ISTANBUL TECHNICAL UNIVERSITY ★ GRADUATE SCHOOL OF SCIENCE ENGINEERING AND TECHNOLOGY

NVH AND STRUCTURAL ANALYSIS OF SHIFT CABLE HOLDER BRACKET

M.Sc. THESIS

Doğukan BİLİCAN

Department of Mechanical Engineering

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Thesis Advisor: Dr. Alper Tolga ÇALIK

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ISTANBUL TEKNİK ÜNİVERSİTESİ ★ FEN BİLİMLERİ ENSTİTÜSÜ

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Doğukan Bilican, a M.Sc. student of ITU Graduate School of Science Engineering and Technology student ID 503131724, successfully defended the thesis entitled "NVH AND STRUCTURAL ANALYSIS OF SHIFT CABLE HOLDER BRACKET", which he prepared after fulfilling the requirements specified in the associated legislations, before the jury whose signatures are below.

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Date of Submission : 04 May 2018
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To my grandfather and my lovely family,

FOREWORD

I am very grateful to work with valuable Dr. Alper Tolga Çalık and Dr. Osman Taha Şen during my master thesis. They opened my view with their experience and guidance all the time with their kind and patience support.

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To my friends who always support me, I appreciate very much.

To my lovely family and especially my spouse I present my love and respect.

May 2018 Doğukan BİLİCAN

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ABBREVIATIONS

ACC	: Accelerometer
BP	: Band Pass
BR	: Band Reject
CAD	: Computer Aided Design
CAE	: Computer Aided Engineering
DOE	: Design of Experiment
ECM	: Engine Control Module
FEA	: Finite Element Analysis
FEM	: Finite Element Modal
FFT	: Fast Fourier Transform
FRF	: Frequency Response Function
HCV	: Heavy Commercial Vehicle
HP	: High Pass
LP	: Low Pass
NVH	: Noise Vibration Harshness
PBN	: Pass By Noise
PT	: Powertrain
RHD	: Right Hand Drive
SDM	: Structural Dynamic Modification
TPU	: Thermoplastic Polyurethane
VECV	: Volvo, Eicher Motors Commercial Vehicles

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NVH AND STRUCTURAL ANALYSIS OF SHIFT CABLE HOLDER BRACKET

SUMMARY

NVH (Noise, Vibration and Harshness) fundamental defines undesirable sound, tactile sensation and subjective impression from particular characteristics of the noise/vibration. Due to legislation and customers' noise sensitiveness increasing with design of more quiet vehicles (increased sound quality) NVH studies became a very significant goal day by day. Confliction of development time, cost etc. and the design of vehicle with good sound quality is major engineering problem. In example of important engineering working for powertrain NVH engineers is not only check the good sound quality but also perception of sound from driveline should be met with customer expectation. In this NVH study, the produced test vehicle will check for focusing of powertrain NVH side especially to transmission and external control system. Data collection from test vehicle will be described with field test and tool specifications. After subjective evaluation "Knocking noise happens especially between 1100 - 1500 rpm" reported and to solve this issue the source identification will be performed from objective data that shows the source of abnormal noise is shift cable holder bracket. Transfer path can be described as shifter cable bracket (source), shifter cable, shifter box and shifter arm. To understand NVH characteristic of shift cable holder bracket experimental modal analysis (hammer excitation test) has been performed to two production brackets from different production batch. Mode frequencies are reviewed different at two brackets hypothesis has been done according to connection stiffness of the bracket mode shapes can be reviewed at different frequencies. Thus computational modal analysis accoplished with Altair Hypermesh and ABAQUS softwares to compare of experimental modals and computation modal, and it is verified experimental modal and computation modal is matched with good connection stiffness. In addition, hypothesis is also verified with lack of connection stiffness can affect mode shape frequencies at computational analysis. NVH solution is also suggested with reinforced bracket design and it is verified with computational analysis and prototype part at vehicle. Furthermore, structural durability analysis performed with comparasion of baseline and reinfored bracket design to double check that proposed design is better than in aspect of the sound quality and structuraly. Structural durability analysis is done with ABAQUS software with data that data mesh and connection stiffness verified computational modal analysis and it is carried out according to fatigue life criteria of Von Mises.

VİTES HALATI BRAKETİNİN NVH VE YAPISAL ANALİZİ

ÖZET

İstenmeyen ses ve gürültünün tanımlanması, titreşim kaynaklı parçalarda meydana gelen dokunsal hissiyat ve bunların özel olarak kullanıcı tarafındaki subjektif algısı NVH (ses, gürültü ve sertlik) esaslarına göre belirlenmektedir. Her geçen gün daha sessiz ve ses kalitesi iyileştirilmiş araç dizaynlarının yaygınlaşmasıyla birlikte kullanıcının ses ve gürültü hassasiyeti artmaktadır. Hem kullanıcının ses hassasiyeti/algısına dayanarak ve regülasyonlara uygun, hem de günümüz şartlarında araç geliştirme süresince uygun fiyatla istenilen iyi derecedeki ses kalitesine sahip araç dizaynı birincil mühendislik problemlerinden biri haline gelmiştir. Örneğin; yeni geliştirilmiş bir güç aktarma organından kaynaklı ses ve gürültü probleminin değerlendirilmesi, ilgili güç aktarma organları NVH mühendisi tarafından sadece ses seviyesi incelemesi olarak değil ayrıca subjektif olarak ses kalitesinin kullanıcı (müşteri) gözüyle değerlendirmesi olarak yapılmaktadır. Ağır ticari kamyona yeni bir şanzıman aplikasyonu uygulandığından, bu çalışmada ilgili aplikasyonun uygulandığı bir test aracında motor ve güç aktarma organları kaynaklı ses ve gürültü değerlendirmesi bu kritere göre yapılmıştır. Bu değerlendirme de özellikle yeni şanzıman aplikasyonu ile birlikte değişen tüm parçalarda, kısacası şanzıman ve vites geçiş sistemine odaklanarak araç için önce subjektif değerlendirme ve sürüş sırasında toplanan ivmeölçer datalarıyla da objektif değerlendirme yapılmıştır. NVH subjektif ve objektif araç saha testinin nasıl yapıldığı, ivmeölçer ve mikrofon datasının toplanışı ile kullanılan araç gereçlerin konumlarıyla beraber spek detayları araç motor ve güç aktarma organları test prosedürü şeklinde çalışmada belirtilmiştir. Aracın ilk subjektif değerlendirmesinde sürücü ve mühendis tarafından "Özellikle 1100-1500rpm arasında vites mekanizmasından gelen ve kabin içinde duyulan vurma sesi" raporlanmıştır. İlgili NVH probleminin raporlanmasına nazaran ivmeölçer dataları yardımıyla ilgili ses ve gürültünün kaynağının bulunması amaçlanmıştır. Kabin içerindeki mikrofon datası frekans süzgeçleri konularak dinlendiğinde ilgili problemin 750Hz frekansında olduğu belirlenmiştir. Şanzıman, vites kulesi, vites levyesi, vites halatı braketi, vites mekanizması ve vites mekanizması levyesi üzerine takılan ivmeölçerler sayesinde objektif data üç eksenli grafikler (colourmap) üzerinden kontrol edilmiştir. Kontroller sırasında şanzıman, vites kulesi ve vites levyesi datalarında bu rezonans bandı ses ve gürültü izine rastlanmamıştır. Fakat vites halatı braketi, vites mekanizması ve vites mekanizması levyesinde bu sesin izleri üç eksenli grafiklerde (colourmap) yardımıyla doğrulanmıştır. İvme tahrik miktarlarına bakıldığında ve ilgili izin görüldüğü kaynakların sesleri dinlendiğinde vites halat braketinin ses ve gürültünün kaynağı olduğunu gözlenmiştir. Vites halatı üzerinden patika yolu izleyerek vites mekanizması, vites mekanizmasının levyesi ve kabin içinde ses problemimi oluşturduğu da kanıtlanmıştır. Dolayısıyla ilgili ses yapısal yolla braketten kabin içine kadar taşınan bir ses ve gürültü problemi olarak sınıflandırılmıştır. Ses ve gürültünün kaynağı olan vites halatı braketinin NVH ve yapısal karakteristliğini detaylı bir şekilde incelemek için deneysel modal analiz (çekiç testi) ITU Otomotiv Laboratuvarında yapılmıştır. Deneysel modal analiz

tamamen aynı dizayna sahip fakat imalatı farklı zamanda yapılmış iki adet şahit numuneye yapılmıştır. Kullanılan üç eksenli ivmeölçerin ve çekicin spekleri verilmiştir. İlgili test serbest koşullu bağlantıyı simule edebilmesi için bağlantı delik lokasyonundan sadece misine yardımıyla askı aparatına bağlanmıştır. Vites halatı braketine evrensel bir koordinat sistemi ve orjin tanımlanmıştır. Yapılan ölçümler sırasında ivmeölçerin lokasyonları ve ivmeölçerlerin eksen takımları evrensel koordinat sistemi ve tanımlanan orjine uygun yapılmıştır. Toplam 22 farklı noktadan ölçüm (ivmeölçer datası) alınmıştır ve çekiç her seferinde aynı noktadan uygulanarak tahrik verilmiştir. Analizör ile toplanan datalardan FRF (Frequency Response Function) ve coherence (tutarlılık) fonksiyonları MATLAB'a aktarılıp incelenmesi amacıyla 22 farklı ivmeölçer lokasyonu için 3 tekrarlı olarak kaydedilmiştir. MATLAB programı yardımıyla öncelikle tanımlamalar (ivmeölçer pozisyonları, eksenlerin girilmesi, grafiklerin çizilmesi vb.) yapılmıştır. Tanımlamalar sırasında braketin şekli nedeniyle bazı ivmeölçer pozisyonları için rotasyon matrisleri kullanılmıştır. Sonuç olarak FRF (Frequency Response Function) grafikleri lineer ve logaritmik çizilerek mod frekans aralıkları belirlenmiştir. Tanımlamalar sayesinde ilgili noktalar arası çizgiler çizdirilerek mod frekans aralıklarında mod şekilleri belirlenmiştir. İki şahit numunenin deneysel modal analiz sonuçlarının farklı olması ve ilk testte düşük frekansta bir mod olması nedeniyle parçalar detaylı olarak tekrardan kontrol edilmiştir. Parçalar arasında kaynak miktarı ve slot bağlantı civatasının torkunun farklı olması nedeniyle bağlantı sertliği farkının mod frekanslarını öne çekme ve ya öteleme oluşturabileceği hipotezi ortaya atılmıştır. Hipotezin doğrulanabilmesi için bilgisayar ortamında da modal analizi yapılarak sonuçların korelasyonu sağlanmıştır. Altair Hypermesh programı kullanılarak sonlu elemanlar modeli kurulmuştur. Hypermesh üzerinde mesh edilen modele malzeme, özellik ve mod frekanslarını bulmak için fonksiyon tanımlanarak model ABAQUS Solver'a aktarılarak çözümlenmiştir. Çözüm sonucunda ilk dört frekans alınarak deneysel modal analiz ile karşılaştırılmış ve ikinci yapılan deneysel modal analiz ile kurulan sonlu elemanlar modal analizinin %0,15-%5,41 arasında hata paylarıyla doğruluğu gösterilmiştir. İlk braket ile yapılan deneysel modal analizin mod frekanslarının neden ötelenmiş olduğunu sonlu elemanlar modelinde göstermek ve kurulan hipotezi desteklemek için Hypermesh modelinde farklı sertlikte bağlantılarla iterasyonlar yapılmıştır. Vites halatının klips ile bağlandığı üst kısım ile ana bağlantı parçası arasında slot bağlantılardan birinin gevşek olmasının üst parçanın görülen mod şekillerinden birini düşük frekansa doğru öne çektiği doğrulanmıştır.

Braket kaynaklı NVH problemini kaynakta çözmek için braketin ilk dizaynında ana parçanın altına bir kaburga (bayrak) destek parçası eklenmesi ön görülmüştür. Bu dizayn ile parçanın sertlik (stiffness) değerinin arttırılması sağlanılarak kafa vurma, eğilme gibi mod şekillerindeki tahrik kaynaklı hareket miktarını azaltacağı düşünülmüştür. Bu tasarım için de sonlu elemanlar modal analizi ile mod frekansları ve mod şekilleri incelemesi yapılmıştır. Bu analizin sonucunda özellikle ilk dizaynda görülen ilk modun (587Hz) yukarı frekansları ötelendiği ve diğer modların frekanslarının da bir miktar yukarı ötelendiği görülmüştür. Ayrıca deplasman değerlerinde de azalmalar olduğu da saptanmıştır. Bilgisayar ortamında doğrulanan çözüm tasarımın prototip numunesi alınarak araç aplikasyonunda subjektif ve objektif olarak değerlendirmesi yapılmıştır. Subjektif değerlendirme sonucunda kabin içerisinde ilgili şikayet tekrarlanmamıştır. Aynı araç testi konfigürasyonu ve bayraklı braket tasarımıyla alınan objektif datalarla ilk durumun karşılaştırılması sonucunda ilgili frekans bandı etrafında düşük deplasmanlı mod frekansları görülmüş ve bunların ses ve gürültü şikayeti oluşturmadığı gözlenmiştir.

Çalışmanın son kısmında braketin ilk ve bayraklı tasarımı dayanımsal olarak yorulma analizine tabi tutulmuştur. Analizde yük olarak vites geçiş kuvveti ilgili araç konfigürasyonunun mekanik oranına nazaran teorik olarak hesaplanmıştır. Analiz sonucunda gerilim değerlerine bakıldığında her iki tasarımın da güvenli bölgede sonsuz ömürlü olduğu açıkça görülmekte ve her iki tasarım arasında kaburga (bayrak) eklenerek tasarımın gerilimde etkisi nerdeyse hiç (0,29%) olmamaktadır. Fakat deplasman değerleri karşılaştırıldığında bayraklı tasarımın etkisinin daha yüksek olduğu ve deplasmanın azaltılmasında ciddi katkı sağladığı (%8,46) raporlanmıştır. Ek olarak, deplasman karşılaştırması braketler üzerinden toplanan ivmeölçer dataları yardımıyla da yapılmış, gerçek iyileşmenin bilgisayar analiziyle sadece %0,77 oranında farklılıkta (%7,69) olduğu raporlanmıştır. İlgili tasarımın hem NVH problemini çözümleyerek hem de yapısal olarak deplasman miktarını azaltarak iyileştirme sağladığı doğrulanmıştır.

1. INTRODUCTION

To accomplish expectations of customer according to NVH characteristic of vehicle interior, both sound level and sound quality are major points have to be considered. Methodology of decrease sound level and satisfy exterior noise reduction is starting with source identification has to be studied to found out leading noise factors. Mainly transmission, mufflers, air intake, turbochargers etc. are the sources for noise contribution. Vibration measurements of component with experimental modal analysis and near field noise can be used to determine the dominant noise source. After defining of dominant factor with comparison of overall noise, modification in the component can be done and pass by noise (PBN) measurements will be done to collect and compare with baseline (Padavala et al, 2017).

The sound quality of vehicle interior is significant assignment and forcefully challenge for the engineers. Since, vehicle production cost and total weight of vehicle are conflicted to the good sound quality goal, engineering task is became as a challenge. The sound quality can be defined as satisfying of individual requirements that are physical, psychoacoustic and psychological. In basic, sound field does not disturb to customer and, perception/evaluation of audit has to be matched with their expectation.

Nevertheless, the undesirable noise inside the vehicle (cabin) may be reported and it is coming from input signals related to modification of the engine or at single transfer path. Source identity has to be found out exactly to diminish or cancel this abnormal noise with transfer path analysis that is airborne noise or structure borne noise. Structural borne noise can be defined as felt as vibration and heard as noise is transmitting through parts (powertrain mounts, bushes or brackets). Air borne noise can be defined radiated noise from parts (orifices, tailpipes etc.) that are contributing vehicle interior noise. In addition frequency range is also classified noise path with upper frequency limit of structural borne noise is 2kHz and air borne noise is considering shares up to 12kHz (Genuit, 2004).

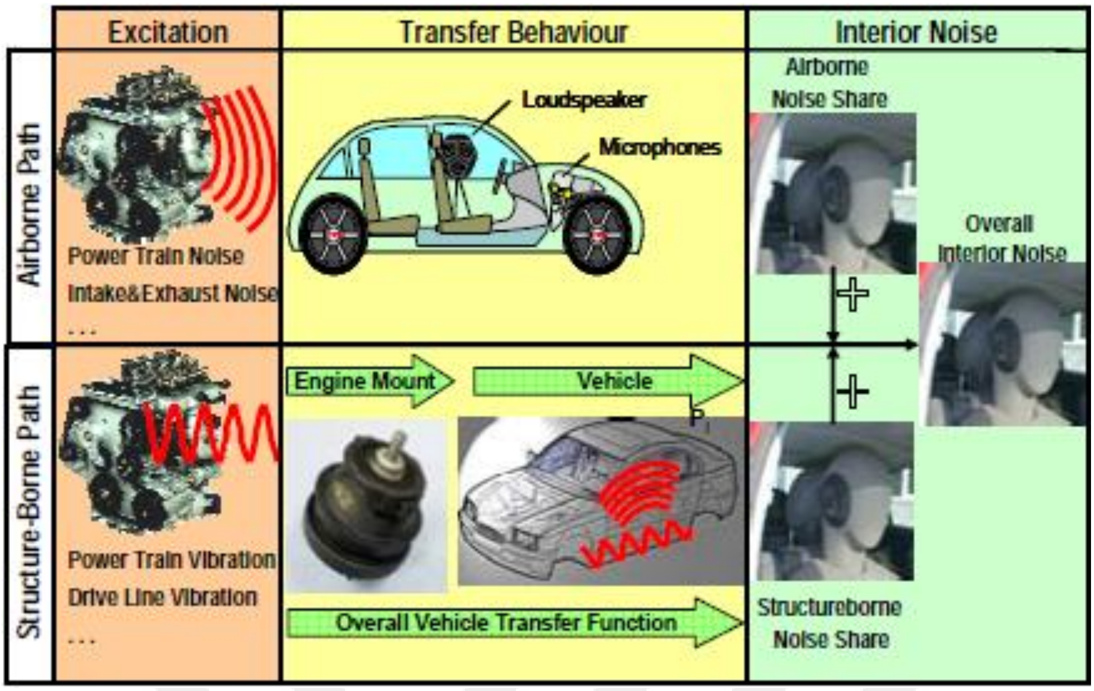

Figure 1.1: Methodology of transfer path (Govindswamy, n.d.).

Due to fact that while powertrain development or new powertrain component installation to vehicle some adjustments have to be performed to satisfy application works functionally and qualified for NVH. According to that whatever components changed at powertrain, NVH characteristic has to be checked all of powertrain elements work together properly to ensure good sound quality.

1.1 Literature Review

Firstly, related to NVH point of view several important studies have been fulfilled. These are focussed to investigate and solve problems from powertrain related to engine, driveline, transmission or external system. Researchers are mainly study to understand abnormal noise source and transfer path for issue by issue and they have performed tools such as experimental modal analysis, finite element modal analysis and source identification.

Brown and Allemang (2007) have been collected research from historical perspective for experimental modal analysis how it is came past to these years and they defined modern era for experimental modal analysis. They did not discuss about how it is started however they informed possible starting point can be taken as development of Prony (1793) or Fourier (1822) or Wheatstone Bridge (1843) experimental tests and modal analysis. Modern era starts may be interpretted to 1960s when definite records can be collected measurements of force and motion. In addition, tracking filter, FFT algoritm, real time analyzer etc. are briefly informed as key technological breakthoughs and role of University of Cincinnati is given at the article.

Avitabile (2001) has been identified how structure vibrate and how the structural dynamic problems can be solved using with some of the tools without mathematical representations. He described modal analysis with basic plate example to understand easily without mathematical explanations. In addition, basic terms that are used at modal analysis literature is defined such as FFT, FRF, coherence, reciprocity mode and mode shapes with graphical representations etc. and shaker test, impact test details have been informed with also data collection details.

Campbell et al. (2005) have been studied of vibration levels at the one of the main customer contact point is the shifter lever. In the study, it is verified that shifter cable is significant transfer path and also each pieces of the cable contributed to the transfer of vibration. To prove this they performed try out with cable by connecting and disconnecting the cable core or conduit from the lever. (The cable assembly either completely assembled as a standard operating configuration, or with the cable core disconnected, or with the cable core and conduit disconnected.) The 240Hz and 300Hz resonances dominated the low frequency region that is focused since this is sensitively range for human tactile interaction. CAE analysis of system performed that is 4th orders engine-excided to these resonances has been correlated. Design of Experiment (DOE) matrix defined to give design recommendation and direction in order to reduce shifter lever vibration using the most influential factors that are increasing cable core stiffness and reducing cable conduit stiffness or clearance between the cable conduit and stranded cable core can be decreased.

As it seen, major transfer path of noise and vibration from transmission to driver contact point is shift cables. Through parts life cycle deformation of NVH materials used at cable systems is investigated by Ruhlander (2006). Study is addressed to importance of isolation materials and giving information to design improved shift system products both satisfy NVH isolation and durability to perform their mechanical functions properly. In addition, study verified that after life cycle test due to isolation material (TPU) deformation extraction and insertion forces dropped to between 28% - 37% and rod end lash increased up to 20%. Increased lash value affected to decrease shift performance however at this study it is still in product limitations. This investigation proves isolation material has to be considered at different cable application according to shift performance, durability and NVH characteristics. Future NVH investigation related to shift cable end fitting can be done as recommended.

Singh et al. (2013) have been conducted research for improving vehicle acoustic with NVH optimization at transmission shifter. Mainly, impact of shifter is investigated due to the design of the top lever (selector) is important to get optimal feel at user contact and mounting of shifter is crucial to increase sound quality. Comparison of different mounting configurations of semi remote shifter (transmission, vehicle body and chassis mounted) have been done according to see impact on vehicle vibration. Effect of mounting location is evident thus optimization work has been performed based on vehicle configuration using mathematical model & NVH analysis with computational software.

Jandav (2014) focus on studing to reduce powertrain noise especially reported as gear rattle and whine. Gear parameters and clutch damper characteristic is optimized to reduce undesirable noise on transmission with source identification. Methodology of investigate the issue at transmission is the source/generator, the transfer path of noise and receiver. An overview and detailed investigation can be reviewed for this two more common transmission noise problem that are gear rattle and whine according to problem categorize: nature of noise, source(s), mean torque load and key factors of noise. Whine and rattle noise classification have been done based on order tracking. Furthermore, optimization details have been studied for transmission gear and clutch dampers to avoid undesirable noise from transmission.

Singh et al. (2015) is also investigated noise contribution of transmission on passenger cabin (interior). Changing of engine characteristics (engine rpm fluctuation increased 25-40 rpm at specific rpm interval) affects the transmission gears to vibrate (rotational vibration) thus generation of noise from driveline occurs. Solving of this problem at the source (engine) is also affects the FE targets of the vehicle and hard to change, thus the issue is solved at transfer path. With vibration sensor at different regions, transfer path is seen transmission to cable holder bracket and cables. (The issue was not reported at different vehicle (RHD) variants that including longer cables.) Cable mode is changing with additional mass at connected to cable and isolation between cable and mounting bracket was satisfied with soft rubber. 6dB improvement has been done with this action.

Kandreegula et al. (2017) have an investigation focused to propeller shaft mounting bracket according to finite element simulation and validation for this analysis. To obtain exact forecast of the performance of propeller shaft mounting bracket they performed simulation with three steps such as modal analysis, static and dynamic analysis (frequency response analysis). FEM model structure is studied with Altair Hypermesh (pre-processor and post-processor) and MSC Nastran (solver). Due to part's behaviour depends on modal characteristic natural frequencies and mode shapes are determined by modal analysis firstly. Static analysis performed according to stress that was observed conditions of braking, cornering and loading instances. Time varying loads are applied to report out dynamic analysis conditions. In addition, simulations have been verified and correlated with experimental test rig setup that is prepared according to VECV standarts.

Chimento et al. (2018) have been presented free and random vibration analysis results to determine dynamic characteristic of Engine Control Module (ECM) assembly that is mounted to chassis frame. Firstly, free vibration study has been performed to determine natural frequencies with hammer excitation experiment. Using with LMS SCADAS and LMS Test Lab recorded data process to obtain mode frequencies and shapes that is 48Hz (bracket bending), 210Hz (bracket twisting) and 315Hz (combination of first and second modes). Finite element modal analysis is also performed to compare with experimental modal that shows FE modal analysis results is directly changing with or without pre-stress effects due to stiffness change. Results of modal analysis with pre-stress effect correlated as modes are 57Hz, 205Hz and 333Hz with hammer impact test. Furthermore, random vibration study is performed to compare experiment and FEA results.

Secondly, structural performance of part is always one of the main criterias that has to be investigated to understand part behavior with modal analysis at firstly described. Researchers are verified component's life is satisfied with vehicle life at least with simulation and analysis.Remnant life analysis, static and dynamic structural analysis and random vibration analysis can be performed according to investigated part behavior.

Shift cable holder bracket that is mounted to transmission clutch housing is not studied directly in articles; however, logic is similar with different bracket applications at any other automotive system brackets. Mainly, the bracket is a supporting element for overhanging object and brackets come in different shapes and sizes for different types of uses in automotive. Other than shift cable holder bracket, different bracket applications are studied by researchers.

Khan et al. (2011) have been evaluated structural/fatigue life of auxiliary heater bracket with random vibration analysis, since bracket is mounted to chassis frame and component subjected to random excitations from the road during its life. This bracket application is different from than driveline bracket parts however it is good example to see random vibration analysis at component mounted to chassis frame. FEM model was created in Altair Hypermesh and it is correlated with application excitations from accelerometers. Fatigue life evaluated according to analysis and design is improved to satisfy required life time correlated again with FEM.

Prasad et al. (2015) have been studied one of the significant driveline brackets is center bearing bracket mount of a propeller shaft which plays crucial role in overhanging the first half of the propeller shaft with ball bearing to the chassis frame. They evaluated failure analysis of this bracket due to if fatigue failure happened that may lead to operational failure of the propeller shaft and result will be transmission failure. In this study, FEM analysis was performed with ANSYS software carried out the static structural analysis according to Von Misses Stress approach.

Subramanian et al. (2017) have an investigation regarding to structural durability performance of gear shift lever bracket according to remnant life. Under two different types of load: inertial load due to engine vibration and gear shift load, crack can observe in the field on bracket and it can affect overall life of component. At this study, high resonance on bracket was occurred while the engine was working at idle. Test methodology provided to this application with existing test sample information. Consequently, it is shown that useful life of the bracket is not majorly effected by crack initiations. It is established with correlations of design parameters and field usage profile data.

1.2 Scope of Research

In this study, due to new transmission and external control system application at heavy commercial vehicle PT NVH assessment has been done. According to assessment the abnormal noise reported “Knocking noise happens especially between 1100 - 1500 rpm”. Thus, based on statement source identification has been performed via color map and microphone data via filters and found out that abnormal noise source is shift cable holder bracket application for fixation of selector cables.

Structural Dynamic Modication (SDM) metholodogy has been followed for shift cable holder bracket has been deeply investigated with experimental and finite element modal analysis. According to developed modal model part characteristic is deeply investigated and rib stiffener addition has been adjusted to part for shifing especially first mode to upper frequency. Proposed SDM (Reinforced Bracket Design) is checked with obtained modal model and prototype part at vehicle and it is verified in terms of modal model frequencies and mode shapes and vehicle objective data with comparison of baseline vs reinforced bracket design.

In addition, structural analysis performed computationaly to compare reinforced design is also shown at least same performance with baseline design based on structural characteristic (Von Mises criteria). Significant improvement reported according to displacement values that are also compared via FEM and accelerometer data collection from vehicle.

For better understanding short description of investigated shift cable holder bracket can be advised. Part is using for attachment point of shift cable with C-clip and it is mounted with three M8 bolts at transmission. Baseline design of braket contains three parts main (L shaped section – green coloured at 3D top view), slot hole connection part (blue coloured) and C-clip connection part (gray coloured). Slot connection has been satisfied with M8 apex bolts and with slot holes design shift cable stoke is adjusted to correct cable stroke with aperture at assembly station. C clips connection part and slot holes connection part is welded inside. Middle hole at slot connection is assembly fixation poke-yoke feature.

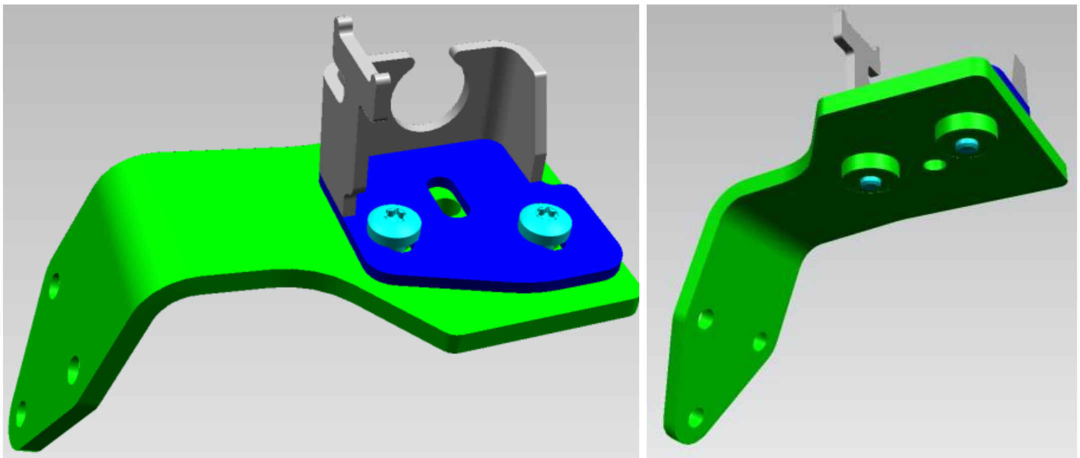

Figure 1.2: 3D CAD view of shift cable holder bracket.

2. POWERTRAIN NVH ANALYSIS

2.1 Data Collection From Vehicle

2.1.1 Vehicle NVH test general remarks

Vehicle test performs at each gears since the transmission order is changed with gear couples work together (corresponding number of teeth that works together) to transmit torque. To calculate transmission order the power flow and number of teeth of gear couples are required. With this information basic calculation may be done for each selected gears to define orders and if any transmission related noise (rattle and whine noise) could be observed at collected data with root cause that which gears are created this noise.

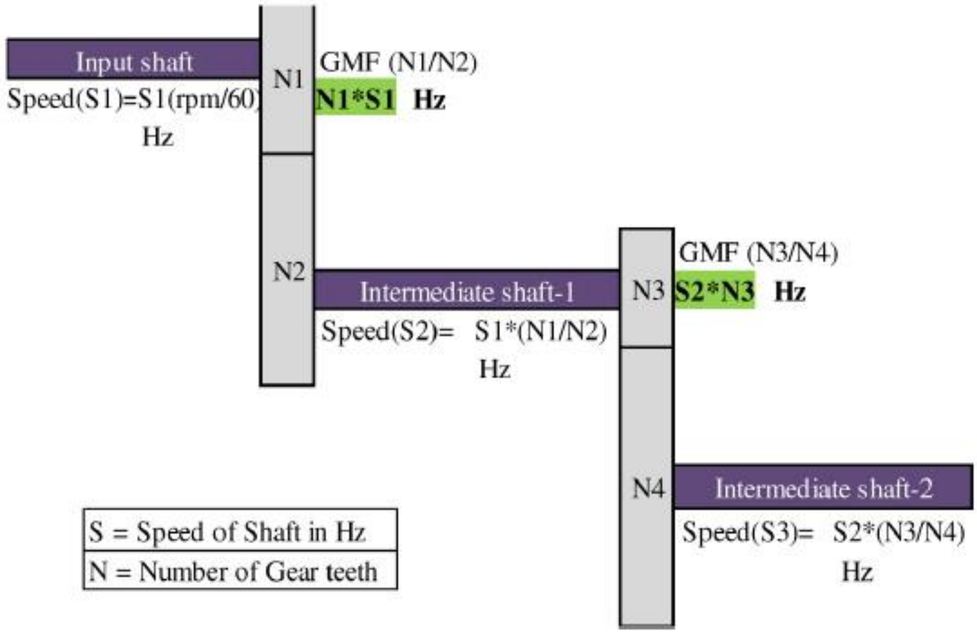

Figure 2.1: Illustration of transmission gear train and computation of gear mesh frequencies (Jadhav, 2014).

To sweep all used rpm range data is collected all gears from low idle (750 Hz) to high idle (2500 Hz). Acceleration and deceleration data is collected with same configuration between idle rpms to observed speed sweep and coast down data.

Multiple runs have to be conducted in each configuration to ensure repeatability. Various operating conditions have to be tested, including idle, part load and full load sweeps in all shifter lever positions.

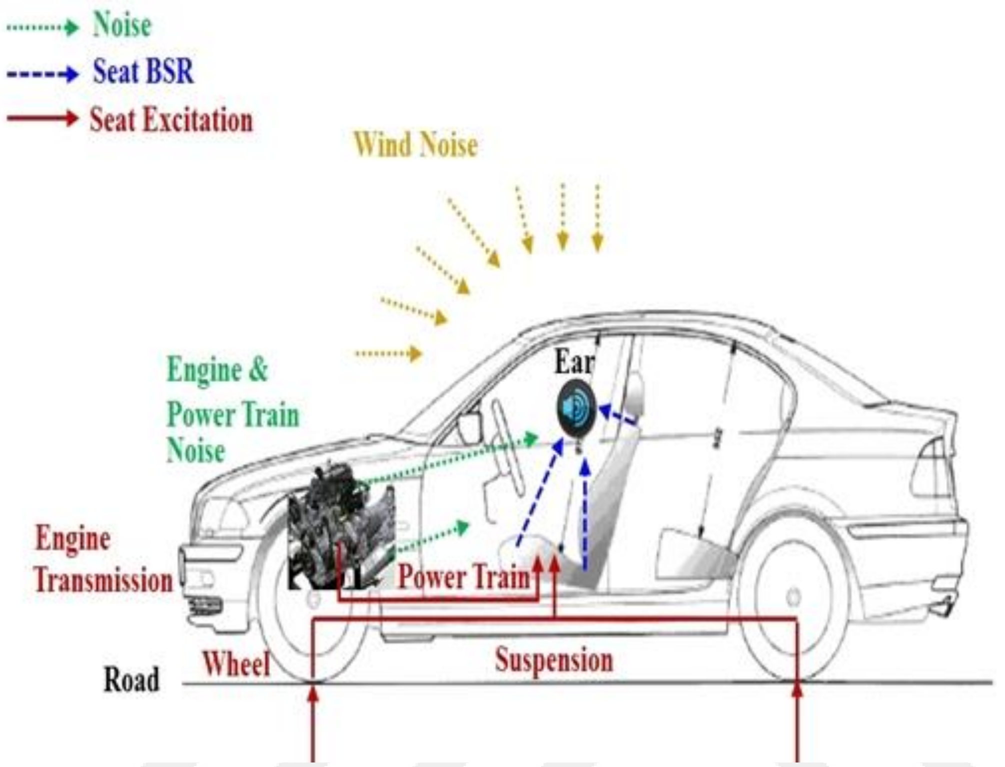

Figure 2.2: Transfer path of vehicle NVH (Park, 2012).

2.1.2 Microphone specification

Microphone is significant to simulate driver's perception to collect data from vehicle interior noise. Two microphones are placed at driver's seat as right and left ear. The microphones are free field type (Brüel & Kjær: ½-inch Prepolarized Free-field Microphone Type 4188) since these microphones are not disturbing the sound field with their own presence it is optimized at 0° incidence for flat free field response . In other words they are designed for compensating their presence in the sound field. (MicW, Technical Note 1401)

- Frequency: 8 Hz to 12500 Hz
- Dynamic Range: 15.8 to 146 dB
- Temperature: –30 to +125 °C (–22 to +257 °F)
- Polarization: Prepolarized

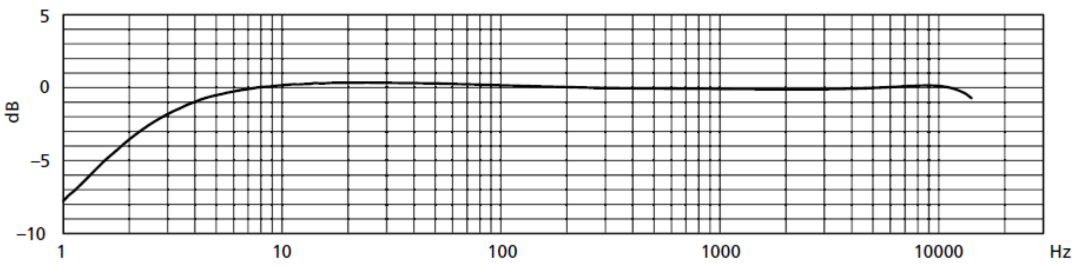

Figure 2.3: Transfer function of free field microphone type 4188 (Product Data Free-field Microphone Type 4188).

2.1.3 Accelerometer specification

The three directional (x,y,z) accelerometer is using to collect data from parts. Example of acceletometer is Kistler Type 8766A500BH that has been used at this field test. For vehicle field test it is suitable to mounted every part (mass is very low and different mouting accessories can be used) and sealing standart is high. Specification details are given at the below Figure 2.4.

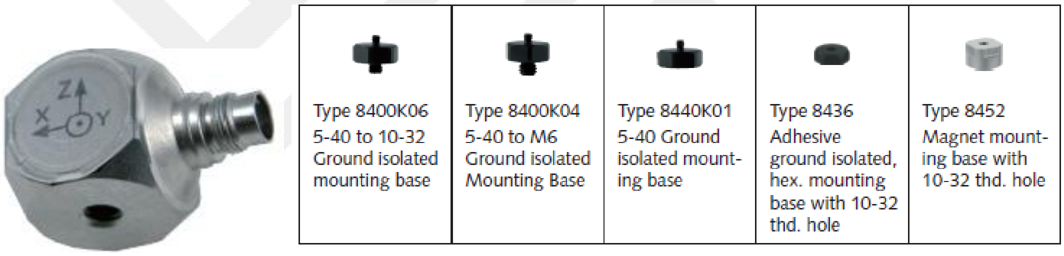

Figure 2.4: Three directional (x,y,z) accelerometer and mounting options (Kistler Miniature PiezoStar® Accelerometer Datasheet: Type8766A250... and 8766A500…).

- Acceleration Range: -+500 g
- Sensivity at 100 Hz: 10 mV/g
- Frequency response: 0,5…12000 Hz
- Sealing housing: IP68
- Mass: 4 gr
- Output full voltage scale: -+5 V

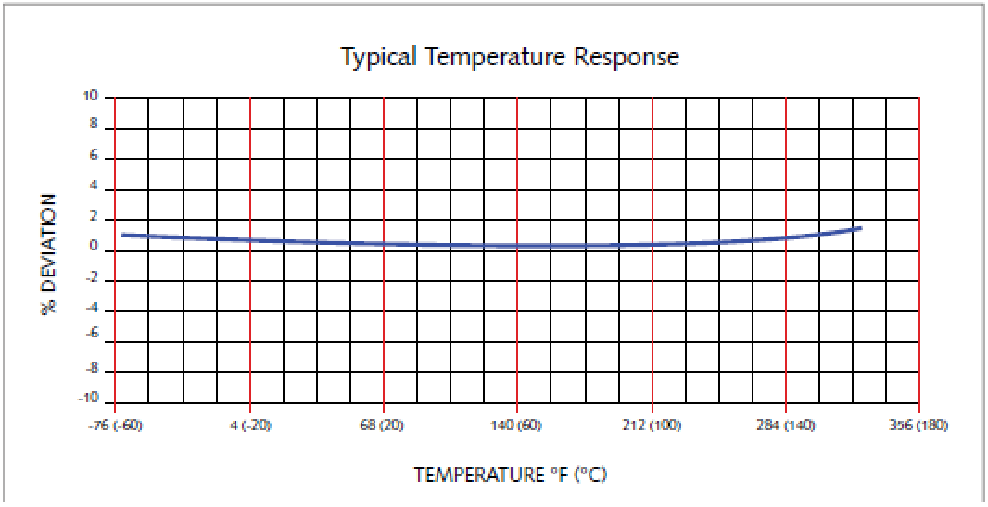

Figure 2.5: Sensivity variation with temperature (Kistler Miniature PiezoStar® Accelerometer Datasheet (Type8766A250... and 8766A500…).

2.2 Source Identification via Vehicle Data

Cables, engine mounts, vehicle body structures etc. is mainly transferring driveline noise through them to inside the vehicle cabin. Source of abnormal noise from the driveline can be different and the following factors have been checked in general.

- Engine Excitation due to firing orders (High fluctuation in the engine rpm)
- Transmission Gear Rattle Noise (High backlash in the rotational components)
- Improper damping at components (inside of transmission – low oil viscosity)
- Improper NVH modes at external control parts (High Resonance bands or high displacements at parts)

Firstly, objective NVH test data collected from vehicle especially at new transmission and external control application parts and while test data collection is continuing, vehicle evaluated with above points for abnormal noise factor. “Knocking noise happens especially between 1100 - 1500 rpm” reported subjectively. According to this statement vehicle data has been investigated to find abnormal noise via source identification from especialy new application parts. Data has been checked accelerometers located transmission, shift turret, shift lever arm, shift cable holder bracket, shifter box, shifter arm and seat track. That configuration can be reviewed at the Figure 2.6.

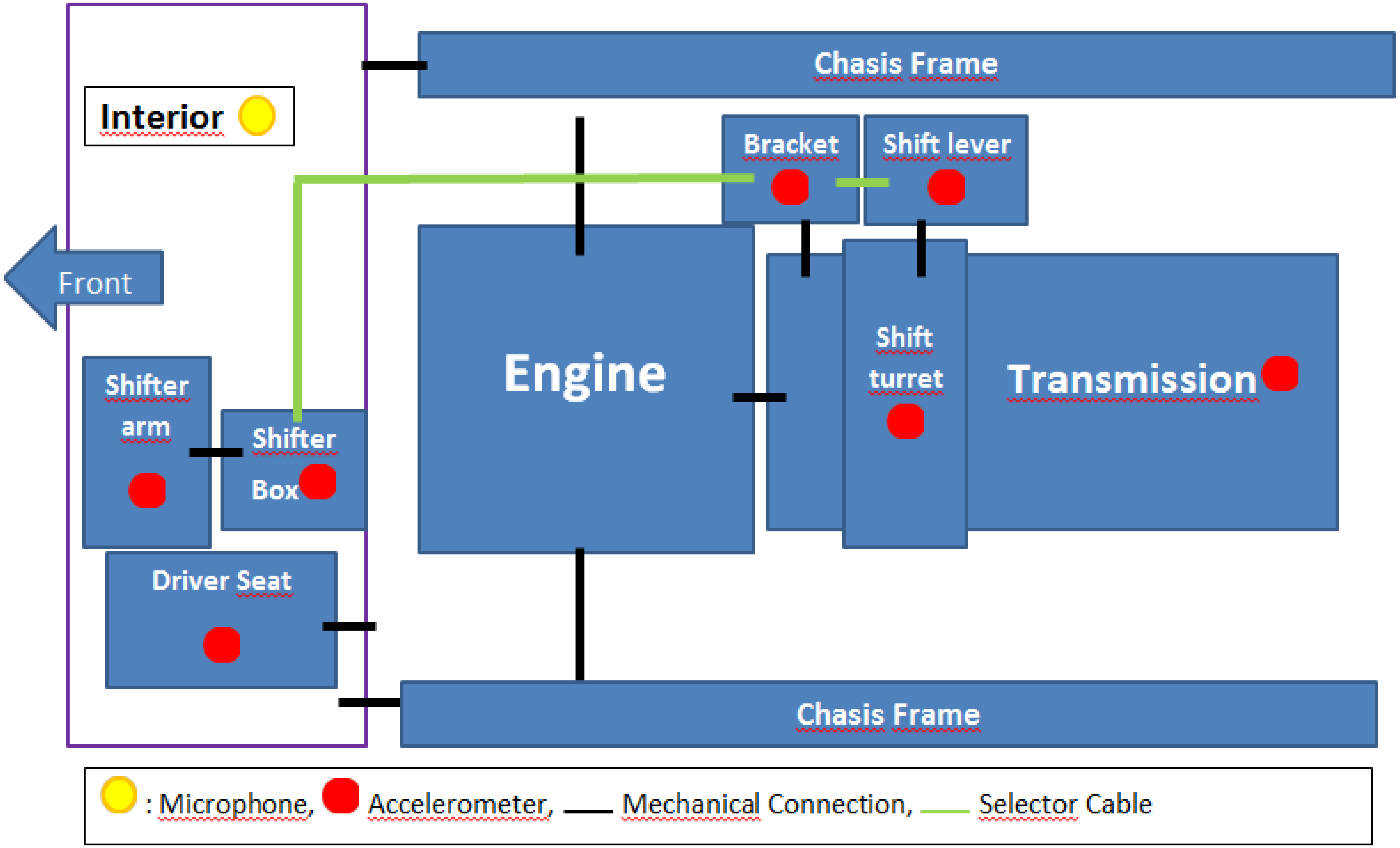

Figure 2.6: Schematic configuration view for accelerometer positions at vehicle.

For source identification of this abnormal noise firstly microphone data has been listening for reviewing which frequency bands that noise was reported. The filter application has been done to separate and find out especially which frequency band is including abnormal noise. Generally, the frequencies that would like to eliminate, preserve or emphasize can be done via filter selection. The filter classification is Lowpass (LP), highpass (HP), Bandpass (BP), Bandreject (BR), notch and resonator. It can be reviewed at the Figure 2.7.

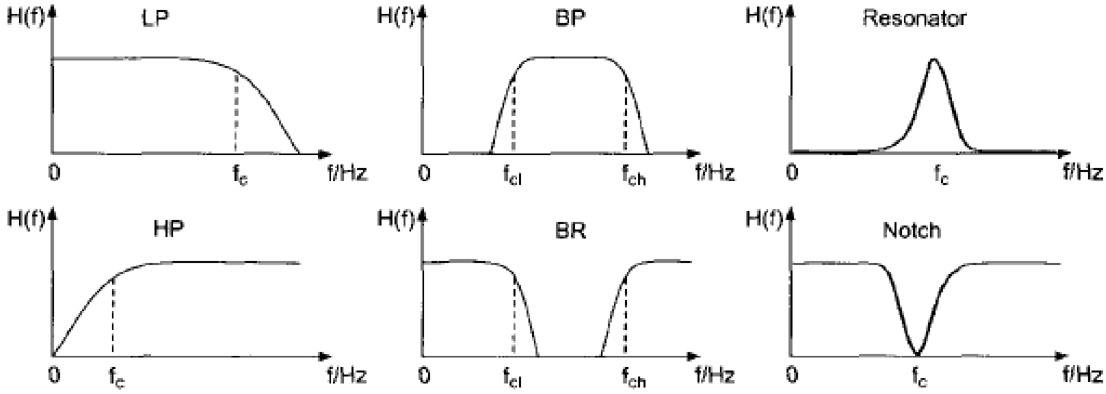

Figure 2.7: Filter classification (Dutilleux and Zölzer, 2002).

When the data reviewed and filtered (BP, BR and notch) around 750Hz wideband, abnormal noise heard from microphone at interior. According to this, noise trace has been checked from parts' accelerometer via colopmaps for source identification.

Objective data have been taken from transmission, shift turret and shift lever arm are not including 750Hz abnormal noise trace (Figure 2.8). (Shift lever arm has also different high interval wideband noise that is not transmitting inside of cabin through cables.)

Moreover, refer to colormaps at the Figure 2.9 shift cable holder bracket has got resonance frequency around checked frequency region and shifter box and shifter arm include 750 Hz wideband noise trace with different excitation form.

According to source of noise is shift cable holder bracket is filtered around 750 Hz and checked again. It is verified the main contributer is bracket and abnormal noise is transferred with path from shift cable though shifter box and shifter arm inside of cabin.

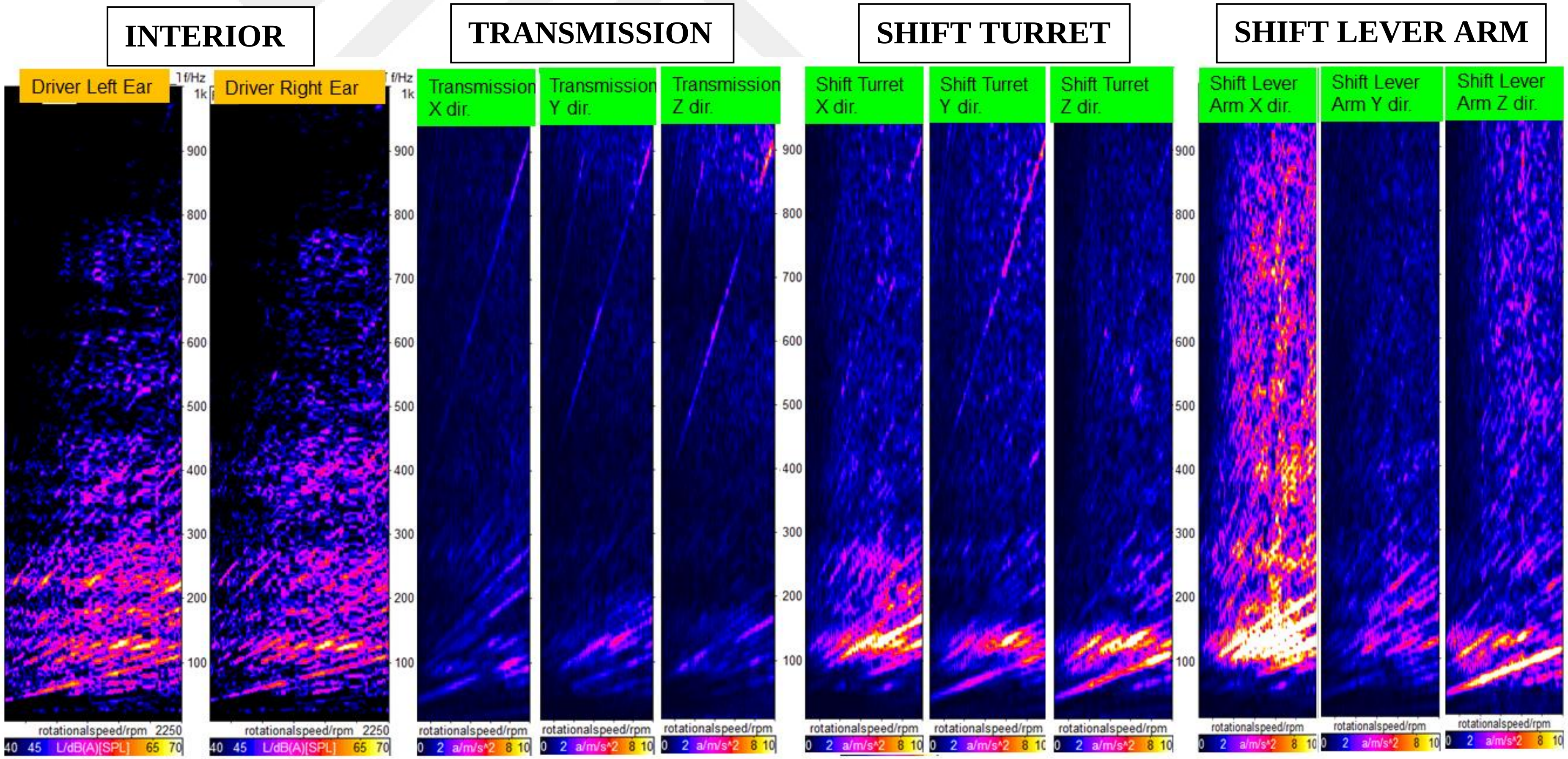

Figure 2.8: Vehicle data at inside of cabin, transmission, shift turret and shift lever arm.

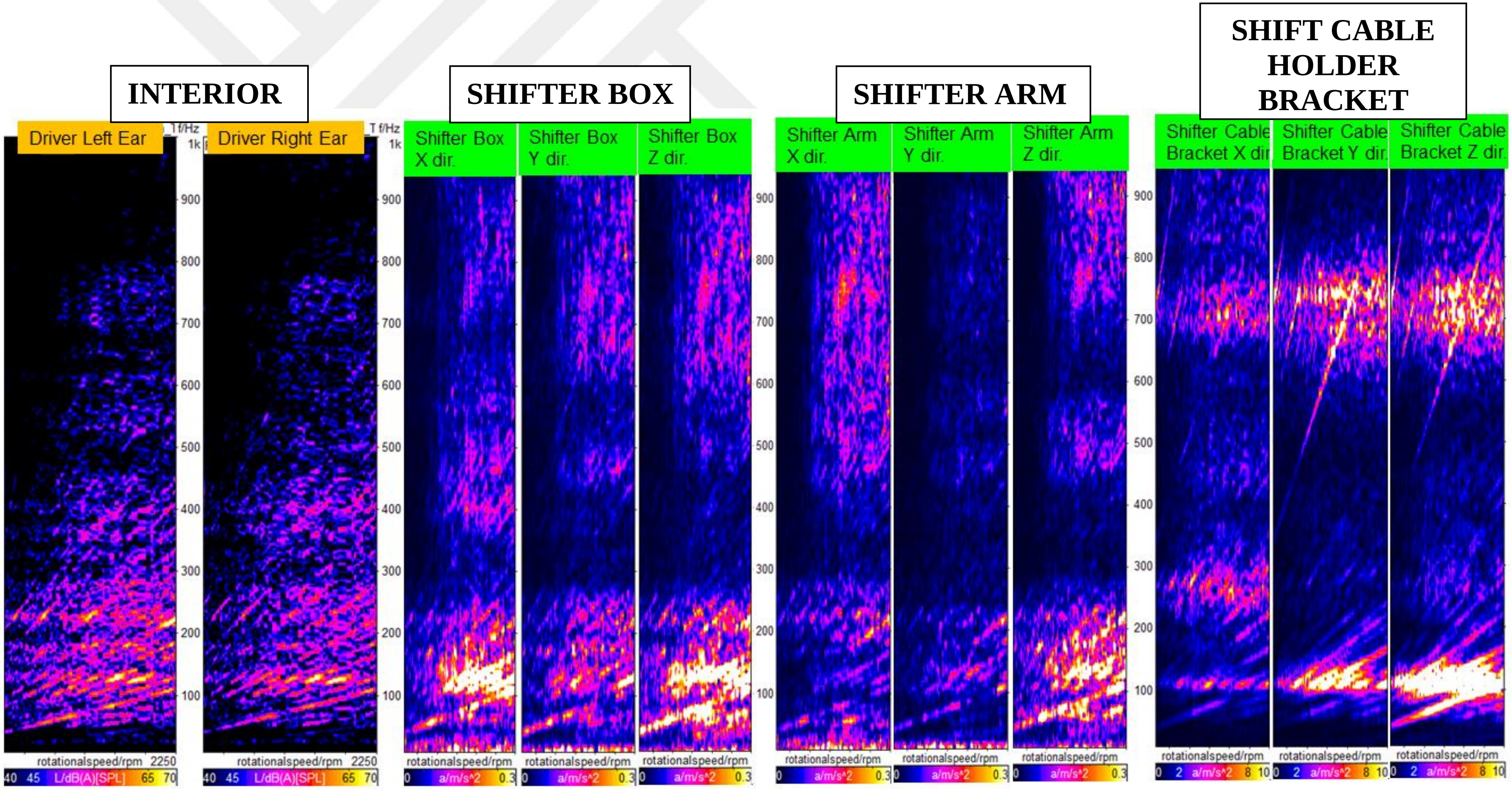

Figure 2.9: Vehicle data at inside of cabin, shifter box, shifter arm and shift cable holder bracket.

Consequently, driver’s claim is verified with vehicle data that is shown 750Hz wideband noise can be heard inside the cabin. Shift cable holder bracket includes the higher amplitude value 10m/s^2 at 750 Hz frequency band can be considered as the main reason. Vibration transfer path can be described as shifter cable bracket (source), shifter cable, shifter box and shifter arm. The 750 Hz is more audible at higher gears can be explained the higher the gear value means excitation derived from driveline also reaches to higher frequency values.

Note that the iterations are repeated again to collect data to verify bracket is root cause for abnormal noise. Since, at high gears (5-6-7-8) vehicle demanded higher torque from engine and transmission and due to test track length is not suitable to continue reach up to 8th gear and continue test until stop at the end for safety precaution 6th gear used.

2.3 Modal Analysis of Baseline Bracket Design

Simulation and design study will be performed to identify areas of weakness of shift holder bracket with development of modal model. Since, the enlightenment and visualization of mode shapes is significant to identification, structural dynamic modification (SDM) methodology is followed. Schematic of the SDM process is shown at the Figure 2.10.

To determine the part’s behavior and effects of changes in the part characteristics modal model that can be reviewed part’s rezonance frequency, mode shapes etc. will be developed with support of experimental modal analysis and finite element modeling (FEM) according to SDM process. Developed model will be correlated with results of experimental modal analysis and FEM, thus model will be reliable to apply structural change proposals and see the affects of these changes on part characteristic. If the evaluation of change is suitable as requested, (in this study, it is requested to no noise transferred to inside of cabin according to part mode shapes and displacement values will be decided.) structural change is approved to apply and use at the vehicle. Until the desired characteristic is obtained, the model will be revised with different stuctural change proposals.

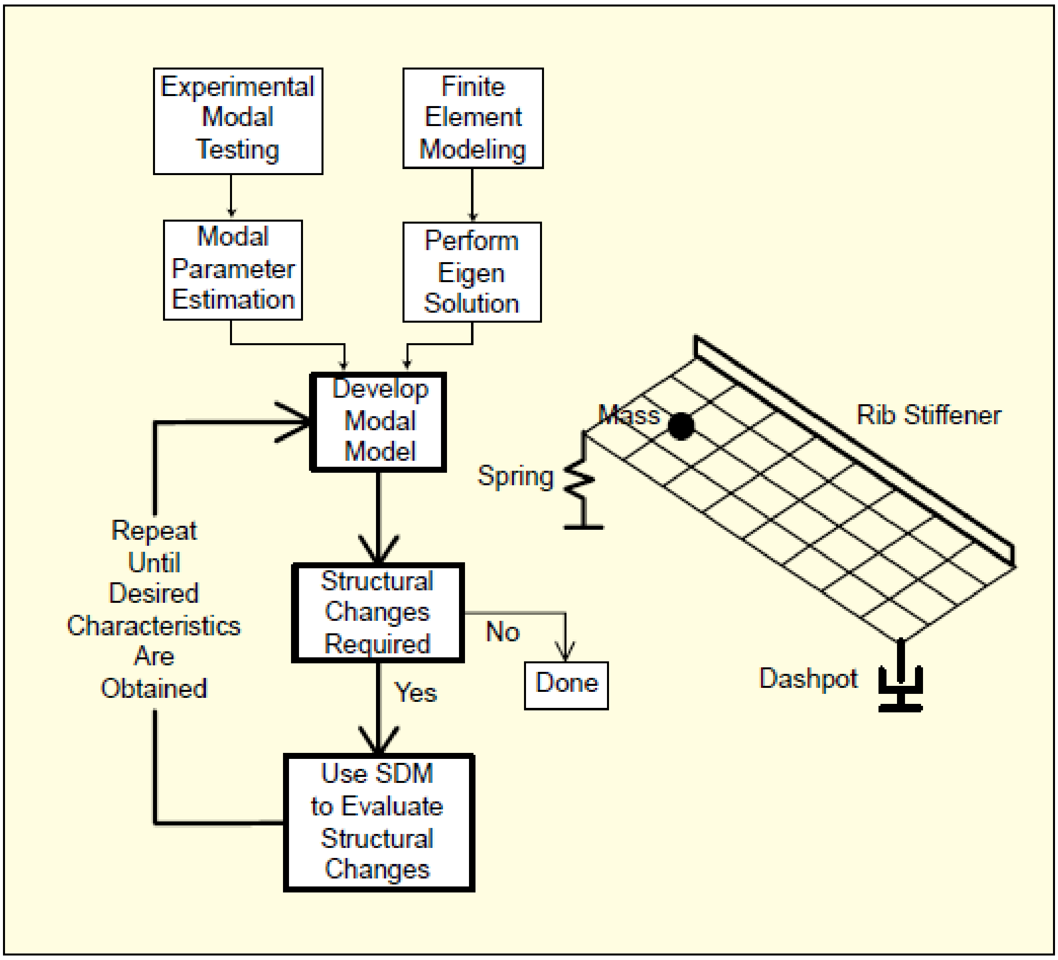

Figure 2.10: Schematic of the SDM process (Avitabile, 2001).

2.3.1 Experimental modal analysis of baseline bracket design

Experimental modal analysis is performed with hammer excitation test at ITU Automotive Lab. Endevco model 2302 modal hammer and three directional (x,y,z) accelerometer (Kistler Type 8766A500BH specification is already given at 2.1.3.) have been used at test.

Hammer specification is;

- Reference sensivity: 2.25 mV/N
- Range: 2000 N
- Maximum impulse: 4448 N
- Frequency range (max): 8000 Hz
- Full scale output: - + 5 V (Modal Hammer Model 23-02 Endevco Datasheet)

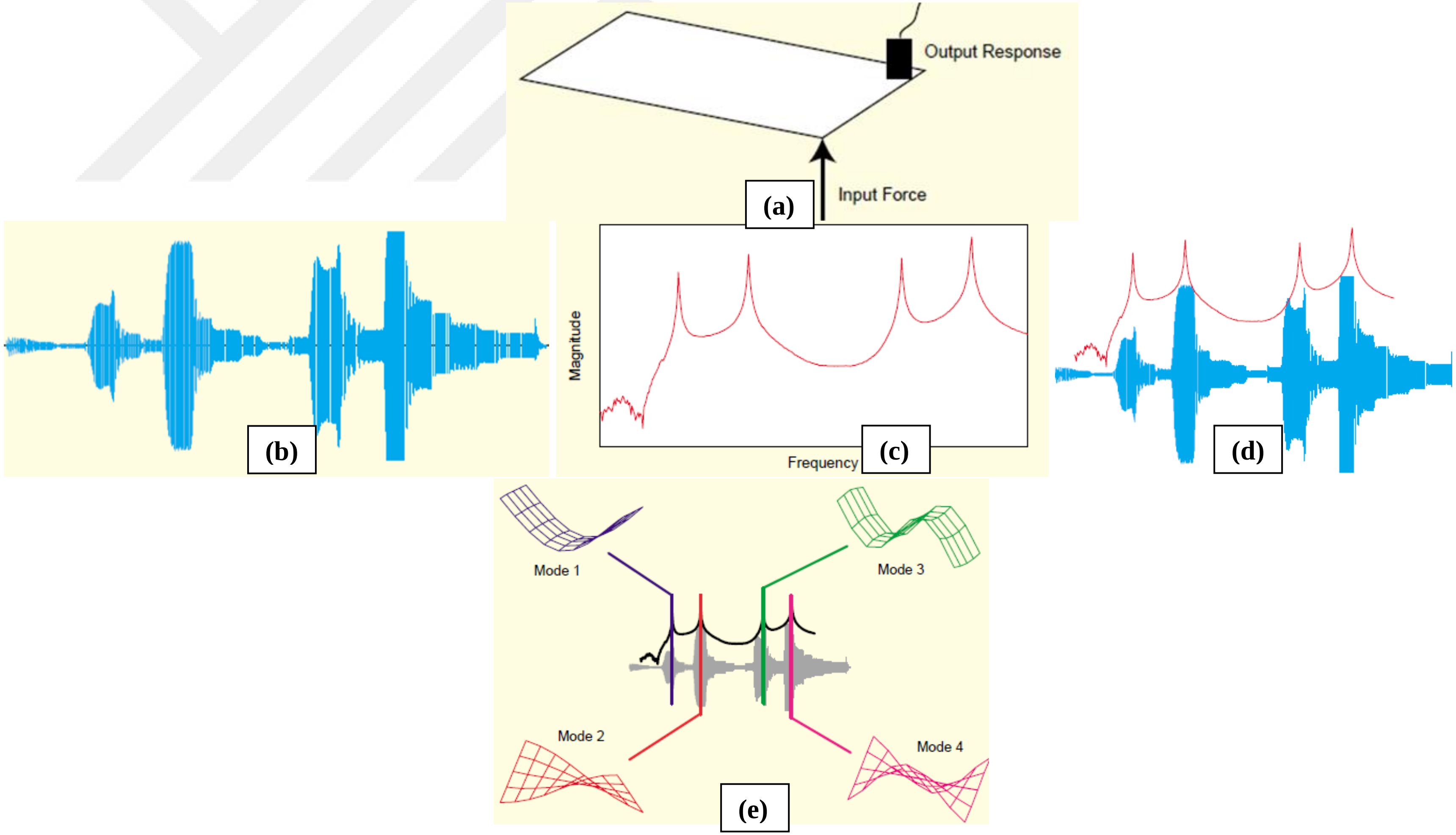

Figure 2.11: Methodology of experimental modal analysis with example of simple plate excitation and response model (a) Simple please excitation/response model (b) Simple plate response (c) Simple plate frequency response function (d) Overlay of time and frequency function (e) Simple plate sine dwell responses (Avitabile, 2001).

Figure 2.12: Endevco model 2302 modal hammer (Modal Hammer Model 23-02 Endevco Datasheet).

Experimental modal analysis methodology has been described with graph figures for simple plate example at the Figure 2.11. According to this methodology hammer test has been performed for shift cable holder bracket and details of this test is given at the below. In addition, hammer figure is given at the 2.12. and test setup is given at the Figure 2.13.

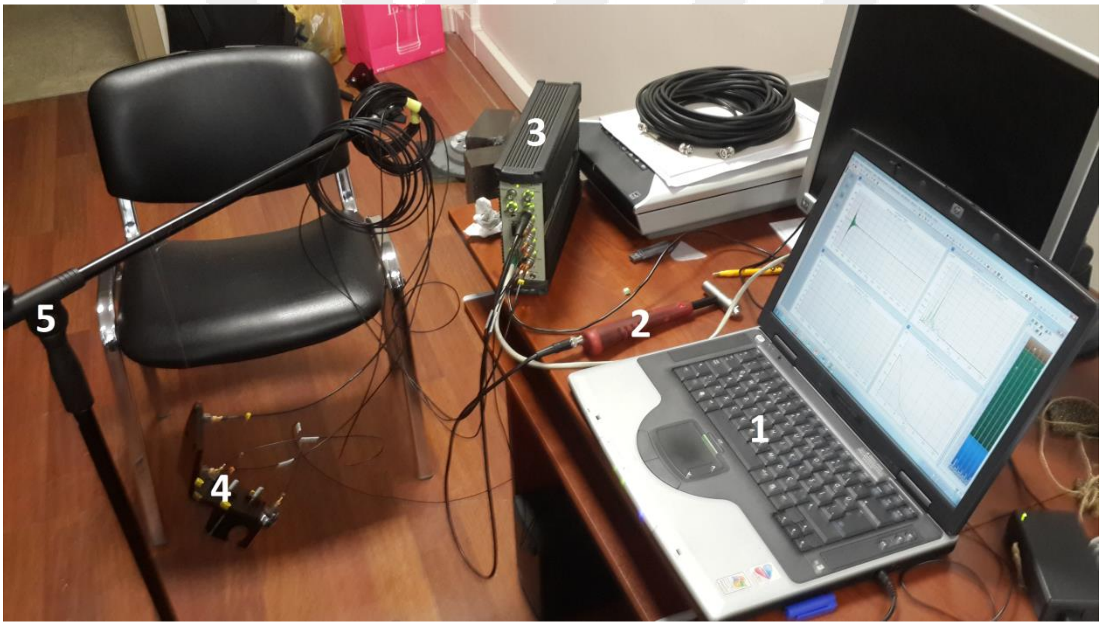

Figure 2.13: Experimental modal analysis configuration (1: PC, 2: Impact Hammer, 3: Analyzer, 4: Part with Accelerometer, 5: Fixation Aperture).

22 measurements have been carried out from different accelerometer positions. While accelerometer mounting position is changed hammer is applied at single reference point for each measurement. Each measurement repeated 3 times to ensure sensible data collection.

Hammer test is applied to two brackets in order to verify the experimental modal analysis and see production order differences and also for correlation between tests.

Except to manufacturing order everything is completely same according to design point of view and production method for brackets.

To assess data quality for how much of the output signal is associated to the input signal, coherence graphs are collected with transfer functions. Collected data for every measurement is processed at MATLAB to review all mode frequencies and mode shapes. To sketch exact mode shapes, coordinate system determined according to first accelerometer as reference point and other acceleration locations have been provided exactly at the Computer Aided Design (CAD) software: CATIA (Figure 2.14 and Table 2.1). In addition, between accelerometers 12 and 18 rotation from X axis has to be defined in MATLAB since bracket angle has been changed for these locations. Due to accelerometers' axis has been changed at cable connection part, accelerometers' axis carefully corrected for acc19, acc20, acc21 and acc22.

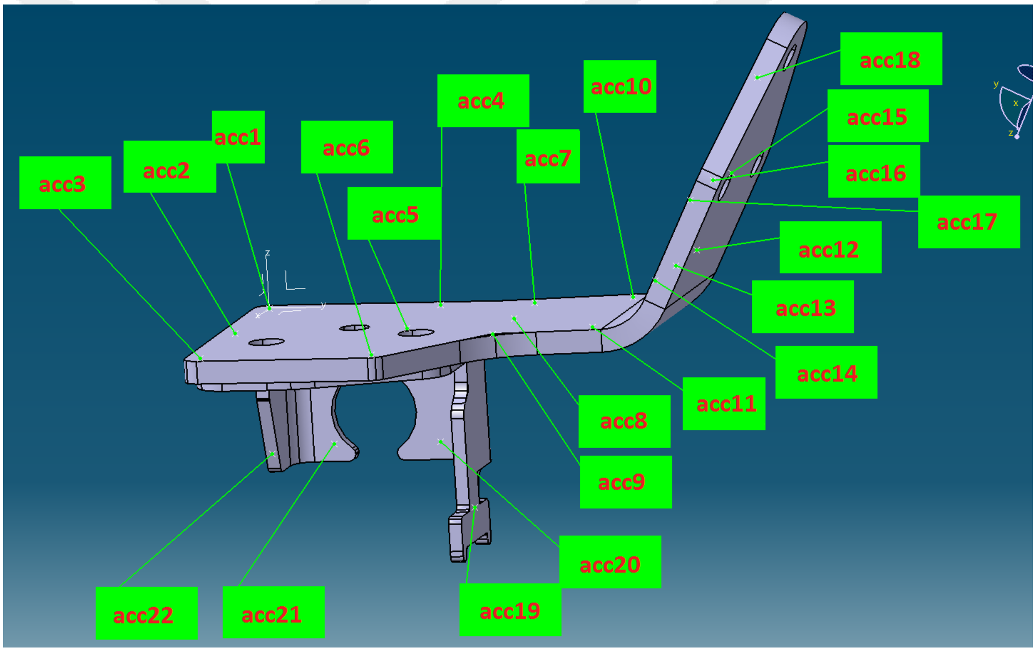

Figure 2.14: CATIA (CAD) measurement view for accelerometer positions.

Firstly, frequency response function (FRF) is plotted that include every 22 measurements at one graph for each axis. Due to parts' structure and geometric shape all FRFs are not exactly overlapped. Graphs can be reviewed linear or logarithmic in MATLAB. While checking linear graphs displacement difference can be observed easily that z axis has higher displacement at each mode frequency. At logarithmic graphs characteristic of each reference points can be illustrated better than linear graphs and mode shape frequency have been found out frequency intervals have been noted at these graphs. Table 2.2 included comparison of intervals that can include mode shape frequencies according to axis and parts. When the compare resonance frequencies are almost same according to axis in same part (only displacement amplitude is different is reviewed at linear graphs). However, parts' mode frequencies are slightly different than each other. Secondly, due to mode frequencies are different from brackets and comparasion of FEM will be done, every mode shapes are also plotted in MATLAB. Why the resonance frequencies are different from each other will be investigated at next sections.

Table 2.1: Accelerometer coordinates from reference acc1.

Accelerometer Position	Coordinates		
	x (mm)	y (mm)	z (mm)
acc1 (Reference)	0	0	0
acc2	42,72	0,01	0
acc3	85,44	0,02	0
acc4	0	53,42	0
acc5	40,72	53,42	0
acc6	85,44	53,43	0
acc7	0	83	0
acc8	26,21	83	0
acc9	52,42	83	0
acc10	-7	112	0
acc11	43,7	112	0
acc12	-10,97	131,65	13,26
acc13	14,05	131,53	13,03
acc14	40,1	131,61	13,2
acc15	-15,7	142,73	35,99
acc16	10,47	143,9	38,4
acc17	37,44	143,31	37,17
acc18	7,9	158,94	69,22
acc19	39,28	70,51	-55,45
acc20	7,04	52,64	-40,75
acc21	7,05	19,41	-40,78
acc22	23	4	-40,77

Table 2.2: Comparison of mode frequencies between axis and sample parts.

Modes	Bracket Sample 1 (Hz)			Bracket Sample 2 (Hz)		
	x	y	z	x	y	z
1	368 - 384	366 - 384	366 - 384	558 - 572	560 - 570	558 - 570
2	524 - 542	522 - 542	524 - 542	666 - 690	666 - 694	670 - 688
3	726 - 760	726 - 766	726 - 766	844 - 870	842 - 870	844 - 860
4	1026-1082	1080	1046 - 1084	1112 - 1138	1116 - 1140	1098 - 1138

Frequency response function (FRF) graphs are given at the below figures in terms of Amplitude versus Frequency for both linearly and logoritmicly.

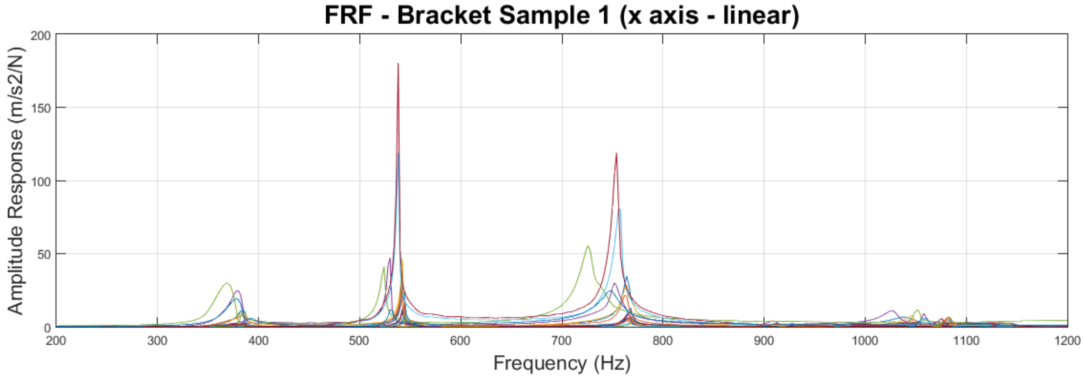

Figure 2.15: Frequency response function graph for 22 measurement points at bracket sample 1 (linear graph – x axis).

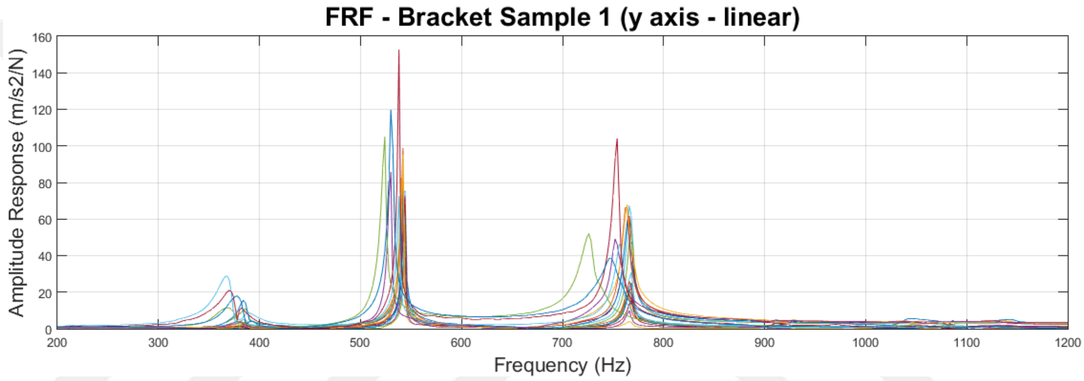

Figure 2.16: Frequency response function graph for 22 measurement points at bracket sample 1 (linear graph – y axis).

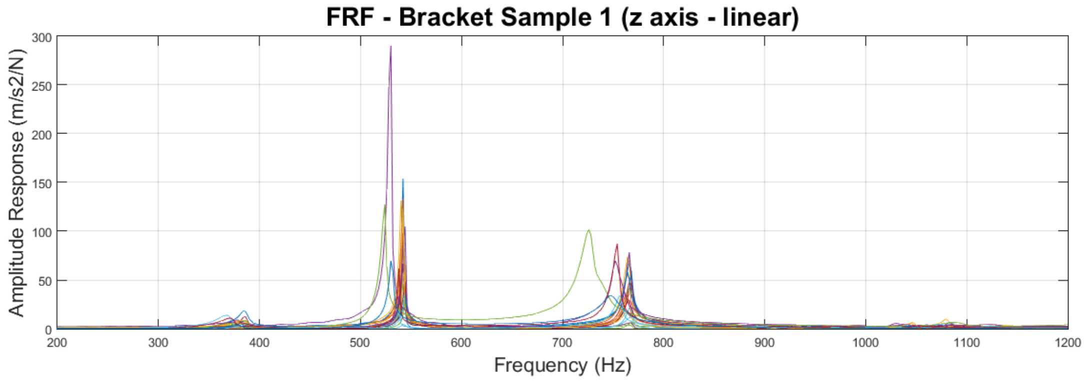

Figure 2.17: Frequency response function graph for 22 measurement points at bracket sample 1 (linear graph – z axis).

Sound intensity level, changing of power level etc. can be shown in terms of bel or desibel (dB) in logaritmic graphs. To better understanding desibel is using in frequency response graphs at logaritmic scale. It is defined according to definition of bel and desibel in MATLAB.

$$Bel = \log(\frac{P}{P_{ref}}) \tag{2.1}$$

$$dB = 10\log(\frac{P}{P_{reference}}) \tag{2.2}$$

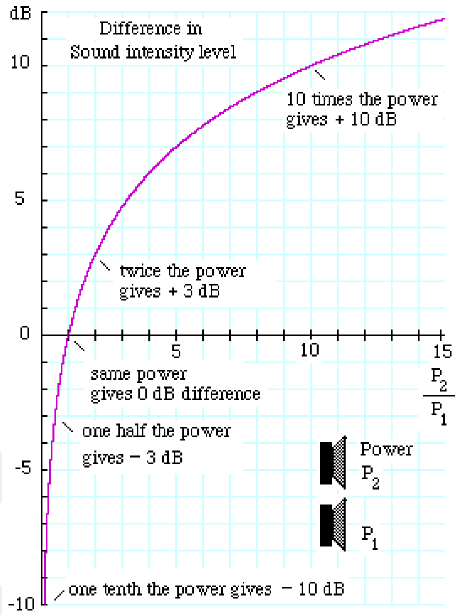

Figure 2.18: Graph for difference in sound intensity level according to dB (UNSW).

However, power is proportional to sound pressure level, acceleration as square of them equation will be defined as,

$$dB = 10\log\left(\frac{a^2}{a_{reference}{}^2}\right) = 10log\left(\frac{a}{a_{reference}}\right)^2 = 20\log(\frac{a}{a_{reference}}) \tag{2.3}$$

In addition, according to equation 2.3 logaritmic graps are plotted as “Amplitude in dB reference 1g” at y axis versus frequency. For example, 1g corresponds to 0dB and 10g corresponds 20dB at the FRFs of experimental modal analysis.

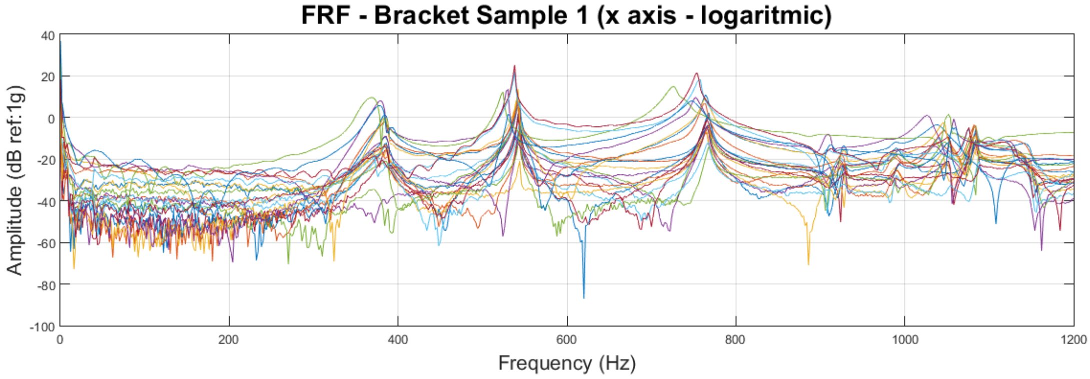

Figure 2.19: Frequency response function graph for 22 measurement points at bracket sample 1 (logarithmic graph – x axis).

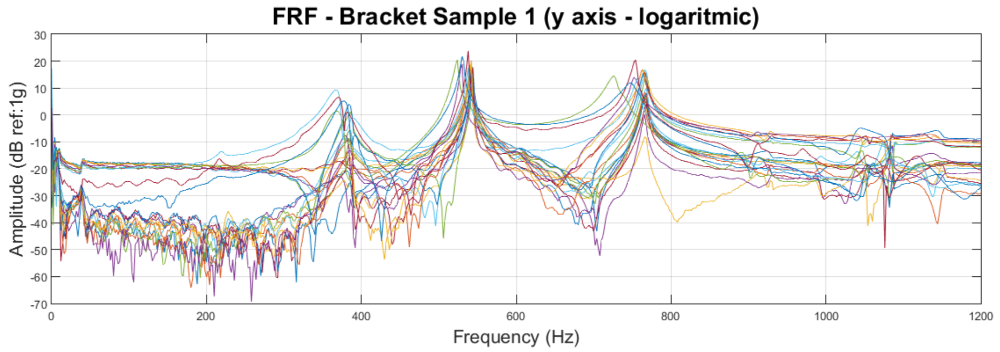

Figure 2.20: Frequency response function graph for 22 measurement points at bracket sample 1 (logarithmic graph – y axis).

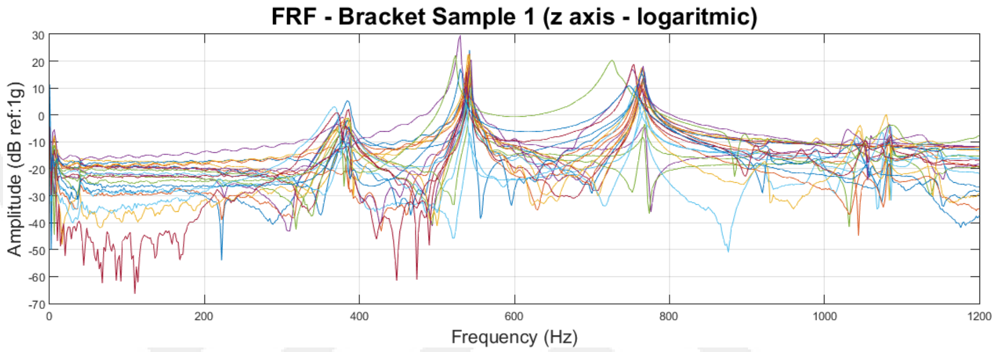

Figure 2.21: Frequency response function graph for 22 measurement points at bracket sample 1 (logarithmic graph – z axis).

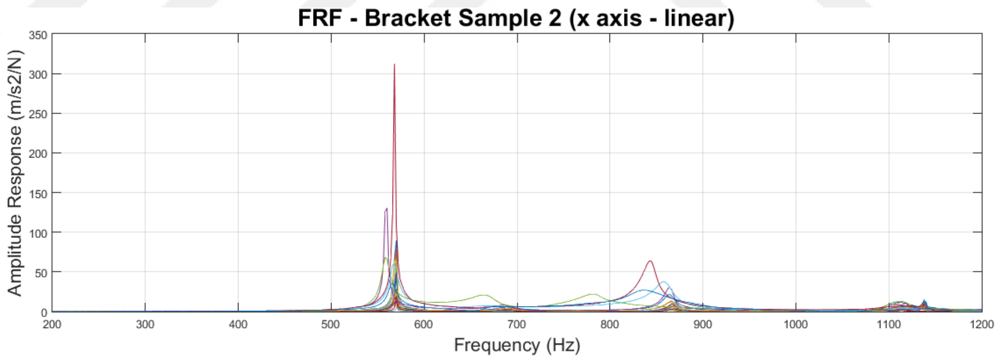

Figure 2.22: Frequency response function graph for 22 measurement points at bracket sample 2 (linear graph – x axis).

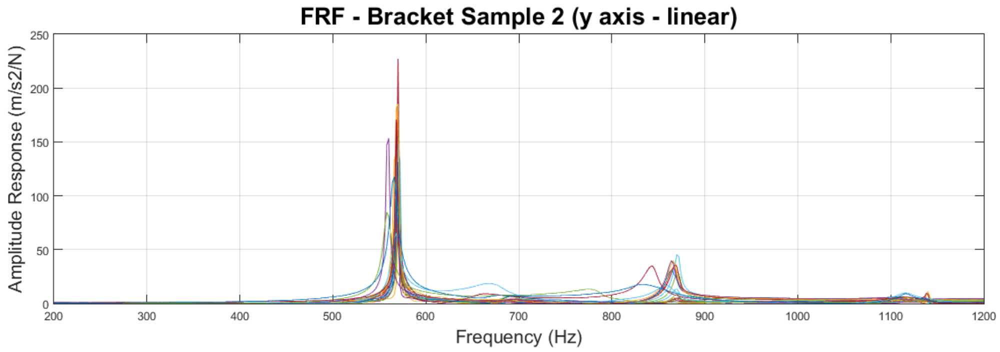

Figure 2.23: Frequency response function graph for 22 measurement points at bracket sample 2 (linear graph – y axis).

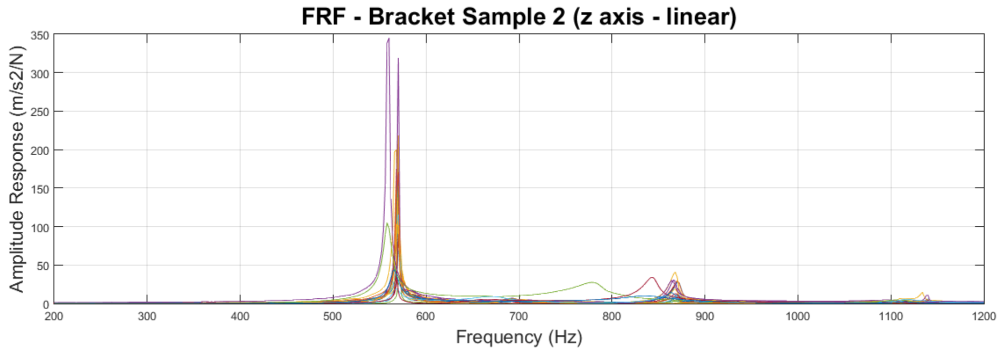

Figure 2.24: Frequency response function graph for 22 measurement points at bracket sample 2 (linear graph – z axis).

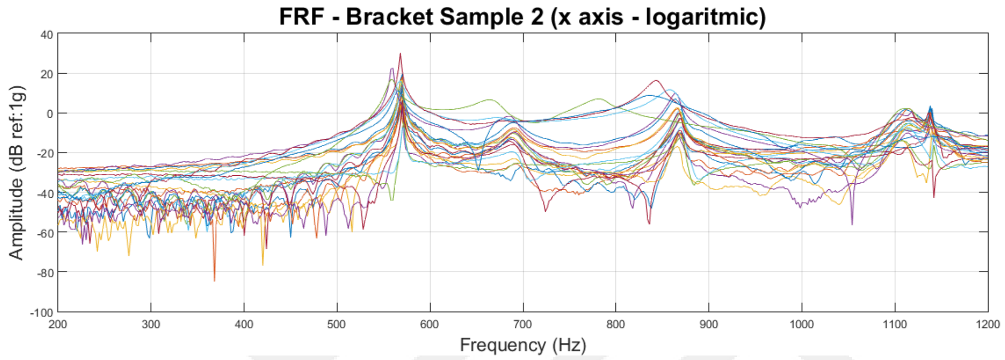

Figure 2.25: Frequency response function graph for 22 measurement points at bracket sample 2 (logarithmic graph – x axis).

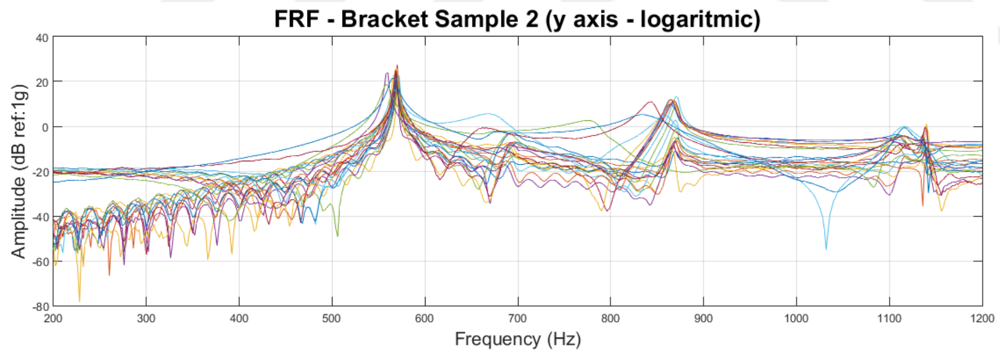

Figure 2.26: Frequency response function graph for 22 measurement points at bracket sample 2 (logarithmic graph – y axis).

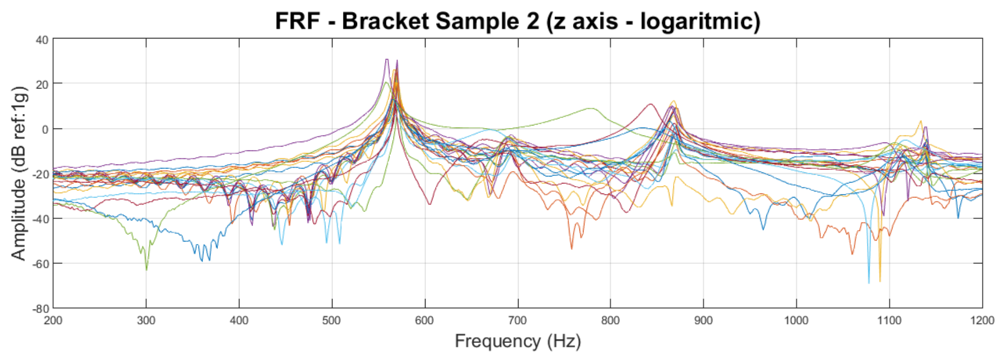

Figure 2.27: Frequency response function graph for 22 measurement points at bracket sample 2 (logarithmic graph – z axis).

To compare mode shapes of experimental modal analysis together with finite element analysis (FEA), mode shape figures will be given at section 2.3.2. According to table 2.2 bracket samples mode frequencies are different however some of the near frequency modes are very similar. For example, bracket sample 1: 540Hz vs bracket sample 2: 580Hz and bracket sample 1: 760Hz vs bracket sample 2 866Hz is looked alike each other. Furthermore, why frequency is different than each other is not completely understood with this information. Experimental modal parts are checked physicly and found out that tigtening torques are slighly different between slot connections and weld filler metal can be a little different. This can change stiffness of connection and parts' all stiffness value.

2.3.2 Finite element modal analysis of baseline bracket design

CAD model (step file) of shift cable holder bracket imported to Altair Hypermesh to prepare data for FEM. Baseline bracket included three body parts that base part (connection to transmission with 3 holes), slot part (connected to base part with two slot holes with apex bolts) and u part (connection for shift cable with clips) are separately meshed as components. Mesh element size is 3.000 and mesh type is always same that is quads. Parts connections are defined with rigidity element types are KINCOUP to satisfy bolt connection at base and slot parts, and weld connection between slot and u parts. Node to node connection assigned between components. After preparation of mesh data material is defined as STEEL and property assigned to parts, load step and output block created to solve eigenvalues as found out mode frequencies between 10 and 5000Hz. Figure 2.28 shows Hypermesh model tree and data view after definition of above parameters.

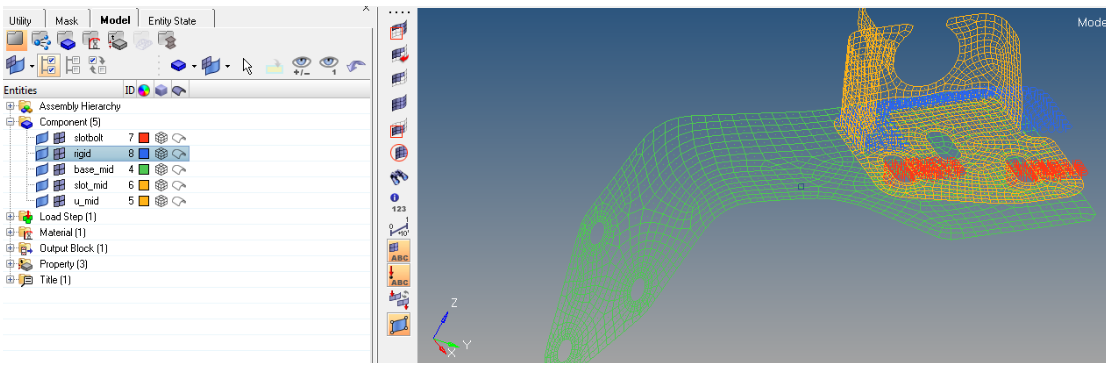

Figure 2.28: Hypermesh view for baseline bracket design.

Pre-process is prepared at the Hypermesh, data is exported to Abaqus with inp format to solve eigenvalue problem. Abaqus solution is reviewed at the Altair Hyperview to obtain mode frequencies and mode shapes at these frequencies. Table 2.3 is shown first four mode freqencies to review and compare with experimental modal analysis. In addition, hyperview figures are shown mode shapes with colour contour for displacement and deformation shape scaled 20 times to review motion easily. In addition, as it mentioned mode shapes from experimental modal analysis that is provided from MATLAB is added to compare with FEM.

Table 2.3: Baseline bracket design (FEM) mode frequencies.

Modes	FEM of Baseline Bracket Design (Hz)
1	587,67
2	652,67
3	867,27
4	1122,5

Blue lines plotted for first position before excitation happened and red lines plotted for mode shapes at defined mode frequencies at MATLAB graphs.

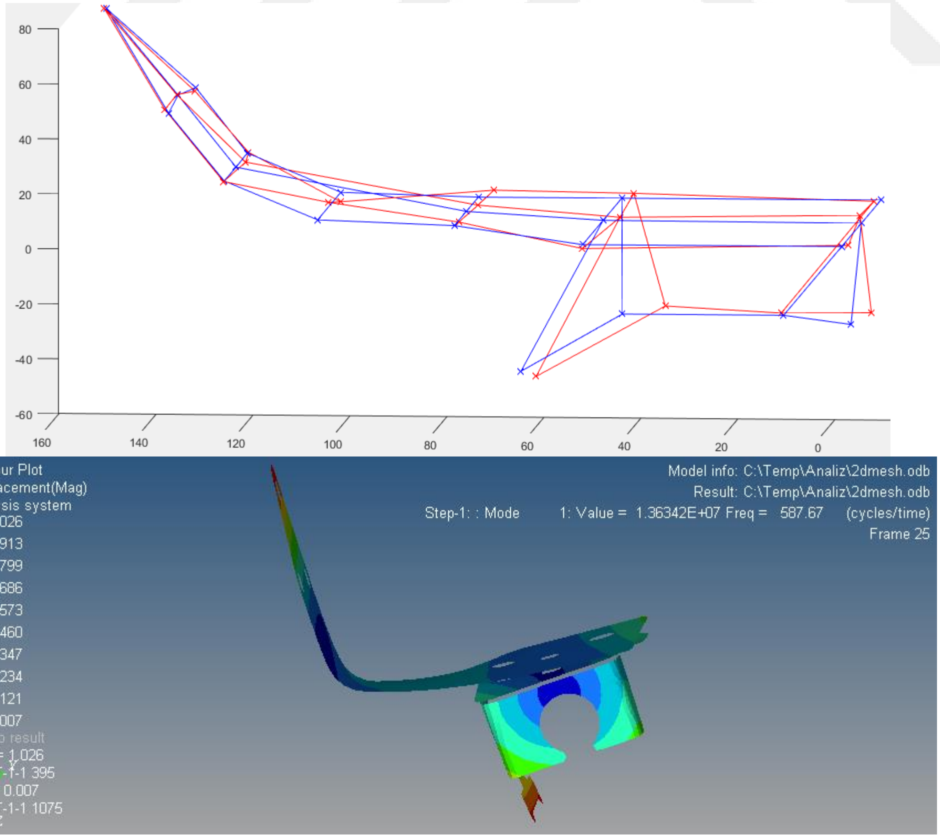

Figure 2.29: Comparison of experimental modal of bracket sample 2 (580Hz) and FEM for baseline bracket design (587,67Hz).

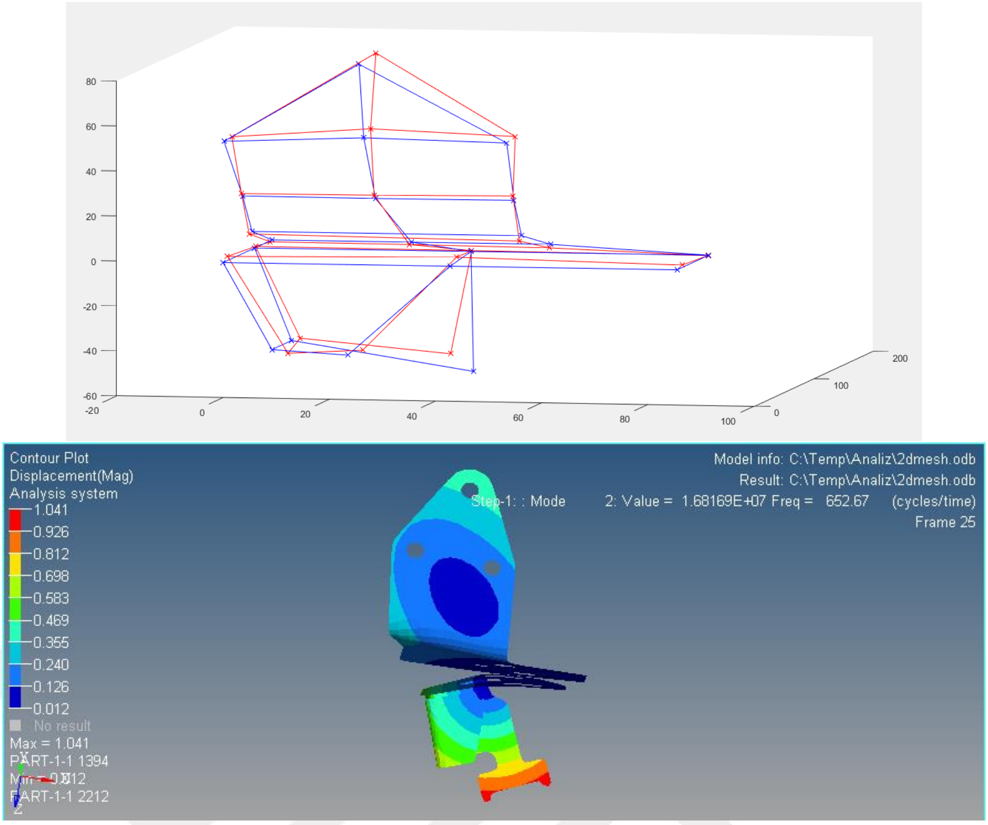

Figure 2.30: Comparison of experimental modal of bracket sample 2 (690Hz) and FEM for baseline bracket design (652,67Hz).

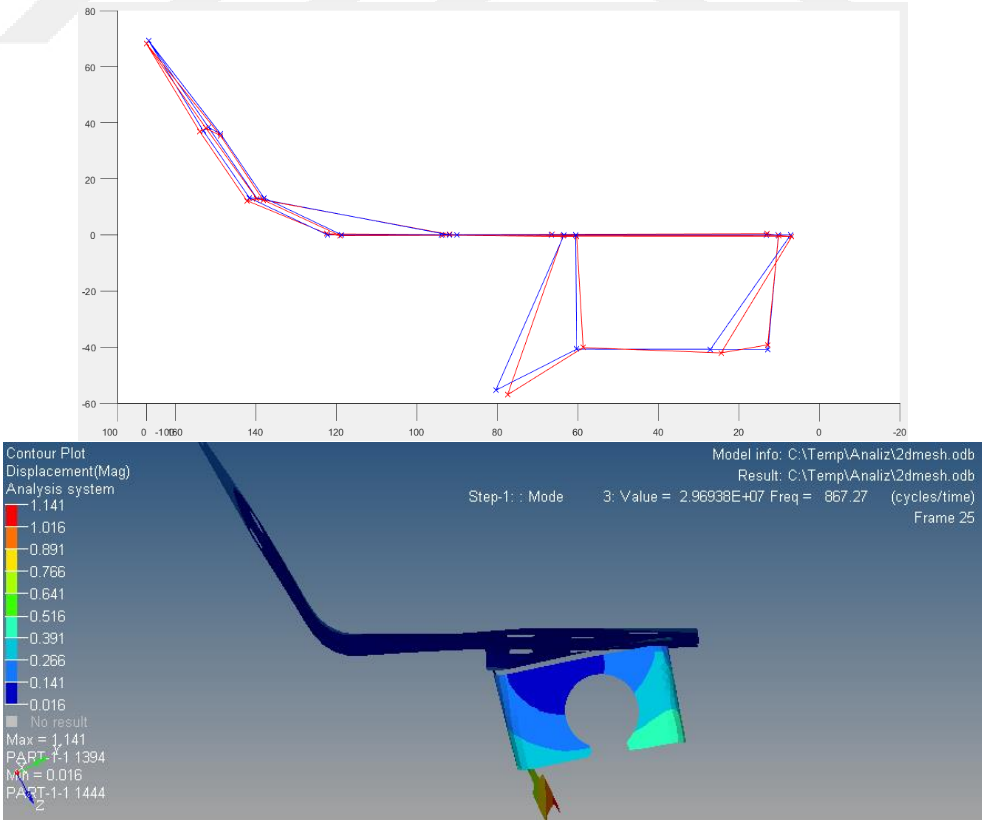

Figure 2.31: Comparison of experimental modal of bracket sample 2 (866Hz) and FEM for baseline bracket design (867,27Hz).

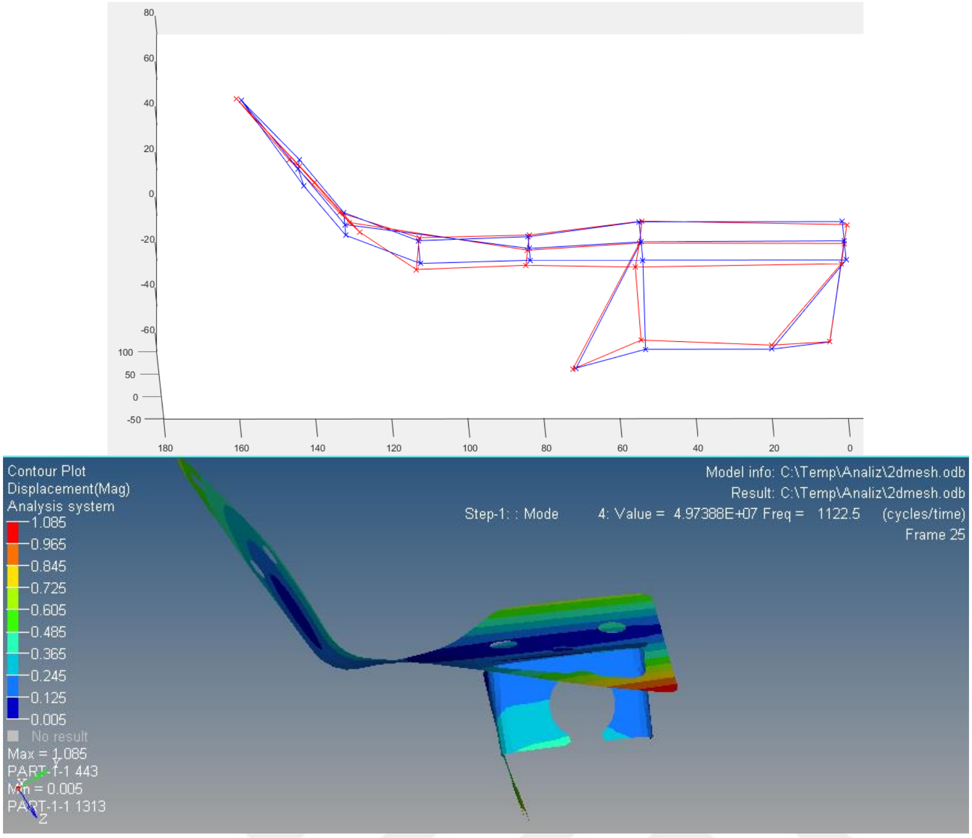

Figure 2.32: Comparison of experimental modal of bracket sample 2 (1138Hz) and FEM for baseline bracket design (1122,5Hz).

Comparison results will be discussed at the 4.1 section.

To simulate hypothesis, mesh data has been iterated with missed connection that is not included every nodes are connected under bolt head as rigid with KINCOUP. Left slot hole location is connected one side as node to node and other side is free, right slot hole location is connected two sides is only with node to node. (First iteration has been perfomed with six nodes to nodes connection at two sides as under the bolt head nodes connected with each other.)

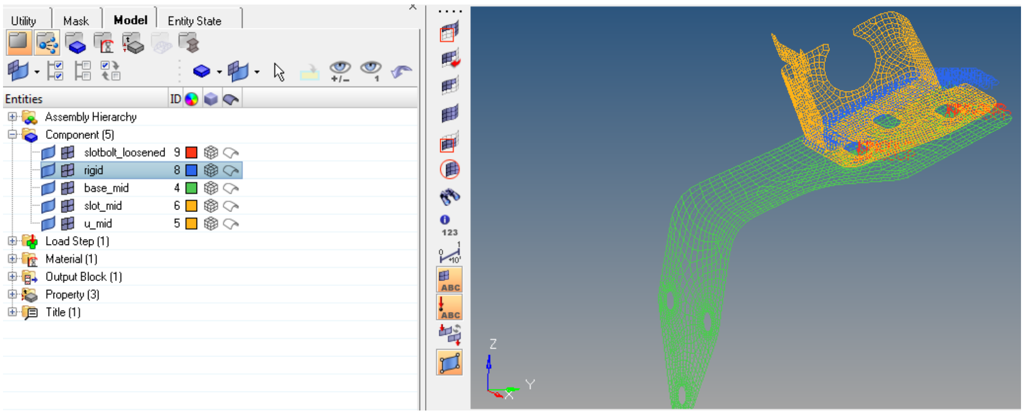

Figure 2.33: Connection definitions at the Hypermesh (slotbolt_loosened and rigidweld).

Table 2.4: Hypothesis bracket (FEM) mode frequencies.

Modes	FEM of Hypothesis Bracket (Hz)
1	239,56
2	514,09
3	651,74
4	869,09
5	1013,9
6	1142,1

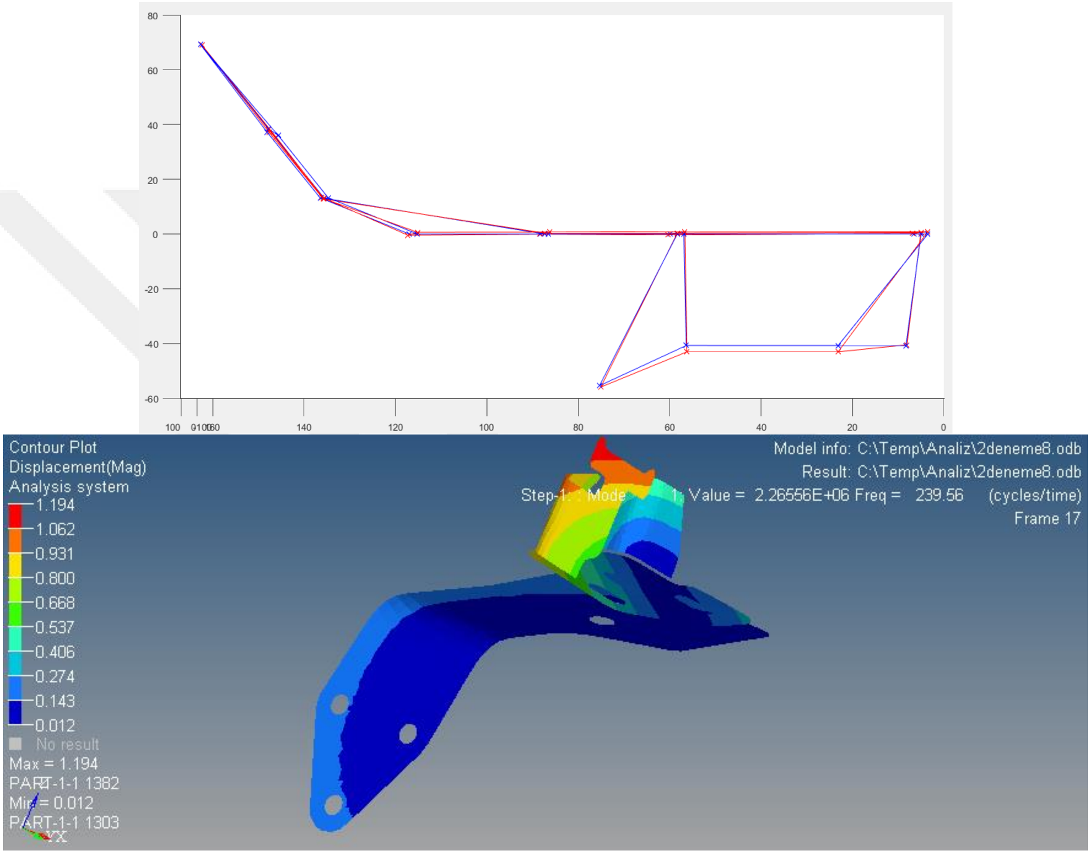

Figure 2.34: Comparison of experimental modal of bracket sample 1 (380Hz) and FEM for hypothesis bracket design (239,56Hz).

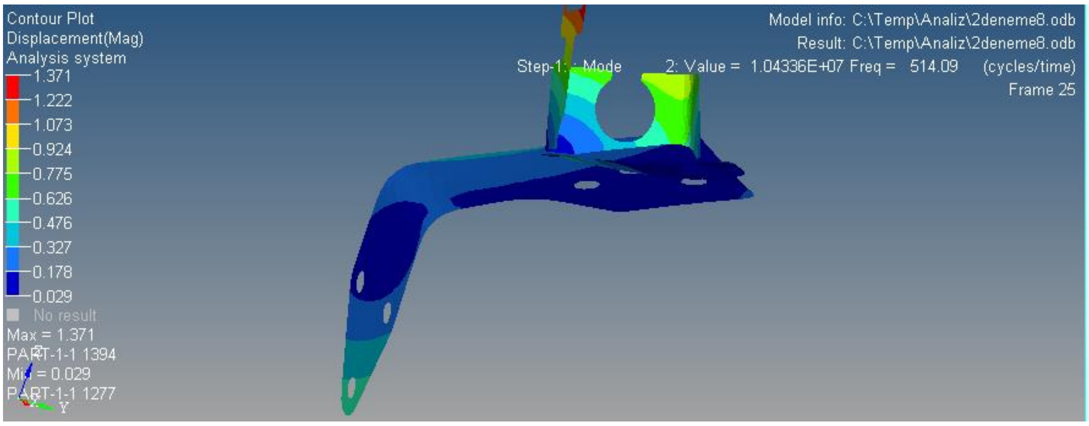

Figure 2.35: FEM for hypothesis bracket design (514,09Hz).

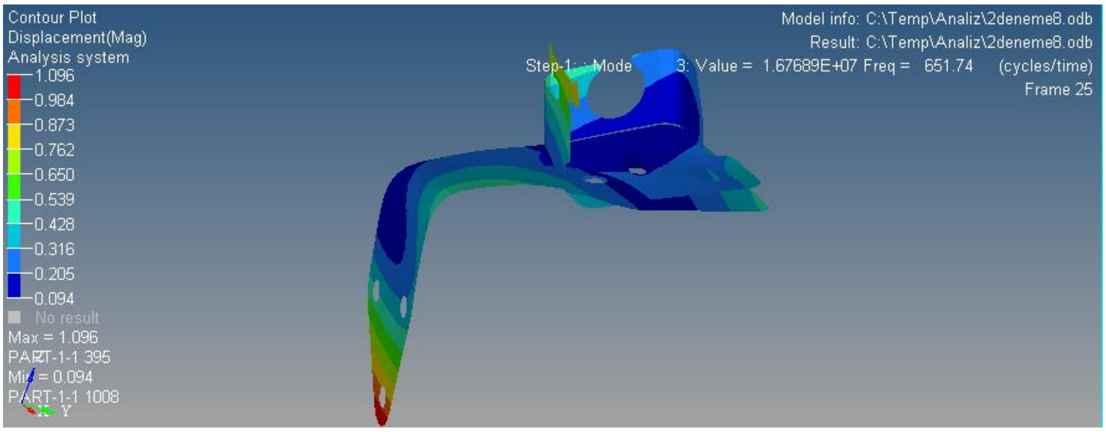

Figure 2.36: FEM for hypothesis bracket design (651,74Hz).

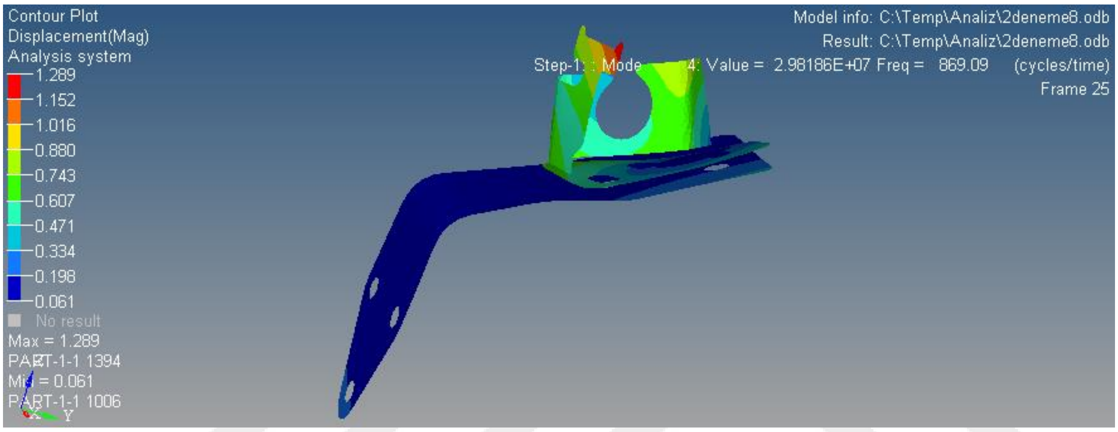

Figure 2.37: FEM for hypothesis bracket design (869,09Hz).

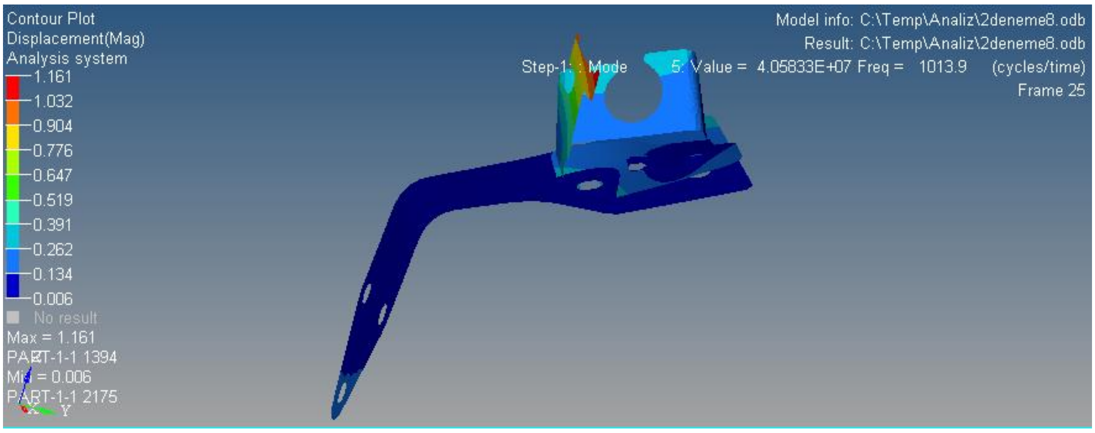

Figure 2.38: FEM for hypothesis bracket design (1013,9Hz).

Consequently, as it can be reviewed from figures 34 to 39 of FEM hypothesis bracket, connection stiffness is affecting mode frequencies have been shifted to lower values definitely. Detailed discussion will be performed at the section 4.1 from summary tables.

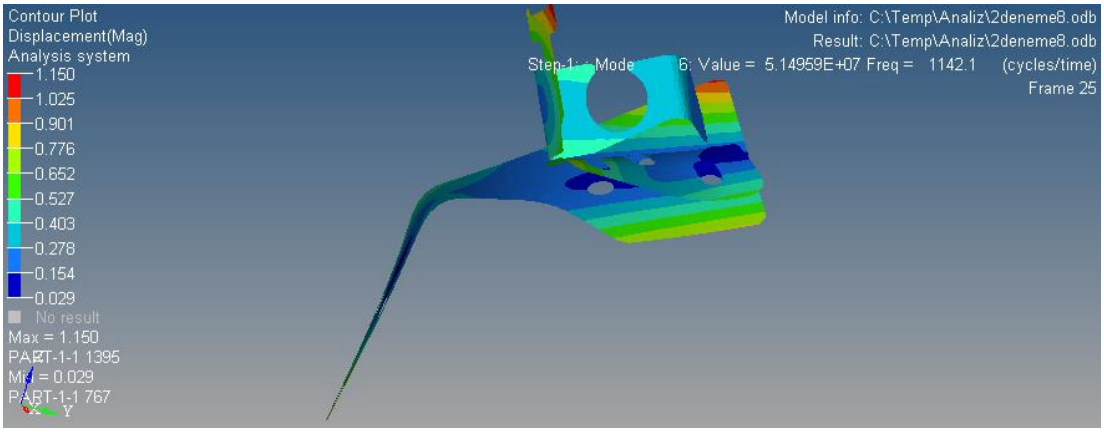

Figure 2.39: FEM for hypothesis bracket design (1141,1Hz).

2.4 NVH Solution with Reinforced Bracket Design

2.4.1 Finite element modal analysis of reinforced bracket design

According to SDM, rib stiffener addition is determined to increase bracket stiffnes for decreasing displacement. In addition, it is estimated that first mode will be shifted since first mode shape is at main part and rib is added under the main part. Reinforced bracket design meshed with same as baseline bracket design configuration. Hypermesh view can be reviewed at the Figure 2.40.

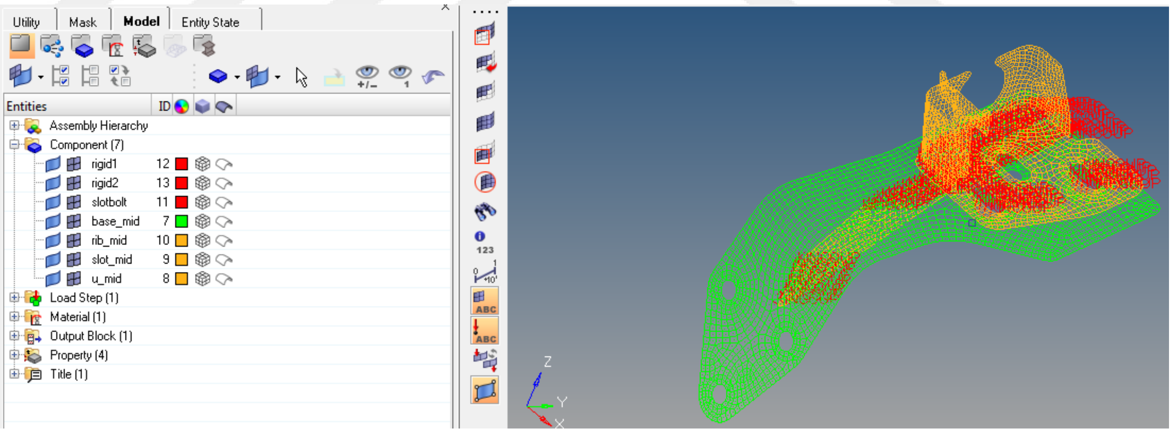

Figure 2.40: Hypermesh view for reinforced bracket design.

FEM result shows mode frequencies are increased and first mode is shifted to upper frequency. First four mode frequency can be reviewed from the table 2.5 and mode shapes framed at the Hyperview figures. From figures that can be reviewed mode shapes are a little different from baseline bracket design since first mode is completely related to main part. With rib stiffener this mode frequency has been shifter very high frequency that is not reviewed at the first four modes.

Table 2.5: Reinforced bracket design mode frequencies at the FEM.

Modes	FEM of Reinforced Bracket Design (Hz)
1	705,96
2	858,22
3	1055,2
4	1478,2

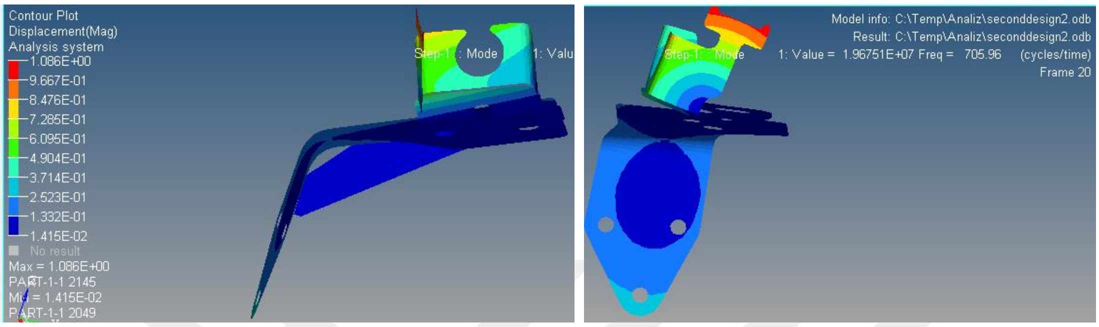

Figure 2.41: Reinforced bracket design mode shape-1 (705,96 Hz) at the Hyperview.

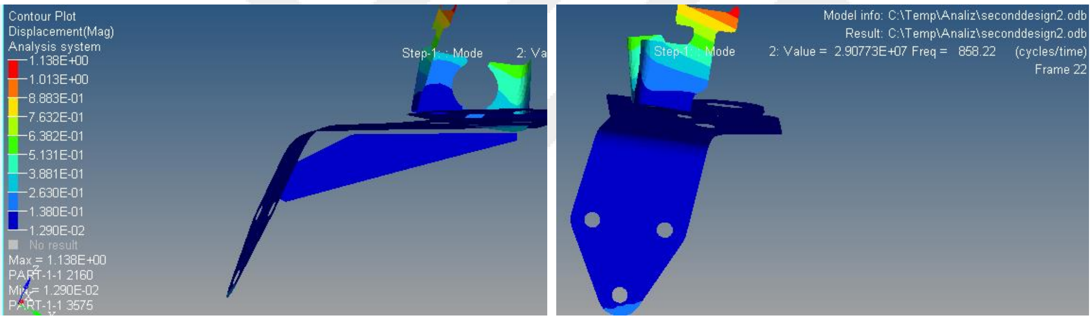

Figure 2.42: Reinforced bracket design mode shape-2 (858,22 Hz) at the Hyperview.

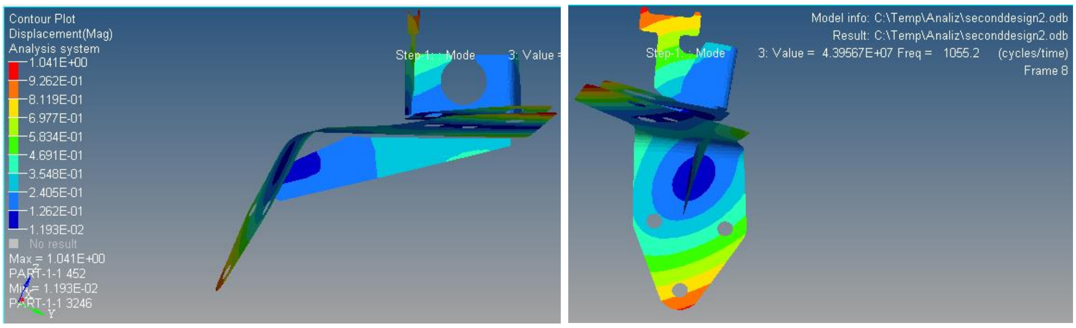

Figure 2.43: Reinforced bracket design mode shape-3 (1055,2 Hz) at the Hyperview.

2.4.2 Data collection from vehicle for reinforced bracket design

Reinforced bracket prototype part is prepared for try out event at the vehicle and objective data is collected to compare NVH chacteristic via colour maps.

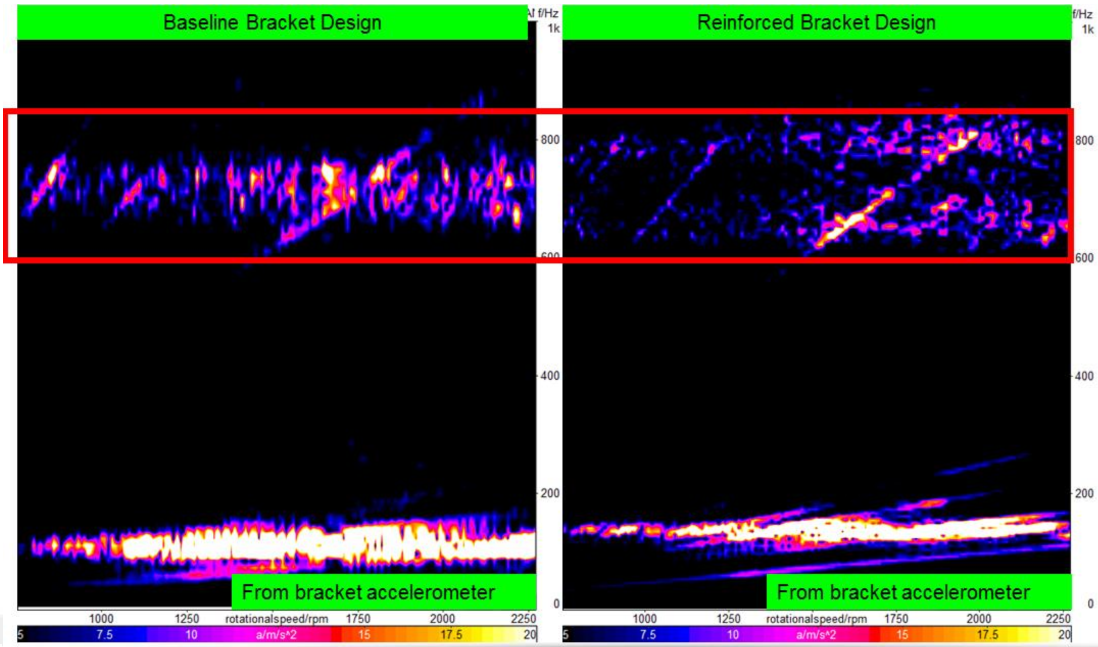

Figure 2.44: Vehicle data comparison for baseline vs reinforced bracket design at the bracket accelerometers.

Baseline shift cable holder bracket has a wide frequency range resonance band around 750 Hz. However, reinforced shift cable holder bracket has resonance frequencies at 650 Hz and 800 Hz with lower excitation values than baseline (figure 2.44).

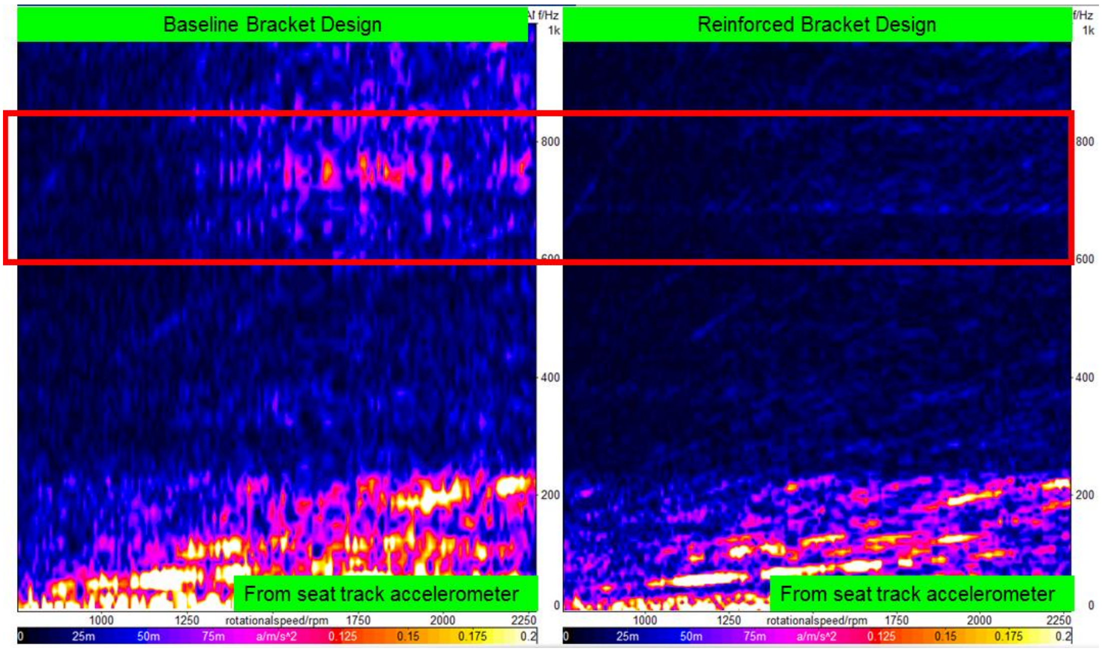

Figure 2.45: Vehicle data comparison for baseline vs reinforced bracket design at the seat track accelerometers.

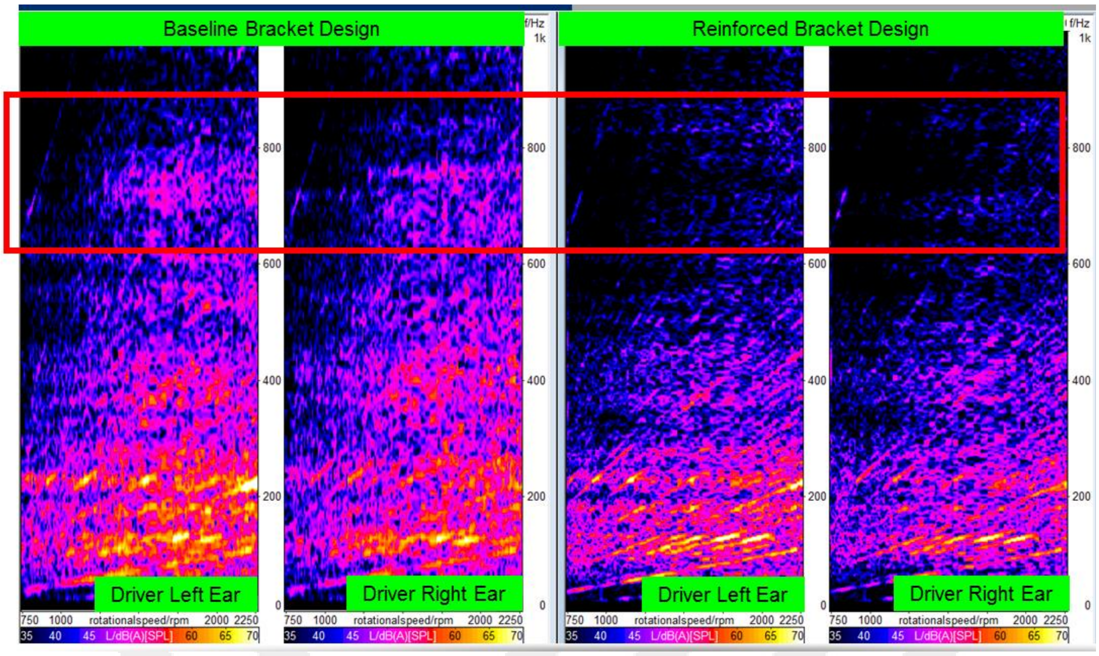

Figure 2.46: Vehicle data comparison for baseline vs reinforced bracket design at interior microphones.

Consequently, due to lower excitations and mode shapes at the frequencies, reinforced bracket design is not caused noise and vibration through cables to inside the cabin. It is verified from accelerometer at the seat track and microphone at the interior.

3. STRUCTURAL ANALYSIS

3.1 General Remarks

The purpose of bracket usage is supporting of overhanging components at the mounting location. In this case, shift cable holder bracket is mounting for selector (shift side) cable that is adjustable with upper part and tightening with M8 apex bolts though slot mounting holes to main part. Slot hole connection usage can be reviewed at the Figure 3.1. The selector cables transmitted driver's intent to change gear via mechanically though themselves to transmission.

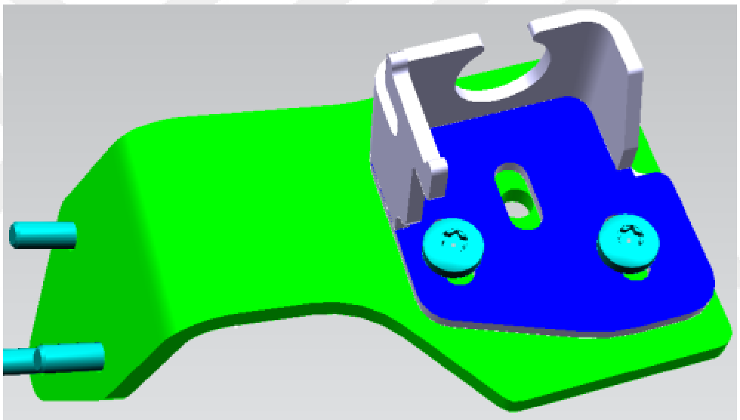

Figure 3.1: 3D top view of bracket.

Two different selection type (select or shift) can be defined to describe gear changing direction, example of shift pattern and selection direction is given at the Figure 3.2. While doing gear selection according to direction select or shift cable will be worked and applied force at gear knob will be transmitted selected cable according to moment arm of shifter that named mechanical ratio. Generally, accepting criteria of shift force is maximum 80N and select force is between 150-180N due to select gate pattern included two selecting option and additional spring to avoid misshift. In addition, mechanical ratio is changed with manual shifter characteric or OEM's request to reach their standart to keep good selection effort.

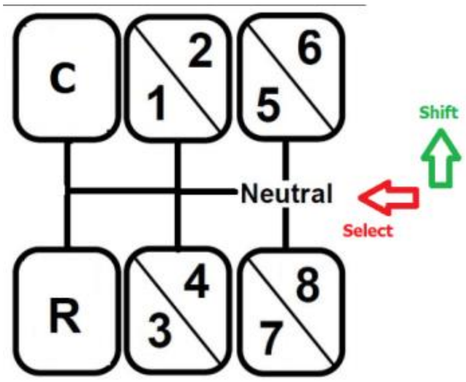

Figure 3.2: 9 speed HCV transmission gear pattern.

According to application force at shift selection that is defined above, structural durability performance of shift cable holder bracket will be investigated. The comparison of base and improved design for NVH characteristic will be also verified for durability sense in terms of Fatigue criteria of Von Mises.

3.2 Theoretical Calculation for Load Condition

The material used for Shift holder bracket is S500MC EN 10149-2. Chemical composition % of steel S500MC is given below.

- Modulus of Elasticity, E = 200 GPa
- Mass Density, ρ = 7850 kg/m3
- Yield Strength = 500 MPa
- Tensile Strength = 550 - 700 MPa

Chemical composition % of steel S500MC (1.0984): EN 10149-2-1996

The sum of Nb, V and Ti shall be max 0.22 %
If agreed at the time of the enquiry and order the sulfur content shall be max 0.01 % (ladle analysis)

C	Si	Mn	P	S	V	Nb	Ti	Al
max 0.12	max 0.5	max 1.7	max 0.025	max 0.015	max 0.2	max 0.09	max 0.15	min 0.015

Figure 3.3: Material chemical composition.

Accepted criteria shifting load is applied force at the customer contact point (shifter). Furthermore, design load condition is applied 80 N shifting load as maximum and it is directly transferring to cable connection part that has surface area 1022,87mm2. Moment arm at the shifter has been defined at the Figure 3.3 with pressure area at the bracket.

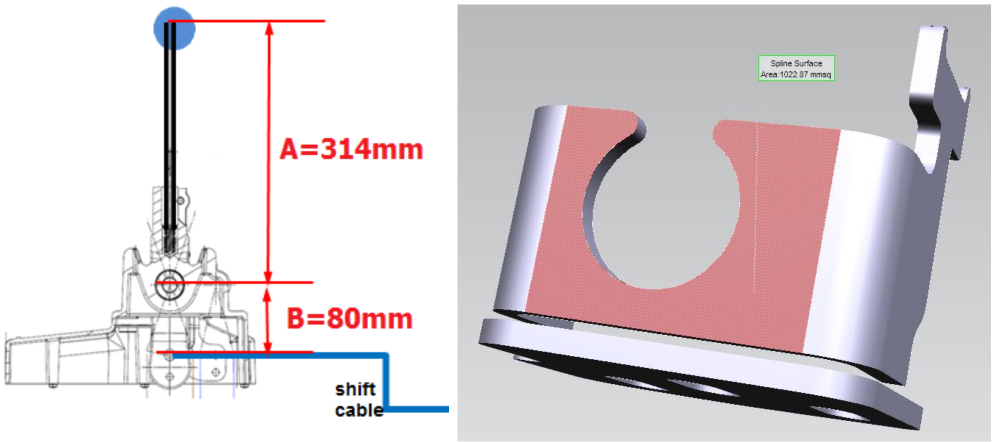

Figure 3.4: Shifter moment arm and pressure area at the bracket.

Table 3.1: Assumptions for shifting load and shift cable efficiency.

Assumptions (From vehicle test at the Ford Otosan)	
Shifting Load	80N
Shift Cable Efficiency	80%

$$\text{Mechanical Ratio} = \frac{\text{A}}{\text{B}} = \frac{314\text{mm}}{80\text{mm}} = 3{,}4 \tag{3.1}$$

Pressure on the bracket in downward direction can be found such as below;

$$\text{Calculated pressure} = \frac{\text{Shifting Load} * \text{Mechanical Ratio} * 2 * \text{Cable Efficiency}}{\text{Bearing Bracket Area}} = \frac{80\text{N} * 3{,}4 * 2 * 0{,}80}{1022{,}87 mm^2} =$$

$$\frac{435{,}2\text{N}}{1022{,}87 mm^2} = 0{,}425\text{MPa (approx)} \tag{3.2}$$

3.3 Structural Analysis for Baseline Bracket Design

Baseline shift cable holder bracket design meshed with same configuration such as finite element modal at the section 2.3.2 and data imported to Abaqus Structural Solver. Boundary condition is defined from three holes that are connected bracket to transmission housing as fixation points. Same as modal analysis three parts connected each other. Base part (green) connected to slot part (blue) with slot bolt rigidity as defined KINCOUP node to node connection. Between slot part and u part (orange) contact surface defined as weld. Force is defined as RB3 to simulate cable transmitting force to bracket from c-clip connection. Force value is taken as 435,2N (0,425MPa) for both direction from theoretical calculation. General view of prepared data at Hypermesh can be reviewed at the Figure 3.5.

Stress analysis and displacement results for baseline bracket design are shared with hyperview figures at the 3.6 and 3.7.

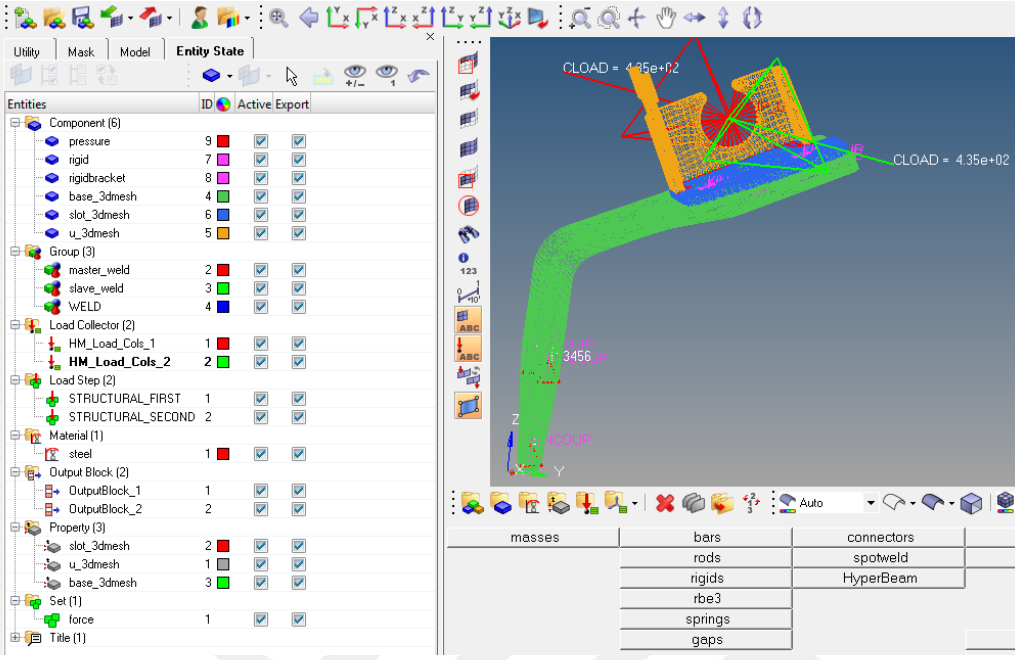

Figure 3.5: Hypermesh view for export data (baseline) to ABAQUS solver.

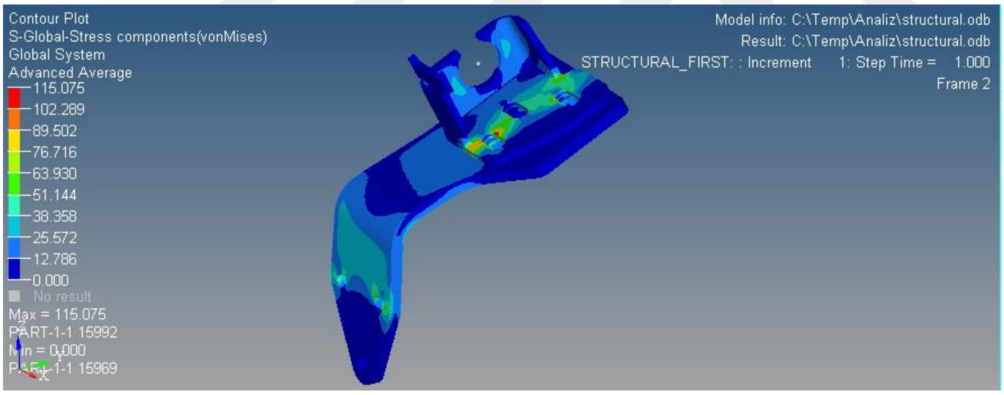

Figure 3.6: Stress analysis results for baseline bracket design according to Von Misses at the Hyperview.

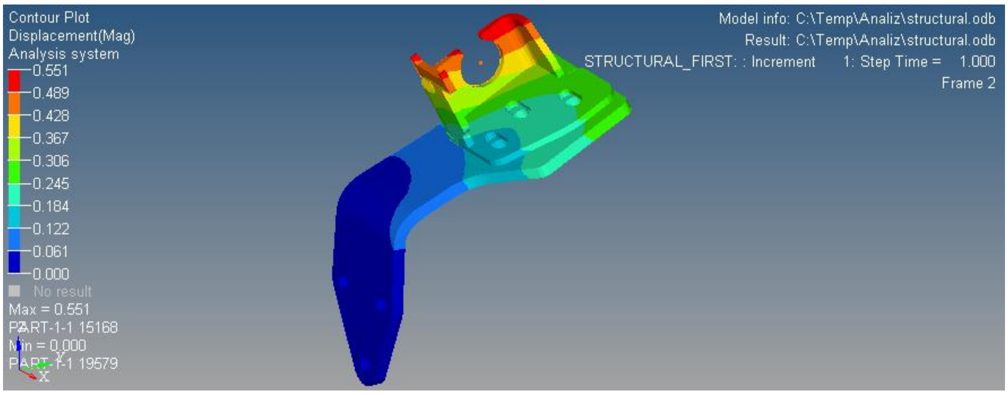

Figure 3.7: Displacement result (Baseline) at the Hyperview.

Structural analysis result shows 115,075 MPa can be seen at the slot holes according to Von Mises criteria. 115,075 MPa is below the half of yield strength of material is 250 MPa; thus, durability criteria of bracket design is oversafe.

Maximum displacement is obtained 0,551 unit for baseline bracket design. Comparison with reinforced design and discussion will be performed at section 4.2.

3.4 Structural Analysis for Reinforced Bracket Design

With same parameters of baseline bracket structural analysis, structural analysis reinforced bracket design performed. Figures 3.8-3.10 are shown modal tree and stress analysis and displacement results for reinforced bracket design.

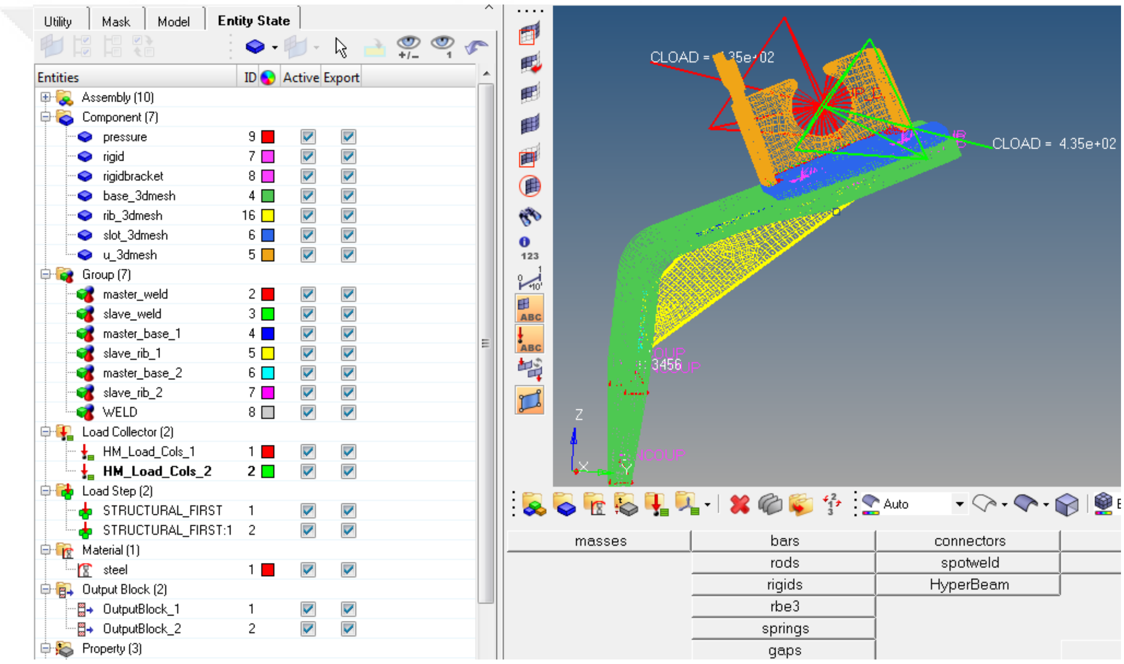

Figure 3.8: Hypermesh view for export data (reinforced) to ABAQUS solver.

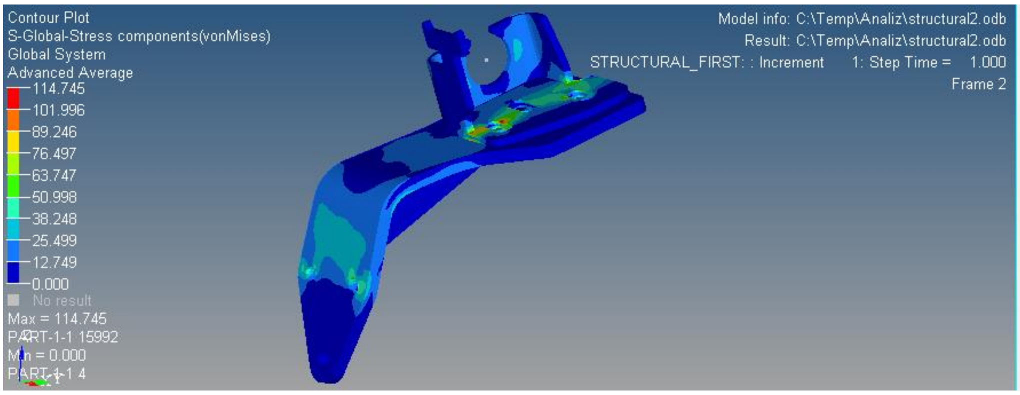

Figure 3.9: Stress analysis results for reinforced bracket design according to Von Misses at the Hyperview.

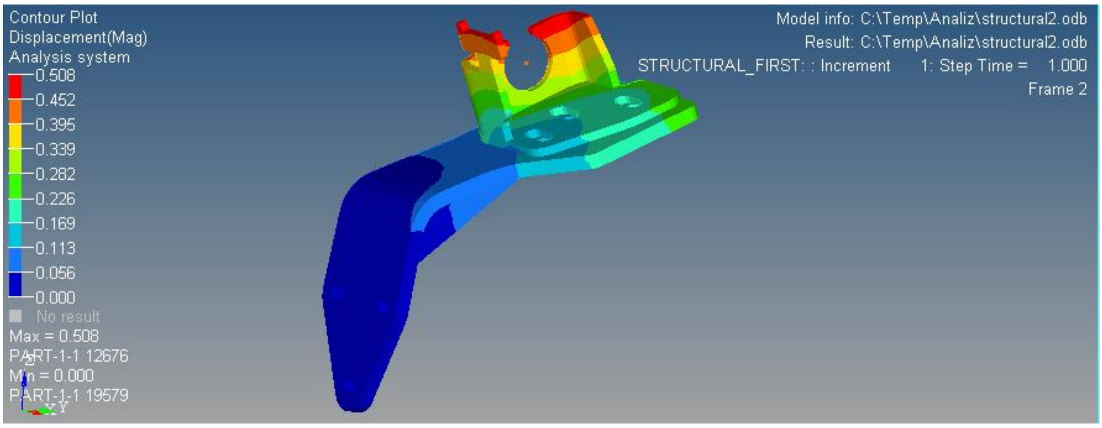

Figure 3.10: Displacement result for reinforced bracket design at the Hyperview.

For reinforced bracket design structural analysis result shows 114,745 MPa can be seen at the slot holes according to Von Mises criteria. 114,745 MPa is below the half of yield strength of material is 250MPa; thus, durability criteria of bracket design is oversafe.

Maximum displacement is obtained 0,508unit for reinforced bracket design. Comparison with baseline design and discussion will be performed at section 4.2.

4. RESULTS AND DISCUSSION

4.1 Discussion for Modal Analysis Results

After subjective and objective evaluation of vehicle with new transmission and external control application, the source of abnormal noise that is shift cable holder bracket is verified with accelerometer and microphone data and solution proposal is provided according to Structural Dynamic Modification (SDM) at the above sections.

While performing SDM to understand part characteristic experimental modal analysis (hammer excitation) and FEM is accomplished however some deviations have been observed. Experimental modal analysis samples show different mode frequencies and first four mode shapes are different from each other. Bracket sample 1 is shown very low mode frequency at first mode shape at the above clip connection part while bracket sample 2 first mode frequency is above than that and first mode shape is shown cantilever working at the main base connection part to transmission. Thus, hypothesis about connection stiffness is also came up with to refer difference between experimental modal analysis samples in addition to correlation of experimental modal with FEM.

Firstly, according to comparison of experimental modal of bracket sample 2 and finite element modal analysis of baseline bracket design, mode frequencies are matched between 0,15% and 5,41% and it is shown at table 4.1. The FEM is correlated with hammer excitation test of bracket sample 2. Mode shapes are also compared at the above sections with MATLAB and Hyperview figures. However this data is not shown why the bracket sample 1 mode frequency and mode shapes are different than bracket sample 2 and FEM. Secondly, previously mentioned hypothesis related to connection stiffness investigated with FEM iterations at the connection especially satisfied with apex bolts between main base part and slot hole connetion part.

Table 4.1: Comparasion summary table of experimental modal of bracket sample 2 and FEM for baseline bracket design.

Modes	Experimental Modal Analysis (Hz)		Finite Element Modal Analysis (Hz)	Correlation of Experimental and FEM
	Bracket Sample 1	Bracket Sample 2	Baseline Bracket Design	Bracket Sample 2 vs Baseline Bracket Design
1	380	580	587,67	1,32%
2	540	690	652,67	5,41%
3	760	866	867,27	0,15%
4	1080	1138	1122,5	1,36%

To verify hypothesis related to connection stiffness iterations have been performed for loosened connection (different connection stiffness) and ABAQUS solution data is imported to hyperview. Table 4.2 prepared to refer mode frequencies to compare experimental modal that is different from each other. Bracket sample 1 included lower mode frequency that mode shape is movement at upper part (u part connected to cable). If the slot bolt is not properly tightened to required torque value (25 Nm) or torque value is not enough to keep parts together mode shape can be observed as lower frequencies is verified with hypothesis bracket FEM.

Table 4.2: Comparison of experimental modal analysis and FEM for baseline and hypothesis bracket.

Modes	Experimental Modal Analysis (Hz)		Finite Element Modal Analysis (Hz)	
	Bracket Sample 1	Bracket Sample 2	Baseline Bracket Design	Hypothesis Bracket
1	380	580	587,67	239,56
2	540	690	652,67	514,09
3	760	866	867,27	651,74
4	1080	1138	1122,5	869,09
5	-	-	-	1013,9
6	-	-	-	1142,1

Consequently, deviation has been explained with FEM and correlated FEM can be used to verify and provide design solution for shift cable holder bracket. Rib stiffener addition has been verified with correlated FEM and prototype part at the vehicle. With rib stiffener first mode of bracket is shifted higher frequencies and excitations have been decreased.

In addition, structural performance and displacement values have been investigated deeply at section 3 and results have been commented at section 4.2.

4.2 Discussion for Structural Analysis Results

Stuctural analysis has been performed for both baseline and reinforced bracket designs and results of analysis have been summarized at table 4.3.

Table 4.3: Summary table for comparison of maximum stress and displacement for baseline vs reinforced bracket designs.

Part Name	Maximum Stress according to Von Misses (Mpa)	Maximum Displacement (Unit)
Baseline Bracket Design	115,075	0,551
Reinforced Bracket Design	114,745	0,508

Yield strengh of material is 250 MPa and maximum stress at parts is lower than half of this thus part design is oversafe.Both of design is almost show same maximum stress at slot hole connection, difference between each other is only 0,33 MPa that is 0,29%. This result can be commented that structural improvement at oversafe designs are not affected highly to decrease maximum stress since part is already infinite life according to Von Mises.

However, rib stiffener addition according to SDM is affected maximum displacement value due to part is working as cantilever and mode shapes are shown movement at the top of part. Displacement improvement is 0,043 unit that is corresponding 8,46%. It can be compared with real vehicle data measurement from accelerometers that are located at shift cable holder brackets. If reinforced bracket design and baseline design compared displacement decreased from 0,065µ to 0,060µ that is corresponding to 7,69% (figure 4.1). Difference between accelerometer data and FEM is only 0,77% according to displacement change percentages.

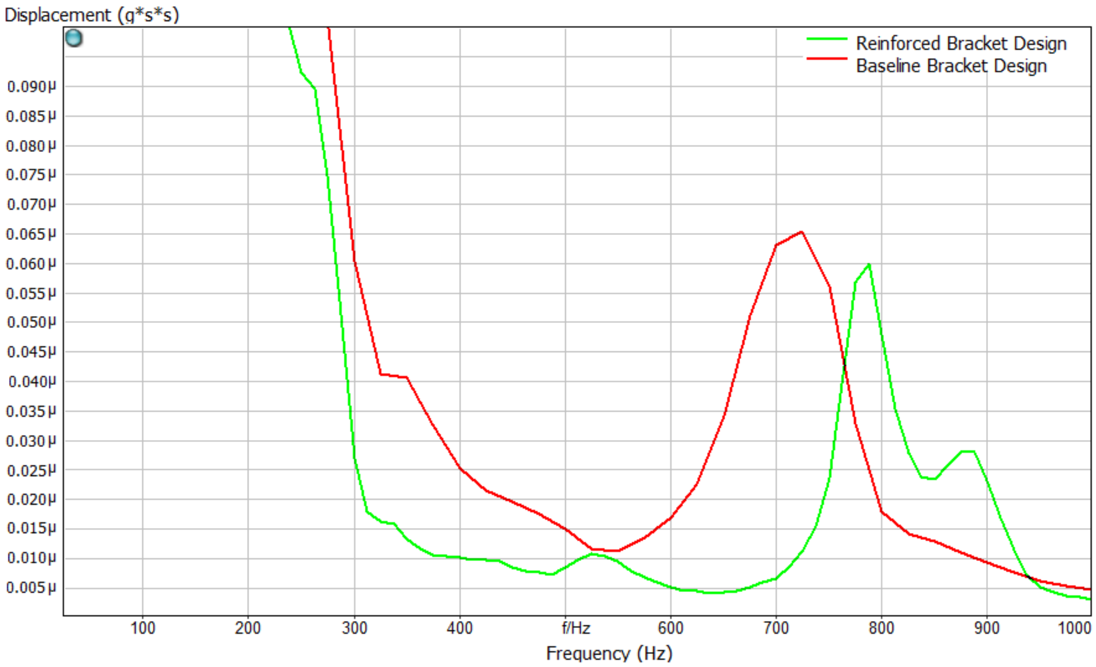

Figure 4.1: Displacement comparison according to accelerometer data at brackets.

5. CONCLUSIONS

In order to perform NVH analysis while adaptation of new transmission and external control system application vehicle test methodology and source identification have been defined with vehicle test and tool specifications. NVH problem at the one of the component of external control system that is shift cable holder bracket is deeply investigated according to structural dynamic modification methodology due to understand mode frequencies and shapes of part. Experimental modal analysis has been performed with two shift cable holder bracket and consistent with results of experimental modal analysis affect of connection stiffness difference has been shown with finite element modal iteration. If the connection stiffness (tightening torque) of apex bolts at the slot hole connection is not satisfied with required torque value undesirable noise & vibration can be shown due to mode frequencies are shifted lower values are provided. Correlated FEM has been used reinforced bracket design as proposal of SDM process. Owing to SDM and provided FEM, noise & vibration problem has been solved with reinforced bracket design. It is reported that first four modes of reinforced bracket design is shifted to upper frequencies especially first mode of baseline bracket that is working such as cantilever beam mode frequency is shifted to higher frequency.

Furthermore, structural analysis has been performed for both baseline and reinforced bracket designs. According to Von Misses fatique criteria two of the bracket designs have infinite life and maximum stress can be occurred at the slot connection holes with almost same stress values. Besides of structural analysis, maximum displacement values have been checked for both designs and reported that reinforced bracket design is improved with rib stiffener to decrease displacement. Rib stiffener affected to cantilever working at the bracket 8,46% improvement resulted at the FEM. Displacement value is compared with accelerometers data from brackets and the results are almost same value according to percentages of improvement (7,69%) with FEM.

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