

ISTANBUL TECHNICAL UNIVERSITY ★ GRADUATE SCHOOL OF SCIENCE
ENGINEERING AND TECHNOLOGY

**ASSESSMENT OF OPERATIONAL TRANSFER PATH ANALYSIS FOR
PRACTICAL APPLICATIONS**

M.Sc. THESIS

Gürkan TUTU

Department of Mechanical Engineering

Machine Dynamics, Vibration and Acoustics Programme

JANUARY 2015

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Thesis Advisor: Prof. Dr. Kenan Yüce ŞANLITÜRK

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ANALİZİ'NİN DEĞERLENDİRİLMESİ**

YÜKSEK LİSANS TEZİ

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To my family

FOREWORD

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ABBREVIATIONS

ADDOF-A	: Average Driving Degree-of-Freedom Acceleration
AH	: Acoustic Holograph
APS	: Auto Power Spectrum
APU	: Auxiliary Power Unit
CPS	: Cross Power Spectrum
CTC	: Crosstalk Cancellation/Compensation
DFT	: Discrete Fourier Transform
DOF	: Degree-of-Freedom
DSM	: Dynamic Stiffness Method
EI	: Effective Independence
EMA	: Experimental Modal Analysis
FE	: Finite Element
FFT	: Fast Fourier Transform
FIR	: Finite Length Impulse Response
FRF	: Frequency Response Function
GCV	: Generalized Cross Validation
GNP	: Gear Noise Propagation
GTDT	: Global Transfer Direct Transfer
GTDT	: Global Transmissibility Direct Transmissibility
IDFT	: Inverse Discrete Fourier Transform
iSPC	: in-situ Source Path Contribution
LSA	: Least-Square Algorithm
MDOF	: Multi Degree-of-Freedom
MDSA	: Multi-Dimensional Spectral Analysis
MIM	: Matrix Inversion Method
MIMO	: Multi Input/Multi Output
MISO	: Multi Input/Single Output
MSM	: Mount Stiffness Method
NTF	: Noise Transfer Function
NVH	: Noise-Vibration-Harshness
OCV	: Ordinary Cross Validation
ODS	: Operational Deflection Shape
OPA	: Operational Path Analysis
OPAX	: Operational Path Analysis with Exogenous Inputs
OTPA	: Operational Transfer Path Analysis
PAS	: Phase Assigned Autospectrum
PC	: Principal Component
PCA	: Principal Component Analysis
PDP	: Plasma Display Panel
POT	: Partial Open Throttle
PQRSM	: Progressive Quadratic Response Surface Method
PSVD	: Partial Singular Value Decomposition

RAD	: Rear Axle Differential
RMS	: Root Mean Square
S/N	: Signal to Noise
SDOF	: Single Degree-of-Freedom
SIMO	: Single Input/Multi Output
SISO	: Single Input/Single Output
SPC	: Source Path Contribution
SVD	: Singular Value Decomposition
TF	: Transfer Function
TPA	: Transfer Path Analysis
TPA-FORM	: TPA From Operational and Reciprocal Measurements
UFF	: Universal File Format
VATF	: Vibro-acoustic Transfer Function
VTF	: Vibration Transfer Function
VVS	: Volume Velocity Source
VVS	: Volume Velocity Source
WOT	: Wide Open Throttle

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ASSESSMENT OF OPERATIONAL TRANSFER PATH ANALYSIS FOR PRACTICAL APPLICATIONS

SUMMARY

Although the design process and manufacturing technology of complex mechanical systems, e.g. automotive and large home appliances, have been improved, the need for vibration and noise reduction has become even more important. In this sense, Transfer Path Analysis (TPA) – also defined as Source Path Contribution (SPC) Analysis – can be used for the identification of significant vibration source(s) and related vibro-acoustic transmission path(s) in a system to solve the Noise-Vibration-Harshness (NVH) problems and to improve the vibro-acoustic design.

Conventional TPA methods are based on overall signal synthesizing, where a contribution is computed from a multiplication of source strength and the Transfer Function (TF) from the source to the receiver. As these methods need identification of TFs between input and output variables as a separate process, they are more time consuming relative to the alternative methods. Furthermore, as the source strengths, also stated as operational input loads, cannot be determined directly but require the use of one of the indirect methods (e.g. Matrix Inversion Method (MIM), Dynamic Stiffness Method (DSM), etc.), conventional TPA methods results in additional uncertainties and less reliable results. Therefore, an alternative approach to classical or traditional TPA methods has been proposed in the mid 2000's. This new method, named as Operational Transfer Path Analysis (OTPA), is based upon signal decomposition technique. Within this approach, the TFs are directly estimated directly from measured operational data without the need for performing separate task for this purpose. Besides, the transmissibility function (the ratio of two responses) is generally used for the operational TFs instead of force to response TFs. In summary, OTPA is more practical than conventional TPA methods because of requiring operational measurements only and reducing total measurement time and effort.

The main purposes of this thesis are (i) to implement the OTPA procedure using the MATLAB code, (ii) to verify implemented OTPA procedure using a simple flat plate test case and (iii) to apply the OTPA to a washing machine. To achieve the objectives, a literature survey including the classical TPA and OTPA methods is performed first and presented in the first chapter. Following this, in the second chapter, the theoretical background of these two methods is explained in detail. To quantify various source(s) and/or transfer path(s) contributions, the background theory of classical TPA consists of two main stages: identification of Frequency Response Functions (FRFs) and determination of operational loads. Determination of operational loads is examined under two subsections as indirect methods for structural and acoustical operational loads, respectively. For the OTPA method, the theory is firstly introduced as a Least-Square Algorithm (LSA) problem for the

estimation of TFs (or FRFs). Then, the enhanced version of the OTPA theory is presented, which makes use of Singular Value Decomposition (SVD) and Principal Component Analysis (PCA) techniques to minimize the measurement noise influences and to avoid numerical problems during the TF estimation.

After introducing the literature survey and theory in detail, the OTPA method is employed for different case studies in this thesis. In chapter 3, this method is numerically and experimentally performed on a simple flat plate to carry out the basic steps of the OTPA method by comparing the measured quantities with the computed counterparts. Dynamic analyses are carried out on the structure to get necessary data for comparison purposes. In addition, the subjects of linear averaging and phase-assigned spectrums are described and used in the content of OTPA application on a simple plate. Finally, the OTPA algorithm using LSA and SVD techniques is implemented into a MATLAB code to process the measurement results to estimate the TFs and compute overall responses. In chapter 4, the OTPA method is applied to a front-loading drum-type washing machine to determine and rank the critical vibro-acoustic transmission paths. The source-transfer paths-receiver model, sensor positions and operational conditions for the analysis are explained in detail. FRF measurements are made and associated modal analysis is carried out on the mechanical system for the identification of appropriate response location during the OTPA analysis. Then, considering two different experimental cases for two different operational measurement sets, the application of OTPA to a washing machine is demonstrated and the results of the contribution analyses are discussed. In the last chapter, the general conclusions are briefly summarized and some recommendations for future studies are presented.

PRATİK UYGULAMALAR İÇİN OPERASYONEL İLETİM YOLU ANALİZİ’NİN DEĞERLENDİRİLMESİ

ÖZET

Otomotiv endüstrisinde olduğu kadar çamaşır makinası, buzdolabı gibi beyaz eşyalarda da göz önünde bulundurulması gereken en önemli hususlardan biri yüksek titreşim ve gürültü seviyesi problemi. Beyaz eşyaların titreşim ve gürültü seviyelerini düşürmeye yönelik yapılan atılımlar, daha ileri seviye bir ölçüm ve sinyal analiz yöntemi gereksinimini doğurmuştur. Müşteri gözünden bakıldığı takdirde, beyaz eşyalarda yüksek gürültü ve titreşim düzeyi istenmeyen durumların başında gelmektedir. Özellikle dengesiz yüklenme ve rezonans durumunda görülen yüksek gürültü ve titreşim seviyeleri, sistemin mekanik parçalarına ciddi oranda hasar vermektedir. Bu durum, sistemin daha fazla enerji harcaması, yorulma ömrünün kısalması ve kullanım süresinin azalmasına neden olmaktadır.

Titreşim ve gürültü sorunlarının çözümü için kullanılacak yöntemin, (i) mekanik sistemi tahrik ederek titreşime sebep olan kaynakları, (ii) oluşan titreşimi hava ve/veya yapı yoluyla ileten iletim patikalarını ve (iii) bu titreşim kaynaklı etkiye karşılık oluşan tepkinin meydana geldiği alıcıyı saptamaya yönelik bir yöntem olması bazı avantajlar sağlamaktadır. Dolayısıyla seçilen yöntemle, araştırılması gereken hususlar: (i) Yapı içerisinde tahrik sebebiyle oluşan vibro-akustik enerji akışının ilerleme biçimi, (ii) yapıdaki titreşim seviyesinin artmasına neden olan kaynak veya kaynakların tespiti, (iii) hangi iletim patikasının veya patikalarının titreşim iletiminde önemli rol oynadığı ve (iv) yapı üzerinde titreşim ve/veya gürültü kaynaklı tepkinin en çok nerede oluştuğunun belirlenmesidir. Bu sebeple, bu tez kapsamında literatürde “Operasyonel İletim Yolu Analizi (OİYA)” olarak isimlendirilen titreşim analizi ve sinyal işleme yöntemi kullanılmıştır.

Bu tez çalışmasında, OİYA’nın temel uygulama adımlarını kavrayabilmek adına, belirli bir alıcı noktasına farklı iletim patikalarından olan titreşim katkılarını belirlemek için, OİYA yöntemi bir beyaz eşya üzerinde farklı çalışma koşulları altında uygulanması hedeflenmiştir. Ayrıca, içerisinde geçirgenlik yaklaşımı, Tekil Değerlerine Ayrıştırma (TDA) ve En Küçük Kareler (EKK) yöntemlerini barındıran OİYA algoritmasının kullanıldığı bir sayısal çözücü programın geliştirilmesi de amaçlanmıştır.

Tezin ilk bölümünde, detaylı bir literatür araştırması sonucunda daha önceden yayınlanmış akademik çalışmalar, endüstriyel uygulamalar ve kullanılan yöntemler incelenmiştir. Literatür incelenmesi neticesinde, “İletim Yolu Analizi (İYA)” veya “Kaynak Patika Katkı Analizi (KPKA)” olarak isimlendirilen bu sinyal işleme ve titreşim analizi yönteminin, çoğunlukla otomotiv endüstrisinde, araç içi titreşim ve gürültünün enazlaması amacıyla kullanıldığı görülmüştür. Otomotiv endüstrisinin yanı sıra, helikopter, tren, kamyon, beyaz eşya gibi karmaşık yapıli sistemlerde, yüksek gürültü ve titreşim seviyelerine neden olan kaynak ve iletim patikasının

saptanması amacıyla kullanıldığı uygulamalara da rastlanmıştır. İncelenen İYA yöntemleri, tezin giriş bölümünde “Klasik İletim Yolu Analizi (KİYA)” ve “Operasyonel İletim Yolu Analizi (OİYA)” olmak üzere iki ana başlık altında toplanarak, ilgili çalışmalardan örnekler verilmiştir. Ayrıca, yöntemlerin olumlu ve olumsuz yönlerine de, literatür araştırması içerisinde yer verilmiştir.

Tezin ikinci bölümünde, İYA yönteminin teorik altyapısına yer verilerek, yöntemin temel uygulama adımları belirlenmiştir. Bu sayede, geliştirilecek sayısal çözücü algoritmanın, yapılacak sayısal ve deneysel çalışmaların temeli oluşturulmuştur. Mekanik sistemlerin içerisindeki ana titreşim kaynağının veya kaynaklarının sistemi uyarması sonucunda sistem üzerinde vibro-akustik yükler meydana gelir. İYA yöntemlerinde, bu vibro-akustik yükler sebebiyle oluşan toplam cevap fonksiyonu, birbirinden bağımsız iletim patikalarından gelen titreşim ve akustik yüklerin toplamı olarak tanımlanır. Bu deneysel analiz yöntemi, daha öncede belirtildiği gibi bir mekanik sistem içerisindeki vibro-akustik iletim patikalarının ve vibro-akustik yükleri oluşturan ana tahrik kaynaklarının belirlenmesi için kullanılır. Mekanik sistem üzerindeki vibro-akustik enerji iletimi, sistemi uyaran yapısal (titreşim) ve akustik yüklerin olduğu aktif bölgeden başlar. Bu iletim, fiziksel bağlantı noktaları üzerinden ve/veya hava yolu ile alıcı bölgesindeki cevap noktalarına ulaşır. Sistem üzerindeki vibro-akustik yüklerden etkilenen ve cevap noktalarını kapsayan bu bölgeye pasif bölge adı verilir. Dolayısıyla mekanik sistem, operasyonel yükleri oluşturan aktif bölge ve bu yüklerle karşılık tepkilerin (çıkıtlar) olduğu pasif bölge olmak üzere iki alt sistem şeklinde ele alınır.

Tezin teori kısmında, sayısal ve deneysel uygulamalarda kullanılacak İYA yöntemini belirlemek amacıyla, hem KİYA hem de OİYA yöntemlerinin teorilerine yer verilmiştir. KİYA'nın teorisi, sentezleme yani tümevarım metodunu esas alır. Bu yaklaşım ile lineer ve zamandan bağımsız bir sistem için toplam cevap fonksiyonu, dolayısıyla da sisteme etkileyen vibro-akustik yüklerin kısmi katkıları, iki fiziksel bileşenin sonucu olarak tanımlanabilir. Bileşenlerden birincisi, sistem üzerinde girdi noktası olarak tanımlanan vibro-akustik yükün etkilediği nokta ile yine sistem üzerinde çıktı noktası olarak belirlenen tepki noktası arasındaki Frekans Tepki Fonksiyonu (FTF)'dur. Diğer bileşen ise, sistem üzerinde vibro-akustik enerji oluşmasına sebep olan ve sistemdeki girdi noktasından etkileyen titreşim ve/veya akustik kaynaklı operasyonel yüküdür. Buradan yola çıkılarak, KİYA yönteminin, FTF'lerin bulunması ve operasyonel yüklerin hesaplanması olmak üzere iki temel aşamada gerçekleştirildiği söylemek mümkündür. KİYA yönteminde birinci uygulama adımı olarak, kaynak (girdi) ve alıcı (çıkıtlar) pozisyonundaki noktalar arası FTF'leri hesaplamak adına, modal çekiç testi veya sarsıcı testi kullanılır. FTF ölçümleri, her bir kaynak ve alıcı noktası arasında ayrı ayrı tekrarlanmalıdır. Ayrıca, bu ölçümler esnasında, iletim yolları arasındaki çapraz bağlaşım etkisini ortadan kaldırmak için, sistemin aktif ve pasif bölgesi arasındaki fiziksel bağlantılar kaldırılmalıdır. Bu durum, KİYA yönteminin kullanıldığı uygulamalarda, ölçüm süresinin uzamasına ve sistemin çalışma şartları altındaki davranışının tam olarak yansıtılamamasına sebep olmaktadır. KİYA yönteminde ikinci uygulama adımı olarak, yapısal ve akustik operasyonel yüklerin belirlenmesi gerekir. Sistem içerisindeki girdi noktalarına algılayıcı yerleşiminin geometrik kısıt nedeniyle zor olması ve operasyonel yüklerin çalışma koşulları altında doğrudan ölçülmesinin mümkün olmaması sebebiyle, operasyonel yükleri belirlemek adına, direk yöntemler yerine dolaylı yöntemler tercih edilmektedir. Tezin teori bölümünde, yapısal ve akustik operasyonel yüklerin hesaplanmasında kullanılan dolaylı yöntemler iki ayrı başlık altında incelenmiştir.

Yapısal operasyonel yüklerin hesaplanması için kullanılan dolaylı yöntemler, dinamik katılık metodu ve matris ters çevirme metodudur. Akustik operasyonel yüklerin (hacimsel hızlar) hesaplanması için kullanılan dolaylı yöntemler ise, noktadan noktaya yüzey örnekleme, ses şiddeti ölçümleri ve matris ters dönüşüm yöntemleridir.

KİYA yönteminin sahip olduğu dezavantajlar sebebiyle, bu yöntemin kullanıldığı uygulamalarda sistemin sahip olduğu gerçek sınır koşulları çalışma şartları altında tam olarak yansıtılamamaktadır. Özellikle, her bir kaynak ve alıcı arasındaki FTF ölçümlerinin ayrı ayrı yapılması ve ölçümler sırasında aktif-pasif bölgeler arası fiziksel iletim yollarının kaldırılması, KİYA yönteminin kullanıldığı uygulamalarda, vibro-akustik kaynak ve iletim yolu ayrıştırması işleminin hatalı sonuçlanmasına sebep olmaktadır. KİYA yönteminin sahip olduğu bu olumsuzluğu ortadan kaldırmak için, son yıllarda yeni bir İYA yöntemi üzerinde çalışılmaktadır. “Operasyonel İletim Yolu Analizi (OİYA)” olarak adlandırılan bu sinyal işleme yönteminde, sadece farklı çalışma şartları altında gerçekleştirilen operasyonel ölçüm verilerine ihtiyaç duyulmaktadır. OİYA’nın teorisi, ayrıştırma yani tümdengelim metodunu esas alır. Bu yöntemde, lineer sistem kabulü ile incelenecek yapı üzerinde, sistemi tahrik eden girdiler ve bu uyarıya tepki veren çıktılar belirlenir. Farklı operasyonel koşullar altında belirlenen girdi ve çıktı noktaları üzerinden ölçümler alınır. OİYA teorisinde öncelikli olarak, bu operasyonel verilerden faydalanarak sistem üzerindeki iletim yollarını tanımlayan transfer fonksiyonlarının (veya FTF) belirlenmesi gereklidir. Transfer fonksiyonlarının hesaplanması için, “Çok Girdili/Çok Çıktılı (ÇGÇÇ)” sistem kabulü yapılarak girdi ve çıktı matrisleri oluşturulur. OİYA teorisi kullanılan algoritmalarda, transfer fonksiyonu matrislerinin elde edilmesi için oluşturulan bu sistematik problem, EKK yöntemi kullanılarak çözülmeye çalışılır. OİYA teorisinde, girdiler ve iletim yolları arası çapraz bağlaşım etkisini ve ölçüm gürültüsünün sebep olduğu etkileri ortadan kaldırmak amacıyla, TDA metodu ile de ÇGÇÇ sisteminin transfer fonksiyonu matrisleri elde edilebilmektedir. Girdi matrisinin tekil değerlerine müdahale edilmediği takdirde, girdi matrisinin tersini almak için kullanılan her iki yöntem (TDA ve EKK) aynı sonucu verir. Oluşturulan sistematik problemin sonucunda, girdi matrisinin tersi çıktı matrisi ile çarpılarak her bir frekans değeri için operasyonel transfer fonksiyonu matrisleri elde edilir. Böylece, incelenen mekanik sistemin gerçek sınır şartları altındaki dinamik davranışı incelenebilir. Elde edilen bu operasyonel transfer fonksiyonu matrisi, ölçülen girdi matrisi ile çarpılarak hesaplanan çıktı matrisine ulaşılabilir ki bu işleme çıktı matrisi sentezleme işleme adı verilir. Ölçülen ve hesaplanan çıktı matrisleri karşılaştırılarak, OİYA yöntemin kullanıldığı uygulamada yöntemin doğruluğu sorgulanabilir. Ölçülen ve hesaplanan çıktılar birbiri ile örtüştüğü takdirde, katkı analizi aşamasında geçilerek baskın vibro-akustik kaynak ve ilintili baskın iletim yolu belirlenir. Bu sayede, gerekli iyileştirme çalışmaları yapılarak, titreşim ve gürültü seviyesi düşürülmüş daha iyi bir vibro-akustik tasarım ortaya konulabilir.

Tezin üçüncü bölümde, bu tez kapsamında geliştirilen sayısal çözücü algoritmanın doğrulanması için, basit bir düz panel üzerinde sayısal ve deneysel OİYA uygulamaları gerçekleştirilmiştir. Sayısal çalışmalarda, yapının sayısal modeli NX-IDEAS programı kullanılarak oluşturulmuştur. Daha sonra, MD-ADAMS programı ile karşılaştırma verilerinin elde edilmesi için yapı üzerinde zorlanmış titreşim analizleri gerçekleştirilerek ilgili noktalar üzerinden kuvvet ve ivme ölçümleri alınmıştır. Deneysel çalışmalarda ise, gerekli ölçümlerin alınabilmesi için bir deney

düzenegi hazırlanmış ve serbest-serbest sınır koşullarını yansıtabilmek adına yapı bir test düzenegi üzerinde elastik ipler vasıtasıyla asılmıştır. Ayrıca, kullanılan deney düzenegi ve ekipmanları, veri toplama sistemi ve yazılımı detaylı olarak açıklanmıştır. Bunun akabinde, gerekli karşılaştırma verilerinin elde edilmesi için yapı üzerinde zorlanmış titreşim analizi uygulanmış ve yapıya farklı tahrik kuvvetleri uygulanarak operasyonel ölçümler gerçekleştirilmiştir. Tez kapsamında geliştirilen MATLAB kodu kullanılarak, ölçüm sonuçları ile hesaplama sonuçlarının karşılaştırması yapılmıştır. Beklenildiği gibi, OİYA yöntemi kullanılarak hesaplanan yapısal tepkiler ve operasyonel ölçümler sonucu elde edilen yapısal tepkiler birbirleri ile örtüşmektedir. Fakat bu durum, düz panel üzerinde uygulanan OİYA yönteminin doğruluğunu kanıtlamak için gerekli fakat katkı analizinin güvenilir bir şekilde yapılabilmesi için yeterli değildir. Bu maksatla, deneysel ve sayısal uygulamalar içerisinde, FTF ve faz açılarının karşılaştırılması yapılmıştır. Burada, “Tek Girdili/Çok Çıktılı (TGÇÇ)” sistem kabulü kullanılarak zorlanmış titreşim sonucu elde edilen FTF’ler ve onların faz açıları, ÇGÇÇ sistem kabulü kullanılarak OİYA algoritması vasıtasıyla hesaplanan FTF’ler ve faz açıları karşılaştırılmıştır. Karşılaştırma sonuçları göstermiştir ki, lineer ortalama alınması ve faz bilgisi içeren autospektrum kullanılması ile daha güvenilir ve ölçüm gürültüsü en aza indirilmiş FTF’ler elde edilmektedir. Burada faz bilgisi içeren autospektrumların kullanılmasının sebebi, FTF’lerin, girdi ve çıktı spektrumlarının faz bilgisini korumak ve algoritmanın işlem süresini kısaltılabilmektedir. TGÇÇ ve ÇGÇÇ FTF’lerinin ve faz açılarının karşılaştırılması, geliştirilen MATLAB kodunun OİYA algoritmasını doğru bir şekilde uyguladığını ve ÇGÇÇ FTF’lerinin güvenilir bir biçimde hesaplanabildiğini ortaya koymuştur.

Tezin dördüncü bölümünde, OİYA yöntemi kullanılarak, önden yüklemeli bir çamaşır makinası üzerinde en baskın titreşim iletim yolu saptanmaya çalışılmıştır. Bölüm içerisinde ilk olarak, çamaşır makinasının çalışma prensibine değinilmiş ve tahrik grubu, süspansiyon sistemi gibi önemli alt sistemleri açıklanmıştır. Çamaşır makinasının en önemli titreşim probleminin, sistemin tahrik grubunun rezonans veya dengesiz yüklenme durumundaki eksantrik hareketinden kaynaklandığı bilinmektedir. Uygulamada ilk adım olarak, OİYA yöntemi için, “kaynak-iletim yolları-alıcı” modeli kurulmuş ve çamaşır makinası üzerindeki önemli vibro-akustik enerji iletim yolları tanımlanmıştır. Yapı üzerinde tanımlanan “kaynak-iletim yolları-alıcı” modeline göre, ana titreşim kaynağı tahrik grubu olup, vibro-akustik yükler süspansiyon sistemi üzerinden çamaşır makinası gövdesine yani panellere aktarılmaktadır. Panellere olan titreşim iletiminin en çok hangi yönde gerçekleştiğini belirlemek için, çamaşır makinası kazanı üzerinde titreşim ölçümü yapılmıştır. Alıcı olarak belirlenen panel yüzeyleri oldukça geniş bir alana sahip olduğundan, en iyi ivmeölçer ölçüm noktalarını, yani alıcı üzerindeki en uygun tepki ölçüm yerlerini, belirlemek için çamaşır makinasının modal analizi yapılmıştır. Çamaşır makinası OİYA uygulamasında, akustik yayılım etkileri ve hava yollu iletim patikaları ihmal edilmiştir. Ayrıca, titreşim ölçümünün sonucu olarak çamaşır makinasının gövdesine, süspansiyon sistemi üzerinden Z yönünde (derinlik boyunca) gerçekleşen titreşim iletimi ve lineer olmayan yapısı nedeniyle körük üzerinden olan titreşim iletimi ihmal edilmiştir. Mekanik sistem üzerinde, tek bir ana titreşim kaynağı olduğundan vibro-akustik iletim patikaları sistem girdileri (input) olarak kabul edilmiştir. Operasyonel ölçümler, farklı tambur devir hızları ve farklı dengesiz yüklenme durumları için ayrı ayrı tekrarlanmıştır. İlave olarak, tek etkili ve çift etkili sönümleyici kullanım farkını gözetebilmek adına iki farklı operasyonel ölçüm seti hazırlanmıştır. Bu sayede, sistemin birbirinden bağımsız çalışma koşulları altındaki dinamik davranışının

modellenmesine olanak tanınmıştır. Operasyonel ölçüm sonuçları PULSE LabShop yazılımı kullanılarak kaydedilmiş, iletim patikası ayrıştırma ve katkı analizi için ise tez kapsamında geliştirilen MATLAB kodu kullanılmıştır. OİYA sonuçlarına göre, toplam panel titreşimine en fazla katkıda bulunan baskın iletim yolunun frekans bantlarına değişiklik gösterdiği görülmüştür. Buna ek olarak, farklı operasyonel koşullarda baskın iletim yolunun değişebileceği fark edilmiştir. Diğer taraftan, kesin olan sonuç ise, çamaşır makinası süspansiyon sisteminde çift etkili sönümleyiciler kullanıldığı takdirde, panelin toplam titreşim seviyesine, sönümleyiciler üzerinden olan titreşim katkısının azaltılabildiği ortaya konmuştur.

Tezin son bölümünde ise, genel sonuçlar yorumlanmış ve ileride yapılabilecek çalışmalarla ilgili fikirler sunulmuştur.

1. INTRODUCTION

Transfer Path Analysis (TPA) method has been employed for Noise-Vibration-Harshness (NVH) analysis, especially in the automotive industry, since the 1970's. TPA is a classical test-based signal processing technique where the flow of vibro-acoustic energy from a main excitation source to a defined receiver location via a group of known structure-borne and airborne transfer paths. The aim of classical TPA procedure is to quantify the contribution of each vibro-acoustic energy transmission path from the source to the receiver. Therefore, the critical system component(s) along that transmission path can be determined to solve the particular NVH problem on the mechanical system by modifying the related components or optimizing the system design. The classical TPA is easy in concept; it provides a systematical approximation to vibro-acoustic design and quicker problem-solving in NVH problems. The determination of Frequency Response Functions (FRFs) or Transfer Functions (TFs) is time-consuming process due to the need for significant amount of measurements and decoupling of the main source(s) from the system. Moreover, it is required to cancel the crosstalk effects between the vibro-acoustic transmission paths that complete system must be divided into subsystems and these transfer paths must be physically isolated for the FRF measurements. For this reason, it is difficult to carry out the classical TPA procedure on a typical mechanical system. As a result, the performed analyses using this classical method may not result in reliable source contribution estimation.

In the mid-2000's, an alternative method to the traditional TPA methods was proposed. The improved method is based on signal decomposition technique. It is defined as Operational Transfer Path Analysis (OTPA) where the TFs are estimated from the operational measurement data. In this method, a response to response TF (transmissibility) is generally used instead of a force to response TF (FRF) to define the transfer path relationship between input and output quantities. Thus, the OTPA method is a time-saving procedure owing to that the source decoupling or transfer path isolation is not required during the FRF measurements. The main advantage of

OTPA over the classical one is that operational effects are taken into consideration wherefore the operational measurements are performed on the operating system. Furthermore, the OTPA method enhanced with Singular Value Decomposition (SVD) technique yields more reliable results and better TF estimation by compensating the crosstalk effect and reducing the influences of external disturbances and measurement noise.

1.1 Problem

To handle the NVH problems is one of the most significant design criteria that need to be considered in the automotive industry as well as in large home appliances (e.g., washing machine, household refrigerator). The requisition for further improvement in the vibration and noise characteristics of large home appliances has generated a requirement for advanced measurement and signal analysis techniques. From the costumers' point of view, the excessive levels of noise and vibration of large home appliances are undesirable and they have becoming important factors affecting the customer decisions. Furthermore, excessive vibrations can cause serious damages on the mechanical components of a system due to unbalanced loading conditions and resonance issues. They can also result in energy wasting and reduction in fatigue life.

To solve the NVH problems, the preferred method should be capable of identifying the source which generates the vibration by exciting the mechanical system, the transfer paths which transmit the vibration via structure-borne and/or airborne, and the receiver where the corresponding responses occur. The required outputs of this method are how the flow of vibro-acoustic energy propagates through the structure, what excitation source(s) causes the most vibration, which transfer path(s) play critical role in the vibration transmission and where the overall response is produced against the operational input load(s).

Although TPA methods are used extensively in various industries, particularly in automotive industry, it appears that TPA methods are either not applied or not published in the literature for home appliances industry. Accordingly, what the TPA can offer for home appliances is not clear. Therefore, the Operational Transfer Path Analysis (OTPA) technique using transmissibility approach and improved with the SVD method is examined in the scope of thesis for better understanding of the dynamic behavior of a washing machine.

1.2 Literature Survey

Transfer Path Analysis (TPA), also known as Source Path Contribution (SPC) analysis, is an experimental and/or simulation-based method to determine and order particular vibration and/or noise contributions by following the vibro-acoustic energy transmission via various structure-borne and/or airborne transfer paths from an excitation source to a defined receiver [1-3]. The aim of TPA procedure is to help engineers to solve Noise-Vibration-Harshness (NVH) problem on a mechanical system by analyzing vibro-acoustic transmission paths and their relative contributions to the overall vibration and/or noise levels [2, 3].

The TPA method is commonly used in the automotive industry to investigate and solve the NVH problems. In the literature, it is noticed that most of the studies (e.g. articles, Ph.D. and M.Sc. thesis, conference papers) involves with TPA are implemented on vehicles. Also, there are other studies about TPA that are applied to the large home appliances, helicopter, etc. Some of these applications are described below.

Van Karsen et al. [4] investigated the applicability of TPA to large home appliances, e.g. washing machine, to investigate the vibration and noise behavior of them. In the study, the first phase of TPA procedure was introduced as the indication of the obvious sources by examining the principal components which are found by applying Singular Value Decomposition (SVD) to the Cross Power Spectrum (CPS) matrix of the response. When the sources are correlative, the single reference TPA approach is applied to the home appliance instead of multiple reference TPA approach that is used when multiple non-correlative sources exist on the structure. In the study, during the second phase of TPA procedure, subsystems were selected and boundary conditions were defined. Also, employing a free boundary condition at the connection points was advised during the measurement of Frequency Response Functions (FRFs) on the receiver subsystem.

Zheng et al. [5] used TPA enhanced with Tikhonov regularization and Acoustic Holograph (AH) method to examine and envision the contribution of the pass-by noise sources of a truck. Sievi et al. [6] implemented the Operational Transfer Path Analysis (OTPA) based on Crosstalk Cancellation (CTC) method on a train for identification of the dominant sound path to the interior noise of train, where the

structure-borne sound is transmitted via the components of bogie system and generated by vibrations of the interior panels. Roozen et al. [7] investigated the use of transmissibility method on the OTPA improved with SVD that was applied to a small gearbox to identify the number of physical transmission paths. Mucchi and Vecchio [8] presented the results of experimental vibro-acoustic TPA on helicopter and compared the measured and synthesized/evaluated Transfer Functions (TFs), operational forces, path contributions and acoustical responses. Kwak et al. [9] applied TPA using Multi-Dimensional Spectral Analysis (MDSA) to determine the computational noise contribution of each main board of a Plasma Display Panel (PDP) television. Sung and Nefske [10] discussed the theoretical framework of the structural-acoustics TPA to estimate the acoustic effect coefficients and the contributions of panel vibration source to the interior noise of a vehicle.

In the literature, Transfer Path Analysis is basically classified into two categories: The first one is the so-called classical TPA which consists of two steps that are measurement of FRFs (or TFs) and determination of operational forces. The second one is Operational TPA (OTPA) that is a single stage procedure where only operationally measured data are used to estimate the Transfer Functions (TFs).

1.2.1 Classical transfer path analysis

Classical TPA techniques based on the Frequency Response Function (FRF) measurements can be categorized into four different approaches [11]:

- The classical TPA approach measuring interface forces [1]
- The Matrix Inversion Method (MIM) [12-16]
- The Mount Stiffness Method (MSM) or Dynamic Stiffness Method (DSM) [2, 21, 22]
- The Gear Noise Propagation (GNP) or component TPA method [23]

FRF-based TPA techniques comprise two stages: FRF measurement and operational force identification for response signal estimation. First of all, FRFs are measured between identified input (reference) and output (response) points comprising vibration and/or sound. If the vibration is structure-borne, shaker(s) or impulse hammer is utilized. On the other hand, if the vibration is airborne, use of loudspeaker(s) is required for FRF measurement. During the second stage, the

operational forces are identified at the reference points and multiplied with FRFs to estimate the responses for the quantification of each source contribution [1, 11].

Plunt [1] introduced the mid and high frequency TPA with an application example of a passenger car that had the medium frequency range periodic noise problem. Due to larger modal overlap, the determination of particular source contributions and/or transmission paths by the conventional TPA procedure using complex FRFs may be inaccurate at higher frequencies [1]. Averaging the Transfer Functions (TFs) over frequency and/or implementing spatial averaging at the receiver (response) location is advised to extend the contribution analysis into mid and high frequency range in this study.

Matrix Inversion Method (MIM) is an indirect method [1, 2] that is used to determine the operational forces in the classical TPA procedure. The operational forces (F_N) are acquired by multiplying the inversion of the FRF (H_{MN}) matrix, measured between structural responses and exciting forces performing at whole interfacing Degrees of Freedom (DOFs) on the receiver side, with the vector of operational accelerations ($\ddot{x}_M$) measured on the receiver side as:

$$\begin{Bmatrix} F_1 \\ \vdots \\ F_N \end{Bmatrix} = \begin{bmatrix} H_{11} & \cdots & H_{1N} \\ \vdots & \ddots & \vdots \\ H_{M1} & \cdots & H_{MN} \end{bmatrix}^{-1} \begin{Bmatrix} \ddot{x}_1 \\ \vdots \\ \ddot{x}_M \end{Bmatrix} \quad (1.1)$$

Here, M indicates the number of responses which should be greater than N indicating the number of force DOFs.

The use of MIM is suggested if the transmission paths consist of rigid connections and/or the mounts are stiffer than the receiving structure [1, 2]. Kim and Kang [12] utilized FRF-based synthesis method and classical TPA procedure to reduce the interior noise of a vehicle compartment by predicting and controlling the alteration in the local stiffness of the mounting point at which operational force is acting on. Zang et al. [13] investigated the feasibility of strategies such as over-determination and singular value rejection to enhance the operational load identification by applying classical TPA model to an automobile. Le et al. [14] carried out a TPA experiment to quantify the transfer paths of steering wheel vibration by observing the effects of paths coupling. Inoue and Tanabe [15] recommended the use of the particle velocity

TPA together with the traditional sound pressure TPA due to supposition of having the same decomposition method and validated experimentally the particle velocity TPA method on a simple test box system. Xie et al. [16] studied a test and simulation combined TPA with optimization method to reduce the steering wheel vibration in idle. Eventually, the MIM is conducted to identify the operational forces in these mentioned publications [12-16].

Inverse techniques such as MIM are used in classical TPA to obtain the operational forces at the excitation locations or at the junction points for the identification of structure-borne transmission paths. The forces obtained may be inaccurate if the inversion of the measured data (TF) matrix is ill-conditioned. In such cases, Tikhonov regularization method is generally preferred to improve the identification of the unknown operational forces.

Choi et al. [17, 18] published two articles about the use of Tikhonov regularization in inverse force determination. In the first article [17], the utilization of a threshold condition number was suggested to enhance the consequences of Tikhonov regularization. A crossover condition number was defined as the most suitable method to be chosen for remodeling the operational forces while implementing the pseudo-inverse method and Tikhonov regularization. In the second article [18], the selection of the optimal regularization parameter in Tikhonov regularization was examined. In this study, a rectangular plate is analyzed to compare the methods for parameter selection like L-curve method, Ordinary Cross Validation (OCV) method and Generalized Cross Validation (GCV) method. The L-curve method gives more reliable consequences than OGV and GCV method at the higher noise levels. On the other hand, OGV method performs better than GCV method when the condition number is high in the matrix inversion. Thite and Thompson [19, 20] published two articles to enhance the conditioning of the matrix inversion in the inverse force determination. In the first publication [19], a rectangular plate was tested for the quantification of the structure-borne transmission paths to demonstrate different techniques used in overcoming problems of ill-conditioned matrix. Owing to the higher modal overlap, specific over-determination gives better results at higher frequencies for decreasing the condition numbers. Besides, the rejection of singular values with an appropriate threshold will be used to decrease the error amplification. In the second publication [20], a rectangular plate was tested for the quantification of

the structure-borne transmission paths to represent the regularization methods utilized for improving the inverse force determination. Tikhonov regularization, utilizing OCV to choose the regularization parameter, performs better than other regularization techniques to specify the particular operational forces and their contributions.

Another indirect method [1, 2] that is applied in the classical TPA approach to obtain the operational forces is the so-called Mount Stiffness Method (MSM). Knowing the complex dynamic transfer matrix for the various DOFs is required in this method where flexible connecting elements are utilized as force transducers. Then, the operational forces are evaluated as:

$$F_{nk}(\omega) = k_{nk}(\omega)(x_{nk1}(\omega) - x_{nk2}(\omega)) \quad (1.2)$$

Here, F_{nk} is the complex force spectrum, k_{nk} is the complex transfer stiffness for mount n in direction k , x_{nk1} is the operating displacement at the source side, and x_{nk2} is the operating displacement at the receiver side.

The use of MSM is advised if the transmission paths include resilient connections and/or the mounts are more flexible than the receiving structure [1, 2]. Plunt [2] examined the idle boom and vibration problem of a front-wheel drive car that has a flexible mounted subframe. Experimental TPA was combined with FE to identify possible transmission paths, e.g. rubber mount connections. To obtain excitation forces, MSM was implemented that results in preferring soft bushings (useful for solving idle NVH) or hard bushings (useful for handling) usage in the mount connections.

Kim and Lee [21] presented the hybrid TPA method to estimate the interior noise generated by the structure-borne transmission path of the powertrain. Instead of the measured force, the simulated excitation force estimated by numerical analysis of the Finite Element (FE) model of a powertrain is assumed as the input force in the hybrid TPA technique. The interior noise was estimated by combining the simulated force by the Vibro-acoustic Transfer Function (VATF) of the vehicle. In addition, MSM was used to evaluate the excitation force that acts the resilient body of a car at the mount points. As a result, the evaluated force and interior noise were compared with the simulated force and interior noise.

Vecchio et al. [22] demonstrated the consequences of an experimental test applied on a hybrid city bus whose driveline system includes two electric motors and an internal combustion engine including a battery pack. In this study, TPA method was used to estimate the transfer functions and to determine structure borne and airborne vibro-acoustic loads which provided to identify the acoustic cavity modes and the responses based on actual operational situation in the cabin. Structure borne contribution of Auxiliary Power Unit (APU) of the hybrid bus was examined with implementing single reference TPA approach. Operating forces were estimated by utilizing MIM due to rigid connections in the subsystems. Required FRFs were obtained from measured structural accelerations and acoustical responses owing to force excitation at transmission path. Moreover, SVD technique was implemented for the inversion of the FRF matrix to minimize numerical errors. For airborne contribution of APU of the hybrid bus, the acoustic reciprocity feature was utilized to determine the Noise Transfer Functions (NTFs), owing to the APU slot size limited the placement of the Volume Velocity Sources (VVSs) in the engine compartment. Near-field intensity measurements under steady state condition were made on the APU in the anechoic room to estimate the operating acoustics loads as the volume velocities. Considering the identified NTFs and the estimated volume velocities, the partial pressure generated by the APU was calculated. On the other hand, the electric motors had structure borne contribution on the vibro-acoustic energy transmission. In this case, the FRFs were determined by using reciprocity property and the operational forces were estimated by applying the Dynamic Stiffness Method (DSM) where the operational accelerations had to be measured on both the active and passive sides of the bushings. At the end of analysis, it was found that the measured and the calculated pressure levels were different from each other due to various reasons such as (i) neglected rotational degrees of freedom in the structure borne case, (ii) the effects of the other source contributions which were not taken into account (use of the single reference TPA approach) and (iii) the difficulty of the determination of the mount stiffness of the electric motor (DSM utilization).

Another FRF-based TPA method is the component TPA method that is also known as Gear Noise Propagation (GNP) method. De Klerk and Rixen [23] discussed the theory of the component TPA by using an example of a vehicle gearbox combination and applied the method on the Rear Axle Differential (RAD) of a vehicle to

investigate the GNP. In this method, the true excitation source is measured individually on a component test setup to identify the equivalent input forces. For the study [23], RAD was considered as the excitation source. Afterwards, the FRF measurement between the subcomponent interface and selected response points is made on the complete system (vehicle) comprising the excitation source (RAD). In the study, the component TPA was corroborated by the GNP application on the RAD of the vehicle. Results showed that the flexibility of the test bench had minor effect on the gear noise forces. It was also found that the effect of uncompensated rotational inertia of the test bench may be reason for difference between the measured and calculated total responses of the driver's ears at the higher frequencies.

Classical TPA methods based on FRF measurement are time-consuming procedure due to one-to-one FRF measurement between defined reference and response locations, and also the active parts of the system is sometimes removed for FRF measurement. Moreover, placement of measurement equipment, e.g., accelerometer, force transducer, onto the mount connection or target point can be a problem owing to insufficient measurement location. Therefore, the procedure needs to be optimized to shorten the testing time and improve the measurement quality. Tcherniak and Schuhmacher [24, 25] reported the main disadvantage of classical TPA in their studies about NVH source contribution analysis by this means it is an obstructive time consuming process owing to the requirement of TF measurements. In the first study [24], a decomposition based-technique based on the estimation of the transmissibility matrix was applied to a simple synthetic numerical test model for the decomposition of the received sound. The theory of the Transmissibility Matrix Method was presented where the transmissibility matrix can be directly predicted from several sets of operating measurements. The method demands solely operational data; thereby FRF measurements are not required to compute exact contributions. Moreover, some practical issues were discussed for the applicability of this method to airborne TPA such as influence of the correlation between the sources, effect of the different type operational conditions, and influence of the neglected source. In the second study [25], the same decomposition method based on the transmissibility matrix was applied on a vehicle to quantify the airborne transfer paths. The consequences of airborne contribution analysis acquired by the transmissibility approach were compared with ones from the classical synthesis

technique obtained by the MIM. Furthermore, the study indicates that the position of indicator sensors affects the computed contributions. The results of the transfer path contribution analysis show that the sum of calculated contributions using the explained method matches very well with the measured noise at the receiver location. Hence, the decomposition method based on the determination of the transmissibility matrix from operational data gives better results in the source separation and overall response synthesization.

Kocoglu [26] compared the classical and Operational TPA methods in the scope of his M. Sc. thesis study. For this reason, the experimental applications were carried out on a prepared test configuration under two different boundary conditions. In the applications of classical TPA, the FRF measurements were performed while the system was on the nonworking condition. Also, the active and passive sides of the system were separated from each other to cut the interaction between transfer paths during the stage of FRF measurements. Hence, the results obtained by using the classical TPA method did not represent the actual system dynamics. Otherwise, in the applications of Operational TPA, the measurements were conducted on the real operating conditions and the FRF were obtained using the operational data. Therefore, the results determined by employing the Operational TPA method represented exactly the system dynamics best.

Lindberg et al. [27] investigated the effect of low sliding speed and low brake pressure on the interior vehicle roughness noise. Here, the roughness noise caused by the disc brake system of the vehicle was denoted as a structure-borne noise. The experimental study did not deal with high brake pressure and high sliding speed to prevent the airborne background noise caused by tyre/roller interaction. TPA was preferred in this study to observe the static brake force in the suspension system of the vehicle. Due to the complexity of examined system and the impossibility of obtaining the exact static preloads in the suspension system during measurement of the NTFs, the NTF matrix may not accurately characterize the operational system. Thus, the vibro-acoustic transmission path cannot be correctly specified by using classical TPA method in this study [27].

Biermayer et al. [28] introduced two TPA optimization procedures called "Mountwise TPA Approach" and "TPA From Operational and Reciprocal Measurements (TPA-FORM)" to enhance the correctness and effectiveness of TPA

processes. Generally, falsities in TPA processes are originated from experimental errors (excitation position, excitation direction, temperature difference between consecutive experiments) and other errors originating from computational aspects of TPA (crosstalk recognition, inertance matrix inversion, etc.). According to [28], experimental and numerical errors may result in approximately 10 dB variety between synthesized and evaluated source contributions to interior vehicle noise. In Mountwise TPA Approach, using the inertances within one mount to compute apparent mass was recommended to diminish condition number for reduction in error amplification. In TPA-FORM method, the inertances were obtained from operational measurements and reciprocally measured FRFs to determine the required operational forces and source contributions. Thereby, the more accurate contribution analysis was made in operational conditions.

Elliot et al. [29] outlined a faster TPA method named as in-situ Source Path Contribution (iSPC) analysis by testing the attachment between the suspension system and body of a vehicle to predict the structure borne noise inside the vehicle. The analysis was performed with neglecting the contributions of airborne noise. The operational tests were carried out without disassembling the mechanical system. Hence, the testing time was reduced by avoiding separation of the system into subsystems. This work concludes that comprising both translational and rotational degrees of freedom in the iSPC analysis improves the SPC verification.

Guash [30] represented the theoretical framework of the Global Transfer Direct Transfer (GTDT) TPA and applied the method to a mechanical system made of springs, dampers and masses. Force identification is unnecessary in the implementation of the GTDT method of TPA where the force TPA methods such as MIM, MSD require force determination. After applying the GTDT approach, a path blocking analysis was performed on the mechanical system by taking off some system parts and by giving zero displacement to a system mass. Hence, if the blocked transmission paths on the discrete mechanical system are known, the displacement of any system mass should be expressed with respect to the displacements of the leftover masses together with the displacement owing to the external forces affecting on it.

Guasch et al. [31] demonstrated the theoretical background of the Global Transmissibility Direct Transmissibility (GTDT) approach to classical TPA and

experimentally proved the method by implementing on seven degrees of freedom (DOF) mechanical system. In the study, the connection between global and direct transmissibilities, and their relation with the transfer and dynamic stiffness matrices were examined. Moreover, the way of using the direct transmissibility of the system was indicated to decompose the response at any system DOF in terms of the remaining system DOFs and of its reaction to the external forces affecting on it.

1.2.2 Operational transfer path analysis

Although classical Transfer Path Analysis (TPA) is a signal processing method that consists of two steps, Operational Transfer Path Analysis (OTPA) is a one-step method where FRF measurements using hammer, shaker or loudspeaker are not required to describe the dynamic behavior of a mechanical system. When OTPA technique is implemented to the system for quantification of vibro-acoustics transmission paths, only the measurements of chosen input (reference) and output (response) signals are required. Afterwards, the matrix of Transfer Functions (TFs) is estimated by using the measured operational data, hence the method is defined as OTPA which uses the transmissibility (response to response TF) approach.

The studies carried out about OTPA in the literature are generally interested in identifying the dominant vibration and/or noise source and transfer paths by contribution analysis. Essentially, the origin of OTPA technique initiates with the studies of Bendat et al. [32, 33] at the beginning of the 1980's [11, 34]. The OTPA theory resembles the Multi Input/Multi Output (MIMO) technique used in Experimental Modal Analysis (EMA), both of these method aims to find the linearly independent TFs [11, 35, 36].

De Klerk and Ossipov [11] presented the theory of the OTPA improved with the Singular Value Decomposition (SVD) method and applied the method on a vehicle to investigate the tire noise propagation. Firstly in the theory of OTPA, the estimation of the TF matrix was explained as a least-squares approximation problem. The article revealed that the smallest singular values were caused by measurement noise, so that the use of SVD algorithm was advised owing to large coherence between input signals. Secondly, the theory of OTPA was improved with SVD to reduce the influences of measurement noise on the estimation of correct TF matrix.

Furthermore, the consequences of the tire noise analysis indicated that the OTPA did not rely on the operational condition such as different tire type or road condition.

De Klerk et al. [35] investigated the applicability of the OTPA on a classic car to specify crucial source contributions and vibration paths inside the vehicle. The operational conditions were simulated in run-up type such as Wide Open Throttle (WOT) and Partial Open Throttle (POT). The experiment was initiated with the measurement of the vehicle on a dynamometer, and then the synthesized outputs were compared with the actual measured outputs to demonstrate if the defined sources patterned the actual response of the system. The second and third phases of the OTPA technique using SVD are analyzing the main path contributions as averaged path contributions and path contributions in individual operating conditions. According to the results of the contribution analysis, the component which had the highest average contribution in the frequency range of interest was determined.

Yan and Jiang [37] combined the Operational Path Analysis (OPA) with Partial Singular Value Decomposition (PSVD) to examine the sound quality contribution of panel radiation noise. The prevailing sound source was identified from examination of the contributions of sound sources to individual sound quality property such as sound intension, loudness and sharpness. One of the highlighted points in the study is that if the OPA method is performed on a Multi Input/Single Output (MISO) system, the PSVD analysis can be employed to estimate the TFs for the characterization of the main inputs and the determination of independent noise source.

OTPA has several advantages over classical TPA. Classical TPA includes two steps which mean that TFs and operational excitation forces are obtained in separate stages. Hence, the method is a complex procedure and the measurement takes too long time. Moreover, the measurement of TF is made when the system is in the non-operating condition. On the other hand, Operational TPA is a one-step method and uses the measured operational data to estimate TFs. Therefore, the implementation of this method to a mechanical system is easier than the classical one. Also, the dynamic behavior of the mechanical system can be modeled well owing to estimation of TFs on the operating condition.

Lohrmann and Hohenberger [38] tested the interior sound pressure of a car and compared four different TPA approximations. In the first case, the Matrix Inversion

Method (MIM) was used and the impact test was utilized with removing the car engine to obtain the Transfer Functions (TFs) of the vehicle. In the second case, the MIM was used again, but the impact test was utilized without removing the car engine to measure the TFs of the vehicle. In the third case, the Crosstalk Cancellation (CTC) method was used and the impact test was applied to the complete vehicle to measure the TFs. In the last case named as OTPA using CTC, the CTC method was used again, but the TFs of the vehicle were estimated from operational data. Due to the correct synthesis of interior sound pressure of the vehicle in real load conditions, best results were acquired in the last case.

De Sitter et al. [36] deduced the theory of the OTPA technique from simple mechanical models by proving the procedure with the help of finite element simulations. According to the study, the technique can be used without disassembling the mechanical system and eliminating operational forces that act on the mechanical system. Thus, the technique has advantages over the classical TPA techniques, which are reducing the total testing time and keeping the real boundary conditions. Furthermore, carrying out the OTPA technique using a deficient group of references, the effect of the number of measurements and uncorrelated forces, and applicability of this method for more complex systems were examined in the study.

Despite the fact that OTPA technique is a time-saving procedure due to use of operational data for estimation of the TFs, it suffers from missing active vibro-acoustic transmission paths, incorrect transmissibility and/or TF estimation, and ill-conditioning problem due to inversion of TF matrix. In the literature, OTPA technique is enhanced with the use of some additional techniques such as parametric load models, SVD, transmissibility approach, coherence analysis, etc. to overcome these problems.

Janssens et al. [34] introduced a different TPA approach called as Operational Path Analysis with Exogenous Inputs (OPAX) that utilizes parametric load models to specify the operational loads. To examine and verify the OPAX method, a simulation study was performed on a 4-cylinder car engine in run-up condition. The OPAX method includes 5 main stages for the evaluation of transfer path contributions for each target [34]. In the first stage, operational measurements are performed to determine path inputs, targets and indicator responses. In the second stage, FRF measurements are performed to obtain FRF's to targets and extra indicators. In the

third stage, parametric load models are identified to express operational loads as a function of measured path inputs. In the fourth stage, operational loads are computed from measured path inputs and identified load models. In the last stage, path contributions are calculated for each target. The prediction of the parametric load models is a numerically stable process [34]. Thus, the use of the parametric load model estimation is suggested to avoid from ill-conditioning problems caused by pseudo-inversion of rank-deficient datasets.

Kim et al. [39] used the OTPA with coherence analysis for identifying the vibration sources in the vehicle suspension system. In this study, a MISO system was developed to model the road noise transmitted to the passenger compartment via the vehicle suspension system. The OTPA method and the coherence function in the frequency range of interest were carried out together to estimate optimum TFs and the coherent output spectrum. As a result, the major transfer path and vibration source were identified to optimize the system employing Progressive Quadratic Response Surface Method (PQRSM) for road noise reduction.

Gajdatsy et al. [40] examined applying the transmissibility approach to OTPA and demonstrated the limitations of implementing this method for a mechanical system. The ratio of two outputs (responses) of a system is stated as transmissibility and it is implemented to OTPA in two different ways. The first way is using the transmissibilities to determine the NTF in place of measuring them (indirect application). The second one is a response-response type approach where transmissibilities are straightly implemented in the OTPA procedure (direct application). According to [40], the direct application to OTPA requires for some preconditions to have well-estimated transmissibilities such as (i) low coherence between input forces, (ii) low cross-coupling between input responses and (iii) including all active paths.

Roozen and Leclère [41] investigated a two-step transmissibility-based OTPA procedure where conventional OTPA procedure is a one-step method. In this procedure, as a first step, an artificial excitation is applied to the machine not in operation and resulting vibro-acoustic responses are used to calculate the transmissibility matrix. The second step of this procedure is implementing conventional OTPA using these computed transmissibility matrix. Moreover in the study, the procedure was implemented on a small gear-box test setup to determine

the number of important transmission paths based on the normalized singular value distribution.

1.3 Objectives and Outline of Thesis

In the scope of this thesis, it is aimed to understand the fundamental utilization steps of the Operational Transfer Path Analysis (OTPA) procedure in order to determine vibration contributions from various transmission paths to a defined receiver location by performing this method on a large home appliance under different operational conditions. It is also aimed to implement the OTPA procedure as a MATLAB code and to validate this using numerical and experimental data.

This thesis contains five different chapters. In the first chapter, as an introduction to the study, the signal processing technique used in the content of this thesis is briefly introduced with a definition of the problem. A literature review of the experimental and numerical applications about the TPA is presented. The positive and negative sides of classical TPA and OTPA methods are explained briefly. The main objectives and outline of the thesis are stated.

In the second chapter, the background theory of classical TPA and OTPA is examined. First of all, the classical TPA theory, consisting of two main stages as identification of Frequency Response Function (FRF) and determination of operational input loads, is presented. Moreover, the ways of how to determine the operational structural and acoustic loads for the classical TPA method are explained in some detail. Secondly, the OTPA theory is introduced using Least-Square Algorithm (LSA) and Singular Value Decomposition (SVD) techniques to estimate Transfer Functions (TFs) and synthesize the overall response.

In the third chapter, an application to a simple flat plate is numerically and experimentally performed utilizing the OTPA procedure to verify the MATLAB code developed in this study. The flowcharts for OTPA procedure for both numerical simulations and experimental studies are schematically illustrated. The preparation of numerical model and measurement setup used in experimental applications are described. During experimental and numerical case studies, forced vibration analysis as well as dynamic analysis is applied to the test structure in order to obtain the required data for comparison purposes. LSA and SVD techniques are implemented

into the OTPA algorithm to estimate operational TF matrix. The importance of using phase-assigned spectrum and linear averaging is explained with simple demonstrations. For the verification, the measured and synthesized overall responses are firstly compared with each other. After that, the computed FRFs and their phase angles using Single Input/Multi Output (SIMO) model assumption are combined with estimated FRFs and their phase angles utilizing Multi Input/Multi Output (MIMO) model assumption.

In the fourth chapter, a washing machine application is carried out employing the OTPA procedure to quantify and rank transfer path contributions from a main excitation source to a selected group of response points. For this reason, establishing the "source-transfer paths-receiver" model, the selection of appropriate input and output sensor positions, the ways of altering operational conditions are clarified with the justifications and assumptions. Vibration measurement and modal analysis are made on the mechanical system to define the direction of vibration transmission from the source to the receiver and possible accelerometer measurement points on the receiver, respectively. The operational measurements are repeated for two different types of operational condition set considering two types of damping units (single effect vs. double effect). Finally, the singular value examination, overall response comparison (measured vs. calculated) and contribution analysis are made in frequency range of interest to determine the critical vibro-acoustic transmission paths.

In the final chapter, main results and findings are discussed and future recommendations for further studies are presented.

2. THEORETICAL BACKGROUND

Transfer Path Analysis (TPA) - an experimental approach based on Frequency Response Function (FRF) measurement - is used to determine the interior sound pressure and/or vibrations in the whole structure by summing up the entire vibro-acoustic transmission path contributions of a particular system. The contributions of the individual transmission paths are computed by multiplication of the evaluated FRFs with the determined operational loads from a measurement, e.g., on a test bench. TPA is considered as an effective method for finding the most dominant transfer paths in the structure to initiate the component optimization [42].

2.1 Classical Transfer Path Analysis

Transfer Path Analysis (TPA) technique is utilized especially in the automotive industry to examine various contributions of vibration and/or noise at a specific receiver position such as the positions of the driver and the passengers in a vehicle. The purpose of the method is to classify transmission paths from different structure-borne and airborne noise sources like engine, intake, exhaust, tire, gearbox noise by making use of the linearity between load and response [43].

The theoretical framework of classical TPA focuses on two subjects: identification of FRF and determination of operational loads. As mentioned before, this experimental method is utilized to define the vibro-acoustic transmission paths in a system. The transmission of this vibro-acoustic energy initiates from the active part of examined system, where the structural and acoustic loads excite the system. The excitation is transmitted through the physical coupling locations and then via airborne transfer paths to the response points at the target locations that are in the passive part of the examined system [34]. Total path contributions, each connected with a single path and operational load, defines the vibration and acoustics responses at the target locations such as seat vibration, interior noise, steering wheel vibration. For instances, a target response are demonstrated as [34]:

$$y_k(\omega) = \sum_{i=1}^n y_{ki}(\omega) \quad (2.1)$$

where

$y_k(\omega)$: the total response at target point k

$y_{ki}(\omega)$: the path contribution of path i

ω : frequency

n : the number of paths

A source-system-receiver model, improved as an alternative formulation in the late 1980s, indicates the every fractional response contribution as the consequence of a singular structural or acoustic load affecting at a positioned interface, and reacting of a system to this interface load [34, 44-46]. The global response is provided by gathering the different path contributions in a vector form. For this reason, the global system is separated at the interface into an active and a passive system. The interface load is generated on the active part and this load generated response on the passive part (Figure 2.1). The acoustic loads from vibrating surfaces or pulsations from nozzles or openings are decomposed by employing omni-directional volume acceleration point sources [34, 47, 48].

Every fractional contribution is defined by the source-system-receiver model as the consequence of the loads generated at every contribution location and FRF from the load location to the response at the target location. Therefore, the TPA formulation, valid for linear and invariant systems, is given as [34]:

$$y_k(\omega) = \sum_{i=1}^n H_{ki}(\omega)F_i(\omega) + \sum_{j=1}^p H_{kj}(\omega)Q_j(\omega) \quad (2.2)$$

where

$y_k(\omega)$: the total response at target point k

$F_i(\omega)$: the structural interface force

$Q_j(\omega)$: the acoustic load (volume velocity or acceleration)

$H_{ki}(\omega)$: the uncoupled passive side FRF from the input structural loads to the target

$H_{kj}(\omega)$: the uncoupled passive side NTF from the input acoustic loads to the target

ω : frequency

n : the number of structure-borne paths

p : the number of airborne paths

As can be seen in the formulation of classical TPA in Eq. 2.2, this method consists of two steps: (i) measuring the FRFs and (ii) estimating the structural and acoustics operational loads. Thus, the theoretical background of classical approach is dealt with two subsections: identification of FRFs and determination of operational loads [34].

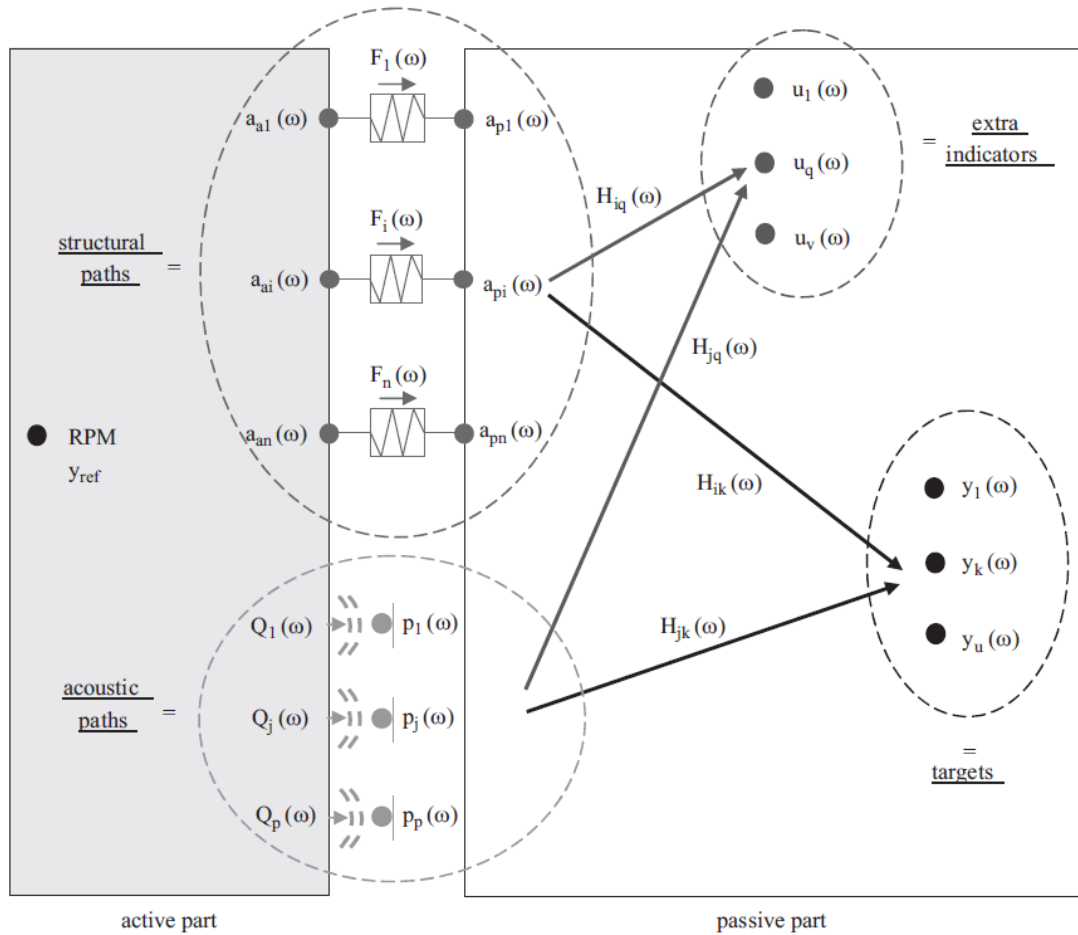


Figure 2.1 : Active and passive parts of the system [34].

2.1.1 Identification of frequency response function

In the classical TPA technique, the Frequency Response Functions (FRFs) of the different transmission paths between source and receiver location are obtained by applying a known artificial excitation on the source location and measuring the response at the receiver location. Generally, a shaker is preferred as a vibration source and a loudspeaker is used as an acoustic source for the artificial excitation. The response (output) to the reference (input) structure-borne load (e.g., excitation force) and/or airborne load (e.g., volume velocity or acceleration) results in FRFs which are obtained by the feature of the linear Multi Input/Multi Output (MIMO) system that is connected with load-response model [43, 49]. Moreover in the classical TPA technique, reciprocal measurement is also used for identification of FRFs where the excitation is implemented on the receiver location and the response is measured at the source location [43, 50]. On the other hand, the FRFs are obtained directly from the operational forces if the use of an artificial excitation (e.g. shaker, loudspeaker) is impossible. In such cases, the operational forces must be exactly obtained from measurement and/or evaluation to correctly identify the FRFs [36, 40, 43, 51].

According the modal testing theory [51], free vibration and forced vibration analyses are two consecutive phases in a complete vibration analysis. For the Single Degree-of-Freedom (SDOF) system, the result of a free vibration analysis gives its modal properties (i.e., natural frequency and damping factor), while a forced vibration analysis, considering a harmonic excitation, results in the determination of the frequency response characteristics of the system. Therefore, the Frequency Response Function (FRF) of the SDOF system is determined by performing forced vibration analysis on the system. It is defined as the ratio of response to force input [51]. The SDOF system is utilized to describe the general expressions and vibration analysis steps for the Multi Degree-of-Freedom (MDOF) systems that have more than one DOF. The first step in the vibration analysis of a MDOF system is establishing the governing equations of motion to determine the spatial properties of this system. The second step is applying a free vibration analysis employing the equations of motion to identify the modal properties of the MDOF system. The third and last step of theoretical vibration analysis is performing the forced vibration analysis to determine the frequency response characteristics of the MDOF system [51]. As a result, the

solution of the equations of motion, when the MDOF system is excited by an input force, hands over the frequency response matrix $[H(\omega)]$. Each element of the frequency response matrix, $H_{jk}(\omega)$, is frequency-dependent and defined as a Frequency Response Function (FRF). It can be expressed in a general form by using the following statement [51]:

$$H_{jk}(\omega) = \frac{X_j}{F_k} \quad (2.3)$$

where X_j represents the response in one of the DOFs, j , induced by an excitation load, F_k , applied on a various DOF, k . All of these variables including $H_{jk}(\omega)$ are expressed utilizing complex algebra to associate the magnitude and phase information [51].

After identification of FRFs, the estimated response at the receiver is evaluated by combining the operational force with the measured FRFs. With an assumption of linearity between excitation and response location, the response at the receiver (e.g. sound pressure level or vibration) is obtained by [43]:

$$\mathbf{y}(\omega) = \mathbf{H}(\omega)\mathbf{x}(\omega) \quad (2.4)$$

where

$\mathbf{y}(\omega)$: the output (response)

$\mathbf{x}(\omega)$: the input (reference)

$\mathbf{H}(\omega)$: the Frequency Response Function (FRF)

The FRF defines the Transfer Function (TF) between reference and response in the classical TPA approach. Besides, the TF is also defined as the Noise Transfer Function (NTF) and/or Vibration Transfer Function (VTF) [42, 43].

It should be noted that the method must consist of all source-path-receiver groups in the experiment to evaluate the complete response at the receiver location. This means that measurement of all noise sources and associated paths is required. The total response at the receiver point is a superposition of all the individual contributions and is determined by adding up the contribution of the result of all

input excitations and transmission paths. The formulation of the total response (i.e. sound pressure and/or vibration) at the receiver location is expressed as [43]:

$$y_m(\omega) = \sum_{n=1}^N H_{mn}(\omega)x_n(\omega) \quad (2.5)$$

where the input excitation force is implemented to the transfer path n and corresponding response is occurred at the receiver point m .

Generally, a structural force or a volume velocity is referred as the input excitation load, and a sound pressure or vibration (e.g. acceleration) is referred as the response. When both structure-born and airborne contributions are interested in the analysis, the total response at the receiver point is described in terms of both structural and acoustic input loads as [43]:

$$p_m(\omega) = \sum_{n=1}^k H_n(\omega)F_n(\omega) + \sum_{j=1}^r H_j(\omega)Q_j(\omega) \quad (2.6)$$

where $p_m(\omega)$ denotes the total response (i.e. sound pressure) at the receiver location m , $F_n(\omega)$ denotes the structural load (i.e. excitation force) at the input location n , and $Q_j(\omega)$ the acoustic load (i.e. volume velocity) at the input location j . Here, the force input is the source of structure-borne noise, as well as the volume velocity input is the source of airborne noise [43]. Therefore, the structure-borne and airborne contributions can be computed in the same formulation for the classical TPA technique.

2.1.2 Determination of operational loads

The first stage of classical TPA method is to obtain the FRFs between the defined set of input and output locations for system identification. Afterwards, the operational loads must be determined to calculate the total response at the receiver point. There are two ways to estimate the operational loads: direct methods and indirect methods. In direct methods, the operational force is directly measured at the excitation point. For example, if a shaker is used as a source of artificial excitation applied to the system at input location, the input force is directly measured by a force transducer.

Indirect methods are usually preferred when it is not possible to measure the operational loads both structural and acoustic directly. For instance, excitation force acting on a mount cannot be directly measured due to inappropriate place to locate a force transducer for an exact force measurement. In such cases, indirect methods are required for obtaining the structural and acoustic operational loads [3].

2.1.2.1 Indirect methods for operational loads

Structure-borne vibro-acoustic transmission paths are quantified by the operational input forces acting on them. When it is not feasible to measure the operational forces directly (e.g. insufficient physical space to place the force transducers between two sources), they are determined using indirect techniques. Basically, there are two indirect methods to estimate operational forces: Dynamic (Complex) Stiffness Method (DSM) and Matrix Inversion Method (MIM) [3].

The Dynamic Stiffness Method (DSM), also defined as Mount or Complex Stiffness Method, is utilized for transmission paths that include resilient connections such as mounts that connect the source sides to the receiver sides. Operational forces are estimated from a known complex dynamic stiffness of the mounts and the differential displacement over the mount during operation. Generally, the displacements are obtained by double integration of measured accelerations. The operational force using DSM is determined as [3]:

$$f_i(\omega) = K(\omega) \cdot (x_s(\omega) - x_t(\omega)) \quad (2.7)$$

where

$f_i(\omega)$: the operational force at transfer path i

$K(\omega)$: the complex dynamic stiffness as function of frequency

$x_s(\omega)$: the operational displacement at the mount connection point on the source side

$x_t(\omega)$: the operational displacement at the connection point on the receiver side

It is essential to measure the operational displacement at both the source and the receiver sides if the DSM is to be utilized. Therefore, locating the accelerometer as close as possible to the mount link points is also important. If the acceleration signals are measured further away from the mount connection points, there can be an

inaccurate estimation of operational loads at higher frequencies. Furthermore, the complex dynamic mount stiffnesses are supposed to be positive either in compression or in tension. They should be able to be obtained as complex data as functions of frequency. Actually, the mount characteristics are defined in terms of force/displacement, alternatively force/velocity or force/acceleration terms can be used to describe the mount characteristics. During the measurement of the dynamic stiffness of mounts, the mounts are pre-loaded as close to the real operational conditions as possible. Furthermore, the temperature should be kept close to the operating temperature [3].

On the other hand, providing a necessary amount of relative displacement for a mount may be very difficult when transmission paths have very rigid connections or when the mount stiffness is very large. For these situations, a method that based on an accelerance matrix inversion measured between structural responses on the receiver side owing to force excitation at all transmission paths is utilized. The source is decoupled from the receiver to measure this accelerance matrix. Afterwards, the measured accelerance matrix is multiplied with operational measurements of structural vibration at the receiver side to determine the operational forces. Singular Value Decomposition (SVD) method is frequently used to reduce numerical problems during the matrix inversion process. The operational force using Matrix Inversion Method (MIM) is determined by [3]:

$$\begin{Bmatrix} f_1 \\ \dots \\ f_n \end{Bmatrix} = \begin{bmatrix} \frac{\ddot{X}_{11}}{F_1} & \dots & \frac{\ddot{X}_{1n}}{F_n} \\ \vdots & \ddots & \vdots \\ \frac{\ddot{X}_{m1}}{F_1} & \dots & \frac{\ddot{X}_{mn}}{F_n} \end{bmatrix}^{-1} \begin{Bmatrix} \ddot{x}_1 \\ \dots \\ \ddot{x}_m \end{Bmatrix} \quad (2.8)$$

where

$\begin{Bmatrix} f_1 \\ \dots \\ f_n \end{Bmatrix}$: the vector of operational forces applied at the input location

$\begin{Bmatrix} \ddot{x}_1 \\ \dots \\ \ddot{x}_m \end{Bmatrix}$: the vector of operational accelerations at the receiver side

$\frac{\ddot{X}_{11}}{F_1}$: the accelerance FRF measured between the acceleration at location i and

a force applied at transfer path j

Implementation of the accelerance MIM requires measuring the receiver-side vibrations in three translational directions under operating conditions. The number of responses (m) ought to at least be equal to the number of predicted input forces (n) to get a unique solution for the operational forces. If the locations of measurement are not limited to the locations of transfer paths, more response measurements can be taken at the receiver side. In such cases, the number of responses (m) becomes greater than the number of predicted input forces (n) that leads to the over-determination of the group of equations where least-squares algorithm is used to determine the operational forces [3].

There is a third indirect method for the determination of operational forces. Actually, this method is the simplified version of the MIM. The operational force for every defined Degree-of-Freedom (DOF) is easily determined by combining the measured point transfer function TF with quantified response at the receiver location. The operational force using the simplified version of MIM is estimated as [43]:

$$F_i = Z_{ij}\ddot{x}_j \quad (2.9)$$

where

F_i : the predicted force input at point i

Z_{ij} : the TF of the force at point i to the response at point j

$\ddot{x}_j$: the quantified response (acceleration) at point j

Despite the fact that this method is easy to utilize, the cross-talk effect at lower frequencies can cause the inaccurate operational force determination [43]. The reason is that the cross-talk contribution from forces exciting at the remaining DOFs can be relatively large at lower frequencies [2, 43].

2.1.2.2 Indirect methods for operational volume velocities

In acoustics, the volume velocity of a source is the ratio of the pressure to the acoustic impedance. It is also defined as the multiplication of the surface area of the source with the normal surface velocity [52]. This definition is solely accurate for flat, plane cross-sections. Generally, containing curved surface, the volume velocity is obtained from a surface integral as [53]:

$$Q = \iint_A \mathbf{V} \cdot d\mathbf{A} = \iint_A \mathbf{V} \cdot \mathbf{n} dA \quad (2.10)$$

where

Q : the volume velocity of the source

$\mathbf{V}$: the normal surface velocity

$\mathbf{A}$: the cross-sectional vector area/surface

$\mathbf{n}$: the unit vector normal to the surface

To obtain the contribution of the source at the receiver location, the volume velocity of the source in operation is multiplied with the TF from the receiver to the source. To determine this TF reciprocally, a known volume velocity is located at the position of the receiver and corresponding pressure at the source is measured. The ratio of the volume velocity at the receiver location to the measured pressure at the source position yields the reciprocal TF. Therefore, the volume velocity can be used to track airborne route. It is essential to accurately determine the effective volume velocity of the source for correct airborne transfer path tracking [52].

Volume velocities which are used to quantify the airborne transmission paths are determined from indirect techniques, in a way similar to the determination of operational forces. There are three indirect methods that can be applied to determine volume velocities: point-to-point surface sampling, sound intensity measurements and matrix inversion. When the airborne contributions are evaluated from a vibrating panel, point-to-point surface sampling is used, and when the airborne contributions are evaluated from an engine, sound intensity measurements are implemented. Moreover, the third method, matrix inversion is applied for identifying the intake and exhaust noise [3].

In point-to-point surface sampling method, the equivalent volume velocity of a defined source can be derived by dividing the global radiating surface into individual sampled scanning surfaces, as illustrated in Figure 2.2. The measured acceleration at a defined point in the normal direction on the surface S is assumed as constant acceleration profile utilized in a given surface area to obtain the volume acceleration of that surface in this method. This assumption necessitates dividing the global radiating surface into individual surfaces (patches). The volume acceleration for every sampled surface S_j is evaluated as [3]:

$$\dot{q}_j = S_j \ddot{x}_{jn} \quad (2.11)$$

$\dot{q}_j$: the volume acceleration of patch j

S_j : the surface area of patch j

$\ddot{x}_{jn}$: the acceleration measured at patch j in the normal direction

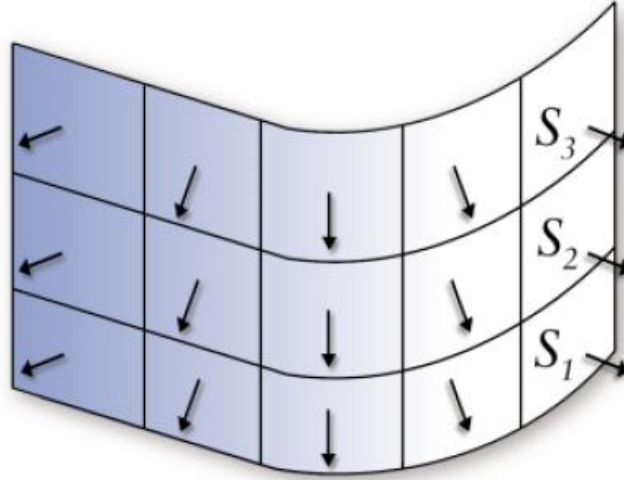


Figure 2.2 : Measurement of the acceleration on panel patches [3].

The volume velocity is described as the movement induced by a sound wave of a unit volume of a sound-transmitting medium through a unit area per unit of time. The operational volume velocities are computed by integrating the volume accelerations obtained from point-to-point surface sampling method. It should be noted that the measured accelerations, $\ddot{x}_{jn}$, must be the surface normal accelerations. As an obligation of measurement aims, the continuous radiating source is sub-divided into

discrete patches where the smallest acoustic wavelength of concern must be at least 2 times bigger than the dimension of these patches. Therefore, the frequency range is associated with the dimension of the patches [3]. In addition, because of using this method for quantification of the contributions from vibrating panels in general, point-to-point surface sampling method is also called as Panel Contribution Analysis [3, 54]. For applying the point-to-point surface sampling technique, the global radiating surface has to be separated into a number of well-defined sub-areas that provides measuring or evaluating the normal acceleration and surface area. As an example, this method is applied to determine the contributions from the various panels in a car cavity [3].

The second indirect method used to determine the operational acoustic loads (volume velocities) is sound intensity measurements. Near-field sound intensity measurements on the source under steady-state conditions should be conducted in an anechoic room, except for in-situ measurements. Therefore, the sound power of the source located in anechoic conditions can be evaluated to obtain estimation for the phaseless volume velocity of the noise source. The volume velocity of the source is considered as invariable with varied acoustic boundary conditions. The mean intensity in a measurement surface S around the noise source is determined using intensity measurements. The time-averaged product of sound pressure and particle velocity yields the mean sound intensity [3]:

$$\vec{I} = \frac{1}{T} \int_0^T p(t) \vec{v}(t) dt \quad (2.12)$$

where

$\vec{I}$: the mean sound intensity vector

$p(t)$: the sound pressure

$\vec{v}(t)$: the particle velocity vector

For the determination of partial contributions, the total surface area S is divided into m sub-areas S_j . The radiated sound power from a sub-area S_j is obtained by integrating the scalar product of the mean sound intensity vector and the normal vector over that sub-area S_j [3]:

$$Pow_j = \int_{S_j} \vec{I} \vec{n} dS \quad (2.13a)$$

where

Pow_j : the radiated sound power from the sub-area S_j

$\vec{n}$: the normal vector for the sub-area S_j

As a result, the sound power Pow_j is determined. If a monopole source radiating into free space is used instead of the radiating source, the following formula is applied to obtain the source strength as [3]:

$$Q_j^2 = Pow_j \frac{4\pi c}{\rho \omega^2} \frac{1}{C} \quad (2.13b)$$

where

Q_j^2 : the source strength

c : the sound velocity

ρ : the air density

In addition, C used in Eq. 2.13b is a factor that is entered by the user. The default value of C is 2 that means a monopole radiating into half space, which is the most general circumstance. Owing to evaluating the source strength (Q_j^2), there is no phase information connected with the determined source strength in this method. As a comparison, sound intensity measurement has fewer limitations than point-to-point sampling owing to complexity of the radiating surface. This method provides better results excluding the low frequency range. On the other hand, the disadvantage of this technique is that the sound intensity measurements have to be made under stationary conditions. Thus, it is impossible to perform the sound intensity measurements under non-stationary conditions such as run-up/run-down measurements [3].

Another indirect way to determine the volume velocity of a patched radiating surface is Matrix Inversion Method (MIM). Measurement of the indicator pressure responses

is made near to the radiating surface in operating conditions. Evaluation of the operational volume velocity of the patched radiating surface requires processing the measured pressure responses together with near-field TFs which are between indicator pressures at the response locations and volume velocities at the radiating surface. Reciprocal measurement of the TF matrix is made by locating microphones at the radiating surface, and loudspeakers at the indicator position (Figure 2.3) [3].

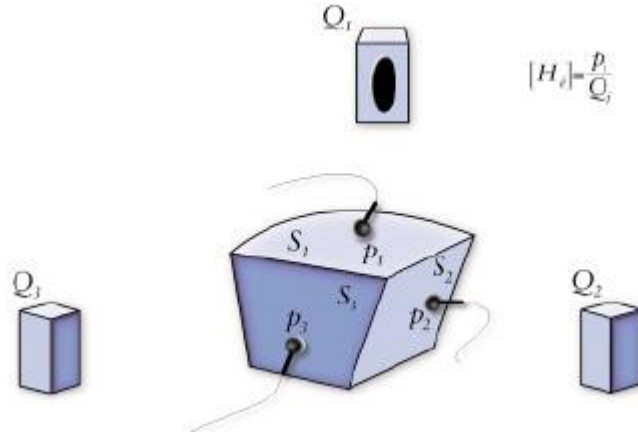


Figure 2.3 : Reciprocal measurement of the TF matrix [14].

The formulation for the explained method is represented as [3]:

$$\begin{bmatrix} Q_1^{oper} \\ \vdots \\ Q_n^{oper} \end{bmatrix} = \begin{bmatrix} \frac{p_1}{Q_1} & \dots & \frac{p_1}{Q_n} \\ \vdots & \ddots & \vdots \\ \frac{p_m}{Q_1} & \dots & \frac{p_m}{Q_n} \end{bmatrix}^{-1} \begin{bmatrix} p_1^{oper} \\ \vdots \\ p_m^{oper} \end{bmatrix} \quad (2.14a)$$

$$P_r^{oper} = \sum_{i=1}^n \frac{p_r}{Q_i} Q_i^{oper} \quad (2.14b)$$

where

Q_i : the operational volume velocity measured at patch i

p_j : the operational pressure responses at microphones j

$\frac{p_j}{Q_i}$: the TF between the volume velocity measured at patch i and the pressure at microphone j

In this case, placing the loudspeaker at location i close to the surface is ordinarily impossible. Thus, the TFs are obtained by reciprocal measurement where the loudspeaker is placed at location j and the microphone is placed at location i . In the MIM, if the number of indicator pressures (m) is received higher than the number of equivalent Volume Velocity Sources (VVS) (n), this situation provides over-determination that enhances the matrix inversion. As a result, the matrix inversion used for obtaining operational forces is technically the same as this method. As a comparison, the MIM can be implemented both under stationary and realistic conditions (source in situ). On the other hand, if the target and pressure indicator microphones are located in the same cavity, this situation can cause ill-conditioned problem. Moreover, so much TF have to be measured for the inversion method and the condition of the matrix inversion may also be a trouble [3].

2.2 Operational Transfer Path Analysis

Operational Transfer Path Analysis (OTPA), utilizing Crosstalk Cancellation (CTC) and Singular Value Decomposition (SVD), is a signal processing technique where the linearized Transfer Functions (TFs) are determined between a group of selected input and output channels from measurements. The algorithm of OTPA compensates for crosstalk between input channels, and the TFs are linearly independent in accordance to each other, thus the process is named as Crosstalk Cancellation or Compensation (CTC). Finding the linearized TFs provides identification of the input and output relationship of the system. The OTPA approach is enhanced with Singular Value Decomposition (SVD) method to reduce the effects of measurement noise. The aim of using SVD method in OTPA approach is to obtain independent principal components defining the Transfer Functions (TFs). The influences of measurement noise on the estimation of TFs can be reduced by removing smaller principal components [11].

De Klerk and Assipov [11] presented the theory of OTPA based on Crosstalk Cancellation (CTC)/Principal Component Analysis (PCA), which aims to find out the linearized TF matrix between chosen measured excitation (inputs) and responses (outputs) of an object described by Degrees-of-freedom (DOF). Moore [43] presented the theoretical background of OTPA including PCA technique. As a first phase, the TF matrix assignment is defined like a Least-Square Algorithm (LSA)

problem. As a second phase, the theory is developed with SVD algorithm to minimize the noise effects on the TF matrix assignment. As a third stage, PCA is implemented into SVD algorithm to eliminate measurement noise and crosstalk influence in the TF estimation. At the last stage, the overall response is synthesized from the noise eliminated TFs.

2.2.1 OTPA theory with least-square algorithm

Considering an arbitrary linear(ized) system model characterized by a group of input and output DOF, indicated as [11]:

$$\mathbf{H}(j\omega)\mathbf{x}(j\omega) = \mathbf{y}(j\omega) \quad (2.15)$$

where $\mathbf{H}(j\omega)$ is the TF matrix connecting the input vector $\mathbf{x}(j\omega)$ to the output vector $\mathbf{y}(j\omega)$ and $(j\omega)$ indicates the dependency on frequency. In the conventional Noise, Vibration, and Harshness (NVH) problems, motions, stated as $\mathbf{u}(j\omega)$, forces, stated as $\mathbf{f}(j\omega)$, and sound pressures, stated as $\mathbf{p}(j\omega)$, are generally the measured signals. Therefore, the input and output vectors are usually gathered from these terms as [11]:

$$\mathbf{x} = \begin{bmatrix} \mathbf{u}_x \\ \mathbf{f}_x \\ \mathbf{p}_x \end{bmatrix}; \mathbf{y} = \begin{bmatrix} \mathbf{u}_y \\ \mathbf{f}_y \\ \mathbf{p}_y \end{bmatrix} \quad (2.16)$$

$$\mathbf{u}_x = [u_x^{(1)}, \dots, u_x^{(k)}]^T; \mathbf{u}_y = [u_y^{(1)}, \dots, u_y^{(n)}]^T \quad (2.17a)$$

$$\mathbf{f}_x = [f_x^{(1)}, \dots, f_x^{(l)}]^T; \mathbf{f}_y = [f_y^{(1)}, \dots, f_y^{(o)}]^T \quad (2.17b)$$

$$\mathbf{p}_x = [p_x^{(1)}, \dots, p_x^{(m)}]^T; \mathbf{p}_y = [p_y^{(1)}, \dots, p_y^{(p)}]^T \quad (2.17c)$$

Here, the dependency on frequency $(j\omega)$ is dropped from equations for the sake of brevity. Besides, any other physical terms can be used as input and/or output. As can be seen in Eq. 2.17a, 2.17b and 2.17c, the numbers of input channels for various terms are stated by indices k, l, m and the numbers of output channels for various

terms are stated by indices n, o, p , respectively. Identifying the input and output groups from measured data is related with how to model the system attitude. Therefore, there is no restriction to select forces, motions or sound pressures as excitation or response. Moreover, there is no obligation to represent whole physical terms in each group indicated in Eq. 2.17a, 2.17b and 2.17c. The dimension of these vectors can be different. Actually, the number of DOF in the response/output vector can be smaller than the number of excitation channels in the reference/input vector. For instance, the TFs are obtained between motions measured on the driveline and the sound pressure at the driver ears in a vehicle analysis [11].

If Eq. 2.15 is arranged, each element in the TF matrix $\mathbf{H}$ can be expressed as [11]:

$$H_{ij} = \left. \frac{y_i}{x_j} \right|_{x_k=0} ; k \neq j \quad (2.18)$$

Generally, Experimental Modal Analysis (EMA) uses this kind of TF matrix element feature represented in Eq. 2.18. In the EMA, an external force $\mathbf{f}$ enforced (e.g. x_j) by impulse hammer or shaker is selected as only input DOF (e.g. $x_k = 0; k \neq j$) and the generated response $\mathbf{u}$ of the system are selected as the output DOFs (e.g. y). Obtained TFs are indicated as receptance Frequency Response Function (FRF) whose frequency peaks represent the eigenfrequencies of the system. Such approach in the OTPA technique requires that the system has to be excited with only one input DOF x_j , meanwhile all other input excitations have to be stopped to establish a column of the TF matrix. Practically, this situation is very difficult to achieve even if the inputs are solely forces. Hence, the specification of the TFs using this approach in OTPA is very difficult and experimentally impractical as it demands so much time and resources [11].

The OTPA technique attempts to identify all elements of the TF matrix from one measurement solely while all input excitations are acting together. To examine this estimation, firstly the transpose of Eq. 2.15 is taken and the equation is rewritten as [11]:

$$[x^{(1)}, \dots, x^{(m)}] \begin{bmatrix} H_{11} & \dots & H_{1n} \\ \vdots & \ddots & \vdots \\ H_{m1} & \dots & H_{mn} \end{bmatrix} = [y^{(1)}, \dots, y^{(n)}] \quad (2.19)$$

where m and n indicate the number of reference (input) and response (output) DOFs. Although Eq. 2.19 has to be computed for each frequency line, a group of synchronized linearly independent operational measurement blocks is required for the frequency analysis. The reason is that the excitations originated from the vibration source change continuously during the measurement period. More precisely, the operational measurements for the computation of synchronous frequency spectrum for each reference and response channel have to be repeated in different operating conditions. It is assumed that the relation between the input and output DOF is linear(ized) and constant during the measurement period, Eq. 2.19 is valid for every individual measurement block. Thus, Eq. 2.19 can be expanded by composing the equation for all measurement blocks r [11]:

$$\begin{bmatrix} x_1^{(1)} & \dots & x_1^{(m)} \\ \vdots & \ddots & \vdots \\ x_r^{(1)} & \dots & x_r^{(m)} \end{bmatrix} \begin{bmatrix} H_{11} & \dots & H_{1n} \\ \vdots & \ddots & \vdots \\ H_{m1} & \dots & H_{mn} \end{bmatrix} = \begin{bmatrix} y_1^{(1)} & \dots & y_1^{(n)} \\ \vdots & \ddots & \vdots \\ y_r^{(1)} & \dots & y_r^{(n)} \end{bmatrix} - \mu \quad (2.20)$$

This statement represents the system model and necessitates that the TFs should be also linearly independent with respect to each other, so that the process is named as CTC. Actually, even if the input terms may be coherent with respect to each other, the estimation of the TF matrix satisfies for it [11].

To solve Eq. 2.20 for the $\mathbf{H}$ matrix, with Least-Square Algorithm (LSA), the number of measurement blocks has to be larger than the total number input DOFs (e.g. $r > m$). Moreover in Eq. 2.20, the residual vector term μ cannot be revealed by the selected group of input DOF [11]. Note that when various physical terms are preferred, a weighted LSA is implemented [11, 18]. Actually, individual measurement blocks can comprise an external distortion owing to, for instance, measurement noise or extra unmeasured excitation input sources which are not considered in the system model [11]. On the other hand, the TF of the system may be variable during the measurement due to nonlinear behavior of the system attitude which may depend on temperature, implemented excitation amplitudes or rotation

speed [11, 42]. Therefore, the additional residue term μ is defined to perform an accurate experiment and to setup a well-established model.

To obtain a solution for Eq. 2.20, firstly the equation is facilitated by composing it in a more clear form in matrices as [11]:

$$\mathbf{X}\mathbf{H} + \mu = \mathbf{Y} \quad (2.21)$$

The identification of H matrix is carried out every individual frequency line of the Fast Fourier Transform (FFT) spectrum. Eq. 2.21 is pre-multiplied by $\mathbf{X}^T$ to unravel the equation for each frequency. This pre-multiplication needs the residue vector to provide the null space of the reference (input) DOF, i.e. $\mathbf{X}^T\mu = \mathbf{0}$. Subsequently, the TF matrix is obtained as [11]:

$$\mathbf{H} = (\mathbf{X}^T\mathbf{X})^{-1}\mathbf{X}^T\mathbf{Y} = \mathbf{X}^+\mathbf{Y} \quad (2.22)$$

Here, matrix $\mathbf{X}^+$ is the pseudo-inverse of matrix $\mathbf{X}$, given as

$$\mathbf{X}^+ \triangleq (\mathbf{X}^T\mathbf{X})^{-1}\mathbf{X}^T \quad (2.23)$$

Since the TF matrix is obtained in Eq. 2.22, the residual vector μ can be determined. As mentioned before, this vector represents the unmodelled portion of $\mathbf{Y}$. The residual vector μ is evaluated by substituting Eq. 2.22 into Eq. 2.21 including some manipulation [11]:

$$\mu = (\mathbf{I} - \mathbf{X}(\mathbf{X}^T\mathbf{X})^{-1}\mathbf{X}^T)\mathbf{Y} \quad (2.24)$$

One can say that when the LSA is implemented into the OTPA technique to estimate TF matrix, the technique is identical to the Multi Input/Multi Output (MIMO) method of obtaining FRF matrix, whether the same input and output quantities are selected. Actually, the term $\mathbf{X}^T\mathbf{X}$ states the averaged Auto Power Spectrum (APS) matrix $\mathbf{G}_{xx}$ of the input signals and the other term $\mathbf{X}^T\mathbf{Y}$ states the averaged Cross Power Spectrum (CPS) matrix $\mathbf{G}_{xy}$ between the input and output signals in Eq. 2.22, assuming that they are not divided by the number of measurement blocks. If this

normalization is revoked, obtaining the TF matrix in OTPA using LSA is equivalent to the MIMO method of estimating FRF as [11]:

$$\mathbf{H} = (\mathbf{X}^T \mathbf{X})^{-1} \mathbf{X}^T \mathbf{Y} = \mathbf{G}_{xx}^{-1} \mathbf{G}_{xy} \quad (2.25)$$

$$\mathbf{G}_{xx} \triangleq 1/r \mathbf{X}^T \mathbf{X} \quad (2.26)$$

$$\mathbf{G}_{xx} \triangleq 1/r \mathbf{X}^T \mathbf{X} \quad (2.27)$$

In general, coherence between input signals is a problem in OTPA method [11, 56, 57]. On the other hand, for example, the shaker signals can be partially coherent in accordance with each other in MIMO methods. Owing to external disturbances such as measurement noise and coherence between input signals, the solution of Eq. 2.25 results in inaccurate FRF estimation. Actually, the small measurement distortions have more effect in Eq. 2.25, thus higher values can cause erroneous FRF determination and more averaging may be needed for correct FRF estimation. Note that Eq. 2.25 can satisfy for residuary coherence between inputs by the off-diagonal terms in $\mathbf{G}_{xx}$. Hence, correct FRFs (or TF for OTPA) are theoretically computed, even though the input signals are almost completely coherent [11].

However coherence between the input signals practically passes 30-40%, the reasons mentioned above further stands for the OTPA computation in Eq. 2.22. Therefore, the OTPA method is enhanced with the use of SVD algorithm to reduce measurement disturbances (e.g. measurement noise). In addition, while MIMO is only used to determine receptance FRF, OTPA can be applied for TF estimates. TFs are also named as transmissibilities that is the ratio among two different quantities. Therefore, amplitude peaks and fall over a frequency do not necessarily mean resonances or anti-resonances of the system. As a result, paying attention to interpret the TFs is important to focus on transmission path contributions [11].

2.2.2 OTPA enhancement with singular value decomposition

In general, linear least-squares problems are best solved by using the Singular Value Decomposition (SVD). The aim of the SVD is to find the singular values (i.e. eigenvalues) of the data set. Actually, the SVD obtains the correlative elements of a

data set, in fact each singular value is non-correlative (i.e. orthogonal) to the other singular values. The most powerful correlative element of the data is stated as the first singular value, remaining a non-correlative residual data set. After that, the SVD is implemented on the residual data set to obtain the latter singular value, again leaving behind a non-correlative residual data set. The process is recurred up to the complete data set is indicated by the SVD [43, 58, 59].

In some cases input signals can compound with measurement noise that makes the input signal highly coherent. For these cases, the transfer matrix $\mathbf{H}$ obtained explicitly by using Eq. 2.22 will lead to wrong estimations. The reason for this is that the inversion term $(\mathbf{X}^T \mathbf{X})^{-1}$ amplifies the measurement noise. Therefore, the SVD method is used to avoid wrong estimations. Essentially, the input matrix $\mathbf{X}$ is decomposed by a SVD as [11]:

$$\mathbf{X} = \mathbf{U} \mathbf{\Sigma} \mathbf{V}^T \quad (2.28)$$

where $\mathbf{U}$ is an $r \times r$ unitary column-orthogonal matrix, $\mathbf{\Sigma}$ is a $r \times m$ matrix with nonnegative numbers on the diagonal (as established for a rectangular matrix) and zeros off the diagonal, $\mathbf{V}^T$ is an $m \times m$ unitary matrix that defines the conjugate transpose of an $m \times m$ unitary column-orthogonal matrix $\mathbf{V}$ [11].

To explain Eq. 2.28 physically [11], the matrix $\mathbf{V}$ is considered as a set of orthonormal or examining basis vector directions for $\mathbf{X}$, and the matrix $\mathbf{U}$ includes a set of orthonormal output basis vector directions for $\mathbf{X}$. In linear algebra, when two vectors in inner product space are orthogonal and both of unit length, or in other words, when their inner product is 0 and the norm of each vector is 1, these two vectors are orthonormal. Furthermore, the matrix $\mathbf{\Sigma}$ includes the singular values that are supposed as scalar gain controls which mean multiplying every input to obtain a matching output. The values Σ_{ii} are ranked in decreasing style; hence the diagonal matrix $\mathbf{\Sigma}$ is individually identified by $\mathbf{X}$ [11].

Generally, the SVD can be utilized to any $r \times m$ matrix. Otherwise, standard eigendecomposition, which is the factorization of a matrix into a canonical form to define the matrix in terms of its eigenvalues and eigenvectors, can solely be utilized to significant square matrices. Even so, eigendecomposition can analytically be used to express the SVD by the following equations [11]:

$$\mathbf{X}^T \mathbf{X} = \mathbf{V} \boldsymbol{\Sigma}^T \mathbf{U}^T \mathbf{U} \boldsymbol{\Sigma} \mathbf{V}^T = \mathbf{V} (\boldsymbol{\Sigma}^T \boldsymbol{\Sigma}) \mathbf{V}^T \quad (2.29)$$

$$\mathbf{X} \mathbf{X}^T = \mathbf{U} \boldsymbol{\Sigma} \mathbf{V}^T \mathbf{V} \boldsymbol{\Sigma}^T \mathbf{U}^T = \mathbf{U} (\boldsymbol{\Sigma} \boldsymbol{\Sigma}^T) \mathbf{U}^T \quad (2.30)$$

Here, the eigendecompositions of the left hand sides are expressed by the right hand sides of the equations. Finally, the non-zero eigenvalues of either $\mathbf{X}^T \mathbf{X}$ or $\mathbf{X} \mathbf{X}^T$ are equivalent to the squares of the non-zero singular values of $\mathbf{X}$. Moreover, eigenvectors of $\mathbf{X} \mathbf{X}^T$ and $\mathbf{X}^T \mathbf{X}$ are the columns of the $\mathbf{U}$ (left singular vectors) and $\mathbf{V}$ (right singular vectors), respectively [11]. Note that implementing the eigendecomposition to identify the SVD matrices is mathematically unsteady in practice, particularly for singular values almost zero. In such situations, the SVD matrices are explicitly identified; hence implementing SVD does not improve the OSPA method. For this reason, specific algorithms (e.g., the one-sided Jacobi orthogonalization and the modified Golub-Reinsch algorithm) are used to deal with computational conditioning troubles [11, 60].

The SVD of $\mathbf{X}$ is implemented in the evaluation of the pseudo-inverse $\mathbf{X}^+$ in a direct way as [11]:

$$\mathbf{X}^+ = \mathbf{V} \boldsymbol{\Sigma}^{-1} \mathbf{U}^T \quad (2.31)$$

where $\boldsymbol{\Sigma}^{-1}$ is the inversion of diagonal singular value matrix $\boldsymbol{\Sigma}$. In Eq. 2.31, the extra rows in the matrix $\boldsymbol{\Sigma}$ were left out in the SVD calculation to make $\boldsymbol{\Sigma}$ a $m \times m$ square matrix [11].

Consequently, the TF matrix $\tilde{\mathbf{H}}$ utilizing SVD technique is estimated by substituting Eq. 2.31 into Eq. 2.22 as [11]:

$$\tilde{\mathbf{H}} = \mathbf{V} \boldsymbol{\Sigma}^{-1} \mathbf{U}^T \mathbf{Y} \quad (2.32)$$

where the residual content μ places in the null space of $\mathbf{U}^T$ with the SVD estimation [11].

It is observed that noise effects and other external disturbances result in small singular values in engineering applications [11, 61]. Thus, the small singular values are undesired and it is advised to remove them. On the other hand, it is noticed that holding all singular values can make the least-squares suitable for the examined measurement better. Nevertheless, the number of utilized singular values in the TF computation of one measurement divulges a trade in the ensuing fit on another analogous measurement. In fact, this cross verification procedure divulges which of the smallest singular values are connected with the noise effects, even if the noise can differ in both measurements [11, 62-64]. Hence, the TF determination is usually developed by considering solely a decreased set of singular values [11].

Although Eq. 2.32 represents estimation for the TF matrix, the measurement noise and crosstalk among measurement channels cannot be identified, and are contained in the analysis data in some circumstances. In such conditions, it is suggested that the measurement noise and crosstalk influence are identified before evaluating the pseudo-inverse $\mathbf{X}^+$, so that the Principal Component Analysis (PCA) can be implemented after the SVD of matrix $\mathbf{X}$ [11, 43, 59, 61].

2.2.3 OTPA with principal component analysis

Principal Component Analysis (PCA) techniques [43] are implemented to find out the data which is irrelevant to the overall response at the receiver location (e.g. measurement noise), and to prevent computational problems in evaluating the TFs. PCA is a method utilized to decrease a data set which includes a lot of interconnected variables to a smaller data which keeps most of the useful information. Exclusively, the PCA contains an orthogonal conversion (i.e. SVD) which transforms the data thus it is non-correlative. These non-correlative variables are defined as the Principal Components (PCs). The PCs are ordered like that the PC with the biggest difference inside the data set is the first PC, the next variable is stated as the second PC, and the process continues until finding the lowest PC. The lowest PCs indicate very small difference which means that the relation between the variables is nearly constant and linear. For instance, if the PCA is implemented on vibration data, the most powerful contribution may be owing to bending mode that is assumed as the first PC. Then, if the PCA goes on the residual data, the next powerful contribution may be torsional wave that can be assumed as the second PC and so on. Also, this method is

implemented in a similar way for sound pressure by ordering the PCs involved with phase relations [43].

The PCA for the OTPA method is carried out in the frequency domain where the Discrete Fourier Transform (DFT) spectra defines the reference data (input variables) for every measurement location. Moreover, the PCA is applied on all of the measurement blocks r , that are thought of being piece of vectors $[x^{(1)} \dots x^{(m)}]$. Substantially, the PCA identifies to what scope the measurement locations participate in mutual signals (i.e. to what scope the signals are correlative) [43, 65].

To obtain as many PCs as the number of data sets (operational conditions) is probable where the maximum number of PCs can be the number of input measurement locations m for the OTPA. Exclusively, the number of meaningful PCs can associate with the number of meaningful DOFs of all excitation sources and/or transmission paths when all sources and/or transmission paths are appropriately specified and contained in the measurement setup. The PCs are obtained from the SVD of a matrix. Indeed, the PCs are the singular values located in the diagonal of the matrix Σ , designated in Eq. 2.28. Notice that the SVD technique is implemented to a signal to distinguish the associated element from the unassociated element; thereby it is crucial to remember that the SVD can solely distinguish sources which are either unassociated or solely partially associated [43].

The PC scores are found by multiplying the matrix of the PCs Σ by the left eigenvector matrix U . According to Eq. 2.28, this multiplication gives the relation below [43]:

$$\mathbf{Z} = \mathbf{XV} = \mathbf{U}\Sigma \quad (2.33)$$

where the matrix of PC scores is stated as $\mathbf{Z}$. It is already known that the external disturbances (i.e. measurement noise) and crosstalk influence causes smaller PCs, thus they should be dispensed with additional calculations by analyzing the matrix of PC scores $\mathbf{Z}$. To obtain a percentage contribution for each PC score, the matrix of PC scores $\mathbf{Z}$ is divided by the total sum of the PC scores, so that the contribution of each PC to the overall signal is examined. The smaller PCs are established as zero to get rid of them, but these values have to be defined again in the evaluation of the inverse of the noise removed PC matrix Σ_r [43].

A definition to determine the crosstalk compensated and noise eliminated TF matrix $\tilde{\mathbf{H}}_r$ is obtained by inserting the noise removed PC matrix $\mathbf{\Sigma}_r$ into Eq. 2.28 [43]:

$$\tilde{\mathbf{H}}_r = \mathbf{V}\mathbf{\Sigma}_r^{-1}\mathbf{U}^T\mathbf{Y} \quad (2.34)$$

Here $\mathbf{\Sigma}_r^{-1}$ is a diagonal matrix that specifies the inverse of the noise removed PC matrix. The diagonal elements of $\mathbf{\Sigma}_r^{-1}$ are specified as [43]:

$$\mathbf{\Sigma}_r^{-1} = \begin{cases} 1/\sigma_n, & \text{if } \sigma_n > \text{thres} \\ 0, & \text{otherwise} \end{cases} \quad (2.35)$$

Here *thres* means a threshold for the minimum value of the PCs to be kept in the analysis. Putting a threshold prevents the consequence of a singular matrix when the inverse of the noise removed PC matrix is computed [43, 58, 61, 66, 67].

2.2.4 OTPA response synthesis

The noise eliminated TF matrix $\tilde{\mathbf{H}}_r$ is estimated by using the SVD method together with the PCA technique. Hence, the response at the receiver location can be synthesized by receiving the convolution of the response matrix with the impulse response function matrix [43]:

$$\mathbf{Y}_s(t) = \mathbf{X}(t) \otimes \tilde{\mathbf{H}}_r(t) \quad (2.36)$$

Here,

$\mathbf{Y}_s(t)$: the synthesized response matrix in the time domain

$\mathbf{X}(t)$: the measured reference matrix in the time domain

$\tilde{\mathbf{H}}_r(t)$: the operational impulse response function matrix in the time domain

$\otimes$ designates a convolution operation of $\tilde{\mathbf{H}}_r(t)$ and $\mathbf{X}(t)$.

It is important to note that Eq. 2.36 is in the time domain, so that the convolution operation must be performed for each time component. The operational impulse response function matrix in the time domain $\tilde{\mathbf{H}}_r(t)$ is actually obtained by taking the Inverse Discrete Fourier Transform (IDFT) of the operational TF matrix $\tilde{\mathbf{H}}_r(j\omega)$,

and it is utilized in the calculation as a vector of Finite Length Impulse Response (FIR) filters that are convolved with the time domain input matrix to synthesize the output [43].

The convolution operation can be conducted in the frequency domain by writing it as the point-wise multiplication of the TF matrix $\tilde{\mathbf{H}}_r$ with the input matrix $\mathbf{X}$ [43]:

$$\mathbf{Y}_s(j\omega) = \mathbf{X}^T(j\omega) \cdot \tilde{\mathbf{H}}_r(j\omega) \quad (2.37)$$

Here,

$\mathbf{Y}_s(j\omega)$: the synthesized response matrix in the frequency domain

$\mathbf{X}^T(j\omega)$: the transpose of the measured reference matrix in the frequency domain

$\tilde{\mathbf{H}}_r(j\omega)$: the operational TF matrix in the frequency domain

The "." means a point-wise multiplication among $\mathbf{X}(j\omega)$ and $\tilde{\mathbf{H}}_r^T(j\omega)$.

The synthesized responses $\mathbf{Y}_s(t)$ and $\mathbf{Y}_s(j\omega)$ are defined as vectors of the output contributions at each input measurement location for each measurement block. The total response at the receiver location is determined by summing up the output contributions from individual input measurement locations. The response synthesis of each measurement block is combined to draw a plot of the response synthesis over the total measurement time. The contribution of each established source to the overall response is examined by summing up the output contributions of the related synthesis vectors [43].

The overall response synthesis (i.e. the total synthesis contributions) is compared with the actual measured total response to represent the usefulness of the OTPA calculations. The measured output at the receiver location is often agreed with the overall output synthesis with respect to the consequences of this crosscheck. Nevertheless, this situation does not definitely refer that the synthesized contributions are accurate [25, 43]. There are some limitations of the OTPA technique that will lead to inaccurate transmission path identification such as neglected sources and/or transmission paths in the measurement setup, cross-coupling between reference measurements, and wrong estimation of transmissibility matrix [25, 43, 56, 68, 69].

3. VERIFICATION OF THE OPERATIONAL TRANSFER PATH ANALYSIS PROCEDURE: APPLICATION TO SIMPLE FLAT PLATE

Operational Transfer Path Analysis (OTPA) is already defined as a signal processing technique to forecast the sum of propagated structure-borne and/or airborne noise and/or vibration by analyzing the linear relationship between selected inputs (references) and outputs (responses) [11, 39, 70]. It is used to determine the linearized Transfer Function (TF) matrix among the group of these chosen references and responses to utilize them for the identification of the critical vibro-acoustic transmission paths and the decomposition of the vibration and/or noise sources [11, 70, 71]. Overall response synthesis is the total of whole synthesized source contributions and it represents the effectuality of the OTPA calculations by comparing the real measured output to the synthesized one [11, 43, 70, 71]. It should be noted that if the real measured output at the receiver location is compared with the overall synthesized output, they will usually match with each other. On the other hand, the consequences of this match do not signify that the synthesized contributions are accurate [25, 43]. Therefore, the TF and phase information comparison can be also used to decide the effectiveness of the OTPA calculations, in addition to comparison of the measured and synthesized output.

In this chapter, numerical and experimental studies about the application of OTPA to a simple plate are presented so as to verify the OTPA procedure and the MATLAB program developed for the investigation presented in this thesis. Dynamic analysis as well as the forced vibration analysis of the test structure is numerically carried out in order to generate data for comparison purposes. An experimental setup is introduced to measure the Frequency Response Functions (FRFs), selected input and output quantities for various operational conditions at the defined coordinates on the structure. The OTPA algorithm developed in MATLAB platform is used to compare measured and synthesized responses at the receiver location. TFs from selected references to responses and their phase information are also examined using predicted as well as measured data. In what follows, numerical simulations are

presented first. Then, in the second part, experimental data are used to establish the transfer functions between the reference and response coordinates.

3.1 Numerical Studies

The OTPA procedure usually includes three main steps:

- Estimation of the TF matrix from measured input and output data,
- Synthesization of the overall response (output),
- Decomposition of the vibro-acoustic transmission paths and their sources (Contribution analysis).

The aim of this section of thesis is the validation of the OTPA procedure implemented in MATLAB by comparing numerically “measured” and the estimated outputs, TFs and their phase angles. Therefore, contribution analysis is excluded in the numerical studies here. In this section, first, a flowchart of the numerical procedure is introduced, and then preparation of the numerical model is presented. Finally, three kinds of numerical simulations are described.

3.1.1 Flowchart for OTPA procedure for numerical applications

As a first move of the numerical study, the structure has to be prepared for the vibration analysis. It is modeled using IDEAS, and then it is imported into ADAMS environment. Boundary conditions, input and output locations are defined on the structure by using appropriate ADAMS commands. Two types of analyses are used to generate desired data for comparison purposes. First of all, forced vibration analysis is carried out so as to obtain the FRFs between selected input and output coordinates. Secondly, dynamic analysis is carried out by acting operational forces on input coordinates and by measuring the structural responses at the defined output coordinates. Due to the fact that the OTPA procedure needs to be carried out in frequency domain, Fast Fourier Transform (FFT) is utilized to transform time-domain data into frequency-domain in ADAMS. After this transformation, the acquired three type data (FRFs, applied operational forces and measured structural responses) are imported into MATLAB environment. A MATLAB code developed for OTPA is run to estimate the complex transfer matrix by multiplying the inverse of the input matrix with the measured output matrix. The algorithm uses two different methods (Least Square Algorithm, LSA and Singular Value Decomposition,

SVD) to estimate the system transfer matrix. Also, overall response (output) matrix is synthesized from the multiplication of the estimated FRF matrix and the “measured” input matrix. In the final step of numerical study, the estimated FRFs and their phase angles are compared with the FRFs and their phases that are obtained using well known Single Input/Single Output (SISO) approach during the forced vibration analysis of the structure. Moreover, the measured responses at the defined output coordinates are compared with the synthesized responses to validate the OTPA implementation. The flowchart of the numerical study is schematically represented in Figure 3.1.

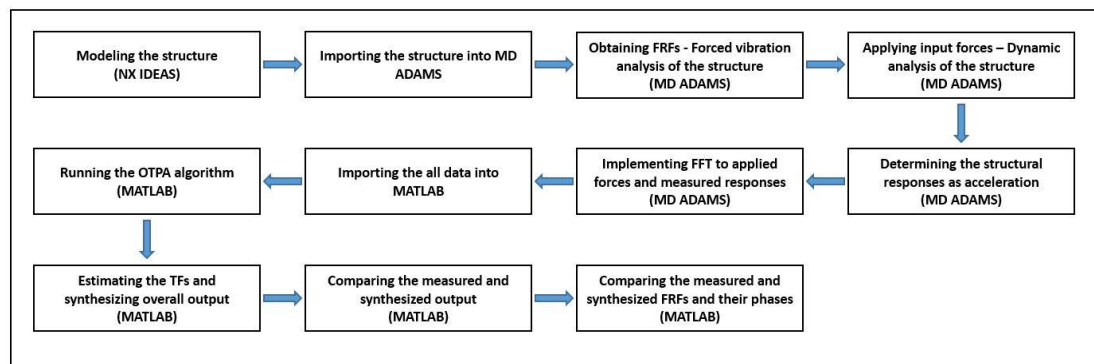


Figure 3.1 : Flowchart representation of the OTPA procedure for numerical case studies.

3.1.2 Simple flat plate numerical model

Some numerical studies about the OTPA method are performed using a simple flat plate. Dimensions of the flat plate studies here are 1700 x 4000 mm and it has 2 mm thickness. The geometry of the plate is firstly modeled and the material type is defined as stainless steel using IDEAS [72]. The flat plate is modeled is divided into finite elements utilizing shell type of elements and is meshed with quad type elements. After generating the finite element model of the structure, the plate is imported into ADAMS environment over ADAMS-Flex module [73]. The boundary conditions are defined for the structure such that one of the short edges of the plate is fixed. For the rest of numerical studies in thesis, this numerical model given in Figure 3.2 is used.

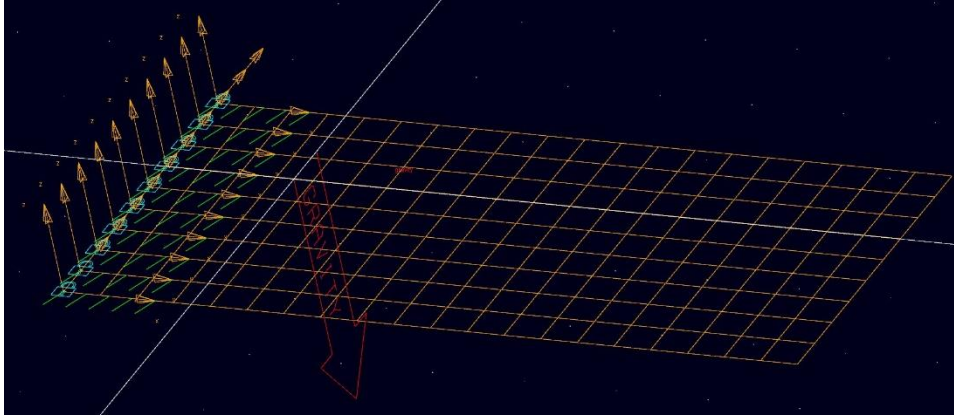


Figure 3.2 : Flat plate numerical model.

3.1.3 Numerical case study I

After modeling the structure and importing the model into ADAMS environment, the forced vibration analysis is performed on the structure using ADAMS-View module [73]. To do this analysis, first of all, input locations, where the operational forces are applied, and the output locations, where the structural responses are measured, must be defined on the flat plate. In this numerical case study, two input locations (F_1, F_2) and three output locations (A_1, A_2, A_3) are chosen as illustrated in Figure 3.3, resulting in six FRFs that can be obtained via this numerical simulation. The operational forces are selected as swept sine excitations whose amplitudes are 1. Swept sine excitation is preferred owing to providing much higher excitation level and much better FRF data quality [74, 75]. Moreover, the structure is excited with an appropriate sweep rate over the frequency range of interest [75, 76].

As a first stage of forced vibration analysis of the flat plate, the first operational excitation load is acting on the first input location (F_1) and the structural responses are measured at the defined three output locations (A_1, A_2, A_3), thus the FRFs (H_{11}, H_{12}, H_{13}) and their phases are obtained between this first operational load and corresponding responses. As a second stage of forced vibration analysis of the flat plate, the second operational excitation load is acting on second input location (F_2) and the structural responses are measured at the defined output locations (A_1, A_2, A_3), so that the FRFs (H_{21}, H_{22}, H_{23}) and their phase angles are obtained between this second operational load and corresponding responses.

As expected, all the FRFs obtained here are complex, containing modulus and phase information. In the example above, responses are due to single excitation; hence

these FRFs can be named as FRFs of the Single Input/Multi Output system (SIMO FRFs). As a result, the SIMO FRFs or TFs can be determined here without the need to consider the effect of the other operational force(s), which means that while an operational input load acts on the structure, other operational load(s) are isolated from the system.

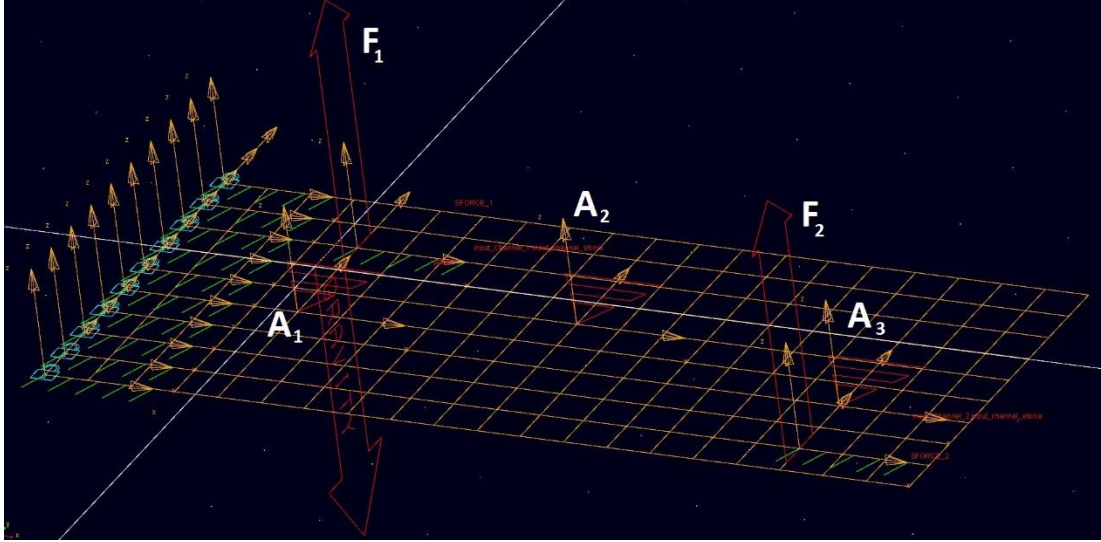


Figure 3.3 : Defined input and output location on the structure for the first numerical study.

The system identification part of the OTPA is actually the same as Multi Input/Multi Output (MIMO) method of identifying FRFs or TFs [11, 70]. For accurate estimation of the MIMO FRFs which are to be compared with SIMO FRFs, dynamic vibration analysis of the structure is required. In this case, the operational loads must simultaneously excite the structure at the defined input locations. Moreover, the measurement of the excitations (input) and the corresponding responses (output) should be made under the operational conditions for modeling the dynamical behavior of the structure. As required, similar measurements are repeated under different operational conditions. It should be noted that all the selected excitations (references or inputs) must be linearly independent from each other, and also the number of chosen references must be smaller than the number of operational conditions due to the requirements of the implementation of the OTPA procedure [11, 43, 70, 71].

First, the operational input loads are selected as random signals. Random excitation is preferred owing to the distribution of the excitation energy over the frequency

range of interest and also for reducing leakage fairly well [75, 77]. The random signals are generated in MATLAB using “randn” command; hence all the operational forces are linearly independent to each other [78]. Then, the operational forces are imported into ADAMS environment and each of them is specified with the related input. As stated before, there are two excitation forces for this numerical simulation, which means that at least three operational conditions must be created, thus six random signals are generated. The first two excitation forces (random signals) are acting on the reference points and the corresponding responses are "measured" as acceleration in time-domain. The same procedure is repeated three times (corresponding to other three operational conditions), each time two operational input loads are applied at two input locations and corresponding responses are measured at the defined output locations. All measured input and output data are stored in time-domain. The OTPA algorithm requires the data in frequency-domain; hence FFT operation is utilized by applying Hanning windowing to transform the time-domain measured data into frequency-domain. Hanning windowing is applied to minimize the leakage effect [79]. All the data obtained in frequency-domain (SIMO FRFs and their phase angles, applied input forces and measured responses for every operational condition) are imported into MATLAB environment to perform the OTPA procedure.

A MATLAB code is developed to process the obtained frequency-domain data using the OTPA algorithm. The linear relationship between a group of selected input and output quantities is setup as it is given in Eq. 2.15. The extended form of this relationship is defined in Eq. 2.20 for every individual frequency term. The operational TF matrix is estimated by multiplying the inverse of input matrix with the output matrix, where LSA and SVD techniques are utilized to take the inverse of input matrix as it is shown in Eq. 2.25 and Eq. 2.32, respectively. The overall responses are synthesized from multiplication of input matrix and estimated TF matrix using Eq. 2.37. Consequently, the required data for comparison purposes to validate the OTPA procedure is obtained within this MATLAB code.

The "measured" responses are also calculated by estimating the TFs first and then by multiplying the operational forces with the TF matrix. As expected, the synthesized (calculated) responses (output) matched well with the "measured" ones as shown in Figures 3.4 and 3.5. It should be also noted that there is no difference between the

results obtained via LSA and SVD techniques utilized to compute the inverse of input matrix. Although all measured response (output) data indicate good agreement with the calculated output data, the comparison of the SIMO FRFs and with the estimated MIMO FRFs (TFs) exhibit some differences as illustrated next. The "measured" via SIMO and the estimated FRFs and their phase angles are presented in the frequency range of interest (0-500 Hz) as it is seen in Figures 3.6, 3.7, 3.8, and 3.9. It is seen that although the SIMO and estimated MIMO FRFs show similar trends, the estimated MIMO FRFs and their phase angles are much noisier than those of SIMO FRFs. As a result, it can be said that the crosstalk effect between operational forces could not be totally compensated. Despite of having well-matched output results, the critical source and/or path decomposition could not accurately be done because of the very noisy TF estimation. It is obvious that, correct estimation of the operational TFs is required to perform successful OTPA. On the other hand, if there is no intervention to the singular values of input matrix (e.g., applying a threshold), LSA and SVD techniques will give the same results when taking inverse of the input (reference) matrix.

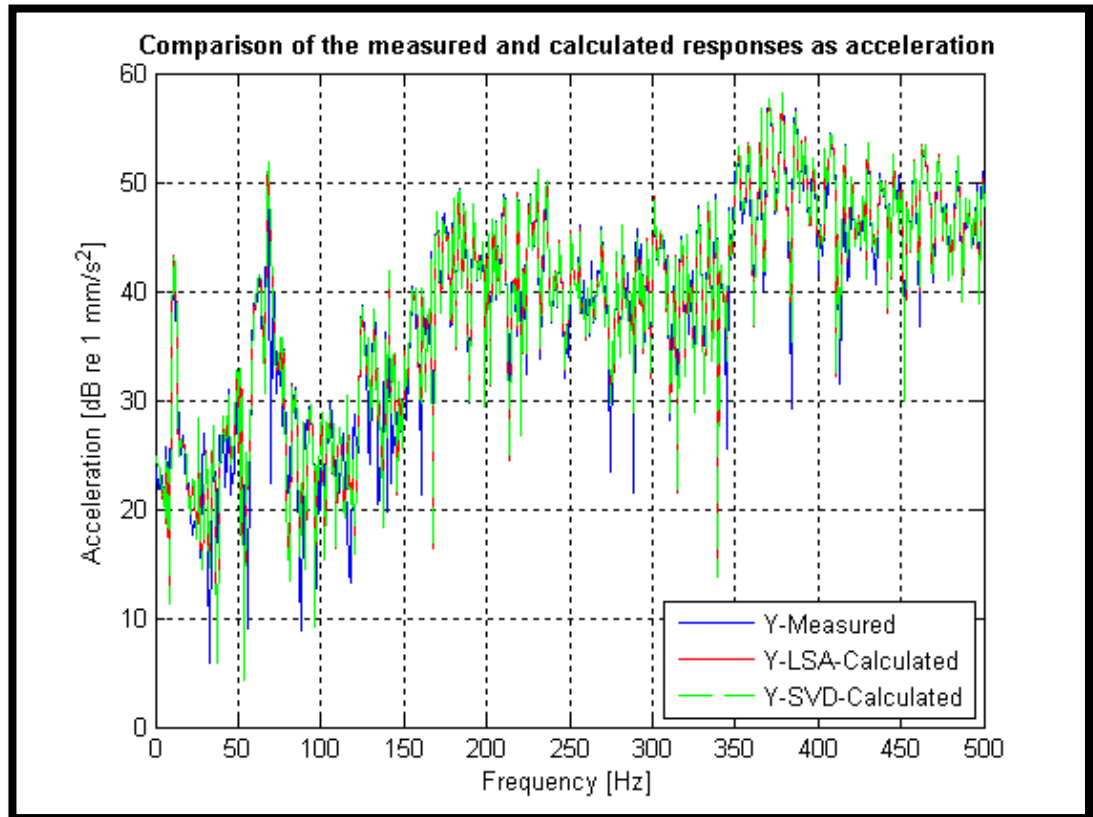


Figure 3.4 : Output comparison of the first response location for the first operational condition $A_1^{(1)}(\omega)$ in the first numerical study.

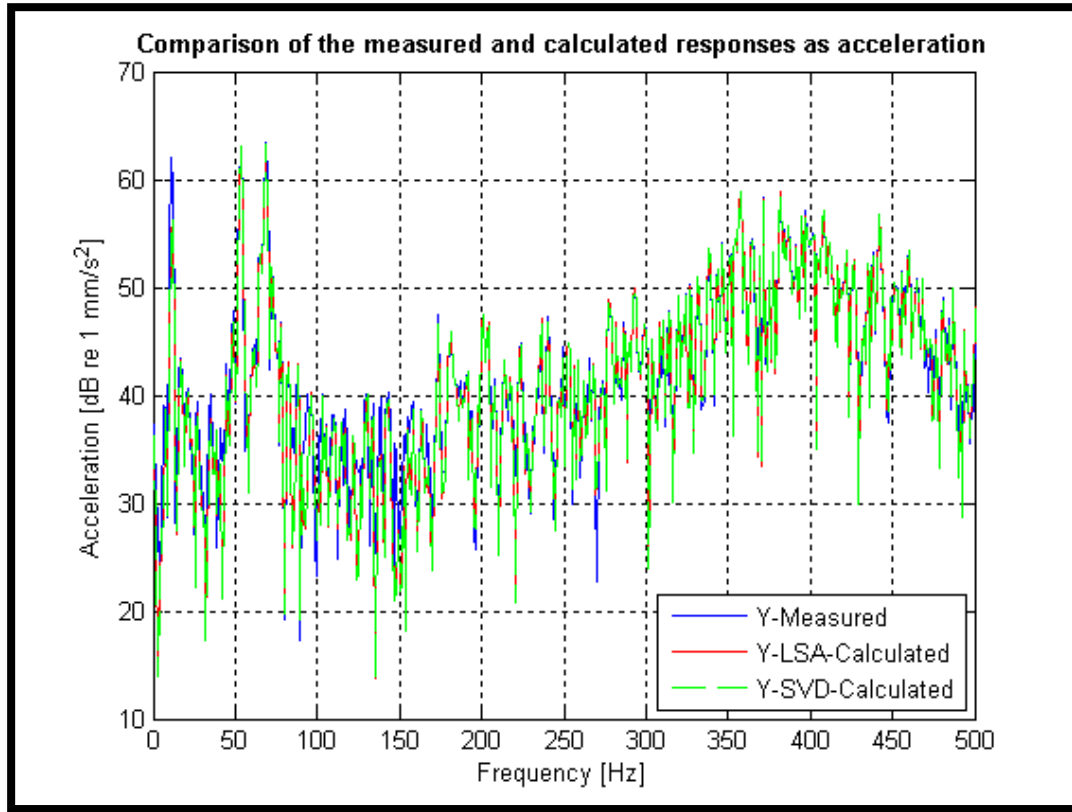


Figure 3.5 : Output comparison of the second response location for the third operational condition $A_3^{(2)}(\omega)$ in the first numerical study.

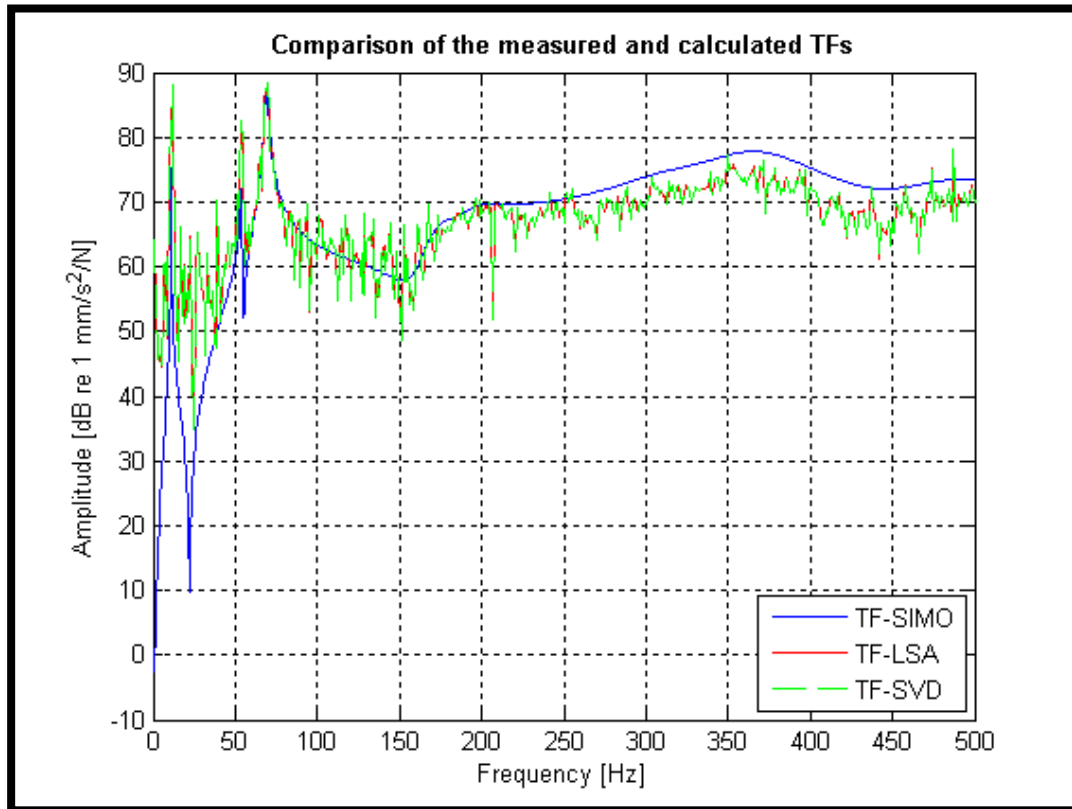


Figure 3.6 : TF comparison for $H_{12}(\omega)$ in the first numerical study.

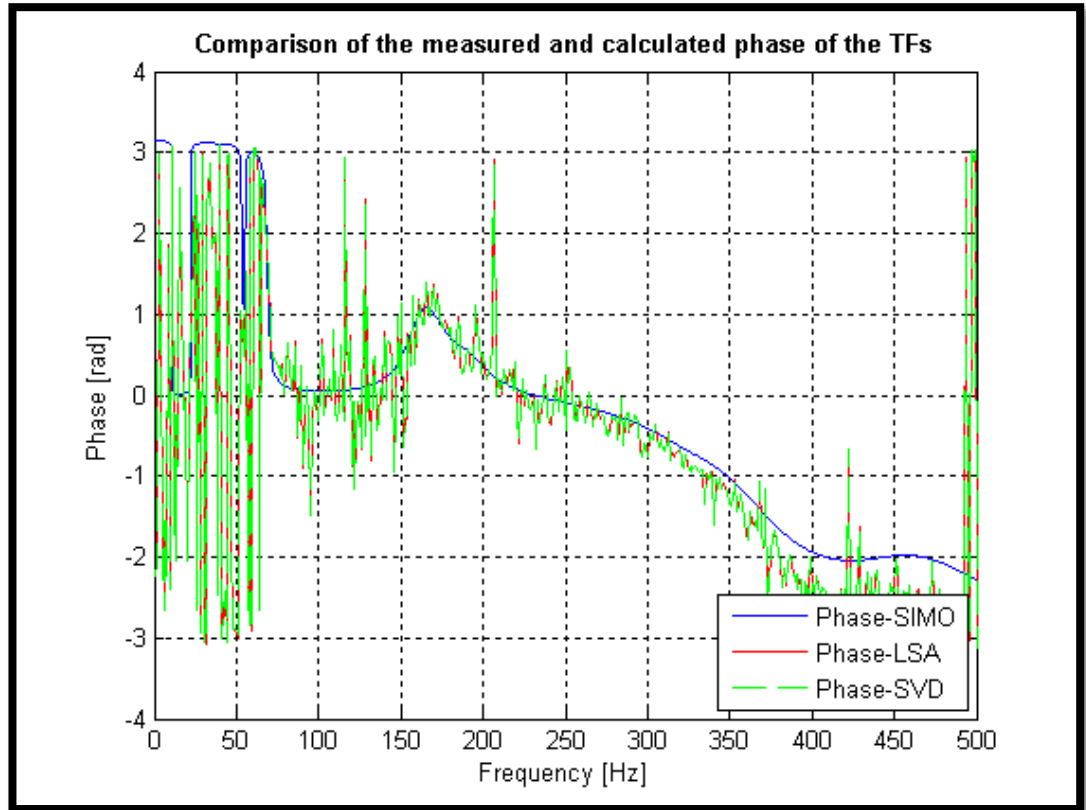


Figure 3.7 : Phase information comparison for the phase of $H_{12}(\omega)$ in the first numerical study.

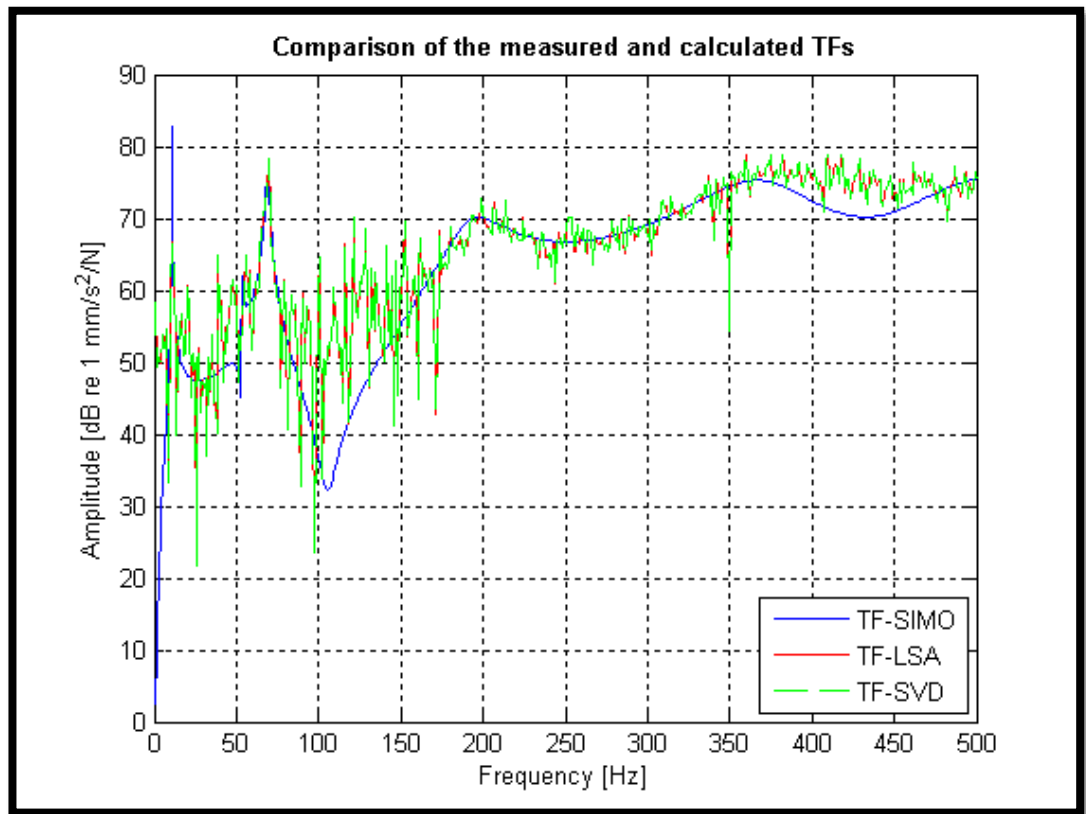


Figure 3.8 : TF comparison for $H_{21}(\omega)$ in the first numerical study.

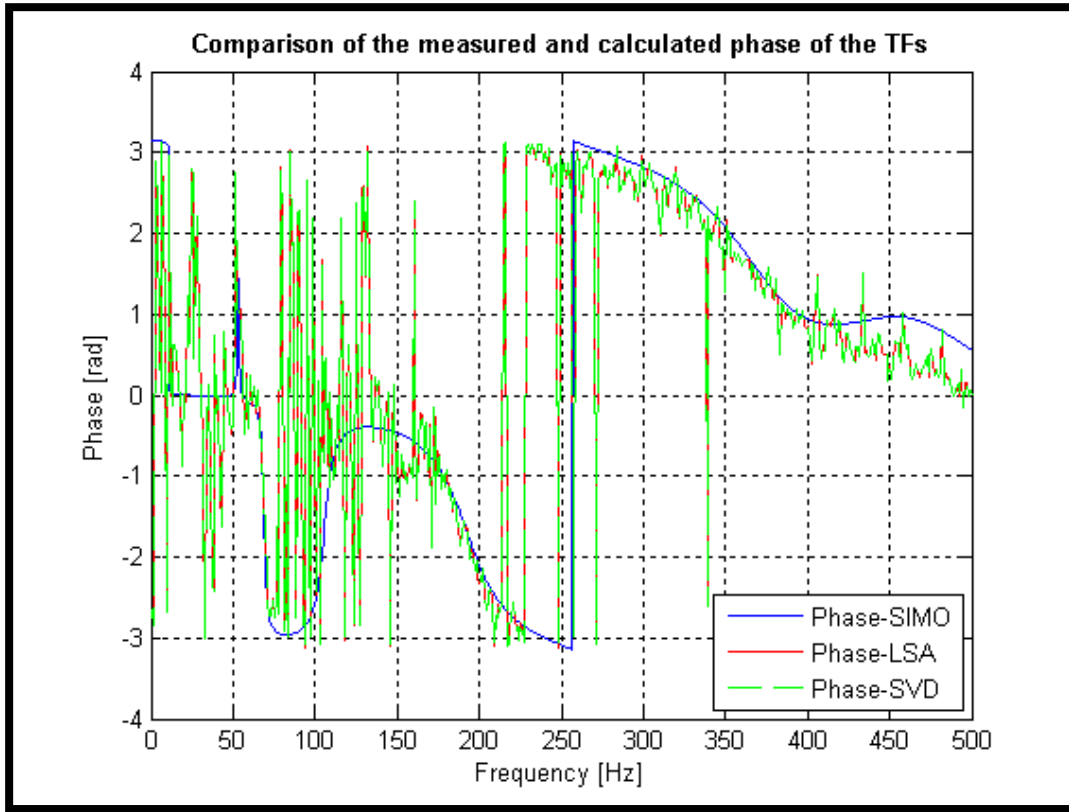


Figure 3.9 : Phase information comparison for the phase of $H_{21}(\omega)$ in the first numerical study.

3.1.4 Numerical case study II

The selection of the input and output locations for the measurement of the reference and response quantities is an important subject in OTPA procedure utilization. In the first numerical case study, it is observed that very noisy TF identification can cause inaccurate vibro-acoustic transfer pathways or vibration source determination. The second numerical study presented here differs from the first one in the sense that different (possibly more appropriate) input and output locations are chosen this time. In this case, it is thought that exciting the structure from relatively more rigid location can provide much better excitation within interested frequency band. Thus, two input locations (F_1, F_2) and two output locations (A_1, A_2) shown in Figure 3.10 are chosen this time, and four FRFs are obtained. In this numerical case study, the same procedure is followed as in the first numerical case study for the estimation of operational TF matrix and the overall structural responses.

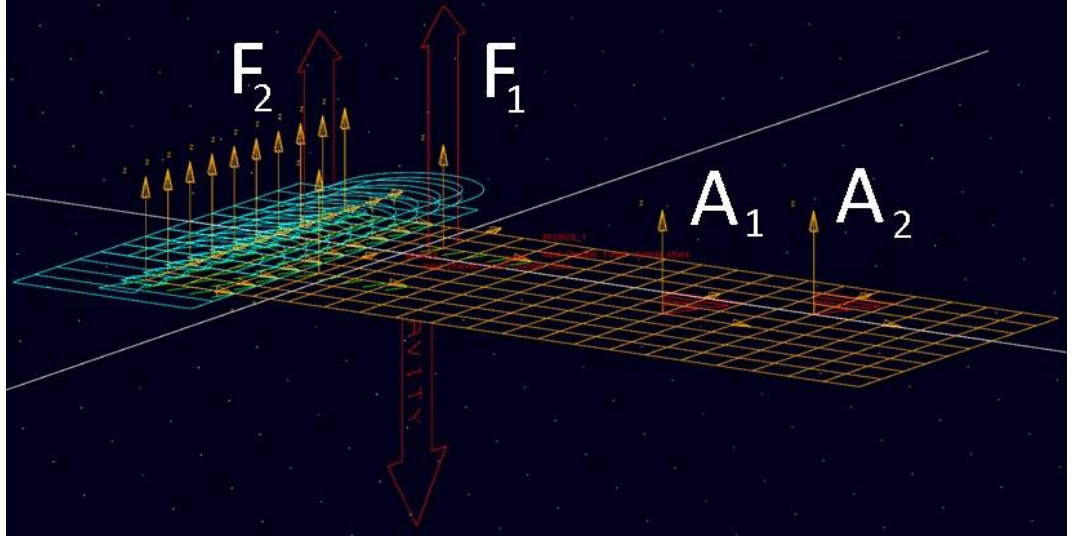


Figure 3.10 : Defined input and output location on the structure for the second numerical study.

In this case, as expected, the synthesized (calculated) output matched with the "measured" one better than before as illustrated in Figures 3.11 and 3.12. Moreover, LSA and SVD techniques produced the same results when none of the singular values are discarded in SVD solution.

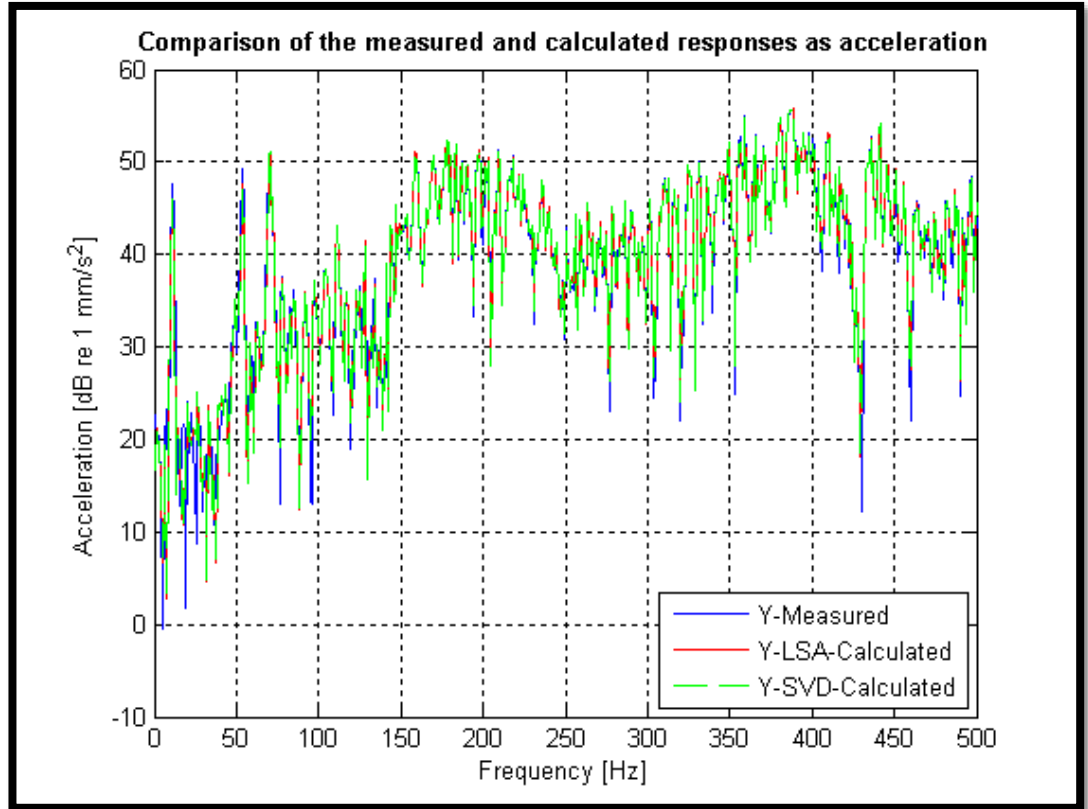


Figure 3.11 : Output comparison of the first response location for the second operational condition $A_2^{(1)}(\omega)$ in the second numerical study.

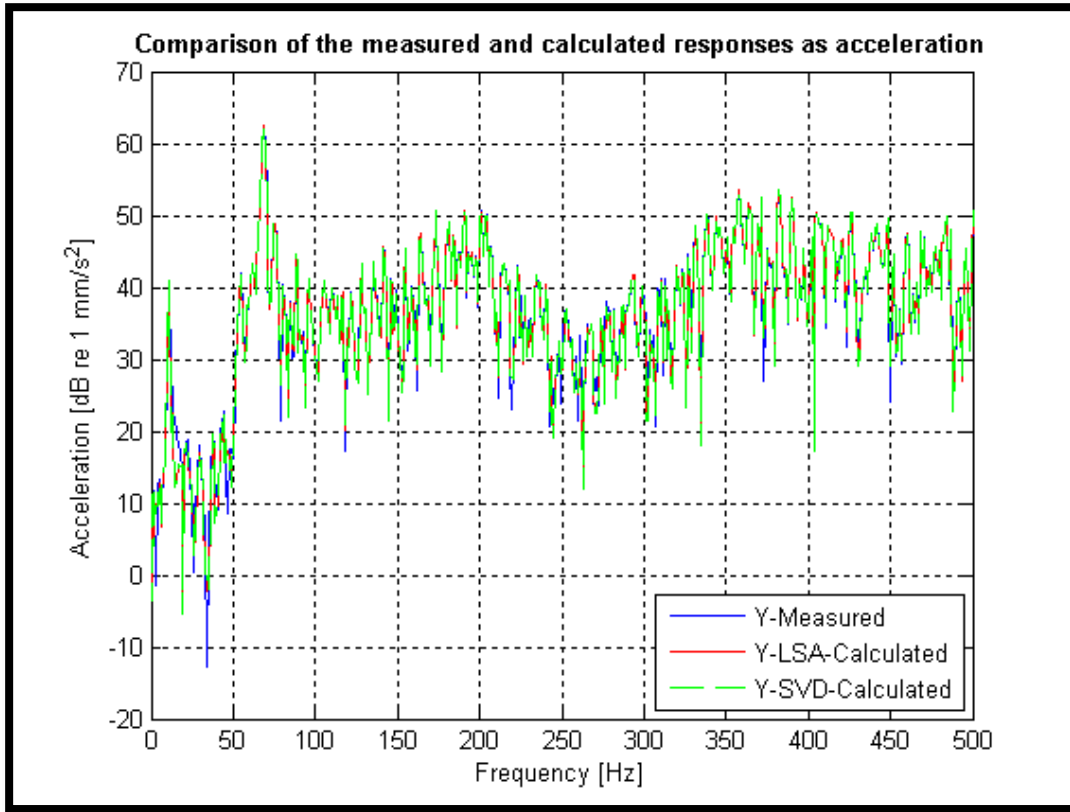


Figure 3.12 : Output comparison of the second response location for the third operational condition $A_3^{(2)}(\omega)$ in the second numerical study.

In this case, the SIMO FRFs "measured" and estimated MIMO FRFs and their phase angles match much better than before within the frequency range of interest (0-500 Hz) as demonstrated in Figures 3.13, 3.14, 3.15, and 3.16. It is noticed however, that the crosstalk effect increases as the locations of the operational input forces get closer to each other. This is not surprising since it gets more and more difficult to make two excitation forces independent of each other as they physically get closer to each other. It is seen that OTPA algorithm can yield better TF estimation if input and output locations are chosen carefully on the examined mechanical system. However, there is still a problem, noticed during this numerical case study, that the estimated TFs and their phase angles contain unexpected behavior as can be seen in Figures 3.15 and 3.16, for instance around 250 Hz. Uncompensated crosstalk effect or unmodeled portion of overall response (given in Eq. 2.21) can cause such problems. Furthermore, the estimated data is still noisy due to numerical errors in the matrix inversion. It is therefore decided that linear spectrum averaging should be applied to signals to reduce noise level and to minimize unexpected noisy peaks in the spectrum.

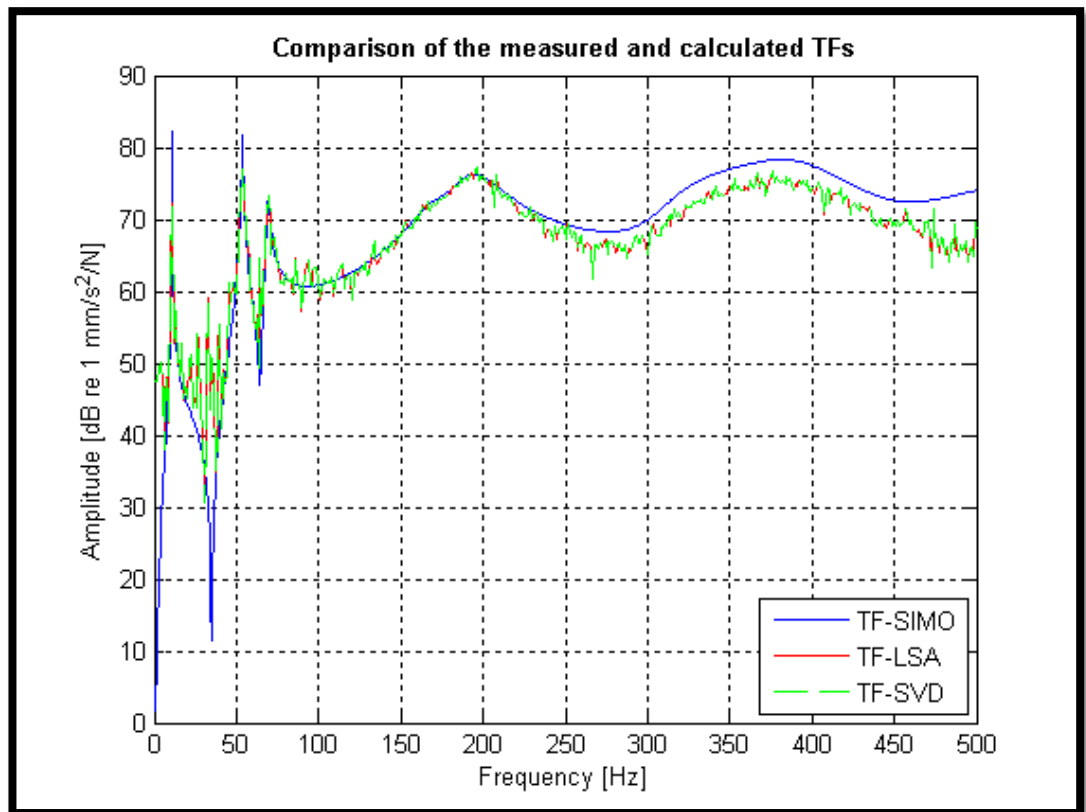


Figure 3.13 : TF comparison for $H_{11}(\omega)$ in the second numerical study.

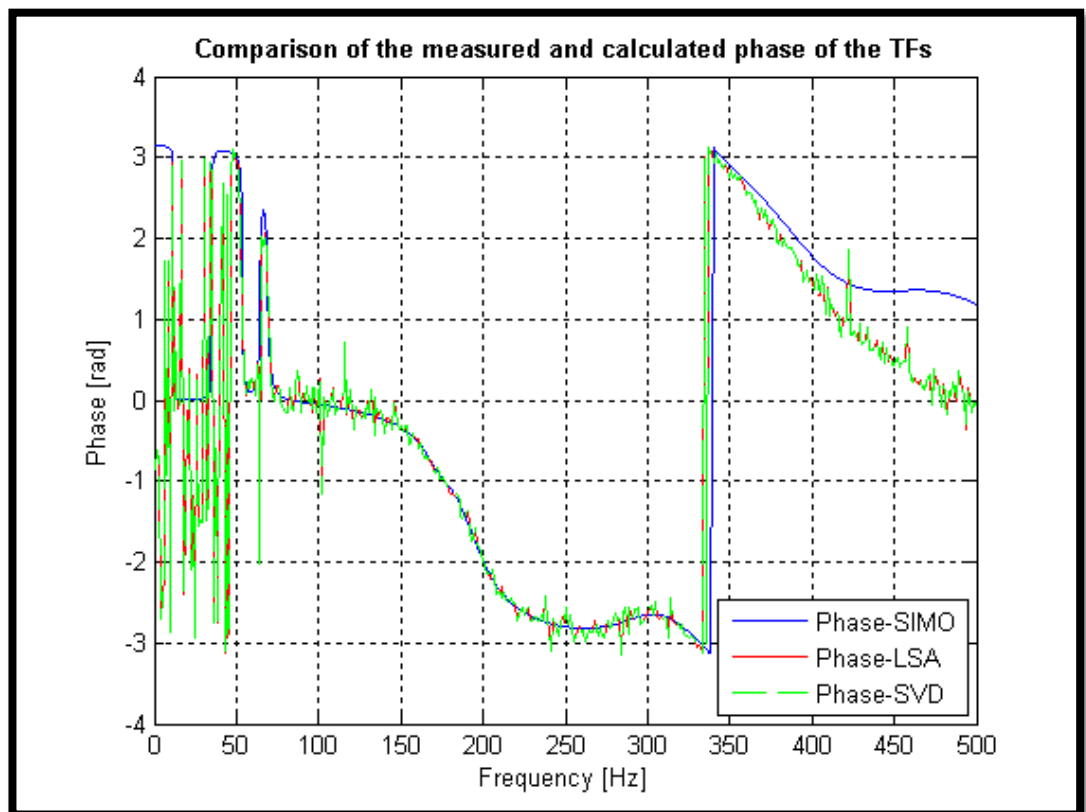


Figure 3.14 : Phase information comparison for the phase of $H_{11}(\omega)$ in the second numerical study.

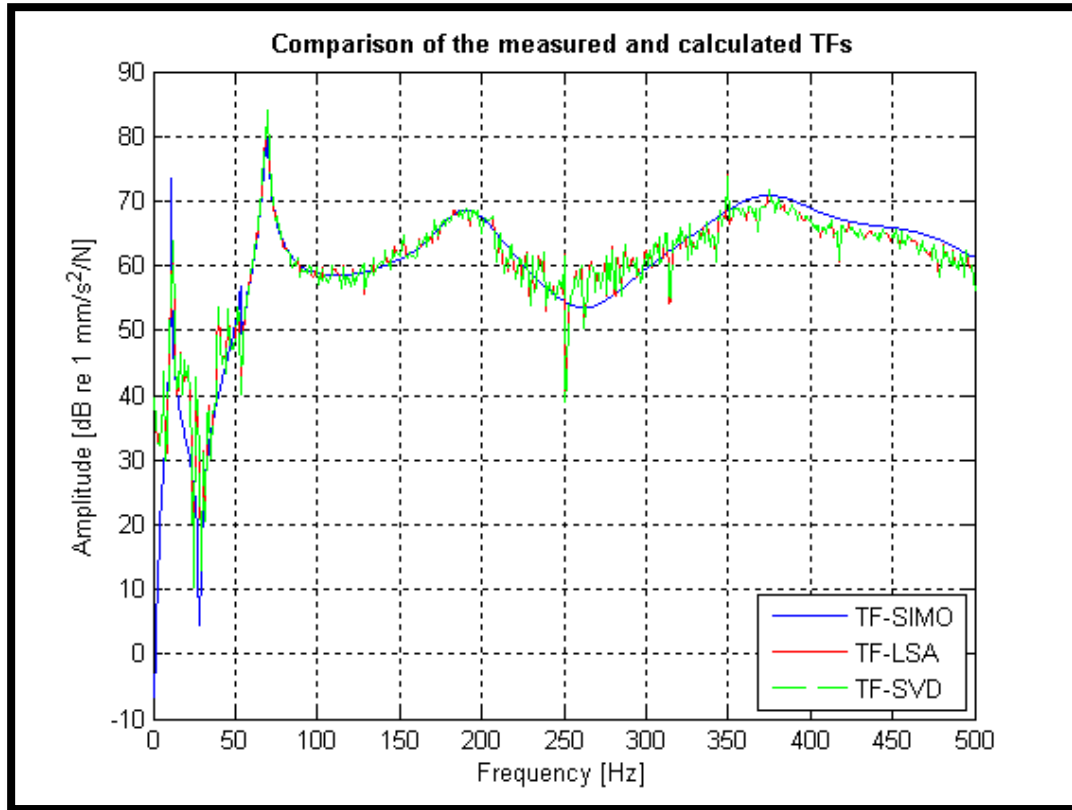


Figure 3.15 : TF comparison for $H_{22}(\omega)$ in the second numerical study.

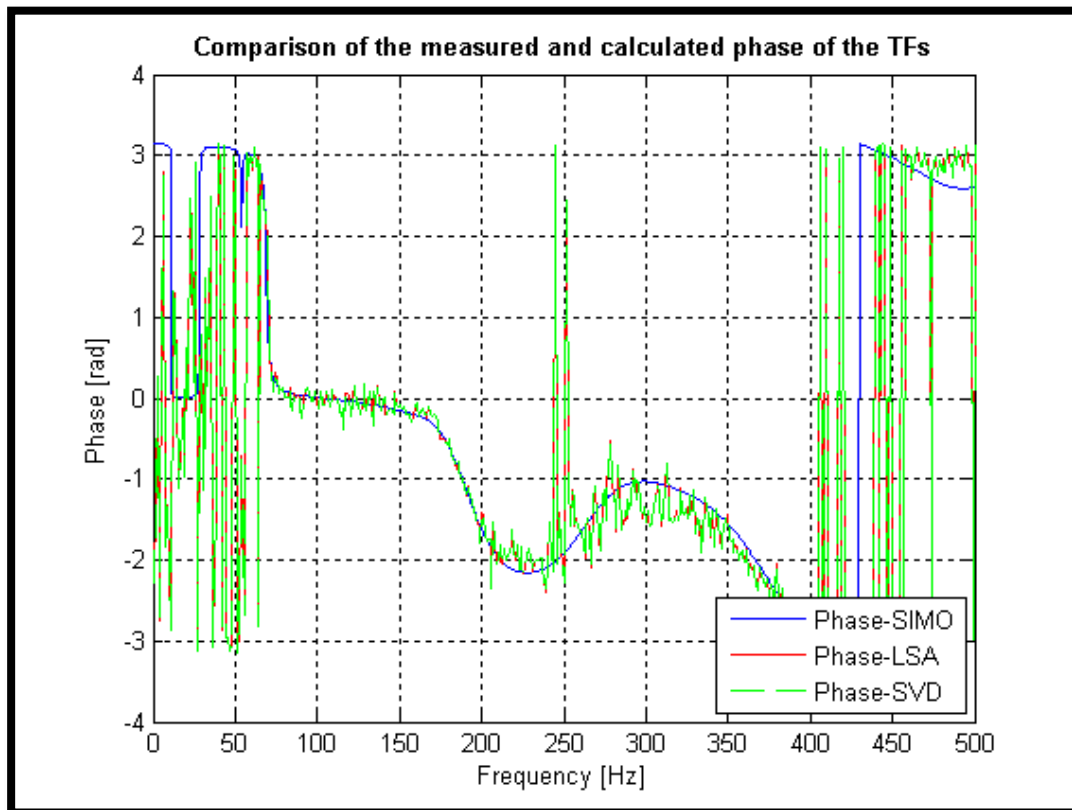


Figure 3.16 : Phase information comparison for the phase of $H_{22}(\omega)$ in the second numerical study.

3.1.5 Numerical case study III

The results of the previous numerical studies so far indicate that LSA and SVD approaches yield the same result when they are used to inverse the input matrix. For this reason, only LSA is considered for the comparison purposes in this numerical case study. In this case, the same procedure is followed as the first and second numerical case studies for the estimation of operational TF matrix and overall structural response computation. The numerical model, input and output locations used in the second case study are also utilized in this case. The difference in this third case study is that, linear averaging is implemented to the "measured" signals in the OTPA algorithm so as to minimize noise. Linear averaging is used to make a positive influence in the measurement Signal to Noise (S/N) ratio and smooth noisy oscillations in the signal [80]. Linear averaging is applied to the Auto Power Spectrums (APS) and Cross Power Spectrums (CPS) of the "measured" signals. Sample results presented in Figures 3.17 and 3.18 confirm that linear spectrum averaging yields less noisy and the most reliable MIMO FRF estimations.

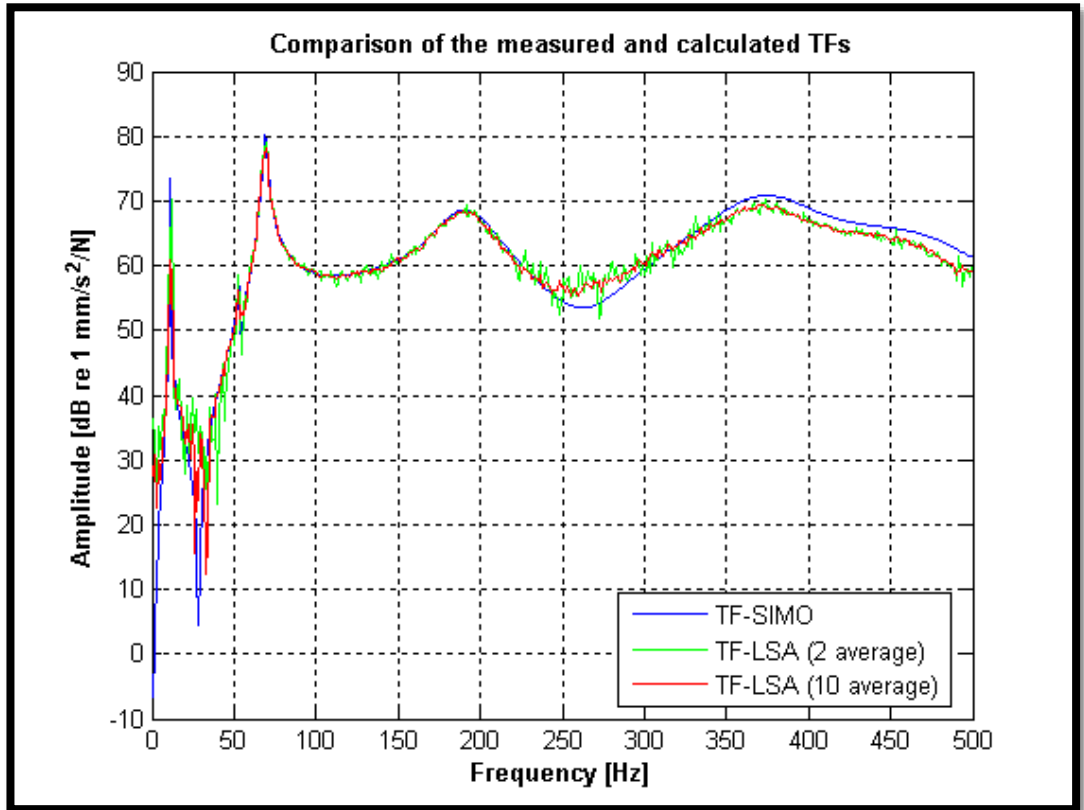


Figure 3.17 : TF comparison for $H_{22}(\omega)$ in the third numerical study.

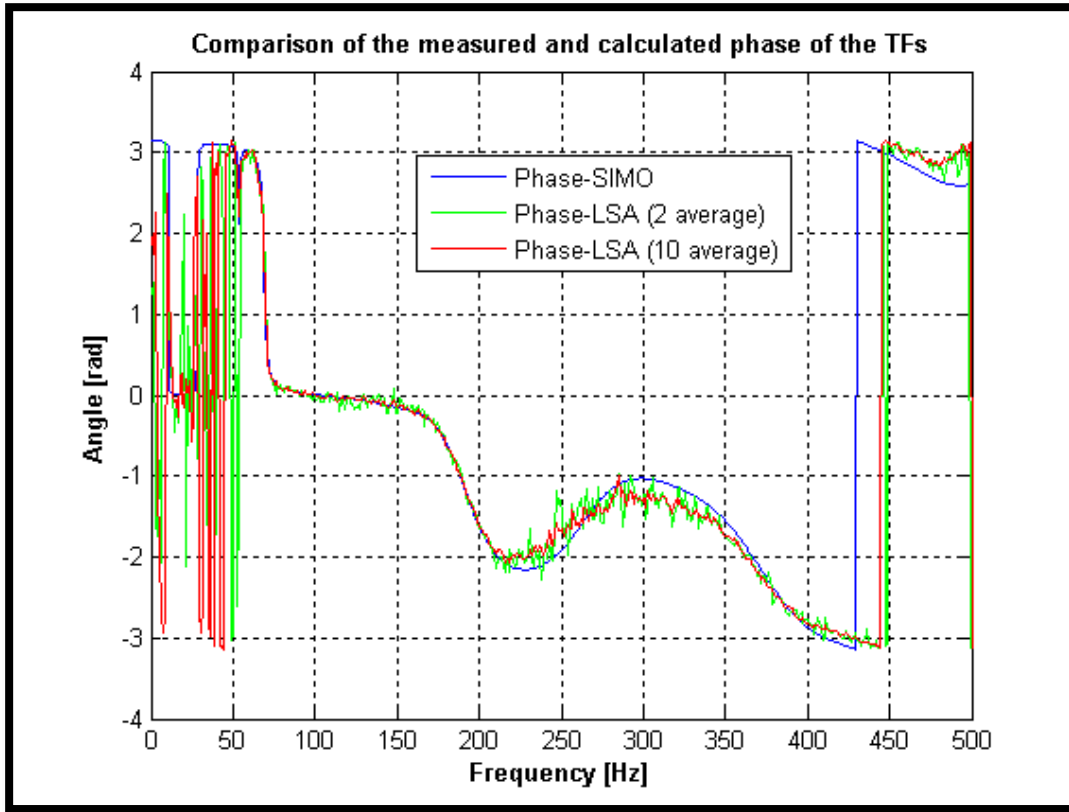


Figure 3.18 : Phase information comparison for the phase of $H_{22}(\omega)$ in the third numerical study.

3.2 Experimental Studies

The purpose of this section of thesis is to verify the implementation of the OTPA procedure by comparing experimentally measured and estimated outputs, TFs and their phase angles. In this section, a flowchart describing the experimental study is firstly introduced. Then, the experimental setup and the data acquisition system are presented. Finally, the need for using the Phase Assigned Autospectrums (PAS) and two experimental applications of OTPA are described.

3.2.1 Flowchart for OTPA procedure for experimental applications

At the beginning of the experimental studies, the structure to be tested is selected. Input (reference) locations, where the operational excitation loads are to be applied, and output (response) locations, where the structural responses are to be measured, are defined on the structure. Considering SIMO model assumption, the single excitation is applied to the input location and structural responses are measured as accelerations at related output locations. Hence, the FRFs are obtained between the selected input and output points (co-ordinates). Then, MIMO model is used to

estimate TFs, i.e., multi operational forces are simultaneously applied to the input locations and the corresponding structural responses are measured as acceleration in time-domain. To transform the time-domain data into frequency domain, FFT operation is carried out for the excitation forces and measured responses via signal analyzer. Later on, the acquired three types of data (FRFs, applied operational input forces and the measured structural responses) are imported into MATLAB environment to process the data according to OTPA algorithm [77]. As a start of the OTPA algorithm, the stored text files are read by the MATLAB code. Input and output matrices are generated for each frequency component. LSA and SVD techniques are utilized to take the inverse of input matrix, and then it is multiplied with the measured output matrix to determine the operational TF (FRF) matrix. This process made it possible to compare the FRFs and their phase angles obtained using SIMO model with those obtained using MIMO model, which in turn made it possible to validate the OTPA implementation in MATLAB. Furthermore, the operational TF matrix is multiplied with the measured input matrix to synthesize overall output matrix. Hence, the measured structural responses can also be compared and matched with these synthesized overall responses to verify the OTPA procedure. The flowchart of the experimental study is schematically presented in Figure 3.19.

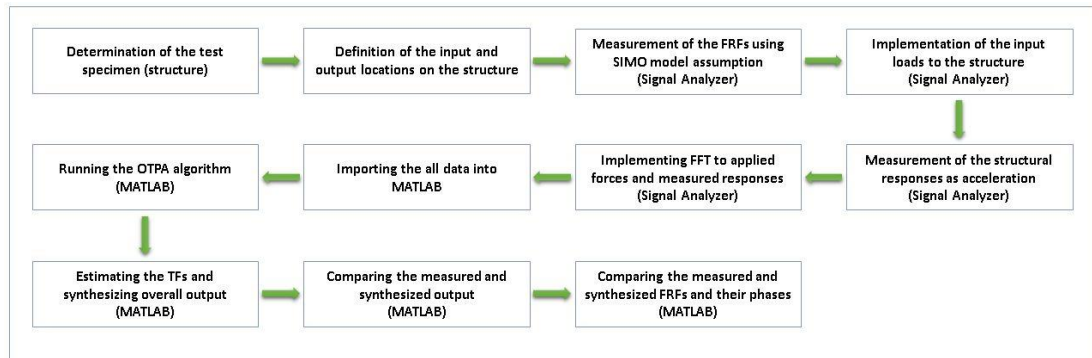


Figure 3.19 : Flowchart representation of the OTPA procedure for experimental studies.

3.2.2 Experimental setup

The rest of this section deals with the OTPA procedure applied to a simple flat plate that has four rectangular holes. The experimental setup is presented in Figure 3.20. The test structure is a steel flat plate with the size of 800 mm x 500 mm x 3 mm. The tests are carried out in free-free boundary conditions. Free-free boundary condition is frequently utilized in modal testing to identify the dynamic behavior of structures

[81, 82] without the need to deal with the difficulties in repeating the clamping boundary conditions. The test structure is hanged from two top corners via very thin flexible plastic rope on a test rig so as to simulate free-free conditions.

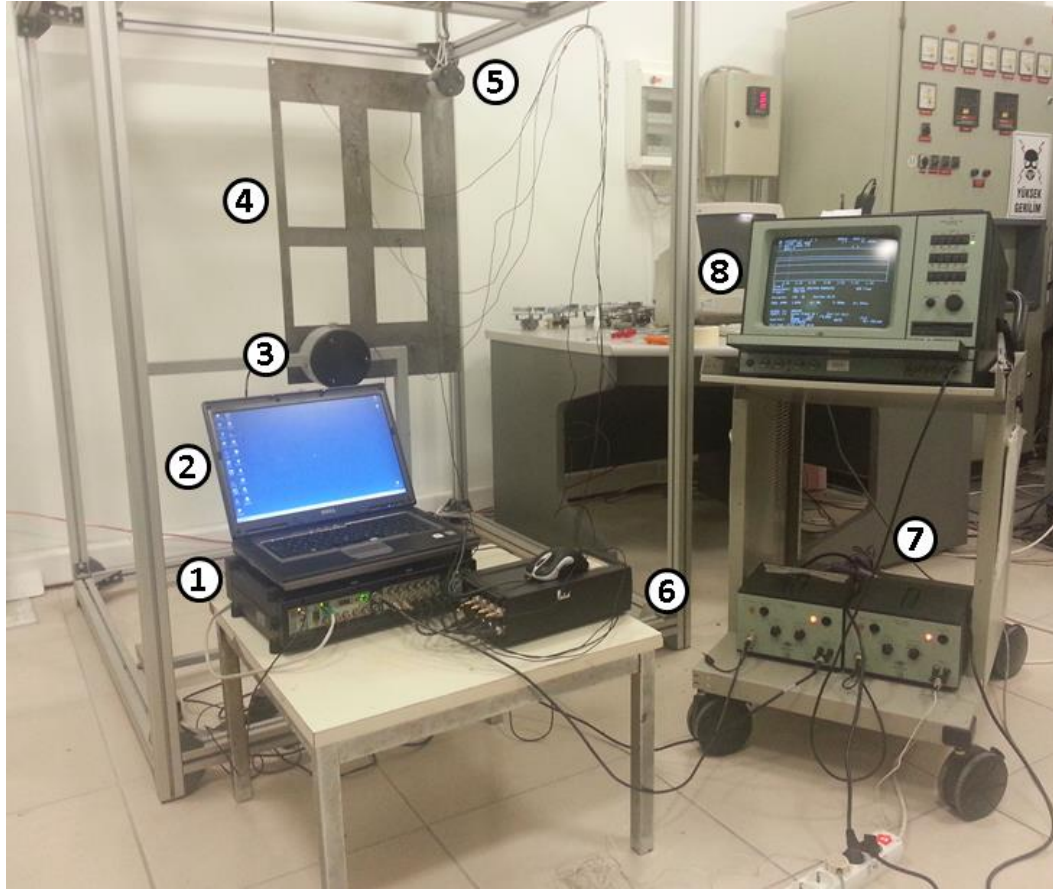


Figure 3.20 : Experimental setup 1) signal analyzer1 2) computer 3) shaker1 4) flat plate 5) shaker2 6) conditioning amplifier 7) power amplifier 8) signal analyzer2.

The responses of the flat plate are described in terms of acceleration (m/s^2). Measurement of the structural response as acceleration of the structure is made by using piezoelectric charge accelerometer. Brüel & Kjaer piezoelectric charge accelerometer Type 4393 [83] is rigidly mounted onto the defined response points on the structure via magnetic side connector (M3 receptacle) for output connection. Its sensitivity is 3.1 pC/g and its frequency response range is 0.1 to 16500 Hz . This piezoelectric accelerometer is a charge type; hence its sensitivity is described as charge per unit acceleration (pC/g) [83]. Owing to the use of piezoelectric sensor, a charge conditioning amplifier is required to convert the charge output into a voltage. Brüel & Kjaer NEXUS charge conditioning amplifier Type 2692-A [84] is attached to the accelerometer for this reason. It has an integrator to generate a voltage output

that is proportional to the integrated value of the input quantity [84]. The charge conditioning amplifier has four input and output modular channels that mean four accelerometers and four signal analyzer input connections can be attached at the same time. The piezoelectric charge accelerometer and the charge conditioning amplifier are demonstrated in Figure 3.21 (a) and 3.21 (b) respectively.

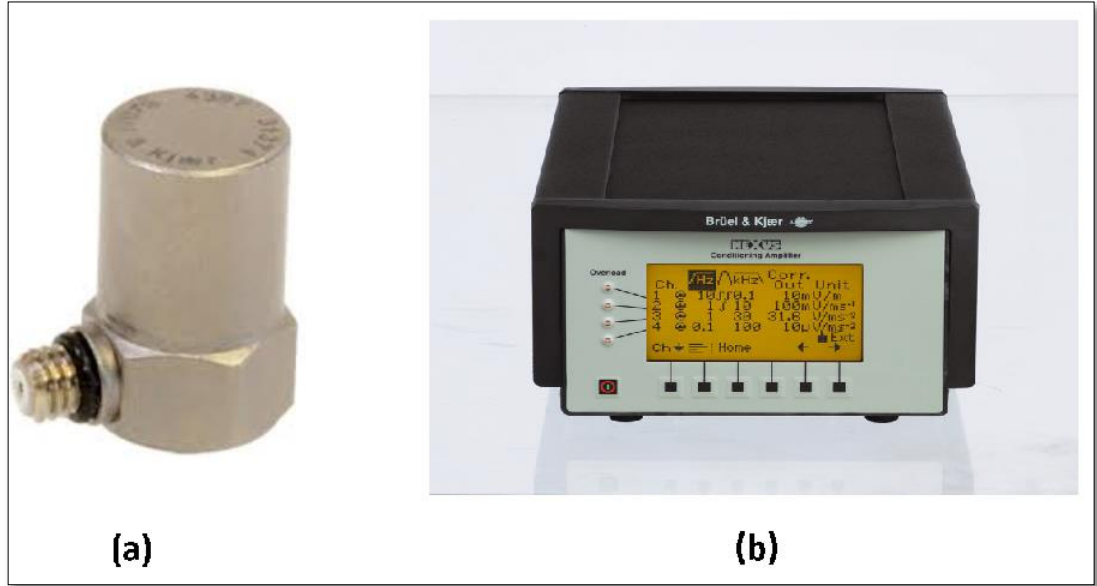


Figure 3.21 : (a) Brüel & Kjaer piezoelectric charge accelerometer Type 4393 [83].
(b) Brüel & Kjaer NEXUS charge conditioning amplifier Type 2692-A [84].

The applied input excitation forces are to be measured in force unit (N). For the application of the operational forces to the defined input locations, two shakers are utilized to excite the structure. Brüel & Kjaer vibration exciter Type 4809 [85] is the first shaker that is called as Input 1 for the rest of thesis. Its force rating is 45 N, it has 10 Hz to 20 kHz excitation frequency range, and also its first axial resonance is 20 kHz [85]. Brüel & Kjaer vibration exciter Type 4810 [86] is the second shaker that is called as Input 2 for the rest of thesis. It is an electrodynamic mini-shaker, its force rating is 10 N, has DC to 18 kHz frequency range, and also its first axial resonance is 18 kHz [86]. Thus, both shakers are suitable to excite structure in the frequency range of interest (0-400 Hz) for the experimental studies included in this section. Measurement of the applied excitation input forces is made by using force transducer. Brüel & Kjaer force transducer Type 8230-002 [87] is directly placed between the test structure and shaker to measure excitation forces in axial direction. The mentioned force transducer has integral electronics to measure dynamic, short-

length static and impact forces [87]. In addition, the connection between the force transducer and the modal shaker is provided by a stinger to remove the influences of cross-axis force inputs and bending moments [88]. The two shakers and the force transducer are illustrated in Figure 3.22 (a), 3.22 (b), and 3.22 (c), respectively.

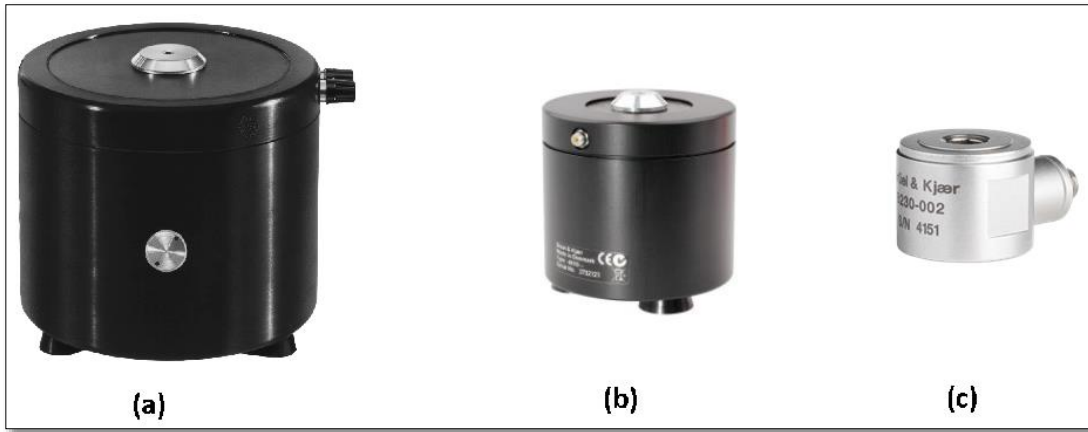


Figure 3.22 : (a) Brüel & Kjær vibration exciter Type 4809 [85]. (b) Brüel & Kjær vibration exciter Type 4810 [86]. (c) Brüel & Kjær force transducer Type 8230-002 [87].

The electrodynamic shakers transform the consecutive signal into an oscillating movement by changing the position of the shaker armature. Inside of the shaker, there is a coil buried in a permanent magnetic field. The current passes through the coil and generates electrodynamic forces. The referred motion is produced via these electrodynamic forces [88]. Therefore, a signal generator is required to produce the desired signals and a power amplifier is also essential to control and amplify the produced signals. Brüel & Kjær generator and sampling module Type 3106 of Brüel & Kjær signal analyzer unit Type 3560C [89] is used as the first signal generator for Input 1, and Brüel & Kjær 5/1 channel input/output controller module type 7540 of Brüel & Kjær signal analyzer unit Type 2035 [90] is utilized as second signal generator for Input 2. On the other hand, some vibration exciters might be directly actuated using the signal from a signal generator. However, the power output of such exciters may not be sufficient for general applications. Therefore, Brüel & Kjær power amplifier Type 2706 [91] is preferred to drive the shakers by amplifying the generated signal power. The employed two signal analyzer units and power amplifier is demonstrated in Figure 3.23 (a), 3.23 (b), and 3.23 (c), respectively.

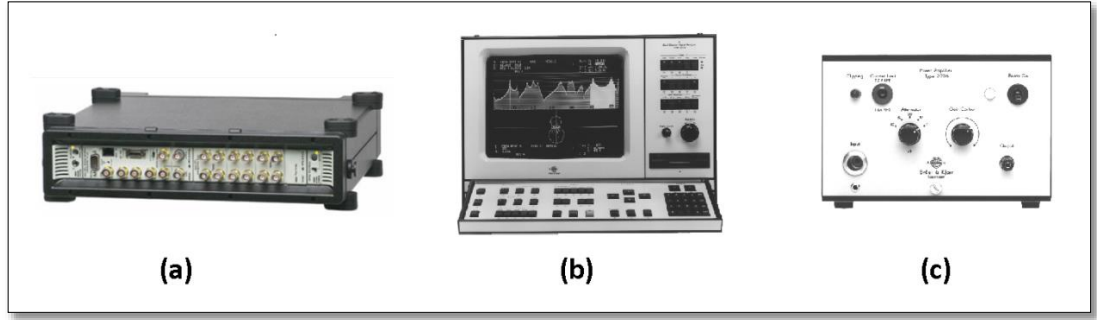


Figure 3.23 : (a) Brüel & Kjaer signal analyzer unit Type 3560C [89]. (b) Brüel & Kjaer signal analyzer unit Type 2035 [90]. (c) Brüel & Kjaer power amplifier Type 2706 [91].

Furthermore, Brüel & Kjaer signal analyzer unit Type 2560C [89] and its software PULSE LabShop [92] are used to process the measured FRFs, operational forces and corresponding structural responses. PULSE LabShop is a data acquisition and signal processing software that can execute fundamental vibration analysis such as FFT analysis, envelope analysis and cepstrum analysis [92]. FFT of vibration and force measurements, APS, CPS, and PAS computations are performed by this signal analyzer and its software.

3.2.3 Use of phase-assigned autospectrums

The definition of a linear system pattern including a group of chosen input and output quantities can be already made as [3.2-3.5]:

$$H(j\omega)x(j\omega) = y(j\omega) \quad (3.1)$$

where $(j\omega)$ defines the dependency on frequency. Hence, all quantities given in Eq. 3.1 carry phase information and the whole measurements should contain complex type data. In the implementation of the OTPA procedure, it should be noted that the phase relationship between various vibro-acoustic transfer ways is as important as the magnitude information. For instance, two different vibro-acoustic transmission paths may have higher individual contributions, but the total overall response might be lower than the individual contributions due to contributions being out of phase with respect to each other [40].

In the numerical studies of this thesis, it is easy to obtain frequency domain data containing complex type values because of utilizing the amplitude spectrums. Even if

linear averaging is implemented to the application, the phase information between input and output quantities can be protected via the use of Cross Power Spectrums (CPS). In the experimental studies of the thesis, phase information is also required between input (excitation) and output (responses) coordinates. This information is obtained by transferring the Phase-Assigned Autospectrums (PAS) from the PULSE LabShop software.

The Phase-Assigned Autospectrum (PAS) is a complex spectrum computed utilizing a signal and a reference signal. Consider a signal X and a reference signal Y , the PAS for signal X is calculated as [93]:

$$PAS_{XX} = G_{XX} \frac{G_{YX}}{|G_{YX}|} \quad (3.2)$$

where G_{XX} defines the Auto Power Spectrum (APS) of the signal X , G_{YX} defines the Cross Power Spectrum (CPS) of the reference signal Y and signal X , and PAS_{XX} defines the Phase-Assigned Autospectrum of signal X [93]. The phase assigned spectrum is also used in Operational Deflection Shape (ODS) analysis [94, 95].

At this stage of the experimental studies, FRF measurements of the flat plate given in Figure 3.20 are performed. The structure is excited with a modal hammer at Input 1 location and the corresponding responses are measured as acceleration at three arbitrary output locations, so that SIMO approach can be used to obtain the FRFs. The same process is repeated five times for spectrum averaging purposes, thus the resulting measured data is the linear average of these five measurements. The FRFs are computed by the signal analyzer using H1 estimator. For the input signal F and the output signal A , the frequency response of the system $H_1(\omega)$ is calculated using H1 estimator as [93]:

$$H_1(\omega) = \frac{G_{FA}(\omega)}{G_{FF}(\omega)} \quad (3.3)$$

where $G_{FF}(\omega)$ defines the input Auto Power Spectrum (APS) and $G_{FA}(\omega)$ defines the Cross Power Spectrum (CPS) between the input signal F and the output signal A [93].

Although, any one of the channels can be chosen as the reference signal for phase assignment, the output transducer (i.e. accelerometer) is chosen here for this purpose. Thus, the first accelerometer signal is chosen as the reference signal to calculate the PASs. Two kinds of FRFs are obtained from the forced vibration test of the structure using a modal hammer. In the first kind, APSs and CPSs were employed to compute FRFs described in Eq. 3.3. For the second kind, the phase-assigned APSs and CPSs were initially calculated using Eq. 3.2, later on the FRFs were obtained from this power spectrums utilizing Eq. 3.3. Finally, the computed FRFs were compared with each other in the frequency range of interest (0-200 Hz). The corresponding FRFs and their phase information are compared in Figures 3.24 and 3.25 respectively.

The result of the comparison indicates that PAS can safely be used to compute FRFs of a linear(ized) system. Additionally, the use of PAS provides data that includes complex type values which carries phase information and reduces the post-processing time of the OTPA procedure implemented in MATLAB in this thesis.

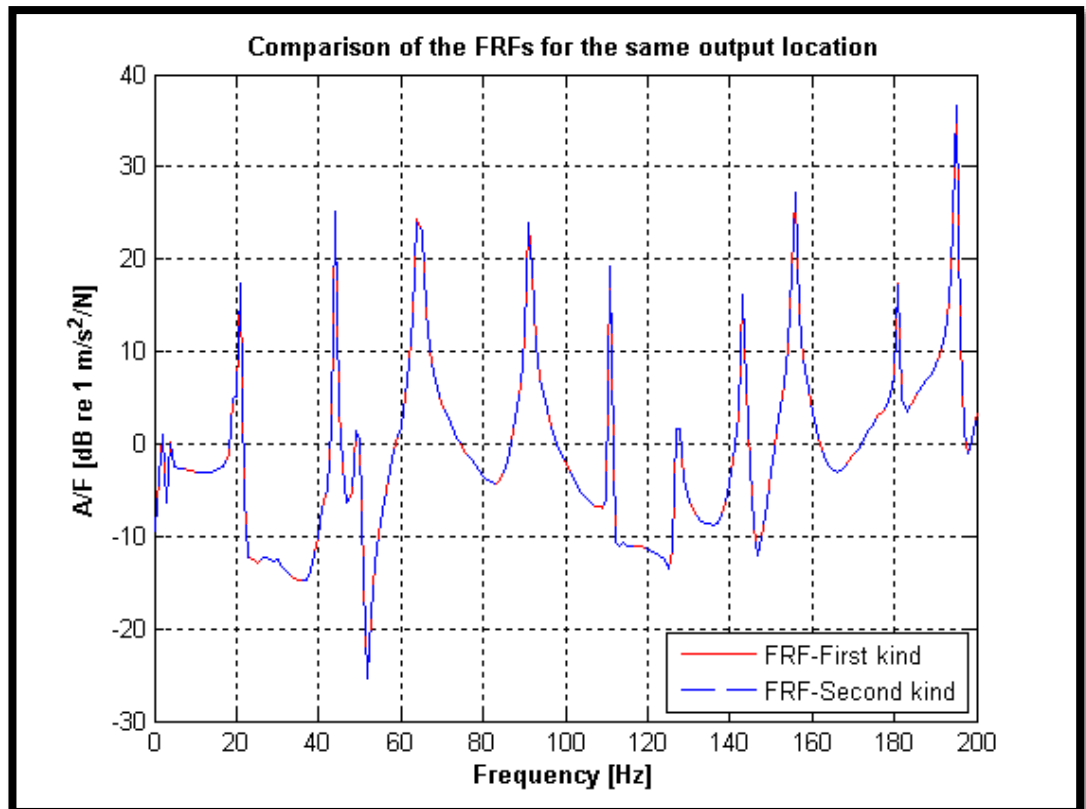


Figure 3.24 : Comparison of the FRFs to represent the use of PAS.

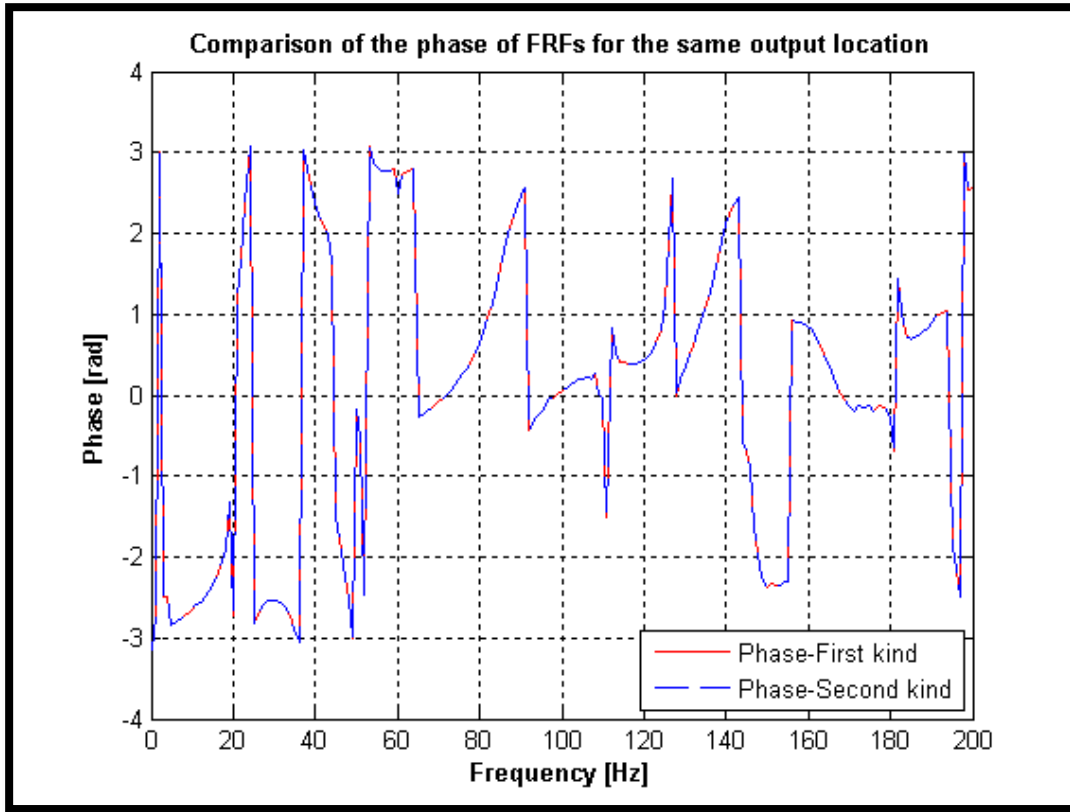


Figure 3.25 : Comparison of the phase of FRFs to represent the use of PAS.

3.2.4 OPTA of a flat plate under free-free boundary conditions

The aim of the experimental applications of OTPA to a simple flat plate, as mentioned before, is the validation of the OTPA procedure by comparing the estimated results with the measured data. The details of the experimental setup and type of stored data that will be utilized into the OTPA algorithm are described in the previous sections. The flat plate with rectangular holes is suspended on the test rig with flexible ropes to simulate the free-free boundary conditions. Two input locations and four output locations are selected on the structure. The required operational forces are applied to the input locations via the implementation of the shaker excitation and the structural responses are measured at the defined output locations using the charge accelerometers. The defined input locations, the attachment of the shakers and the positions of the accelerometers are shown in Figure 3.26. These locations will be used in all experimental applications performed on this simple plate in this section. For the rest of this section, the input and output quantities are denoted as $F_r^{(m)}$ and $A_r^{(n)}$ respectively where m is the number of references

(input-applied operational forces), n is the number of responses (output-measured structural responses), and r is the number of operational conditions.

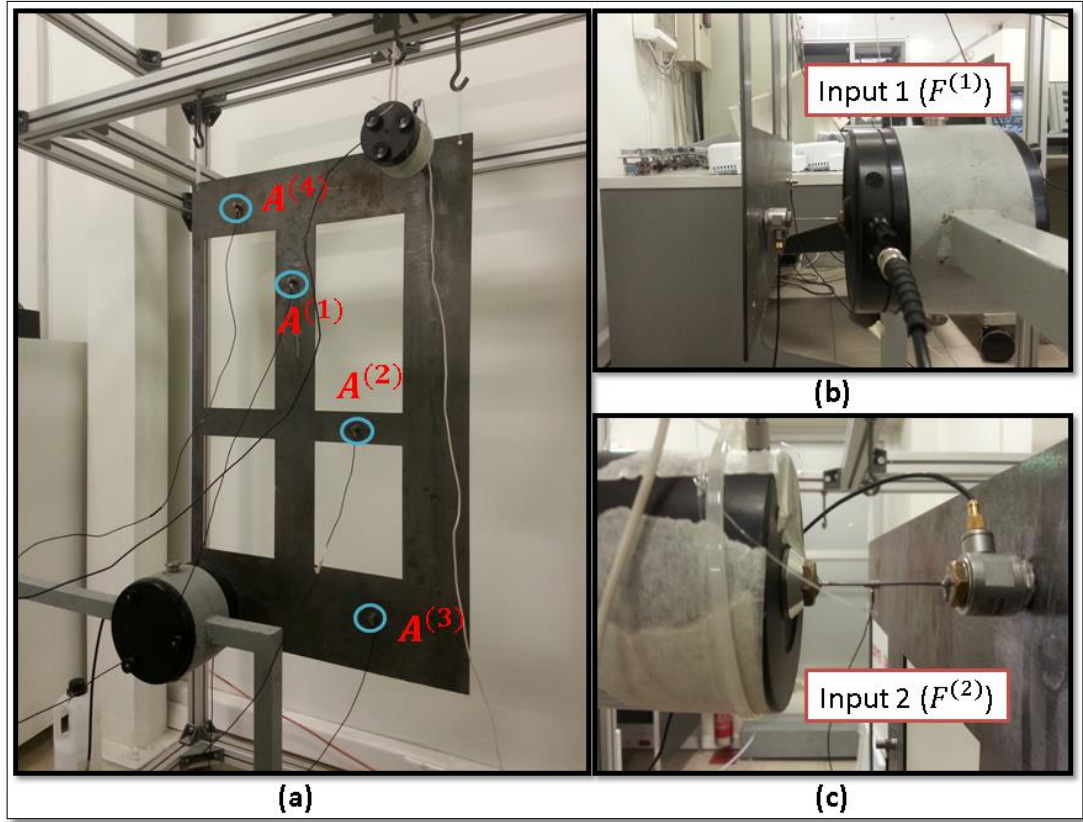


Figure 3.26 : (a) Defined output locations ($A_r^{(n)}$). (b) Attachment of the first shaker (Input 1). (c) Attachment of the second shaker (Input 2).

As a first step, the FRFs are estimated using the SIMO model assumption. A random signal generated by the first signal generator is fed to Input 1 to excite the structure, and the corresponding responses are measured at the output locations as acceleration at the same time. The same process is repeated for the Input 2, and then the SIMO FRFs are computed by the signal analyzer and stored to compare them with the operational TFs (or FRFs) that will be estimated via the OTPA algorithm.

In the second step, the aim is to determine the FRFs by implementing the OTPA algorithm which is based on MIMO model. It is impossible to execute the OTPA with one measurement performed under only one operational condition. For a mechanical system containing m input and n output quantities, the number of operational conditions r must be greater than the number of references m to estimate the TF matrix utilizing OTPA algorithm [11, 43, 70, 71]. There are two input

excitation points assigned on the structure, so that more than three various operational conditions are required to avoid ill-conditioning.

As an information for this section of third chapter, there are two types of experimental studies performed on the simple flat plate, given in Figure 3.26 (a). In the first experimental case study, the signal type is selected as pseudo-random signal for input excitation. In the second experimental case study, random type excitation signal is chosen as signal type.

In the first case, the simple flat plate is simultaneously excited via two shakers by applying the pseudo-random signals and corresponding structural responses are measured at the same time for five different operational conditions. Thus, it can be considered that the operational input forces are made linearly independent to each other owing to use of pseudo-random signal excitation type with alternating amplitude levels for each operational condition. The amplitudes of the implemented excitation signals are changed in every operational condition where the details of applied operational forces are given in Table 3.1.

Table 3.1 : Details of the operational conditions for the first experimental application on the plate.

# of Operational Condition	Signal Type	Excitation Frequency Range [Hz]	First Shaker (INPUT 1)	Second Shaker (INPUT 2)
			Amplitude Level [Vrms]	Amplitude Level [V]
1	Pseudo-Random	0 - 1600	0.1	1
2	Pseudo-Random	0 - 1600	0.2	0.9
3	Pseudo-Random	0 - 1600	0.3	0.8
4	Pseudo-Random	0 - 1600	0.4	0.7
5	Pseudo-Random	0 - 1600	0.5	0.6

First, the input and the output signals are acquired in time-domain by linear averaging 100 times considering an overlap value of % 66.7. Subsequently, the FFT operation is carried out to the averaged signals and the phase-assigned spectrums are computed using Eq. 3.2 by selecting the Input 1 signal as the reference signal. As a result, input and output signals are obtained in frequency-domain within 0-800 Hz frequency range with a frequency resolution of 1 Hz. All the calculations up to here are made by the signal analyzer. A MATLAB code developed in this thesis is used to read these frequency domain data and to perform the OTPA algorithm. The linear relationship between a set of assigned input and output quantities is assumed as it is stated in Eq. 2.15. The extended form of this relationship can be represented using

Eq. 2.20 for each frequency component. The operational TF matrix is estimated by multiplication of the inverse of input matrix and the output matrix, where LSA and SVD methods are utilized to obtain the inverse of the input matrix as it is given in Eq. 2.25 and Eq. 2.32, respectively. The overall responses are computed by multiplying the measured input matrix with the estimated TF matrix according to Eq. 2.37. At the end, the required data for comparison and verification the OTPA procedure are obtained via the MATLAB code developed in this thesis.

The first set of experimental results indicates that the measured structural responses are matched well with the computed (synthesized) responses. The measured and synthesized results for one of the outputs are compared in Figure 3.27 which shows excellent agreement. Furthermore, comparisons of the SIMO FRFs with those estimated via MIMO procedure within the MATLAB code developed show that MIMO FRF estimation procedure yields good TFs as depicted in Figures 3.28 and 3.29, where one of the FRF and their phase angles are compared. Moreover, it can be said that LSA and SVD method yield the same results in the process of input matrix inversion.

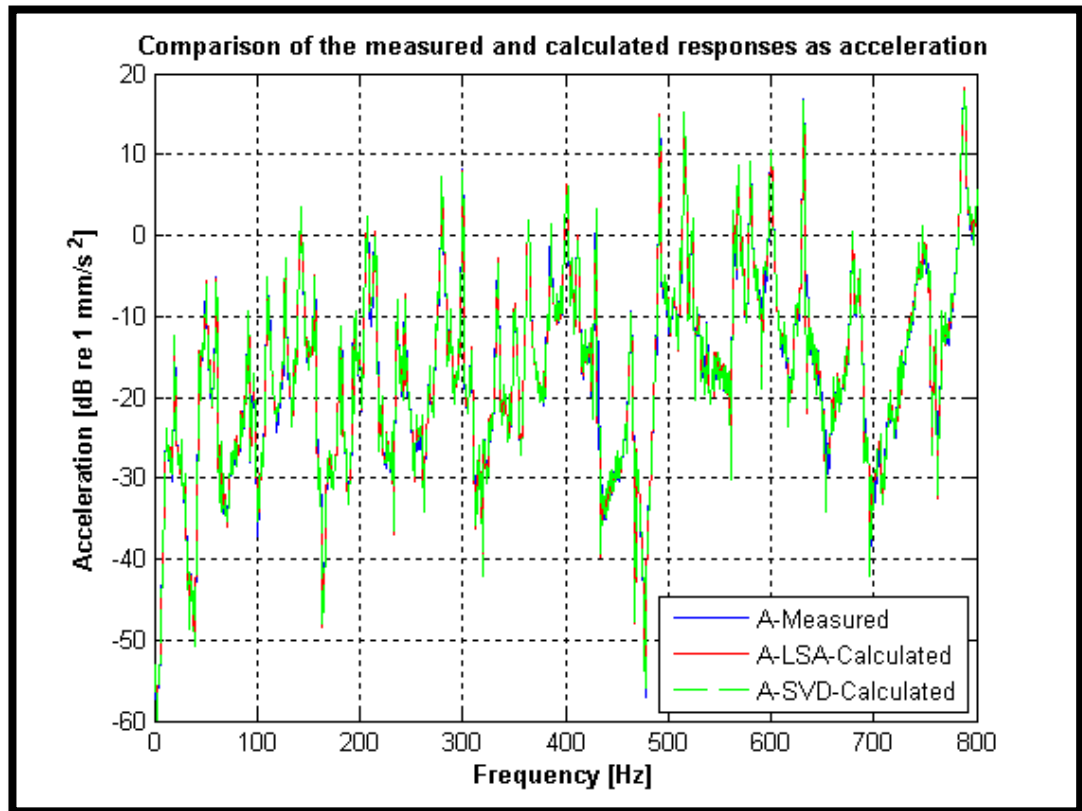


Figure 3.27 : Output comparison of the second response location for the fourth operational condition $A_4^{(2)}(j\omega)$ in the first experimental study.

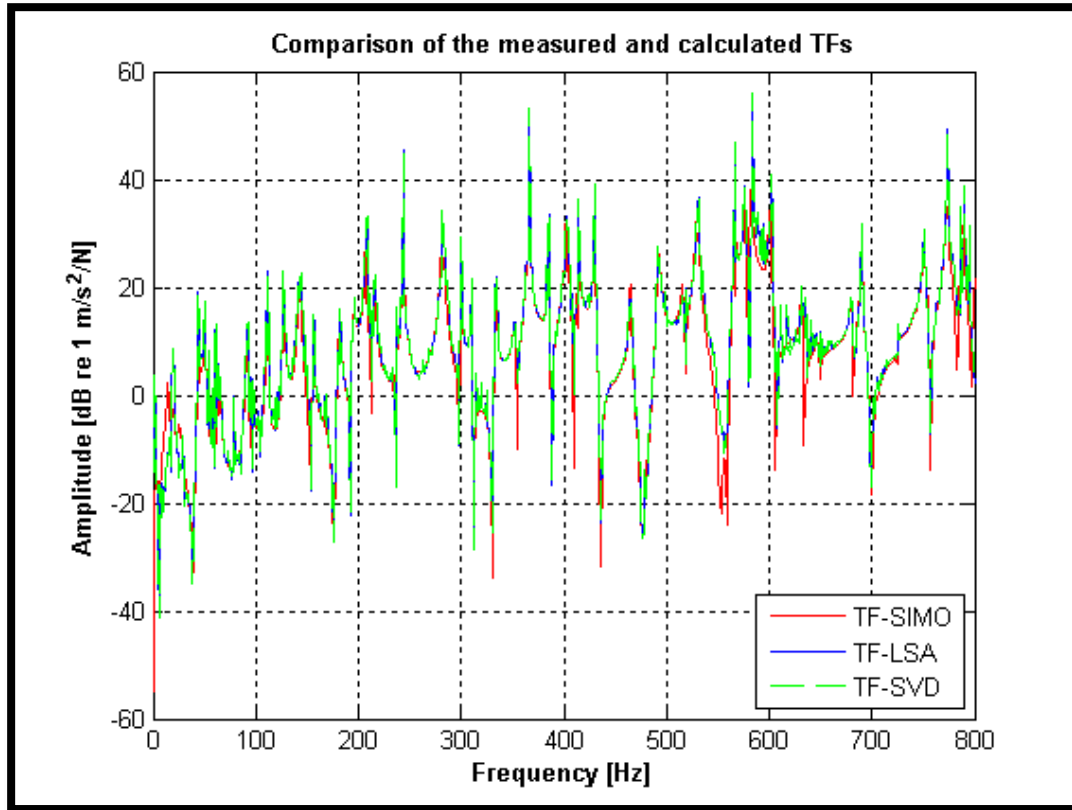


Figure 3.28 : TF comparison for $H_{12}(\omega)$ in the first experimental study.

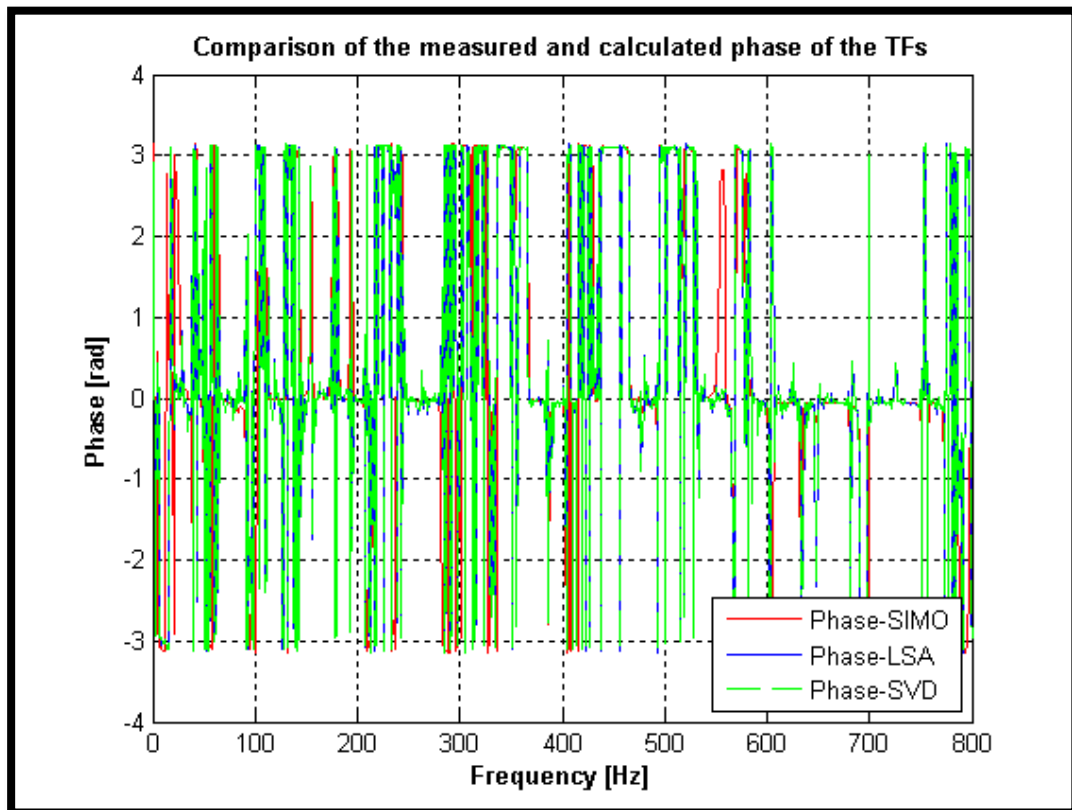


Figure 3.29 : Phase information comparison for the phase of $H_{12}(\omega)$ in the first experimental study.

In the second case, the structure, simple flat plate, is simultaneously excited via two shakers by using random (white noise) signals and corresponding structural accelerations are measured at the same time for eight different operational conditions. The amplitudes of the applied excitation signals are changed in every operational condition where the details of applied operational forces are represented in Table 3.2. This is to ensure that the matrix to be inverted shall not be singular and to minimize the undesirable effects of noise as much as possible.

Table 3.2 : Details of the implemented operational conditions for the second experimental application on the plate.

# of Operational Condition	Signal Type	Excitation Frequency Range [Hz]	First Shaker (INPUT 1)	Second Shaker (INPUT 2)
			Amplitude Level [Vrms]	Amplitude Level [V]
1	White Noise	0 - 1600	1.4	1
2	White Noise	0 - 1600	1.3	0.9
3	White Noise	0 - 1600	1.2	0.8
4	White Noise	0 - 1600	1.1	0.7
5	White Noise	0 - 1600	1	0.6
6	White Noise	0 - 1600	0.9	0.5
7	White Noise	0 - 1600	0.8	0.4
8	White Noise	0 - 1600	0.7	0.3

The same procedure as in the first case of experimental study is implemented to the second case; but this case, Hanning windowing is applied to the input and the output signals so as to minimize the leakage effects. Although the length of time record is adjustable in the signal analyzer, it is required to be chosen from a group of predefined values. Since random signals are generally nonperiodic in the predefined measurement time periods, a window must be implemented to set aright the leakage to obtain the correct FFT signal amplitude level. Hence, a Hanning window is applied to a random signal in the time-domain to reduce the influences of leakages to better represent the frequency spectrum of the data.

As expected, the second set of experimental results also confirms that the synthesized responses are almost the same as measured ones, as demonstrated in Figure 3.30. Furthermore, there is also a good match between the estimated operational TFs (or FRFs) and the measured FRFs via utilizing SISO or SIMO approach. This can be seen in Figures 3.31 and 3.32, where one of the FRF and their phase angles are compared. Once again, it can be said that LSA and SVD method yields the same results in the process of input matrix inversion.

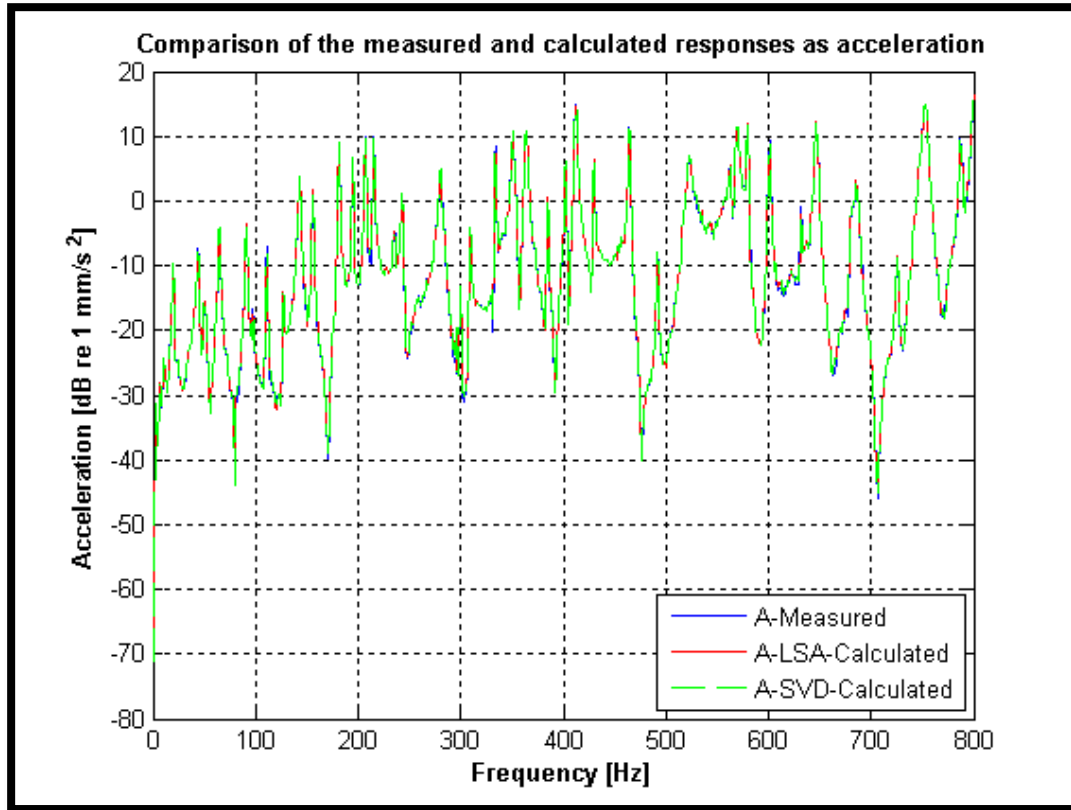


Figure 3.30 : Output comparison of the third response location for the sixth operational condition $A_6^{(3)}(j\omega)$ in the first experimental study.

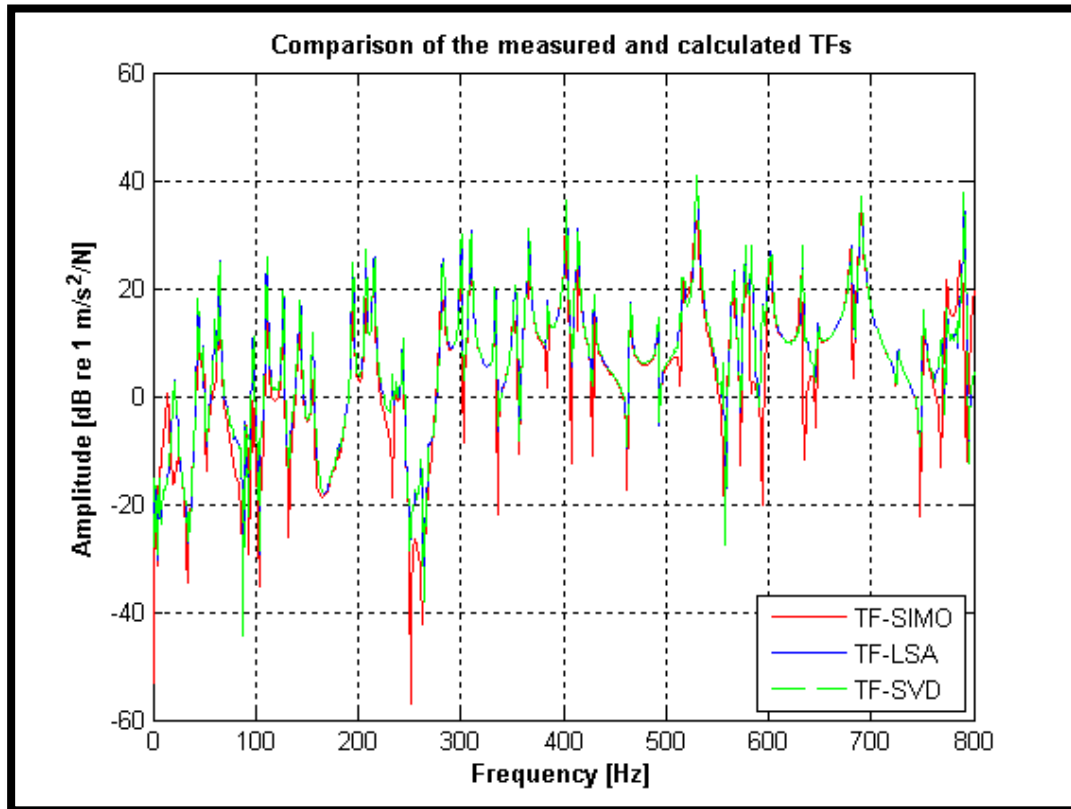


Figure 3.31 : TF comparison for $H_{11}(\omega)$ in the second experimental study.

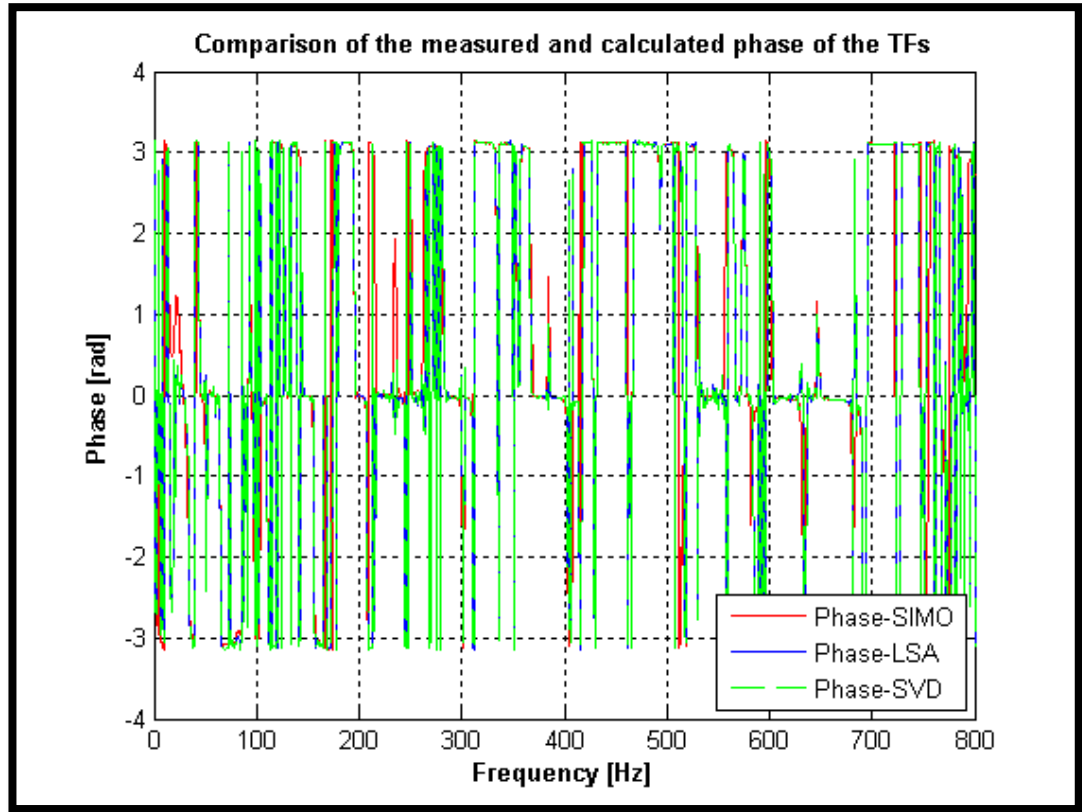


Figure 3.32 : Phase information comparison for the phase of $H_{11}(\omega)$ in the second experimental study.

3.2.5 Contribution analysis

Another main objective of the OTPA is to analyze contributions from various sources. The response signal, e.g. the accelerometer at any defined output location on the test structure, is separated in individual source contributions utilizing the OTPA method. The result of sample contribution analysis for an arbitrary output measurement location is illustrated in Figure 3.33.

For the analysis presented here, the operational TFs were estimated from the first experimental study on the flat plate, where the pseudo-random signal were used as input excitation loads. In Figure 3.33, the green line shows the total synthesized response on the selected measurement location. Furthermore, the red and blue lines show the contributions of Input 1 ($F^{(1)}$) and Input 2 ($F^{(2)}$), respectively for this receiver point. An overview of the contributions from the different inputs indicates that the highest computed contributions originate from Input 2 over the entire frequency range of interest (0-800 Hz), if compared by the amplitude frequency spectrum in logarithmic scale.

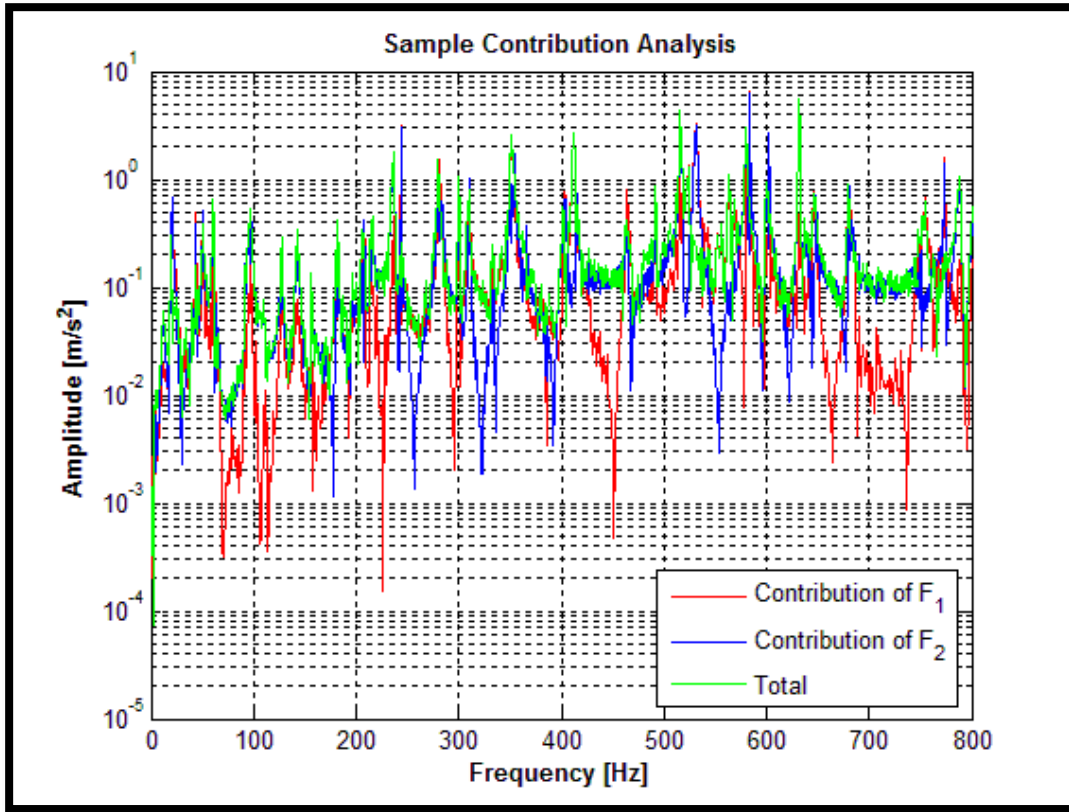


Figure 3.33 : Contributions to the selected output measurement location on the flat plate at an arbitrary operational condition with pseudo-random signal excitation, of the two different inputs as $F^{(1)}$ (red) and $F^{(2)}$ (blue).

3.3 Summary

The purpose of this chapter was to verify the MATLAB code developed for carrying out fundamental steps of the OTPA procedure. For this reason, numerical and experimental applications were performed on a simple flat plate. In both of the numerical and experimental studies, forced vibration analysis and dynamic analysis of the structure are made to obtain the required spectrums. A MATLAB code was developed in this thesis to process the OTPA and to compare the computation results with the measurement results as output, FRF and phase information. Linear spectrum averaging was used to reduce the noise level and to get more reliable FRFs from MIMO FRF estimations. The use of Phase-Assigned Autospectrum (PAS) was also demonstrated with an application to obtain complex type data. The main reason for using PAS was to preserve the phase information as well as to minimize the post-processing time. As expected, a good match between the synthesized and measured structural response was obtained. It is worth restating here that MIMO FRF estimation is essential for any OTPA procedure. The comparisons of the SIMO and

MIMO FRFs in this chapter verified that the MATLAB code developed in this thesis was capable of MIMO FRF estimations with acceptable accuracy.

4. APPLICATION OF THE OPERATIONAL TRANSFER PATH ANALYSIS TO A WASHING MACHINE

In this chapter, the Operational Transfer Path Analysis is employed for the case of a washing machine for a quantitative ordering of contributions from a main excitation source over various vibro-acoustic transmission paths to a receiver. For the analysis, source-transfer path-receiver model was defined on the machine to figure out the root causes of vibration problems. Modal testing of complete assembly is made so as to decide the output measurement locations on the receiver side. Vibration measurements on the vibration source are performed to determine the input sensor positions at the connection points. The selection of suitable sensor positions and operational conditions are explained; also the measurement setup is introduced. Several multichannel measurements including simultaneous data collection for structure-borne vibration in various operational conditions are carried out for the OTPA. The main source excitation is changed by altering the spinning cycle speed of the washing machine and total imbalance inside the washing machine drum. The Transfer Functions (TFs) are computed using Singular Value Decomposition (SVD) and Least Square Algorithm (LSA) considering transmissibility approach. Overall responses of panel vibrations of a washing machine under different excitations are also synthesized using computed TFs. The synthesized responses are compared with the original measurement results for the validation of OTPA procedure. Finally, contribution analysis is performed to decompose the contributions of vibration transfer paths in the frequency range of interest. The results are evaluated to analyze the contributions of main excitation source via the strongest transmission paths to the panel vibrations.

4.1 Establishing the Source-Transfer Path-Receiver Model

According to the basic principles of OTPA procedure, an accurate establishment of source-transfer path-receiver model is required for reliable decomposition of critical vibro-acoustic transmission paths and/or sources. Source is considered here as the

initial point of vibro-acoustic energy flow to the system. The vibro-acoustic energy is transmitted to other system parts via structure-borne and/or airborne transfer paths; hence the transfer paths can be stated as vibro-acoustic energy transmission paths. The corresponding responses generated from main source excitation are measured and identified at the receiver location. Thus, building up the correct source-transfer path-receiver model for the TPA is required for accurate modeling of the vibro-acoustic energy transmission in a system. Moreover, it provides accurate implementation of the TPA procedure to the system for quantifying the source powers and ranking the contributions of the several sources through the several transfer paths for each given receiver location.

In this part of thesis, test specimen is a conventional washing machine where the OTPA procedure will be applied to. The mentioned washing machine, as shown in Figure 4.1, contains a tub that contains some water during washing, a rotary drum that is located inside the tub to house laundry, and a motor that produce a driving force to rotate the drum. The drum rotates around a horizontal axis; hence the mechanical system is named as a front-loading washing machine [96]. The tub, the rotary drum and the motor are the members of the drive group of the washing machine. The drive group is divided into two subgroups as the tub group and the drum group. The tub group contains the tub assembly, a front counter weight and a top counter weight; where the drum group contains the rotary drum, a rotary shaft and flange members [97]. The drive group with a transmission unite located in a cabinet is also called as suspended mass in the literature. The suspended mass is connected to the front panel of the cabinet with a bellow, attached to the side panels of the cabinet by two coil springs and joined to the cabinet base by two damping units [98].

The washing machine executes a sequence of processes like washing cycle, rinsing cycle, spinning (spin-dry) cycle. When the rotary drum is rotated in inequilibrium condition during performing one of these processes (especially in spinning cycle), the laundry can be nonuniformly distributed [96]. Inequilibrium condition occurs if the geometric centre of the drum does not coincide with the gravity centre of the total mass inside the drum [98, 99]. In such cases, a force caused by unbalance is applied to the rotary shaft and it causes an eccentric motion of the drum, hence resulting in the vibration of the tub. Thereby, vibrations in a washing machine are generated by

the motion of the drive group (or suspended mass) during spinning cycle stage in the unbalanced state. These motions are transferred to the washing machine cabinet via the suspension system. In addition, the spinning cycle stage is the most irritating operational condition as it causes the most noise [98].

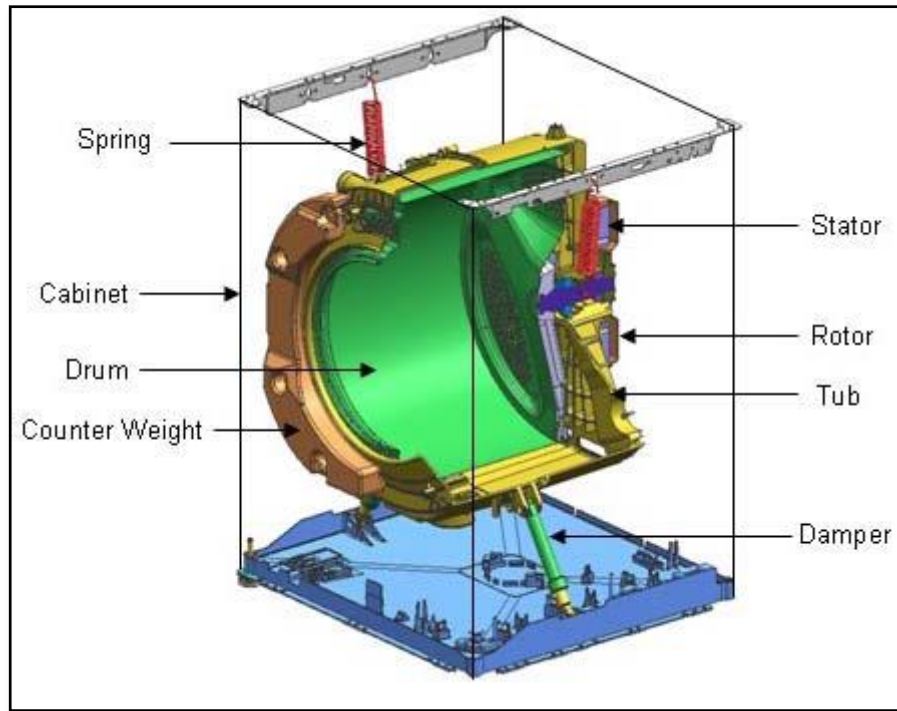


Figure 4.1 : Schematic view of drum type washing machine [100].

As a result, if one wants to establish the source-transfer path-receiver model for the OTPA considering the washing machine, the following definitions can be made:

- Source: The main vibration source of the characterized front-loading washing machine is the drive group that contains the tub, the rotary drum and the motor.
- Transfer paths: The vibrations of the machine are transmitted from the tub to the cabinet through the suspension system that includes two damping units, two coil springs and the bellow.
- Receiver: The resulting responses are generated at various parts of the washing machine, including top panel, front panel, side panels, back panel and base.

In this thesis, only structure-borne vibration effects are considered by neglecting the airborne and structure-borne noise influences. Also, the vibration transmissions from

the drive group to the cabinet through the bellow are not taken into account as a transfer path. Therefore, the source-transfer paths-receiver model for this experimental application may be established as:

- The main source is the drive group of the washing machine where the vibro-acoustic energy flow initiates.
- For the transfer paths, it is assumed that the vibration transmission propagates from the drive group to the washing machine cabinet over two coil springs and two damping units (the attachment of two coil springs to the side panel and the attachment of two damping units to the cabinet base).
- The receiver is the side panels of the washing machine.

The details of assumptions made and source-transfer path-receiver model design are described with the justifications in the following sections.

4.2 Selection of the Sensor Positions

Establishing the source-transfer path-receiver model is an important issue for the application of the OTPA procedure for successful representation of the system's dynamics. Another critical issue is the selection of sensor positions on the defined source, transition and response locations for accurate prediction of the vibro-acoustic energy propagation in the system.

For successful applications of OTPA to a mechanical system, the following recommendations are available in the literature [11, 70, 71]:

- Response (output) data measured on various excitation sources should be selected as input (reference) signal to decompose a measured overall response (e.g., at the driver's ear) in the divergent source scopes.
- Different responses on single excitation source should be selected as input quantities to determine wherewith the vibro-acoustic energy is transmitted from the main source to its vicinal component(s).
- Transfer paths should be selected as input variables to specify the position(s) that transmit the most vibration without pointing out the origin of the source.
- Several physical variables, as well as input and output locations should be included and combined in one OTPA setup to figure out vibration and/or noise propagation throughout the structure.

- Extra outputs (responses) within the system should be employed to determine the vibration transmission into the system via subsequent parts.

The main purpose of the application of the OTPA to a washing machine is to identify which transition locations have higher contributions as carrying vibration from the drive group to the receiver group, i.e., the side panels of the washing machine cabinet. For this reason, the transition locations (transfer paths) are accepted as OTPA inputs considering the drive group is the main excitation source. Furthermore, some assumptions are made about the transfer paths. **The first assumption** is that only structure-borne transfer paths are considered during the application of the OTPA procedure to the washing machine, thereby the airborne transfer paths are neglected. The washing machine tub vibration, which is generated by the eccentric motion of the drum in an unbalance state, is transmitted to the cabinet panels through five main transfer paths: two coil springs, two damping units, and the bellow, as illustrated in Figure 4.2. The placement of sensor near to bellow is difficult due to geometric restriction of the structure. Moreover, number of input channels of the signal analyser unit is limited (for the analyser available, maximum 12 input channels can be simultaneously used for a measurement) hence which signals need to be measured is an important question and this needs some optimisation. . Thereby, **the second assumption** is made here such that the vibration propagation via bellow to the panels of the washing machine cabinet is ignored.

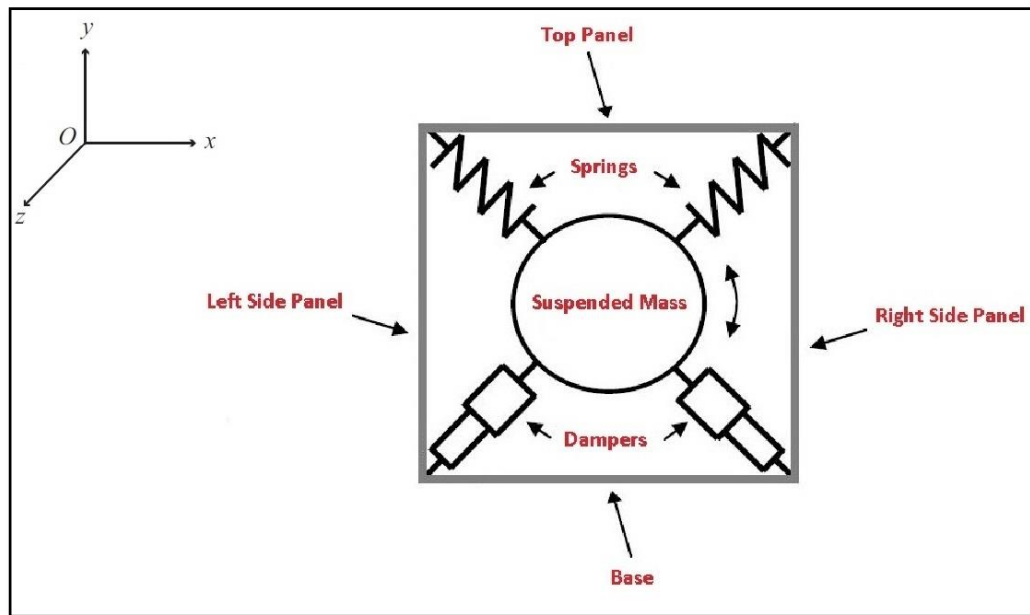


Figure 4.2 : Schematic view of the washing machine representing the transfer paths.

4.2.1 Selection of the input sensor positions: displacement measurement in the washing machine

To decide the most significant transfer path which transmits the most vibration to the receiver, vibration levels of the washing machine tub in a steady-state condition were measured in terms of displacements. Displacement measurement is required to optimize the number of input variables. Measurement of the tub displacements is made via the utilization of a triaxial piezoelectric charge accelerometer as shown in Figure 4.3 (b). Brüel & Kjaer triaxial piezoelectric charge accelerometer Type 4321 [101] was rigidly attached to the top periphery on the tub as it is demonstrated in Figure 4.3 (c). The accelerometer has three independent outputs for synchronous measurements in three perpendicular directions [101]. These measured acceleration signals were passed over the charge conditioning amplifier and turned into the displacement signals by double integration using the charge amplifier [84]. The converted displacement signals were simultaneously recorded by the signal analyzer [89] while processing real-time analysis utilizing PULSE Data Recorder that provides synchronous real-time analysis and time-data recording for post-processing [102]. The sampling frequency was 4096 Hz and the measurement period was defined as 60 second for each measurement.

In order to measure the horizontal (X), cross (Z) and vertical (Y) displacement of the washing machine tub, an operating test for the washing machine in the spinning cycle process was carried out. The displacement of the washing machine tub in a steady state vibration was measured. The measurements of the steady-state vibration were performed on the washing machine drum while keeping the rotation speed of the spinning drum constant. The rotation speed of the spin drum was increased from 400 rpm to 1400 rpm with a 100 rpm increment. The steady-state vibration measurements were repeated for each stage of the spinning cycle process. Two 1500 gr and one 2000 gr unbalanced masses were located along the circumference of the inner wall of the drum with equal angular distances. Their locations were referred as the midpoints between the two consecutive wings of the drum, which has 3 wings in total. Rubber plates were preferred as unbalanced masses in place of housing laundry inside the washing machine drum in the stage of spinning cycle as it is demonstrated in Figure 4.3 (a). The recorded time-domain displacement measurements were imported into MATLAB platform to draw the orbits.

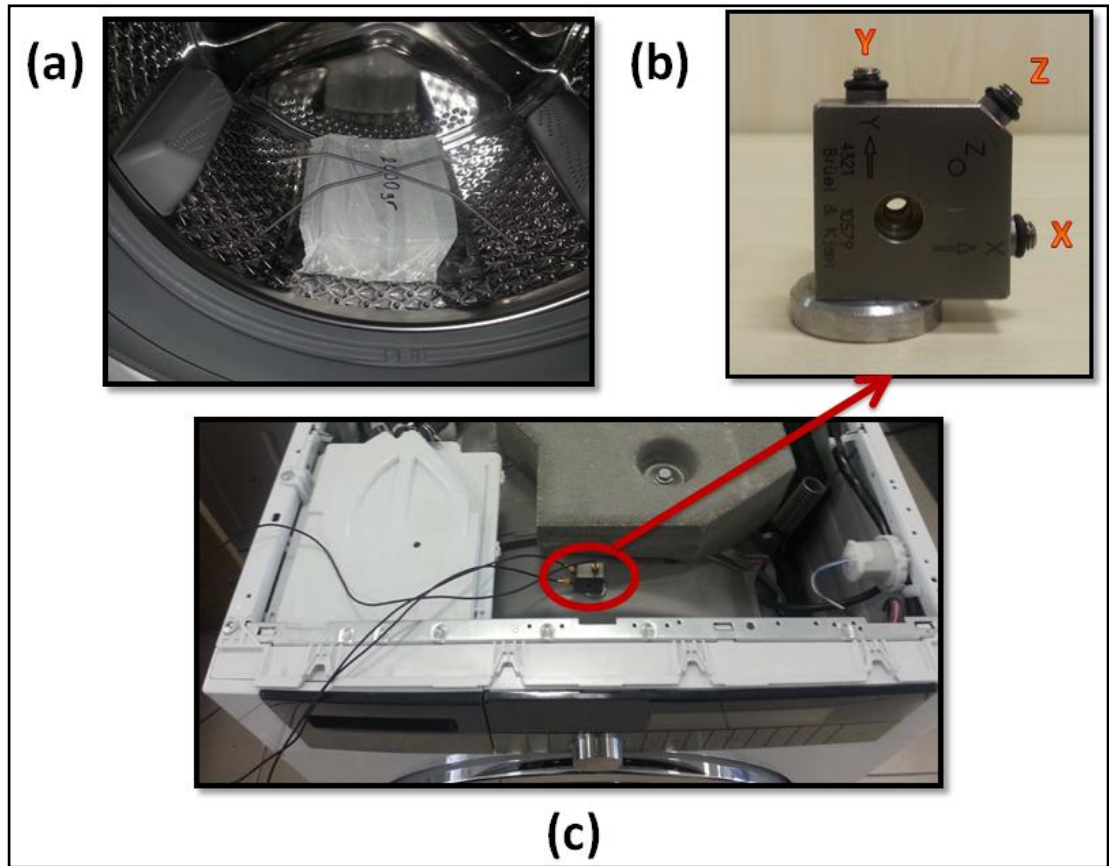


Figure 4.3 : (a) Inserting the unbalanced mass into inner wall of the drum. (b) Brüel & Kjaer triaxial piezoelectric charge accelerometer Type 4321 and displacement measurement directions. (c) Placement of the triaxial accelerometer on the tub.

Consequently, three orbits (displacement in X axis vs. displacement in Y axis, displacement in X axis vs. displacement in Z axis, and displacement in Y axis vs. displacement in Z axis) were drawn for each rotational speed of the drum using MATLAB. Sample displacement measurement results for 1200 rpm and 1400 rpm spinning cycles are given in Figure 4.4. It is observed that displacements in Z direction are much smaller than displacements in X and Y directions. For this reason, one can comment that vibration transmission in the Z direction from drive group to the cabinet through the suspension system is less than vibration transmission in X and Y directions. As a result, **the third assumption** made here is that the propagation of vibro-acoustic energy flow in Z direction to the washing machine cabinet is assumed to be small hence ignored. The same assumption appears to be used to in a study for the determination and examination of unbalance in washing machines [103].

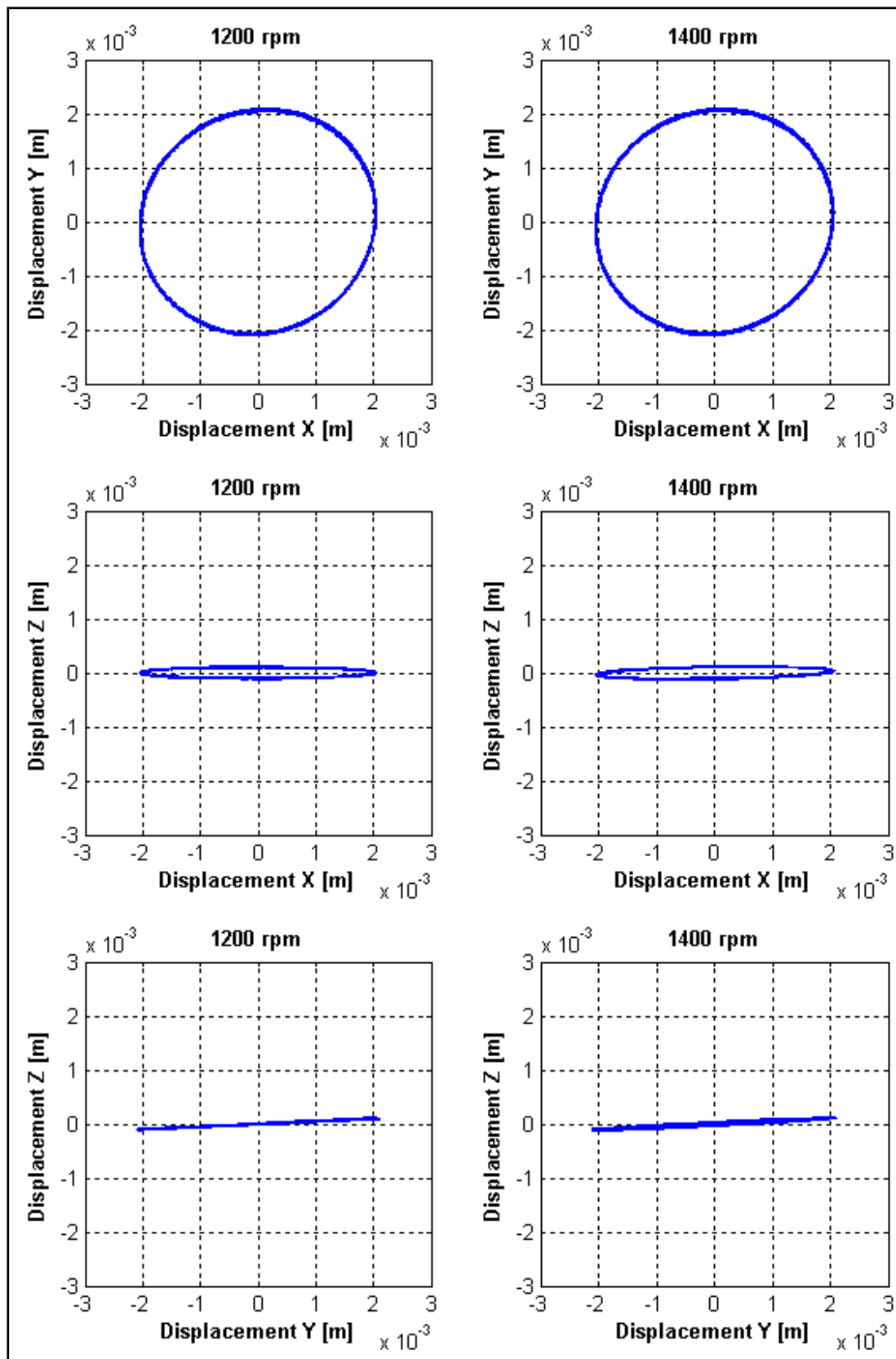


Figure 4.4 : Orbits for 1200 rpm and 1400 rpm spinning cycles considering 500 gr unbalanced mass in total.

For a mechanical system, the vibro-acoustic energy transfer initiates from the active side elements producing the operational excitation forces and acoustic loads, and moves through the physical attachments and via airborne transmission paths, to the passive side elements generating the corresponding responses [34]. Therefore, the active and passive parts of the examined mechanical system must be stated. Additionally, the physical attachments between active and passive sides of complete assembly and airborne transfer paths should also be defined. In this application, the drive group (suspended mass) of the washing machine is the main excitation source that acts like the active system component. The washing machine cabinet contains the response target locations, thereby it can be considered as the passive system component. The suspension system including coil springs and dampers provide the physical connections the active and passive system components. Although, the OTPA minimizes almost the majority of crosstalk effect, it is advised that the sensors are usually located as near as possible to each reference point on the source to keep apart the effects of the other sources [71]. Furthermore, it is an important issue to select the reference points as close as feasible to the physical connection on the active part of the system for good determination of the operational input quantities. In this application, four physical connection points between the active and passive system components shown in Figure 4.1 are considered. It is difficult to place the input sensors on the active part of the system owing to geometric restrictions of the structure and the movement of the active part in the vertical and horizontal directions. Hence, the input sensors (accelerometers) are positioned as close as possible to the physical connection on the passive part of the system as represented in Figure 4.5. Accelerometer #1, #2, #3 and #4 are used to measure the vibration signals as acceleration, propagated through left and right springs and dampers, respectively, in horizontal (X) and vertical (Y) directions. In this case, it is determined that there are 8 input quantities in total utilized for the application of OTPA to the washing machine. Thus, **the fourth assumption** made here is that transition locations are defined as input variables to determine which transfer path transmits the most vibrations to the response locations on the receiver. It should be noted that, transfer paths are also defined as vibro-acoustic transmission paths and transition locations in the literature.

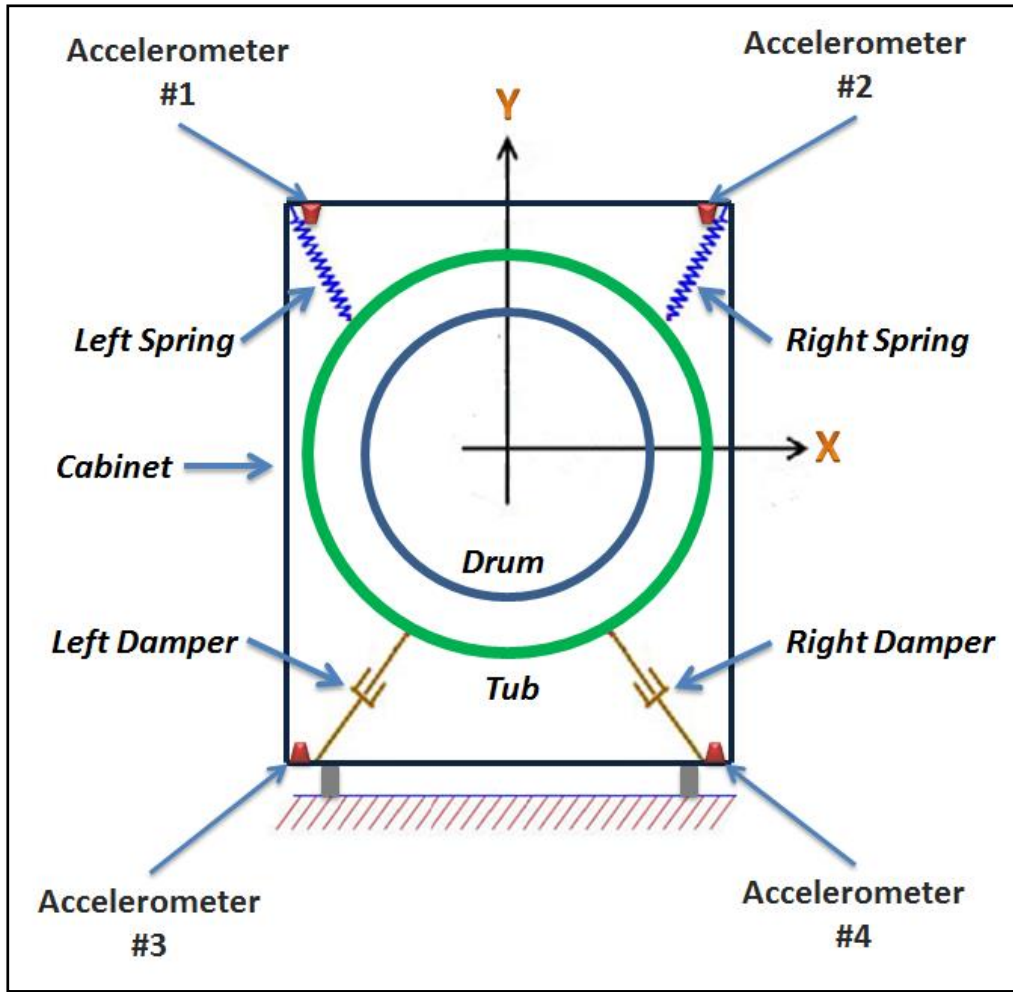


Figure 4.5 : Input sensor positions inside the washing machine.

4.2.2 Selection of the output sensor positions: modal analysis of the washing machine

To determine the response points on the receiver location, experimental modal analysis of washing machine was performed. Modal analysis is used to determine the basic dynamic features of a mechanical system such as natural frequencies and mode shapes [104]. The main purpose of carrying out the experimental modal analysis here is to optimize the number of output variables. First of all, the numerical model of the washing machine cabinet excluding base, as given in Figure 4.6, was created in ICATS platform using MESHGEN module [105]. MESHGEN is utilized to produce the experimental meshes; thereby creating a (.DSP) file, which includes the position of the measurement points as nodes to simulate the experimental model [105]. The same measurement points were marked on the cabinet of experimental model as presented in Figure 4.6.

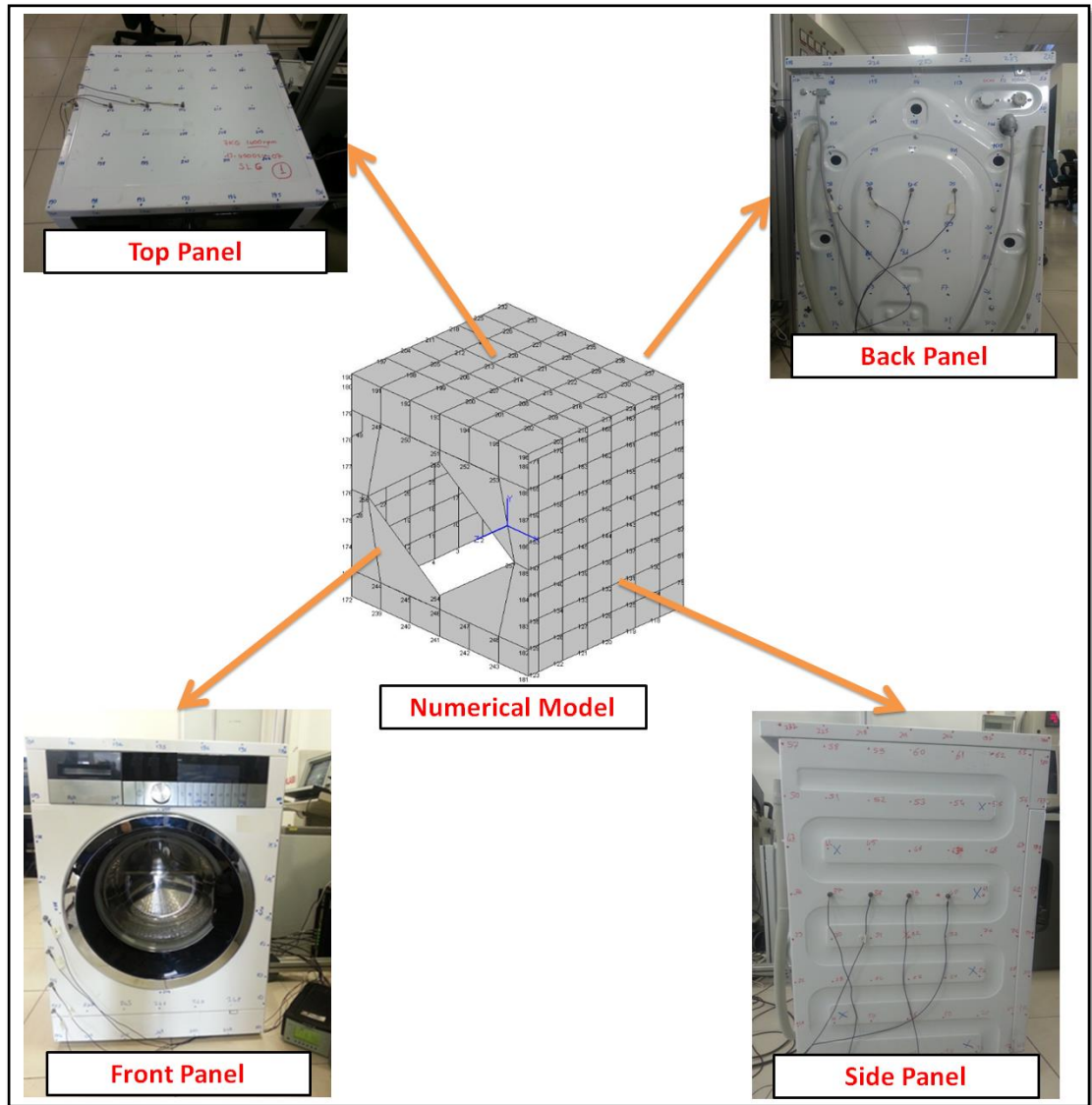


Figure 4.6 : Numerical model of the washing machine cabinet excluding the base.

For the modal analysis of the washing machine, the experimental setup given in Figure 4.7 was used. The same procedure was used as in the forced vibration analysis of the simple flat plate described in third chapter of this thesis. The shaker [86] was attached to the right panel of the washing machine to excite the structure using a random signal. The corresponding responses were simultaneously measured as accelerations using accelerometers [83] at each of the defined measurement points. The measurement directions were taken as in Z direction for the front and back panels, in Y direction for the top panel, and in X direction for the side panels. The FRFs between measured forces using force transducer [87] and accelerations were computed by the signal analyzer [89] and stored in Universal File Format (.uff).

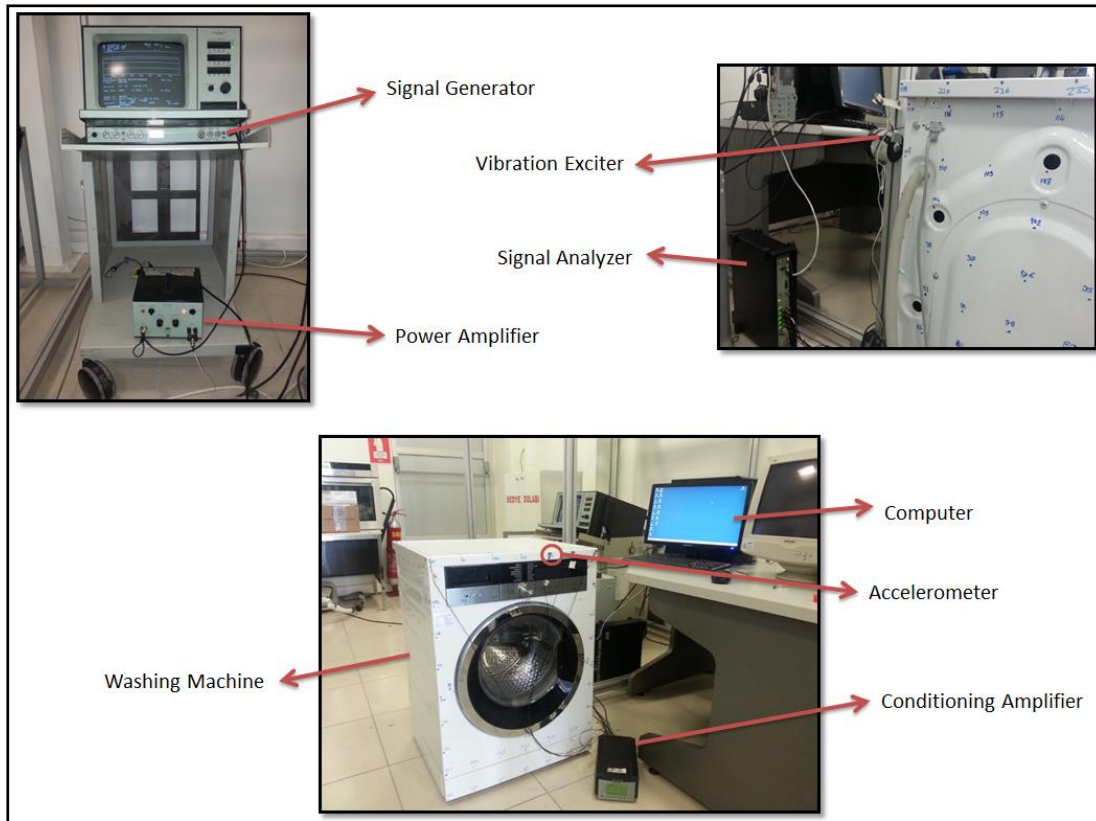


Figure 4.7 : Experimental setup used for the modal analysis of washing machine.

The computed FRFs were imported into ICATS platform and analyzed using MODENT module [106] to obtain natural frequencies and corresponding complex mode shapes. MODENT is utilized to examine single/multi FRF data in SIMO/MIMO format to determine the model properties of the experimental model [106]. For the experimental modal analysis of washing machine, the FRF data was analyzed up to 120 Hz that resulted in extracting a total of 17 modes within the frequency range of interest. All FRF plots and their phase angles were overlaid for the accurate extraction of natural frequencies, and the determined mode shapes were animated in ICATS platform using MODESH module [107]. The first 5 mode shapes obtained from modal analysis of the washing machine are given in Figure 4.8. After the extraction of the natural frequencies as well as the complex mode shapes, MODPLAN module [108] was used to find the best response points in ICATS platform. MODPLAN is employed to determine the best response points so as to establish the most favorable excitation, suspension and response points [108].

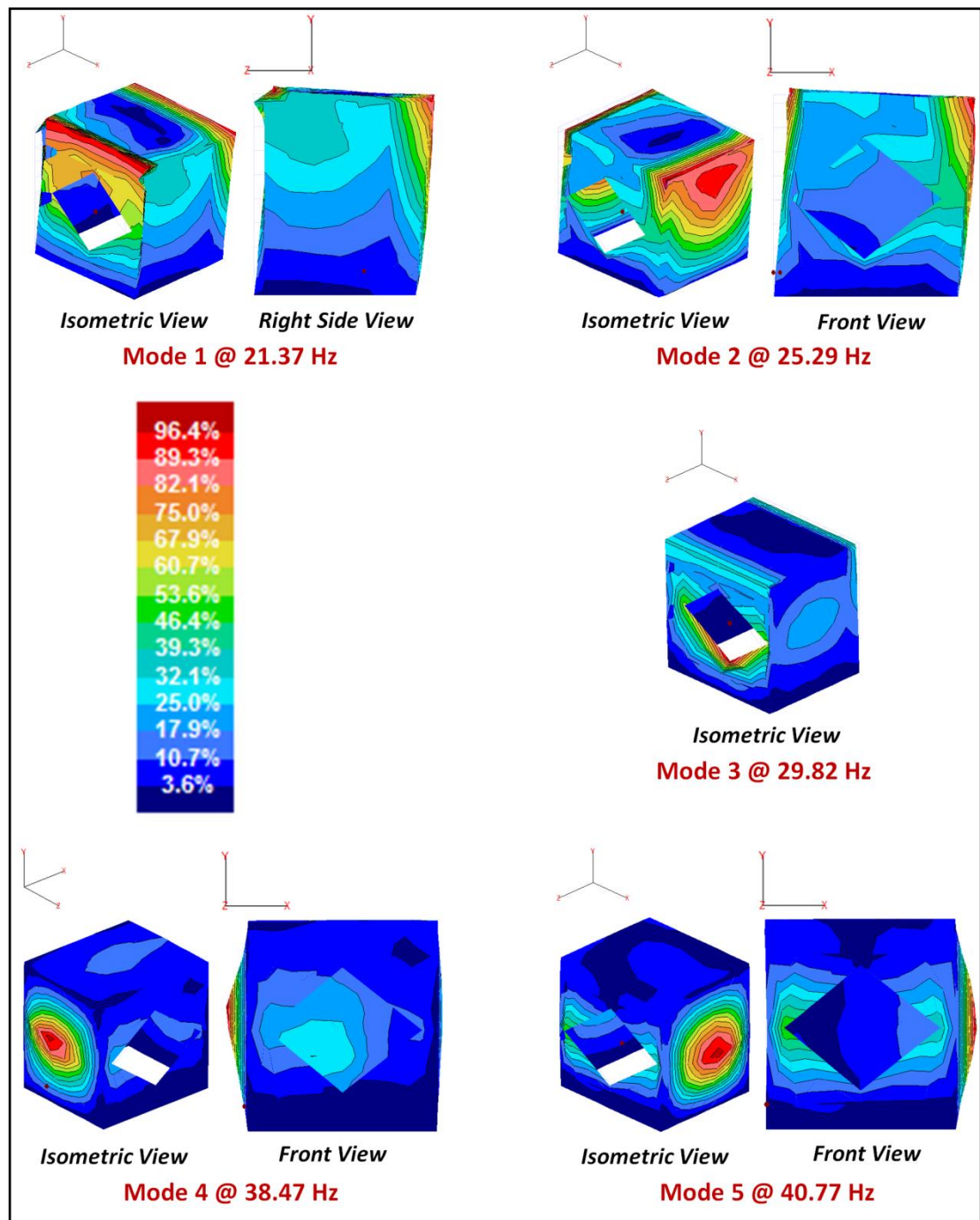


Figure 4.8 : The first 5 mod shapes obtained from modal testing.

In MODPLAN, each measurement points (nodes) defined on the ICATS model of washing machine, shown in Figure 4.6, can be seen as a Degrees-of-Freedom (DOF). For the identification of best response DOFs, two parameters are required, which are Average Driving DOF Acceleration (ADDOF-A) and Effective Independence (EI). These parameters are named as the test planning tools in MODPLAN [108].

Considering a set of natural frequencies ω and corresponding mode shapes $[\Phi]$, these test planning tools available in ICATS can be described as follows [108]:

ADDOF-A defines the average acceleration response of the structure. It is determined by the following statement [108]:

$$ADDOFA(j) = \sum_{a=1}^b \Phi_{j,a}^2 \quad (4.1)$$

where a denotes the mode index and b denotes the number of modes. The index j differs from 1 to the number of DOFs.

EI is another test planning tool to find the best measurement points. The components placed in the main diagonal of the EI matrix give the additive of each DOF to the global rank of the mode shape matrix. Thus, best measurement coordinates is indicated by the components that have high values [4.14].

The supposed Fischer information matrix $[A]$ is determined by the following expression [108]:

$$[A]_{bxb} = [\Phi]_{bxN}^T [\Phi]_{Nxb} \quad (4.2)$$

The estimation matrix, $[E]$, of the mode shape matrix is found by the following equation [108]:

$$[E]_{NxN} = [\Phi]_{Nxb} [A]_{bxb}^{-1} [\Phi]_{bxN}^T \quad (4.3)$$

The estimation matrix is a symmetric idempotent matrix. The elements of the main diagonal of this matrix give the optimal measurement DOFs, where the elements with high-values are the best measurement DOFs [108].

Consequently, the calculation of EI x ADDOF-A vector yields the best accelerometer points; thereby the optimum output sensor locations can be specified by ranking accelerometer points. The result of this calculation for the side panels of washing machine considering the mode shapes along horizontal (X) direction is presented in Figure 4.9.

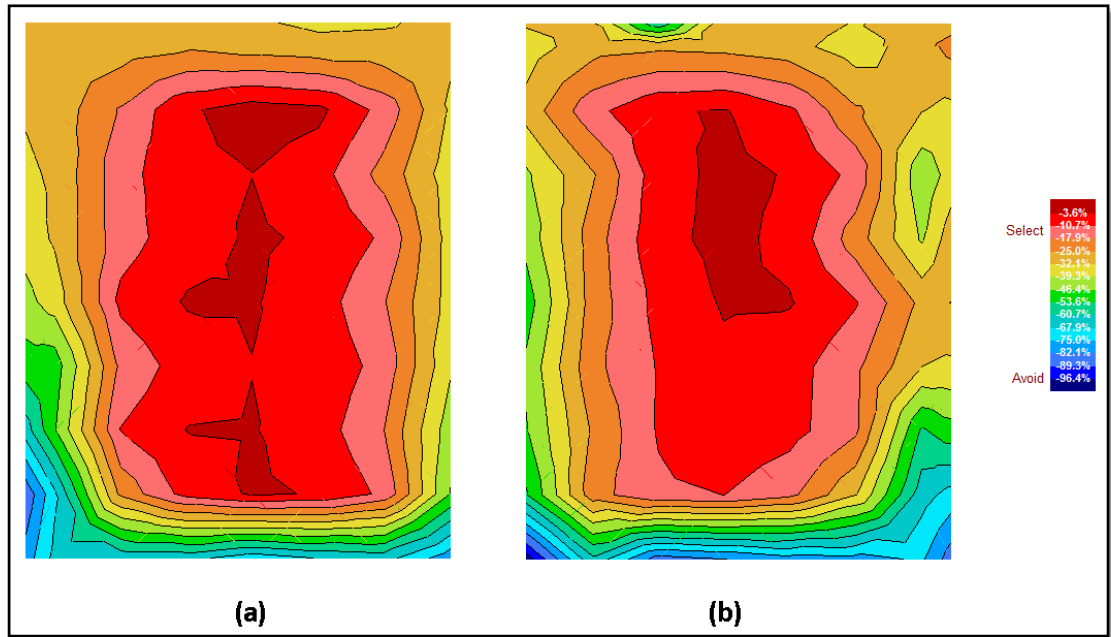


Figure 4.9 : Best accelerometer response points for (a) right side panel (b) left side panel of the washing machine.

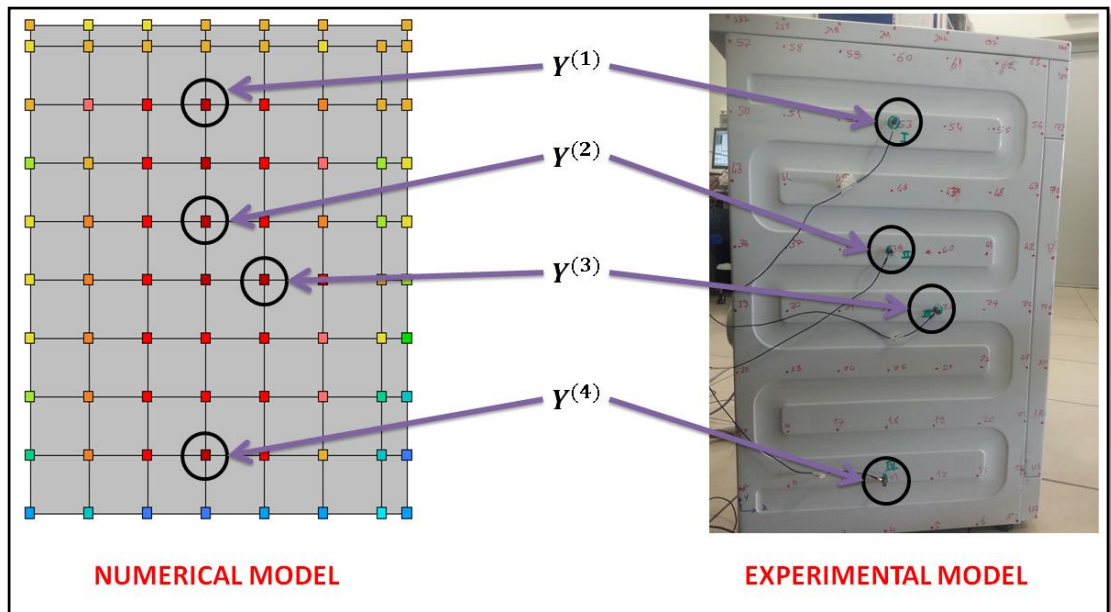


Figure 4.10 : Selected output sensor positions on the defined receiver location.

As a result, four output sensor positions were selected with regarding the consequences of best accelerometer response determination procedure. Selected output sensor points, given in Figure 4.10, are located on the left side panel of the washing machine. For the rest of this chapter, the input and output quantities are denoted as $X_r^{(m)}$ and $Y_r^{(n)}$ respectively where m is the number of references (input-

applied operational forces), n is the number of responses (output-measured structural responses), and r is the number of operational conditions. It should be noted that the variables $X_r^{(m)}$ and $Y_r^{(n)}$, and the directions X and Y have completely different meanings.

4.3 Selection of the Operating Conditions

In order to predict the TFs, another essential issue for OTPA is the selection of appropriate operational conditions for the measurements. There are two main targets to obtain better OTPA consequences as (i) to constitute entire characteristic operational conditions during the measurement period, and (ii) to manage low coherence between the frequency spectrums utilized in order to compute the TFs. It is advised that the measurements are carried out during the time period that the operating conditions are altered over the all characteristic operating span for the prediction of the TFs [11, 70, 71]. As a requirement, the number of operational conditions (frequency spectrums) r must be greater than the number of reference (input) signals m , i.e. $r > m$ [11, 43, 70, 71].

As described before, the front-loading washing machines generally contain cabinet, tub, drum, spring, counter weight, stator, rotor and damper, as represented in Figure 4.1. The tub is mounted to keep the water for washing process; the rotor is directly linked to the drum to provide power for the rotation; the stator is coupled on the back of the tub inside the washing machine cabinet. During the rotation of the drum, the mass unbalance owing to the eccentric motion of laundry generates the vibration. The vibrations of drum and tub, i.e. the vibration of the drive group, are transferred to the cabinet over the suspension system assuming the springs and the dampers [100]. Therefore, the existence of the centripetal force can be mentioned here due to the rotation of reference frame so as to the inertia of motion. For the spinning cycle of the washing machine, the inner wall of drum applies a centripetal force on the laundry while the drum rotates at higher speed. The direction of centripetal force is along the radius of the circle, from the object towards the center. It is computed utilizing the following expression:

$$F_{cp} = m\omega^2 r \quad (4.4)$$

where F_{cp} denotes the centripetal force, m is the mass of the object, rotating with a rotational frequency ω along an orbit with radius of curvature r , as shown Figure 4.11.

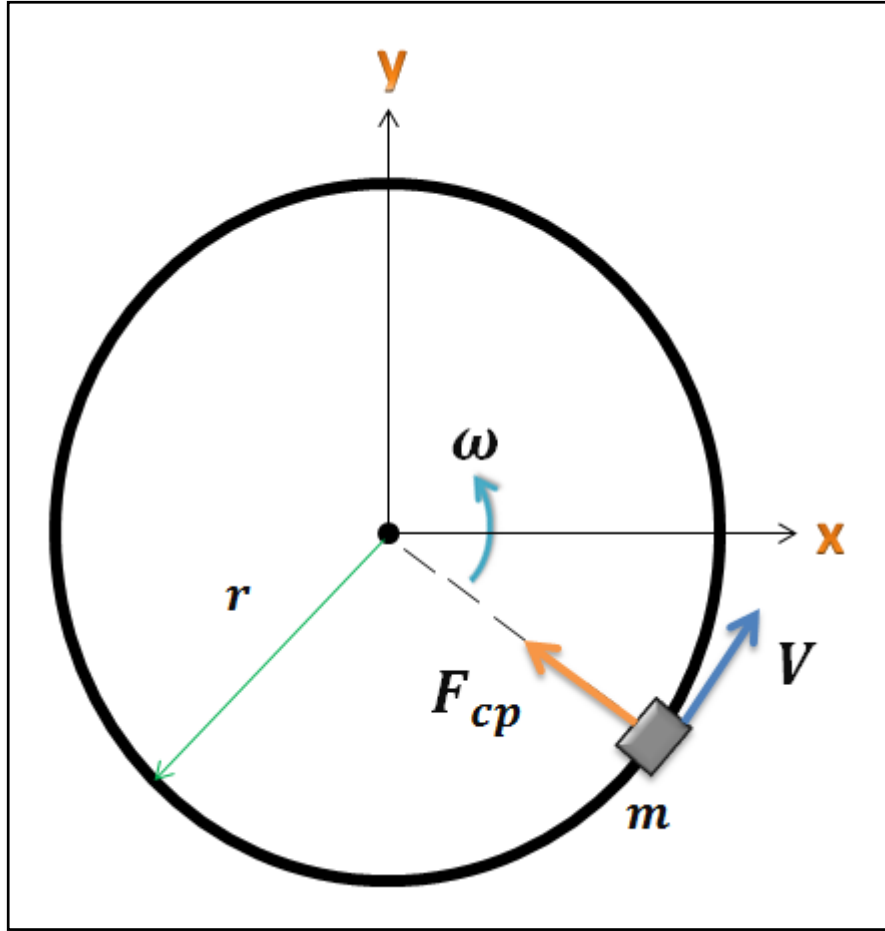


Figure 4.11 : Direction of centripetal force.

From this expression (Eq. 4.4), it is understood that the necessary operational conditions for the application of OTPA to a front-loading washing machine can be provided by changing the influence of centripetal force. The effect of this force can be changed in three ways:

- Increasing/decreasing the mass of the object (m)
- Raising/reducing the rotational frequency (ω)
- Expanding/narrowing the radius of orbit (r)

Altering the main source excitation, which is assumed as centripetal force here, leads to vary the operational conditions. For this washing machine application, the main source of excitation was altered by changing the mass of the object and the rotational

frequency while the radius of rotation path was kept constant. More precisely, the operational conditions are varied by changing the rotational speed of the washing machine drum and total unbalance mass inside it. During the spinning process, the rotational speed of the drum can increase from low to high level; thereby the complete assembly can be excited over a wide frequency range. Moreover, the operational measurements can be made by creating different unbalance conditions in the drum (e.g. increasing total mass located inside the drum) at the steady-state condition of the spinning process with a constant rotational speed.

4.4 Experimental Setup and Measurement Details

The main purpose of the application of OTPA to a washing machine application was to analyze contributions of different transfer paths to the side panels of the washing machine. The washing machine tested is a horizontal axis front-loading drum-type. As described before, the main component groups are the drive group (also named as the suspended mass) the suspension system and the cabinet. The suspension system provides the connection between the suspended mass and the cabinet. Main vibration source in the washing machine is the suspended mass, its eccentric motion during a spinning cycle in an unbalanced situation, causing vibrations in the washing machine. Vibration transmission propagates from the suspended mass to the cabinet via the transfer paths, i.e., the suspension system. The suspension system contains two springs that link the drive group to the side panels of the cabinet, two friction dampers that link the drive group to the cabinet base, and the bellow that links the drive group to the front panel of the cabinet. In this application, vibration transmission to the cabinet through the bellow was ignored.

For the analysis, 8 input and 4 output variables were defined considering the structure-borne vibration transmission locations. The effects of structure-borne and airborne sound as input quantity were neglected. Piezoelectric charge accelerometers [83] were used to measure the input and output accelerations for structure-borne vibration. Due to the use of piezoelectric charge type sensors, the charge conditioning amplifiers [84] were utilized to convert the charge output into a voltage. Measurements were processed with PULSE LabShop software [92] and corresponding, synchronized signal analyzer unit [89] to obtain the required phase-assigned spectrums. The signal analyzer unit has 12-channel Input Module Type

3038; hence maximum 12 input channels could be simultaneously used for signal measurement. The measured data was analyzed with the developed MATLAB code for the estimation of TFs and for the decomposition of the critical transfer paths. The measurement flowchart is illustrated in Figure 4.12.



Figure 4.12 : Measurement flowchart for the washing machine application.

The reference (input) sensors were located as close as possible to physical connection between the transfer paths (the suspension system of washing machine) and passive side components of the system (the washing machine cabinet). Therefore, which transfer paths transmit the most vibration from one main excitation source to the receiver location could be determined by employing transfer paths as input variables for OTPA. 8 input signals in total for each operational condition were simultaneously measured at the positions given in Table 4.1. It should be stated that structure-borne vibration transmission to the receiver location in the Z-direction through the suspension system is weaker; hence it was not considered as an input variable. The corresponding structural responses were also simultaneously measured at four defined measurement locations on the left side panel of washing machine. The four different response positions were defined in the X-direction as a result of modal analysis of the washing machine and were the best acceleration response locations on the left side panel; hence the propagation of the vibro-acoustic energy could be identified. It should be restated that panel vibrations are mostly along the X-direction.

Table 4.1 : Reference (input) sensor positions and directions.

Source	Position	Direction	Number of
Structure-borne	Left Damper	Horizontal (X)	1
		Vertical (Y)	1
	Left Spring	Horizontal (X)	1
		Vertical (Y)	1
	Right Damper	Horizontal (X)	1
		Vertical (Y)	1
	Right Spring	Horizontal (X)	1
		Vertical (Y)	1

As explained in subsection 4.3, it is essential to modify or change the main source excitation, i.e., the input quantities, throughout the operational measurement. Strong alterations reduce the coherence between the selected input quantities and provide much better conditioning [11, 70]. The washing machine was tested at different rotational speeds, from 400 to 1400 rpm, with a step of 100 rpm. Every stage of spinning cycle lasts for 90 seconds. Moreover, two critical spin cycles at 1200 rpm and 1400 rpm were retested for different unbalanced weight, completely placed inside the washing machine drum. Two variations were created for the placement of the unbalanced weight in the drum. In the first variation, single unbalanced weight was attached to the periphery of the spin drum where rubber plates were used as unbalanced masses. For the second variation, three rubber plates were positioned to the inner wall of the drum with equal angular distances. The details of the positions of the unbalanced masses for both variations are illustrated in Table 4.2. Furthermore, two types of damping units, so called single effect and double effect friction dampers, were utilized for the operational measurements. In addition, the input and output accelerations in a steady-state vibration were measured.

Table 4.2 : Details of unbalanced masses positioned in the drum.

# of Operational Condition	Variation 1	Variation 2
	Single Unbalanced Mass [gr]	Trio Unbalanced Masses [gr]
1	100	800-800-100
2	150	800-800-150
3	200	800-800-200
4	250	800-800-250
5	300	800-800-300
6	350	800-800-350
7	400	800-800-400
8	450	800-800-450
9	475	800-800-475
10	500	800-800-500

The entire operational measurements described above were employed to predict the TFs from each input location to the output location. All measured signals were obtained as phase-assigned spectrums in 0-1600 Hz frequency range and a frequency resolution of 0.25 Hz. Also, a MATLAB code was developed to process the measurement results using the OTPA procedure to obtain the required data for the identification of critical transfer paths.

4.5 Results and Discussion

After establishing the source-transfer paths-receiver model with the appropriate selection of sensor positions and operational conditions for the OTPA, the evaluation of all measurements initiates with the comparison of synthesized (calculated) output signals and their originally measured counterparts. It is already realized in third chapter that LSA and SVD yield the same operational TF estimation and overall response synthesization; hence SVD method is preferred for the computations. Actually, the estimation of TFs from Eq. 2.32 considering a least-squares approach leads to determine the responses $\mathbf{Y}_s$ by employing the TFs as FIR filters in Eq. 2.37 with the real measured inputs $\mathbf{X}$ [11, 43, 70, 71]. Comparison of the synthesized response (i.e., the output produced by the multiplication of the TFs and measured inputs) and the measured response signals can be seen as the first step of the verification of the OTPA procedure. After this validation, if the actual measured response signal is matched well with the synthesized one, the overall response signal can be separated in individual path contributions utilizing the OTPA method; thereby the name of this step is also called contribution analysis. As a result, the critical transfer paths which transmit the most vibration can be determined.

As explained before, variation in washing machine's excitation can be provided in two different ways: changing the rotational speed of the spinning drum and generating various unbalance conditions in the drum at the stage of spinning process. Therefore, the consequences of application of OTPA to washing machine are examined under two subsections. In the first subsection, the results belong to the experimental measurements where the operational conditions were varied by changing the rotational speed of the drum. For the second subsection, the consequences refer to the experimental measurements where the operational conditions were altered by creating different unbalance conditions inside the washing machine drum. Moreover, two types of damping units were used in the washing machine to examine the OTPA application with different setups.

4.5.1 Results of experimental case I: changing the rotational speed of the washing machine drum

In this experimental case, the main source excitation was altered by changing the rotational speed of the washing machine drum. The operational measurements were

repeated for different spin velocities, from 400 to 1400 rpm, with a step of 100 rpm. Every stage of spinning cycle lasts for 90 seconds; hence the measurement period was set as 60 sec for each operational condition. Instead of housing laundry inside the drum, rubber plates were preferred as unbalanced masses. For this reason, one 2000 gr and two 1500 gr unbalanced masses were placed along the circumference of the inner wall of the drum with equal angular distances. The acceleration measurements for input and output quantities defined on the washing machine were made in a steady-state condition. The spin velocities from 1200 rpm to 1400 rpm were the working conditions where the highest vibration levels were measured. Also, two types of damping unit were used in the measurements for this experimental case. Single and double effect friction dampers were utilized in the first and the second operational measurement sets of experimental case I, respectively.

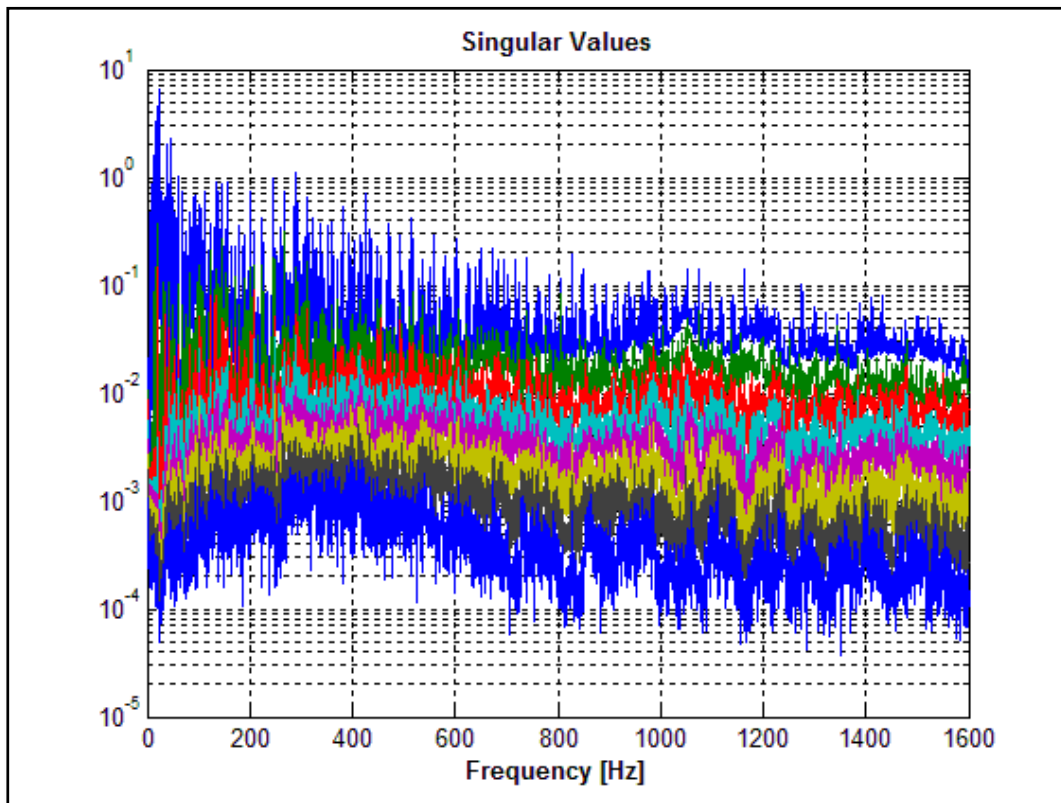


Figure 4.13 : All of the singular values of first operational measurement set of experimental case I as a function of frequency in logarithmic scale.

As an initial step for the verification of the OTPA procedure, the actual measured response (output) signals are compared with the synthesized (calculated) response signals. While the main source excitation for the front-loading drum type washing machine altered over the rpm range, two stationary operating tests of the rotation

speed of the drum, with 1200 rpm and 1400 rpm were chosen for the verification. Figure 4.13 shows the entire singular values of first operational measurement set of experimental case I as a function of frequency in logarithmic scale, where the single effect friction dampers were used in the suspension system of washing machine. It is observed that the singular values are representing bound characters; hence the established OSPA model can describe the response vibration measured on the left side panel of washing machine with 8 different structure-borne input quantities.

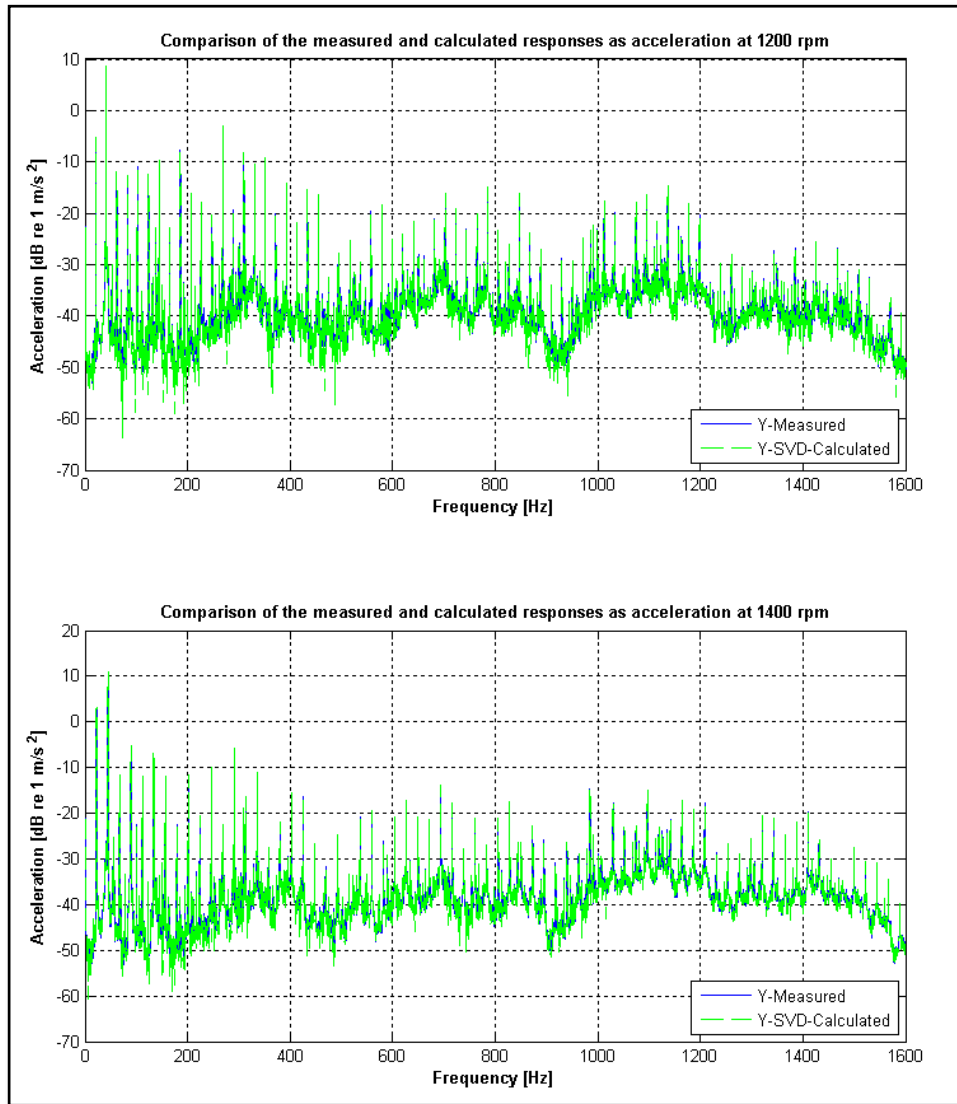


Figure 4.14 : Comparison of measured (blue) and calculated (green) response acceleration signals $Y_r^{(n)}$ in dB scale for a drum rotational speed of 1200 rpm (top) and 1400 rpm (bottom) with single effect friction damper set.

Figure 4.14 illustrates the comparison of measured and calculated (synthesized) responses as acceleration in dB scale for an arbitrary output measurement location

using single effect friction dampers. From this figure, it can be understood that there is a good accordance between the measured and synthesized response signals for both of the 1200 rpm and 1400 rpm measurement conditions. The results of the first operational measurement set of experimental case I indicate that the selected sensor locations and the defined major vibration transmission paths as input variables were appropriate for the analysis.

After the comparison process for the validation, the dominant transfer paths are identified by the contribution analysis where the overall response is separated into individual path contributions. It is important to express that the all measured input and output acceleration signals were determined as complex type Root Mean Square (RMS) values by the use of signal analyzer. Individual path contributions from each of the 8 input/source channels were separately obtained using the OTPA method for every assigned response location on the left side panel of washing machine, shown in Figure 4.10. To indicate an overall vibration level for the receiver, the absolute RMS values of the separated individual path contributions for the four response measurement locations were linearly averaged. In order to evaluate the overall vibration level of each contributor for each operational condition (the spin velocity of drum), total RMS values in the frequency range of interest were computed using the following statement:

$$J_{indcon}(r, m) = \sqrt{\sum_{a=N_i}^{N_f} \left(\bar{Y}_r^{(m)} \right)_a^2} \quad (4.5)$$

where $J_{indcon}(r, m)$ defines the total amount of energy of the accelerations as RMS value, the index r is from 1 to the number of operational conditions and the index m is from 1 to the number of input quantities. $\bar{Y}_r^{(m)}$ denotes the linearly averaged absolute RMS values of individual contributor (input channels), a denotes the frequency component index, N_i and N_f represents the initial and final frequency components included in the frequency range of interest, respectively.

The most important operational condition for the first operational measurement set of experimental case I is the rotational speed of 1400 rpm where the highest vibration level was measured. Therefore, the pie chart given in Figure 4.15 illustrates the result

of contribution analysis as percentage values for the 1400 rpm measurement condition over the entire frequency band (0-1600 Hz). According to this contribution analysis, the most dominant transfer path is found to be the horizontal (X) component of the right spring that transmits the most vibration from the drive group (main source) to the left side panel of washing machine (receiver) for the whole frequency range.

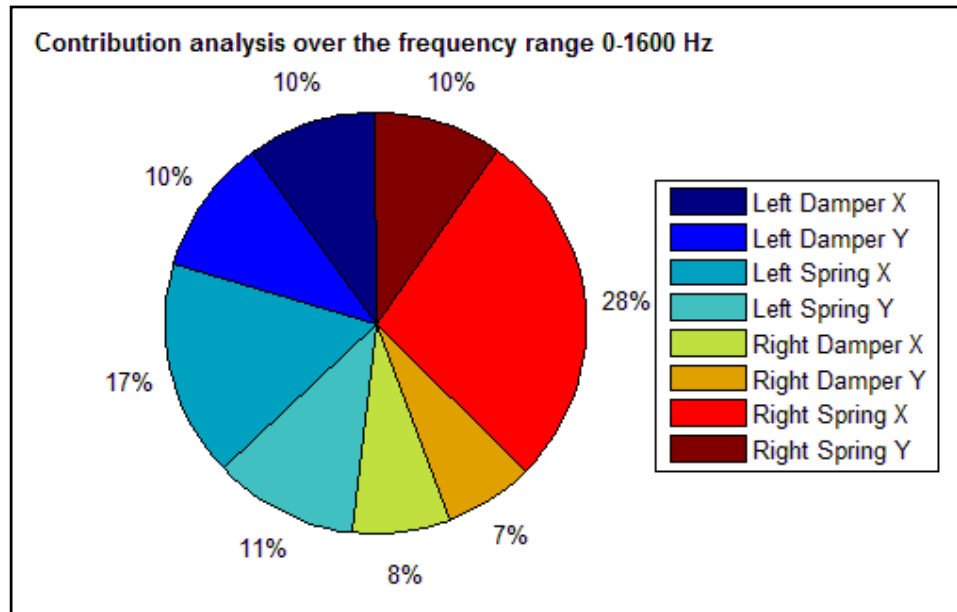


Figure 4.15 : Result of contribution analysis belongs to experimental case I, where the single effect friction dampers were used, obtained as percentage values for the 1400 rpm measurement condition over the complete frequency range 0-1600 Hz.

For a detail observation of the contribution analysis, different frequency ranges could also be examined. The reason is that the vibration problems can occur in a frequency span narrower than the complete frequency range of interest. Moreover, the dominant transfer paths can change from one frequency span to another frequency span. Figures 4.16 and 4.17 show the results of contribution analysis as percentage values for the 1400 rpm measurement condition at lower frequencies, over the frequency ranges 0-100 Hz and 100-200 Hz, respectively. According the result of contribution analysis over the frequency range 0-100 Hz given in Figure 4.16, two most dominant vibration transmission paths are the horizontal (X) component of the right spring and the vertical (Y) component of the left spring. On the other hand, if one examines the contribution analysis over the frequency range 100-200 Hz illustrated in Figure 4.17, it can be realized that the vertical (Y) component of the left damper and the

horizontal (X) component of the left spring seem to be the inputs that transmit the most vibration to receiver location.

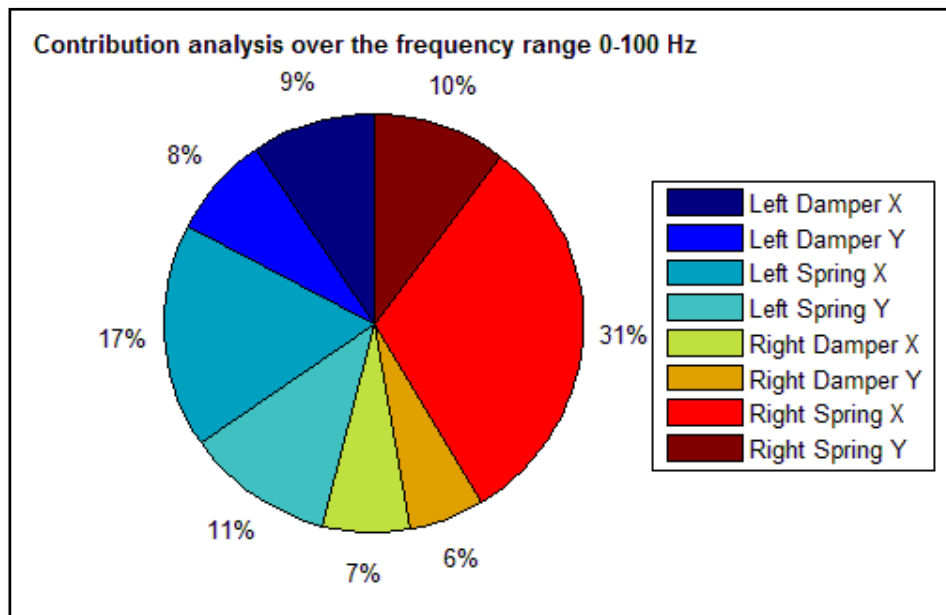


Figure 4.16 : Result of contribution analysis belongs to experimental case I, where the single effect friction dampers were used, obtained as percentage values for the 1400 rpm measurement condition over the frequency range 0-100 Hz.

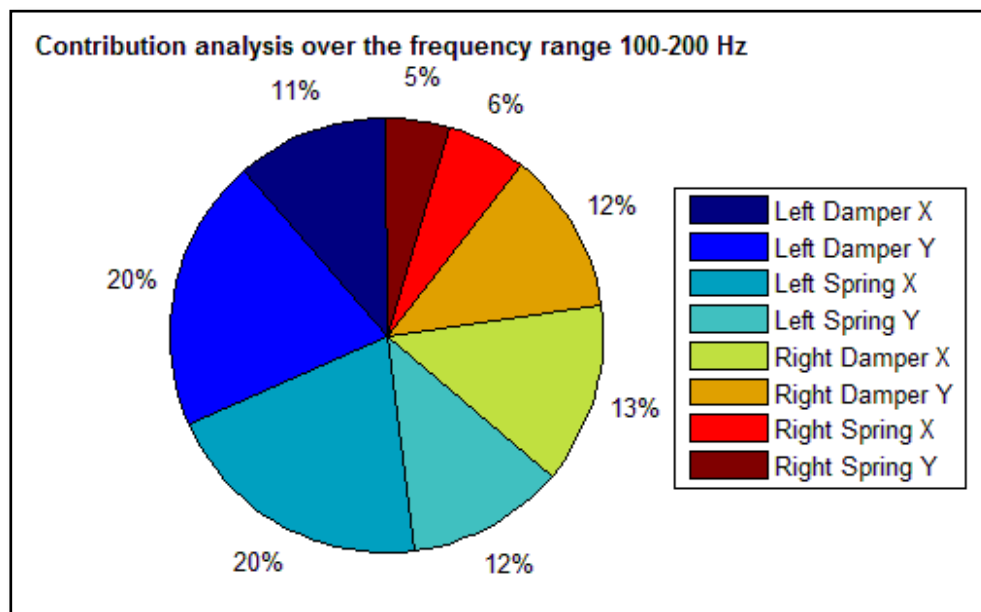


Figure 4.17 : Result of contribution analysis belongs to experimental case I, where the single effect friction dampers were used, obtained as percentage values for the 1400 rpm measurement condition over the frequency range 100-200 Hz.

In the second part of experimental case I, a double effect friction damper set was used in the suspension system of washing machine. Figure 4.18 shows the singular values of the input matrix $\mathbf{X}_r^{(m)}$ belongs to the second operational measurement set of experimental case I, where the double effect friction dampers were used. The singular values contain the first 8 principal components of each measurement data block, including only input (indicator) signal DOFs. It is seen that 8 physical transmission paths are required to be involved up to 1600 Hz. Furthermore, it should be considered that the examination of singular values, i.e. principal components of the input (reference) matrix, is required. The reason is that the OTPA is a signal processing method which is based on the signal decomposition at the defined receiver points, the vibration contributions of the mistakenly ignored transfer paths can be distributed between the indicator transmission channels which are taken into account for the OTPA [7, 56].

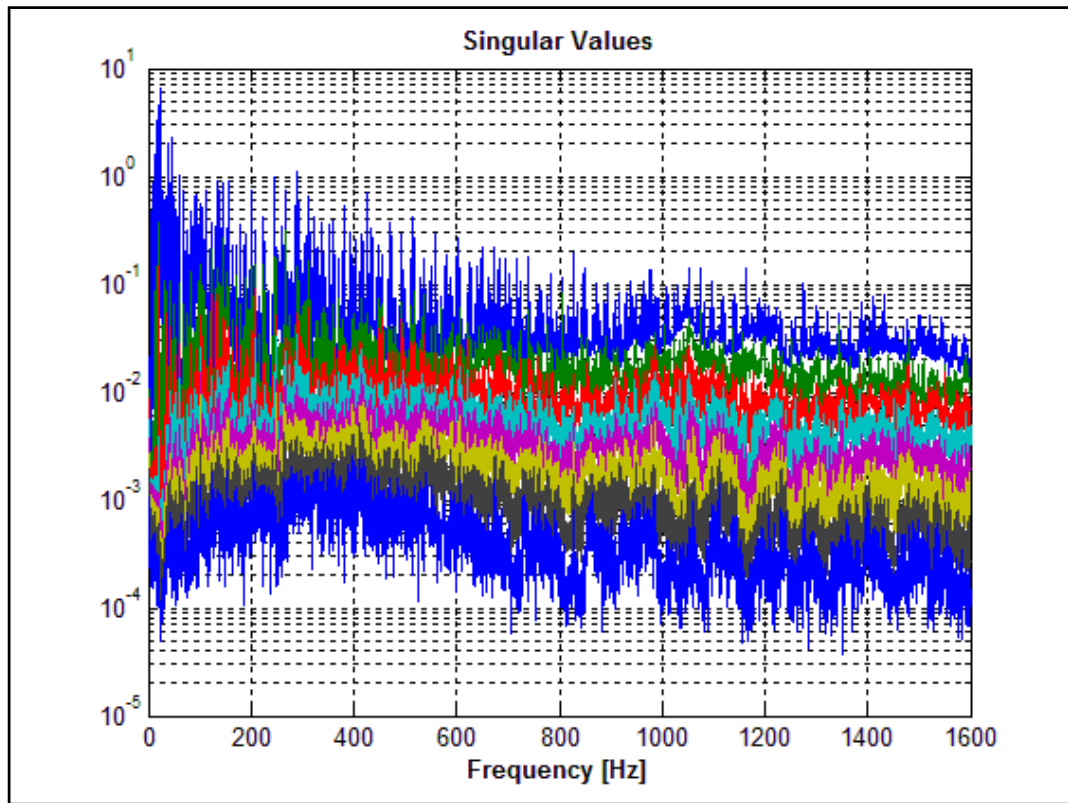


Figure 4.18 : All of the singular values of second operational measurement set of experimental case I as a function of frequency in logarithmic scale.

A good combination among the synthesized and actually measured response signals on an arbitrary output location for the 1200 rpm and 1400 rpm operational

measurement conditions can be seen in Figure 4.19. This comparison indicates that the utilized OTPA model is capable of the identification of critical vibro-acoustic transmission paths. Also, transmission paths (or sources) can be ranked at response target locations by comparing the individual transmission path contributions.

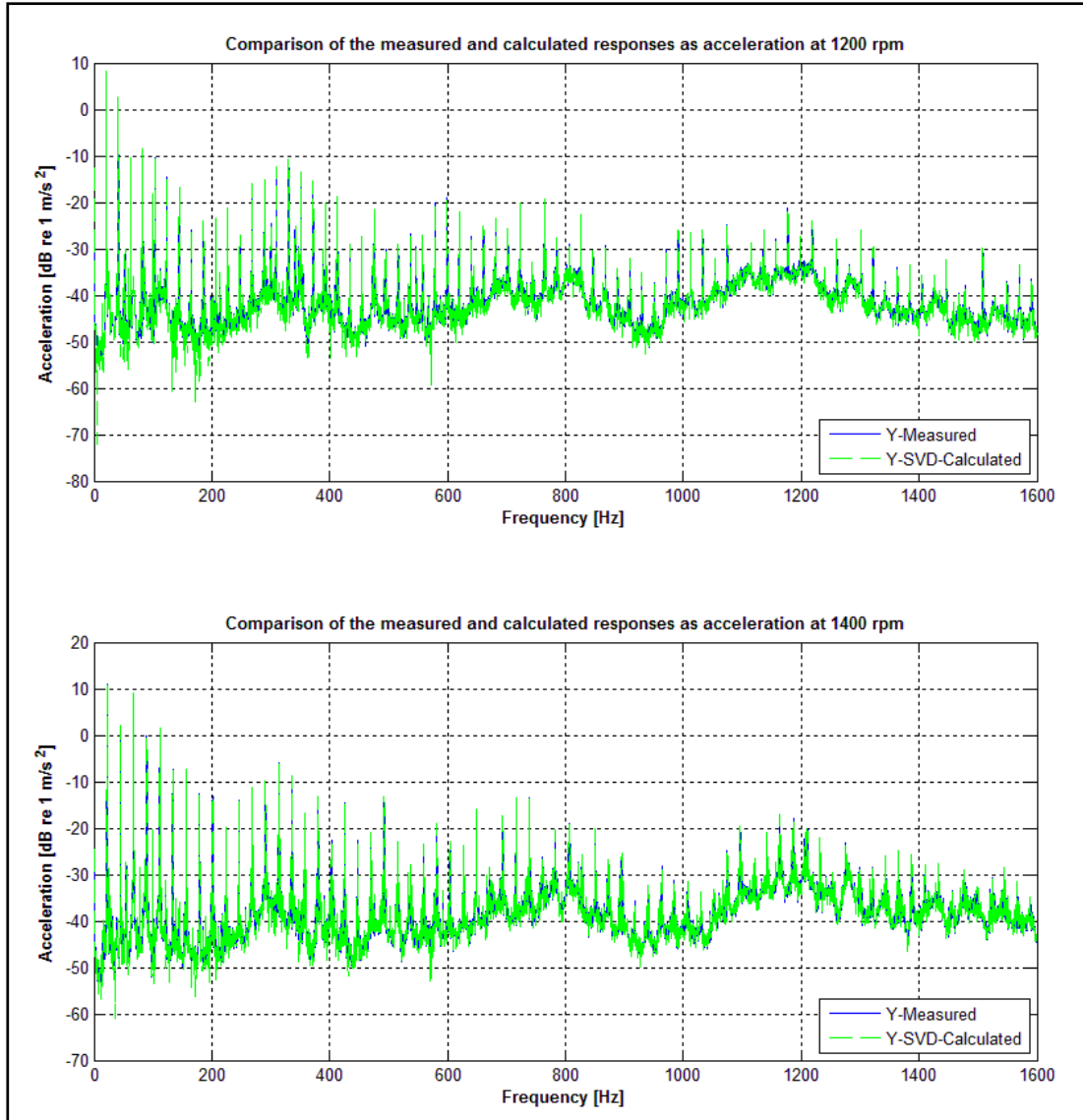


Figure 4.19 : Comparison of measured (blue) and calculated (green) response acceleration signals $Y_r^{(n)}$ in dB scale for a drum rotational speed of 1200 rpm (top) and 1400 rpm (bottom) with double effect friction damper set.

For the contribution analysis, the same procedure was traced as it is in the first part of experimental case I where the single effect friction dampers were used. Using Eq. 4.5, the individual path contributions of each input variables for every operational condition were computed from linearly averaged overall response data. As said

before, the highest vibration levels were measured at the rotational speed of 1400 rpm during the steady-state situation of the spinning process. Therefore, 1400 rpm operational measurement condition was selected for the decomposition of output signals. The contribution analysis for the selected operational condition over the complete frequency range (0-1600 Hz) is illustrated as percentage values in Figure 4.20. With respect to consequences of this contribution analysis, the most two significant contributors for this experiment are the horizontal (X) component of the right spring and vertical (Y) component of the left spring respectively.

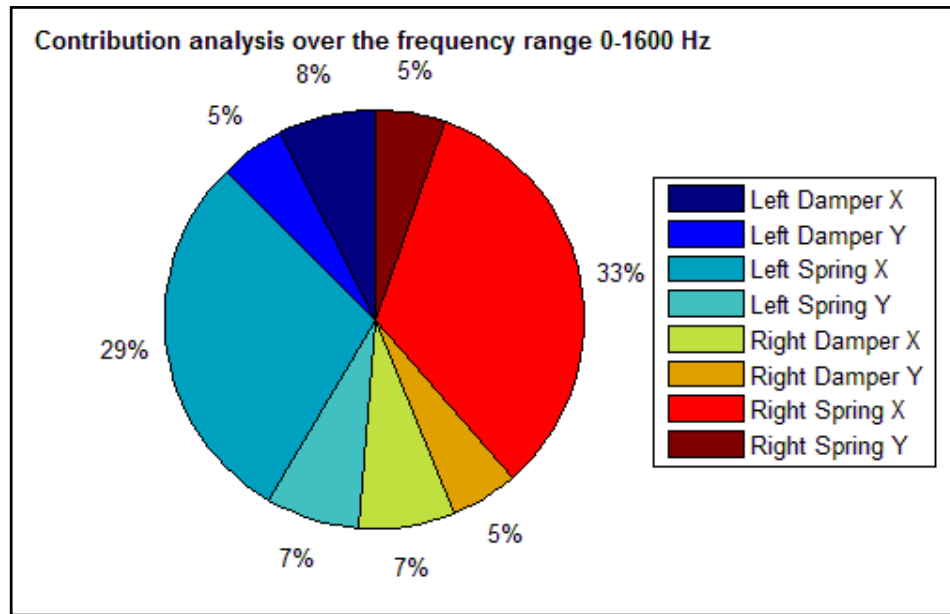


Figure 4.20 : Result of contribution analysis belongs to experimental case I, where the double effect friction dampers were used, obtained as percentage values for the 1400 rpm measurement condition over the complete frequency range 0-1600 Hz.

As mentioned before, the individual path contributions of each contributor can change when different frequency bands are considered. For example, if the contribution analysis is performed over the frequency ranges 0-100 Hz (see Figure 4.21) and 100-200 Hz (see Figure 4.22), it can be observed that the major and minor contributors are different in these examined frequency ranges. The most dominant transfer path over the frequency range 0-100 Hz is the horizontal (X) component of the right spring, as shown in Figure 4.21. On the other hand, the major contributor for the frequency range 100-200 Hz is the horizontal (X) component of the left spring, as demonstrated in Figure 4.22.

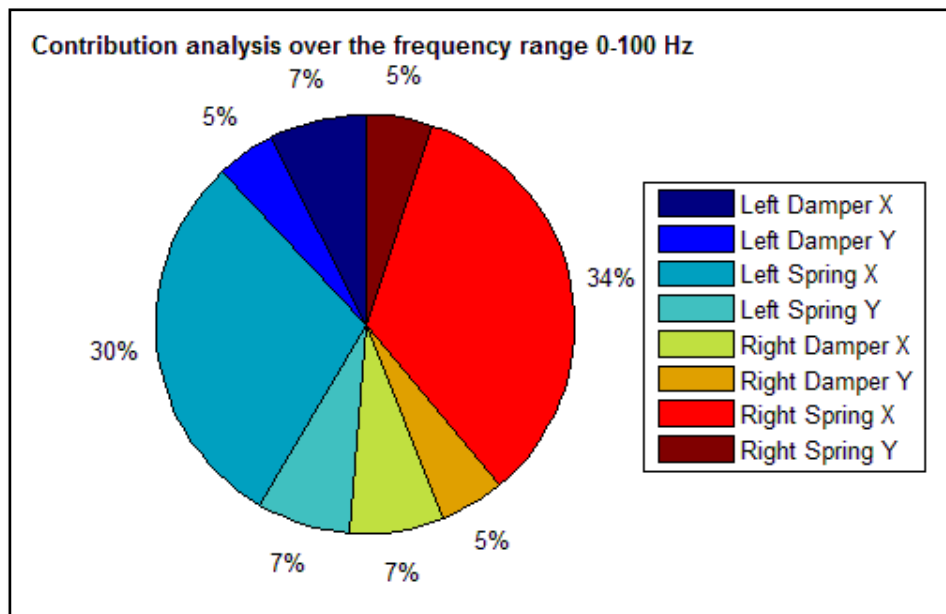


Figure 4.21 : Result of contribution analysis belongs to experimental case I, where the double effect friction dampers were used, obtained as percentage values for the 1400 rpm measurement condition over the frequency range 0-100 Hz.

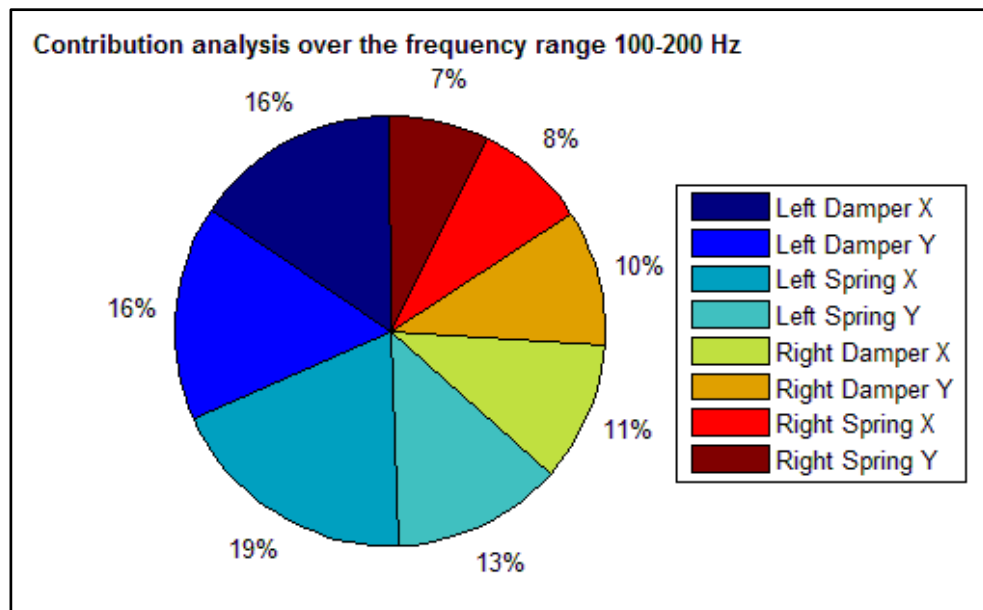


Figure 4.22 : Result of contribution analysis belongs to experimental case I, where the double effect friction dampers were used, obtained as percentage values for the 1400 rpm measurement condition over the frequency range 100-200 Hz.

As a summary, two different operational measurement sets were carried out for the application of OTPA to a washing machine. The main source excitation was altered by changing the rotational speed of spinning drum from 400 rpm to 1400 rpm, with a

step of 100 rpm. In first part of experimental case I, the single effect friction dampers were used, and in the second part, the double effect friction dampers were utilized in the suspension system of washing machine. Individual single effect friction damper can apply up to 80 N friction force to dissipate the vibration oscillations coming from the drive group of washing machine. The double effect friction dampers, on the other hand, transmits different amount of force at different vibration amplitudes. They are designed to transmit larger amount of force (for the damper used 120 N) when the vibration amplitudes are high (when the system is near or at resonance), and lower level of force when the system is away from resonance (i.e., at spinning speeds). Therefore, it is expected in experimental case I explained in this chapter that total contribution of friction dampers to the overall vibration response of the left side panel should be less when the double effect friction damper set is used.

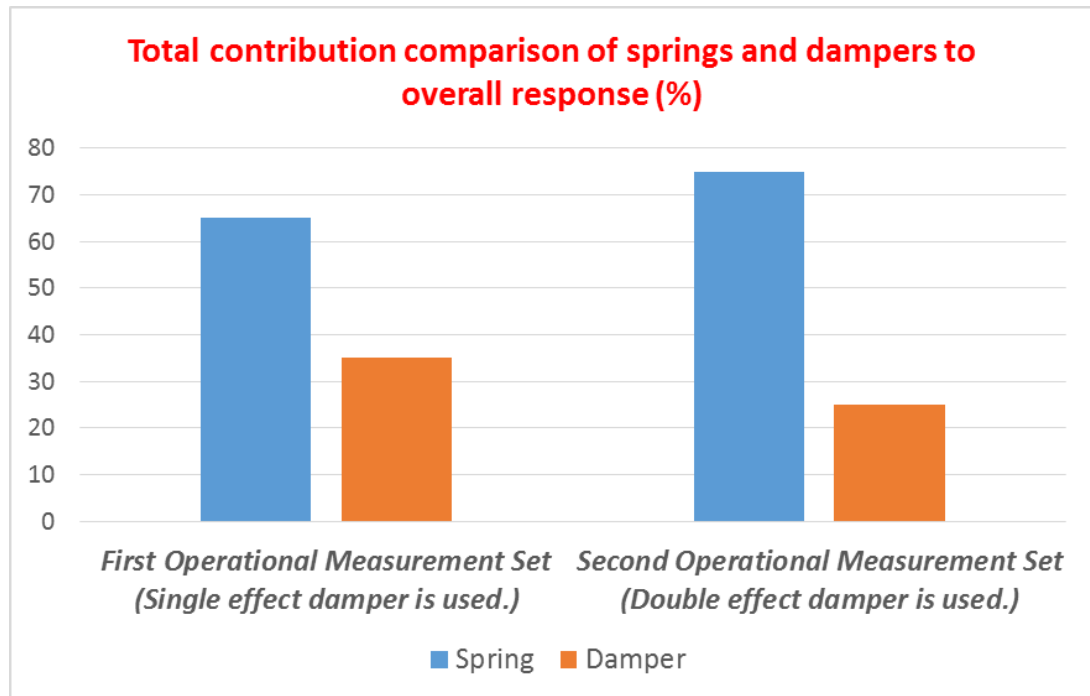


Figure 4.23 : Comparison of total contributions of the springs and dampers to overall response for the 1400 rpm measurement condition.

Considering the steady-state vibration measurements during the spinning process of mentioned washing machine for the OTPA, total contributions of springs and damping units to the overall response of the receiver were compared in Figure 4.23. The contribution of dampers to the panel vibration is around 35% for the analysis using first operational measurement set where the damping units are single effect friction dampers. In the analysis utilizing second operational measurement set, the

contribution of dampers to the panel vibration is reduced to 25% with the use of double effect friction dampers. It should be noticed that this comparison belongs to the result of the contribution analysis at 1400 rpm operational measurement condition where the highest vibration levels were measured. As a result, an overview of the contributions from the various input quantities characterized by the accelerometers at the corresponding measurement locations for this experiment represents that the highest predicted contributions come from the springs. It is vital to remember that only structure-borne vibro-acoustic transmission paths were considered for the washing machine application and the accelerations in a steady-state vibration were measured when the rotation speed of the spin drum was varied.

4.5.2 Results of experimental case II: creating different unbalance conditions for the washing machine

In this section, a parametric study is performed to determine how the condition number changes by creating various unbalance conditions for the application of OTPA procedure to the front-loading drum-type washing machine. For this reason, the study was divided into two parts, and each part has four different operational measurement sets. Every step of spinning cycle lasts for 90 sec; so that, as before, the measurement period was set as 60 sec for each operational condition. The input and output measurement locations (sensor positions) were kept the same as it was in experimental case I of this chapter. It is useful to remember that the input and output sensor positions are illustrated in Figure 4.5 and 4.10 respectively. To alter the operating conditions for the analysis, the operational measurements were made under different unbalance conditions while keeping the rotational speed of drum constant this time. The accelerations in a steady-state vibration were measured to obtain input and output quantities. Moreover, the various unbalance conditions were created by locating different amount of rubber plates inside the spinning drum for each operational measurement. The details of the locations of the unbalanced masses are given in Table 4.2.

For the determination of TF matrix here, SVD technique is implemented in the OTPA algorithm. The major step with SVD technique is the inversion of the singular values matrix [4]. The input matrix $\mathbf{X}$ is described by SVD with the following statement [11, 70]:

$$[X] = [U][\Sigma][V]^T \quad (4.6)$$

where $[\Sigma]$ is the singular value matrix, $[U]$ and $[V]$ are the unitary matrices, $[V]^T$ defines the complex conjugate transpose of $[V]$ [11, 70]. If diagonal elements of $[\Sigma]$ are very small non-zero values, the matrix is assigned as ill-conditioned and causes numerical error in the inversion operation. The condition number of the input matrix $[X(j\omega)]$ is stated as [4]:

$$c(X) = \|X\| \|X^{-1}\| \quad (4.7)$$

The condition number of $[X(j\omega)]$ is also determined as ratio of the largest singular value of $[X(j\omega)]$ to the smallest singular value of $[X(j\omega)]$ [4]. The condition number of the inverted matrix is proportional to the error dissemination in the operation of matrix inversion [4, 109]. When the condition number is small, the equations are expressed as well-defined and result in more reliable answers [4]. If the error amplification is limited by the condition number, this reduction in condition number can provide reduction of error amplification [28].

In the first part of experimental case II, the single effect friction dampers were used in the suspension system of washing machine. First of all, the rotational speed of drum was adjusted to 1200 rpm. The variation 1 and variation 2 given in Table 4.2 were used to position the unbalanced masses inside the drum for the first and second operational measurement sets respectively. After that, the spin velocity of drum was set to 1400 rpm. The variation 1 and variation 2 shown in Table 4.2 were used to position the unbalanced masses into the drum for the third and fourth operational measurement sets, respectively. Summarizing, the first part of this study contains four different operational experimental sets for the OTPA and the operational measurements were repeated for ten various operating conditions in each measurement set. The comparison of condition numbers for the explained operational measurement sets of the first part is given in Figure 4.24. Actually, the condition numbers of four operational measurement sets are close to each other. But, the smaller condition numbers are noticed when the rotational speed of drum was 1200 rpm and single unbalanced mass with varying amount of unbalance was positioned

inside the drum (variation 1). Therefore, this operational measurement set was chosen for OTPA.

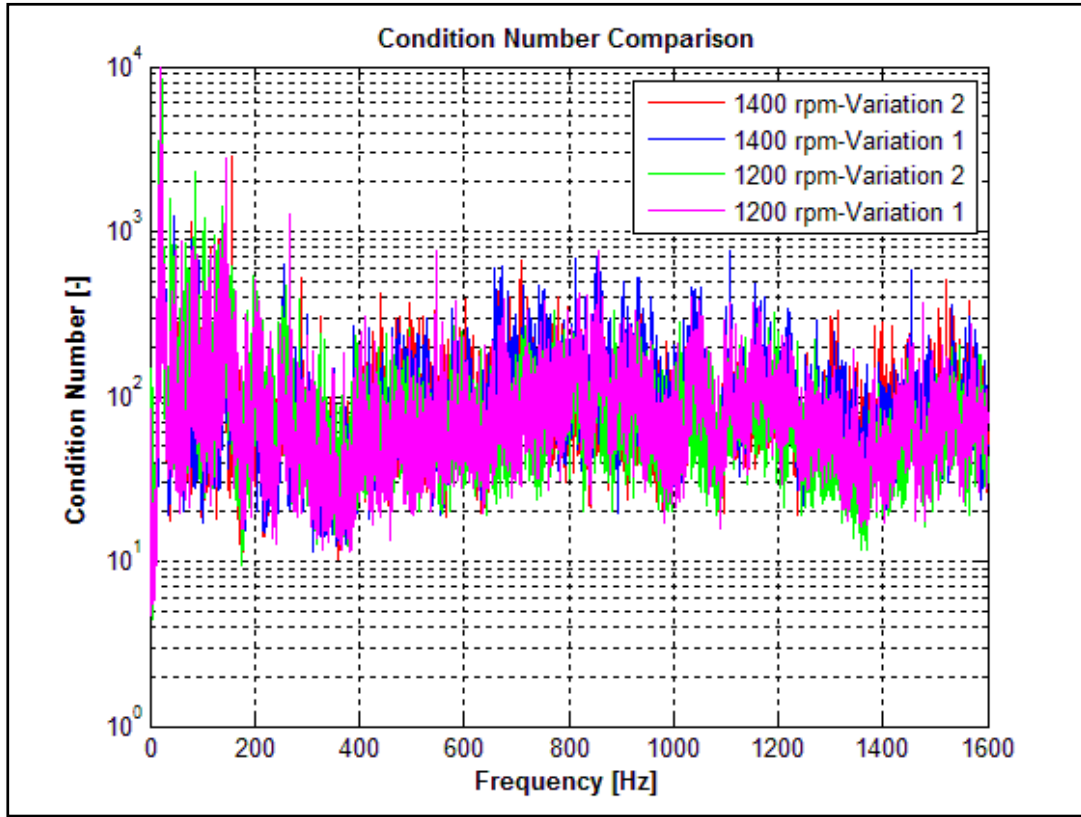


Figure 4.24 : Condition number comparison for the first part of experimental case II in logarithmic scale.

The MATLAB code developed that could run the OTPA algorithm was used for the operational TF estimation, overall response synthesization and contribution analysis in this experimental case. SVD technique was used for the inversion of the input matrix; thereby the singular values that are utilized in the TF matrix determination can be examined. Figure 4.25 represents all of the singular values of this experimental measurement; hence it should be observed that 8 different input variables exist for the analysis. Initially, the measured and synthesized response signals were compared for an arbitrary output location at an arbitrary operational measurement condition, as shown in Figure 4.26. As expected, these two output signals match very well with each other; so that the vibration of left side panel can be described by the utilized OTPA model with 8 structure-borne channels.

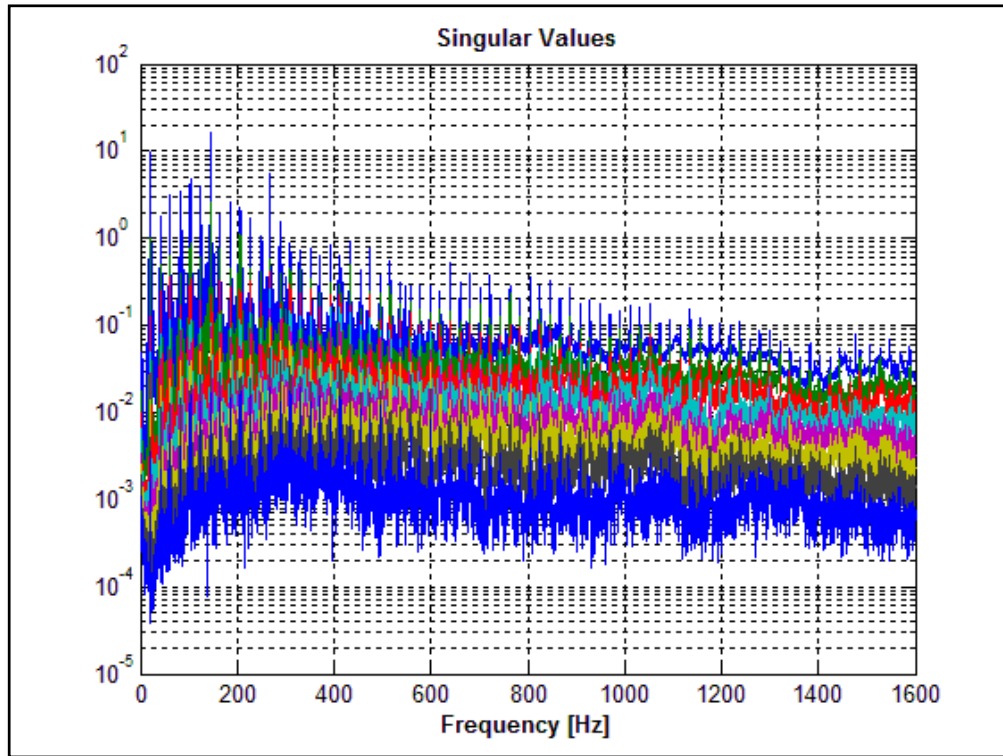


Figure 4.25 : All of the singular values of selected operational measurement set for the first part of experimental case II as a function of frequency in logarithmic scale.

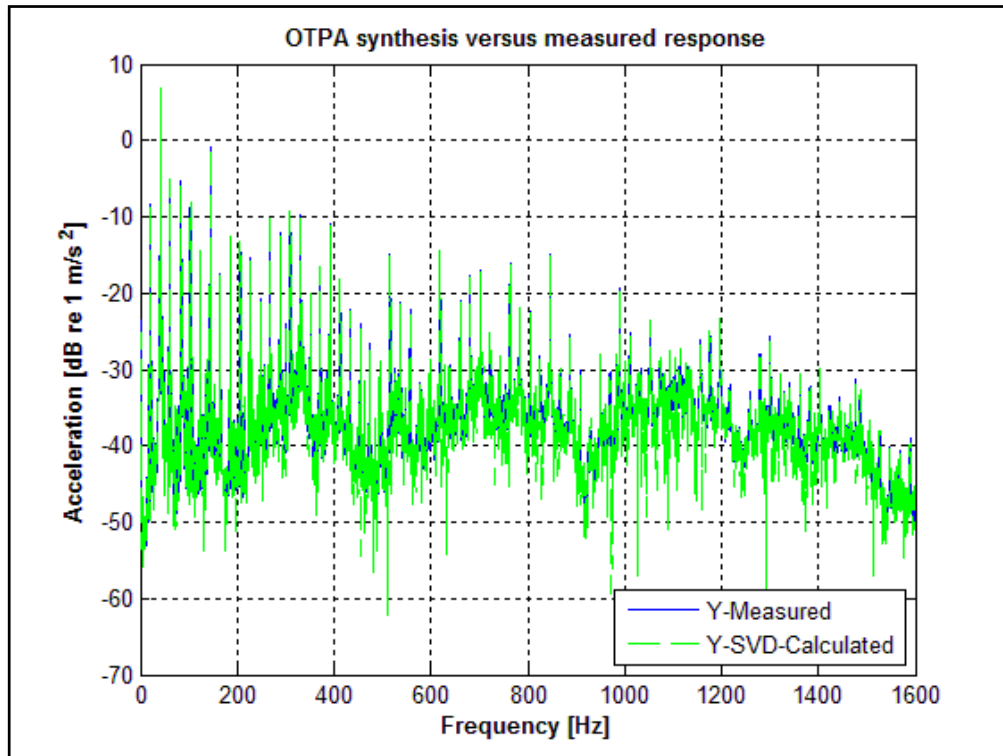


Figure 4.26 : Comparison of measured (blue) and calculated (green) response acceleration signals $Y_r^{(n)}$ in dB scale for a constant drum rotational speed of 1200 rpm in the first part of experimental case II.

For ranking the vibration transfer paths, the contributions of each input quantities at every output measurement locations were determined for each operational condition. Firstly, the contributions of each contributor in one operational condition were linearly averaged for four response locations to demonstrate complete vibration characteristic of the receiver. After that, entire phase-assigned spectrum measurement blocks of the complete operational conditions were averaged to obtain an overall indication of the model's fit. Therefore, averaged individual path contributions of each input quantity were used to identify the most critical vibration transfer path. The averaged path contributions of the washing machine from all of the selected operational measurement set over the complete frequency range (0-1600 Hz) is shown as percentage values in Figure 4.27. According to results of this contribution analysis, the most two significant contributors for this experiment are the vertical (Y) component of the left spring and the horizontal (X) component of the right damper, respectively. It is important to note that the computed percentage values of each contributor symbolize the total RMS value of the corresponding signal for the desired frequency range.

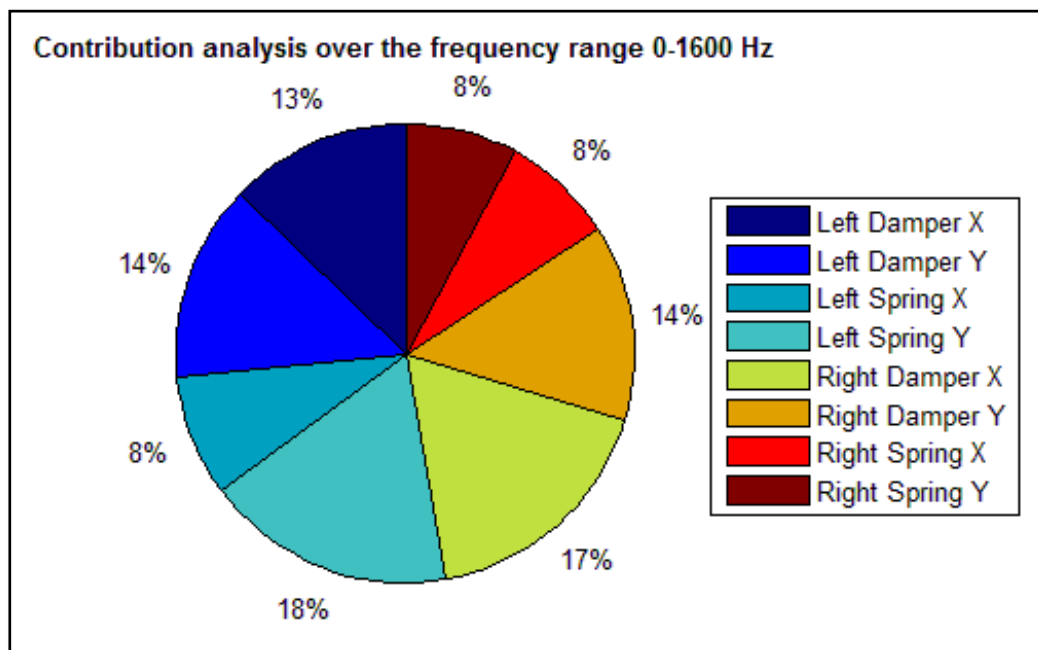


Figure 4.27 : Result of contribution analysis belongs to experimental case II, where the single effect friction dampers were used, obtained as percentage values by averaging entire measurement blocks over the complete frequency range 0-1600 Hz.

The ranking of the average path contributions of individual contributors can differ in every frequency band. For instance, if the contribution analysis is carried out over the frequency bands 400-800 Hz (see Figure 4.28) and 1000-1200 Hz (see Figure 4.29), it can be observed that the most dominant vibration transfer paths are different in these examined frequency ranges.

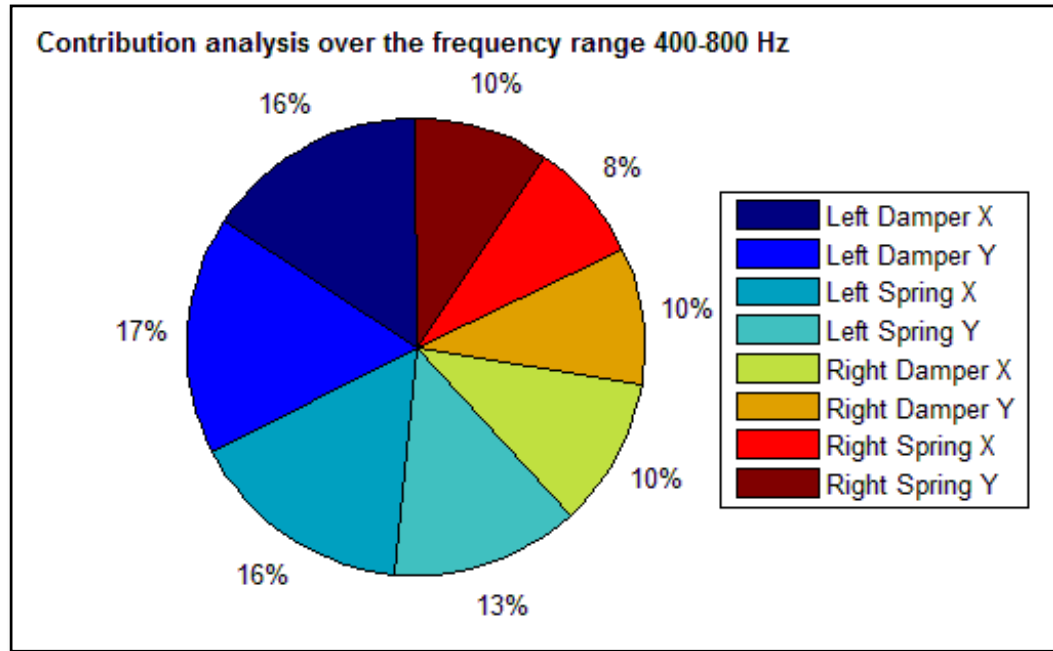


Figure 4.28 : Result of contribution analysis belongs to experimental case II, where the single effect friction dampers were used, obtained as percentage values by averaging entire measurement blocks over the frequency range 400-800 Hz.

The most dominant vibro-acoustic transmission paths over the frequency range 400-800 Hz seem to be the horizontal (X) and vertical (Y) components of the left damper, as illustrated in Figure 4.28. In addition, the horizontal (X) and vertical (Y) components of the left spring have relatively high contributions to overall vibration level of the receiver. On the other hand, the major contributors for the frequency range 1000-1200 Hz is the horizontal (X) and vertical (Y) components of the right damper, as shown in Figure 4.29. Besides, the horizontal (X) component of the left damper has higher contribution than the other remain vibro-acoustic transfer pathways. Consequently, it is vital to remember that examining the results of contribution analysis in different frequency ranges is useful to identify the most dominant vibration transmission path in the related frequency range.

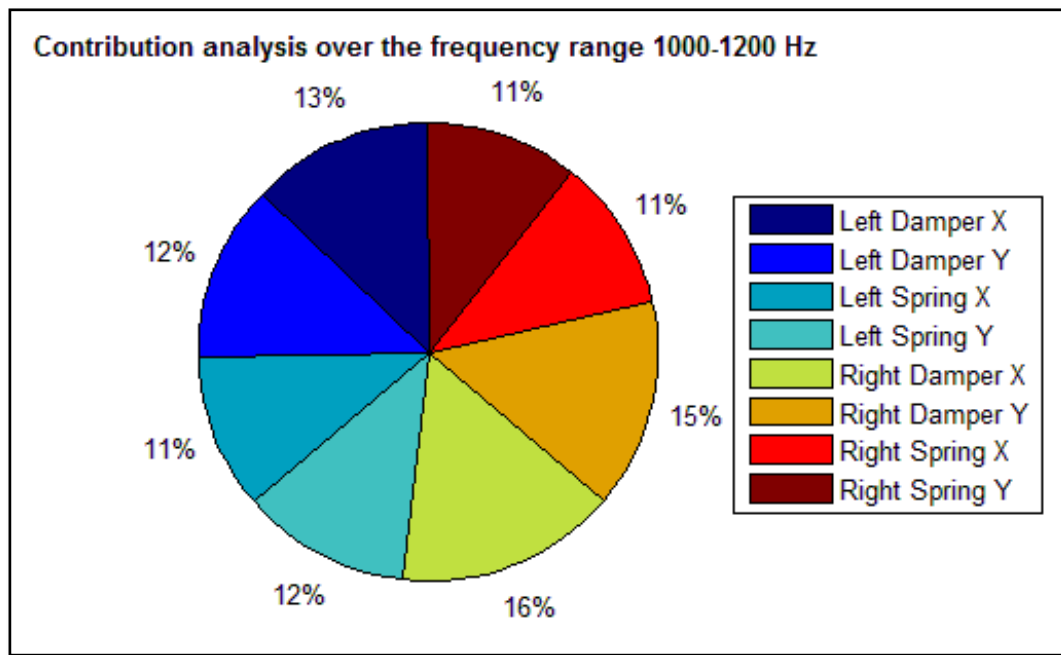


Figure 4.29 : Result of contribution analysis belongs to experimental case II, where the single effect friction dampers were used, obtained as percentage values by averaging entire measurement blocks over the frequency range 1000-1200 Hz.

In the second part of the experimental case II, the double effect friction dampers were employed in the suspension system of washing machine. Initially, the spin velocity of drum was set to 1200 rpm. The variation 1 and variation 2 illustrated in Table 4.2 were used to place the unbalanced masses into the drum for the first and second operational measurement sets respectively. After that, the rotational speed of spinning drum was adjusted to 1400 rpm. The variation 1 and variation 2 shown in Table 4.2 were used again to position the unbalanced masses inside the drum for the third and fourth operational measurement sets respectively. As a summary, the second part of this study includes four various operational experiment sets for the OTPA and the operational measurements were repeated for ten various working conditions in each measurement set. The comparison of condition numbers for the explained operational measurement sets of second part is given in Figure 4.30. Indeed, the condition numbers of four operational measurement sets are close to each other. However, slightly smaller condition numbers are seen when the rotational speed of drum was 1200 rpm and single unbalanced mass was positioned inside the drum (variation 1). For this reason, this operational measurement set was chosen for the analysis.

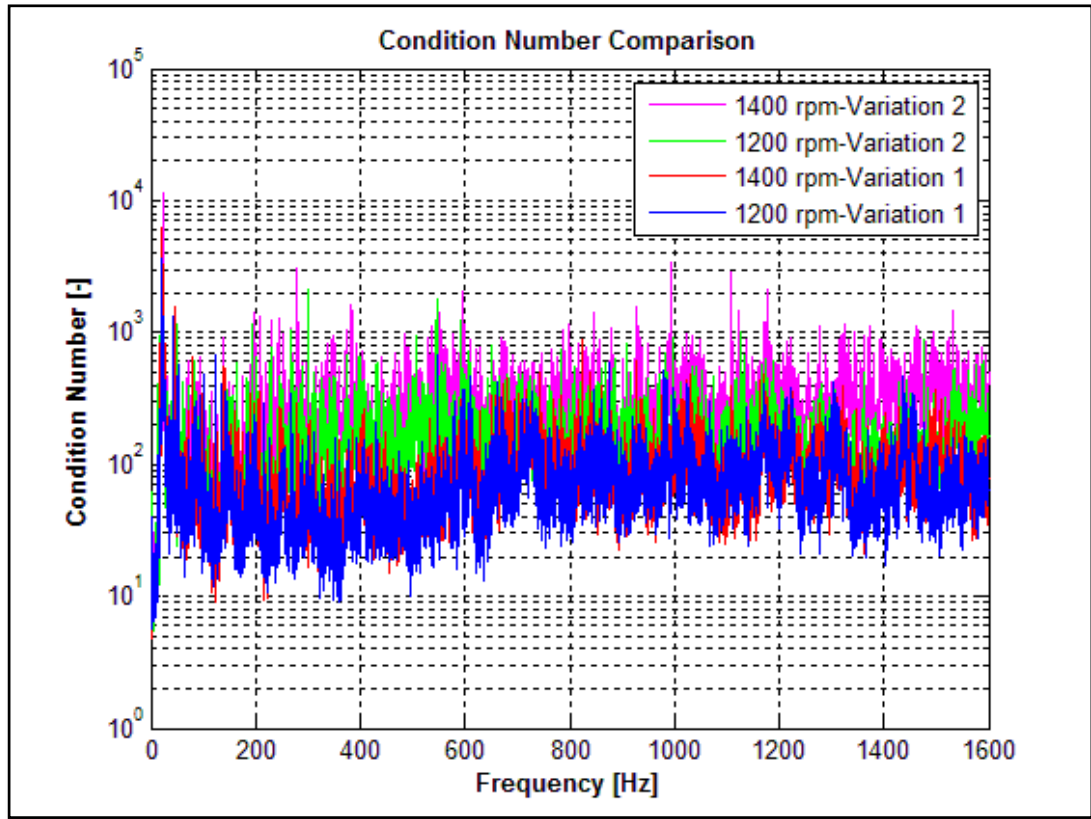


Figure 4.30 : Condition number comparison for the second part of experimental case II in logarithmic scale.

With the use of developed MATLAB code that could run the OTPA algorithm, the operational TFs were estimated, overall responses were resynthesized and vibration transmission paths were decomposed in this experimental case. As explained in the previous chapter, if the singular values of input matrix are not changed by applying a threshold or not normalized by using a reference value, LSA and SVD techniques can give the same results in the process of input matrix inversion. For this reason, SVD method was used for the inversion of input matrix; so that the singular values that are used in the TF matrix estimation can be investigated. Entire singular values of this experimental measurement are given in Figure 4.31; thus it can be said that there are 8 independent input quantities for the analysis. After that, the measured and calculated response signals were compared for an arbitrary output location at an arbitrary operational measurement condition, as demonstrated in Figure 4.32. As expected, the originally measured and synthesized output signals compare very well with each other; hence the established OTPA model can be used to describe the vibration response of left side panel with 8 structure-borne input quantities.

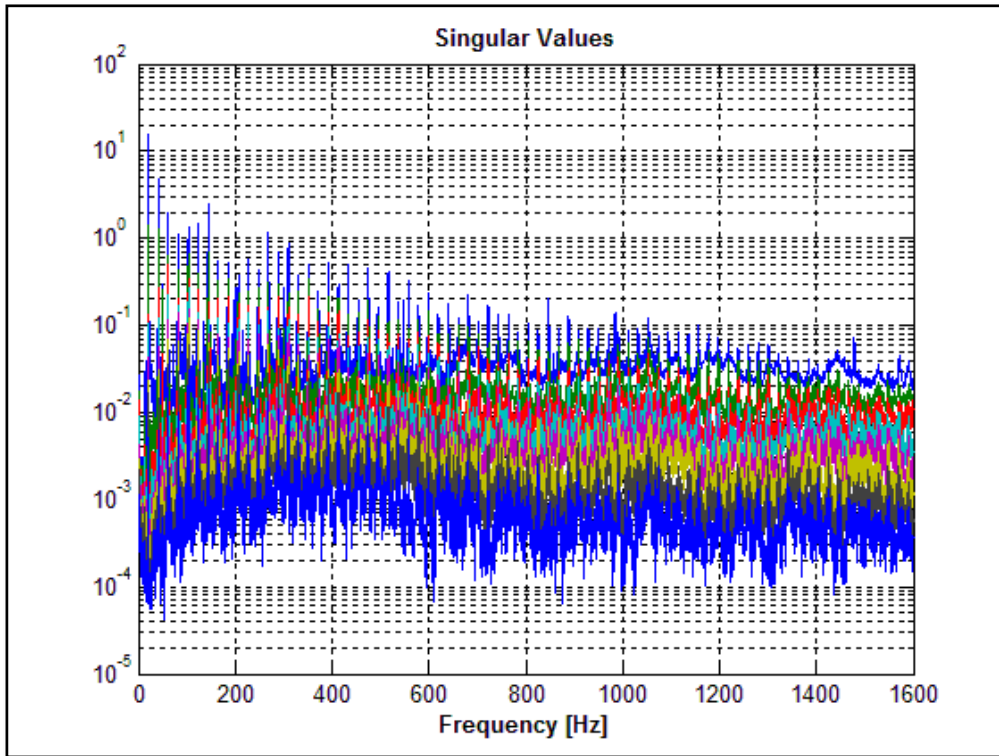


Figure 4.31 : All of the singular values of selected operational measurement set for the second part of experimental case II as a function of frequency in logarithmic scale.

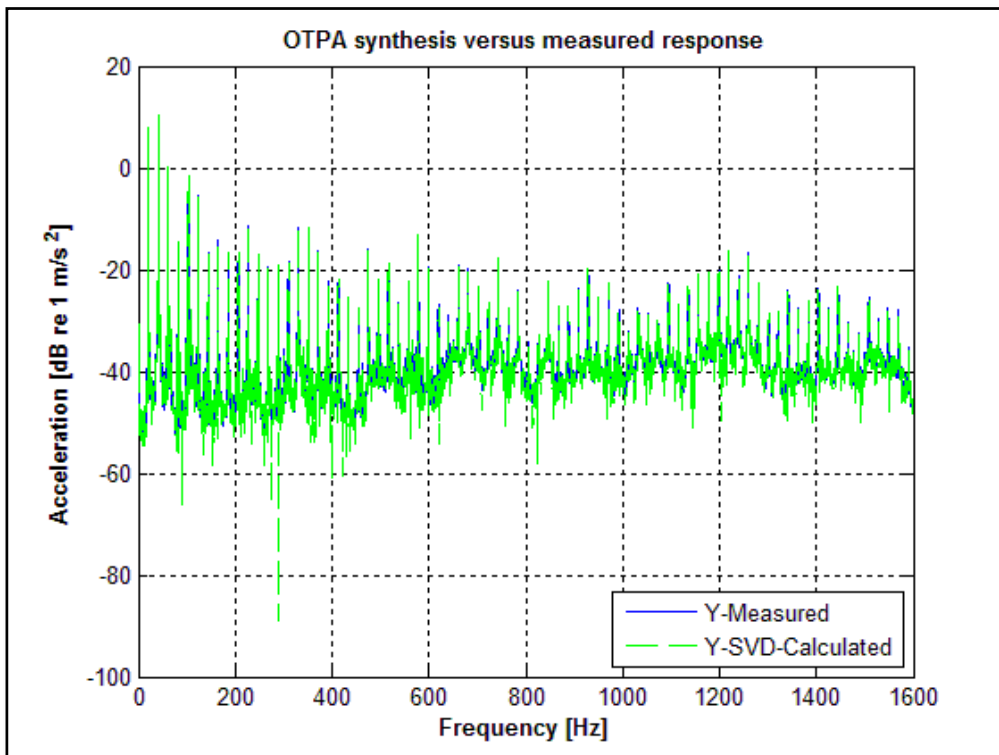


Figure 4.32 : Comparison of measured (blue) and calculated (green) response acceleration signals $Y_r^{(n)}$ in dB scale for a constant drum rotational speed of 1200 rpm in the second part of experimental case II.

To rank the vibration transfer paths, the contribution analysis was applied using the same procedure as in the first part of the experimental case II. The averaged path contributions of the washing machine from all of the selected operational measurement set over the complete frequency range (0-1600 Hz) is illustrated as percentage values in Figure 4.33. This contribution analysis suggests that the vertical (Y) component of the left damper is the most dominant transfer path for this experiment. It is essential to remember that the computed percentage values of each contributor symbolize the total RMS value of the corresponding signal for the desired frequency range. It is seen once again that the averaged individual path contributions can vary depending on the frequency band selected. For example, when the contribution analysis is applied over the frequency bands 400-800 Hz (see Figure 4.34) and 1000-1200 Hz (see Figure 4.35), it can be noticed that the most dominant vibration transfer paths are different for these examined frequency ranges. The most dominant transfer path over the frequency range 400-800 Hz is the horizontal (X) and vertical (Y) components of the left damper, as given in Figure 4.34. On the other hand, the major contributor for the frequency range 1000-1200 Hz is the vertical (Y) component of the right damper, as represented in Figure 4.35. It should also be noted that X and Y contributions of the right spring are also relatively high.

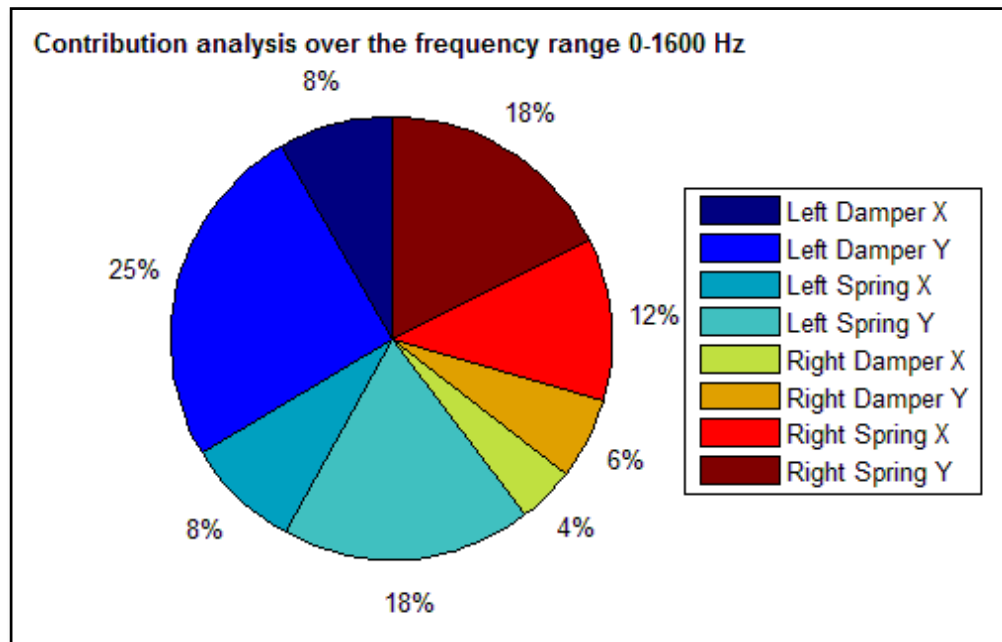


Figure 4.33 : Result of contribution analysis belongs to experimental case II, where the double effect friction dampers were used, obtained as percentage values by averaging entire measurement blocks over the complete frequency range 0-1600 Hz.

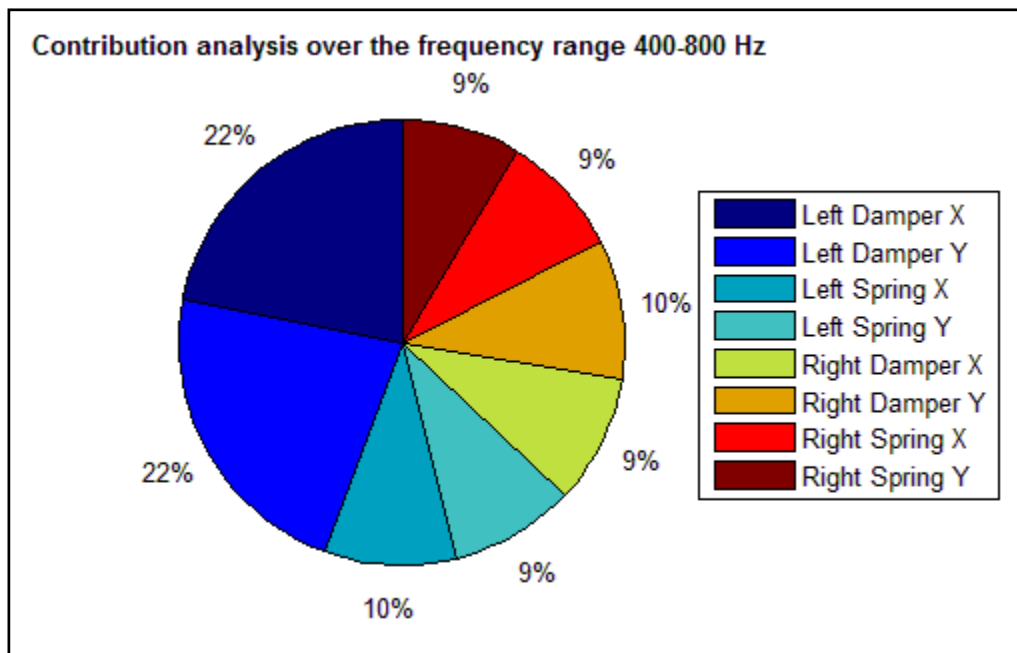


Figure 4.34 : Result of contribution analysis belongs to experimental case II, where the double effect friction dampers were used, obtained as percentage values by averaging entire measurement blocks over the frequency range 400-800 Hz.

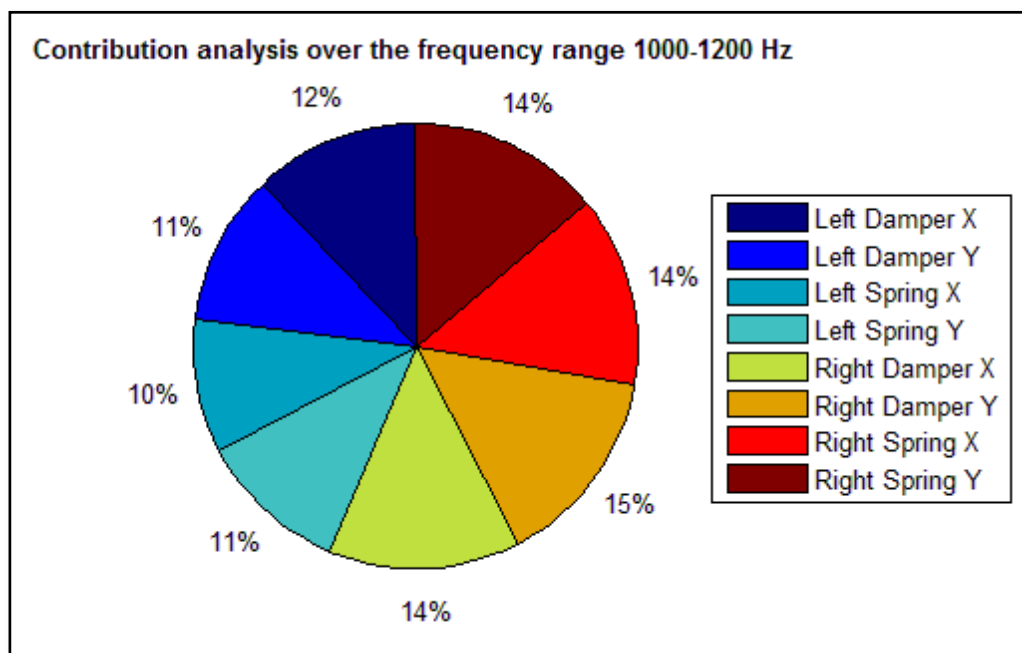


Figure 4.35 : Result of contribution analysis belongs to experimental case II, where the double effect friction dampers were used, obtained as percentage values by averaging entire measurement blocks over the frequency range 1000-1200 Hz.

To sum up, two various operational measurement sets were carried out by comparing the condition numbers for the application of OTPA to a washing machine. The main

source excitation was altered by creating different unbalance conditions inside the washing machine drum. In first part of experimental case II, the single effect friction dampers were used, and in the second part, the double effect friction dampers were utilized in the suspension system of washing machine.

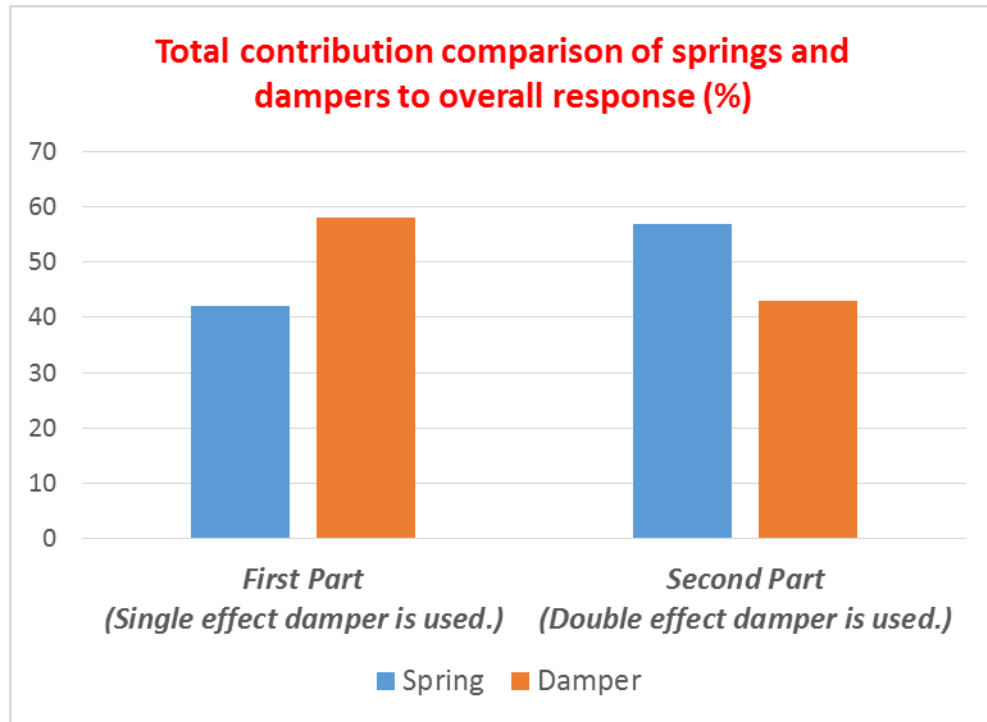


Figure 4.36 : Comparison of total averaged contributions of the springs and dampers to overall response for experimental case II.

Assuming the steady-state vibration measurements during the process of spinning cycle of the washing machine for the OTPA, total averaged contributions of springs and friction dampers to the overall averaged response of the receiver were compared in Figure 4.36. The averaged contribution of friction dampers to the panel vibration is around 58% for the OTPA of first part using selected operational measurement set, where the damping units are single effect friction dampers. In the OTPA of second part using chosen operational measurement set, the contribution of dampers to the panel vibration is reduced to 43% with the use of double effect friction dampers. Finally, an overview of the contributions from the different input channels described by the accelerometers at the corresponding measurement places for these experiments indicates that the highest estimated contributions come from the dampers in the first part of experimental case II. On the other hand, the highest estimated contributions originate from the springs in the second part of experimental

case II. It should be noted that the airborne vibro-acoustic transmission paths were ignored for the washing machine application.

4.6 Summary

The aim of this chapter was to identify the most dominant vibration transfer paths of a washing machine. The critical components of a front-loading drum-type washing machine were firstly explained. The source-transfer paths-receiver model for the OTPA was established with certain assumptions. Modal analysis as well as vibration measurements were performed on a washing machine for the selection of appropriate output and input sensor positions. The ways of how the operational working conditions could be varied (how the main source excitation could be altered) were presented with requirements. The measurement setup was introduced, and experimental measurements were familiarized in detail. A MATLAB code was developed to process the OTPA algorithm for the estimation of TF matrix, overall response signal synthesization and contribution analysis. At the end, two experimental cases for the washing machine were presented with the consequences. The results of contribution analysis demonstrated that the separated individual path contributions of each input variables can change depending of frequency range of interest. OTPA confirms that the use of double effect dampers in the suspension system of washing machine reduces the vibration transmitted through the damping units to a panel of the washing machine.

5. CONCLUSIONS AND FUTURE RECOMMENDATIONS

5.1 Conclusions

The main objective of Transfer Path Analysis (TPA) is to determine vibration and/or noise contributions from various sources (i.e. source contributions) or to identify vibration and/or noise transmitted to a receiver through various transfer paths (i.e. source path analysis). The Operational TPA is more practical, reliable and accurate signal processing technique than the classical TPA. In this thesis, it is aimed to validate the OTPA procedure implemented in MATLAB environment and then to perform the OTPA on a more complex system, e.g. washing machine. Therefore, two main applications were the subjects of this thesis. For the verification of the OTPA procedure implemented in a MATLAB code, an application to a simple flat plate is presented. After that, an application to a washing machine is introduced for a more detail examination of OTPA.

In Chapter 1, the problem is defined and a detailed literature survey is presented to figure out the application areas of TPA. The objectives and outline of thesis are given. In Chapter 2, the theoretical backgrounds of Classical and Operational TPA are explained to realize the analytical framework behind the numerical and experimental applications employed in this thesis.

In Chapter 3, an application to a simple flat plate is carried out in various numerical and experimental case studies to validate the MATLAB code developed for understanding the basic stages of the OTPA procedure. It is observed in both of the numerical and experimental studies that the use of phase-assigned spectrum and linear spectrum averaging are required for the correct implementation of the OTPA method. It is shown that accurate synthesizing of overall response signal is not sufficient for satisfactory contribution analyses, correct estimation of MIMO FRFs and their phase angles is also required to get more reliable results. Furthermore, it is noticed that LSA and SVD techniques yield the same result for the prediction of

operational TF (or FRF) matrix. It is also shown that the MATLAB code developed in this thesis is able to estimate MIMO FRFs and their phase angles accurately.

In Chapter 4, OTPA is applied to a washing machine for the identification of vibration problems generated from the drive group. The source-transfer paths-receiver model for the analysis is established and the significant vibro-acoustic transmission paths are defined on the structure. Displacements in a steady-state vibration are measured on the washing machine tub to identify in which direction the most vibration transmitted to the panel. Modal analysis of washing machine is made to determine the best accelerometer response locations as the significant receiver locations. The airborne transfer paths, the vibration transmission to the washing machine cabinet along the Z-direction and the contribution of the bellow are ignored in OTPA. The inputs to the transfer paths are assumed as input variables; hence transmissibility approach is used for the operational Transfer Function (TF) estimation. The operational conditions are varied by changing the rotational speed of the drum and generating various unbalanced conditions inside the spinning drum. The operational measurements are recorded using PULSE LabShop software. The required data for the contribution analysis is obtained by the MATLAB code developed in this thesis. At the end, two different experimental cases for two different damping units are presented with the consequences. It is observed that the major contributor to overall panel vibration can change from one frequency band to another. Moreover, this may change under different main source excitation. On the other hand, OTPA clearly shown under different operational measurement conditions that the use of double effect dampers provides reduction in the transmitted vibrations compared to those of single effect dampers.

5.2 Future Recommendations

The MATLAB code developed in the scope of this thesis can be used for the application of OTPA to more complex mechanical systems; thereby the accuracy of the numerical tool developed here can be tested further. The airborne transfer paths can be taken into account for a new washing machine application to separate the transfer paths more accurately. Also, the contribution of bellow to the overall vibration response of washing machine can be considered as an input variable.

Finally, the regularization methods can be implemented into the OTPA algorithm to obtain more reliable solutions.

REFERENCES

- [1] **Plunt, J.**, (1998). Strategy transfer path analysis (TPA) applied to vibro-acoustic systems at medium and high frequencies, *Proceedings of the 23rd International Seminar on Modal Analysis (ISMA)*, Katholieke Universiteit Leuven, Leuven, Belgium, September 16-18, Vol. 2, pp. 1025-1030.
- [2] **Plunt, J.**, (2005). *Finding and fixing vehicle NVH problems with transfer path analysis*, Sound and Vibration, Vol. 39, no. 11, pp. 12-16.
- [3] **LMS International**, (2005). *Transfer path analysis: The qualification and quantification of vibro-acoustic transfer paths*, < http://www.plm.automation.siemens.com/en_us/products/lms/ >, date retrieved 20.03.2014.
- [4] **Van Karsen, C. D., Gwaltney, G., and Blough, J.**, (2000). Applying transfer path analysis to large home appliances, *Proceedings of the 17th International Modal Analysis Conference (IMAC)*, Vol. 17.
- [5] **Zheng, S., Dan, J., Hao, P., and Lian, X.**, (2013). Contribution of pass-by noise sources with acoustic holograph and dynamic transfer path model, *Proceedings of the FISITA 2012 World Automotive Congress*, Springer Berlin Heidelberg, pp. 637-643.
- [6] **Sievi, A., Martner, O., and Lutzenberger, S.**, (2012). Noise reduction of trains using the operational transfer path analysis - Demonstration of the method and evaluation by case study, *Noise and Vibration Mitigation for Rail Transportation Systems*, NNFM 18, Springer Japan, pp. 453-461.
- [7] **Roozen, N. B., Leclere, Q., and Sandier, C.**, (2012). Operational transfer path analysis applied to a small gearbox test setup, *Proceedings of Acoustics 2012*, Nantes, France.
- [8] **Mucchi, E., and Vecchio, A.**, (2013). Experimental vibro-acoustic transfer path analysis on helicopters, *Proceedings of the ASME 2013 International Design Engineering Technical Conferences and Computers and Information in Engineering Conference (IDETC/CIE)*, Portland, Oregon, USA, August 4-7, pp. V008T13A008-V008T123A008.
- [9] **Kwak, H. T., Yoon, J. H., Yang, I. H., Lee, J. Y., and Oh, J. E.**, (2010). Noise source identification and noise reduction of PDP TV using transfer path analysis, *Proceedings of the ASME 2010 International Mechanical Engineering Congress and Exposition (IMECE)*, Vancouver, British Columbia, Canada, November 12-18, pp. 101-103.

- [10] **Sung, S. H., and Nefske, D. J.,** (2012). Transfer path analysis of body panel participation using a structural-acoustic finite element model, *Proceedings of the ASME 2012 Noise Control and Acoustics Division (NCAD) Conference at InterNoise 2012*, New York City, NY, USA, August 19-22, pp. 351-357.
- [11] **De Klerk, D., and Ossipov, A.,** (2010). Operational transfer path analysis: Theory, guidelines and tire noise application, *Mechanical Systems and Signal Processing*, Vol. 24, Issue 7, pp. 1950-1962.
- [12] **Kim, K. S., and Kang, Y. J.,** (2011). Local stiffness control for reducing vehicle interior noise by using FRF-based synthesis method, *Journal of Mechanical Science and Technology*, Vol. 25, Issue 1, pp. 81-88.
- [13] **Fang, Z., Hong, Z., and Erbing W.,** (2012). Matrix inversion method for load identification in transfer paths analysis, *Future Communication, Computing, Control and Management*, Springer Berlin Heidelberg, Vol. 141, pp. 517-524.
- [14] **Le, M., Hong Z., Erbing W., Haiqing, X., and Lang, S.,** (2012). Research on the effect of paths coupling of transfer path analysis method, *Future Communication, Computing, Control and Management*, Springer Berlin Heidelberg, Vol. 141, pp. 543-550.
- [15] **Inoue, A., and Tanabe, Y.,** (2012). Development of particle velocity transfer path analysis, *Proceedings of the ASME 2012 Noise Control and Acoustics Division (NCAD) Conference at InterNoise 2012*, New York City, NY, USA, August 19-22, pp. 359-368.
- [16] **Xie, R., Liu Z., Long, S., and Tian, Z.,** (2013). Test and simulation integrated transfer path analysis and optimization of the steering wheel vibration in idle, *Proceedings of the FISITA 2012 World Automotive Congress*, Springer Berlin Heidelberg, pp. 405-415.
- [17] **Choi, H. G., Thite, A. N., and Thompson, D. J.,** (2006). A threshold for the use of Tikhonov regularization in inverse force determination, *Applied Acoustics*, Vol. 67, Issue 7, pp. 700-719.
- [18] **Choi, H. G., Thite, A. N., Thompson, D. J.,** (2007). Comparison of methods for parameter selection in Tikhonov regularization with application to inverse force determination, *Journal of Sound and Vibration*, Vol. 304, Issue 3, pp. 894-917.
- [19] **Thite, A. N., and Thompson, D. J.,** (2003). The quantification of structure-borne transmission paths by inverse methods. Part 1: Improved singular value rejection methods, *Journal of Sound and Vibration*, Vol. 264, Issue 2, pp. 411-431.
- [20] **Thite, A. N., and Thompson, D. J.,** (2003). The quantification of structure-borne transmission paths by inverse methods. Part 2: Use of regularization techniques, *Journal of Sound and Vibration*, Vol. 264, Issue 2, pp. 433-451.

- [21] **Kim, S. J., and Lee, S. K.,** (2008). Prediction of interior noise by excitation force of the powertrain based on hybrid transfer path analysis, *International Journal of Automotive Technology*, Vol. 9, Issue 5, pp. 577-583.
- [22] **Vecchio, A., Van der Auweraer, H., Kindt, P., Houthoofd, T., and Sas, P.,** (2005). Experimental transfer path analysis of a hybrid bus, *SAE Technical Paper*, No. 2005-01-2335.
- [23] **De Klerk, D., and Rixen, D. J.,** (2010). Component transfer path analysis method with compensation for test bench dynamics, *Mechanical Systems and Signal Processing*, Vol. 24, Issue 6, pp. 1693-1710.
- [24] **Tcherniak, D., and Schuhmacher, A. P.,** (2008). Application of decomposition-based technique in NVH source contribution analysis, *Proceedings of ISMA-2008*, Leuven, Belgium.
- [25] **Tcherniak, D., and Schuhmacher, A. P.,** (2009). Application of transmissibility matrix method to NVH source contribution analysis, *Proceedings of IMAC-XXVII, Society for Experimental Mechanics*, Orlando, Florida, USA, February 9-12.
- [26] **Kocoglu, S. T.,** (2012). Operational transfer path analysis and application on test rig, *M.Sc. Thesis*, Istanbul Technical University, Istanbul, Turkey. (in Turkish)
- [27] **Lindberg, E., Hörlin, N.-E., and Göransson, P.,** (2012). An experimental study of interior vehicle roughness noise from disc brake system, *Applied Acoustics*, Vol. 74, Issue 3, pp. 396-406.
- [28] **Biermayer, W., Brandl F., Hoeldrich R., Sontacchi, A., Brandl, S., and Pribsch., H. H.,** (2008). Efficient transfer path analysis for vehicle sound engineering, *Proceedings of the JSAE 2008 Annual Congress*, Yokohama, Japan, May 21-23, Paper No. 20085030.
- [29] **Elliot, A. S., Moorhouse, A. T., Huntley, T., and Tate, S.,** (2013). In-situ source path contribution analysis of structure borne road noise, *Journal of Sound and Vibration*, Vol. 332, Issue 24, pp. 6276-6295.
- [30] **Guasch, O.,** (2009). Direct transfer function and path blocking in a discrete mechanical system, *Journal of Sound and Vibration*, Vol. 321, Issue 3, pp. 854-874.
- [31] **Guasch, O., Garcia, C., Jove, J., and Artis, P.,** (2013). Experimental validation of the direct transmissibility approach to classical transfer path analysis on a mechanical setup, *Mechanical Systems and Signal Processing*, Vol. 37, Issue 1, pp. 353-369.
- [32] **Bendat, J. S.,** (1976). System identification from multiple input/output data, *Journal of Sound and Vibration*, Vol. 49, Issue 3, pp. 293-308.
- [33] **Bendat, J. S., and Pierson, A. G.** (1980). *Engineering applications of correlation and spectral analysis*, Wiley, New York.
- [34] **Janssens, K., Gajdatsy, P., Gielen, L., Mas, P., Britte, L., Desmet., W., and Van der Auweraer, H.,** (2011). OPAX: A new transfer path analysis method based parametric load models, *Mechanical Systems and Signal Processing*, Vol. 25, Issue 4, pp. 1321-1338.

- [35] **De Klerk, D., Lohrmann, M., Quickert, M., and Foken, W.,** (2009). Application of operational transfer path analysis on a classic car, *Proceedings of NAG/DAGA 2009 International Conference on Acoustics*, Rotterdam, Holland, March 23-26.
- [36] **De Sitter, G., Devriendt, C., Guillaume, P., and Pruyt, E.,** (2010). Operational transfer path analysis, *Mechanical Systems and Signal Processing*, Vol. 24, Issue 2, pp. 416-431.
- [37] **Yan, L., and Jiang, W.,** (2014). Research on the procedure for analyzing the sound quality contribution of sound sources and its application, *Applied Acoustics*, Vol. 79, pp. 75-80.
- [38] **Lohrmann, M., and Hohenberger, T.,** (2008). Operational transfer path analysis: Comparison with conventional method, *Journal of the Acoustical Society of America*, Vol. 123, no. 5, pp. 3534.
- [39] **Kim, Y.-D., Jeong, J.-E., Park, J.-S., Yang, I.-H., Park, T.-S., Muhamad, P.-B., Choi, D.-H., and Oh, J.-E.,** (2013). Optimization of the lower arm of a vehicle suspension system for road noise reduction by sensitivity analysis, *Mechanism and Machine Theory*, Vol. 69, pp. 278-302.
- [40] **Gajdatsy, P., Janssens, K., Desmet, W., and Van der Auweraer, H.,** (2010). Application of the transmissibility concept in transfer path analysis, *Mechanical Systems and Signal Processing*, Vol. 24, Issue 7, pp. 1963-1976.
- [41] **Roozen, N. B., and Leclère, Q.,** (2013). On the use of artificial excitation operational transfer path analysis, *Applied Acoustics*, Vol. 74, Issue 10, pp. 1167-1174.
- [42] **De Klerk, D.,** (2009). Dynamic response characterization of complex systems through operational identification and dynamic substructuring, *Ph.D. Thesis*, Delft University of Technology, Delft, Holland.
- [43] **Toome, M.,** (2012). Operational transfer path analysis: a study of source contribution predictions at low frequency, *M.Sc. Thesis*, Chalmers University of Technology, Gothenburg, Sweden.
- [44] **Magrans, F. X.,** (1981). Method of measuring transmission paths, *Journal of Sound and Vibration*, Vol. 74, Issue 3, pp. 321-330.
- [45] **Verheij, J.,** (1986). Multipath sound transfer from resiliently mounted shipboard machinery, *Ph.D. Dissertation*, Technische Physische Dienst TNO-TH, Delft, The Netherlands.
- [46] **De Vis, D., Hendricx, W., and Van der Linden, P. J. G.,** (1992). Development and integration of an advanced unified approach to structure boom noise analysis, *Proceedings of the 2nd International Conference on Vehicle Comfort*, ATA.
- [47] **Van der Linden, P. J. G., De Meester, K., and Le Martret, J. P.,** (1996). Airborne source strength measurements on the basis of pressure indicator inversion applied to an engine valve cover, *Proceedings of the 4th ICSV*, St-Petersburg, Russia.

- [48] **Van der Linden, P. J. G., Keppens, T., Raff, J., and Schnur, J.,** (2001). Determination of the noise contributions of engine surfaces, *Proceedings of the SAE Noise and Vibration Conference*, US, Paper No: 2001-01-1482.
- [49] **Bendat, J. S.,** (1976). Solutions for the multiple input/output problem, *Journal of Sound and Vibration*, Vol. 44, Issue 3, pp. 311-376.
- [50] **Schuhmacher, A.,** (2010). Techniques for measuring the vibro-acoustic transfer function, *Sound and Vibration*, Vol. 44, no. 3, pp. 6-13.
- [51] **Ewins, D. J.,** (2000). *Modal testing: Theory, practice and applications*, 2nd Edition, Research Studies Press Ltd, Baldock, Hertfordshire, England.
- [52] **Url 1,** < <http://www.diracdelta.co.uk/> >, Volume Velocity, date retrieved 11.12.2014.
- [53] **Url 2,** < http://en.wikipedia.org/wiki/Volumetric_flow_rate >, Volumetric flow rate, date retrieved 11.12. 2014.
- [54] **Van der Linden, P. J. G., and Varet, Ph.,** (1995). Experimental determination of low frequency noise contributions of interior vehicle body panels in normal operation, *SAE-paper 960194*, Detroit, Michigan, USA, February 26-29, pp. 61-66.
- [55] **Dekking, F., Kraaikamp, C., Lopuhaä, H., and Meester, L.,** (2004). *Probability and statistics for the 21st century*, Delft University of Technology, Delft, The Netherlands.
- [56] **Janssens, K., Gajdatsy, P., and Van der Auweraer, H.,** (2008). Operational transfer path analysis: a critical review, *Proceedings of the International Conference on Noise & Vibration Engineering (ISMA)*, Katholieke Universiteit Leuven, Leuven, BE, pp. 3657-3672.
- [57] **Magrans, F.,** (2009). Path analysis, *Proceedings of the NAG/DAGA Conference*, pp. 768-771.
- [58] **Brick, H.,** (2009). Application of the boundary element method to combustion noise and half-space problems, *Ph.D. Thesis*, Chalmers University of Technology, Gothenburg, Sweden
- [59] **Press, W. H., Teukolsky, S. A., and Vetterling, W. T.,** (2007). *Numerical recipes: The art of scientific computing*, 3rd Edition, Cambridge University Press, New York, USA.
- [60] **Golub, G., and Kahan, W.,** (1965). Calculating the singular values and pseudo-inverse of a matrix, *Journal of the Society for Industrial and Applied Mathematics, Series B: Numerical Analysis*, Vol. 2, no. 2, pp. 205-224.
- [61] **Noumura, K., and Yoshida, J.,** (2006). Method of transfer path analysis for interior vehicle sound by actual measurement, *Japan Society of Automotive Engineers, Proceedings of the Annual Congress*, Vol. 55, no. 6, pp. 7-12.

- [62] **De Klerk, D.**, (2009). Determining the significant number of singular values in experimental dynamic substructuring, *Proceedings of the 27th International Modal Analysis Conference, Society for Experimental Mechanics*, Orlando, FL, USA.
- [63] **Devijver, P. A., Kittleri J.**, (1982). *Pattern recognition: A statistical approach*, Prentice-Hall, London, England.
- [64] **Kohavi, R.** (1995). A study of cross-validation and bootstrap for accuracy estimation and model selection, *International Joint Conference on Artificial Intelligence (IJCAI)*, Vol. 14, no. 2, pp. 1137-1145.
- [65] **Müller-BBM Vibro-Acoustik Systeme**, (2012). PAK v5.6 computer software, Müller-BBM VAS.
- [66] **Jackson, J. E.** (1991). *A user's guide to principal components*, John Wiley & Sons Inc., New York, USA.
- [67] **Jolliffe, I. T.**, (2002). *Principal component analysis*, 2nd Edition, Springer-Verlag, New York, USA.
- [68] **Gajdatsy, P., Janssens, K., Gielen, L., Mas, P., and Van der Auweraer, H.**, (2008). Critical assessment of operational transfer path analysis: effect of coupling between path inputs, *Proceedings of Acoustics '08*, Paris, France, June 29-July 4, pp. 5821-5826.
- [69] **Gajdatsy, P., Janssens, K., Gielen, L., Mas, P., and Van der Auweraer, H.**, (2008). Critical assessment of operational transfer path analysis: mathematical problems of transmissibility estimation, *Proceedings of Acoustics '08*, Paris, France, June 29-July 4, pp. 5463-5468.
- [70] **De Klerk, D., and Ossipov, A.**, (2010). Operational transfer path analysis: Theory, guidelines and tire noise application, *Proceedings of the ISMA2010 including USD2010*, pp. 3961-3976.
- [71] **Putner, J., Fastl, H., Lohrmann, M., Kaltenhauser, A., and Ullrich, F.**, (2012). Operational transfer path analysis predicting contributions to the vehicle interior noise for different excitations from the same sound source, *Proceedings of the Internoise 2012*, New York City, NY, USA, August 19-22.
- [72] **NX I-DEAS**, (2006). Version 11, Structural Dynamics Research Corporation.
- [73] **MD ADAMS R3**, (2008). MSC.Software Corporation.
- [74] **Napolitano, K., and Linehan, D.**, (2009). Multiple sine sweep excitation for ground vibration tests, *Proceedings of the IMAC-XXVII, Society for Experimental Mechanics*, Orlando, FL, USA, February 9-12.
- [75] **Pauwels, S., Michel J., Robjins, M., Peeters B., and DeBille, J.**, (2006). A new MIMO sine testing technique for accelerated, high quality FRF measurements, *Proceedings of the IMAC-XXIV, Conference & Exposition on Structural Dynamics*, St Louis, Missouri, USA, January 30-February 2.
- [76] **LMS International**, (2005). *LMS Test.Lab MIMO Sine Testing*, < http://www.plm.automation.siemens.com/en_us/products/lms/ >, date retrieved 20.11.2014.

- [77] Heylen, W, Lammens, S, and Sas, P., (1998), *Modal analysis theory and testing*, Katholieke Universiteit Leuven, Leuven, Belgium.
- [78] MATLAB 8.2, (2013). Version 1.7.0_11 (R2013b), MathWorks.
- [79] Stanford Research Systems, (n.d). *About FFT Spectrum Analyzers – Application Note #1* < <http://www.thinksrs.com/downloads/> > date retrieved 15.11.2014.
- [80] Gatti, P. L., and Ferrari, V., (2003). *Applied structural and mechanical vibrations: theory, methods and measuring instrumentation*, Taylor & Francis e-Library.
- [81] Cakar, O., and Sanliturk, K. Y., (2005). Elimination of suspension effects from measured frequency response functions, *Trends in the Development of Machinery and Associated Technology, IX. International Research/Expert Conference, TMT 2005*, Antalya, Turkey, September 26-30.
- [82] Uslu, A. A., (2007). Force estimation using vibration data, *M.Sc. Thesis*, Istanbul Technical University, Istanbul, Turkey. (in Turkish)
- [83] URL 3, < <http://www.bksv.com/Products> >, Brüel & Kjaer Piezoelectric Charge Accelerometer - Type 4393, date retrieved 19.11.2014.
- [84] URL 4, < <http://www.bksv.com/Products> >, Brüel & Kjaer NEXUS Charge Amplifier - Type 2692-A, date retrieved 19.11.2014.
- [85] URL 5, < <http://www.bksv.com/Products> >, Brüel & Kjaer Measurement Exciter - Type 4809, date retrieved 19.11.2014.
- [86] URL 6, < <http://www.bksv.com/Products> >, Brüel & Kjaer Measurement Exciter - Type 4810, date retrieved 19.11.2014.
- [87] URL 7, < <http://www.bksv.com/Products> >, Brüel & Kjaer 8230-002-DeltaTron Force Transducer, date retrieved 19.11.2014.
- [88] Kazanci, M. V., (2013). Active vibration control of a cantilever beam, *M.Sc. Thesis*, Istanbul Technical University, Istanbul, Turkey.
- [89] URL 8, < <http://www.bksv.com/Products> >, Brüel & Kjaer Signal Analyzer Unit - Type 3560C, date retrieved 19.11.2014.
- [90] URL 9, < http://www.pearl-hifi.com/07_Facility_Tour/B_K_2035.pdf >, Brüel & Kjaer signal analyzer unit Type 2035, date retrieved 19.11.2014.
- [91] URL 10, < https://www.atecorp.com/ATECorp/media/pdfs/data-sheets/Brueel-Kajaer-2706_Datasheet.pdf >, Product data: Power Amplifier – Type 2706, date retrieved 20.11.2014.
- [92] URL 11, < <http://www.bksv.com/Products/pulse-analyzer/pulse-platform/pulse-labshop.aspx> >, PULSE LabShop, date retrieved 20.11.2014.
- [93] PULSE LabShop V.12.5 Help.
- [94] Salim, M. A., Noordin, A., Ismail, A. N., Hassan, M. F., Shafizal, M. A. T., Mansor, M. R., and Rosli, M. A. M., (2010). Fast fourier transform and operational deflection shape analysis for vibration on handheld tools. *Canadian Journal on Mechanical Sciences & Engineering*, Vol. 1, no. 1, pp. 1-11.

- [95] Kozaczka, E., Domagalski, J., and Gloza, I., (2010). Investigation of the underwater noise produced by ships by means of intensity method, *Polish Maritime Research*, Vol. 17, no. 3, pp. 26-36.
- [96] Patent: EP 2594675, Washing machine having vibration reducing apparatus and vibration reducing method using the same, < <http://www.google.com/patents/EP2594675A1?cl=en> >, date retrieved 24.11.2014.
- [97] Varol, İ., (2008). Determination of dynamic properties of viscoelastic members using vibration tests, *M.Sc. Thesis*, Istanbul Technical University, Istanbul, Turkey. (in Turkish)
- [98] Spelta, C., Savaresi, S., Fraternali, G., and Gaudiano, N., (2008). Vibration reduction in a washing machine via damping control, *Proceedings of the 17th World Congress of International Federation of Automatic Control (IFAC)*, July, Seoul, Korea, pp. 11835-11840.
- [99] McDonald, K. T., (1998). Physics in the laundromat, *American Journal of Physics*, Vol. 66, Issue 3, pp. 209-211.
- [100] Lim, H. T., Jeong, W. B., and Kim, K. J., (2010). Dynamic modeling and analysis of drum-type washing machine, *International Journal of Precision Engineering and Manufacturing*, Vol. 11, No. 3, pp. 407-417.
- [101] URL 12, < <http://www.bksv.com/Products/> >, Brüel & Kjaer Triaxial Piezoelectric Charge Accelerometer - Type 4321, date retrieved 26.11.2014.
- [102] URL 13, < <http://www.bksv.com/Products/analysis-software/data-acquisition/data-recording/time-data-recording-7708?tab=overview> >, Time Data Recording - Type 7708, date retrieved 26.11.2014.
- [103] Çelikel, S., (2010). Detection and identification of imbalance in washing machines, *M. Sc. Thesis*, Istanbul Technical University, Istanbul, Turkey. (in Turkish)
- [104] Zhao, L. D., Fouladi, M. H., Haw, and Y. K., (2014). Model correlation procedures for air-conditioning copper discharge tube, *Journal of Engineering Science and Technology*, EURECA 2013 Special Issue August, pp. 40-58.
- [105] ICATS, MESHGEN Users Guide, (2008). Imperial College Testing Analysis and Software, Imperial College, London.
- [106] ICATS, MODENT Users Guide, (2008). Imperial College Testing Analysis and Software, Imperial College, London.
- [107] ICATS, MODESH Users Guide, (2008). Imperial College Testing Analysis and Software, Imperial College, London.
- [108] ICATS, MODPLAN Users Guide, (2008). Imperial College Testing Analysis and Software, Imperial College, London.

- [109] **Mas, P., Sas, P., Wyckaert, K.,** (1994). Indirect force identification based upon impedance matrix inversion: A study on statistical and deterministical accuracy, *Proceedings of ISMA-19: Tools for Noise and Vibration Analysis*, Leuven, Belgium, September 12-14, Volume III, pp. 1049-1065.

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