource limitations. The nature and number of constraints to be included in the formulation depend on the objective function; and they may have or may have not exact mathematical expressions. For example, if maximum stress is a constraint of a structure and if a structure has regular shape they have an exact mathematical relation of maximum stress with dimensions. But in case of irregular shape, finite element simulation software may be necessary to compute the maximum stress [9].

The following two types of constraints appear from most considerations [7]:

- i. Inequality type Constraints: Inequality constraints state that the functional relationship among variables is greater than, smaller than or equal to a resource value.
- ii. Equality type Constraints: Equality constraints state that functional relationships should exactly match to a resource value. It is very difficult to handle the equality constraints in the algorithms. In such cases, equality constraint is relaxed by including two inequality constraints.

Many optimization problems require constraints on the design problem that must be solved before the optimization can be considered complete. The constraint equations are mathematically summarized below.

$$\begin{aligned}
 &\text{Minimize: } f_1(x), f_2(x), \dots, f_L(x) \\
 &\text{Subject to: } g_j(x) \leq 0, \quad j= 1, 2, 3, \dots, m \\
 &\quad h_k(x) = 0, \quad k= 1, 2, 3, \dots, n \\
 &\quad \text{and } (x_i)_l \leq x_i \leq (x_i)_u
 \end{aligned} \tag{2.4}$$

From the above equation x is the variables to be adjusted through optimization, f is the optimization objective function, h represents the constraint equality equation and g

represents the constraint inequality equations and i and j represent the number of constraint equations.

2.4.3. Objective Functions

An objective function is a numerical value that is to be maximized or minimized [2, 3, and 7]. For example, we may wish to maximize profit or minimize weight a structure. Many solution methods work only with single objectives. When using these methods, we normally weights the various objective functions and sums them to form a single objective. Other methods allow multi objective optimization, such as the calculation of a Pareto front [2].

An optimization program requires the definition of an objective function to be optimized by varying the parameters associated with the design objective [2]. The constraints are applied to the optimization parameters since the parameters can be interrelated by physical laws or must be constrained to ensure physical compatibility, or to simplify the model. A problem that has no inequality constraints is said to be unconstrained, however there is still a possibility to be bounds on the parameter values. In this paper optimization is applied to the design of the vehicle model in order to improve the bending stiffness to weight ratio of the underframe.

Since weight and stiffness are the major factors in load carrying structural design analysis [1], optimization has been done on minimizing the major influencing factor that is weight to stiffness ratio of the wagon underframe.

A general optimization problem, or mathematical modeling of the problem, can be formulated as:

Minimize: $f(x)$

Subject to: $g_j(x) \leq 0$ 2.5

In this formulation, the inequality constraints, g contains all three types of constraints of the former formulation in equ.2.3. This is achieved by replacing each equality constraint

by two inequality constraints and by including these, together with the upper and lower limits on the design variables in the constraint vector (g). [5]

The next task in the formulation procedure is to find the objective function (f) in terms of the design variables (x) and other problem parameters. The common engineering objectives involve minimization of overall weight of a component, or maximization stiffness to increase total life of a product or others.

Although most of the objectives can be quantified (expressed in mathematical form), there are some objectives (such as aesthetic aspect of a design, ride characteristics of a vehicle suspension design and reliability of a design) that may not be possible to formulate mathematically [7, 13]. In such a case an approximating mathematical expression is used.

In real world optimization, there could be more than one objective that the designer may want to optimize simultaneously. The multiple objective optimization algorithms are complex and computationally expensive. Therefore the most important objective is chosen as the objective function and the other objectives are included as constraints by restricting their values within a certain range [2, 3]

The objective function can be of two types. Either it is to be maximized or it has to be minimized. Usually the optimization algorithms were written for minimization problems or maximization problems. Although in some algorithms, some minor structural changes would enable to perform either minimization (or) maximization; this requires extensive knowledge of the algorithm.

The duality principle helps by allowing the same algorithm to be used for minimization or maximization with a minor change in the objective function instead of a change in the entire algorithm. If the algorithm is for solving a minimization problem, it can be easily changed to a maximization problem by multiplying the objective function by -1 and vice versa. [2, 5]

2.4.4. Variable Bounds

The final task of the formulation procedure is to set the minimum and the maximum bounds on each design variable. Certain optimization algorithms do not require this information. In these problems, the constraints completely surround the feasible region. Other problems require the search algorithm with in these bounds.

In general, all N design variables are restricted to lie within the minimum and the maximum bounds as follows.

$$(x_i)_l \leq x_i \leq (x_i)_u \text{ for } i= 1,2,3,4,\dots N \quad (2.6)$$

In any given problem the determination of the variables bounds $(x_i)_l$ and $(x_i)_u$ may be difficult. One way to remade this situation is, to make a guess about the optimal solution and set the minimum and maximum bounds so that the optimal solution lies within these two bounds. [4, 7] If any design variable corresponding to the optimal solution is found to lie on or near the minimum or maximum bound, the chosen bound may be adjusted and optimization algorithm may be simulated again. And after the above four tasks are completed, the optimization problem can be mathematically written in a special format, known as linear or nonlinear programming (LP or NLP).

2.4.5. Convergence

The check for convergence can be completed in a number of ways, depending on the problem and optimization method being used. Convergence is checked by comparing the current output function with the previous output function. If the difference between these values is less than a specified target or tolerance value then convergence has been achieved. This is illustrated using Equation 2.3 shown below.

$$|\Delta F_k| = |F_k - F_{k-1}| < \epsilon F \quad (2.7)$$

In the above equation ΔF_k is the change in objective function, F_k is the current objective

function value F_{k-1} is the previous iteration's objective function value and εF is the tolerance or target value used for the convergence [2, 7].

Constrained optimization problems are generally more difficult to solve than unconstrained optimization [2]. Recent developments have focused on reformulating constrained optimization problems as unconstrained optimization [9]. A number of optimization programs are available for the problem of being solved in this work, but the goal attainment method was selected for the optimization of the wagon underframe structural design optimization.

The goal of this method is to reduce the objective functions defined by F below a set of goals defined for each objective. The vector x defines the input parameters that are varied during each iteration starting at an initial condition. The vectors l_b and u_b define the lower bound and upper bound values respectively for the input parameters defined in x , the subscript i is an index used to show that the bounds are not uniform for each entry of the input parameters. The goal attainment method will attempt to reduce the objective functions below their defined goals, however the goals are not always initially known [1].

2.5. Optimization Objectives

A multi objective optimization process that is held in this paper utilizes the following objective functions, and defined as follow.

Objective function: minimize weight to stiffness ratio;

Subject to: von-mises stress \leq allowable stress;

Total deformation max. $\leq \delta_{\text{allowable}}$;

And $x_{\text{lower}} \leq \text{plate thickness} \leq x_{\text{upper}}$ 2.8

On the other hand the objective function in equ.2.8 can be express as:

Minimize (Obj = W/kb) (2.9)

In the above objective function Obj stands for objective function, W is the structure weight, and K_b the bending stiffness. Prior to performing the optimization process the initial objective functions were calculated using the uniform values. The initial objective function has been used as the basis for improvement and the optimization process that are going to be used in reducing these values. The initial values for the optimization are shown below in chapter 3 of tables 3.1, 3.2, and 3.3.

Chapter Three

3. Physical and Mathematical Modeling

3.1. Initial Parameters for Optimization

In the present literatures, the optimization process has been described in many ways. The initial data for optimization process is taken from different European and Chinese standards, and physical measurement with some assumptions.

3.1.1. Initial Assumptions

While beginning this research it is assumed that:

- The underframe has center sill, cross bars, head stock etc. as its components. Assume center sill is the main load carrying component and give focus on center sill weight optimization.
- Center sill was a solid element initially and now it is a shell element of plate thickness ranging from 15mm to 63mm. possible plate thickness for rail freight construction ranges from 5mm to 63mm [25, 26, 27, and 28].
- A Center sill is composed of upper, lower, and side plates which may have different plate thickness. But in our work it is assumed that all center sill plates have the same plate thickness.
- The load applied on the underframe is a uniformly distributive load in the form of pressure.
- Assume maximum total deformation is bounded by its upper boundary $\leq 5\text{mm}$. which is associated to wheelbase [20].

3.1.2. Material Selection

In this research, the material chosen is structural steel (S235) which is commonly found in existing heavy duty rail vehicle underframe structures. The recommended material and its

properties are taken from European standard (EN 12663-2) [20], and are shown below in tables 3.1, and 3.2 respectively.

Table 3 1: standard steel grade commonly used for underframe design [20]

Description	Limit stress N/mm ²		
	S235	S275	S355
Parent metal	235	275	355
Parent metal in the immediate vicinity of weld	214	250	323

Table 3 2: material properties for S235 steel grade [11, 20]

Property	Value	Units
Elastic modulus	2.1xe11	N/m ²
Poissons ratio	0.28	
Shear modulus	7.9xe10	N/m ²
Mass density	7800	Kg/m ³
Tensile strength	3.6xe8	N/m ²
Yield strength	2.35xe8	N/m ²
Thermal expansion coefficient	1.1xe-005	/k
Thermal conductivity	14	W/(m.k)
Specific heat	440	J/(kg.k)

The above data together with design variables and state variables are used as input for both weight and bending stiffness optimization and static analysis of the underframe.

3.1.3. Physical Modeling of the Underframe /Center Sill

During the optimization process, the underframe 3D model has been called for analysis several times, each time with different design parameters. So the model has to be in parametric form that enables it to change the parameter whenever required [2]. So a parametric model of the underframe structure is being modeled using a modeling software

CATIA V5R20 which is compatible with design optimization and analysis tool (ANSYS 14.5).

The Sensitivity analysis of the underframe is performed to find the effect of the objective function and the state variables (stress, deformation, and/or safety factor) on the variation of geometric parameters (length, width and thickness). Among the geometric parameters that influence the state variables, plate thickness is more and is considered as design variable for the optimization study.

Before going to mathematical expression, it is important to model the object in its physical model to relate the constraints and the objective functions to the design variables. The following figures show what the model looks like and how it is modeled.



Figure 3 1: flat wagon (container transport wagon) [11]

The above figure shows a flat wagon with the loading capacity 70t dedicated for container transportation with the gauge 1435mm. It can provide the loading capacity for 70t and can transport two containers with the dimension 6m and 12m each; or three containers of 6m each at the same time. The flat wagon can also carry refrigeration containers and become a refrigerated car when it is required.

3.1.4. Key Parameters and underframe Dimensions

While we are modeling freight wagon underframe structure, there are key parameters on which we offer special attention. These key parameters are loading capacity of the rail freight vehicle, type of load applied on the freight wagon underframe, rail gauge standard

and curve radius on which the freight vehicle is going on, underframe dimension and the like. The overall dimension of the underframe is shown fig.3.2 and fig.3.3 below.

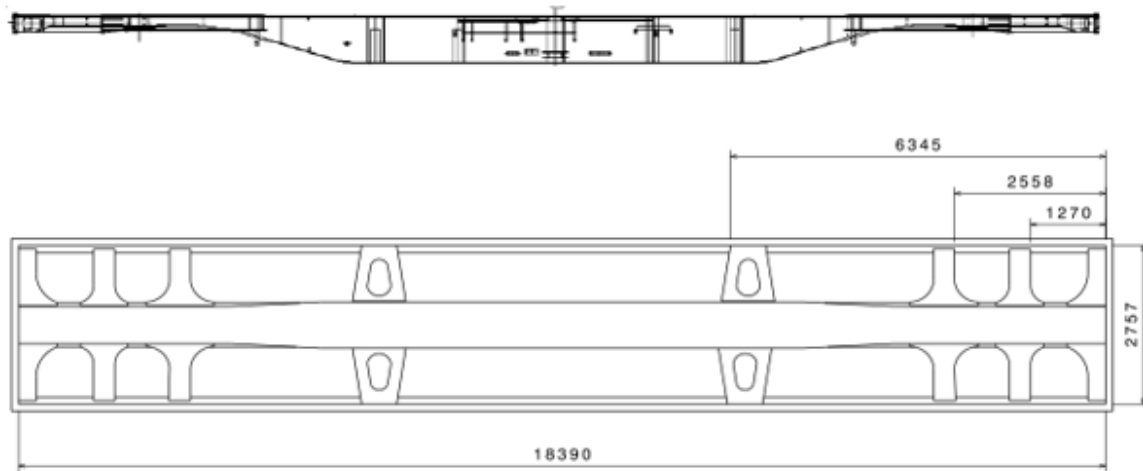


Figure 3 2: flat wagon underframe structure [11, 20, and 23]

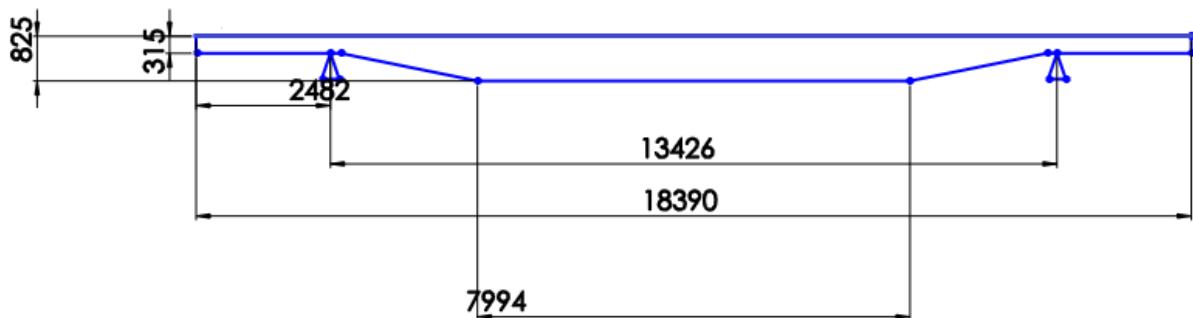
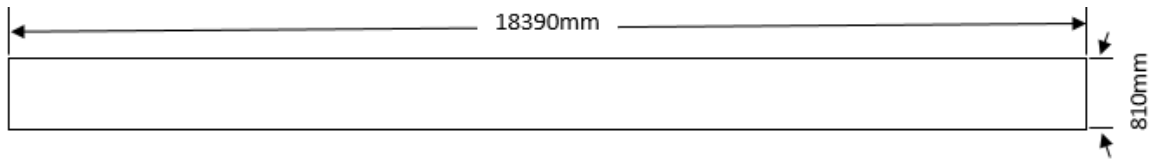


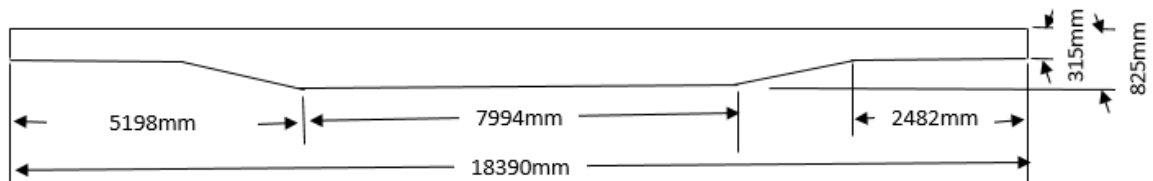
Figure 3 3: gonadal flat wagon underframe /center sill dimensions

Center sill, cross bars, head stock and etc.... are some of structural parts of flat wagon underframes. Among the flat wagon underframe structural parts, Center sill carries the majority of the loads that the flat wagon underframe is supposed to handle [1]. Therefore, the analytical calculation is based on this center sill structure and physical modeling of the center sill with its upper, lower and side plates of thickness t is shown in fig.3.4 (d).

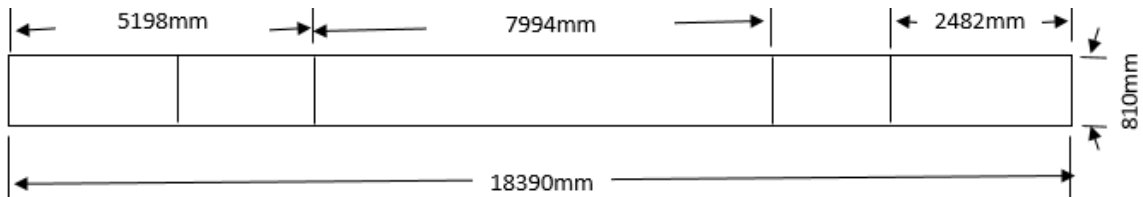
The area of each plates and side frames is calculated from the geometry, where the dimensions of each upper, sides and lower plates of the center sill and it's sub assembly are showed in fig.3.4 (a), (b), (c), and (d) respectively.



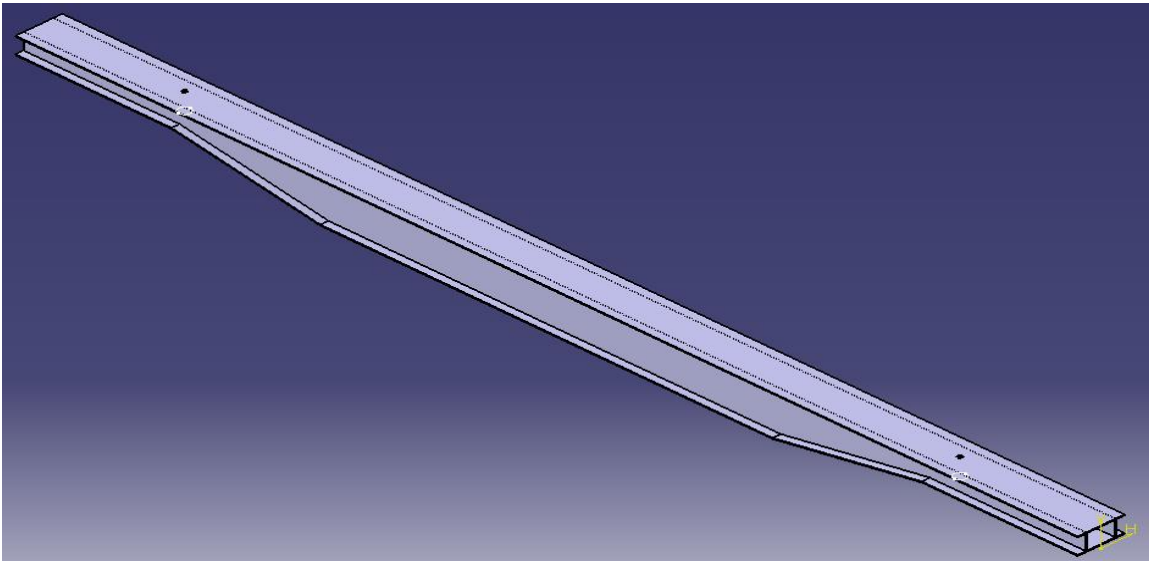
(a). dimensions of center sill upper plate with thickness t



(b). dimensions of two center sill side plates with thickness t



(c) Dimensions of lower plates of the center sill with thickness t



(d). 3D model of the center sill, (modeled as shell element) by using CATIA

Figure 3 4: center sill components and its sub assembly (welded center sill)

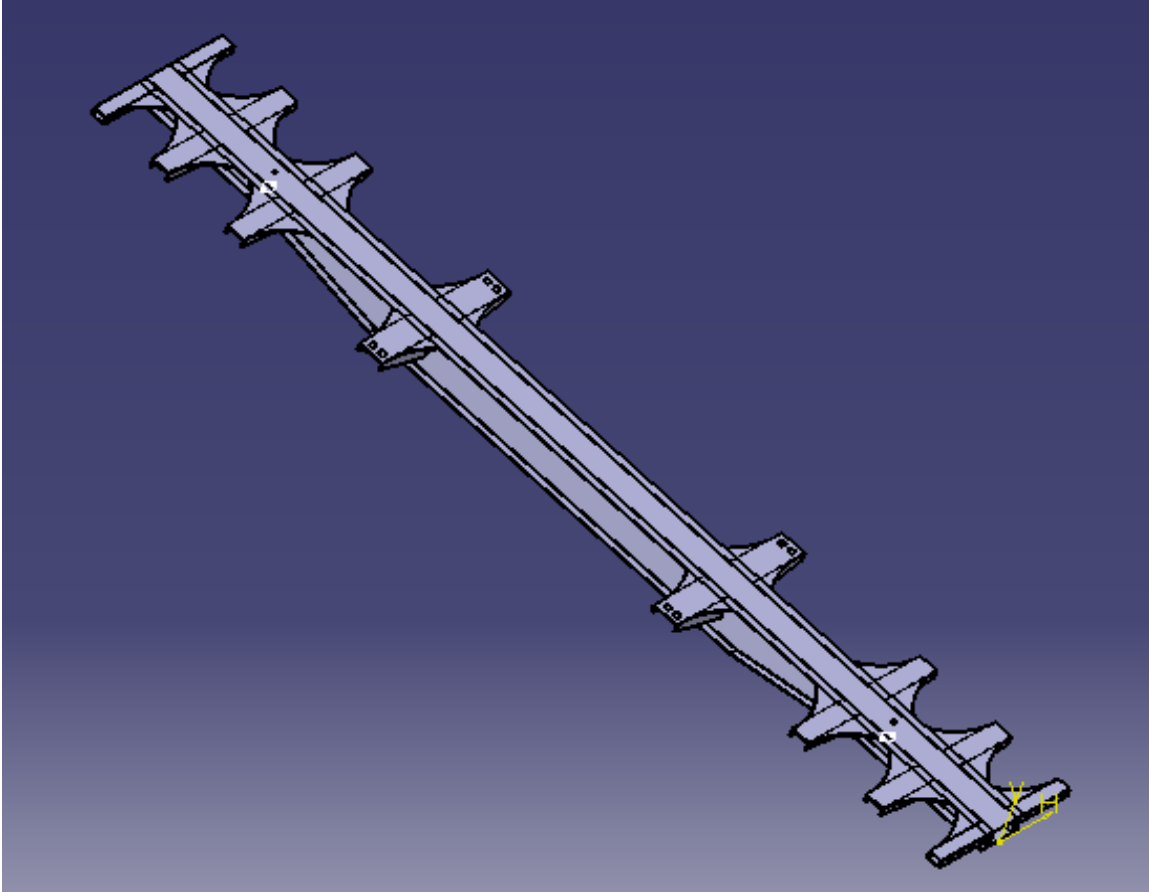


Figure 3 4: 3D model of the flat wagon underframe (center sill with side frames) done by CATIA V5R20.

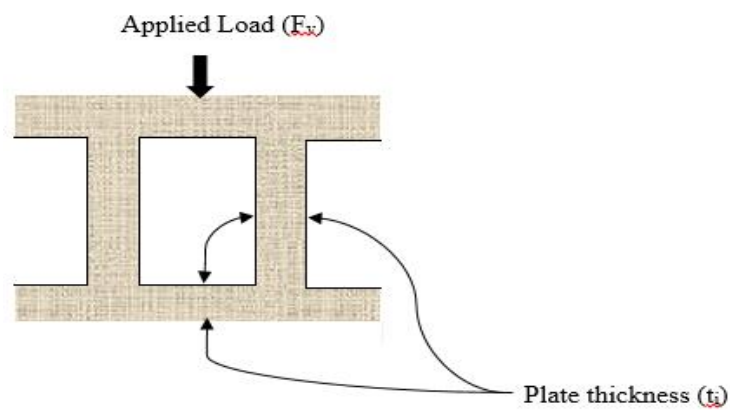


Figure 3 5: cross-sectional representation of center sill

3.2. Mathematical Modeling

This section describes the process of transforming a selected design physical model into a well-formulated optimize able design problem. The formulation of an optimum design problem involves translating a descriptive statement of the physical model into a well-defined mathematical statement.

There are five steps involved in formulation procedure for design optimization problems [7]: problem description; data and information collection; definition of design variables; optimization criterion; and formulation of constraints.

To develop a mathematical formulation for the problem, one needs to gather information on material properties, performance requirements, resource limits, cost of raw materials, and so forth. The next step in the formulation process is to identify a set of variables that describe the system. The formulation process begins by developing a descriptive statement for the problem called the design variables.

In multiple objective functions there arises a possibility of conflicts between the constraints and the objective variable. One simple way to handle the problem is to construct or model an overall objective function as a combination of the conflicting multiple objective functions.

The objective function of the research is minimizing the overall weight of the underframe by minimizing the material consumption or material volume of the wagon underframe /center sill. To simplify the process, the volume of the underframe structure as a function of the area and thickness of the plate is expressed as an objective function and stress and deflection as a constraint with the concerned point to be less than a specific limit. That is minimizing weight of the underframe /center sill, subject to equivalent stress less than or equal to the allowable stress of a welded steel parent and total deformation and plate thickness are bounded by their upper and lower boundaries.

The stress $\sigma(x)$ developed anywhere in the underframe must be smaller than or equal to the allowable strength ($S_{\text{allowable}}$) 214 N/mm² of the parent metal in the immediate vicinity of welded steel (S235) [20], while maximum total deformation is bounded by its upper limit 5mm [20], and lower and upper plate thickness is limited by its possible lower and upper boundaries of 15mm and 63mm respectively [25, 27, and 28].

Mathematically it is expressed as: $\sigma(x) \leq S_{\text{(allowable)}}$,

Subject to: $\delta(x) \leq \delta_{\text{(allowable)}}$ and

Plate thickness lower bound \leq plate thickness \leq plate thickness upper bound.

Where, $\delta(x)$ is maximum total deflection, $\delta_{\text{(allowable)}}$ is maximum total deflection of the underframe at normal design pay load and shall not exceed 0.3% of the wheelbase or bogie pivot pitch from the initial positions [20], and it is calculated and assumed to be 5mm. That means the total deflection $\delta(x)$ of a point in the center sill is limited by the upper bounds of constraints as $\delta(x) \leq 5\text{mm}$. The actual plate thickness is bounded by its upper and lower bounds and assumed to be 15mm and 63mm respectively. The lower bound assumption is taken from physical measurement of the underframe side plate and the upper bound is taken from the combination of different steel plate manufacturing countries and company's standards [25, 26, 27, and 28].

The bending stiffness of the structure is the structural resistance to an applied bending load and can be calculated using the Equation,

$$K_b = F_v / \delta. \quad (3.1)$$

In the above equation K_b represents the bending stiffness; δ is the vertical deflection of the underframe and it is measured at the middle of the frame as this represents the greatest magnitude of vertical deflection. F_v represents the down ward distributive maximum force due to the material mass of the underframe and payload at static condition.

Table 3 3: Definition of Design mass [20]

Definition	Symbol	Description
Design mass of the vehicle body in working order	M_1	The design mass of the vehicle body in working order according to EN 15663 without bogie masses.
Design mass of the bogie or running gear	M_2	Mass of all equipment below and including the body suspension. The mass of linking elements between vehicle body and bogie or running gear is apportioned between.
Normal design payload	M_3	The mass of the normal design payload as specified in EN 15663.

At maximum loading capacity, the under frame is subjected with loads due to design mass of the under frame, M_1 and design payload of the vehicle, M_3 . Design mass or material mass, M_1 of the under frame can be expressed as:

$$M_1 = v * \rho \quad (3.2)$$

Where, v is material volume, and ρ is material density of the underframe with plate thickness, t .

The design weight /material weight of the underframe, W_1 can be calculated as

$$W_1 = M_1 * g \quad (3.3)$$

And it is also related to the material volume of the plate and can be expressed as

$$W_1 = \rho * g * v \quad (3.4)$$

Where, g is gravitational acceleration.

As we observe from equation 3.4 material weight can be expressed as material volume. Since our objective is to express material weight with respect to underframe plate

thickness, we have to relate material weight to other parameters such as plate thickness and plate areas as follow:

$$W_1 = \rho * g * \sum_{i=1}^n (A_i * t_i) \quad (3.5)$$

Where, A_i stands for plate areas of the upper, lower and left and right side plates of the center sill as well as side frames and bars, t_i stands for thickness of each underframe plates and side frames and bars and i stands for number of plates, frames and bars.

Assuming each upper, lower, left and right side plate and side frames and bars has the same plate thickness t , then equation 3.5 can be rewritten as:

$$W_1 = \rho * g * t * \sum_{i=1}^n A_i \quad (3.6)$$

$$W_1 = \rho * g * t * [(14.896 + 14.973) + 2 * 10.979 + 10.35]$$

$$W_1 = 4,757,659.686 * t \text{ N} \quad (3.7)$$

In the above equation, the area is calculated by using simple geometrical relation of each plates, frames and bars (see fig.3.4 (a), (b), and (d)).

Design payload, M_3 of the proposed flat wagon is 70 tones; which has a loading effect of 686,700N, i.e.

$$W_3 = 686,700 \text{ N} \quad (3.8)$$

When the vehicle is at normal loading condition, the underframe is subjected to a compound load of W_1 and W_2 .

$$W = W_1 + W_3$$

$$W = 4757659.689 * t + 686700 \text{ N} \quad (3.9)$$

When the vehicle is at its maximum loading condition, the compound load can be calculated as follow;

$$W = 1.3*(4757659.689*t+686700) \text{ N} \quad (3.10)$$

Where 1.3 is maximum loading factor and taken from European standard [20]:

The load in equation (3.10) also can be expressed in the form of pressure P ; that is $P = \frac{W}{A}$ where, P is a downward uniformly distributive force in the form of pressure applied from the top of the underframe, W is a compound load applied on the underframe, A is the area of the upper plate where the load or pressure is applied on it.

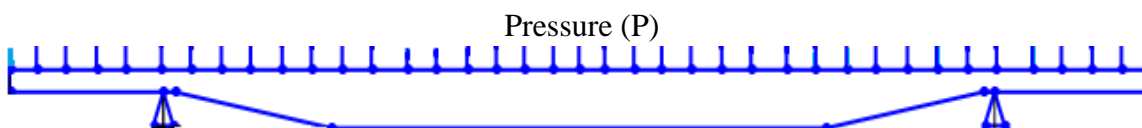


Figure 3 6: underframe with uniformly distribute load or Pressure (P)

In actual cases the freight wagon may not be loaded all over its area; sometimes the wagon may carry a single container of 6m or 12 meter. In this research it is assumed that the wagon is loaded at its maximum loading capacity and the load is uniformly distributed all over the underframe as shown in fig.3.4 above.

$$P = \left(\frac{6184957.5957*t + 892710}{20.1} \right) \text{ N/m}^2 \quad (3.11)$$

In the above equations t is a design variable, thickness bounded by its lower and upper value of 0.015m and 0.063m respectively [25, 26, 27, 28]. i.e.

$$0.015\text{m} \leq t \leq 0.063\text{m} \quad (3.12)$$

For the case of convenience, we can optimize the material weight by using uniformly distributive force in the form of pressure as objective function, stress and deflection as constraint functions and plate thickness as input design variable. After the optimizing the weight, the material volume corresponding to plate thickness has been taken. The optimum plate thickness again verifies the optimum weight of the underframe as described in equation (3.7).

Consider equation (3.11) as an objective function, that is minimize material weight of the underframe of plate thickness t , subject to allowable stress ≤ 214 pa, deflection ≤ 0.005 m, plate thickness is bounded by its upper and lower limits of 0.063m and 0.015m respectively. The upper limit of the plate thickness is put by assuming there is a hollow space for brake pipes and lines, electric signals and communication lines besides assumptions based on section 3.2 [24, 25, 26 & 28].

The objective function is now mathematically expressed as:

$$\text{Minimize: } f(t) = \left(\frac{6184957.5957*t + 892710}{20.1} \right) \text{ N/m}$$

$$\text{Subject to: } \sigma \leq 214(\text{Pa})$$

$$\delta_{\max} \leq 0.005\text{m}$$

$$0.015\text{m} \leq t \leq 0.063\text{m} \quad (3.13)$$

3.3. Design Optimization using ANSYS Workbench

Design optimization is an iterative process. Iterative implies analyzing several trial one after another until an acceptable design is obtained [8]. It is important to understand the concept of trial design. In the design process, the designer estimates a trial design of the system based on experience, intuition, or some simple mathematical analyses. The trial design is then analyzed to determine if it is acceptable, and the design process is terminated. In the optimization process, the trial design is analyzed to determine if it is the best. Depending on the specifications, “best” can have different connotations for different systems. In general, it implies that a system is cost-effective, efficient, reliable, and durable.

3.3.1. Optimization Procedures

An optimization program is implemented to improve the previously structural design with the goal of reducing weight while holding the stiffness of the vehicle in its bounding limit by changing a number of model parameters.

The first step in structural optimization is parameterizing the input and output parameters or design and state variables; and this is done after the model has been imported in to the ansys workbench (during structural analysis and optimization process). Based on the requirement, we can parameterize design variables like length, thickness, width and state variables such as material volume, material density, young's modulus, yield strength, material mass, total deformation, equivalent stress, safety factor, etc. as input and output parameters. Figures 3.8 up to 3.11 show how these input and output parameters are addressed and what they look like during the optimization process.

	A	B	C	D	E
1	Property	Value	Unit	<input checked="" type="checkbox"/>	<input checked="" type="checkbox"/>
2	Density	7800	kg m ⁻³	<input checked="" type="checkbox"/>	<input checked="" type="checkbox"/>
3	Isotropic Secant Coefficient of Thermal Expansion			<input type="checkbox"/>	<input type="checkbox"/>
6	Isotropic Elasticity			<input type="checkbox"/>	<input type="checkbox"/>
7	Derive from	Young's M...			
8	Young's Modulus	2.1E+11	Pa	<input checked="" type="checkbox"/>	<input checked="" type="checkbox"/>
9	Poisson's Ratio	0.28		<input checked="" type="checkbox"/>	<input checked="" type="checkbox"/>
10	Bulk Modulus	1.5909E+11	Pa	<input type="checkbox"/>	<input type="checkbox"/>
11	Shear Modulus	8.2031E+10	Pa	<input type="checkbox"/>	<input type="checkbox"/>
12	Alternating Stress Mean Stress	Tabular		<input type="checkbox"/>	<input type="checkbox"/>
16	Strain-Life Parameters			<input type="checkbox"/>	<input type="checkbox"/>
24	Tensile Yield Strength	2.35E+08	Pa	<input checked="" type="checkbox"/>	<input checked="" type="checkbox"/>
25	Compressive Yield Strength	2.35E+08	Pa	<input checked="" type="checkbox"/>	<input checked="" type="checkbox"/>
26	Tensile Ultimate Strength	3.6E+08	Pa	<input checked="" type="checkbox"/>	<input checked="" type="checkbox"/>

Figure 3 7: material properties as input parameters in ANSYS work bench

Besides input material properties, state variables like material mass and material volume are parameterized during the optimization process see figure 3.9. The geometric parametrical values in the figure below are default input values and automatically changed after optimization process and these parameters are considered as an output parameters after the optimization process.

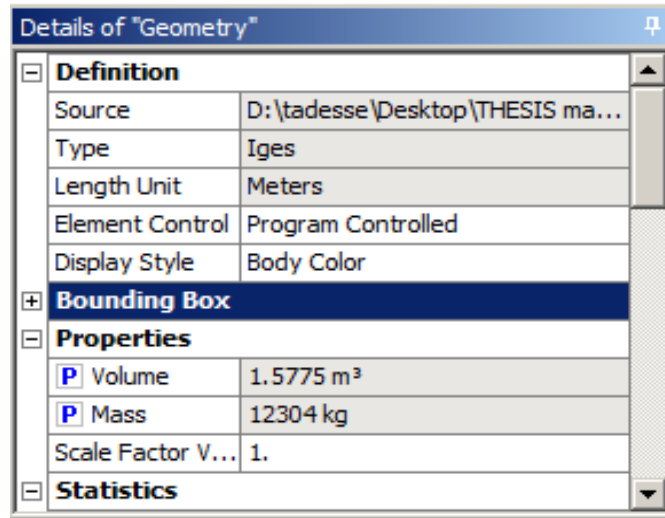


Figure 3 8: geometric mass and volume parametrizing

During the optimization process the 3D model has been done a mesh discretization process using FEM. Different meshing methods and mesh properties also have been used, (see figure 3.10).

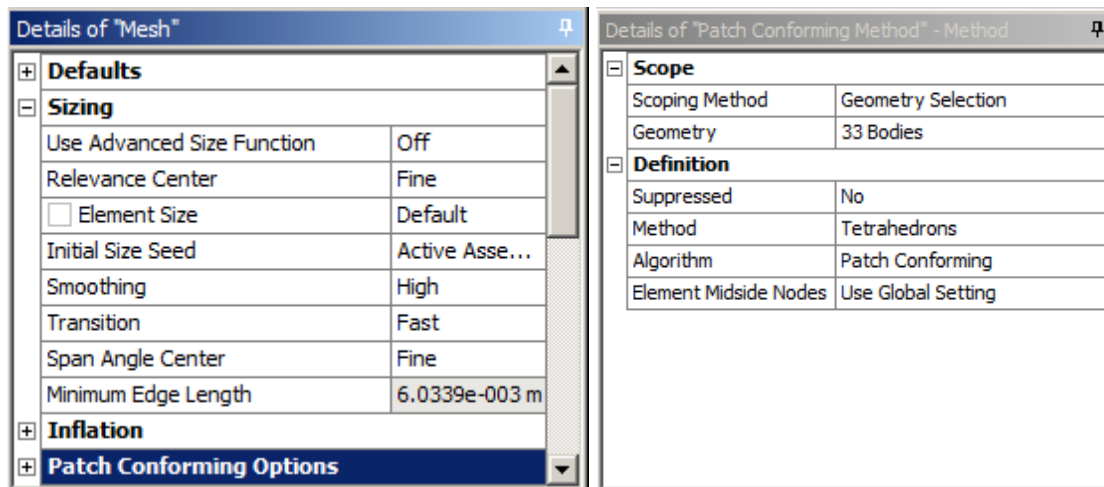


Figure 3 9: mesh details and methods

The overall distributive load in the form of pressure, applied on the upper surface of the underframe is also parameterized as input parameter in the optimization process. This parameter is used as objective function and the following figure shows how pressure is parameterized during optimization process. The pressure magnitude value indicated in the

following figure is an arbitrary value of a sample model and varies as a function of design variable (plate thickness).

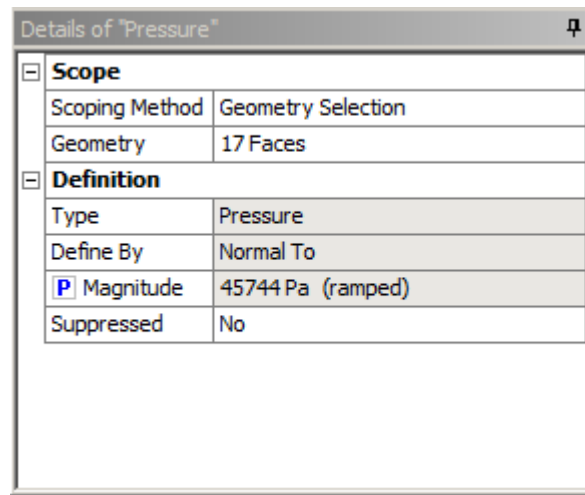


Figure 3 10: pressure parametrizing

After accomplishing the analysis, we get maximum total deformation, maximum equivalent stress and safety factor. We have to parameterize maximum total deformation and maximum equivalent stress so that the optimization process consider the effect of these parameters in relation to the input parameters. The following figure shows how total deformation max. And equivalent stress max. are parameterized in ANSYS workbench optimization.

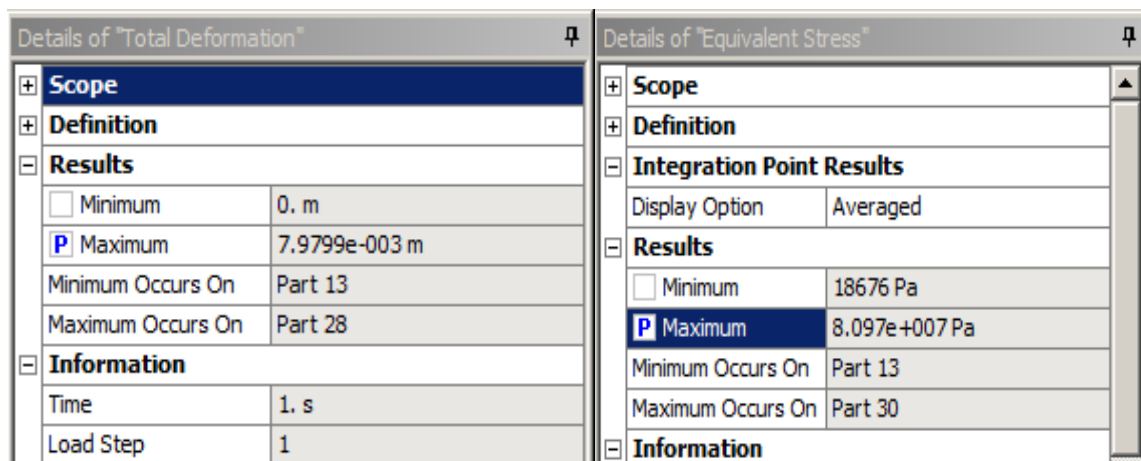


Figure 3 11: parametrizing total deformation max. and equivalent stress max.

The second step is undergoing optimization process and setting the input and output parameters within the response surface, response surface optimization and six sigma analyses as shown in figure 3.13 and figure 3.14 below. While doing this step we have passed through so many important sub steps in order to have none ambiguous optimization results. Among them the following are some of the sub steps that we follow during optimization.

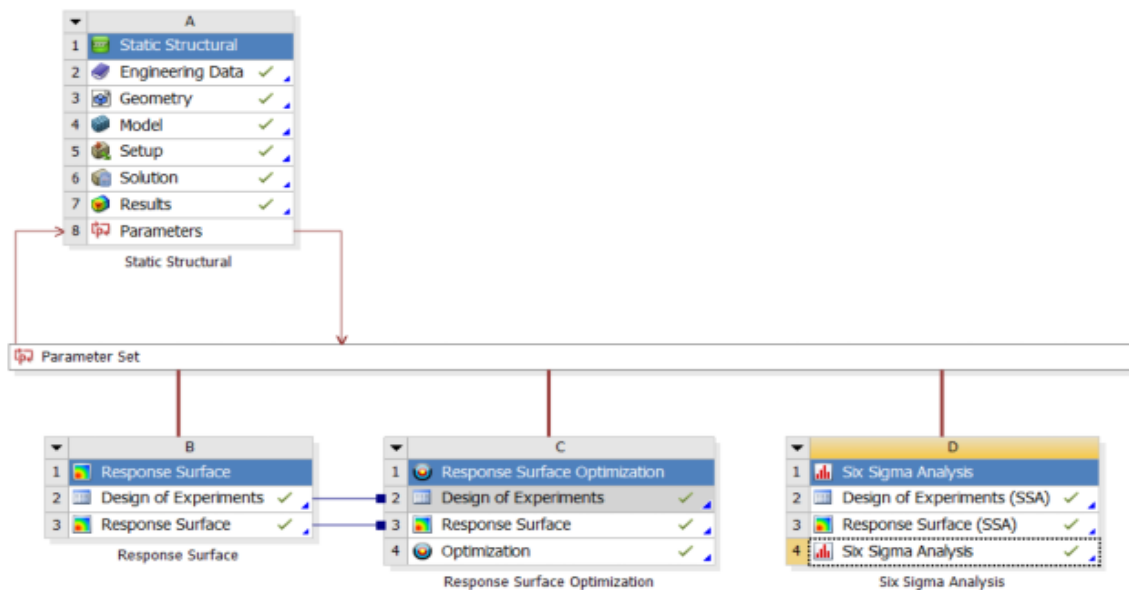


Figure 3 12: setting optimization parameters

Outline: No data				
	A	B	C	D
1	ID	Parameter Name	Value	Unit
2	[-] Input Parameters			
3	[-] [S] Static Structural (A1)			
4	[P] P1	Tensile Ultimate Strength	3.6E+08	Pa
5	[P] P2	Compressive Yield Strength	2.35E+08	Pa
6	[P] P3	Tensile Yield Strength	2.35E+08	Pa
7	[P] P4	Poisson's Ratio	0.28	
8	[P] P5	Density	7800	kg m ⁻³
9	[P] P8	Pressure Magnitude	40750	Pa
*	[P] New input parameter	New name	New expression	
11	[-] Output Parameters			
12	[-] [S] Static Structural (A1)			
13	[P] P13	Geometry Mass	12304	kg
14	[P] P14	Geometry Volume	1.5775	m ³
15	[P] P9	Safety Factor Minimum	2.5769	
16	[P] P10	Equivalent Stress Minimum	10920	Pa
17	[P] P11	Equivalent Stress Maximum	9.1195E+07	Pa
18	[P] P12	Total Deformation Maximum	0.0048881	m
*	[P] New output parameter		New expression	
20	Charts			

Figure 3 13: input and output parameters

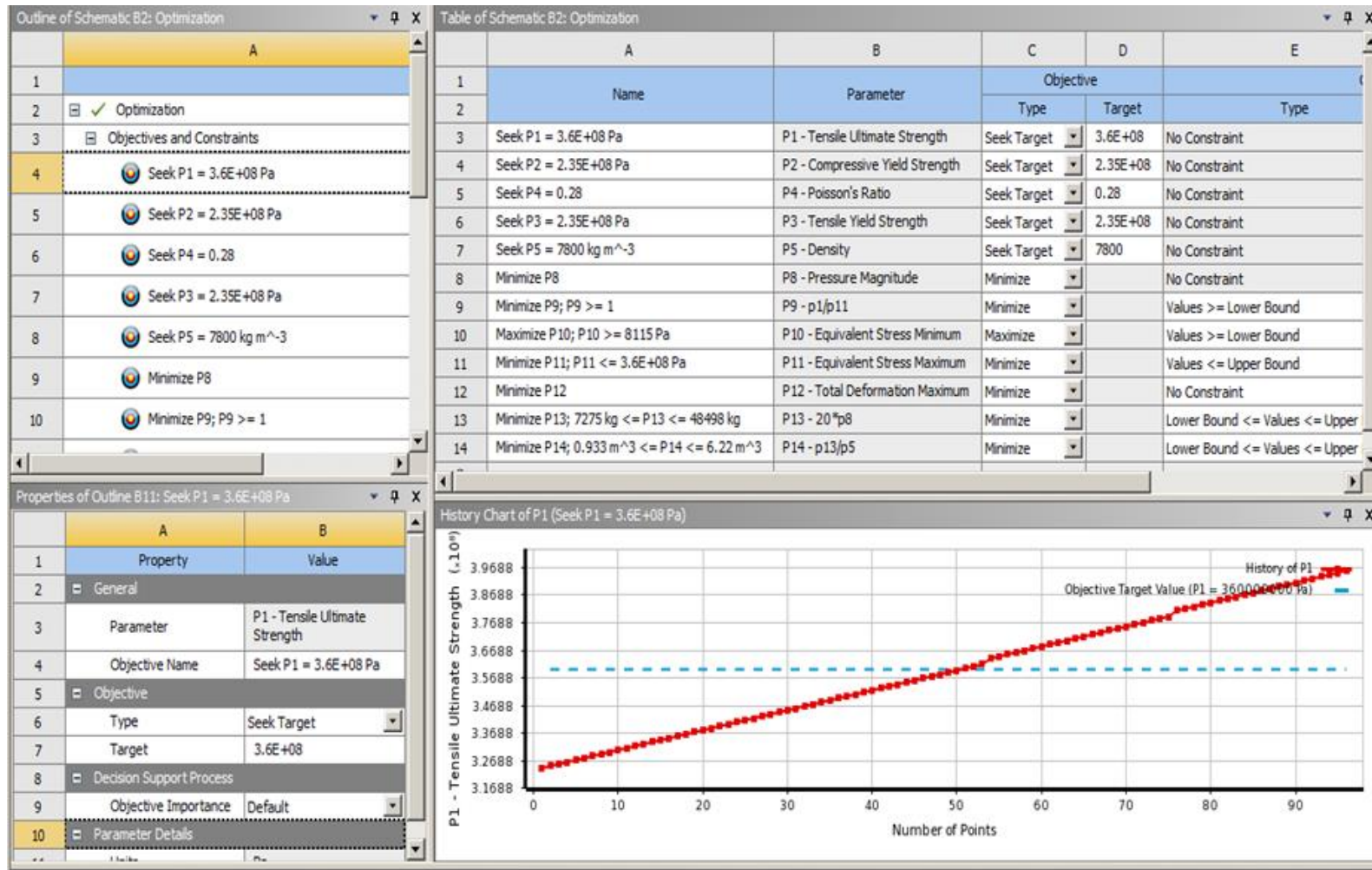


Figure 3 14: input and output parameters, objectives and constraints in ANSYS optimization work bench

3.3.2. Response Surface Method

Response surface is an efficient way to get the variation of a given objective function with respect to input parameters [3]. It provides a continuous variation of the overall uniformly distributive load in the form of pressure over a given variation of the input design variable (plate thickness). And its accuracy can be controlled well for up to 10 input parameters and it is also a great basis for further analyses of optimizations, like six-sigma analysis.

3.3.3. Design of Experiments (DOE)

A Design of Experiment (DOE) is a scientific way to conduct a series of experiments with a given set of parameters, each with a range that minimizes the number of runs needed to understand the influence of the parameters.

3.3.4. Goal Driven Optimization (GDO)

Determines candidate designs based on our design goals. It uses DOE/response surface results to quickly explore parameter space and states a series of design goals to generate design candidates.

3.3.4.1. Goal Driven Optimization Methods

There are three optimization methods in Design explorer

1. Screening (Shifted Hamersley) [default]

- Direct sampling method by a quasi-random number generator
- Good for preliminary designs

2. MOGA (Multi-objective Genetic Algorithm)

- Multi-goal optimization
- Provides several candidates

3. NLPQL (Non-linear Programming by Quadratic Lagrangian)

- Fast gradient based local optimization algorithm for single objective.

As we have said in the above literature, MOGA is our optimization goal driven optimization method that we have implemented here.

The next step in the procedure is interpreting the results and deciding the optimum result. ANSYS optimization process shows the relation between input and output parameters with the desired variable(s). We have to interpret different graphical and tabular results as we

wish to be; and decide where the optimum result would be. This is done by realizing the objective function to meet its initial target under the given constraints.

The initial condition for the optimization process is the uniformly distributive load in the form of pressure values (objective function) corresponding to the best observed data run of plate thickness (design variable). The final target is to have the plate thickness values which give the optimum underframe weight, which in turn maximizes the ratio between stiffness and weight.

Chapter Four

4. Results and Discussions

4.1. Optimization Results

There are various steps involved in graphical solution procedure such as coordinate system setup, inequality constraint boundary plot, identification of the feasible region for an inequality, identification of the feasible region, plotting of objective function contours, and identification of the optimum solution. Most of the procedures are involved during optimization parameter setup and a few of them are fetched from graphical and tabular interpretation of the final optimization results in Appendix-I and Appendix-II.

4.1.1. Graphical and Tabular Results

As we observe from graphical and tabular results in Appendix- I, there can be many feasible optimum results for a system on which some are better than others. The question is how we compare the graphical and tabular results and designate one as better than another. To do such a decision, we must have a criterion that associates each input and output parameters and state variables with design variable (plate thickness).

The next step in the formulation process is to identify all constraints and state variables and develop expressions for them with respect to the design variable (plate thickness). For convenience, graphical and tabular relation of parameters in optimization results shown in Appendix-I can be re-tabulated (see Appendix-II) and re-graphed by taking the plate thickness as a design variable.

4.1.2. Results when plate thickness is considered as design variable

In this section the graphical solution of the overall process is taken and re-finishing the optimization process by introducing several concepts related to optimum design problems. Here, optimization problems involve only one design variables and can be solved by observing how they are graphically represented. The input and output parameters and state

variables value corresponding to the design variable (plate thickness value) after running the optimization process are shown graphically (see fig.4.1 to fig.4.7).

All constraint functions (parameters) are re-plotted against the plate thickness, and a set of feasible designs (the feasible set) for the problem is identified. Objective function contours are then drawn, and the optimum design is determined by visual inspection. To implement the step-by-step procedure and obtain a graphical solution for the problem Microsoft excels is used.

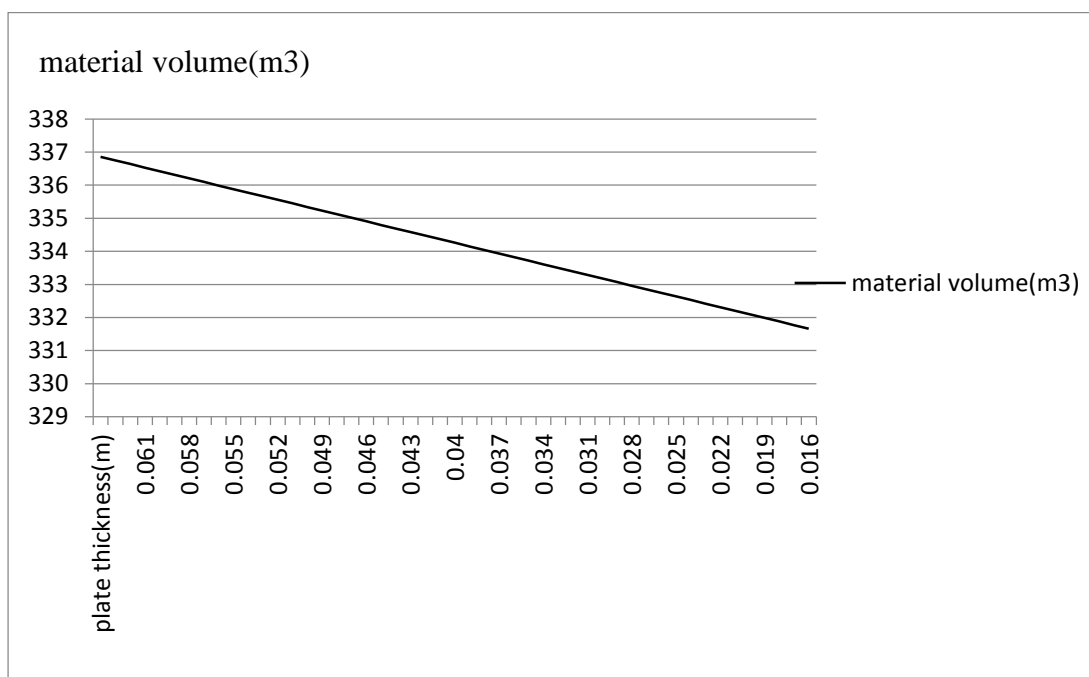


Figure 4.1: material volume vs plate thickness

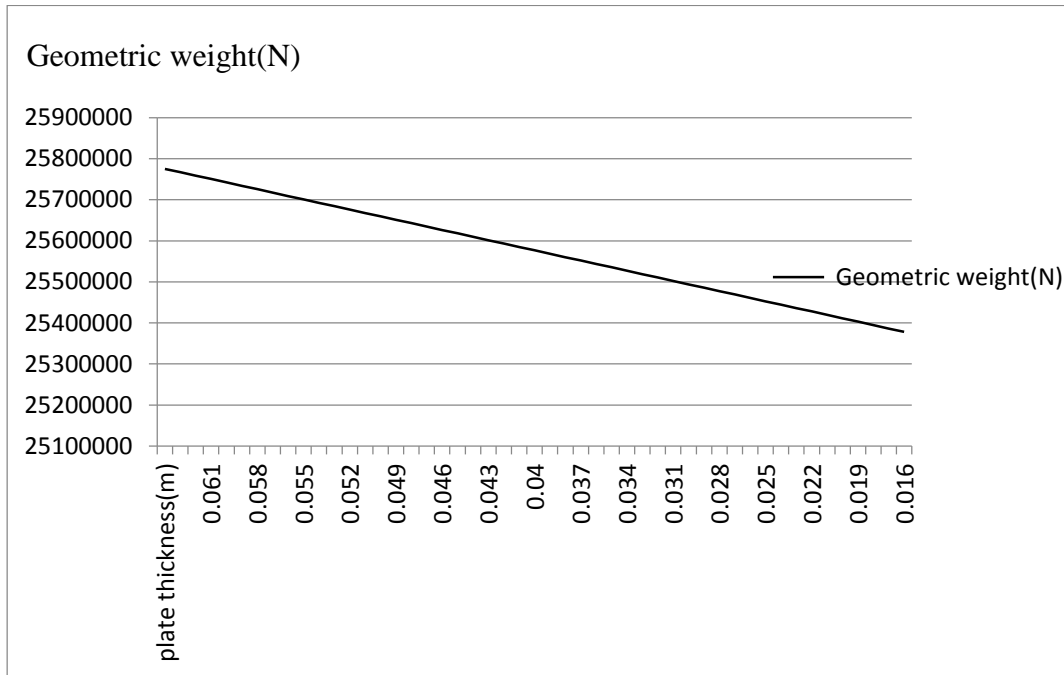


Figure 4.2: weight vs plate thickness

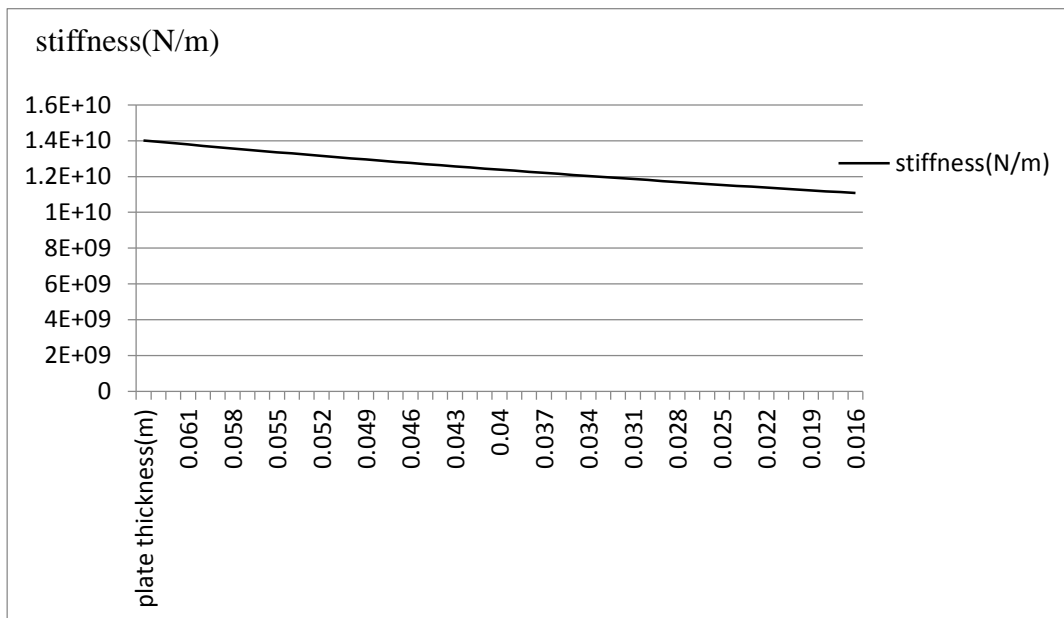


Figure 4.3: stiffness vs plate thickness

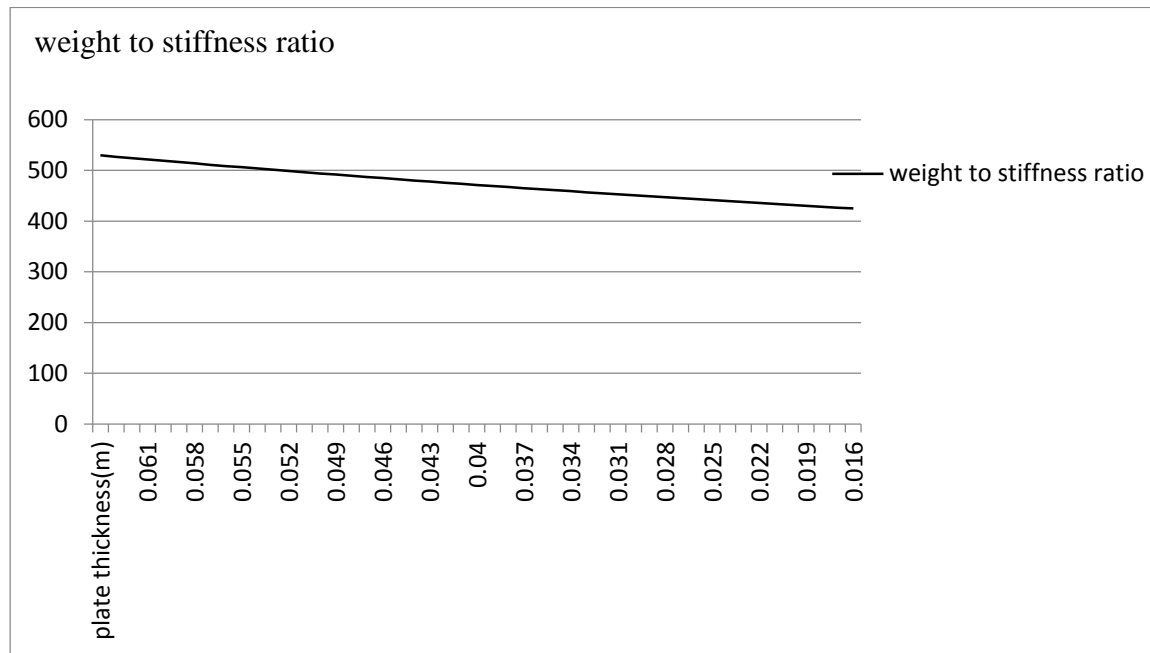


Figure 4.4: weight to stiffness ratio vs plate thickness

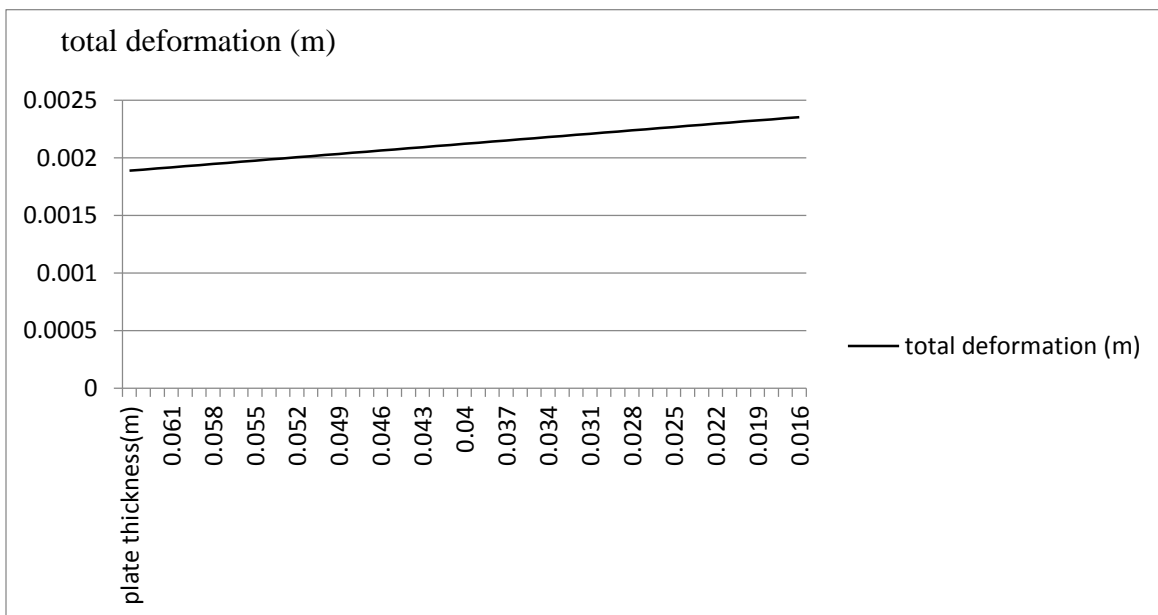


Figure 4.5: total deformation vs plate thickness

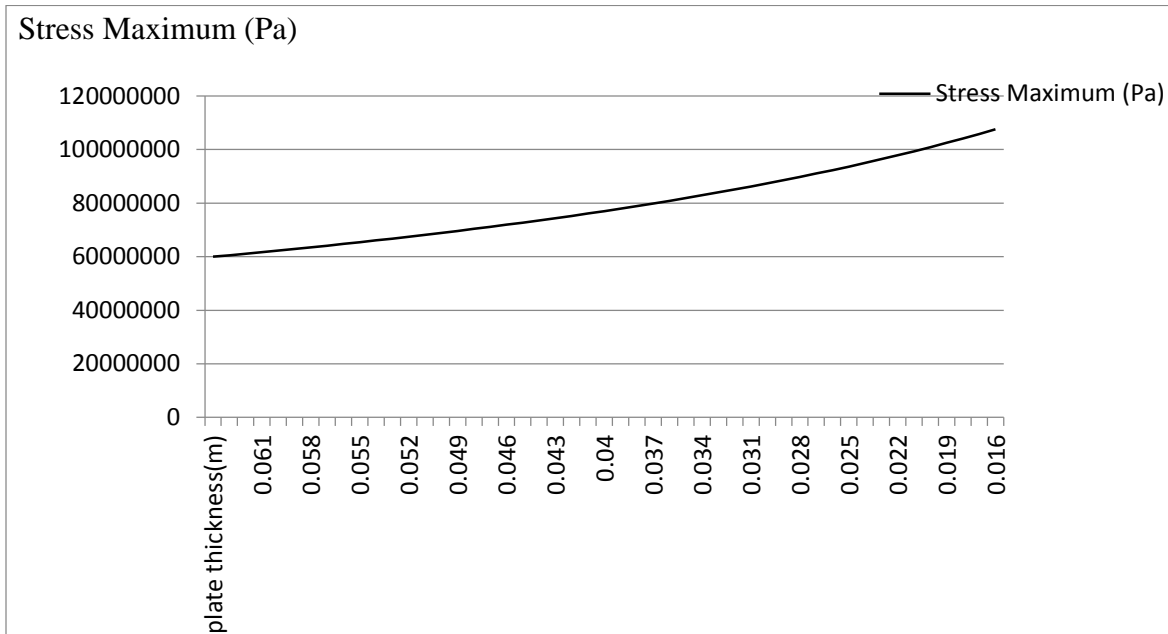


Figure 4.61: equivalent stress max. Vs plate thickness

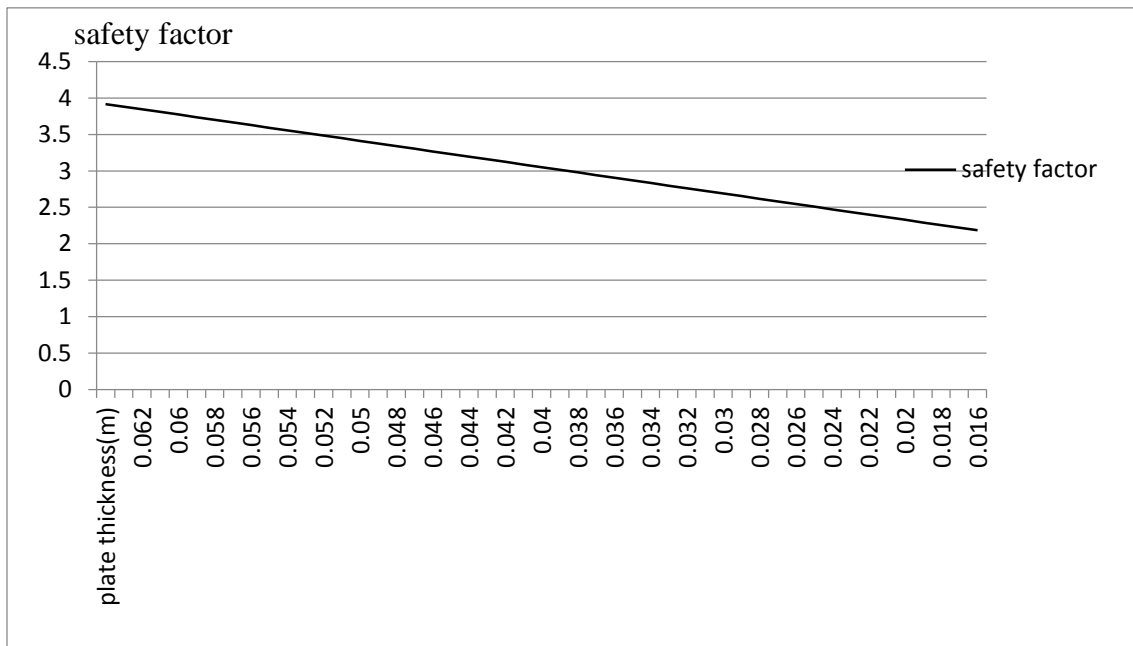


Figure 4.7: Safety factor min. vs plate thickness

4.2. Discussions on Structural Optimization

Observing the above graphs, total displacement and equivalent stress increase, and safety factor values goes minimum; as we reduce the plate thickness of the underframe. The above graphs of figures. 4.1 - 4.7 are indirectly fetched from ANSYS optimization workbench tabular result on convenience way for decision making process. Based on our design objectives of minimizing structural weight by keeping the stiffness and other parameters in its allowable range so that weight to stiffness ratio of the underframe is minimized, we interpret the graphs and decisions have been made on the design variable (plate thickness).

From the above graphs, we can observe that there are numerous solutions that can be suggested as an optimized result. But, based on our design objective of underframe weight optimization by keeping bending stiffness and other design parameters in their allowable range, and we can choose the most convenient value of the design variable (plate thickness), so that our design objectives to be achieved.

When we consider safety factor of figure 4.7, all ranges are in relatively safe condition and plate thickness of the underframe can be any value in the range 0.015m to 0.063m, but safer in diminishing direction. When we consider total deformation (see figure 4.5), it is also safe throughout all thickness range, but safer in increasing direction. But when we consider maximum equivalent stress (see figure 4.6), the graph clearly shows that the equivalent stress becomes maximum in the diminishing plate thickness direction. Therefore, from the above explanation and other parametrical relations from figures 4.1 to figure 4.7, the design variable (thickness) value can be predicted approximately as 0.023m, and equivalently the following optimization results (table 4.1.) are forwarded.

Table 4 1: optimization results

Plate thickness (m)	Material mass (kg)	Material volume (m ³)	Material weight (N)	Stress max. (pa)	Total deformation (m)	Safety factor	Stiffness (N/m)	Weight to stiffness ratio
0.0231	35500.9208	4.5514	348264.033	0.9502e8	0.0022758	2.4732	454762967	439.399778

From our initial assumption, that is when the center sill is a solid element its weight is 425.21 KN and its material volume is 5.437 m³. As we compare these results with optimized results we found the following comparison (see table 4.2).

Table 4 2: comparisons of results before and after optimization

Optimization parameters	Before optimization	After optimization	Difference	Reduction In %
Material volume (m ³)	5.437	4.5514	0.8856	16.288
Material Weight (kN)	425.21	348.265	76.8757	18.07

As we see from the above table, we save 16.288% of material volume and 18.07 % of material weight by making the center sill as shell element.

Chapter Five

5. Structural Analysis Done on Modified Underframe to Check the Optimality

5.1. Need of Re-modeling

Design Optimization is an iterative process. It implies analyzing several trial designs one after another until an acceptable design is obtained. In this section the previously conducted optimization results of the underframe (center sill) structure have been checked with finite element analysis software (ANSYS); and it is expected that the iterative design optimization result is the best or acceptable during the analysis.

The output of this process is an optimized structural model with low weight and minimum weight to stiffness ratio that assures readiness for detailed design. While re-modeling the modified 3D underframe (center sill) structure the following assumptions have been done.

5.2. Assumptions for Re-Modeling of the Underframe Structure

1. The wagon underframe (center sill) structure is modeled with shell element (hollow beam).
2. All the components such as rack structure, brake equipment, traction motors, air compressor, Sand weight, Coupler weight, pipes and cables etc. are modeled as mass elements.
3. All the components weights are modeled as mass element at center of gravity of equipment weight.
4. Bolted and riveted joint of the center sill structure (if any) are modeled as rigid elements.
5. Static analysis and simulation is carried out based on the condition of the vehicle being stationary.
6. During static analysis the underframe is treated as simply supported hollow beam,

and the load is the maximum operating load and is considered in accordance with EN12663-2 standards required for freight wagon design [20].

5.3. Physical Modeling

The physical modeling of the re-modeled underframe structure has been done in the same manner as in section 3.1.3, except with different plate thickness of 0.023m.

5.3.1. Modeling Details

A three dimensional shell geometry of the underframe (center sill) structure with plate thickness of 0.023 meter is modeled by using a 3D modeling software CATIA V5R20. And the underframe (center sill) structure has been analyzed using Finite Element Software ANSYS 14.5 workbench.

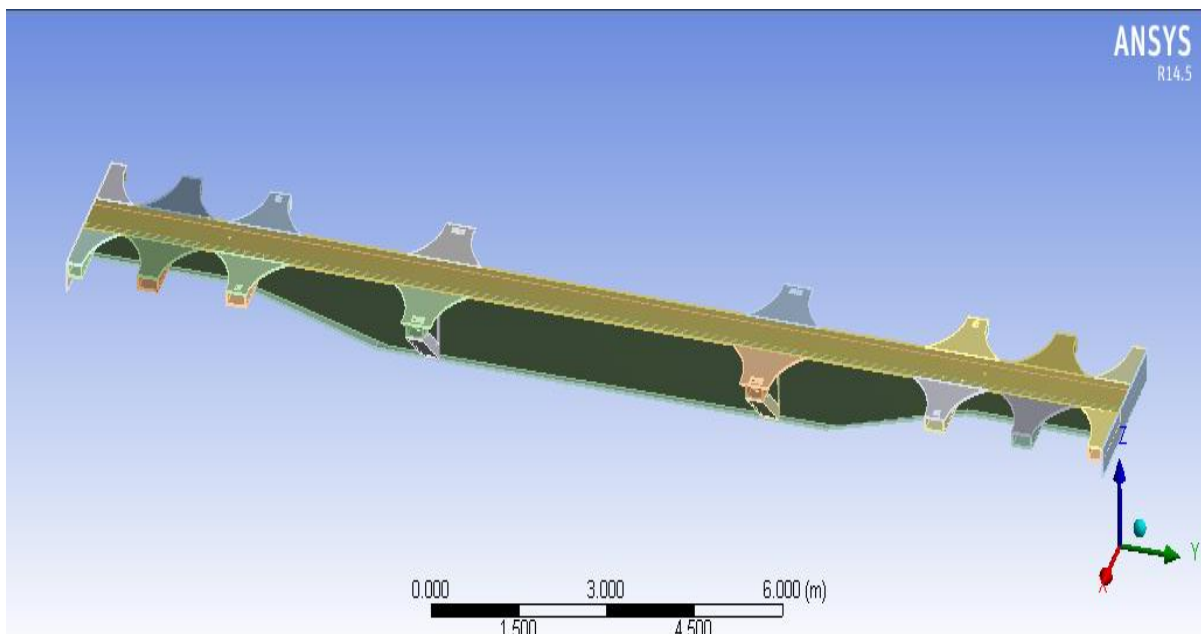


Figure 5 1: imported 3D model done by CATIA V5R20

The model is imported to ANSYS workbench as an IGS file, and the geometry is ready to prepare for meshing. This means that some of the lines in the imported model were toggled from edge lines to suppressed (or manifold) lines so that they would not represent an artificial edge that would force the finite elements to unnecessarily align themselves. The

misreading of lines happens at the locations of fillets and radii features created during modeling, as the features get falsely interpreted as distinct surfaces in the IGS transformation [13].

Once the geometry is cleaned, the design space volume is filled with tetrahedral elements using the auto-mesh features of tetrahedral Mesh. The full underframe FEA model contains 16918 elements and 38132 nodes. The weights of all equipment are included as mass elements.

The resulting mesh that is used as the design space for the structural design optimization study and other simulation results has been shown in Figure 5.2 below.

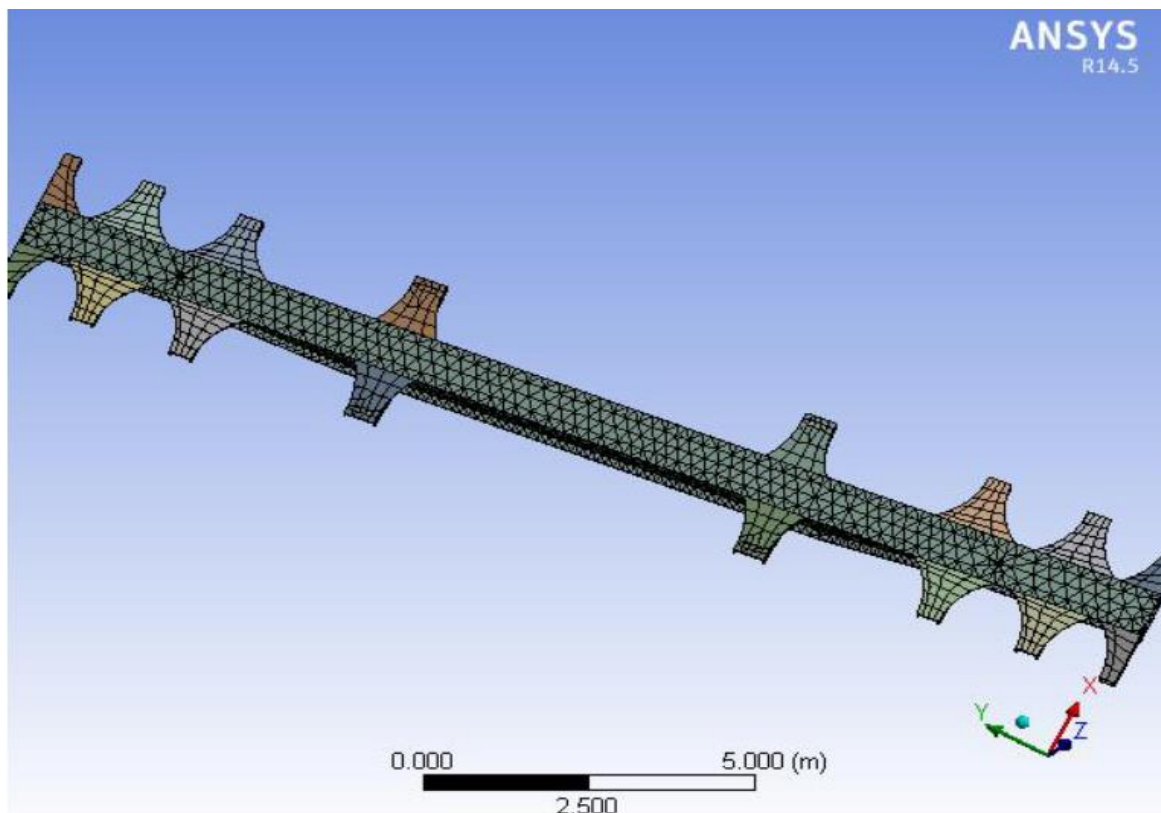


Figure 5 2: meshed model

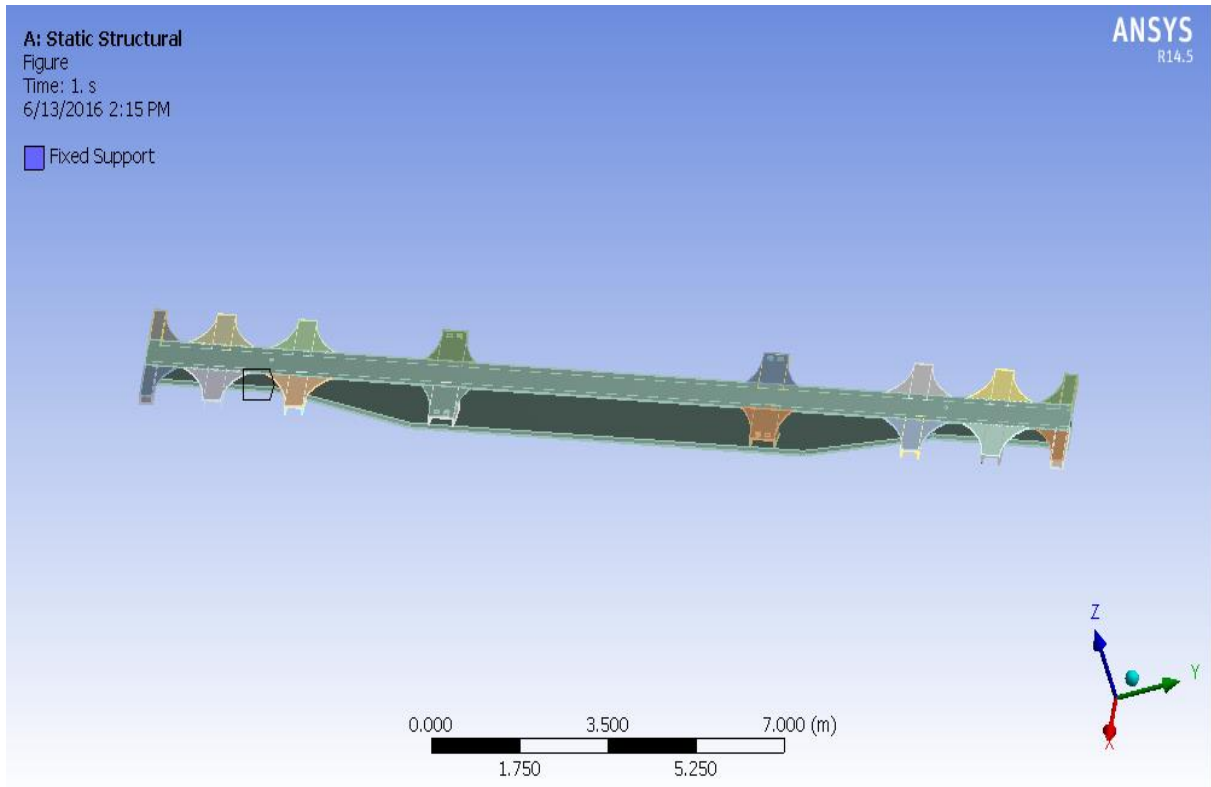


Figure 5 3: fixed support

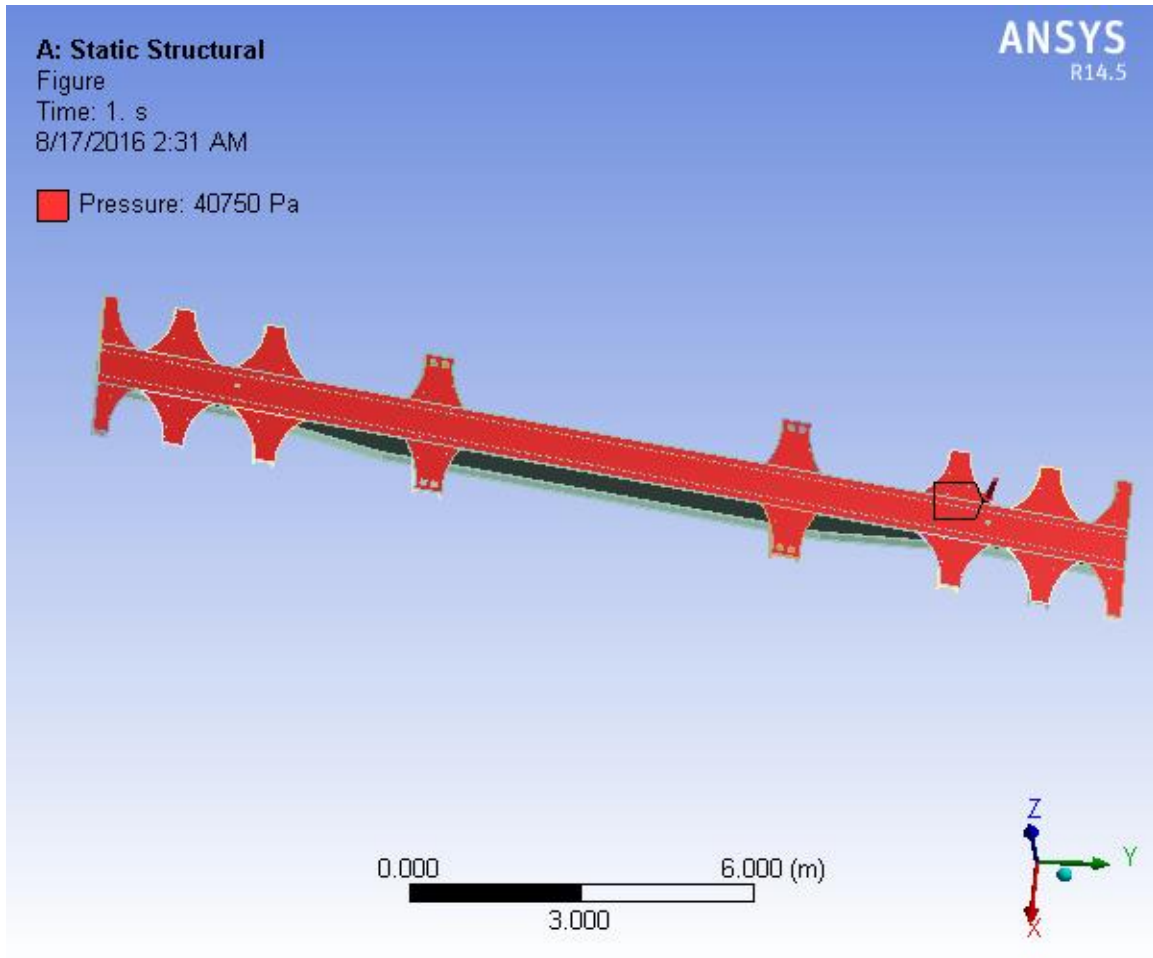


Figure 5 4: applied load in the form of pressure

5.4. Static Results after Optimization

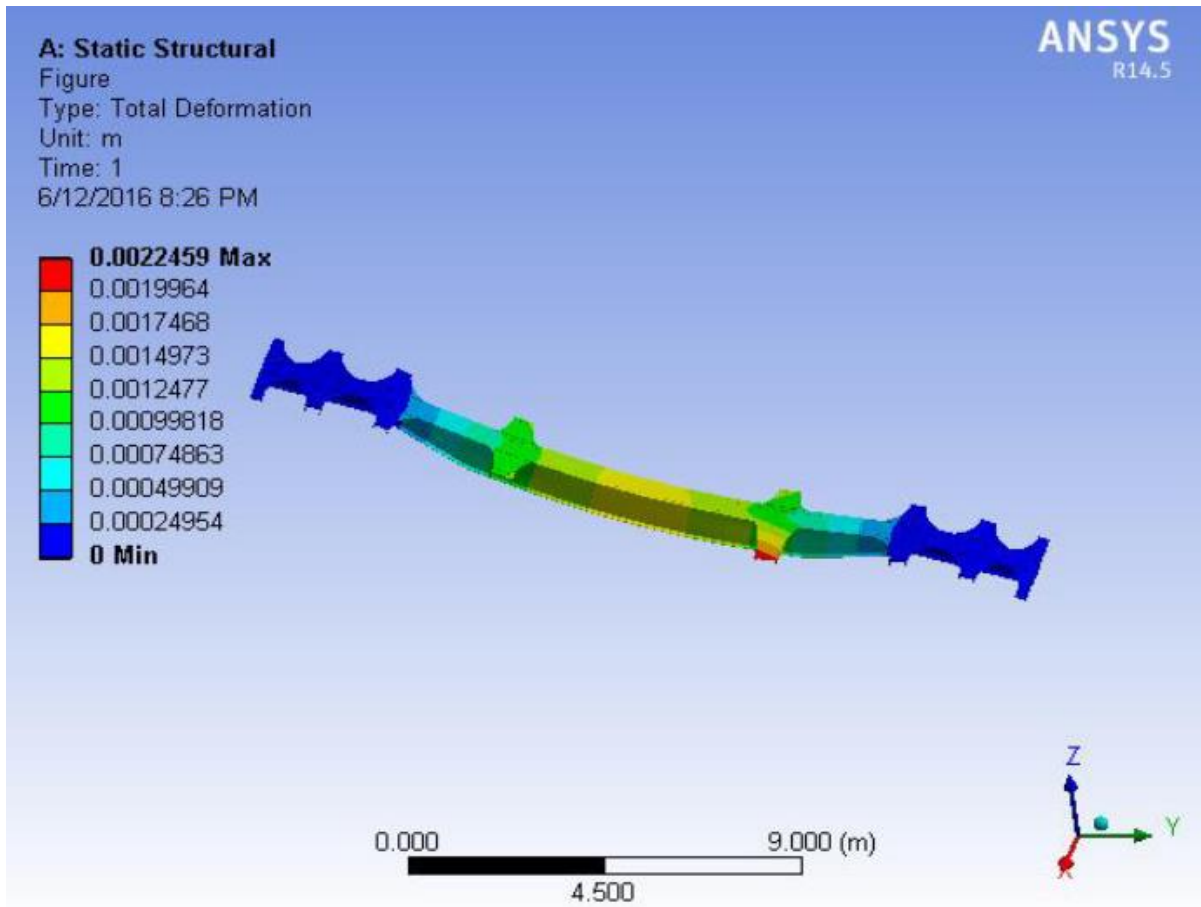


Figure 5 5: total deformation

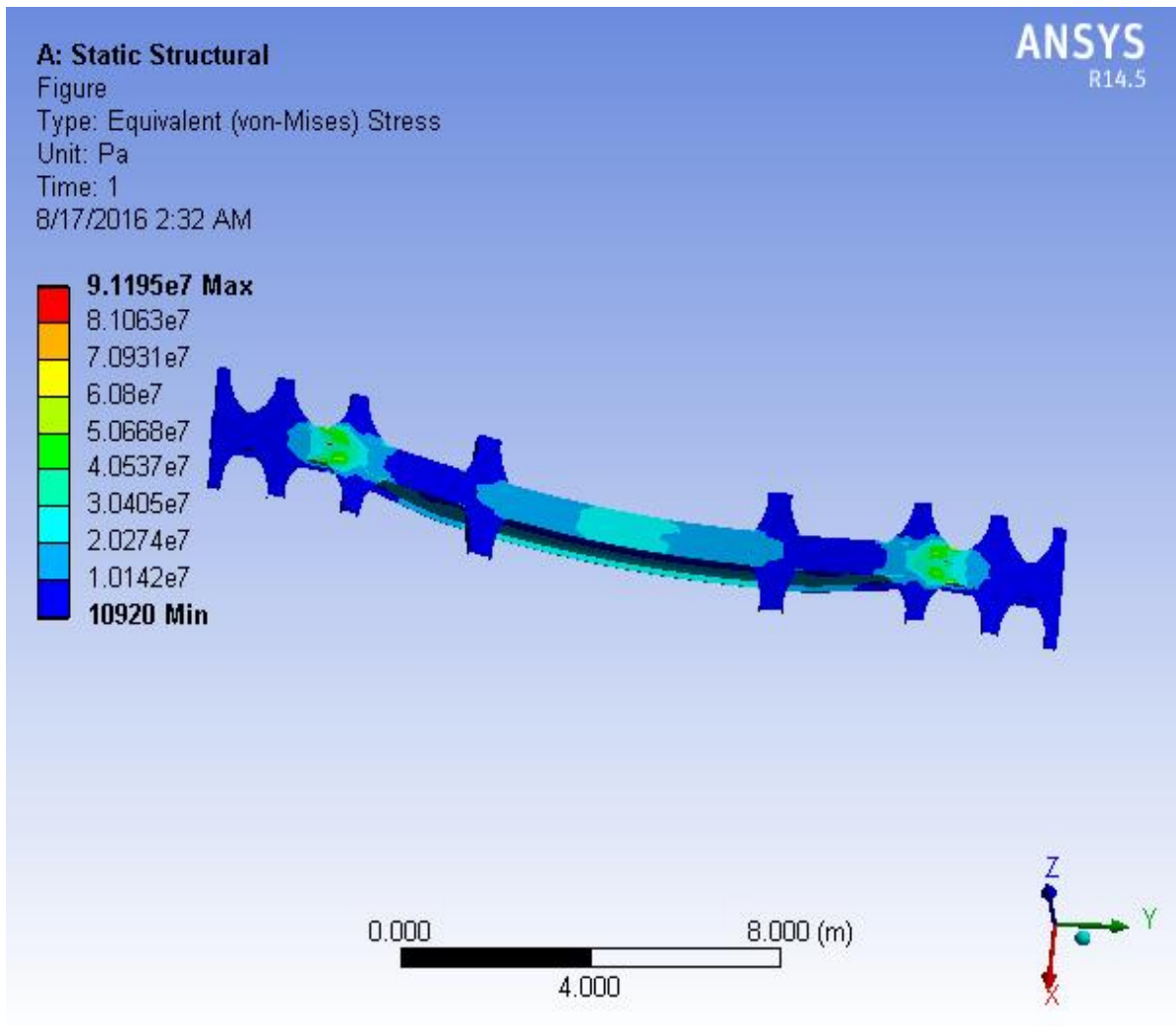


Figure 5 6: equivalent (von-mises) stress

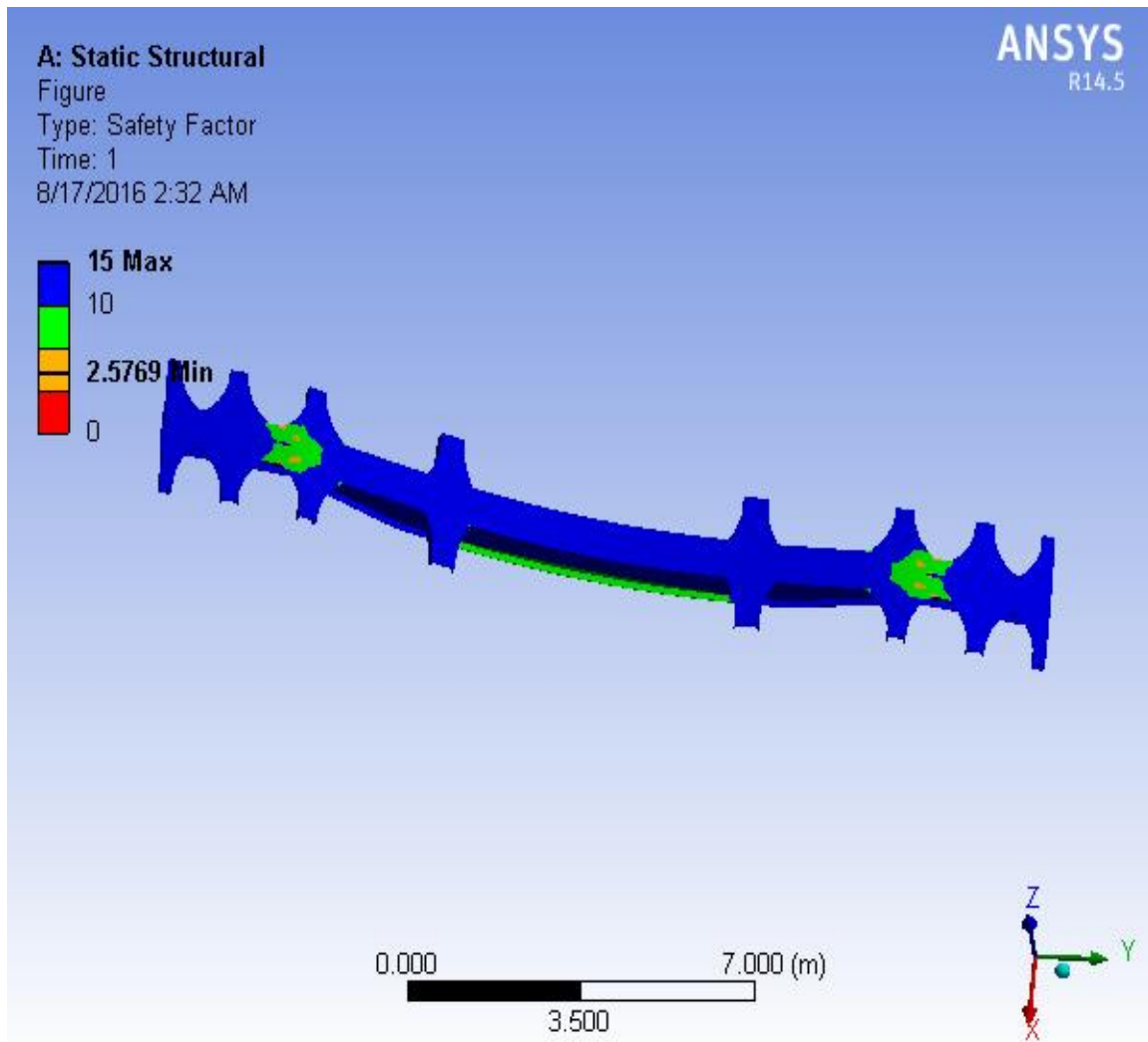


Figure 5 7: safety factor

5.5. Comparison of Static Results with optimization Results

As we have stated in section 3.1.2., the material chosen for the underframe structural optimization is structural steel (S235) with yield strength of 235Mpa and allowable maximum total deformation 0.005m (not exceeding 0.3% of the wheelbase [29]).

Concerning analytical and optimization results, maximum equivalent stresses are 95.02Mpa and 91.2Mpa with their safety factor of 2.473 and 2.577 respectively. When we

multiply each of the yield strength by their safety factor, we get 234.98Mpa and 235Mpa respectively and they are almost the same and implies that the underframe is within the safe condition.

Maximum total deformation of the optimization and analytical result is 0.00228m and 0.00225m respectively. This shows that both results are below the maximum boundary of 0.005m, which satisfies our objective function and constraints. And comparison of the static analysis results with that of the optimization results are shown in table 5.1 below.

Table 5 1: comparison of the re-modeled static analysis and optimization results

Properties	Total deformation max. (m)	Equivalent stress max. (pa)	Safety factor
Optimization results	0.0022758	9.5018138e7	2.473212
Re-modeled Static analysis Values	0.0022459	9.1195e7	2.5769
Difference in %	+1.313	+0.4	-4.2

As we see from the comparison table above, structural response of the re-modeled underframe and optimization results are almost the same. A slight difference comes due to mesh sizing and some other design errors during undergoing static analysis of the re-modeled underframe.

Chapter six

6. Conclusion and Recommendation

6.1. Conclusion

The following conclusions can be drawn from the present work:

We know that weight minimization directly affects structural stiffness and it has a significant impact on the load that we want to apply on the flat wagon underframe. The objective of this paper is to reduce weight of the underframe by determine or optimizing center sill plate thickness so that the optimum ratio between structural weight and stiffness is achieved. The optimization process pursued to determine the center sill plate thickness that minimizes the introduced objective function (weight of the underframe) by maintaining the overall bending stiffness and other parameters in their safe condition.

While reducing the vehicle structural weight by optimizing center sill plate thickness, we are reducing energy consumption and material utilization of the underframe structure to be economical in material cost. That is the optimization process helps in reducing 18.02% of the overall weight of the underframe; and this again reduces energy consumption from 9.01% to 12.614% of the overall energy consumption (i.e. a 10% reduction in mass is related to a 5% to 7% reduction in energy consumption). When we compare the material volume before and after optimization, material volume of the underframe is reduced by 16.288%, which in turn reduces the material cost of the underframe with safe design.

As a conclusion, we can say the optimization process minimizes weight of the underframe by maintaining center sill plate thickness at its optimum value and bending stiffness and other design and optimization parameters in their safest conditions. We achieve all what we say and wish in doing the optimization process; and therefore, the optimization result is convenient for the present load minimization objective at static condition.

6.2. Recommendations and Future Works

As we say above in chapter one, due to some uncertainties and limitations, this paper mainly focused on freight wagon underframe structural optimization. The optimization process is carried at static condition; It doesn't tell us whether the underframe is safe or not at dynamic conditions. To be successful in a complete design analysis of a rail ways fright vehicle wagon underframe, there should be also a clear understanding about dynamic stress analysis (fatigue analysis) and weight optimization of the underframe in relation to torsional stiffness; and it should be analyzed and checked before approving freight wagon underframe structural optimization process.

Based on uncertainties and limitations described above the following future work is possibly suggested as future work of this paper.

- Freight wagon underframe weight optimization in relation to torsional stiffness (fatigue).

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Appendix-I

Optimization results when plate thickness is considered as design variable

Table of Schematic C2: Design of Experiments (Central Composite Design : Auto Defined)

	A	B	C	D	E	F	G	H	I	J	K	
1	Name	P1 - ...	P2 - Co...	P3 - T...	P4 - ...	P5 - ...	P8...	P...	P10 - ...	P11 - E...	P12...	P1
2	45	3.8081E+08	2.4858E+08	2.4858E+08	0.29618	8250.9	43106	2.5854	11550	9.6151E+07	0.0051785	13C
3	44	3.3919E+08	2.4858E+08	2.4858E+08	0.29618	8250.9	38395	2.9026	10287	8.5643E+07	0.0046125	13C
4	43	3.8081E+08	2.2142E+08	2.4858E+08	0.29618	8250.9	38395	2.9026	10287	8.5643E+07	0.0046125	13C
5	42	3.3919E+08	2.2142E+08	2.4858E+08	0.29618	8250.9	43106	2.5854	11550	9.6151E+07	0.0051785	13C
6	41	3.8081E+08	2.4858E+08	2.2142E+08	0.29618	8250.9	38395	2.5854	10287	8.5643E+07	0.0046125	13C
7	40	3.3919E+08	2.4858E+08	2.2142E+08	0.29618	8250.9	43106	2.3028	11550	9.6151E+07	0.0051785	13C
8	39	3.8081E+08	2.2142E+08	2.2142E+08	0.29618	8250.9	43106	2.3028	11550	9.6151E+07	0.0051785	13C
9	38	3.3919E+08	2.2142E+08	2.2142E+08	0.29618	8250.9	38395	2.5854	10287	8.5643E+07	0.0046125	13C
10	37	3.8081E+08	2.4858E+08	2.4858E+08	0.26382	8250.9	38395	2.884	9313.3	8.6193E+07	0.0045983	13C
11	36	3.3919E+08	2.4858E+08	2.4858E+08	0.26382	8250.9	43106	2.5689	10456	9.6768E+07	0.0051625	13C
12	35	3.8081E+08	2.2142E+08	2.4858E+08	0.26382	8250.9	43106	2.5689	10456	9.6768E+07	0.0051625	13C
13	34	3.3919E+08	2.2142E+08	2.4858E+08	0.26382	8250.9	38395	2.884	9313.3	8.6193E+07	0.0045983	13C
14	33	3.8081E+08	2.4858E+08	2.2142E+08	0.26382	8250.9	43106	2.2881	10456	9.6768E+07	0.0051625	13C
15	32	3.3919E+08	2.4858E+08	2.2142E+08	0.26382	8250.9	38395	2.5689	9313.3	8.6193E+07	0.0045983	13C
16	31	3.8081E+08	2.2142E+08	2.2142E+08	0.26382	8250.9	38395	2.5689	9313.3	8.6193E+07	0.0045983	13C
17	30	3.3919E+08	2.2142E+08	2.2142E+08	0.26382	8250.9	43106	2.2881	10456	9.6768E+07	0.0051625	13C
18	29	3.8081E+08	2.4858E+08	2.4858E+08	0.29618	7349.1	38395	2.9026	10287	8.5643E+07	0.0046125	11!
19	28	3.3919E+08	2.4858E+08	2.4858E+08	0.29618	7349.1	43106	2.5854	11550	9.6151E+07	0.0051785	11!
20	27	3.8081E+08	2.2142E+08	2.4858E+08	0.29618	7349.1	43106	2.5854	11550	9.6151E+07	0.0051785	11!
21	26	3.3919E+08	2.2142E+08	2.4858E+08	0.29618	7349.1	38395	2.9026	10287	8.5643E+07	0.0046125	11!
22	25	3.8081E+08	2.4858E+08	2.2142E+08	0.29618	7349.1	43106	2.3028	11550	9.6151E+07	0.0051785	11!
23	24	3.3919E+08	2.4858E+08	2.2142E+08	0.29618	7349.1	38395	2.5854	10287	8.5643E+07	0.0046125	11!

Static Structural (A1)

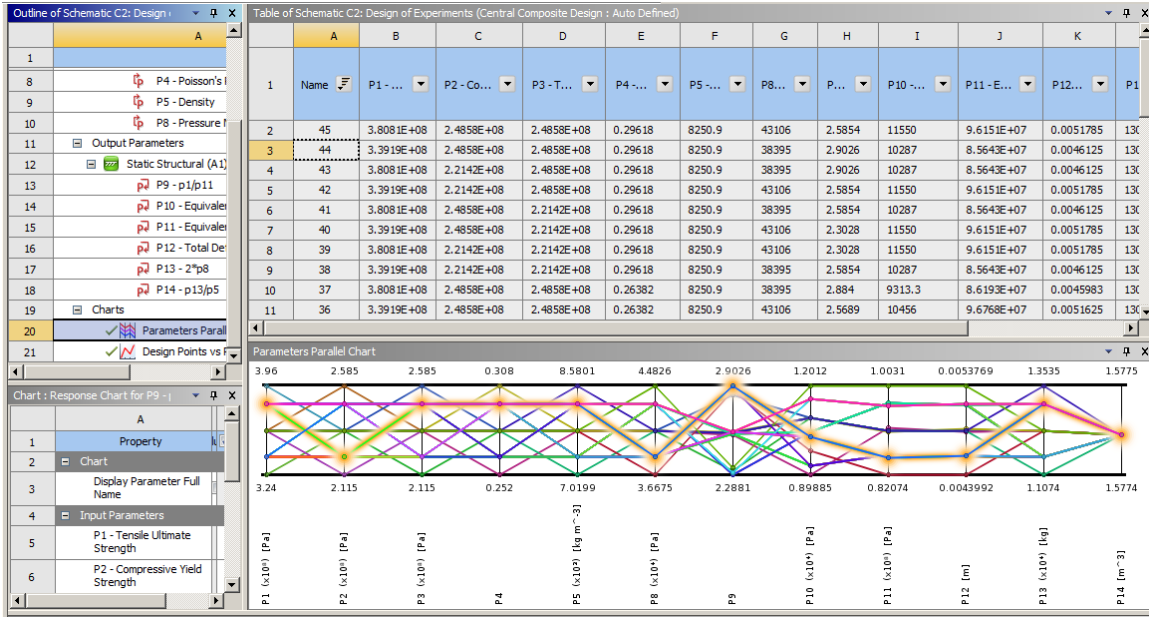
- P10 - Equivalent Stress I
- P11 - Equivalent Stress I
- P12 - Total Deformation
- P13 - 2* p_8
- P14 - p_{13}/p_5
- Min-Max Search
- Goodness Of Fit
- Response Points
- Response Point
- Response
- Local Sensitivity
- Local Sensitivity Curves
- Spider

Name	Value	Unit
Strength	3.6E+08	Pa
Yield Strength	2.35E+08	Pa
Ultimate Strength	2.35E+08	Pa
Volume	7800	kg m ⁻³
Yield Modulus	40750	Pa
New expression		

	B	C	D	E	F	G
	P9 - p1/p11	P10 - Equivalent Stress Minimum	P11 - Equivalent Stress Maximum	P12 - Total Deformation Maximum	P13 - 2* p_8	P14 - p_{13}/p_5
1	★	★ 0.99954	★ 1	★ 1	★ 1	★ 1
1	★	★ 0.99942	★ 1	★ 1	★ 1	★ 1
0	★	★ 0.72071	★ 0	★ 0	★ 0	★ 0
9.5373E-05		16.771	0.029279	1.303E-12	2.6277E-08	2.2204E-16
0	★	★ 0.15111	★ 0	★ 0	★ 0	★ 0
0.1206	✗	10.002	★ 0	★ 0	★ 0	★ 0
0.0392	★	★ 1.2015	★ 0	★ 0	★ 0	★ 0

Charts

- P13
- P14
- P9
- P10
- P11
- P12
- New output parameter



Appendix-II

Optimization parameters related to plate thickness:

plate thickness (m)	Pressure Magnitude (Pa)	Geometric weight(N)	Geometric mass(kg)	Material volume (m ³)	Stress Maximum (Pa)	safety factor	over all weight(N)	total deformation (m)	Stiffness (N/m)	weight to stiffness ratio
0.063	1282362.4	595662.35	60719.913	7.7846	59986194	3.9176	1282362.35	0.00189	6.79E+08	529.64087
0.062	1276177.4	589477.39	60089.439	7.70377	60544240.7	3.8815	1276177.39	0.0019	6.72E+08	526.9354
0.061	1269992.4	583292.44	59458.964	7.62294	61112767.9	3.8454	1269992.44	0.00191	6.66E+08	524.25744
0.06	1263807.5	577107.48	58828.489	7.54211	61692073.5	3.8092	1263807.48	0.00192	6.59E+08	521.60655
0.059	1257622.5	570922.52	58198.014	7.46128	62282467	3.7731	1257622.52	0.00193	6.53E+08	518.98234
0.058	1251437.6	564737.56	57567.539	7.38045	62884269.9	3.737	1251437.56	0.00194	6.46E+08	516.38441
0.057	1245252.6	558552.6	56937.065	7.29962	63497816.1	3.7009	1245252.6	0.00195	6.4E+08	513.81235
0.056	1239067.6	552367.65	56306.59	7.21879	64123452.6	3.6648	1239067.65	0.00196	6.33E+08	511.26579
0.055	1232882.7	546182.69	55676.115	7.13796	64761540.5	3.6287	1232882.69	0.00197	6.27E+08	508.74434
0.054	1226697.7	539997.73	55045.64	7.05713	65412455.2	3.5926	1226697.73	0.00198	6.21E+08	506.24765
0.053	1220512.8	533812.77	54415.165	6.9763	66076587.3	3.5565	1220512.77	0.00199	6.15E+08	503.77534
0.052	1214327.8	527627.81	53784.69	6.89547	66754343.6	3.5204	1214327.81	0.00199	6.09E+08	501.32706
0.051	1208142.9	521442.86	53154.216	6.81464	67446147.6	3.4843	1208142.86	0.002	6.03E+08	498.90246
0.05	1201957.9	515257.9	52523.741	6.73381	68152440.8	3.4482	1201957.9	0.00201	5.97E+08	496.5012
0.049	1195772.9	509072.94	51893.266	6.65298	68873683	3.412	1195772.94	0.00202	5.91E+08	494.12294
0.048	1189588	502887.98	51262.791	6.57215	69610354	3.3759	1189587.98	0.00203	5.85E+08	491.76737
0.047	1183403	496703.02	50632.316	6.49132	70362954.2	3.3398	1183403.02	0.00204	5.79E+08	489.43414

0.046	1177218.1	490518.07	50001.842	6.41049	71132006	3.3037	1177218.07	0.00205	5.73E+08	487.12295
0.045	1171033.1	484333.11	49371.367	6.32966	71918054.6	3.2676	1171033.11	0.00206	5.68E+08	484.83348
0.044	1164848.2	478148.15	48740.892	6.24883	72721669.9	3.2315	1164848.15	0.00207	5.62E+08	482.56544
0.043	1158663.2	471963.19	48110.417	6.168	73543447.4	3.1954	1158663.19	0.00208	5.57E+08	480.31851
0.042	1152478.2	465778.23	47479.942	6.08717	74384009.9	3.1593	1152478.23	0.00209	5.51E+08	478.09241
0.041	1146293.3	459593.28	46849.468	6.00634	75244008.9	3.1232	1146293.28	0.0021	5.46E+08	475.88685
0.04	1140108.3	453408.32	46218.993	5.92551	76124126.4	3.0871	1140108.32	0.00211	5.4E+08	473.70155
0.039	1133923.4	447223.36	45588.518	5.84468	77025076.9	3.051	1133923.36	0.00212	5.35E+08	471.53623
0.038	1127738.4	441038.4	44958.043	5.76385	77947608.7	3.0148	1127738.4	0.00213	5.29E+08	469.39061
0.037	1121553.4	434853.44	44327.568	5.68302	78892506.8	2.9787	1121553.44	0.00214	5.24E+08	467.26443
0.036	1115368.5	428668.49	43697.093	5.60219	79860594.6	2.9426	1115368.49	0.00215	5.19E+08	465.15742
0.035	1109183.5	422483.53	43066.619	5.52136	80852736.2	2.9065	1109183.53	0.00216	5.14E+08	463.06933
0.034	1102998.6	416298.57	42436.144	5.44053	81869839.6	2.8704	1102998.57	0.00217	5.08E+08	460.99991
0.033	1096813.6	410113.61	41805.669	5.3597	82912858.7	2.8343	1096813.61	0.00218	5.03E+08	458.9489
0.032	1090628.7	403928.65	41175.194	5.27887	83982796.8	2.7982	1090628.65	0.00219	4.98E+08	456.91605
0.031	1084443.7	397743.7	40544.719	5.19804	85080709.6	2.7621	1084443.7	0.0022	4.93E+08	454.90114
0.03	1078258.7	391558.74	39914.245	5.11721	86207708.9	2.726	1078258.74	0.00221	4.88E+08	452.90392
0.029	1072073.8	385373.78	39283.77	5.03638	87364966	2.6899	1072073.78	0.00222	4.83E+08	450.92416
0.028	1065888.8	379188.82	38653.295	4.95555	88553716	2.6538	1065888.82	0.00223	4.79E+08	448.96163
0.027	1059703.9	373003.86	38022.82	4.87472	89775262.2	2.6176	1059703.86	0.00224	4.74E+08	447.01611
0.026	1053518.9	366818.91	37392.345	4.79389	91030980.8	2.5815	1053518.91	0.00225	4.69E+08	445.08738
0.025	1047333.9	360633.95	36761.87	4.71306	92322326.1	2.5454	1047333.95	0.00226	4.64E+08	443.17522

0.024	1041149	354448.99	36131.396	4.63223	93650836.1	2.5093	1041148.99	0.00227	4.59E+08	441.27943
0.023	1034964	348264.03	35500.921	4.5514	95018138.6	2.4732	1034964.03	0.00228	4.55E+08	439.39978
0.022	1028779.1	342079.08	34870.446	4.47057	96425957.9	2.4371	1028779.08	0.00229	4.5E+08	437.53608
0.021	1022594.1	335894.12	34239.971	4.38974	97876121.9	2.401	1022594.12	0.0023	4.46E+08	435.68812
0.02	1016409.2	329709.16	33609.496	4.30891	99370570.5	2.3649	1016409.16	0.0023	4.41E+08	433.8557
0.019	1010224.2	323524.2	32979.022	4.22808	100911363	2.3288	1010224.2	0.00231	4.36E+08	432.03863
0.018	1004039.2	317339.24	32348.547	4.14725	102500691	2.2927	1004039.24	0.00232	4.32E+08	430.23672
0.017	997854.29	311154.29	31718.072	4.06642	104140882	2.2566	997854.285	0.00233	4.28E+08	428.44978
0.016	991669.33	304969.33	31087.597	3.98559	105834418	2.2204	991669.327	0.00234	4.23E+08	426.67762
0.015	985484.37	298784.37	30457.122	3.90476	107583946	2.1843	985484.369	0.00235	4.19E+08	424.92006