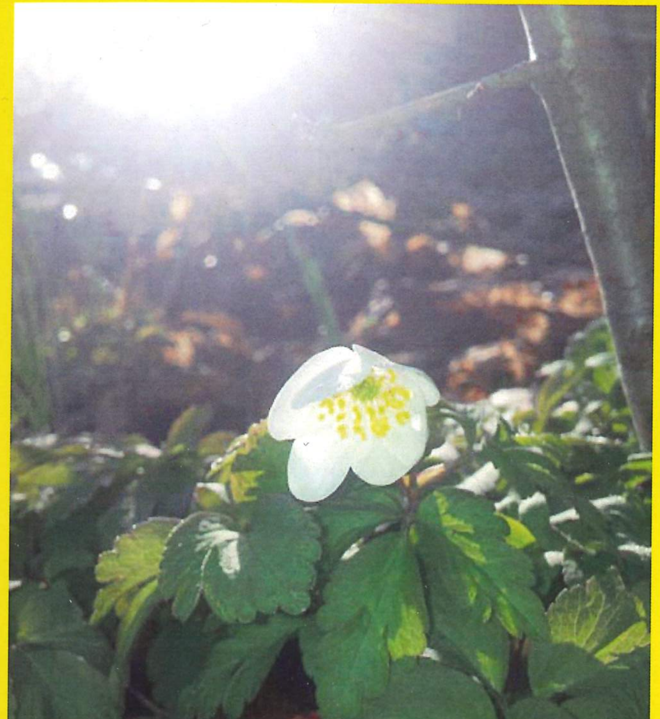
SVIB
VIBRATIONS
NYTT



Medlemsblad för Skandinaviska Vibrationsföreningen

Årg 31 Nr 1 2013
ISSN 0281-6830

Innehållsförteckning

Annonskostnad för 2 nr (exkl moms)	
Dubbeluppslag	10000 SEK
Baksida	7000 SEK
Hel insida	5000 SEK
Halvsida	3000 SEK
Medelmsavgift:	550 SEK
Pensionärer	280 SEK

Foto framsida AnnaLotta Skarp

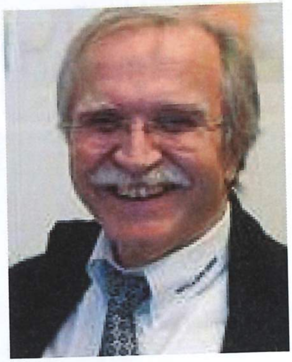
Ordföranden har ordet

Som jag nämnde i förra numret har vi en rejält förnyad styrelse och vi har dessutom fått formellt OK för en revidering av stadgarna. Ni hittar protokollen från både årsmötet och det extra möte vi hade i januari tillsammans med de reviderade stadgarna i detta nummer. Den viktigaste förändringen är att verksamhetsåret nu blir jämnt kalenderår. Förutom att medlemmarna får några månader extra för medlemsavgiften för 2013, som alltså gäller från september 2012 t o m december 2013, blir även bokförings och deklarationsrutinerna enklare. Dessutom innehåller de nya stadgarna även möjligheten till företagsmedlemskap. Mer om också detta längre fram.

I skrivande stund har äntligen lite mer vårvärme kommit. Det har varit en högst märklig vår i ett regnhål som Göteborg med en hel del soliga dagar men ständiga frostnätter och bara ett par mm nederbörd under hela mars!

Vi håller på få ordning på föreningens medlemsregister och skall lansera en ny hemsida kopplat till denna under året. Medlemmarna själva skall kunna gå in och uppdatera sina personliga uppgifter, anmäla sig till våra arrangemang mm. Vi skall också göra tre fullmatade SVIB-nytt under året (april, september, december). Dessutom behöver vi aktivt gå ut och värva nya medlemmar, såväl individuella som företag.

Vi har börjat planera för nästa vibrationsdag och årsmöte som blir i Danmark på våren 2014. Vi har börjat också planera för ett specialsymposium/kurs med tema mätteknik/signalanlys under hösten i år.



Som ansvarig för Svib-nytt avslutar jag med att gnälla lite. Hur intressant innehållet i vårt medlemsblad blir hänger på alla er som medlemmar. Jag får i stort sett inga bidrag från andra än personer i styrelsen eller dem som är nära styrelsen och så har det varit under lång tid. Jag har till och med fått någon gliring för det. Jag hoppas ändå att det nummer ni håller i handen är läsvärd, men den kunde varit ännu bättre om ni alla funderade på med vad ni kunde bidra. Jag tror att bredden på artiklar, notiser och inlägg skulle må bra av det.

Det har också varit svårt att få våra annonsörer att skriva ihop ett par sidors presentation av sina företag. Det vore trevligt att få veta mer om dem som ligger bakom annonserna. Jag kommer att kontakta alla annonsörer och påminna dem om den möjligheten via mail.

En skön Valborg och sommar önskar

Juha

Information om företagsmedlemskap

SVIB erbjuder nu företag som har flera individuella medlemmar i SIB att hantera dessas medlemskap enklare. Vi har erfarenhet av att det ofta är det administrativa krånglet, t ex med ett antal individuella fakturor, som får medlemmar att tröttna på att få sina medlemskap förnyade.

Det är inte medlemsavgiften för sig som är ett hinder.

Genom en stadgeändring har SVIB nu möjlighet att även ha företag som medlemmar. Den främsta fördelen för företagen (och även SVIB:s administration) är främst att en enda faktura administreras årligen, även om antalet individer i företaget som ansluts kan vara stort.

Andra fördelar är

- Rabatter på deltagande i SVIB-arrangemang
- Rabatt som utställare vid SVIB-arrangemang
- Rabatt på annonsering i SVB-nytt
- Plats på sponsorsida i SVIB-nytt och på SVIB:s hemsida
- Gratis plats- och kursannonsering på SVIB:s hemsida

Olika nivåer för företagsmedlemskap

Olika företag har naturligtvis olika antal medarbetare med intresse för SVIB:s verksamhet. Vi inför därför tre olika nivåer av företagsmedlemskap, kalla det "brons", "silver" och "guld" eller "bas", "premium" och "VIP" nivå.

Det nya medlemskapet bör dessutom ge företag incitament att ansluta fler medarbetare som får del av SVIB:s information och aktiviteter. Sammanfattningsvis gäller följande för de tre nivåerna:

Paket nivå	*Medlems antal*	*Annons rabatt*	*Deltagar-rabatt**	*Utställar-rabatt*	*Sponsor-sida*	*Pris/år (2013)*
Brons	6	10%	10%	10%	Namn+logo	3000
Silver	10	20%	20%	20%	Namn+logo+ text (1/4 sida)	6000
Guld	20	25%	25%	25%	Namn+ Logo+ text (1/2 sida)	12000

*Deltagarrabatten gäller från den andra deltagaren, första från företaget betalar fullt pris.

För samtliga nivåer gäller:

- Avgiftsfri plats- och kursannonsering på SVIB hemsida
- Företaget anmäler individuella medarbetare via en utsedd kontaktperson mot SVIB.

Denne är också fakturamottagare.

Styrelsen Informerar

Protokol
Årsmöte i Skandinaviska Vibrationsföreningen, SVIB
Onsdagen den 14. november 2012 på Aeroseum i Göteborg.

a) Fastställande av dagordning
Dagsorden belv godkendt af de fremmødte.

b) Val av mötesordförande och mötessekreterare
Elisabeth Blom blev valgt til mødeleder, Olsson. Anders Olsen blev valgt til mødesekretær.

c) Val av justeringsmän för dagens protokoll
Kjell Ahlin og Anders Brandt er valgt som justeringsmän

d) Fråga om stämman blivit behörigen kallad
Enighed om at der er indkaldt senest 10 dage før på hjemmesiden. Godkendt af de fremmødte.

e) Avgående styrelsens redogörelse för verksamheten under det föregående
Elisabeth Blom gennemgår firmaberetning/virksomhedsberetning, der godkendes af de fremmødte

f) Revisionsberättelse
Elisabeth Blom gennemgår revisionsberetning. Redovisning er i orden iht, alle regler.

g) Fastställande av resultat- och balansräkningar

Torbjörn Kloow gennemgår balance og resultat rapport. Årsresultat på + 24 tkr. Og en egenkapital på +185 tkr. Pr. november har foreningen 325 medlemmer,hvoraf de 20 er fra mødet i Oslo i 2011.

Resultat og balance godkendes af de fremmødte.

h) Beslut rörande ansvarsfrihet för den avgående styrelsen

Forsamlingen godkender ansvarsfrihed for den afgående bestyrelse.

i) Fastställande av budget för kommande verksamhetsår

Torbjörn Kloow gennemgår forslag til nyt budget for 201-2013. Gevinst/tab = 0 kr. TK foreslår, at det nye styre bruger ans forslag som arbejdsbudget. Nyt styremøde skal afholdes snarrest.

Forslag godkendes af de fremmødte.

j) Fastställande av föreningens årsavgift för medlemmar och pensionärer

Årsafgift på SEK 550 fastholdes. For pensionærer fastholdes SEK 280.

Juha Plunt præsenterer forslag om firmamedlemskab. Forslaget sendes til videre behandling i det nye styre, og det besluttes at styret arbejder videre.

k) Val av ordförande för tiden t o m nästkommande årsmöte

Juha Plunt modtager valg

l) Val av vice ordförande för tiden t o m nästkommande årsmöte

Elisabeth Blom modtager valg til vice ordførende, og bytter plads med Juha Plunt.

m) Val av sekreterare

Anders Olsen modtager valg som SVIB sekretær

n) Val av skattmästare

Det nye styre skal vælge en ny skattemester, da TK fratræder.

o) Val av övriga styrelsemedlemmar

Jessica Fronell modtager valg, WSP.
Anders Brandt modtager valg, SDU.
Åsa Collet modtager valg, ÅF.

p) Val av två revisorer och revisorssuppleant

Ralf Collin utsågs till revisor.Revisor nr 2 er vacant. Örjan Johansson er Revisorsuppleant.

q) Tillsättande av en valberedning

TK, örjan Johansson og Søren R K Nielsen modtager valg.

r) Övriga ärenden

§3.3 ændres til....

§4 ændring af virksomhedsår

§5.1 Udskottsordførende og AU erstattes af projektgrupper

§7 Ændring i 1 og 2

§8 1/9-31/8 ændres til kalenderår. Før 1 oktober ændres til: Senest 3 uger før årsmødet.

§9 Ved omröstning skal skriftlig votum ændres til : Medlem kan röste via skriftlig fuldmagt.

Ændringerne som styret foreslår godkendes af de fremmødte.

Elisabeth Blom orienterer om hjemmesiden. Vi håber at ny webside kommer op. PCB kommer at være ansvanlig for siden.

Medlems register er vigtig, og skal opdateres.

På den nye hjemmeside, skal det være muligt for medlemmer selv at opdatere oplysninger.

TK står fortsat for bogføring i SVIB.

Mötessekreterare: *Anders Olsen*
Justeringsmän: *Anders Brandt*
Kjell Ahlin

Protokol

Extraordinært möte i Skandinaviska Vibrationsföreningen, SVIB Onsdagen den 23. januar 2013 hos Müller-BBM Göteborg.

a) Fastställande av dagordning

Dagsorden belv godkendt af de fremmødte.

b) Val av mötesordförande och mötessekreterare

Juha Plunt blev valgt til mødeleder.
Anders Olsen blev valgt til mødesekretær.

c) Val av justeringsmän för dagens protokoll

Jessica Fromell og Åsa Collet er valgt

d) Fråga om mötet er behörigen kallad

Ale de fremmødte samt fuldmagter godkender kallad

e) Slutlig beslut om stagdeändringer enligt bifogat forslag

Alle stagdeändringer godkendes og er 100 % enstemigt vedtaget af fuldmagter og de fremmødte.

Der er modtaget fuldmagter fra følgende:

Claes Fredö: Ja
Johan Granlund: Ja
Jan Sjölin: Ja
Johan Larsson: Ja
Anders Brandt: Ja
Kim Albert Schmidt: Ja

Mötessekreterare: *Anders Olsen*
Justeringsmän: *Jessica Fromell*
Åsa Collet

Stadgar

Stadgar för Skandinaviska Vibrationsföreningen Fastställda 1983-03-22. Reviderade 1985-11-26, 1996-11-11, 1997-11-25 och 2013-01-23.

§1 **Namn och hemort** Föreningens namn är Skandinaviska Vibrationsföreningen (SVIB) och dess hemort Stockholm.

§2 Ändamål Föreningen skall verka inom vibrationsområdet för kunskapsutveckling och informationsspridning. Föreningen skall anordna seminarier och årlig Vibrationsdag. Vidare skall föreningen i medlemsblad informera sina medlemmar om bl a normarbeten, forskning och utveckling samt utbildning.

§3 Medlemskap

1. Varje enskild person som har intresse för vibrationsfrågor kan upptas som medlem i föreningen. Inval beslutas av styrelsen.
2. Till hedersledamot kan styrelsen kalla person som på ett utomordentligt förtjänstfullt sätt främjat föreningens syften. Hedersledamot är befriad från medlemsavgift.
3. Styrelsen kan bevilja företagsmedlemsskap.
4. Styrelsen kan på saklig grund genom enhälligt beslut utesluta medlem ur föreningen.

§4 Sammanträden

1. Allmänt sammanträde hålls efter beslut av styrelsen eller efter framställning hos styrelsen från minst 1/3 av medlemmarna. Årsmöte hålls i anslutning till årlig Vibrationsdag senast 6 månader efter kalenderårets slut.

Vid årsmötet skall följande ärenden behandlas:

a) Fastställande av dagordning
b) Val av mötesordförande och mötessekreterare
c) Val av justeringsmän för dagens protokoll
d) Fråga om stämman blivit behörigen kallad
e) Avgående styrelsens redogörelse för verksamheten under det föregående verksamhetsåret
f) Revisionsberättelse
g) Fastställande av resultat- och balansräkningar
h) Beslut rörande ansvarsfrihet för den avgående styrelsen
i) Fastställande av budget för kommande verksamhetsår
j) Fastställande av föreningens årsavgift k) Val av ordförande för tiden t o m nästkommande årsmöte
l) Val av vice ordförande för tiden t o m nästkommande årsmöte
m) Val av sekreterare
n) Val av skattmästare
o) Val av övriga styrelsemedlemmar
p) Val av två revisorer och revisorssuppleant
q) Tillsättande av en valberedning
r) Övriga ärenden

2. Kallelse med föredragningslista skall publiceras högst sex och minst två veckor före allmänt sammanträde eller årsmöte.
3. Vid omröstning och beslut i frågor som ej rör ändring av föreningens stadgar eller föreningens upplösande gäller enkel majoritet.

Vid lika röstetal för och emot ett förslag äger ordföranden vid sammanträdet utslagsröst.

§5 Styrelse och revisorer

1. Styrelsen utses vid årsmöte. Den skall bestå av ordförande, vice ordförande, sekreterare, skattmästare och tre (2008:11) övriga ledamöter.
2. Styrelsen är beslutsmässig när minst tre ordinarie ledamöter är närvarande, varav en är ordförande eller vice ordförande.
3. Successiv förnyelse av styrelsen bör eftersträvas. Ordförande får ej väljas på längre sammanhängande period än 5 år. Övriga styrelseledamöter bör inte omväljas mer än fem gånger.
4. Val av ordförande, vice ordförande, sekreterare, skattmästare och övriga styrelseledamöter samt revisorer och revisorssuppleant förbereds av en valberedning om tre personer som utses vid årsmöte.
5. Föreningens revisorer, till antalet två, jämte en suppleant väljs vid årsmöte.
6. Styrelsen skall utarbeta en arbetsordning för föreningens verksamhet. Denna skall fastställas på årsmöte. Varje medlem skall vid inträde i föreningen erhålla ett exemplar av arbetsordningen.
7. Föreningen tecknas av ordförande ensam eller vice ordförande i förening med ytterligare en ordinarie ledamot.

§6 Arbetsutskott

Ett antal fasta arbetsutskott förutses, med uppgift att svara för seminarieverksamhet och andra återkommande uppgifter. Styrelsen beslutar om tillsättande av arbetsutskott och väljer utskottsordförande. Utskottsledamöter utses av styrelsen på förslag av utskottsordförande.

§7 Avgifter

1. Årsavgifterna fastställs vid årsmöte.

§8 Räkenskaper och revision

1. Föreningens räkenskapsår utgör kalenderår. Efter räkenskapsårets utgång skall styrelsens berättelse jämte föreningens räkenskaper med verifikationer överlämnas till revisorerna senast 3 veckor före årsmötet.
2. Revisionsberättelsen skall föredras vid årsmöte.

§9 Ändring av stadgarna

För ändring av stadgarna erfordras beslut därom med minst 3/4 majoritet av de röstande vid två på varandra följande allmänna sammanträden. Medlem kan rösta via skriftlig fullmakt.

§10 Föreningens upplösande

1. För föreningens upplösande erfordras beslut därom med minst 3/4 majoritet av de röstande vid två på varandra följande årsmöten. Förslag härom skall tillställas styrelsen före årsmötets utlysande. Vid omröstningen skall skriftligt votum från icke närvarande medlemmar medräknas.
2. Vid eventuellt upplösande överlämnas föreningens tillgångar till en föreningen närstående sammanslutning. Beslut härom fattas av det andra av de i moment 1 nämnda årsmötena.

Insändare

Hej Juha!

Jag känner mig tyvärr tvingad att protestera mot hur mitt bidrag om aktiv vibrationsdämpning refereras (se bilaga).

Du skriver bl. a. följande:

- *"Han visade ett par riktigt underhållande videoklipp som illustrerade hur hoppandet gick till och som visade svängningarna hos arenans ståplatsläktare för hemmasupporters".*
  Faktum är att jag visade ett klipp där man ser hur publiken hoppar på Gamla Ullevi. Jag framförde också att Gamla Ullevis läktarkonstruktion är styv med egenfrekvens väl över hoppfrekvensen. Jag visade även ett klipp från en tyska arena där ståplatsläktaren syntes vibrera kraftigt under hoppkrafterna från publiken, som ett varnande exempel på hur det hade kunnat se ut om Gamla Ullevi haft en sämre konstruktion. Jag har mycket svårt att tro att någon annan än Du har kunnat uppfatta det som att detta klipp skulle visa Gamla Ullevis läktare!
- *"Ett system av aktiva vibrationsdämpare har installerats och vibrationer uppmätta i stört hus ser ut att kunna ha minskat. Dock är spridningen i uppmätt vibrationsamplituder stor både före och efter att systemet tagits i bruk".*
  Jag tycker att Du med dessa formuleringar fabricerar ett ifrågasättande av systemets effektivitet. Jag framförde inget annat än att den aktiva vibrationsdämpningen efter en inkörningstid är effektiv, men att ett visst slutligt intrimningsarbete pågår. Vi ser ingen "spridning" relativt vad vi förväntar oss med hänsyn till publikantal och vilka lag som spelar, och din beskrivning stämmer inte med min presentation.

Jag tycker att SVIB-medlemmarna kan förvänta sig ett sakligt referat av anförandena under vibrationsdagen.

m.v.h.
Gunnar Widén

PS: För den som är intresserad finns en artikel om systemet i Bygg & Teknik nr 1 2013.

Measurement and Analysis of Valvetrain Dynamics and Dual-Mass Flywheel Characteristics

Seán Adamson, Vispiron Rotec GmbH, Munich, Germany sean.adamson@vispiron.de

Abstract

Over the last decade the automotive industry has achieved significant progress in improving both fuel economy and driving comfort. In order to make accurate assessments of engine and transmission capabilities state-of-the-art testing equipment in the development phases needs to be used. This paper presents experimental investigations involving two areas of current interest: valvetrains and dual-mass flywheels. Instrumentation used, test configurations, test parameters and analysis software are discussed.

Higher valvetrain efficiency is a key factor in reducing fuel consumption and emissions. Methods for measuring valve motion are outlined. The increasing complexity of valvetrain systems demands application-specific test and analysis software. Valve lift, valve velocity and valve acceleration curves versus camshaft speed and angle are presented. Measured data are compared with kinematic curves and the dynamics of valve opening, closing and seating are discussed.

The dual mass flywheel or DMF is increasingly used in vehicles fitted with manual transmissions in order to optimise driving comfort by reducing noise and vibration in the driveline. The DMF eliminates excessive gearbox rattle and reduces gear change/shift effort. Measurement data are used to describe the pros and cons of single and dual mass flywheel systems. Typical analyses include DMF wind-up angles and the comparison of torsional vibration amplitudes on the engine and transmission sides.

1 Equipment and Methods for Measurement and Analysis

The data acquisition and analysis equipment used in the applications described here is developed, manufactured and marketed by Vispiron ROTEC GmbH [1, 2, 3]. The company's core products are pc-based signal acquisition and analysis systems which are fitted with both digital and analogue input channels. Rotational speed signals are acquired using 10GHz with digital counter channels which record the time intervals between TTL pulses. Analogue channels facilitate conditioning and capture of a variety of signals with sampling rates up to 400kHz. The analysis software, working primarily in the angle domain, provides comprehensive analyses in the time and spectral domain (FFT order and frequency analysis). Near real-time capability with display and analysis of all channels allows for adjustment of test parameters during the measurement. A distinctive feature of ROTEC systems is the phase-matched acquisition of all signals: speed signal acquistion with variable discretisation of time (angle-equidistant sampling) and acquisition of analogue signals – displacement, velocity, acceleration, force, pressure, torque, etc. – at constant time intervals (time-equidistant sampling). The main purpose of this paper is to give an overview of the test and analysis possibilities offered by ROTEC equipment. For further details on sensors, test setups and interpretation of results the reader should refer to the technical reference works cited by the author.

2 Dynamic Valvetrain Testing and Analysis

2.1 Valvetrains in Internal Combustion Engines

The valvetrain controls the gas exchange process, i.e. the intake of fuel mixture and the discharge of exhaust gases [4]. To allow the maximum amount of mixture to be drawn into the cylinders, the inlet valves should open as quickly as possible and remain open for as long as possible followed by rapid closing. There is generally a conflict between fast valve opening and closing, which means large valve accelerations, and the requirement of limiting loads, which means small accelerations. Development tasks include ensuring that valvetrain components satisfy strength and durability requirements and that they operate to a high degree of accuracy.

Development requirements include dynamic valvetrain testing on both running engines and motored test rigs either for speed ramps or steady state speeds. The sensors required must measure the valve lift (and valve velocity if possible). Camshaft speed and angular position are ideally measured using a rotary encoder attached to the camshaft [5]. In addition, quantities such as shaft torques and spring forces are also measured. Comprehensive valvetrain testing requires data acquisition equipment which enables fast, synchronous acquisition of all test data. The testing equipment must enable the behaviour of valve trains of different cylinders in the same engine to be quickly compared and also confirm the repeatability of successive tests for the same valve train. Prerequisites for the equipment include high resolution, sufficiently large memory

(particularly for speed ramp data which are acquired at high sampling rates) and application-specific valvetrain motion software.

2.2 Valvetrain Data Collection

Figure 1 shows the minimum signal requirements for valvetrain testing.

1. Reference pulse for triggering the measurements
2. Camshaft speed signal
3. Valve lift signal

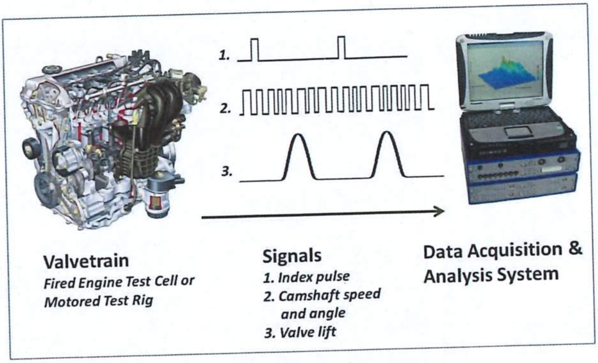


Figure 1. Valvetrain testing

Modern instrumentation and testing equipment enables measurements to be performed on both single and multiple valves. Depending on the type of valvetrain testing performed – component test stand, motored cylinder head test bench, fired engine test cell, etc – the following additional data may be collected and analysed:

- Valve velocity (when using laser vibrometers)
- Torque
- Displacement (e.g. spring displacement)
- Forces (e.g. using strain gauges)
- Torsional vibration
- CAN signals (pressure, temperature, etc).

2.3 Valvetrain Analysis Software

Software features available in the ROTEC valvetrain analysis include:

- Online display of valve lift, valve velocity and valve acceleration versus angular position, time or speed.
- Automatic measurement and analysis of speed sweeps and speed steps.
- 3D plots of lift, velocity and acceleration (z-axis) versus angular position (x-axis) and speed (y-axis).
- Quick calculation of seating velocities, closing angles, lift-off angle, lift loss, curve max. / min. values, etc.
- Comparison of measured and theoretical lift, velocity and acceleration profiles.

2.3.1 Motored Cylinder Head Test Bench

Test benchs powered by electric motors are widely used for validating valvetrain capability and dynamics [6, 7]. In comparison to fired engine test cells, their main advantages are that instrumentation is easy to install and comprehensive measurement and analysis of valvetrain dynamics is possible. The valve motion results presented below were obtained using Polytec's High Speed Laser Vibrometer (HSV) which can measure valve lift and also valve velocity up to 30m/s [8]. Its high bandwidths combined with the high resolution of the ROTEC equipment enhance valve acceleration calculations. Figure 2 shows typical waterfall plots of valve lift, velocity and acceleration from 500 to 3000rpm. The camshaft speed and angular position was measured with a 3600 line rotary encoder. The valve lift and velocity signals were acquired at 400kHz sampling frequency. Valve velocity is calculated in m/rad and acceleration in m/rad^2.

Figures 4 and 5 show two further standard analyses in the ROTEC software:

- 2D plots of valve lift, velocity and acceleration at selected speed values (Figure 4)
- Plots of valve closing velocity and valve closing angle at a specified value of lift in the valve closing range (Figure 5)

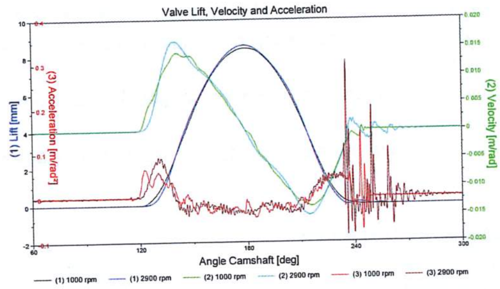


Figure 4: Valve lift, velocity and acceleration at 1000 and 2900rpm

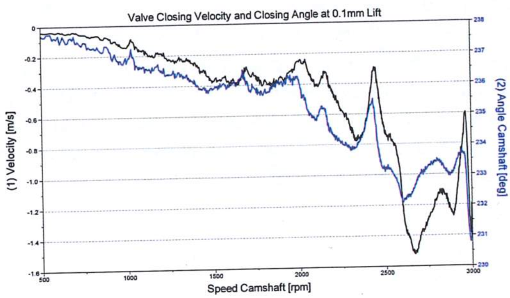


Figure 5: Valve closing velocity and closing angle at 0.1mm lift versus speed

In Figure 3 the valve lift, velocity and acceleration are shown for various camshaft speeds. Before and after maximum valve lift, effects due to valve spring oscillations may be seen in both the velocity and acceleration curves. Acceleration data can be directly related to valve loads which peak before and after valve seating. Over the range of valve seating, an unwanted phenomenon appears – valve bounce. This valve bouncing phenomenon will accelerate seat wear, results in a loss of power and may cause engine damage. In the valve closing range, valve bounce can be observed in the valve lift, valve velocity and acceleration profiles. The peaks which appear in these curves are due to this phenomenon.

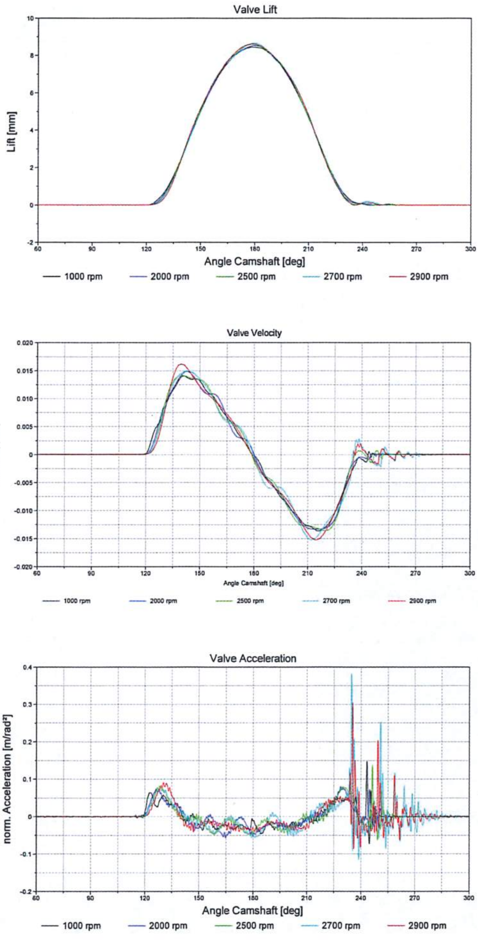


Figure 3: Valve lift, valve velocity and valve acceleration at different speed values

Figures 4 and 5 show two further standard analyses in the ROTEC software:

- 2D plots of valve lift, velocity and acceleration at selected speed values (Figure 4)
- Plots of valve closing velocity and valve closing angle at a specified value of lift in the valve closing range (Figure 5)

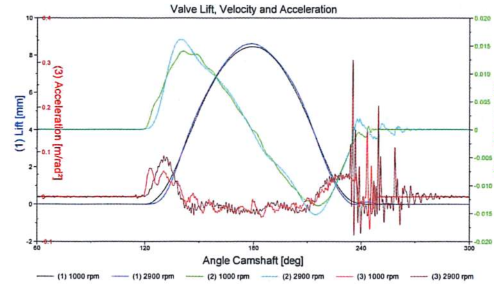


Figure 4: Valve lift, velocity and acceleration at 1000 and 2900rpm

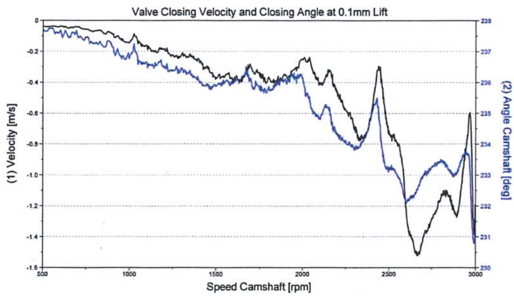
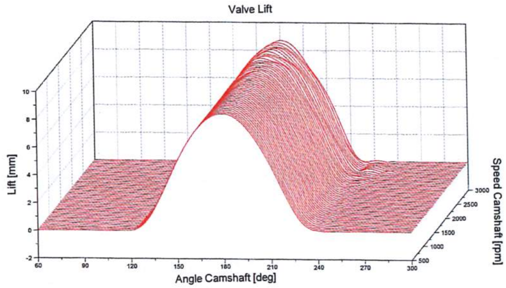


Figure 5: Valve closing velocity and closing angle at 0.1mm lift versus speed

Measured lift curves versus speed may be compared with either 2D theoretical lift curves or reference lift curves measured at low speeds. These comparisons highlight another dynamic valve phenomenon – valve jump at high engine speed (Figure 6). Under normal conditions, the valvetrain components including the valve, tappet, spring retainer etc. are pressed against the cam surface by the valve spring and move up and down while maintaining contact with the cam lobe as the camshaft rotates. However, due to high acceleration the valve can unexpected lift up and lose contact with the cam profile. Besides poor NVH behaviour this can cause wear (pitting) on the cam profile where the valve clashes back onto the cam (closing ramp). In severe cases, heavy lift overshoot can result in valve-piston contact leading to engine damage.



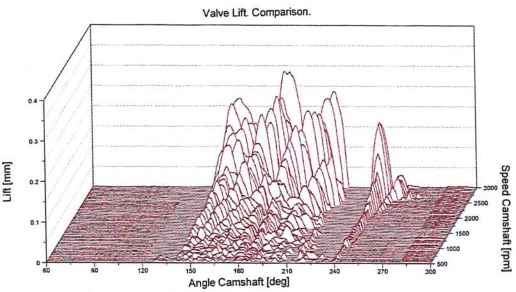


Figure 6: 3D difference lift plot versus speed (this results from subtraction of a 2D single lift cycle measured at low speed

Figure 7 shows a speed ramp performed on another cylinder head test bench powered by an electric motor. This test bench is designed to simulate a 4-cylinder engine. In this case the camshaft torque and valve spring forces were also measured. The data may be used in investigating valve spring dynamics. Frequency analysis of both the spring force and torque curves can reveal spring natural frequencies typically in the range of 400 to 500Hz.

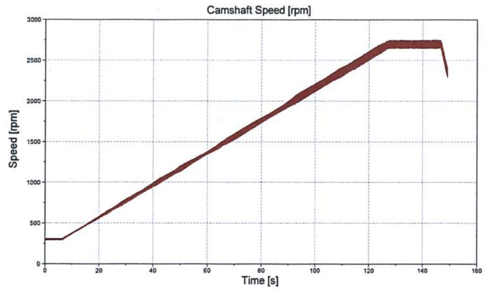


Figure 7: Speed ramp from 500 to 2500rpm camshaft speed

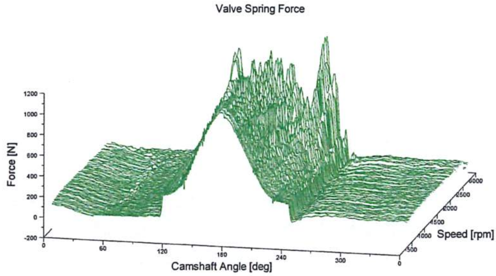


Figure 8: 3D waterfall plot of valve spring force

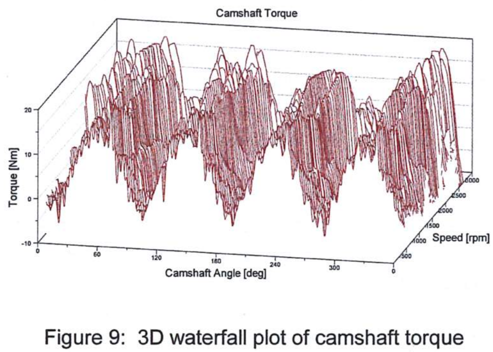


Figure 9: 3D waterfall plot of camshaft torque

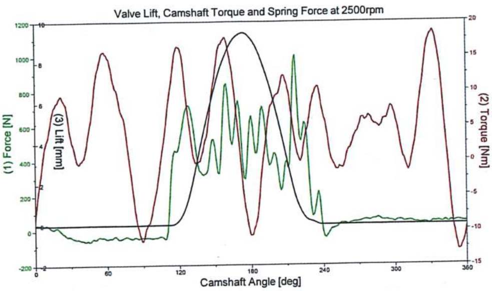


Figure 10: Valve lift, camshaft torque and spring force at 2500rpm camshaft speed

2.3.2 Valvetrain Testing on Fired Engines

Compared to motored test benches, valvetrain testing on running engines reflects real working conditions and, in particular, temperature effects. Valve displacement can be measured using both specially designed transducers and commercially available sensors. Apart from the difficulties encountered in mounting the sensors they are often limited in the range of valve motion which can be measured. In recent years magnetoresistive sensors mounted inside the cylinder in the valve guide have been successfully used for measuring valve lift [7, 9, 10]. Major advantages of these sensors include small installation space requirements and operation at high temperatures. In order to measure valve displacement the sensor requires a (ferromagnetic) tooth structure machined into the valve stem. Movement of the valve, and thus the teeth, produces a change in the sensor's electrical resistance due to the varying magnetic field. The sensor outputs simultaneous sine and cosine signals resulting from the valve movement. These signals are input to ROTEC 400kHz analogue channels. A ROTEC software algorithm generates a valve lift curve from the sine and cosine signals.

The accuracy of this valve lift curve may be determined by performing measurements with both the magnetoresistive sensor and high speed vibrometer in parallel on a motored test bench before the machined valve, sensor assembly, etc. are used on the fired engine. Figure 11 shows the measured magnetoresistive sensor and laser signals while Figure 12 shows the result of the comparison where the valve lift calculated from the magneto-resistive sensor signals has been subtracted from the lift curve measured with the laser. The maximum deviation is of the order of 20 microns which is acceptable for all practical purposes.

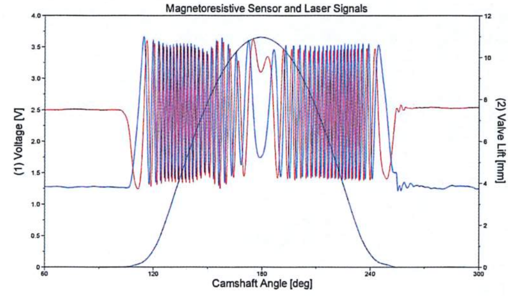


Figure 11: MR sensor sine and cosine voltage outputs and valve lift signal (HSV laser)

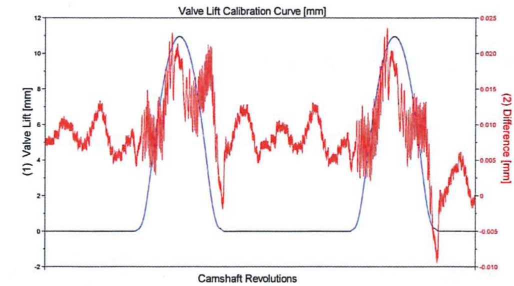


Figure 12: Measured lift difference between HSV laser and magnetoresistive sensor

3 Dual Mass Flywheel Testing

Another frequent application for ROTEC equipment is in-vehicle testing of clutches and dual mass flywheels (DMF). To identify dynamic torsional vibration effects in torque transmission, simultaneous measurement of speed on both sides of the clutch or DMF is required. On the engine side, the starter ring gear may be scanned with a magnetic sensor. On the transmission side, a gear within the gearbox must be accessed or, alternatively, a toothed rim can be attached to the gearbox input shaft. This angular sampling provides a fixed number of data points per revolution which is independent of the rotational speed. The momentary angular velocity of rotating shafts is thus measured, i.e. the mean velocity from pulse to pulse. The vibration angle and the angular acceleration are obtained by integrating and differentiating the measured angular velocity respectively. These two calculations are important when investigating torsional vibration problems. Another important calculation is the angle between the two speed channels (angular displacement, slip).

3.1 Clutch measurements

Figure 13 shows test results from a speed run-up in fourth gear (4-cylinder engine, conventional clutch). Magnetic speed sensors were used on the starter gear (132 teeth) and on a gearwheel on the gearbox input shaft (27 teeth).

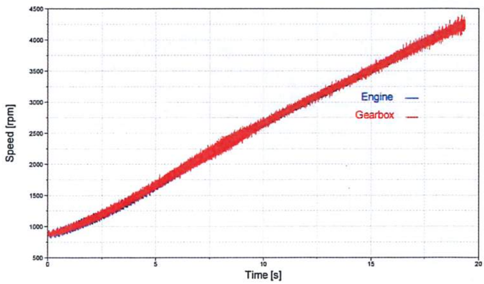


Figure 13: Speed ramp. Time history data on both sides of the clutch

Figure 14 shows two revolutions of the speed curves in detail. The data points from the speed measurements are also shown. The engine's firing order can be clearly seen (2nd order, twice per revolution).

An FFT analysis shows additional orders since cyclical combustion is not a purely sinusoidal process. The 3D waterfall plots (Figure 15) show order analyses of angular acceleration as a function of speed. The gearbox acceleration peaks which occur at resonance-critical speeds contribute, for example, to the unwelcome audible noise known as gearbox rattle. These 2nd order resonance peaks can be reduced by modifying clutch friction and torsional stiffness. Alternatively, a dual-mass flywheel (DMF) may be used. The main purpose of a DMF is to counter gearbox rattle by damping the input torsional vibrations.

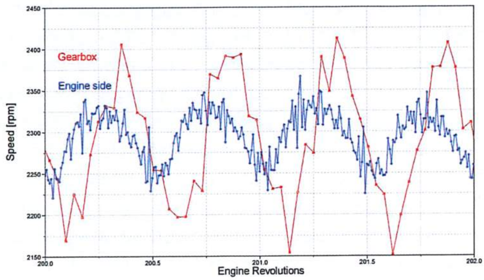


Figure 14: Speed data over two revolutions

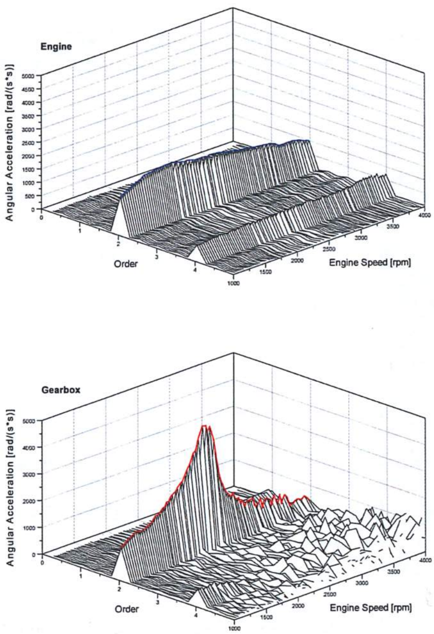


Figure 15: Angular acceleration [rad/s^2], engine and gearbox

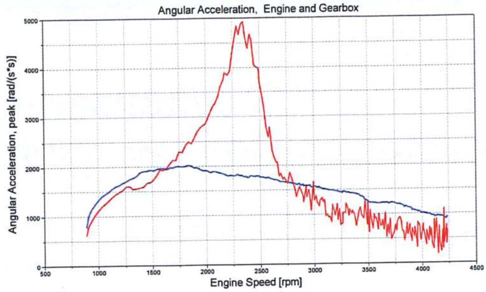


Figure 16: 2nd order angular acceleration [rad/s^2], engine and gearbox

3.2 DMF Measurements

Dual mass flywheels (DMF) are installed in many modern vehicles with manual transmissions. The DMF is located between the engine and the gearbox replacing the conventional flywheel. The DMF isolates the driveline from engine excitation thus increasing driving comfort. The primary inertial mass is connected to the output shaft of the engine. The secondary inertial mass is connected to the input shaft of the gearbox thus increasing the moment of inertia of the transmission. These two de-coupled masses are linked by an elastic spring system which permits relative rotary motion between the primary and secondary masses (Figure 17). The use of the secondary mass and appropriate spring constants have the effect of shifting the resonance speeds which excite the natural rattling frequency below the engine idling speed, i.e. outside the normal driving ranges. The secondary mass also forms one of the two friction surfaces for the clutch disc.

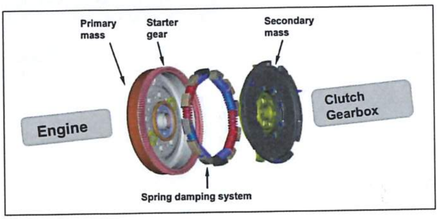


Figure 17: Dual-mass-flywheel, schematic assembly

The main disadvantage of the DMF is that every time the engine is started or stopped it has to run through this resonance point. Problems can then arise if e.g. the vehicle gets stuck in resonance. A further disadvantage of the DMF is an increase in engine-side torsional vibrations due to the DMF's lower effective moment of inertia on the engine side compared to a conventional flywheel. Figure 18 shows time history rotational speed data taken on a diesel engine both with and without a dual-mass-flywheel. The measurements were performed at idle speed (flywheel and gearbox input). The DMF causes an increase in engine-side vibrations (red curve).

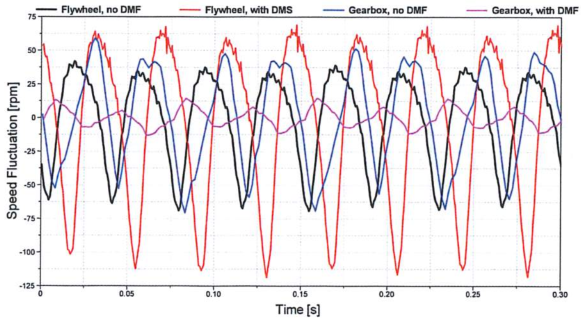


Figure 18: Speed data with and without a DMF

The main disadvantage of shifting transmission resonance below idle speed is that when the vehicle is started it has to run through resonace. Figure 19 shows data from a normal engine start – the engine runs through the resonance point and reaches idle speed. The DMF angular displacement reaches a maximum of about 45 degrees.

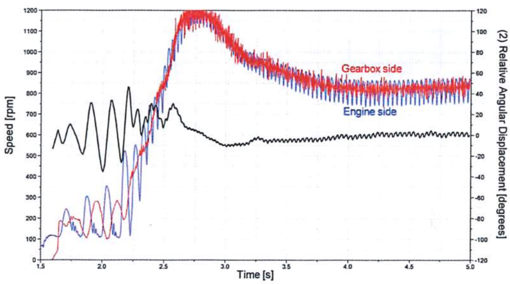


Figure 19: Good engine start behaviour. Time and angular displacement

Figure 20 shows data from a bad engine start. The magnitude of the DMF angular displacement causes the spring components to lock-out. The maximum DMF angular displacement is 60 degrees in both directions. The resulting torque causes the friction clutch disc to slip through and the angular displacement curve runs away. In this case, poor engine management parameters (injection volume, duration and time) in combination with a high DMF spring rate caused the bad engine starting behaviour. The aim of this type of measurement is to optimise the starting conditions and keep the DMF angular displacement below a set value (for this particular DMF below 45 degrees).

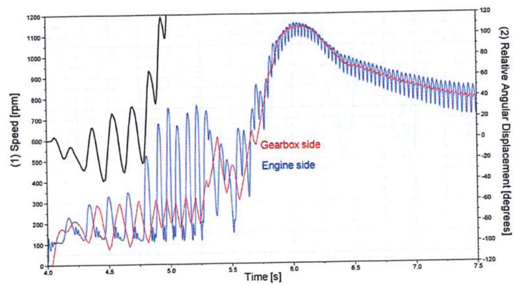


Figure 20: Bad engine-start behaviour. Time and angular displacement curves

Figures 21 and 22 show a case of deliberate vehicle abuse (total misuse duration of around one minute). The vehicle is sharply braked in 4th gear until drive train vibrations arise below idle speed. Figure 21 shows the speed data. ROTEC speed sensors with directional recognition were used (starter gear with 132 teeth, toothed rim with 170 teeth attached to the secondary side). The data show large speed fluctuations when resonance is reached and the direction of rotation reverses for a short time.

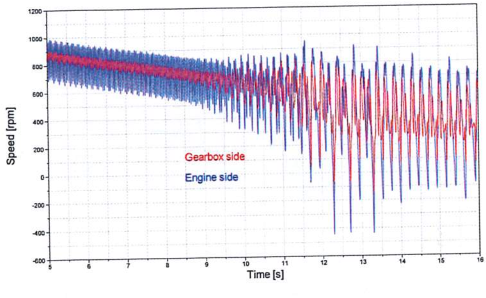


Figure 21: DMF driveline resonance. Time histories

With decreasing engine speed, the excitation in the transmission increases until, at resonance, the primary to secondary mass angle reaches 120 degrees and the maximum permissible engine torque is exceeded (Figure 22). This can damage the DMF or other driveline components. Torque overload protection is provided by a torque limiter in the DMF. When the torque limit is exceeded the DMF slips between the primary and secondary masses. The drivetrain remains in resonance unless the clutch is activated or the vehicle is brought to rest.

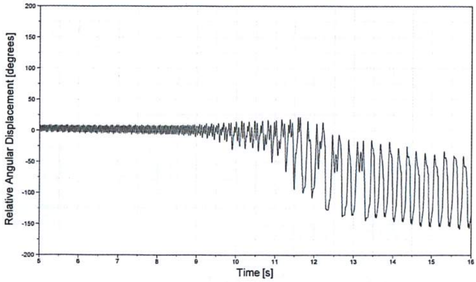


Figure 22: Driveline resonance. DMS relative angular displacement

Acknowledgements

The author wishes to thank Dirk Beismann of Ford-Werke GmbH in Köln for useful discussions on the valvetrain data presented in Section 2 and Ralf Till of ZF AG, Schweinfurt for providing the measurement data presented in Section 3.

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Operational transfer path analysis: Practical considerations for selecting sensor positions

Mihkel Toome
Müller-BBM Scandinavia AB
Marieholmsgatan 9, 405 22 Göteborg, Sweden
Mihkel.Toome@MuellerBBM.se

1. Introduction

Operational transfer path analysis (OTPA) is an alternative to classical transfer path analysis (TPA) as a method used to predict the noise or vibration source/path contributions to the response of a system. Both methods are based on the assumption that there is a linear relationship between input (reference) and response positions; however, while the classical TPA method uses a known input force to compute transfer functions as FRFs, which are then multiplied by a known (calculated or predicted) input force to compute the contributions to the response at the receiver position, the OTPA method uses operational measureable quantities to compute both the transfer functions (in this case transmissibilities) and the response at the receiver.

Although OTPA is currently used predominantly for vehicle noise, vibration and harshness (NVH) assessment, the method is useful for any noise/vibration assessment where a ranking of the source/path contributions at the receiver position(s) is desired, e.g. industrial installations, building HVAC installations, complex machinery and appliances, trains, aircraft, ships, submarines, construction equipment.

The goal of this article is to introduce the underlying theory behind the OTPA method, as well as to highlight some practical considerations for selecting sensor positions and the OTPA post-processing. The practical considerations are highlighted through the description of a case study and by recreating the results of the case study in a simple, virtual OTPA test setup simulation.

2. Theory

Similarly to classical TPA, OTPA is based on a linear relationship between the source(s) and receiver(s), which can be described as:

$$\mathbf{Y}(j\omega) = \mathbf{X}(j\omega) \cdot \mathbf{H}(j\omega) \tag{1}$$

Where $\mathbf{Y}(j\omega)$ is a matrix of the output responses at the receiver measurement positions (MPs) , $\mathbf{X}(j\omega)$ is a matrix of the measured quantities at input reference measurement positions and $\mathbf{H}(j\omega)$ is a matrix of the transfer functions. In this case, the equation is relative to frequency and phase, $j\omega$, meaning that the TPA computation is repeated for each discrete frequency line of the discrete Fourier transform (DFT). OTPA may be conducted similarly in the time domain, however, in the following discussion, all computations are assumed in the frequency domain.

The key to the OTPA computation is in the set up of the matrices: Referring to equation (2), note that the **X** and **Y** matrices are set up such that the columns consist of the measurement positions (MPs), denoted by n for the number of response MPs, and m for the number of reference MPs, and the rows consist of the data for each measurement time-block of data, denoted by *r* for the number of measurement blocks.

$$\begin{bmatrix} y_1^{(1)} & \cdots & y_1^{(n)} \\ \vdots & \ddots & \vdots \\ y_r^{(1)} & \cdots & y_r^{(n)} \end{bmatrix} = \begin{bmatrix} x_1^{(1)} & \cdots & x_1^{(m)} \\ \vdots & \ddots & \vdots \\ x_r^{(1)} & \cdots & x_r^{(m)} \end{bmatrix} \begin{bmatrix} H_{11} & \cdots & H_{n1} \\ \vdots & \ddots & \vdots \\ H_{1m} & \cdots & H_{mn} \end{bmatrix} \tag{2}$$

This relationship is particularly useful for the analysis of, for example, a vehicle run-up. Of course, the equation is used under the assumption that the linear relationship between **X** and **Y** holds true for each measurement block. The number of rows should normally be much larger than the columns.

Further, the reference, **X**, and response, **Y**, matrices may include measurements of different units, for example force, vibration and sound pressure level. However, if this is the case, the data must be normalized to represent the relative contributions at the response position(s).

Prior to computing the transfer function, **H**, the cross-talk between the reference MPs must also be minimized. The cross-talk is the contributions to the measurement at a reference MP from forces acting at other reference points. This is done by singular value decomposition (SVD) of **X**, as shown in equation (3).

$$\mathbf{X} = \mathbf{U}\Sigma\mathbf{V}^T \tag{3}$$

Here, **U** and **V** are eigenvectors corresponding to the matrix of singular values (i.e. eigenvalues) in **Σ**.

The SVD is also used because it is an efficient method to compute the least-squares estimate of the inverse of a matrix, in this case the matrix **X**. The SVD pseudo-inverse of **X** is shown in equation (4).

$$\mathbf{X}^+ = \mathbf{V}\Sigma^{-1}\mathbf{U}^T \tag{4}$$

A principal component analysis (PCA) is then conducted where the lowest ranked principal components (PCs), which constitute measurement noise, are disregarded from the analysis. The PCA is described by equation (5),

$$\Sigma_r^{-1} = \begin{cases} 1/\alpha_n & if \quad \alpha_n > thres \\ 0 & otherwise \end{cases} \tag{5}$$

where *thres* represents the threshold for the minimum size of the PCs to be included in the analysis.

The transfer function **H** is then computed by combining equation (1) with equations (4) and (5). The result is a “noise removed and cross-talk cancelled” estimate of the transfer function matrix, denoted $\mathbf{H}_r$.

$$\mathbf{H}_r = \mathbf{V}\Sigma_r^{-1}\mathbf{U}^T\mathbf{Y} \tag{6}$$

However, the statement "noise removed and cross-talk cancelled" should be taken with a degree of skepticism – many factors come into play which may impede the effectiveness of the cross-talk cancellation method. These are discussed in the following sections of this paper.

In computing the transfer function, it is critical to include reference measurements of several relevant operating conditions in order to ensure that all of the relevant modes are excited and included. Once the transfer function, $\mathbf{H}_r$, has been calculated, measurements are repeated for the operating condition of interest (i.e. new measurement data for matrix **X**), and the response synthesis, $\mathbf{Y}_s$, is calculated.

The resulting response synthesis, $\mathbf{Y}_s$, is a matrix containing the contribution of each reference MP with respect to each block of measurement data. The overall response at a specific point in time is therefore the sum of the contributions from all the reference MPs.

The reliability of the OTPA results is often checked by comparing the actual measured response to the response synthesis. However, although the measured response and response synthesis may match well, the source/path contribution predictions may not be accurate. The potential sources of error for OTPA are discussed further in section 3.

The OTPA theory is described in detail in references [1, 2, 3, 4].

3. Potential sources of error

It is important to keep in mind that the accuracy of the OTPA results depends on the correct placement of the sensors – that is to say it is advantageous to have an understanding of the system response prior to setting up the measurement. Further, it should be kept in mind that cross-talk between highly correlated paths is not identifiable using the SVD; however, a good signal to noise ratio can allow for the contribution to be accurately evaluated, even between strongly correlated paths.

A literature review indicates that there are three main potential sources of error in using the OTPA method [5, 6, 7, 8]:

1. Neglected sources/paths in the measurement setup.
2. Cross-coupling between input measurements.
3. Incorrect estimation of the transfer paths.

These sources of error are discussed in the following.

3.1 Neglected sources/paths in the measurement setup

It was previously mentioned that it is advantageous to have an understanding of the system's response prior to setting up the OTPA measurement. An incorrect setup, such as accidentally neglecting critical sources and paths in the measurement setup will introduce errors, particularly if the neglected paths are highly correlated with those included in the analysis.

The resulting contribution predictions will be skewed to account for the energy of the neglected correlated paths – the contribution of the included paths will appear to be higher than they actually are. Further, the overall response synthesis will match the measured response – which could lead to incorrect conclusions regarding the reliability of the data.

If the neglected sources/paths are uncorrelated or only partly correlated to those included in the analysis, the error in the measurement setup will be evident upon comparison of the overall response synthesis to the actual measured response – they will not match.

3.2 Cross-coupling between input measurements

Difficulties in the cross-talk cancelation (CTC) technique, which is based on a SVD of the matrix of reference MPs, arise when two or more highly correlated paths exist for the system. The SVD cannot separate highly correlated paths, therefore, the results may yield erroneous contribution estimates.

Cross-talk between uncorrelated or even partly correlated paths can however be successfully separated using the CTC technique.

3.3 Incorrect estimation of the transfer paths

Because the OTPA transfer function is based on transmissibilities between the reference and response MPs, if all the significant modes of the system are not included in the measurement, the calculated transfer paths will be incorrect. The potential for this error can be alleviated by including measurements for several relevant operational conditions, e.g. run-up and run-down for several gear positions for powertrain analysis.

Further, the calculation of the transfer function relies on the SVD in order to calculate the pseudo-inverse of the matrix of reference/input measurements. In this case, the pseudo-inverse is an approximation, thus the results may not be completely accurate.

4. Case Study: OTPA of a road tractor

An OTPA study was conducted on a trailer-less road tractor, which initially produced some puzzling results.

The analysis was conducted for an idling (i.e. stationary) condition. Microphones were placed to cover the airborne sources (ABS) and accelerometers were mounted to cover the structure-borne sources and paths (SBS).

The microphone positions are shown in Figure 1, where the indicated MPs are separated into two groups consisting of: six MPs in the vicinity of the cab for the engine, exhaust, intake and transmission, and six additional MPs spread out at the tires, engine cooling fan and rear axle. The response MP was a microphone at the driver position.

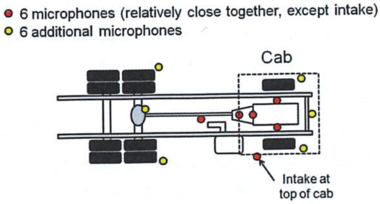

Figure 1: OTPA microphone positions

The source/path contribution analysis results of an engine idle condition produced some strange results in the low frequency, specifically at 25 Hz (dominating 3rd order). The contribution analysis indicates that the ABS sources/paths are the entirely significant contributors to the overall sound level at the response position in the tractor cab, while the SBS paths are insignificant contributors, with the overall SBS contribution approximately 20 dB lower than the ABS contribution, refer to Figure 2a.

A comparison of the overall response synthesis to the actual measured response indicates that the results may be accurate. However, all previous experience suggests that a much higher contribution from the SBS paths would be expected in this low frequency range.

Given that the analysis was conducted at engine idle operating condition, it was determined that the tire, rear axle and engine cooling microphone positions (refer to Figure 1, the MPs denoted as *6 additional microphones*) are not actually measuring any significant ABS source, and may therefore be excluded from the analysis.

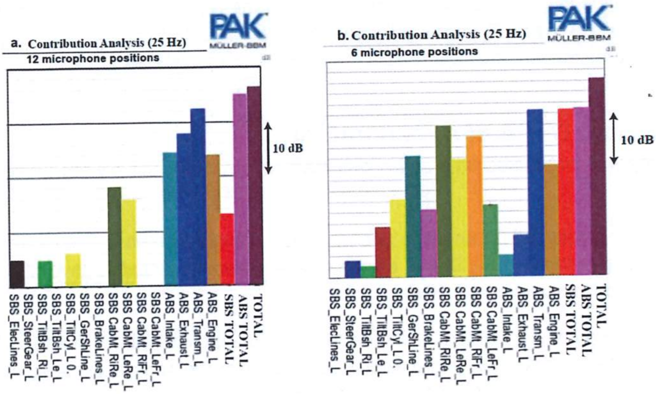

Figure 2: Contribution analysis with a. 12 mic positions included b. 6 mic positions included

The OTPA post-processing was repeated with the *6 additional microphone* MPs excluded from the analysis. Referring to Figure 2b, we see that the resulting contribution analysis indicates that the ABS and SBS contributions seem to be equal.

Comparing Figure 2a with Figure 2b, note that although the SBS and ABS contributions are different, the total computed sound pressure level remains constant for both analysis cases.

The results displayed in Figure 2b were believed to be closer to the correct results – subsequently, modifications to the tractor to reduce both the ABS and the SBS contributions were implemented which lead to a reduction of the overall cab sound level. Thus, it was confirmed that the results of Figure 2b are indeed sufficiently accurate for selection of countermeasures, while the results of Figure 2a are erroneous.

These results indicate that by excluding the six additions ABS MPs the energy is shifted towards the SBS sources/paths in the contribution analysis calculation.

The question that remained was what causes the over/under prediction to occur, and are there any rules of thumb that can be followed to avoid such erroneous OTPA results?

5. OTPA test setup simulation

To study the OTPA method with respect to the results discussed in the road tractor case study, a simplified, numerical simulation of an OTPA study was created. The purpose of the simulation was to investigate the OTPA method in a highly controlled environment where all of the input parameters (i.e. the "measurement" parameters) could be accurately defined. Further, the analysis results obtained should not be influenced by any unknown sources of error.

Several simulated measurement cases were studied in [4]; however only the case that best illustrates what has occurred in the road tractor OTPA case study is discussed in the following.

A simple test setup consisting of a source with one SBS path and one ABS path was created. The ABS source was modeled as a radiating spherical source of a size comparable to that of an engine, in a free field environment, and radiating a single tone at 25 Hz. The reference MP and the ABS response MP were positioned at a similar distance to the microphones for the actual measurement described in section 4.

Further, an additional ABS reference MP was included in the analysis – the distance of the additional MP was varied from $1/16\lambda$ to $1/2\lambda$, where λ represents the wavelength of the radiating tone.

The SBS path was defined to be fully correlated and contributing the same amount as the ABS path at the response position.

In summary, the described OTPA test setup simulation represents the case with highly correlated SBS and ABS paths, and with an additional ABS MP included in the analysis that is not measuring a significant source.

5.1 Simulation results

The contribution analysis results indicate that an over-prediction of the ABS path (and under-prediction of the SBS path) occurs, refer to Figure 3a. The results in Figure 3a represent the case where both the ABS reference MPs are at an equal distance from the source, thus they "measure" the same amplitude and phase. The calculated contribution (the grey bar) is approximately +2.5 dB higher than the actual signal (the black bar) for the ABS contribution, and -3.5 dB lower for the SBS contribution. Comparing the calculated ABS and SBS contributions to the total response at the receiver, note that the ABS accounts for two-thirds of the contribution, while the SBS accounts for one-third of the contribution. The total calculated response at the receiver for this case matches the actual response.

Because the reference MPs (two ABS and one SBS) are all fully correlated, the resulting SVD yields only one singular value. The scaling of the contributions is determined by the eigenvector – In this case the scaling is the same for each reference MP. Since the contribution for each path (SBS and ABS) is summed, two-thirds of the contribution is calculated to be coming from the ABS path (two reference MPs, which are summed) and one-third from the SBS path (only one reference MP).

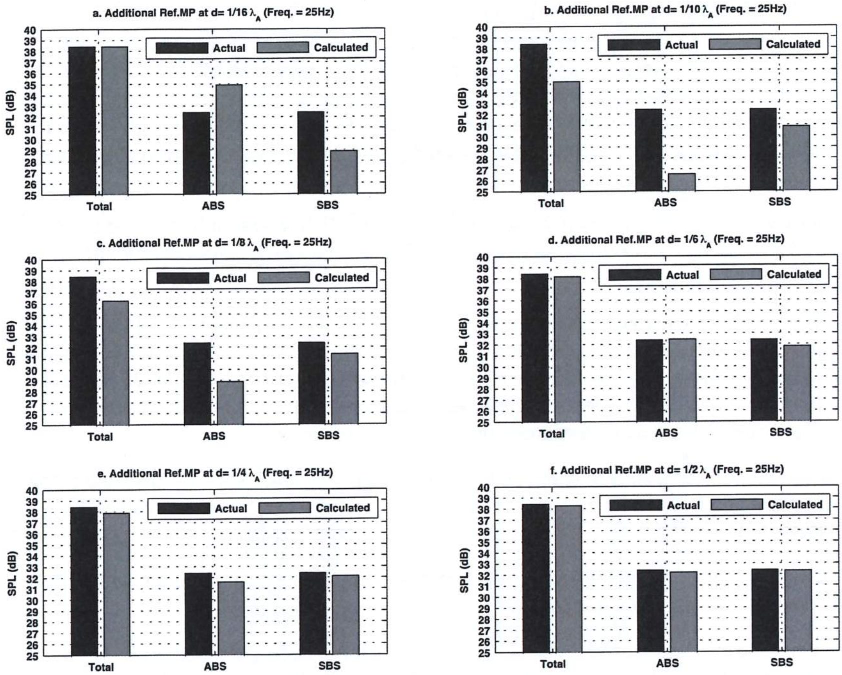

Figure 3: Contribution analysis with an additional ABS ref. MP at varying distance to the source

In Figure 3b, we see that the calculated contributions relative to the actual contributions have changed again, and in this case the calculated total response does not match with the actual total response. What is occurring here is the phase difference between the two ABS reference MPs causes a lower calculated contribution when the MPs are summed. The error in the measurement setup here is however apparent upon comparison to the actual response at the receiver position.

As the distance between the additional ABS reference MP increases, the relative amplitude at this reference MP reduces, which leads to a reduction in the contribution prediction error as well. At a distance of approximately 1/6λ (Figure 3d) the error is already less than 1 dB.

6. Discussion

The outcome of the OTPA simulation gives some insight into the likely cause of the erroneous contribution prediction that occurred in the road tractor OTPA case study: The additional microphone MPs (at tires, rear axle and engine cooling fan, refer to Figure 1) are not actually measuring a significant additional source since the road tractor is stationary. Therefore, these ABS reference MPs are mainly measuring cross-talk from the engine. Further, in the low frequency range it likely that the SBS path is highly correlated to the ABS path, thus the CTC method is unable to separate the contributions and the energy is simply spread out amongst the reference MPs, scaled by the relative amplitudes of the reference signals.

Recalling the potential sources of error for OTPA described in section 3, the cause for the error in the road tractor case appears to be due to *cross-coupling between input measurement positions*. The results of the OTPA simulation further support this conclusion.

It has been proven that number of reference MPs for a particular source is insignificant, since the energy is simply distributed amongst the reference MPs, and later summed to yield the predicted contribution [8]. The results of this study further shows that when two or more paths are highly correlated, and exhibit similar contributions at the response position, the number of reference MPs will influence the results – the sources/paths will be weighted according to the number of reference MPs.

The result of this study emphasises that consideration must be given to the physics when including MPs in the OTPA post-processing. The correct number of MPs and proper placement is critical, particularly for airborne sound: In general, the number of ABS MPs used in the OTPA should be adapted according to frequency range: At low frequency (i.e. the size of the source is much smaller than one-sixth of the wavelength), the source radiates uniformly as a simple point source, thus fewer microphone positions are required; whereas at high frequency the source can be seen as a combination of multiple sources, and will therefore exhibit directivity in the radiation pattern so several microphone positions are required to properly measure the source.

Note that the number of MPs actually used in the OTPA analysis can be adapted during post-processing – thus the measurement does not have to be repeated; only measurement data is either included or excluded from the analysis.

Further, the Principal Components (PCs) should be examined: a strong contribution from very few PCs is an indication that the reference signals are highly correlated. Referring to Figure 4, we see from the cumulative contribution of the PCs for the case study of the road tractor that at 25 Hz, the first PC contributes to over 75% of the overall signal. This gives an indication that there is a strong correlation between the reference MPs and that, consequently, the SVD used by the CTC method may not be able to suppress crosstalk between the signals efficiently.

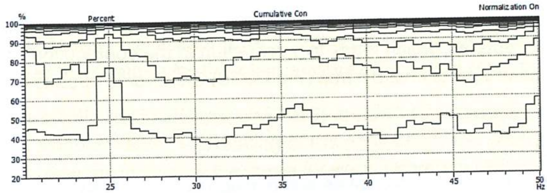

Figure 4: Road tractor OTPA case study, principal component analysis

A further conclusion derived from the OTPA simulation results is that even with highly correlated signals and with a good signal to noise ratio (SNR) for the ABS reference MPs, the contribution prediction results will lead to the correct practical conclusions. Particularly for ABS reference MPs, it is critical to ensure that the MPs are measuring a critical source – reference MPs associated with non-existing sources (at certain running conditions) will mainly measure cross-talk (considered as noise when judging the SNR), which will lead to incorrect source contribution prediction results, and subsequently erroneous conclusions.

7. Summary and conclusions

Upon comparison of the road tractor OTPA case study results and the OTPA numerical simulation results, several conclusions leading to practical considerations regarding sensor positioning and post-processing were conceived. These practical considerations are summarized in the following:

- The correct number of sensors and proper placement is critical for accurate source contribution prediction results, particularly for airborne sound sources/paths.
- The number of microphone positions per source used in the analysis should be adjusted during post-processing depending on frequency range: less microphone positions in the low frequency range, and more in the high frequency range.
- The correlation between reference measurement positions (i.e. the ability for successful cross-talk cancellation) should be checked by looking at the contribution of the PCs.
- A good SNR is important, particularly for airborne reference MPs, in order to minimize cross-talk (to be considered as noise). Microphones should be placed as close as possible to their assigned source.
- Microphone signals should be checked to ensure that they are not measuring significant cross-talk from another source. This could be checked by PCA, or by simply examining the relative amplitude and phase relationship at the microphone positions for the various sources.

By critically examining the OTPA data during post processing and keeping the suggestions listed above in mind, OTPA can be a convenient diagnostics tool leading to sufficiently accurate source/path contribution conclusions.

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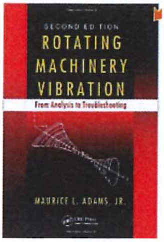

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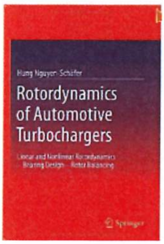

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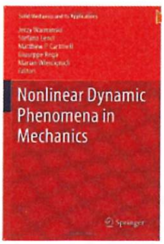

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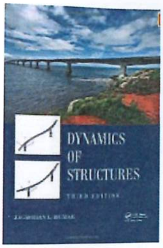

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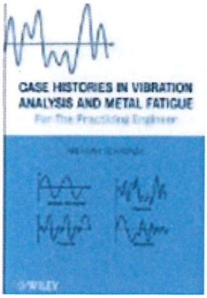

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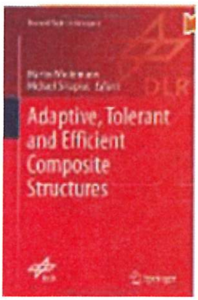

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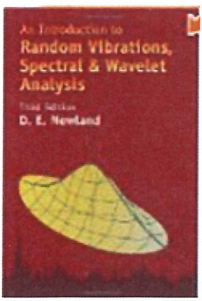

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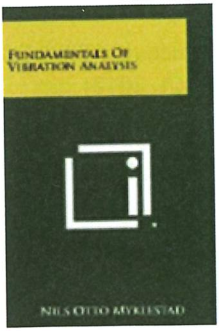

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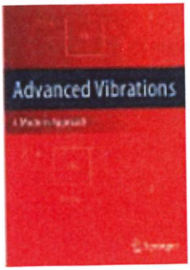

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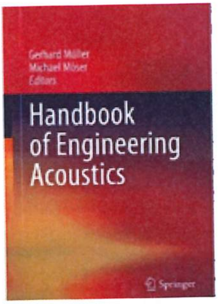

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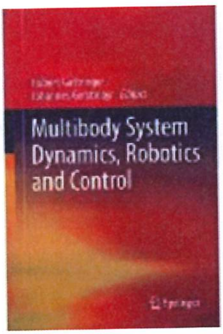

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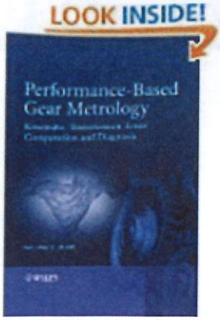

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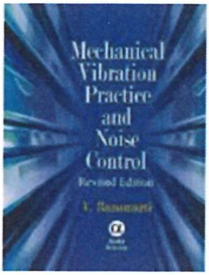

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Lindevangen 2 DK-3490 Kvistgård,
DenmarkTel +45 491 132244
Mob:+45 20223325
ao@vibratec.dk

- **Skattmästare**

Jessica Fromell
WSP Environmental
Box 13033, 402 51 Göteborg
Tel: +4631 7272916
jessica.fromell@wspgroup.se

Övriga ledamöter

Anders Brandt
Institute of Technology and Innovation
University of Southern Denmark
Niels Bohrs Allé 1, DK-5230 Odense M
Tel: +45 6550 7451
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