ISTANBUL TECHNICAL UNIVERSITY ★ GRADUATE SCHOOL OF SCIENCE ENGINEERING AND TECHNOLOGY

ANALYSIS AND PREDICTION OF NOISE PROPAGATION FOR COMMERCIAL VEHİCLE REAR AXLE DESİGN

M.Sc. THESIS

Mustafa YILDIRIM

Department of Aeronautics and Astronautics Engineering

Interdisciplinary Aeronautical and Astronautical Engineering Graduate Program

MAY 2015

ISTANBUL TECHNICAL UNIVERSITY ★ GRADUATE SCHOOL OF SCIENCE ENGINEERING AND TECHNOLOGY

ANALYSIS AND PREDICTION OF NOISE PROPAGATION FOR COMMERCIAL VEHICLE REAR AXLE DESIGN

M.Sc. THESIS

Mustafa YILDIRIM
(511111127)

Department of Aeronautics and Astronautics Engineering

Interdisciplinary Aeronautical and Astronautical Engineering Graduate Program

Thesis Advisor: Prof. Dr. Vedat Ziya DOĞAN

MAY 2015

İSTANBUL TEKNİK ÜNİVERSİTESİ ★ FEN BİLİMLERİ ENSTİTÜSÜ

TİCARİ ARAÇLARIN ARKA AKS TASARIMINDA GÜRÜLTÜ OLUŞUMUNUN ANALİZİ VE TAHMİNİ

YÜKSEK LİSANS TEZİ

Mustafa YILDIRIM
(511111127)

Uçak ve Uzay Mühendisliği Anabilim Dalı

DAP Uçak ve Uzay Mühendisliği Yüksek Lisans Programı

Tez Danışmanı: Prof. Dr.Vedat Ziya DOĞAN

MAYIS 2015

Mustafa Yıldırım, a **M.Sc.** student of **ITU Graduate School of Science Engineering and Technology** student ID 511111127, successfully defended the **thesis** entitled “**ANALYSIS AND PREDICTION OF NOISE PROPAGATION FOR COMMERCIAL VEHICLE REAR AXLE DESIGN**”, which he prepared after fulfilling the requirements specified in the associated legislations, before the jury whose signatures are below.

Thesis Advisor :	**Prof. Dr. Vedat Ziya DOĞAN** İstanbul Technical University	....
Jury Members :	**Prof. Dr.Metin Orhan KAYA** İstanbul Technical University	....
	Prof. Dr.Ata MUĞAN İstanbul Technical University	....

Date of Submission : 04 May 2015
Date of Defense : 29 May 2015

To my advisors,friends and family,

FOREWORD

First of all, I would like to thank my supervisor Prof.Dr.Vedat Ziya Doğan for his advice and supportsduring my thesis's process and I also thank my friends Arda Küçüksarı and Hare Dursun for their helps during my education. Additionally I am gratefull to my supervisor Selçuk Tabak for his guidance.Last but not least, I thank my parents for their supports and endeavour on me.

May 2015

Mustafa YILDIRIM
Makine Mühendisi

TABLE OF CONTENTS

Page

ABBREVIATIONS

C	**:** Damping Coefficient
CAD	**:** Computer Aided Design
CAE	**:** Computer Aided Engineering
CCW	**:** Counter Clockwise
CFD	**:** Computational Fluid Dynamics
DOF	**:** Degree of Freedom
FDR	**:** Final Drive Ratio
FEA	**:**Finite Element Analysis
FEM	**:**Finite Element Method
FMC	**:** Ford Motor Company
FWD	**:** Front Wheel Drive
GUI	**:** Graphical User Interface
K	**:** Stifness Matrix
M	**:** Mass Matrix
MBD	**:**Multi Body Dynamics
N&V	**:** Noise and Vibration
NVH	**:**Noise Vibration Harshness
OEM	**:**Original Equipment Manufacturer
RSS	**:**Root Squaire Sum
RWD	**:** Rear Wheel Drive
TE	**:** Transmission Error
ω	**:** Natural Frequency

LIST OF TABLES

Page

LIST OF FIGURES

Page

ANALYSIS AND PREDICTION OF NOISE PROPAGATION FOR COMMERCIAL VEHICLE REAR AXLE DESIGN

SUMMARY

Noise is one of the major and subsequent problem for automotive industry and prevention of this problem sometimes can be easy if it is component based or isolation problem but if the problem is caused by design and additionally if it is hard to predict this problem's effects on unmanufactured vehicle and it is always hard to prevent this problem because of automotive design duration and phase layout.Therefore it is significant to predict the design sourced problems before the design frozen and vehicle manufactured.

In the absence of prototypes, analytical methods such as finite element analysis are very useful in resolving noise and vibration problems, by predicting dynamic behavior of the automotive components and systems. Finite Element Analysis (FEA) is a simulation technique and involves making assumptions that affect analytical results.

This study handed out an encountered problem of an vehicle rear axle system components design parameters and try to predict this noise propagation according to experienced targets values.

Ford Transit medium commercial Jumbo bus vehicle model investigated for this study.According to customer expectations and marketing strategy, new axle variant(Single Rear Wheel-SRW) analysed in nvh perspectives.Since the most important nvh problem is gear whine for a rear axle, the connection between gears modeled specifically with using Hypoid gear simulation tool(HGSiM). Modal and FRF analysis were performed at Optistruct tool.

Results compared with previous axle model(Double Rear Wheel-DRW) and accordingly compared with test results for future studies.As design effect to noise and vibration, rear axle tube thickness and final drive ratio are investigated and compared.

This study help to understand a vehicle design related nvh problem and direct the design to prevent issues before the vehicle manufactured. Numerical studies make benefits to manufacturer as a cost and time saving improvements.Therefore this kind studies are very important for automotive industry.

TİCARİ ARAÇLARIN ARKA AKS TASARIMINDA GÜRÜLTÜ OLUŞUMUNUN ANALİZİ VE TAHMİNİ

ÖZET

Gürültü günümüzde önemini gittikçe arttıran ve insanların konfor algısının bir parçası haline gelen bir etmendir. Bu konfor algısı ile insanların daha iyiye sahip olma isteği gün geçtikçe artmaktadır. Bu durum dolayısıyla mühendislik disiplinlerine yansımakta, yeni disiplinler doğmakta ve önem kazanmaktadır.Bu kapsamda üretilen ürünlerin titreşim karakteristiği ve dolayısyla oluşturabileceği gürültü ürünlerin tasarım aşamasında mühendisler tarafından incelenmeli ve irdelenmelidir.

Titreşim ve gürültü karakteristiği parçanın, sistemin tasarımı ve malzemesi ile doğrudan ilişkilidir. Bu ilişkiler parça bazında malzeme özellikler ve geometrik şekli ihtivasıyla parçanın katılığı, esnekliği, sönüm karakteristiği bakımından incelenmektedir. Bu incelemeler yapılırken çeşitli yaklaşımlar ile yeni metodolojiler geliştirilmeye çalışılmaktadır. Bu metodolojiler parça ve sistemin doğal frekanslarını belirleme ve parçayı gerekli durumlarda daha katı veya esnek olması için mühendise, tasarımcıya yardımcı olmaktadır. Bu yaklaşımlar genel olarak test ve analiz gibi iki kısma ayrılmaktadır. Bu iki yaklaşım çeşitli mühendislik uygulamalarında tasarımın doğruluğu ve korelasyonu bakımından uygulamaya geçmekte olan ve üzerine sürekli çalışılarak geliştirlmekte olan yaklaşımlardır.

Parçaların karakteristiği doğal frekansları parça titreşimlerini ve yapıların tahrik kuvvetlerine karşı tepkisini etkileyen en önemli kriterdir.Bu karakteristik özellikler şekil ve malzeme özelliklerine bağlı olarak değişmektedir ve bu özellikler genel olarak kütle, atalet, katılık ve sönüm karakteristiği olarak ayrılmaktadır. Bu özellikler parçaların doğal frekanslarına etki eden etmenlerdir. Parça ve sistemlerin doğal frekansları çekiç testi ile kolaylıkla bulunabilmekte ve parça karakteristiği belirlenebilmektedir ancak test maliyetleri ve imkanları bu opsiyonu zaman zaman kısıtlamaktadır. Test yerine daha hızlı, kolay ve az maliyetli bir çözüm olan bilgisayar destekli analizler gün geçtikçe daha da önem kazanmaktadır ve bu doğrultuda yeni metodolojiler ve yaklaşımlar geliştirilerek bu sektörün ileride test verifikasyonlarının yerine alması beklenmektedir.

Otomotiv endüstrisi için gürültü oluşumu önemli ancak sonradan anlaşılan bir problemdir, problemi önceden tahmin etmek normal şartlarda çok güçtür.Tasarım aşamasında öngörülemeyen gürültü, eğer problem izolasyon malzemeleri ile önlenebilecek ise çözümü kolaydır ancak problem bu şekilde önlenemezse probleme sebep olan tasarımın değişikliği oldukça zor ve masraflıdır.Bu değişiklikler otomotiv imalatçısının maliyetlerini ciddi şekilde arttırmaktadır. Bundan dolayı problemi daha tasarım aşamasında ve araç imalatından önce tahmin etmek ve önlemek maliyet ve zaman kaybı açısından çok önemlidir.

Araç üretim süreçleri tasarım fazı ile başlar ve tasarımın doğruluğu ileriki aşamalarda üretilen prototip araçlar ile ve öncesinde bilgisayar destekli araçlar ile sağlanır.Prototip araçlar imal edilmeden önce sonlu elemanlar gibi yöntemler ile araç aktarma organlarının parça ve sistem dinamik davranışlarının tahmini, titreşim ve gürültü problemlerin çözümünde çok kullanışlıdır.Sonlu elemanlar analizi analitik sonuçları etkileyen parametreleri ve kabul kriterleri kolaylıkla simüle edilebilir.

Böylelikle sistem ve parça etkileşimleri daha tasarım fazında üreticiye bilgi verir ve parça, araç imal edilmeden gerekli aksiyonların alınmasına olanak sağlar.

Bilgisayar destekli analizler ile otomotiv endüstrisi sürekli gelişmekte ve ilerlemektedir. Bu gelişim öncelikli olarak tasarım aşamasında başlamakta ve sonrasında tasarlanan parçaların analizlerinde kendini göstermektedir.Bu çalışma kapsamında analizler yapılarak parça bazlı olabilecek problemler önceden gün yüzüne çıkartılmıştır.Bu analizler genel olarak parça karakteristiğini belirlemek için yapılan modal analiz ve frekans cevap fonksiyonu analizidir.Modal analiz ile parça ve sistemlerin doğal frekansları ve mod şekilleri bulunmaktadır.Ancak karmaşık sistem ve yapılarda bu doğal frekansların hangi parça ve sisteme ait olduğunu anlamak zor olabilir, parçalar ilişkide olduğu parçalardan tahrik olabilir ve doğal frekansı varmış gibi gözükebilir. Bu durumun önüne geçmek ve parça modlarını birbirinden ayırt etmek için bu analizlere ek olarak frekans cevap analizi yapılmaktadır. Bu analiz ile parça doğal frekansları faz açıları da incelenerek doğru bir şekilde elde edilebilmektedir.

Bu çalışmada izlenen yol öncelikli olarak parçaların doğal frekanslarını bulmak ve bu değerleri birbiri ile olabildiğince uzaklaştırmak üzerine kurulmuştur. Bu kapsamda araçlara uygulanmak istenen tek tekelekli arka aks sitemi modellenmiş ve akabinde sistem bazlı analizler ile sonlu elemanlar modeli test sonuçlarına göre uyarlanmıştır. Parça bazlı test sonuçlarına göre yapılan analiz benzetimleri daha karmaşık yapıda olan sistemlerin(aktarma organları) analizinin yapılmasına ve doğruluğunun sağlanmasında önemlidir. Doğruluğu istenilen seviyeye gelen sonlu elemanlar modeli daha sonra ile parça bazlı analiz benzetimleri baz alınarak elde edilen benzetimler ile araç aktarma organlarının sonlu elemanlar modeli kurulmuş ve analizler gerçekleştirilmiştir.

Bu çalışmada arka aks sisteminde kullanılan parçaların tasarım parametreleri irdelenmiş ve bu parametrelerden aks kovan kalınlığının, aks boyunun ve diferansiyel sistemin parçası olan hipoit dişlinin analizleri yapılarak gürültü oluşumuna etkisi analizler vasıtasıyla incelenmiş ve oluşturulabilecek gürültü daha önceden analiz ve testleri yapılmış baz araç modeli ile kıyaslama yapılarak gürültü tahmini yapılmıştır.

Orta ticari araç olarak Ford Transit modeli ele alındı.Bu araç uzun şasi boyuna sahip yüksek tavanlı modeli test edildi.Bu aracın şasi ve karoser modeli analzilere etki etmeyeceğinden dolayı modellenmedi ancak tüm aktarma organları titreşim ve frekans değerlerini etkileyecek tüm parçalar dahil edilerek hazırlandı.

Bu çalışma kapsamında yeni tasarlanan tek tekerli arka aks sistemi, titreşim ve gürültü perspektifinde sonlu elemanlar olarak modellendi ve analiz edildi.Arka aks gürültüsünde en etkili parça olan dişli sistemi ve bağlantısı kinematik bir hipoit dişli simülasyon aracı ile modellendi. Optistruct programı ile sistemlerin doğal frekansları bulundu ve frekans cevap fonksiyonu analizi gerçekleştirilerek bu frekansların parça bazlı indirgenmesi ve parçaların ilk frekans değerlerinin ayrıştırması sağlandı.

Analiz sonuçları test sonuçları ile karşılaştırılarak model benzetimleri ve doğruluğu sağlandı ve akabinde mevcut çift tekerli aks ile karşılaştırıldı.Tasarımın arka aks titreşim ve gürültüsüne etkisi olarak, aks kovan kalınlığı ve dişli aktarım oranları araştırıldı ve etkileri karşılaştırıldı.

Bu çalışma araç aktarma organları arka aks sistem kaynaklı gürültü problemini anlamaya, tahmin etmeye ve tasarıma araç daha üretilmeden yön vermeye katkı sağlamıştır.Bunun gibi sayısal hesaplamalar ile otomotiv imalatçıları zaman ve özellikle para kaybının önüne geçebilir ve problem daha ortaya çıkmadan çözüme kavuşturulabilir.Bu bakımdan bu tür çalışmalar otomotiv sektöründe büyük öneme sahiptir.

1. INTRODUCTION

Automotive industry is maintaining itself presence deliberatley within time and importance of this industries sustain its challenge.Most of the improvements of the automotive interior and exterior parts specifications increasing up the automotive sector’s standards such as performance,comfort,qualityetc.New motodologys are entering the this endustries to obtain more cheap and fast results as possible.

Real vehicle tests could not be possible to perform in everytime even tough their results are precise.Therefore new metodologys are need to be developed to obtain right results without using any vehicle and devices.

This thesis focus on medium commercial vehicle rear axle design effect to noise propagation with using finite element method.Powertrain systems are modeled and analysed accordingly results evaluated and compared with vehicle test.Correlation of base vehicle variant help to model others variations to observe the most convenient condition in nvh perspective.

1.1 Purpose of Thesis

This study is mainly focused on developing a methodology for commnercial vehicle powertrain systems desıng using numerical calculations.Design and manufacture phase of an automotive is quite interminable duration therefore before the last design phase , manufacturer want to be able to anticipate and designed part problems of vehicles otherwise produced vehicles can not be corresponds customer expectations.To satisfy customer expectation and to prevent late design change is possible via developing CAE calculations for vehicles powertrain component.

The specific aim of this study is foresee and predict the nvh problems of commercial vehicle powertrains, especially rear axle sourced noise.The design parameters investigated according to contribution of them to noise propogation.

1.2 Literature Review

The essential idea of modal analysis is to describe complex phenomena in structural dynamics with simple constituents like natural vibration modes. This idea is very much like atomism which attempts to find the most basic elements for varieties of different substances, or the concept is of Fourier series which represents a complicated waveform by a combination of simple sine and cosine waves. In this sense, the origin of modal analysis may be traced way back in history. However, there are two landmarks of such atomistic comprehension in the recent history of science which paved the way for the birth of modal analysis. Newton, from his observation of the spectrum of sunlight, confirmed its composition of colour components. Fourier, based on earlier mathematical wisdom, claimed that an arbitrary periodic function with finite interval can always be represented by a summation of simple harmonic functions. The Fourier series and spectrum analysis laid a solid foundation for the development of modal analysis.

Theoretical modal analysis can be closely identified with the wave equation which describes the dynamics of a vibrating string. From the solution, we can determine its natural frequencies, mode shapes and forced responses- constituents so accustomed by today's modal analysts. This stage of modal analysis, developed during the nineteenth century, was largely dependent upon mathematics to solve partial differential equations which describe different continuous dynamic structures. The elegance of the solutionis evident while the scope of solvable structures is limited.

The concept of discretization of an object and introduction of matrix analysis brought about a climax in theoretical modal analysis early in the last century. Theory was developed such that structural dynamic analysis of an arbitrary system can be carried out when knowing its mass and stiffness distribution in matrix forms. However, the theory could only be realized after computers became available. In that aspect, theoretical(or analytical) modal analysis is very much numerical modal analysis.

The foundation of experimental modal analysis, a name contrived long after the engineering practice it embodies, was laid early in the last century. The core of experimental modal analysis is system identification. As a result, it was nourished by the development in electrical engineering. The analogy of electric circuits and a mechanical system enabled the application of some circuit analysis theory into the study of mechanical systems. This gave forth system identification, mechanical impedance and sub-system analysis in structural dynamics.The invention of the fast Fourier transform (FFT) algorithm by J.W. Cooly andJ.W. Tukey in 1965 finally paved the way for rapid and prevalent application of experimental technique in structural dynamics. With FFT, frequency responses of astructure can be computed from the measurement of given inputs and resultant responses.

The theory of modal analysis helps to establish the relationship between measured FRFs and the modal data of the tested specimen. Efforts were focused on deriving modal data from measured FRF data. The first, and perhaps the most significant method of experimental modal analysis was proposed by C.C. Kennedy and C.D.Pancu in 1947 before FFT was conceived. Their method was largely forgotten until FFT gave life to experimental modal analysis. Since then, numerous methods have been proposed and many have been computerized, including those time domain methods which are based on free vibration of a structure rather than its frequency responses.

The experimental development also helped to advance the theory of modal analysis.Traditional analytical modal analysis based on the proportional damping model was expanded into the non-proportional damping model. The theory of complex vibration modes was developed. Modal analysis evolves more in parallel with control theory.Inverse structural dynamic problems such as force identification from measured responses were actively pursued. Nonlinear dynamic characteristics were studiedexperimentally.

Today, modal analysis has entered many fields of engineering and science.Applications range from automotive engineering, aeronautical and astronautical engineering to bioengineering, medicine and science. Numerical (finite element) and experimental modal analysis have become two pillars in structural dynamics[1].

While modal analysis and frequency response analysis widely used at many different areas, this study focused it for powertrain sytems, driveline systems and mostly for rear axle. At recent years there are many projects for rear axle but this thesis handed in some of literature studies as guiding light.

Rear axle noise prediction according to design parameters firstly investigated by Hussien A Hussien and Ahmed A. Shabana at 1998 and 1999, their study is including the finite element and dynamic model of rear axle eigen values and mode shapes change investigation and understanding effect of them to modal frequencies values[2].And also Hirasaka and Sugita et.al. investigated rear axle pinion and ring gear interaction and transmitting error effect to gear whine noise, with the help of more fine accurate modelling of gears, components contribution to noise investigated deeply as design effect[15].

2. POWERTRAIN

The term power train refers to the group of components that generate power and deliver it to the driving wheels. It includes the energy generating engine, the clutch, the transmission, the various drive shafts and the differential. The driveline is the portion of a vehicle after the transmission which changes depending on whether the vehicle is front-wheel drive, rear-wheel drive, or four-wheel drive.Figure 2.1 represent a vehicle powertrain systems for rear wheel drive.

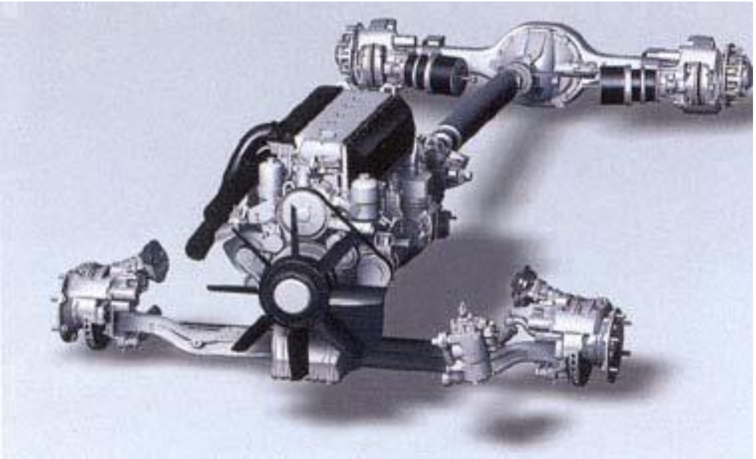

Figure 2.1 : Powertrain systems of a vehicle

A schematic representation is shown at Figure 2.2 which consist of mainly 4 systems(Engine,transmission,drive shaft and rear axle).Those systems can vary according vehicle to vehicle such as driveshafts numbers can be 3 if vehicle is long or driveshaft can be replaced by linkshaft if vehicle front wheel drive.

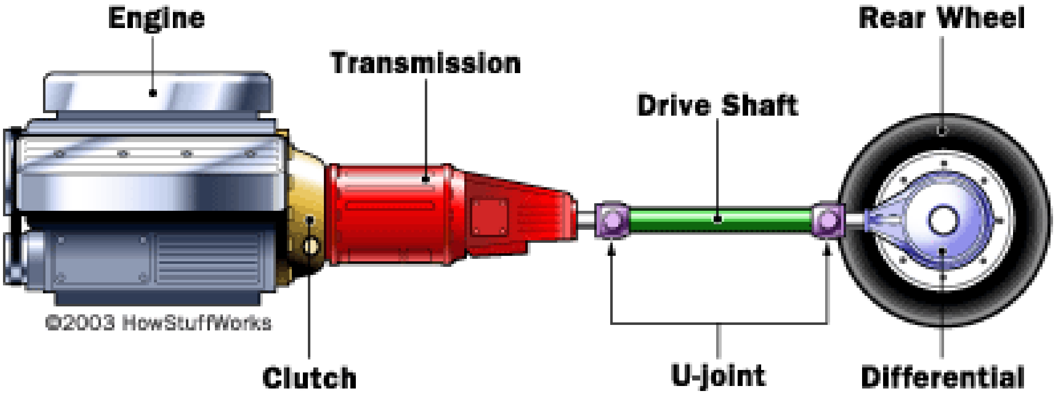

Figure 2.2 : Powertrain systems side view[3]

East-West based powertrain system is mostly used at passenger car. This E-W system where traction performed by front wheel(FWD), is more efficient for fuel economy because of compact system provide less weight contribution to vehicle. Also it is very good for packaging and this provide more space to vehicle interior volume. Figure 2.3 shows the FWD and RWD variant of a vehicle.

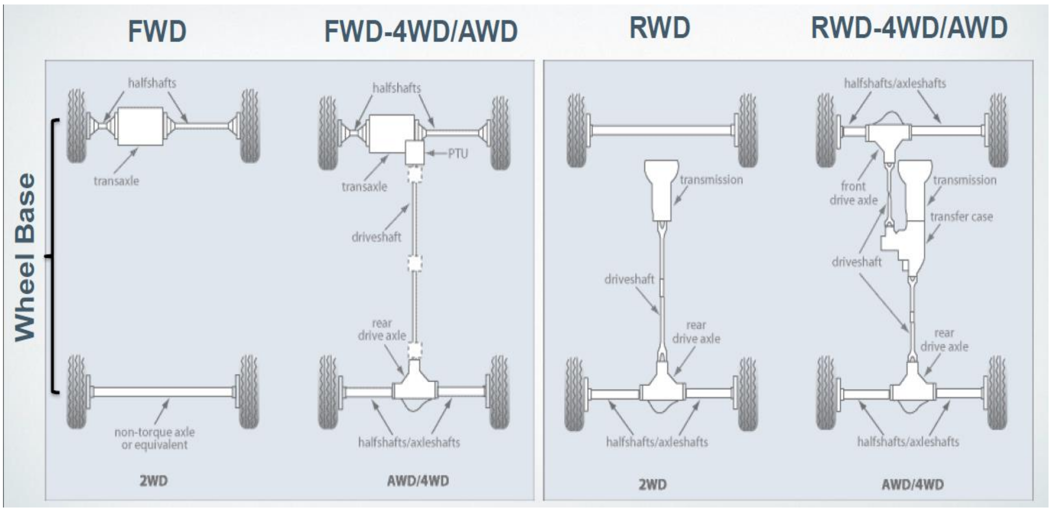

Figure 2.3 : Powertrain of a vehicle for AWD and RWD[4]

Rear wheel drive system generally used at commercial vehicle.This system located as North-South, where North is front of vehicle and South is rear of vehicle.This NS provide more balanced weight distrubution to wheels and increase the durability performance of vehicle.More payload increase the traction force on rear wheel drive(RWD) that increase the acceleration.

Since this study consist of a commercial vehicle rear axle noise,RWD system is mainly investigated for this purpose.

2.1 Engine

Internal combustion engines are devices that generate work using the products of combustion as the working fluid rather than as a heat transfer medium. To produce work, the combustion is carried out in a manner that produces high-pressure combustion productsthat can be expanded through a turbine or piston.

There are three major types of internal combustion engines in use today: (1) the spark ignition engine, which is used primarily in automobiles; (2) the diesel engine, Figure 2.4 shows the a sample engine for Diesel,which is used in large vehicles and industrial systems where the improvements in cycleefficiency make it advantageous over the more compact and lighter-weight spark ignitionengine; and (3) the gas turbine, which is used in aircraft due to its high power/weightratio and also is used for stationary power generation[5].

Engine is the main excitation source of all components and this also make it noise generator for a vehicle.There are mainly two kind of noise types for engine which are combustion related explosion and mechanical component interactions related noise.

This study is not focused on engine noise but rear axle noise excitation comes directly from engine.For that reason engine is included at analysis model due to it is excitation source.

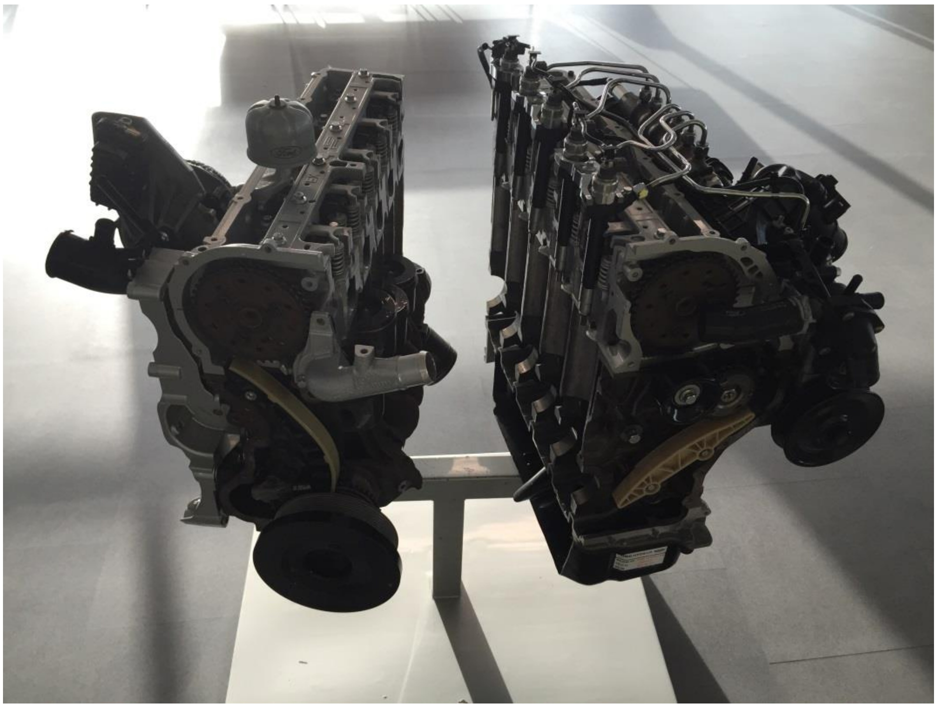

Figure 2.4 : Cutaway I5 Transit engine

2.2 Transmission

The function of the vehicle transmission is to transfer engine power to the driving wheels of the vehicle. Changing gears inside the transmission allows matching of the engine speed and torque with the vehicle's load and speed conditions. In manual transmissions, the driver must shift from gear to gear, whereas in automatic transmission the shifting is performed by a control system. There has been a gradual refinement in gearbox design over recent decades and a move towards an increasing number of gear ratios to improve overall performance and efficiency.Figure2.5 represent the commercial vehicle transmission.

Vehicles are traditionally equipped with gearboxes and differentials. The number of gears in vehicle transmissions range from three for older cars to five, six and even eight in newer ones. The differential provides a constant torque amplification ratio (final drive) and acts as a power split device for left and right wheels. The role of a gearbox is to provide different torque amplification ratios from the engine to the wheels when necessary at different driving conditions [7].

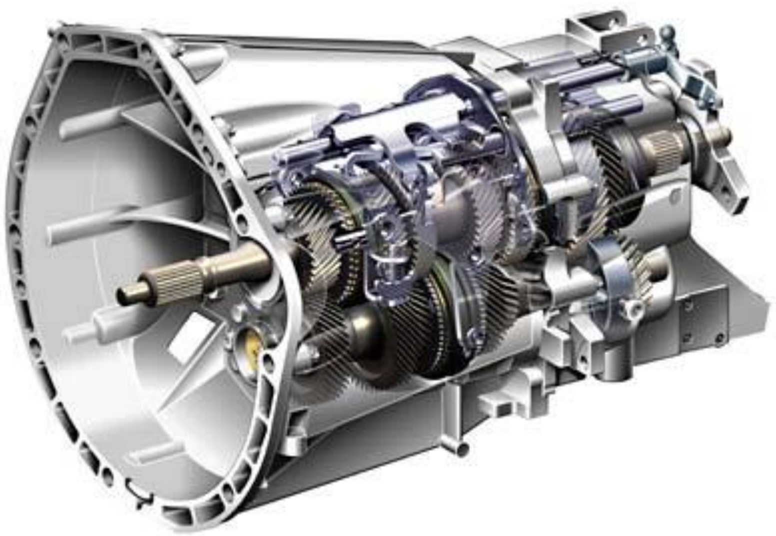

Figure 2.5 : Transmission[8]

2.3 Propshaft

Propshaft is the system which transmits torque from transmission to rear axle whilst compensating suspension movements on RWD vehicles. It is consist of few shafts, center bearing, and universal joints. Figure 2.6 represent the Transit propshaft system and its connection with other systems.

It is mainly provide below contribution to powertrain system;

- Transmits torque – through lifetime including bump stop torques
- Absorbs displacements coming from suspension system and engine roll movement
- Adjusts alignment difference between engine & transmission and rear axle

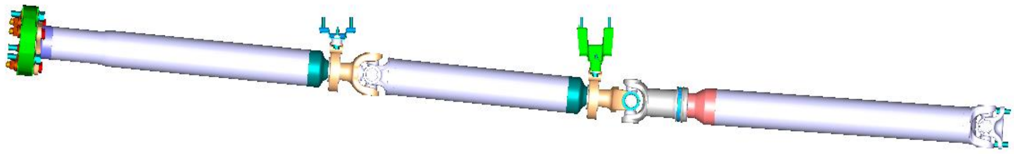

Figure 2.6 : Propshaft of Transit vehicle

2.4 Rear Axle

Rear axle system transmitt the torque and force from propshaft to wheels and also provide connections from wheels to frame via leaf spring and absorbers.Figure2.7 represnet the rear axle assembly.In most motor vehicles the rear axle has a housing, tubes for the axle shafts, a final drive.The final drive and axle shafts are part of the motor vehicle transmission. Wheel brakes are mounted on the rear axles, as are the hubs for the wheels. The rear axle is connected to the frame or body of motor vehicle by a suspension. When the rear wheels are independently suspended, the differential is attached directly to the frame of the motor vehicle, and in this case, the pivoting axle shafts have ball and socket joints. The housing of the rear axle is a hollow supporting beam and is the axle of the automobile[10].

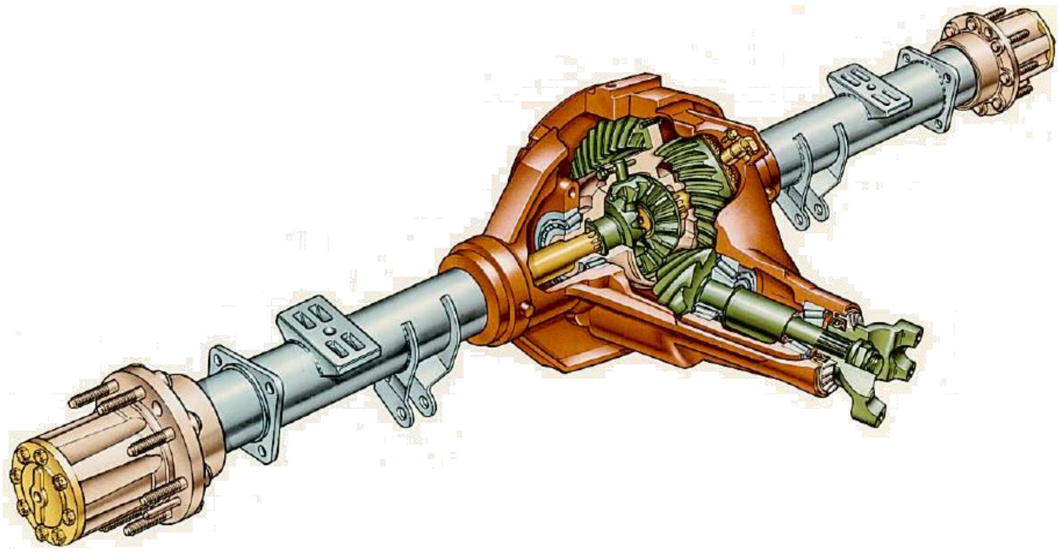

Figure 2.7 : Rear axle of a commercial vehicle[11]

Many NVH problems generally sourced by rear axle system especially diferential which changes direction of rotation.This rotation axis change is generated by pinion gear and ring gear contact.This connection is very important in terms of NVH.The interactions are the major source of gear whine/rattle noise while vehicle is ongoing.

Commercial vehicles rear axle can be sorted out in different variants such as single reduction, double reduction or single rear axle, double rear axle or tridem rear axle. Ford Transit vehicles use generally SRW, which stand for single rear wheel, and DRW double rear wheel axle. Its applications changes according to vehicle types, payload, fuel efficiency and customer wants.

2.4.1SRW-Single Rear Wheel

Single rear wheel axle is mostly used for commercial vehicles if it is not designed to carry excessive payload. The advantages of SRW to DRW are like below;

- Cost of ownership
    - Operational costs related with worn tyre replacements
    - Improved fuel economy on SRW compared to DRW
- Ride comfort for people mover/coach bus intended transportation
- Customer expectation,satisfaction.

Figure 2.8 shows typical single rear wheel sheme for a vehicle.

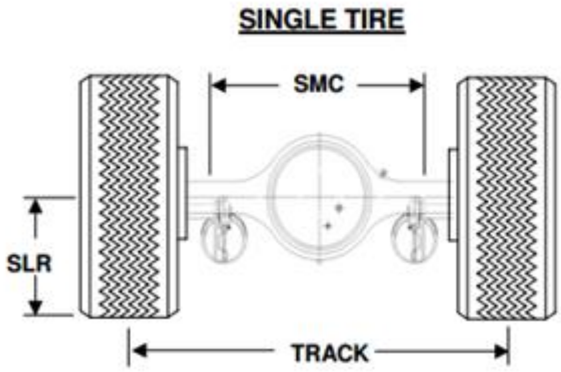

Figure 2.8 : Single tire rear axle[12]

2.4.2 DRW-Double Rear Wheel

DRW axle is designed to carry excessive payloads and this application result as worsening on riding comfort. Additionally two wheels generate more traction force on wheels and this is turn over as cost to owner. In case of payload increase, wheel number can be increased to distribute loads equally and more balanced. Figure 2.9 shows typical double rear wheel sheme for a vehicle and Figure 2.10 represent the variant at vehicle.

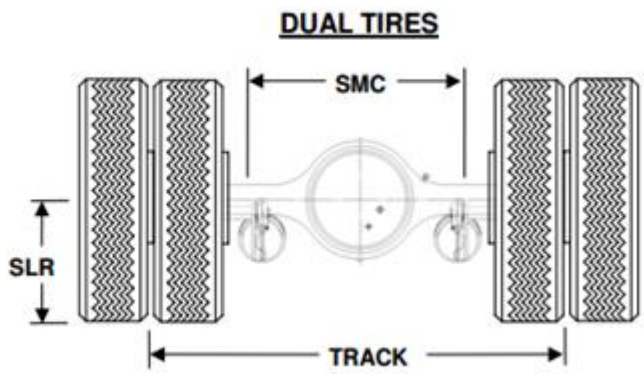

Figure 2.9 : Dual tire rear axle[12]

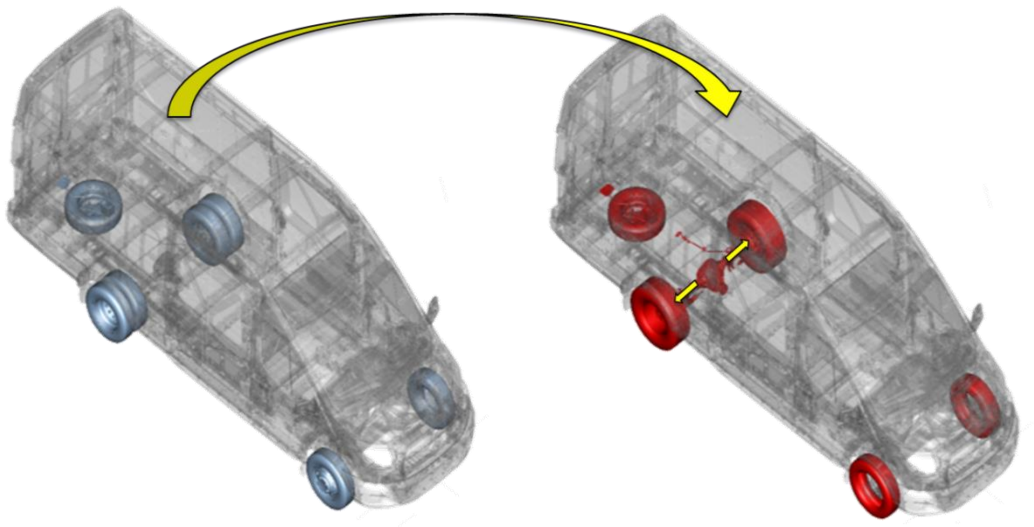

Figure 2.10 : DRW to SRW proposal

2.4.3 Hypoid Gear-Ring Gear

Hypoid gear and ring gear are main component of differential system. These components provide the rotation change of velocity and torque. Engine torque transmitted to propshaft, propshaft is connected to hypoid gear with gear mechanism. At this stage torque and velocity only transmitting torque without any direction change. Figure 2.11 shows typical hypoid gear couples.

Hypoid gears function is to change the direction of the power transmitted by the drive shaft through 90 degrees to the driving axles. At the same time. it provides a fixed reduction between the speed of the drive shaft and the axle driving the wheels[22].

Gear noise is often described as a source-path-receiver problem where the excitation

Occurs in the gear mesh and is then transmitted via the gear body, shafts and bearings to the gearbox housing where it is radiated as noise[21]

The transmission error of a gearbox is the expression of the difference between the theoretical angular position of the driven gear (output shaft) and its actual position while running the driving gear (input) at constant speed.

The transmission error quantifies the gearbox's imperfections when transferring energy from input to output, resulting in a metric of the gearbox's efficiency. The higher the transmission error, increase the risk of an amplified dynamic variation of the shafts rotational speed or torque. This would imply an increased risk of noise or vibration problems in the gearbox or elsewhere in the driveline[20].

Static TE is commonly used as excitation in order to calculate bearing forces, enabling the prediction of vibration and sound radiation from the housing[21].

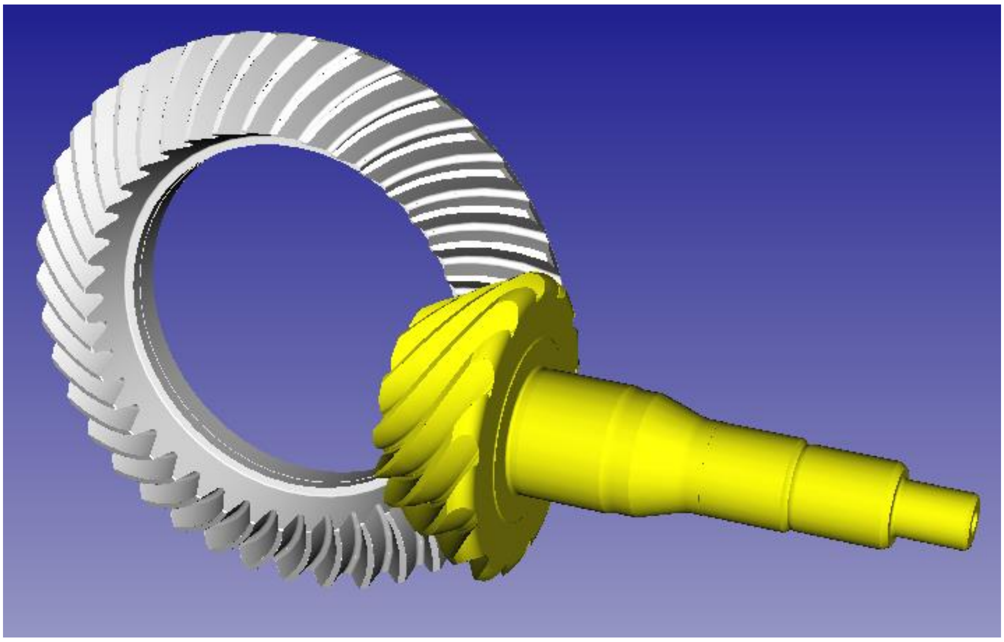

Figure 2.11 : Hypoid bevel gears

Hypoid ring gear couples and FDR-final drive ratio is most important parameters. Gear transmission error, mesh forces, interactions are main sources of axle whine noise therefore the study comprehend the pinion-ring gear couples to investigate the effect to vibrations-noise levels.

3. NVH

Noise and vibration are constantly present in our society. Noise causes serious problems both at home and in the workplace and the task of reducing noise is a main subject currently focused on by authorities. Similarly, manufacturers of mechanical products with vibrations causing acoustic noise, more and more find themselves forced to compete on the noise levels of their products. Such competition has so far occurred predominantly in the automotive industry, where the issues with sound and noise have long attracted attention [13].

NVH related problems (such as noise, vibrations, etc.) come out at a test and problems can fix if parts or internal component sources it. Isolation system can cover these kind problems but if noise source is vehicle body design, therefore it can be too late to fix this noise because of cost and time limitations.

Automotive design completed as a prior phase and lately design change cannot be implemented to vehicle because long lead-time of parts and cost of change therefore design issue must be understand and prevent by manufacturer at initial phase of vehicle production plan.

Under the constraints of improved vehicle refinement, automotive OEMs are challenged to improve vehicle noise, vibration and harshness (NVH) characteristics, reduce vehicle weight, and streamline manufacturing and assembly processes. In support of these objectives, alternate methods of vehicle noise control are being investigated[14].

Nvh test can be too late to understand and prevent powertrain systems structural or airborne noise issue therefore a computer-aided analysis is needed to solve this kind of problems. With this perspective, nvh methodology has to be developed and correlation of result must be compared with experimental results.

In order to decrease the cost of the development process or late changes applied to the car, it is crucial to have reliable simulation in advance to the realization of the first physical prototype.

3.1. Modal Analysis Theory

Finite element analysis is a computer modeling approach has provided engineers with a versatile design tool, especially when dynamic properties need to be perused. This numerical analysis requires rigorous theoretical guidance to ascertain meaningful outcomes in relation to structural dynamics. An important part of dynamic finite element analysis is modal analysis[1].Modal analysis is the theory dealing with the dynamics of mechanical systems described by modes.

Modal analysis is the process of determining the inherent dynamic characteristic of a system in forms of natural frequencies, damping factors and mode shapes, and using them to formulate a mathematical model for its dynamic behavior. The formulated mathematical model is referred to as the modal model of the system and the information for the characteristics are known as its modal data[1].Below equations simply describe the relation of force, mass and damping of an equation of motion definition**(3.1).**

$$m\ddot{x}(t) + c\dot{x}(t) + kx(t) = f(t) \qquad \textbf{(3.1)}$$

Where, M means mass matrix; C means damper matrix; K means stiffness matrix; x(t) means response; f(t) means force input[6].Below equations describes the natural frequency, damping and frequency response function**(3.2)**.

$$\omega_n = \frac{\sqrt{\mathrm{k}}}{\sqrt{\mathrm{m}}} \qquad \textbf{(3.2)}$$

The dynamics of structure are physically decomposed by frequency and position. This is clearly evidence by the analytical solution of partial differential equations of continuous systems such as beams and strings**(3.3).**

$$\xi = \frac{c}{\sqrt{2km}} \quad \textbf{(3.3)}$$

Modal analysis is based upon the fact that vibration response of a linear time-invariant dynamic system can be expressed as the linear combination of a set of simple harmonic motions called the natural modes of vibration**(3.4).**

$$H_{(\omega)} = \frac{\omega^2 n - \omega^2}{(\omega_n^2 - \omega^2)^2 + (2\xi\omega\omega_n)^2} \quad \textbf{(3.4)}$$

This concept is akin to use of a Fourier combination of sine and cosine waves to represent a complicated waveform. The natural mode of vibration are inherent to a dynamic systems and are determined completely by its physical properties (mass, stiffness, damping) and their spatial distributions. Each mode is described in terms of its modal parameters: natural frequency, the modal damping factor and characteristic displacements pattern, namely mode shape. The mode shape may be real or complex. Each corresponds to a natural frequency. The degree of participation of each natural mode in the overall vibration is determined both by properties of the excitation sources and by the mode shapes of systems.

Modal analysis embraces both theoretical and experimental techniques. The theoretical modal analysis anchors on a physical model of a dynamic system comprising its mass, stiffness and damping properties. These properties may be given in forms of partial differential equations. A more realistic physical model will usually comprise the mass, stiffness and the damping matrices. These matrices are incorporated a linear dynamic system enables us to transform these equations in to a typical eigenvalue problem. Its solution provides the modal data of system. Modern finite element analysis empowers the discretization of almost any linear dynamic structure and hence has greatly enhanced the capacity and scope of theoretical modal analysis[1].

Structural modification makes changes on mass, stiffness or damping of a dynamic system. These physical changes will certainly alter the dynamic behavior of the system. Using the modal model of the system, simulation and prediction of 'what if' can be conducted. The effect of hypothetical physical changes on the dynamic behavior can be derived without another complete analysis or the actual structural changes.

For instance, if a lumped mass is to be added to a part of the system, then the existing modal model and the mass together should predict a new modal model 'after' the structural modification. This is particularly useful in an early design stage to optimize dynamic characteristics of a new design, or to improve a structure's dynamic behavior after its modal model is derived from measurement. However, this approach of structural modification often forbids large-scale structural changes[1].

All numerical solutions of computer-based programs are based on finite element method and equation of motion. According to mass, stiffness matrix and degrees of freedoms of systems; eigenvalue and Eigen vector can be calculated and accordingly natural frequencies mode shapes can be solved.

3.2. Noise Paths

Noise can be transmitted to interior environment in two way. One of them is structural which noise transmitted via structural components of vehicle to interior. For rear axle; noise generated generally at differential case, it cause vibration and vibration transmits to springs via tubes, springs transmit it to chassis and chassis transfer it to inside the vehicle. Figure 3.1shows the noise paths for a passenger vehicle.

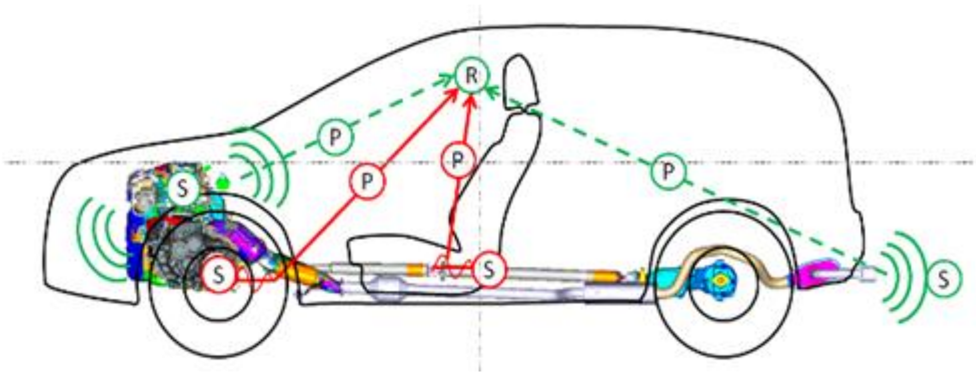

Figure 3.1 : Structure Borne and Airborne noise path[11]

Other noise transfer path is air, structural vibrations causes pressure wave at air, the pressure changes generates noise.There are mainly two kind of noise sources classification according to propagation of noise which is structural-borne and airborne.

3.3.1. Structural-Borne Noise

Structural-borne noises are generally composed by external vibration and forces and propagated , transmitted by automotive systems parts and components to the interior of automotive. Structure borne noises are defined at low and mid frequency range. Figure 3.2 represent the range of structure and airborne noise.

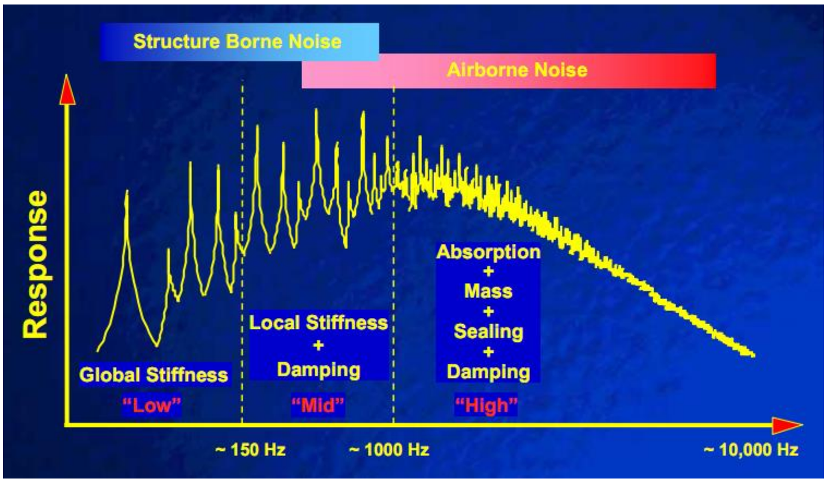

Figure 3.2 : Structure Borne and Airborne noise frequency range[15]

Sound levels are commonly described in terms of the sound power (W) output of noise sources and the sound pressure (Pa) amplitude but decibel scales (dB) are useful due to the wide range of sound powers and sound pressure amplitudes[15].

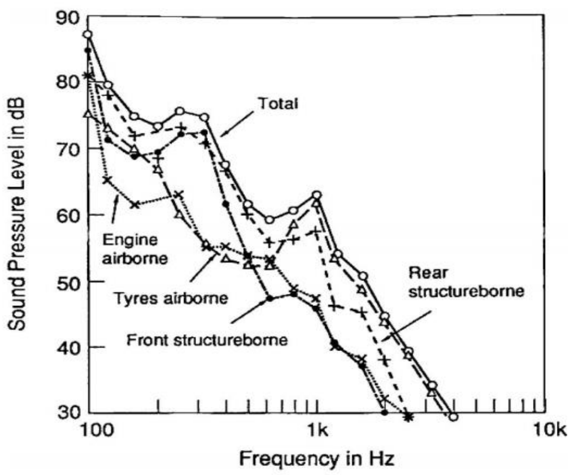

Figure 3.3 : Sound pressure levels with corresponding frequency according to sources and types[15]

It can be seen from Figure3.3 that below 500 Hz the interior noise is dominated by structure-borne inputs.

3.3.2 Airborne Noise

Airborne noises are composed via interaction of parts and flows, propagation and transmission the noise to the interior of vehicle by air. Most significant airborne noises are tire noise and wind noise. Figure 3.4 shows frequency mapping of driveline NVH performance attributes.

As well as tire/road and engine noise, wind noise is also becoming increasingly important for similar reasons, although the problem is generally only significant at high speeds (wind noise increases at approximately 60 dB per decade of speed, which is a much higher rate than the other sources)[15].

Airborne noise radiates spherically in a free field so that the sound intensity will decrease with the square of the distance from the source.

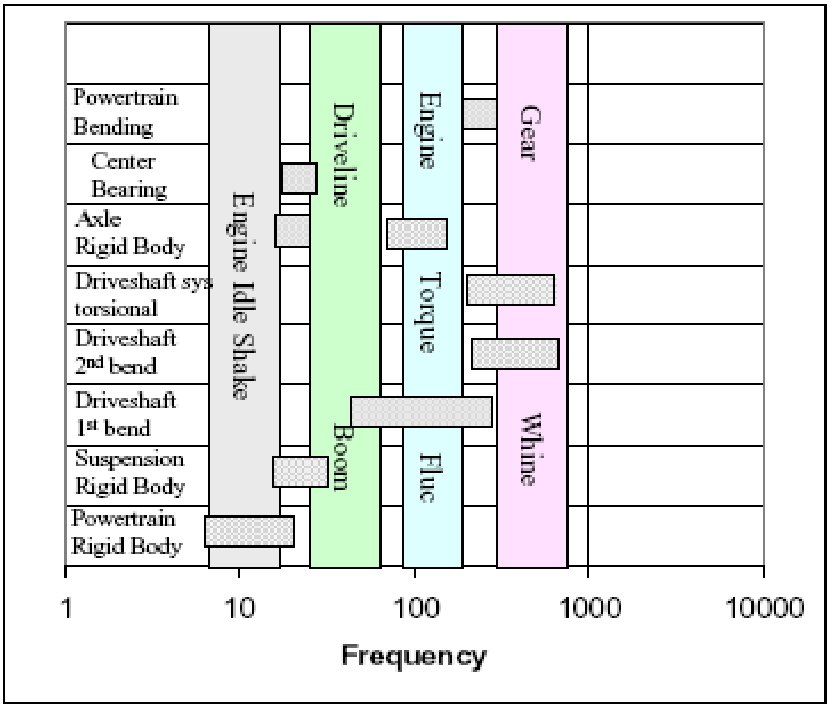

Figure 3.4 : Frequency mapping of primarily driveline NVH performance attributes[16]

3.4 Noise and Vibration Analysis

Noise and vibrations analysis can be performed in several ways and methods. Analytic analysis of vibration is most commonly done using the finite element method(FEM) through normal mode analysis. In order to successfully model vibrations, usually models with much detail such as finer grid meshes, correctly selected element types, etc. need to be used, compared with models sufficient for static analysis. In addition, dynamic analysis using FEM requires good knowledge of boundary conditions. For many of these inputs to FEM software, experiments can help refine the model. This is the main cause of much experimental analysis of vibrations today.

For acoustic analysis, acoustic FEM can be used as long as the noise is contained in a cavity. For radiation analysis, the boundary element method(BEM) is getting more common. At this method, known patterns from a FEM analysis can be used to model how the sound radiates and builds up an acoustic field.

FEM and BEM are usually restricted to low frequencies, where the mode density is low. For higher frequencies, statistical energy analysis(SEA) can be used. As the name implies, this method deals with mode density in a statistical manner, and is used to compute average effects[15].

3.5 Experimental Analysis

In many cases it is necessary to measure vibrations or sound pressure to solve vibration problems, because the complexity of problems often make them impossible to foresee through analytical model such as FEM. This is often referred to as trouble-shooting. Another important reason to measure and analyze vibration is to provide input data to refine analytical models. Particularly, damping is an entity, which is usually impossible to estimate through models, it needs to be assessed by experiment.

Experimental analysis of noise and vibrations usually done by measuring accelerations or sound pressures. In order to analyze vibrations, the most common method is by frequency analysis, which is due to nature of linear systems. The main tool for frequency analysis is the FFT(Fast Fourier Transform) which is readily available today through some software.

Some of the analysis necessary to solve many noise and vibrations problems needs to be done in the time domain. An example of such analysis is the fatigue analysis, which is incorporates cycle counting, and data quality analysis, to assess the quality of measured signals. For a long time the tools for noise and vibrations analysis were focused on frequency analysis, partly due to limited computer performance and cost of memory. Today, however sophisticated time domain analysis can be performed at a low costs[15].

Estimation of frequency response turns out to be rater complicated in terms of all the errors involved due to the spectral estimation as well as errors due to unwanted noise in the measured force and response signals.

Computer modeling alone cannot determine completely the dynamic behavior of structures, because certain structural properties such as damping and nonlinearity do not conform with traditional modeling treatment. There are also boundary condition uncertainties which modeling needs additional help to work. Substantial advances in determination techniques have complemented modeling with the experimental determination of structural properties[1].

Computer aided engineering analysis is not included at this chapter due to next chapter mainly approach the CAE analysis and its results.

4. FEM MODEL

Finite elements method is the main approach to solve interactions and equations between components and systems. Its transmits the force, displacement etc. to near field equivalence elements. Modeling components as finite element help to define and solve physically nature of component more accurately.

Rear axle interacts with systems like propshaft and indirectly transmission and engine. So to define the physic of rear axle more accurately, other systems also included at model. Components of powertrain system are imported to preprocessor tool(Hypermesh) as 3D.The 3D data's divided to finite elements according to Motor Company(FMC) CAE guideline. This guideline includes all connections of major components at powertrain system. From grid number identification to property of elements are defined at this guideline so model is composed based on this procedure.

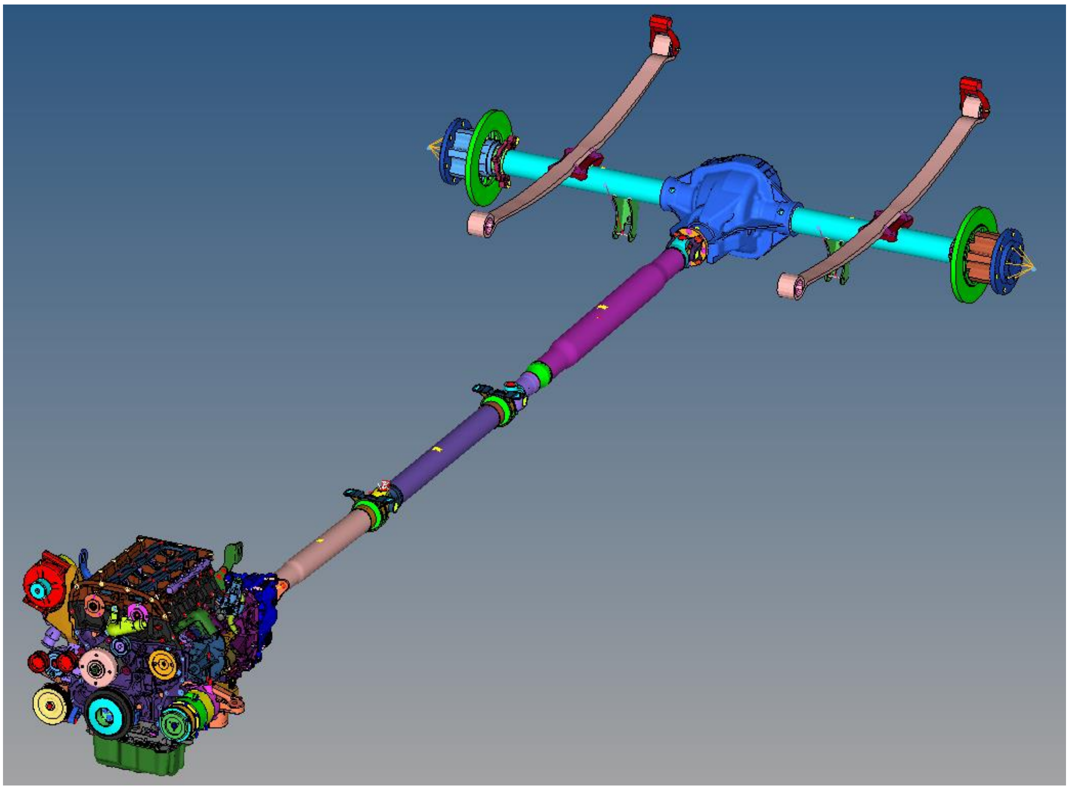

Figure 4.1: FE model of powertrain system with SRW axle

Figure 4.1 represent medium commercial powertrain systems as FE model. This model includes all relevant powertrain systems and components, some parts are not modeled due to ineffective behavior to modal frequency. This model contains majority of important components as fem but some components on engines is taken only mass due to it has not effect on analysis.

This powertrain model includes5388654 node, 3673382 elements and 1593 components,443 materials and 971 property card.

Figure4.2 shows the SRW axle finite element model which includes leaf spring, axle tube, differential housing. Interior part of this model is showed at Figure 4.3 without tubes and leaf springs.

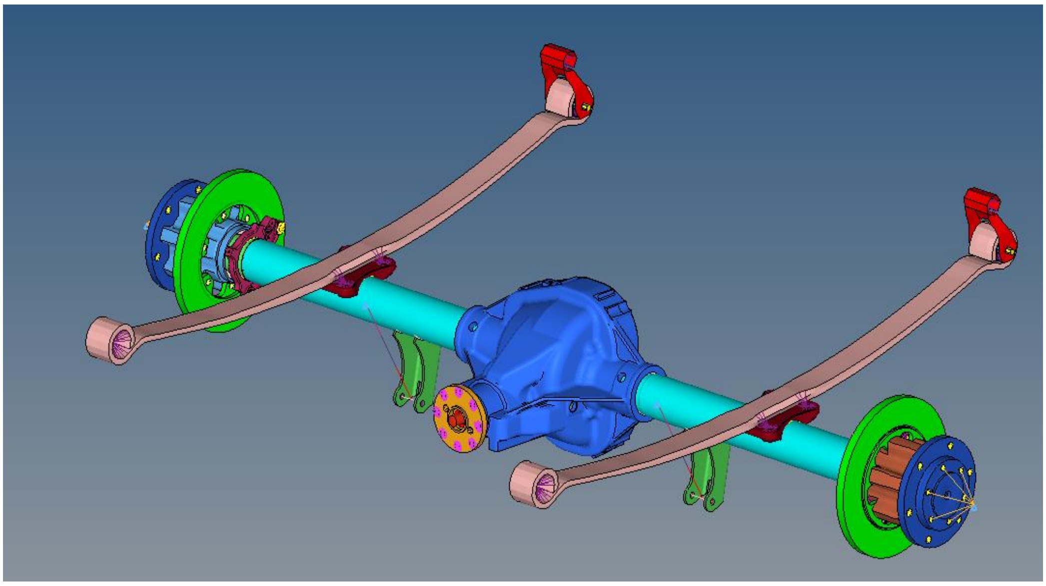

Figure 4.2: SRW axle finite element model

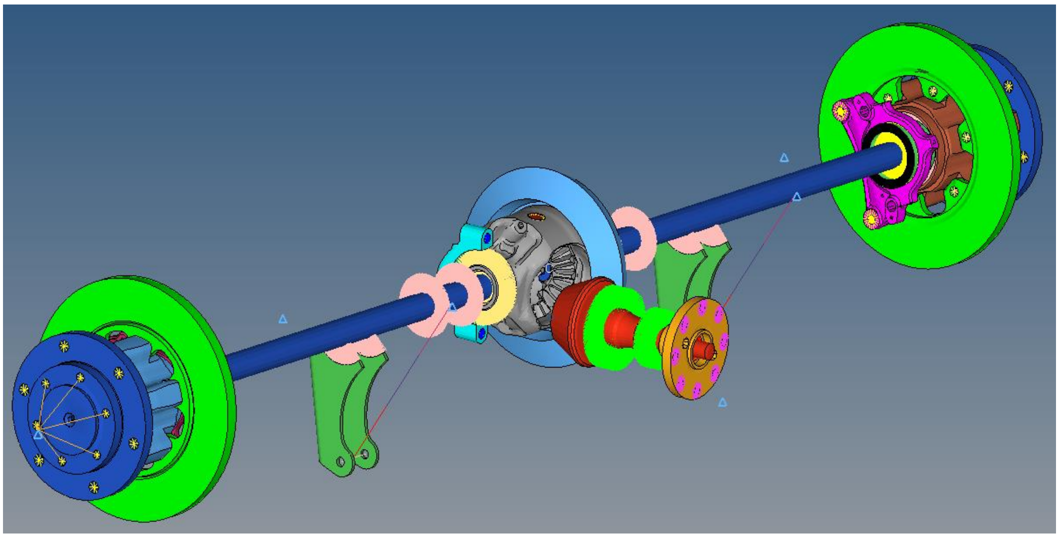

Figure 4.3: SRW axle interiour finite element model

FEM model includes variety of elements to represent the real life conditions. Hypermesh preprocessor has easy to use gui and has lots of elements type to build more accurate finite elements model.

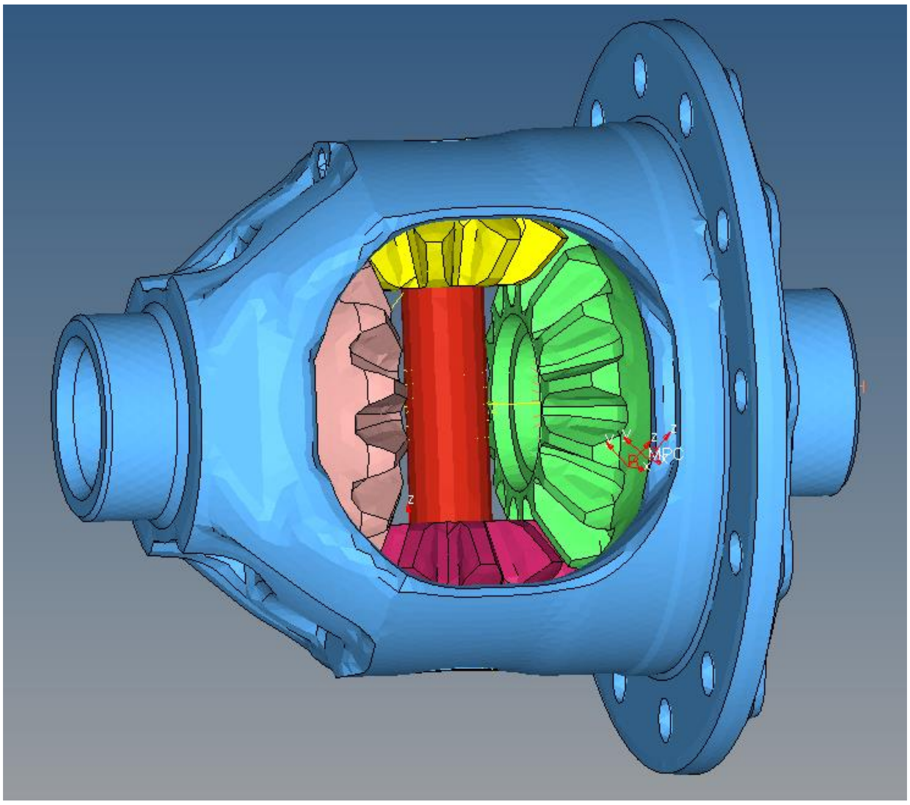

Figure 4.4: Differential case FE model

Figure 4.4 shows that differential case and gear connections inside it. The gears connected to each other with 4 MPC(Multi Point Constrained) elements which defines the relation of gears according to teeth values.

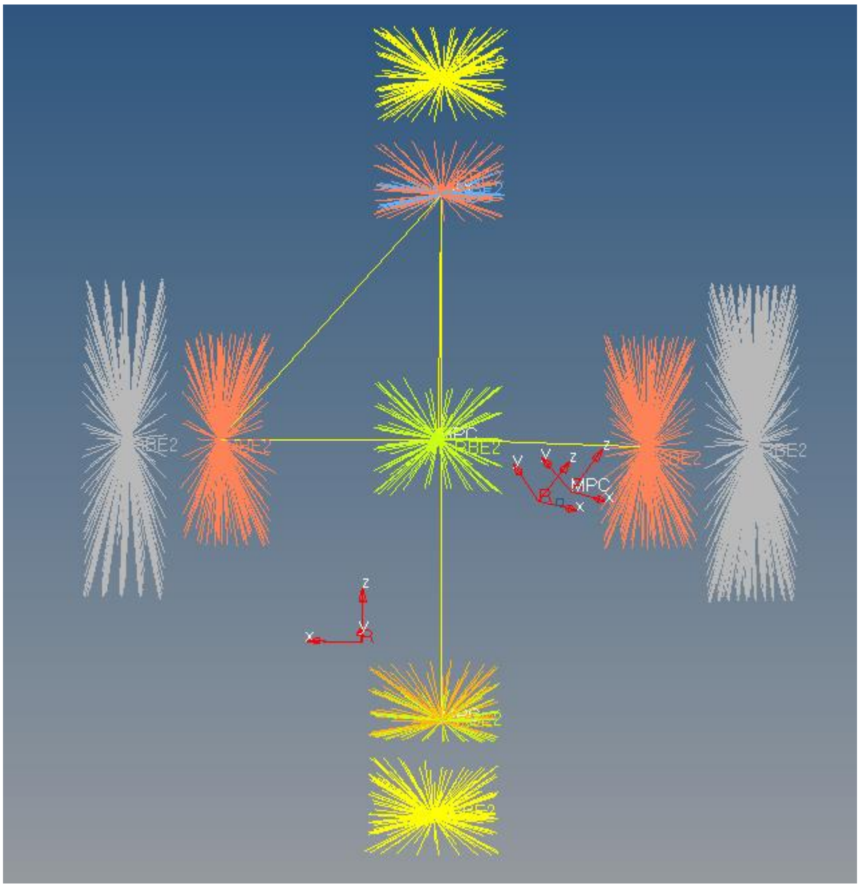

Figure 4.5: Differential case gear connections

Figure 4.5 represent the gear connections with Rbe2 and MPC elements. Additionally Figure 4.6 shows 3 pieces propshaft and figure 4.7 shows the U bracket connections in detail.

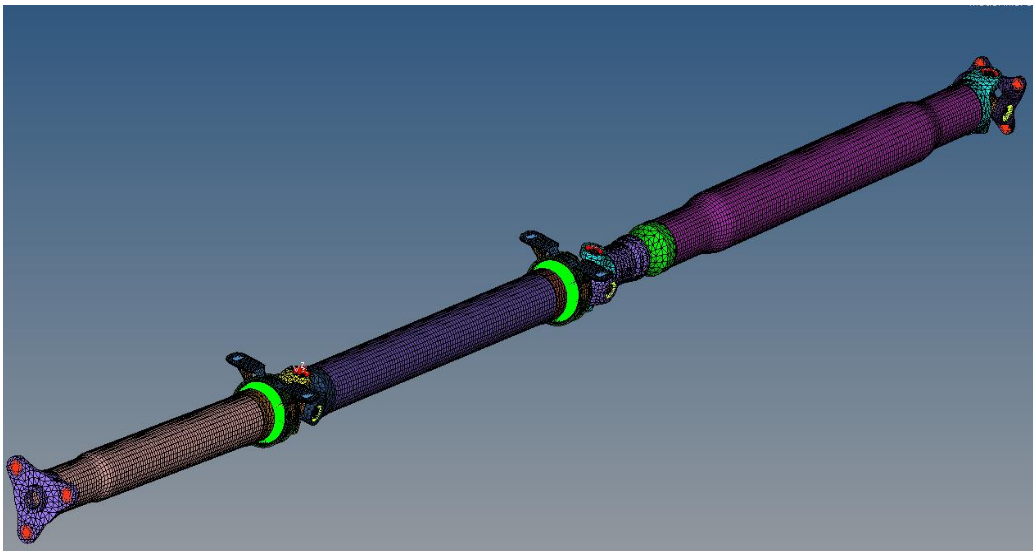

Figure 4.6: FE model of propshaft

The propshaft models consist of 3 pieces, according to length propshaft supported with U bracket to the vehicle frame.

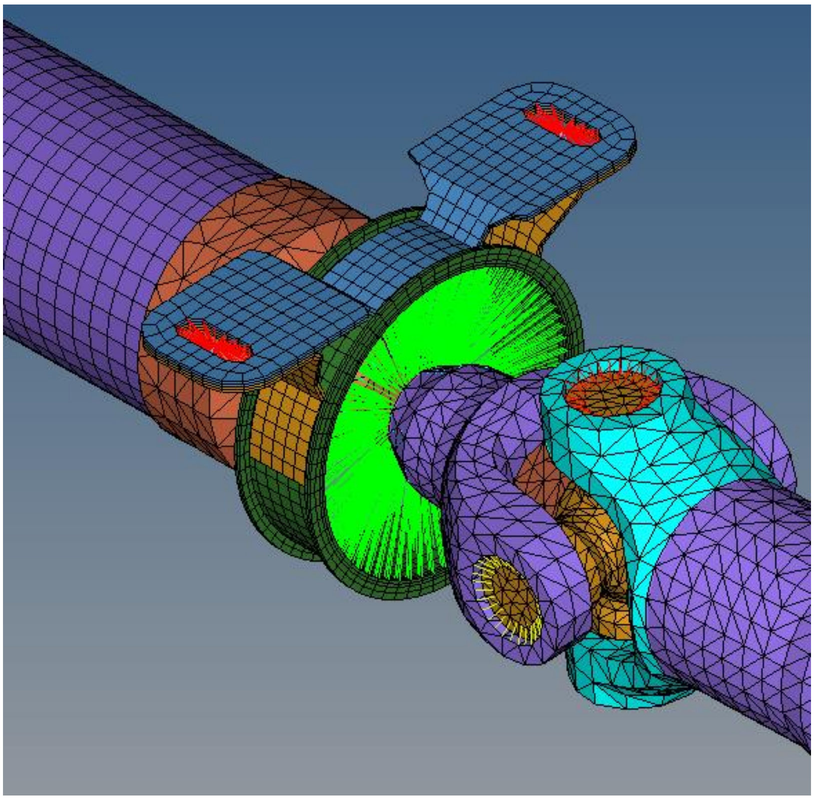

Figure 4.7: U bracket connection

Powertrain FE model has some of the elements types and it is explained according to dimension(1D,2D,3D).

4.1. 1-D Element

1-D elements perform a critical function in Finite Element Analysis as they can be used to connect nodes together, attach dissimilar meshes, distribute loads and in general provide a quick and easy way to attach things together. There are numerous types of 1-D elements ranging from infinitely rigid simple connections to complex cross sectioned elements that can be stressed.

Hypermesh currently supports bar2, bar3, rigid links, rbe3, plot, rigid, rod, spring, weld, gap, and joint one-dimensional elements. The following is a list of the different element types and their purpose, which is mostly used at fem model:

- bar2: supports complex beams
- bar3: supports complex beams (bar3 elements contain a third node designed to support second order beams)
- rbe3: supports NASTRAN RBE3 elements
- rigid: supports rigid elements
- rod: supports simple beams
- spring: supports springs or damper
- weld: supports weld elements[17]

Although the model includes different type of 1D element, some special connections like gear is defined more specifically.

4.1.1. Hypoid Gear Connection

Hypoid gear the most important and one of the most complex connection for FE model. It is important because gear interactions are the main reason of whine noise therefore it should be modeled accurately.

The FE model hypoid gear connections provided by 4 spring elements and 2 coincident multi point constrain to simulate whine noise. Gear parameters inputted to HGSiM tool is used to calculate gear mesh point, line of action and stiffness and accordingly output files transferred to Hypermesh. Figure 4.8 shows the gear connections in schematic for Nastran solver.

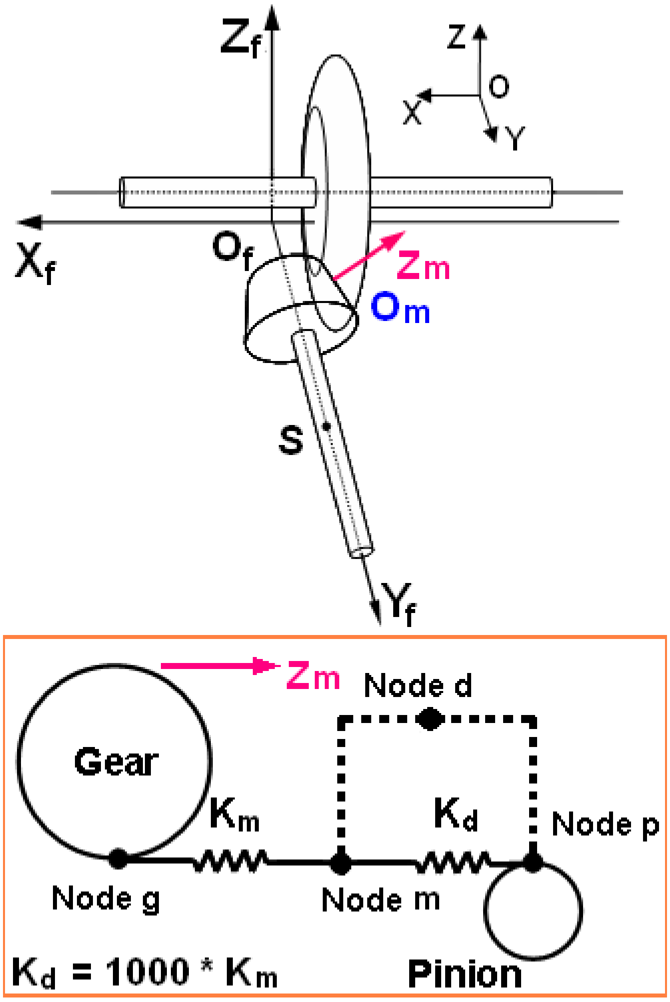

Figure 4.8: Hypoid gear connection[19]

There is two-spring element, which is Celas2, used to represent right connections. Km is the gear mesh stiffness and Kd is simply equal 1000 times Km which is used to represent transmitted error. Node d is used to provide equations of gears, which is special connection of gear rotations represented with MPC element. Zm is vector of line of action, which are very important parameters for gear connections to transmit force and moment truly. Figure 4.9 shows line of action scheme.

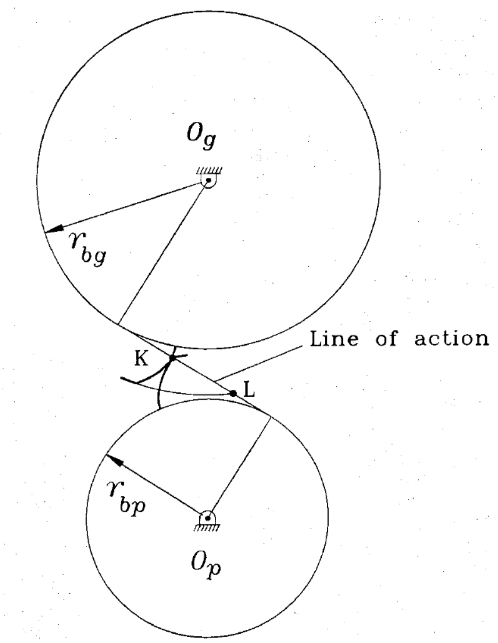

Figure 4.9: Line of action[20]

Line of action represent the connection line vector between the gear teethes, it is the line where force is transmitted. Figure 4.10 shows the line of action coordinates for finite element model.

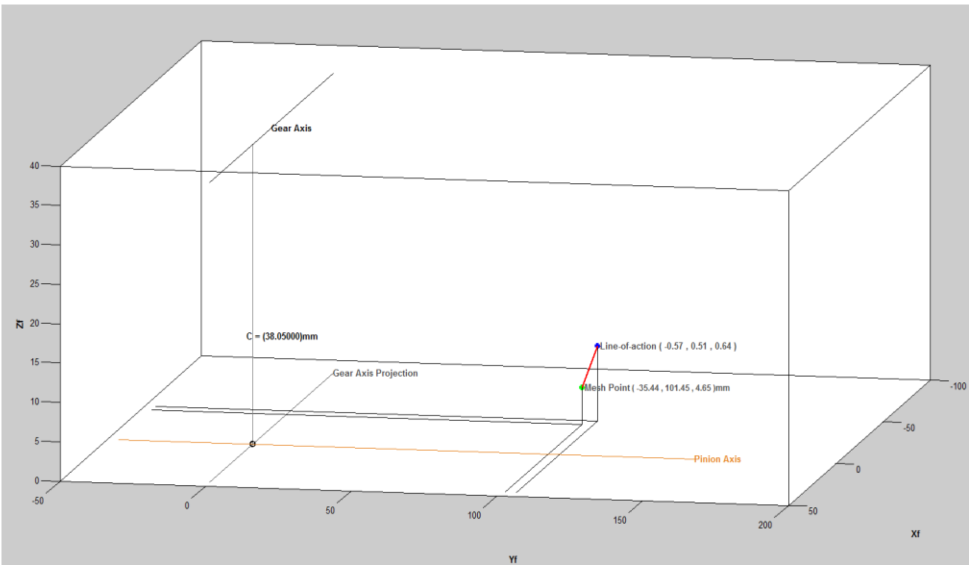

Figure 4.10: HgSiM mesh point and line of action coordinates

4.1.2.2-D Element

Some of powertrain components are designed as sheet metal and simple geometry. For those components, 2D elements are used to make model easier to change parameters and to decrease calculation time. Mostly used elements as 2D are CTRIA3 and CQUAD4.

CTRIA3 is three-node isoparametric flat plate element. Commonly used for mesh transitions. May have excessive stiffness particularly for membrane strain.CQUAD4 is four node isoparametric flat plate element, behaves well when irregularly shaped; good results can be obtained with skew angles up to 45 degrees

4.1.3. 3-D Element

There are varieties of element type for solid meshing but powertrain components generally are meshed withTetra4, Tetra10, Hexa8, Hexa20 elements in case of complex geometry, below other elements are also used to shape model accurately. Figure 4.11 shows basic element types.

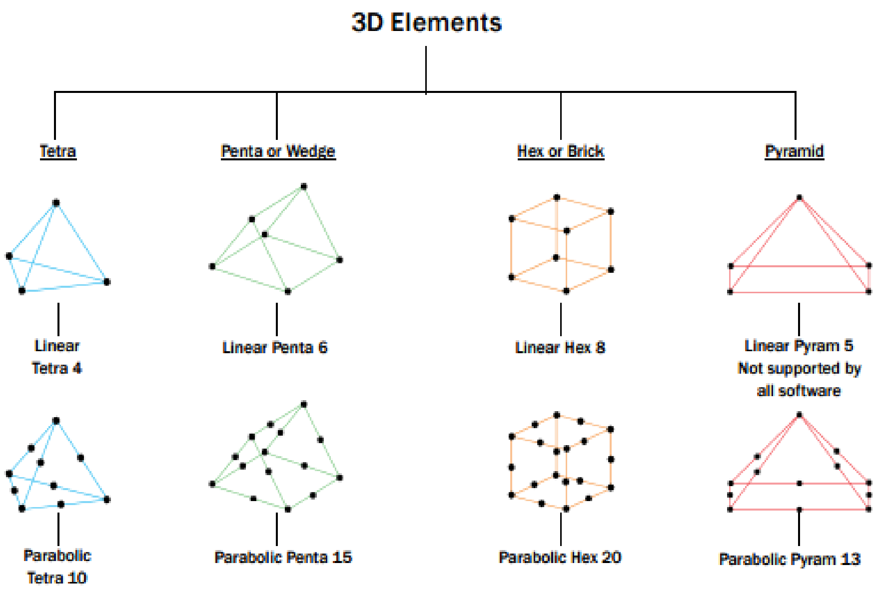

Figure 4.11: 3D element types[18]

Although second order elements provides more accurate results , it increase calculation time and computational cost therefore if component is not critical for analysis it is better to model it as first order.

5. ANALYSIS&SOLUTIONS

There are 3 different case to compare the effect of systems and changes which are;

- DRW&SRW comparison
- Tube thickness change
- Final drive ratio change

Case 1 is based on axle type change for real purpose for currently used commercial vehicle. Case two and there is compared inside as a design of experiment action for future changes.

5.1. Case 1-DRW&SRW Comparison

There is a change proposal on axle wheel number where DRW has 4 wheels are changes to SRW which has 2 wheels. The change also requires axle and tube length increasement almost 300 mm. Track width increased approximately 50 mm per each side. Figure 5.1shows the basic difference between DRW and SRW axles.

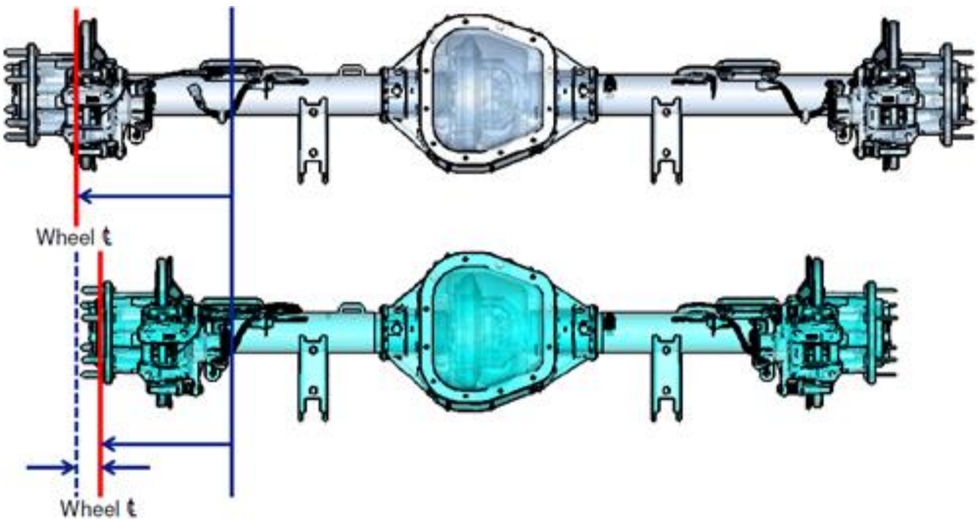

Figure 5.1: DRW&SRW rear axle view

The most acceptable method to give a decision about new system is comparing it with previous one. It is known that current DRW rear axle is not generating noise so comparing natural frequencies of new SRW axle with DRW axle is the best decision criteria. Of course to perform nvh test is more accurate and certain decision criteria but at this stage of proposal, there is no manufactured SRW axle and also there is no produced car with this axle. For that reason two system compared according their natural frequencies.

There are general rules for powertrain driveline system's modal analysis to decide if it is generate noise or not. If components natural frequencies being coupled where close two components has same natural frequency it generate noise and also get in resonance therefore it will broke down. Therefore, while designing a system it is requested that two related or close component has some natural frequency difference, which is modal separation. Figure 5.2 shows SRW axle rear propshaft natural frequencies values.

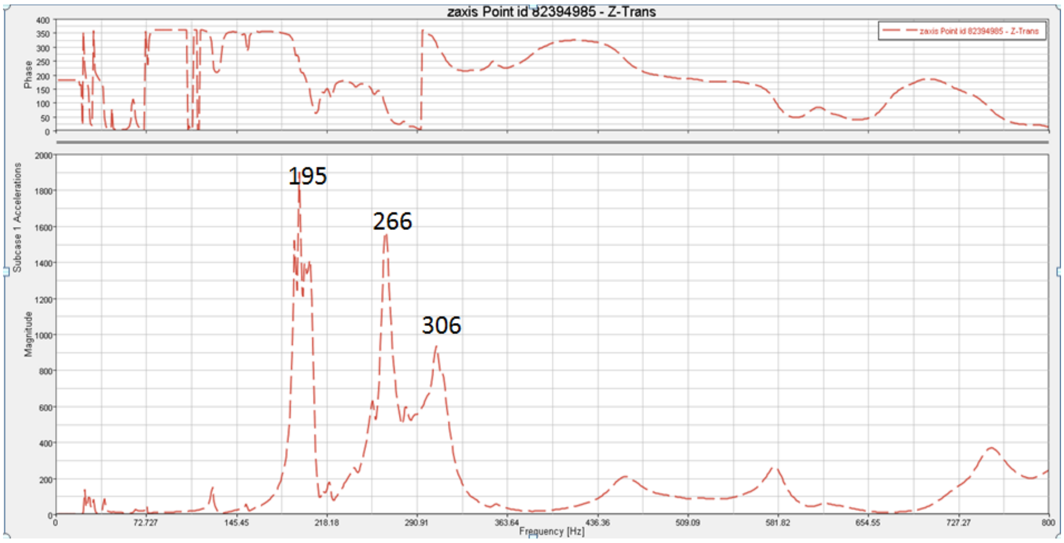

Figure 5.2: SRW rear propshaftnatural frequecies

Figure 5.3 shows the mode shapes of this propshaft.It is observed that second and third mode of rear propshaft is affected by first and middle propshafts.Also it is clear at Figure 5.2, the second and third modes amplitudes are lower than fist mode amplitude and it can be figured out that modes are affected by other components.

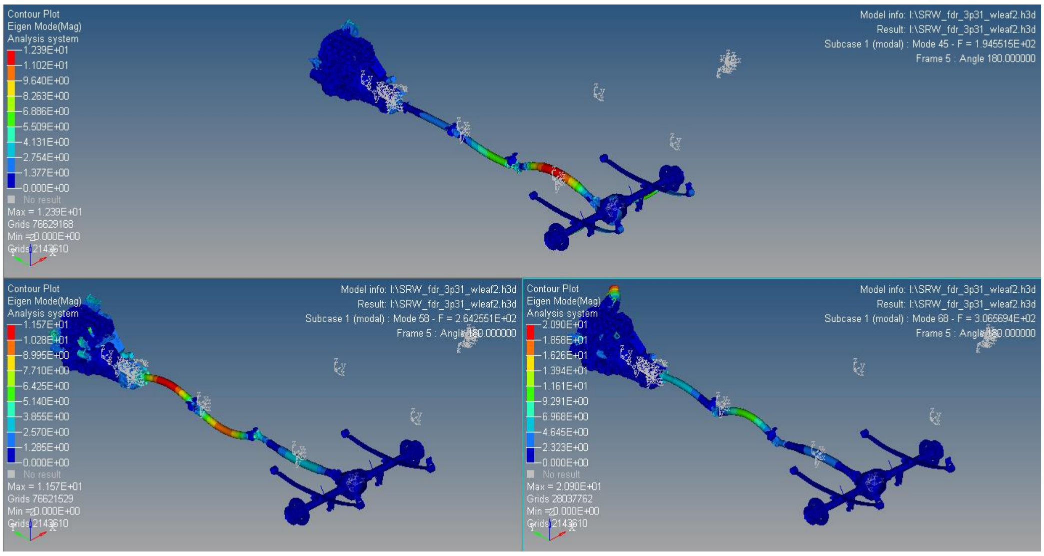

Figure 5.3: SRW rear propshaft mode shapes

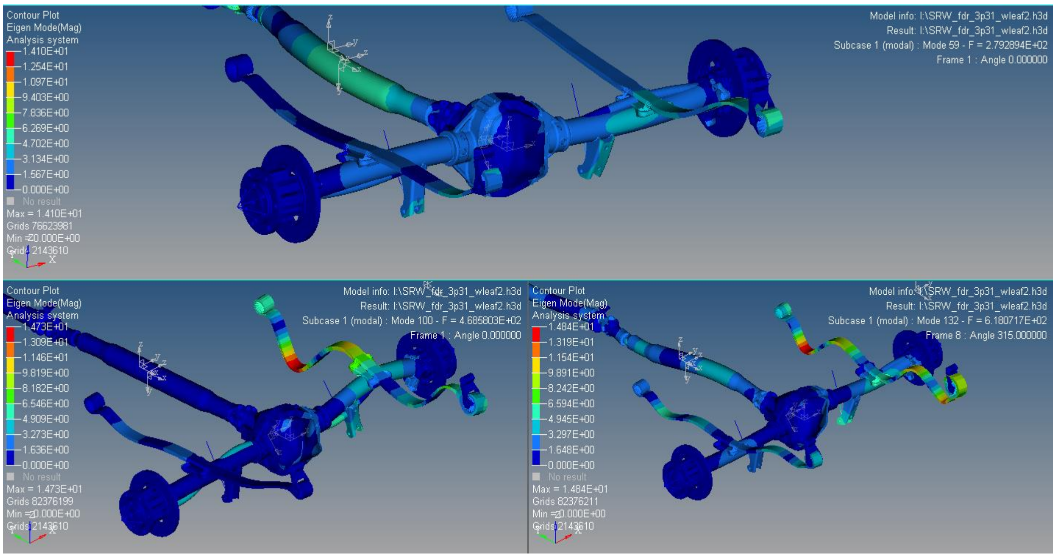

Figure 5.4: SRW rear axle tube mode shapes

Table 5.1 shows the natural frequencies of both axle types. Left axle represent the left side of vehicle when person looks to vehicle at front of it. The DRW and SRW axle modes are different from each other. Since the methodology is modal separation of components natural frequencies, it is observed that DRW rear axle mode gaps greater than the SRW. The difference is 29 Hz between the modes which is 13 Hz for SRW axle. Therefore, it is figured out that DRW axle is modal separation is better than SRW and it should be less nosier than SRW.

Table 5.1: Natural frequencies of SRW and DRW axle

	Modes(Hz)	Left Axle	Right Axle	3rd Propshaft
SRW	1st Mode	183	182	195
	2nd Mode	203	199	264
	3rd Mode			307
DRW	1st Mode	166	166	195
	2nd Mode	181	181	265
	3rd Mode			

5.2. Case 2-Tube Thickness Change

Tube is the payload carrier component of rear axle, design of tube, especially thickness change according to vehicle weight or cost reduction purpose need to be investigated deeply. Since the thickness affect the bending characteristic of axle, natural frequency of rear axle system can be affected due to mass or stiffness change.

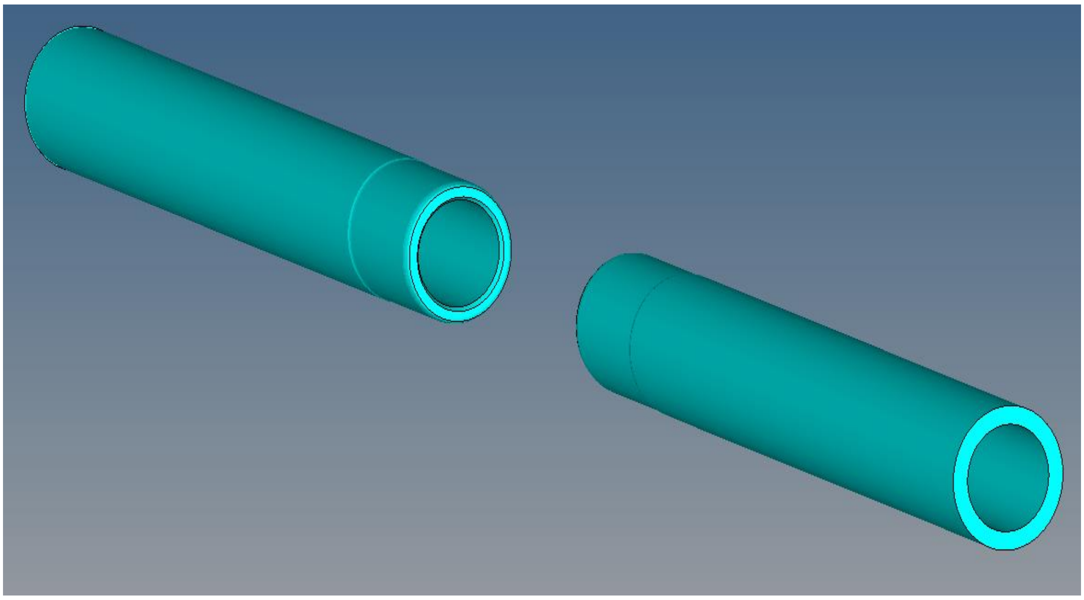

Figure 5.5: Tube of rear axle system

Same as axle type change the modal separation of components is the main and basic criteria to decide the new design is noisy or not.Same aproach is used for tube thickness change.

The natural frequency of a component can be figured out basically with below equation:

$$W = \sqrt{\frac{K}{m}} \quad \textbf{(5.2)}$$

The weight reduction of rear axle tube is directly effects the modes, mass decreasement will increase the frequency value if the stiffness still same or not decrease as mass(5.1).Since the stiffness value depend on length of the tube, modulus of elasticity and mass moment of inertia, the elasticity modul and length still same, only inertia is effected by mass and the decreasement ratio is need to be find. Additionally the modes of structures must be analysed as system and effect of mass decreasement on modal separation should be figured out accordingly. Figure 5.6 showh the thickness change effect to 3rd propshaft mode.

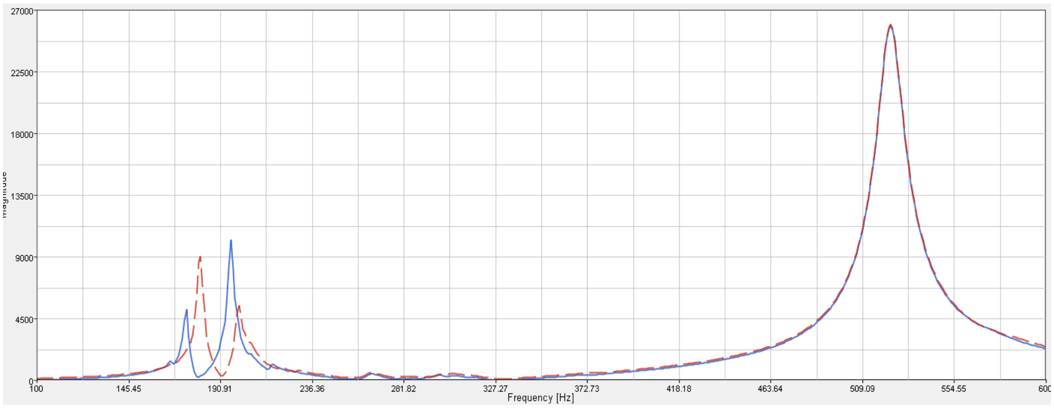

Figure 5.6: Thickness effect on 3rd propshaft

Some shifting can be observed first and second mode of shaft where it was 174 Hz for 6 mm thickness become 180 Hz for first mode, it changes from 196 Hz to 200 Hz for second mode and still same for 3rd mode.

Table 5.2 shows that the eigenvalues of axle and 3rd shaft for two different thickness.

Table 5.2: Eigenvalues of axle for 6 mm and 9.9 mm thickness

DRW	Modes(Hz)	Left Axle	Right Axle	3rd Propshaft
T9.9	1st Mode	167	167	180
	2nd Mode	182	181	200
	3rd Mode		272	523
T6	1st Mode	167	167	174
	2nd Mode	181	181	196
	3rd Mode		263	522

It is observed that thickness change could not affect the axle modes but 3rd propshaft modes changed. The difference between the modes decreased for 6.6 mm tube thickness. The differences between 3rd propshaft and axle first mode is 13 Hz for 9.9 mm but it becomes 7 Hz for 6 mm tube. Same as first mode, second modes are also decreasing for 6.6 mm where it is 19 Hz for 9.9 mm becomes 14 Hz.

It can be figured out that from modal and frf analysis result, tube thickness reduction is worsening the nvh characteristic of rear axle. The modal separation of 6.6 tube thickness is less than 9.9 mm therefore modal coupling of components are more obvious and it is the major factor to get in resonance situation.

5.3. Case 3-FDR Ratio Change

There are a few FDR ratio variations, which is currently used at Transit vehicle, but this thesis focus only two most used ratio, which is 3.31 and 4.1. This value directly comes from ring gear ratio to pinion gear ratio, below equation shows the relation(5.2).

$$FDR = \frac{Ring\ gear\ number}{Pinion\ gear\ number} \tag{5.2}$$

The higher axle ratio, 4.1:1 for instance, would increase acceleration and pulling power but would decrease fuel economy. The engine would have to run at a higher rpm to maintain an equal cruising speed. The lower axle ratio 3.31:1 would reduce acceleration and pulling power but would increase fuel mileage. The engine would run at a lower power rpm while maintaining the same speed[22].

Final drive ratio is very important parameter for gear whine issue for that reason two different ratio investigated to see the effect of gear parameters to axle whine noise. Table 5.3 shows the gear parameters for two different gear ratio.

Table 5.3: Hypoid gear parameters

Parameter	Ratio:3,31	Ratio:4,10
Num of pinion teeth	13	10
Num of gear teeth	43	41
Offset [mm]	38	38
Gear Face Width [mm]	41	40
Pinion mean cutter radius [mm]	210	210
Gear pitch diameter(outer pitch) [mm]	250	250
Gear pitch radius (outer pitch) [mm]	125	125
Pinion Spiral Angle (mean) [deg]	NA	NA
Pinion Face Width	NA	NA
Backlash micro scale [mm]	NA	NA
Mesh stiffness variation (% of mean)	NA	NA
Hand of pinion spiral	LH	LH
Rot direction	CCW	CCW
Gear Type	Face-Hobbed	Face-Hobbed
Gear Cutter Radius	105	105
Number of blades(Nw)	19*2	19*2
Gear Spiral angle (mean) [deg]	28	32
Pinion Pitch angle	NA	NA
Gear pitch angle	62	67

Modal analysis and frf analysis is performed to distinguish component modes but for different FDR ratios, it is not observed any difference between variants. Below table 5.4 represent the modes of components for both FDR 3.31 and 4.1.

Table 5.4: Eigenvalues of components according to FDR

SRW	Modes(Hz)	Left Axle	Right Axle	3rd Propshaft
3p31	1st Mode	182	182	195
	2nd Mode	199	199	264
	3rd Mode			307
4p1	1st Mode	183	183	195
	2nd Mode	200	200	264
	3rd Mode	299	299	307

Additionally Figure 5.7 shows the pinion nose displacement graph of FDR 3.31 and 4.1 ratio. It can be seen the graph that 3.31 FDR ration pinion nose displacement is little bit higher than FDR 4.1, its vibration is also higher therefore it can be said that it generates more noise than FDR 4.1 ratio.

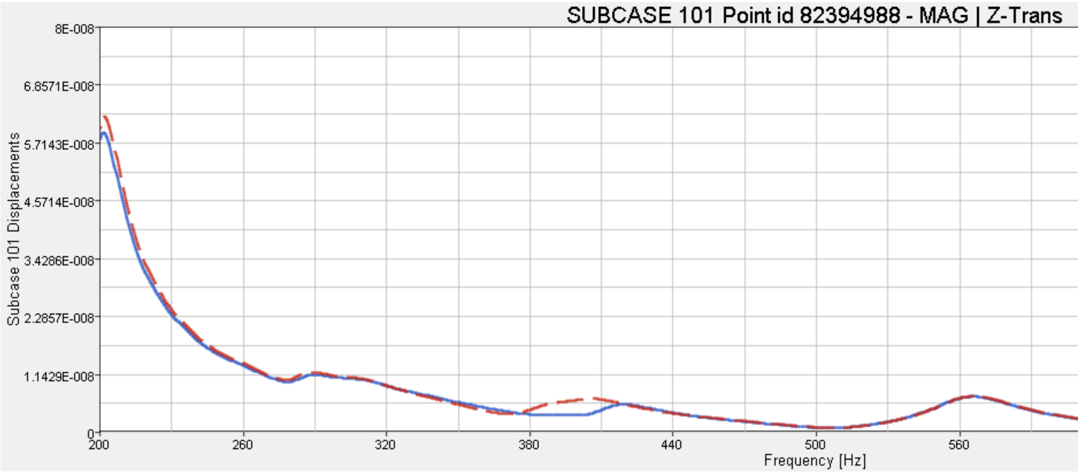

Figure 5.7: FDR 3.31 and 4.1 pinion nose displacement

According to modal analysis and mode separation method, it is not clear to make a comment about the fdr ratios effect on noise propagation. To make an evaluation, other analysis methods such as compliance or radiated noise need to be investigated. Addition to that dynamic model of those systems should be composed after that a good evaluation can be made.

6. TEST AND VERIFICATION

The DRW to SRW change is the important and major update for vehicle variant. Therefore, this proposal is investigated deeply and tested in few departments. Powertrain NVH team tested the new axle in system level and Vehicle NVH team tested it for driver sensibility inside the vehicle.

Since the powertrain test is in steady state condition and measurement of acceleration of driveline system gives the natural frequencies, therefore analysis results directly compared with them.

6.1. Powertrain Level Tests

Transit Jumbo bus vehicle with SRW axle is tested to find natural frequency of components. To perform this, hammer test, which represent frequency response analysis, is carried out. Figure 6.1 shows the measurement locations on powertrain system and figure 6.2 shows schematic representation the accelerometer on powertrain system.

Figure 6.1: Test measurement locations

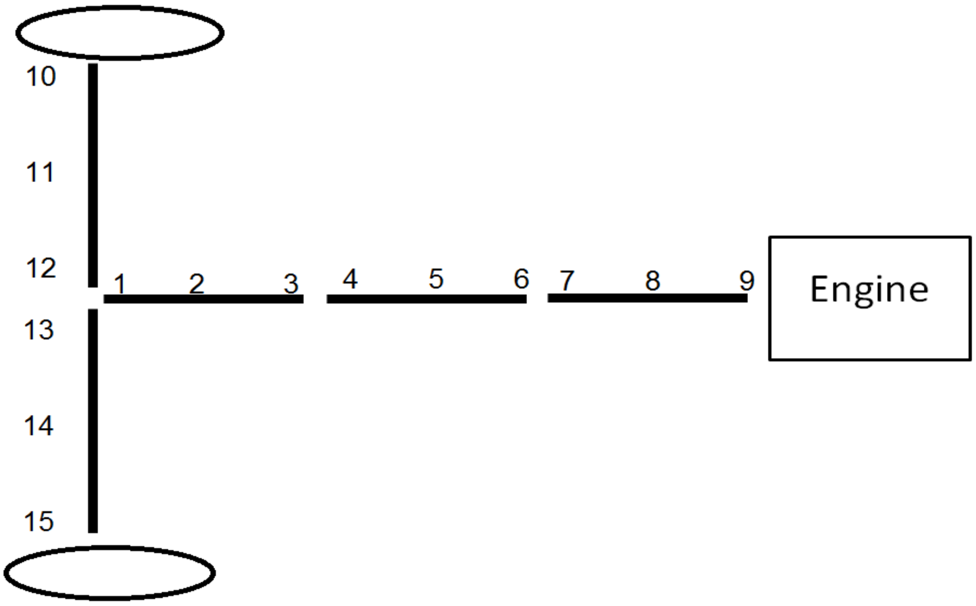

Figure 6.2: Test measurement points of SRW vehicle

Acceleration sensors located on driveshaft and rear axle tube and an hammer give impact on those components,accelorometer measure the vibration and gives info about the natural frequency.

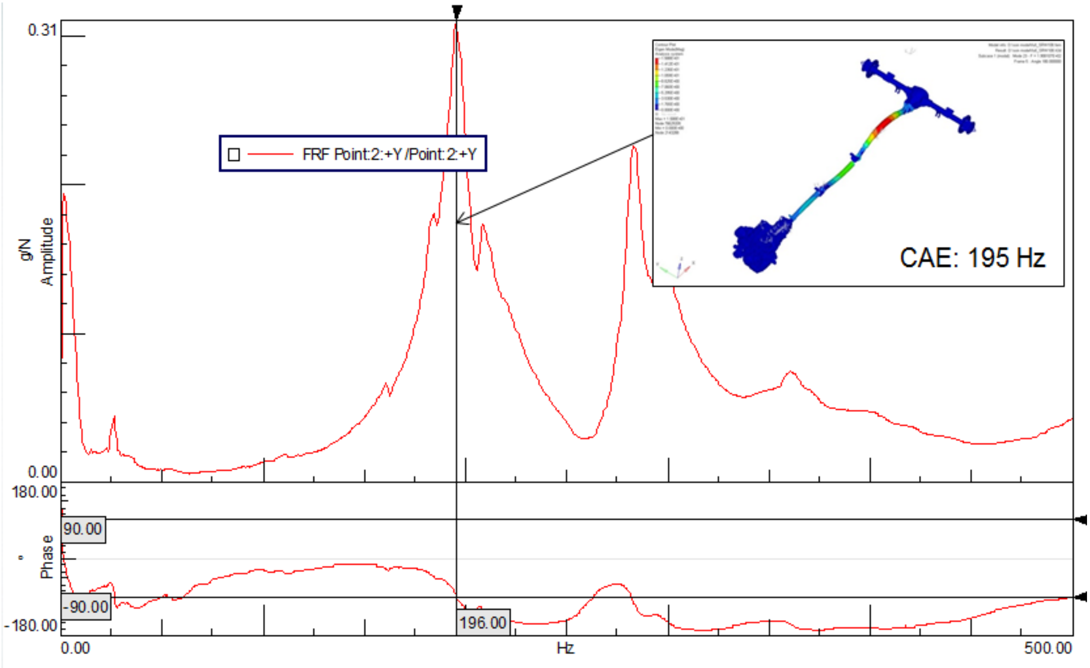

Figure 6.3: Rear propshats vertical bending mode

Figure 6.3 shows that the rear propshaft test result and it is value is quite close with Cae result. The vertical bending modes is 195 Hz at Cae and it is measured 196 Hz at test. The results shows the good correlation for rear propshaft.

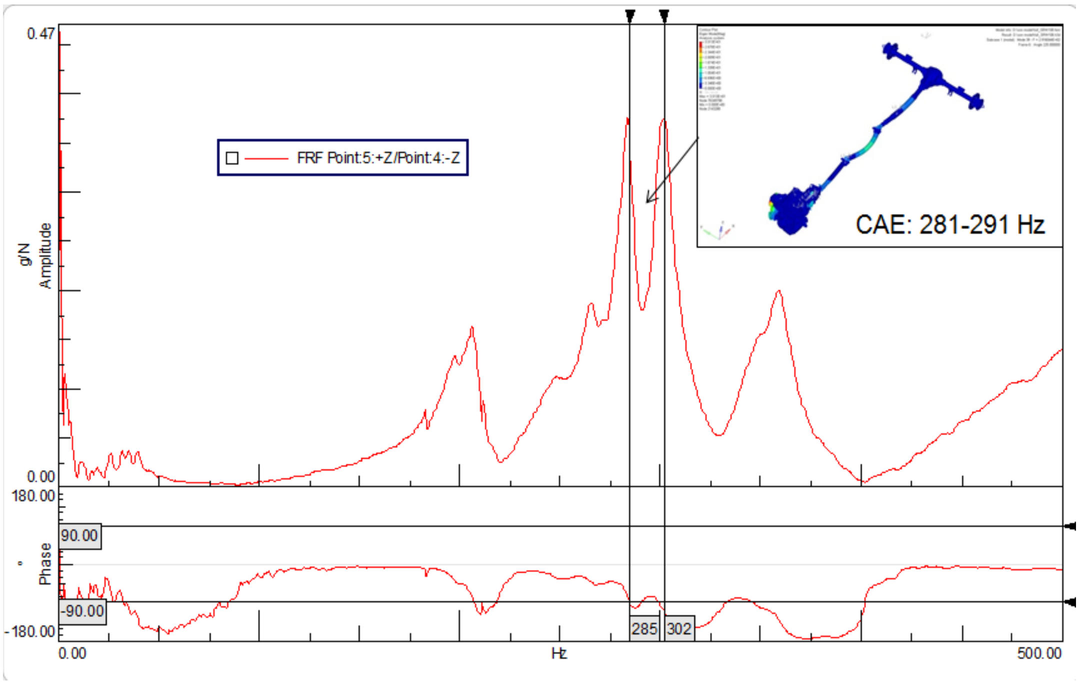

Figure 6.4: Middle propshats bending mode

Figure 6.4 shows the test result of middle propshaft. The first mode of propshaft is 285 Hz, which is 281 Hz at Cae, and second mode is 302 Hz and it is 291 Hz at Cae.

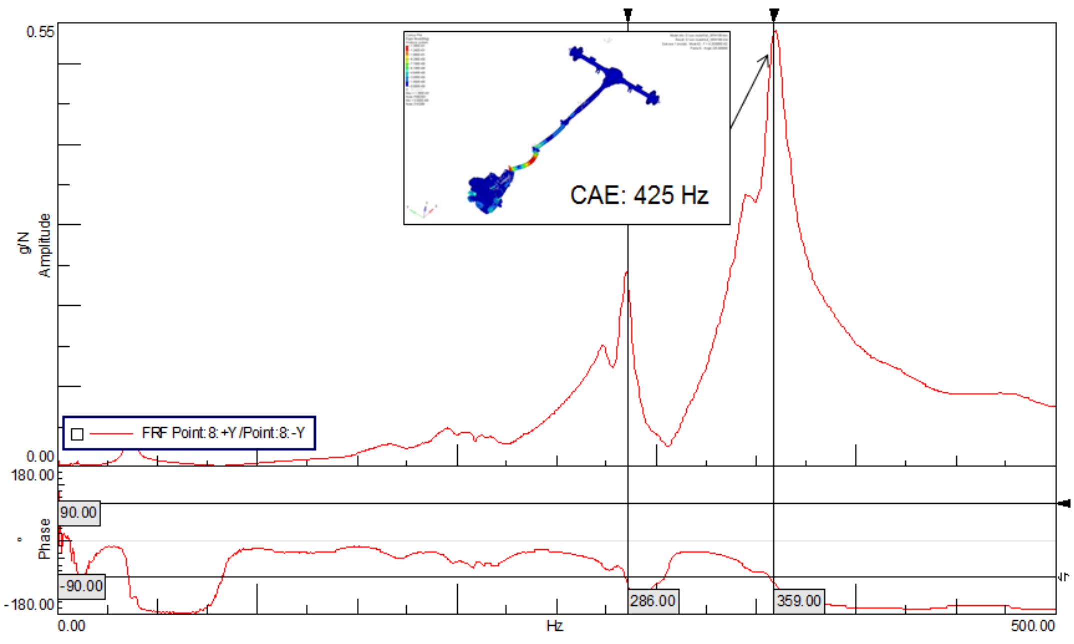

Figure 6.5: First propshaft bending mode

Figure 6.5 is showhs the hammer test result of first propshaft.The vertical bending mode is 359 Hz which is 425 Hz at Cae.Since the results are quite different finite element model should be modified.Stiffness values should be updated to make correlation with test.

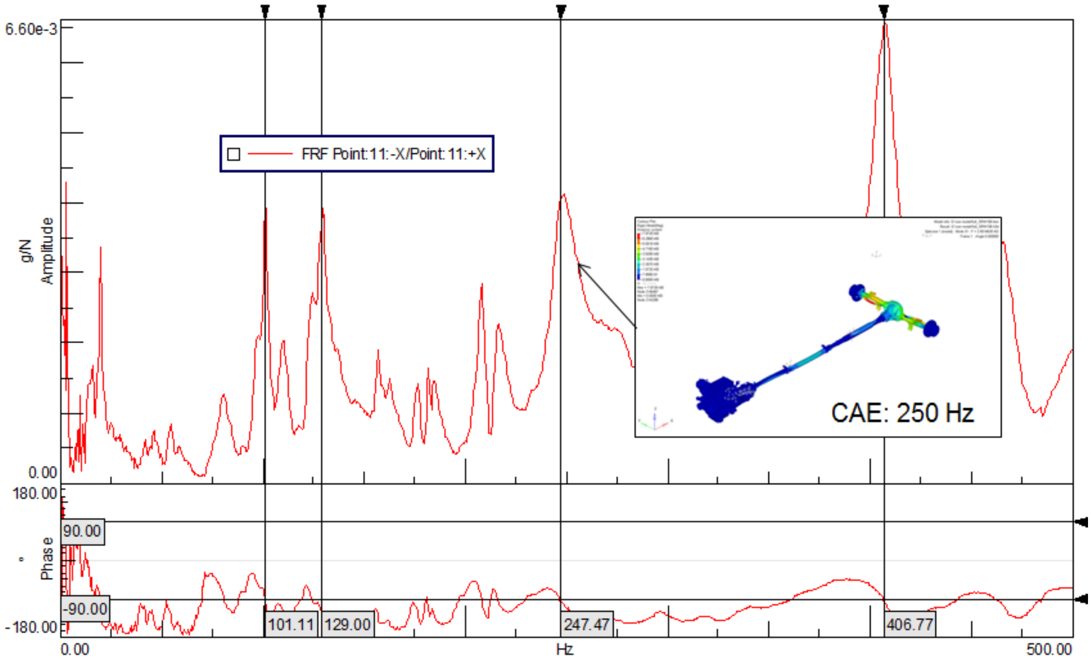

Figure 6.6: Rear axle tube lateral bending mode

Figure 6.6 shows the rear axle tube lateral bending mode. Analysis result is quite close with test where 247 Hz test measurement is 250 Hz at analysis. The good correlation between results shows that the finite element model is represent the physical situation very well and it is well enough to replace the tests.

Other test type is to measure noise level directly inside the vehicle at driver ear position. Since the vehicle level test represent the driver ear, it is not directly related with analysis results due to analysis only shows the natural frequencies but vehicle level test measure noise level directly. Even though it is not directly related the analysis results modal separation can be associate with measurements and for future studies this relationship can direct engineers to have idea about it.

For Proposal of axle change, SRW based vehicles were manufactured and nvh tests were performed. The comparison of SRW and DRW axle were performed in db and also according to vibration characteristic. Loudness level is measured on drive condition at driver ear location for both case.

6.2. Vehicle Level Tests

Vehicle level test is performed to understand how driver or passenger is affected by this change and to see the new axle is better or worse the current axle. To perform this test vehicle is donated by microphones, which is located at driver and passenger ear level to represent real situation. Figure 6.7 shows the vehicle interior measurement points.

Figure 6.7: Microphone location to record noise

Figure 6.8 shows the articulation index comparison of vehicle interior noise level. Articulation index is the parameter of passenger conversation convenience.

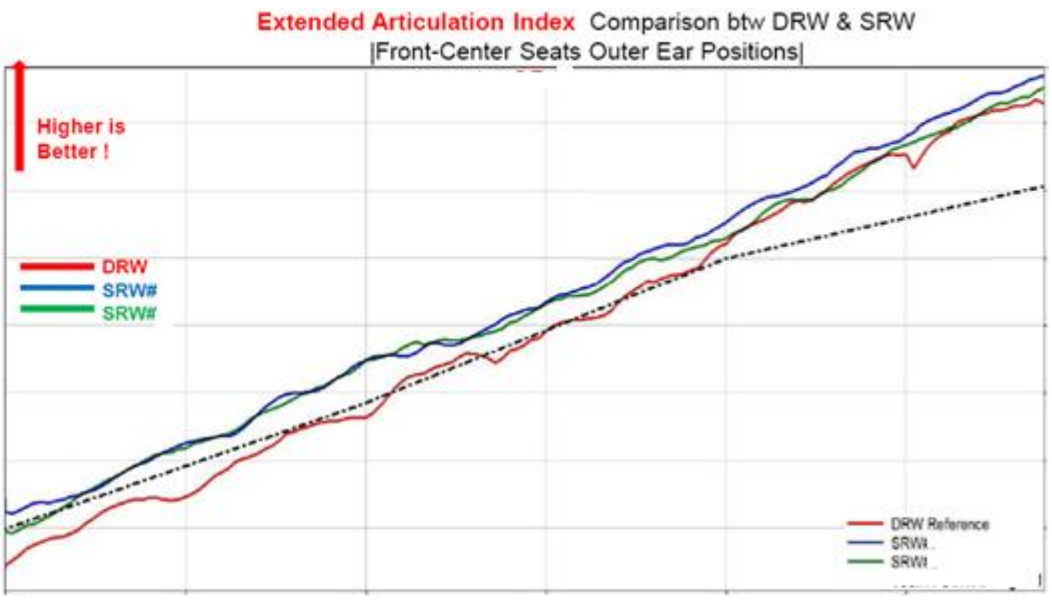

Figure 6.8: Articulation index comparison

According to graphs DRW axle worse than SRW axle. The articulation index is not directly related with axle variant, so more robust and true evaluation parameter is checking the RSS level of vibration. Figure 6.9 shows the RSS vibration comparison of DRW and SRW axle.

Figure 6.9: DRW and SRW axle RSS comparison

This graph shows the difference between DRW and SRW vibration level. Figure 6.9 clearly shows that SRW axle is generating more noise than DRW axle.

Additionally, order based comparison for different FDR ratios has been performed. 3th order chosen as dominant order because of pinion gear mesh number is 13.Colormap diagram is generally used for to see the order based noise and the red line shows the noise effect on diagonal line

Figure 6.10 shows the axle whine comparison of FDR ratios for SRW axle. The color map gives info about the noise according to order level where fdr 3.31 has 13 teeth, it is looked that 13th order and fdr 4.1 has 10 teethes therefore most dominant order is 10th for this ratio. Color map shows that interior noise perception of axle whine with 3,31 FDR rear axle is slightly louder than 4,10 FDR rear axle.

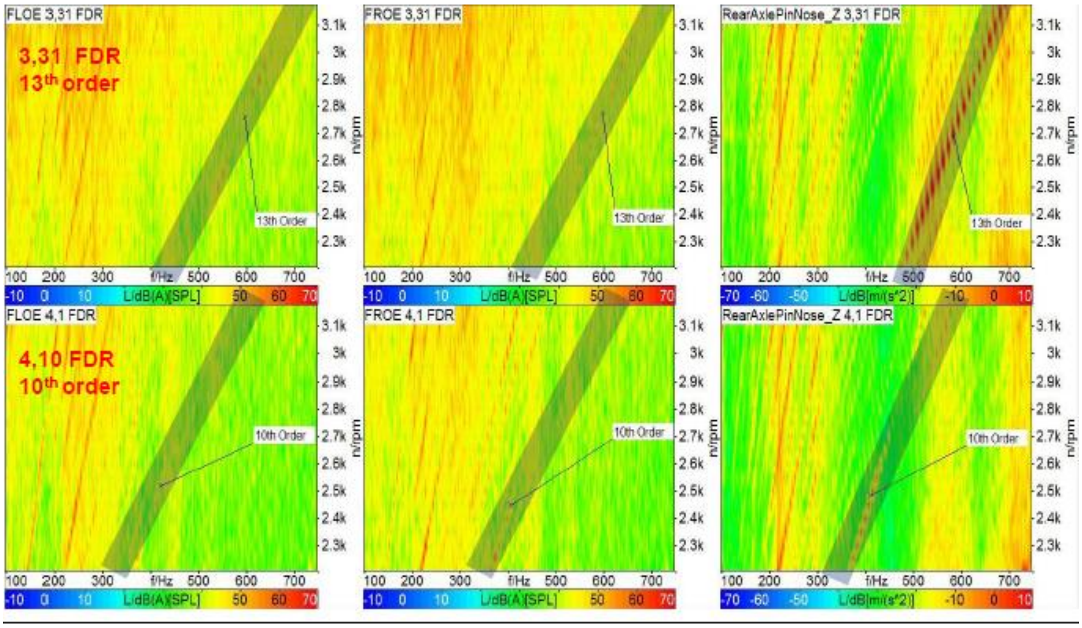

Figure 6.10: 10^{th} and 13^{th} order comparison of SRW axle

7. CONCLUSIONS AND RECOMMENDATIONS

Structural borne noise is a major problem for a manufactured vehicle and it is hard to understand whether a noise problem exists or not before the vehicle build. For this purpose automotive manufacturers use traditional method such as vehicle based test with using prototype vehicles which has significant cost for OEM(Original Equipment Manufacturer).According to the test result OEM changes design if problem observed but it generally turn back as a high budget cost changes and some changes could not be possible to implement to the vehicle because of time shortage of build plan. Therefore it is vitally important to predict the design sourced noise before the manufacturing a vehicle.

This thesis focused on design change of a commercial vehicle rear axle system and try to analyses it as finite element method and accordingly make prediction of noise generation. There are three different cases, which are axle type change from double rear wheel to single wheel, axle tube thickness change as a weight reduction item and final drive ratio change. The three different cases investigated and compared in between and according to analysis result, modal separation between components investigated.

The main rule is the separate modes as possible as much, which prevents the coupling of systems and also prevent the resonance and noise generation. According to analysis result DRW axle modal separation is higher than SRW axle therefore, it can be figured out that SRW axle generates more noise. This statement concurred with test results where 13^{th} order color map graphs compare the two cases.

Axle tube thickness change is investigated to change structure stiffness and also as a possible weight reduction item. It is figured out that tube thickness reduction worsening the current modal separation and therefore become convenient to generate more noise due to modal coupling.

FDR ratio change is not affected the structure stiffness so it is not possible to make an evaluation with current method. For this case pinion nose displacement is taken as comparison target and figured out that 3.31 FDR is generating more noise than 4.1 ratio

As a result, modal separation is the very convenient decision criteria to investigate the noise generation of systems. This criterion is mainly used for powertrain systems by most of the OEM and it is better enough to replace rig tests. The methodologies prevents time and money loss for companies and direct the design to produce more comfortable vehicles.

For further studies, structural borne noise analysis should be supported by radiated noise analysis which represent the noise generation by vibration of outer surface of components. The combination of two noise propagation analysis will provide more accurate results of systems

REFERENCES

[1]**He J. and Fang Z.** (2001).Modal Analysis 1st edition

[2]**Hussien A. and Shabana A.** (1999).Finite Element Dynamic Analysis Of The Rear Axle System, International Congress and Exposition Detroit, Michigan, March 1-4

[3]<http://auto.howstuffworks.com/auto-basics-channel.htm>date retrieved at 03.02.2015

[4]**Sen A**. (2012), Driveshafts_Training_v3,Ford Otosan PTI Awareness

[5]<http://authors.library.caltech.edu/25069/6/AirPollution88-Ch4.pdf>date retrieved at 25.04.2015

[6]**Dogan V.Z. (**2012), Structural Daynamics Lesson Notes

[7]**Crolla D. And Mashadi B.** (2012), Vehicle Powertrain Systems: Integration and Optimization

[8]Ford Motor Company Driveline Nvh Course Notes, 2001

[9]Ford Motor Company Driveline Nvh Course Notes, 2005

[10]<http://encyclopedia2.thefreedictionary.com/Rear+Axle> date retrieved at 21.03.2015

[11]**Carter M. (**2010), Driveline Nvh Course, Ford Motor Company

[12]Roadranger, Dana Spicer Drive Axles Application Guideline(2006)

[13]**Brandt A. (2011),** Noise and Vibration Analysis

[14]**Lilley K.M., Fasse M.J and Weber P.E. (**2001).A Comparison of NVH Treatments for Vehicle Floor pan Applications. Society of Automotive Engineers, Inc.

[15]**Hirasaka N., Sugita H. and Asai M. (**1999). A Simulation Method of Rear Axle Gear Noise

[16]**Steyer G. Voight M and Sun Z.** (2005). Balancing Competing Design Imperatives to Achieve Overall Driveline NVH Performance Objectives, American Axle and Manufacturing.

[17] <http://training.altairuniversity.com/modeling/meshing-2/1d-2/> date retrieved at 13.04.2015

[18]**Goalke M.** (2014).3D Meshing, Altair University Hypermesh Training Documents

[19]HGSIM 5.1 User Manual (2015)

[20]Transmission Error: The Metric for Gearbox Nvh Quality, Siemens PLM Software (2014)

[21]**Henriksson M. (**2009)On noise generation and dynamic transmission error of gears

[22]<http://constructionmanuals.tpub.com/14273/css/14273_188.htm> date retrieved at 25.04.2015

CURRICULUM VITAE

Name Surname: Mustafa Yıldırım

Place and Date of Birth: Bursa-06.03.1989

E-Mail: mustafa.yildirim16@yahoo.com

EDUCATION:

B.Sc.: Sakarya University

PROFESSIONAL EXPERIENCE AND REWARDS:

- Erasmus Student-06.2010-08.2010-Applied Science of Dusseldorf University
- Chamber of Mechanical Engineers membership(2011)
- Mechanical Engineer at Ford Otosan(02.01.2012/-)