

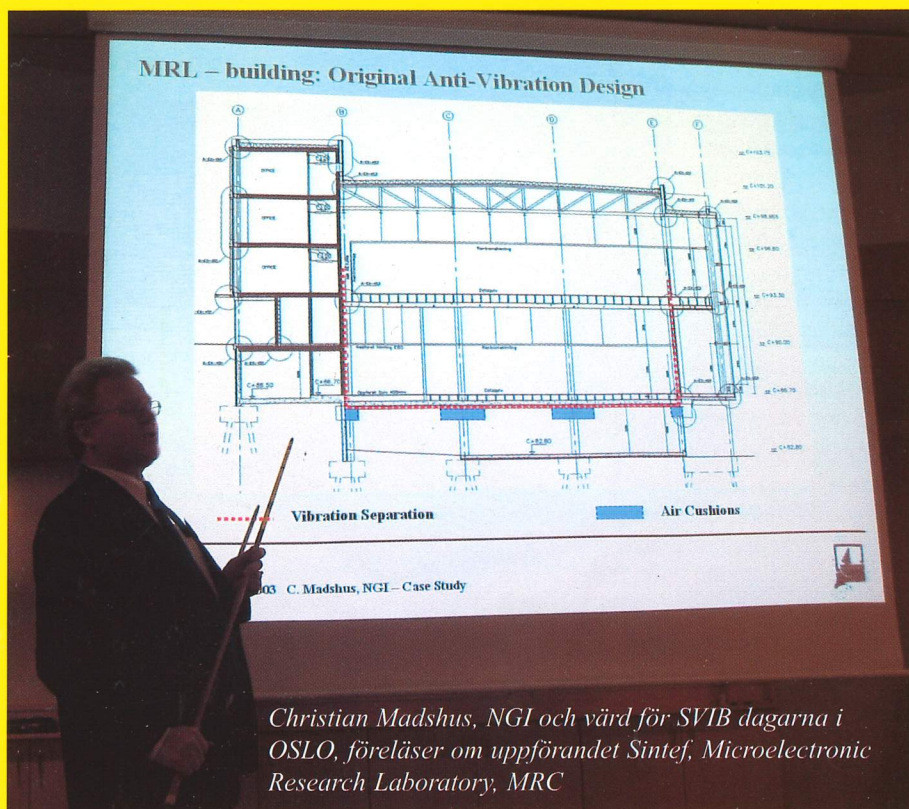
SVIB

VIBRATIONS

NYTT

Medlemsblad för Skandinaviska Vibrationsföreningen

Årg 21/22 Nr 4 December 2003 / Nr 1 Mars 2004



MRL - building: Original Anti-Vibration Design

Vibration Separation

Air Cushions

2003 C. Madshus, NGI - Case Study

Christian Madshus, NGI och värd för SVIB dagarna i OSLO, föreläser om uppförandet Sintef, Microelectronic Research Laboratory, MRC



SKANDINAVISKA VIBRATIONSFÖRENINGEN

Familjen *PULSE*

PULSE™, vårt bästsäljande Multianalys-system, är resultatet av mer än 60 års innovationer och erfarenheter. PULSE familjen kan tack vare valmöjligheterna med mjukvara och hårdvara anpassas till en mängd olika tillämpningar inom ljud och vibrationsanalys.

- Helt PC-baserad (dvs inga DSP-kort)
- Från små portabla system upp till hela nätverk
- Modulärt system från 0 upp till mer än 200 kanaler
- Batteridrivet och/eller nätdrivet
- Supporterar TEDS-givare (Transducer Electronic Data Sheet)
- Enastående dynamik tack vare 24-bitars A/D-omvandling
- Standard LAN-anslutning mellan dator och mätenhet, möjlighet till trådlös överföring
- Rapport i Word/Excel med aktiva diagram

För mer information titta in på www.bksv.se och www.bksv.com eller kontakta ditt lokala Brüel & Kjær kontor.



Huvudkontor: DK-2850 Nærum • Danmark
Telefon: +45 45 80 05 00 • Fax: +45 45 80 14 05
www.bksv.com • info@bksv.com

Brüel & Kjær Sverige • seinfo@bksv.com
Borås-kontoret: Tel: 033-22 56 22 • Fax: 033-123140
Huddinge-kontoret: Tel: 08-449 86 00 • Fax: 08-449 8610

PULSE i olika former

Brüel & Kjær 

Erfarenhet

Det känns bra att jobba ihop med personer med erfarenhet. Jag trivs med mitt jobb, med kollegor på min arbetsplats och på andra ställen. Personer som vet hur man ska göra, vad som kan gå fel och som lär sig av de fel som görs. Som konsult vill jag dela med mig av erfarenheter. Det känns bra för mig att få uppskattning och oftast betalt när att jag delar med mig av kunskaper och erfarenheter.

På en tidigare arbetsplats hade jag och en kollega diskussion med kunder om att ta fram en programvara för beräkning och med en databas där mätresultat ingick som underlag. Vi var tvungna att satsa lite själva för att få merparten finansierat. Det visade sig då att våra kollegor i företaget, på ett annat kostnadsställe, inte ville vara med och finansiera denna lilla del med hänvisning till att vi säljer ut vår kunskap. Vi kommer att få färre uppdrag i framtiden. Så blev det inte. Kunderna fick istället respekt för vår kunskap och erfarenhet och uppdragen har inte minskat.

En person med stor erfarenhet var Olle Backteman. Olle har avlidit, 68 år gammal. Minnesord över Olle presenteras på annan plats i tidningen. Jag träffade Olle första gången 1986 eller 1987 när jag arbetade på 3K Akustikbyrå. Jag minns Olle särskilt väl för att han var nyfiken och verkade tycka mycket om sitt jobb och kontakter med kollegor och kunder. Min senaste kontakt med Olle var efter att jag skrivit ledaren i höstnumret av

SVIB-Nytt om värdefulla mätresultat. Han skrev uppmuntrande till mig och skickade dessutom mail till alla Ingemanssöner att "noga läsa Martin Almgrens inlägg och begrundar detta".

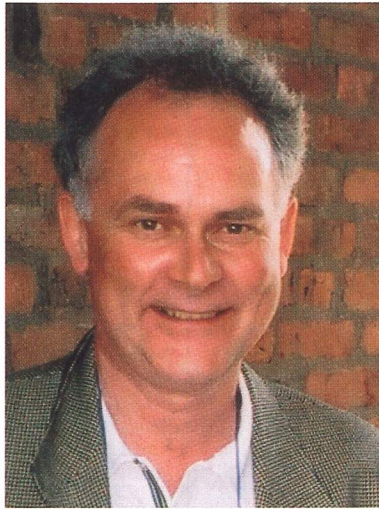
En annan person med stor erfarenhet är Henry Sandström. Henry har varit drivande inom standardisering inom vibrationsområdet för

byggverksamhet. Standardarbetet har lett till fem svenska standarder. Henry håller på att skriva en bok om sitt liv: "Norrlandspojken som gick sprängningsvägen". Det tycker jag är ett lovvärt initiativ. Kontakta Henry om ni vill veta mer.

Deltagande i standardiseringsarbete är ett sätt att dela med sig av kunskap och erfarenhet. Jag har tidigare beklagat mig över att standardisering av metoder och bedömningsgrunder som rör tredje man har allt svårare att hitta finansiärer. Det är

visserligen trevligt att få beröm och uppmuntran för det man gör, men man måste ju leva också. I synnerhet när man "säljer ut sin kunskap och erfarenhet". Myndigheterna måste ta ett större ansvar för att bevaka skattebetalarnas och övriga medborgares intressen.

På nästa styrelsemöte i Skandinaviska Vibrationsföreningen har vi många punkter på agendan som så småningom förhoppningsvis leder till välbesökta symposier och diskussionsmöten där erfarenheter sprids och utbyts. EUs direktiv om vibrationer i arbets-



SVIB styrelse

Ordförande

Martin Almgren, tekn dr
Ingemansson Technology AB
+4631 774 74 54

Vice ordförande

Ole Døssing, civ ing
Rovsing A/S
+45 44 20 08 29

Sekreterare

Alf Olsson, ing
SVIB Normtjänst AB
+4670 717 29 15, tis-tors +468 444 14 10 (SEK)

Övriga ledamöter

Thomas Abrahamsson, prof
Chalmers Tekniska Högskola
+4631 772 15 06

Christian Madshus, Dr scient
Norges Geotekniske Institutt
+47 22 02 30 44

Ylva Vidhög, civ ing
Forsmarks Kraftgrupp
+46173 821 30

Arbetsutskott

AU 1 & 2 Vibrationsmätning och -analys samt fordons- och maskinvibrationer

Kjell Ahlin, tekn lic
Saven EduTech AB
+468 473 50 94

AU 3 Trafik-, mark- och Byggnadsvibrationer

Tomas Odebrant, civ ing
Ingemansson Technology AB
+468 744 57 80

AU 4 Vibrationer inverkan på människan

Johan Granlund
Vägverket Konsult
+46 243 75388

AU 5 Forskning och utbildning inom vibrationsområdet

Thomas Abrahamsson, prof
Chalmers Tekniska Högskola
+4631 772 15 06

Innehåll:

Erfarenhet - <i>Martin Almgren, Ordförande, SVIB</i>	1
Protokoll från årsmöte med SVIB - <i>Alf Olsson</i>	4
Digital Vibrationssensor - <i>Christer Svensson</i>	7
Guldaccelerometern 2003 -	10
OlleBackteman- minnesord - <i>Erik Rudolphi</i>	12
Vibrations from the demolition of a large offshore platform - <i>H. Heyerdahl...</i>	13
CEEES Conference -	23
Kurser och Konferenser -	24
Steam turbine vibration characteristics - <i>Mike McGuire</i>	30
Jernbaneverket i Norge utfører fullskalaforøk -	49

Annonskostnad (inkl moms)

Mittuppslag	22 500:- /4 nr
Dubbeluppslag	20 000:- "
1 baksida + 3 helsidor	15 000:- "
Hel insida	11 250:- "
Halvsida	6 250:- "

Medlemsavgift 450:- / år

SVIB NORMTJÄNST AB

Box 1288, 164 29 KISTA
Alf Olsson +4670 71729 15
Postgiro 405 36 55-9
Bankgiro 59 57-5076

Kansli och redaktion:

SVIB
Box 1288, 164 29 KISTA
Telefon 0533-91590
Telefax 0533-81602
E-post svib.ntab@swipnet.se
Hemsida www.svib.se

Ansvarig utgivare:

Alf Olsson
e-post: olsson@sekom.se

miljön och som ska göras om till föreskrifter i medlemsländerna, är ett sådant ämne. Ett annat ämne är frågan om maxnivåer för vibrationer från tåg- och vägtrafik. Vad är en maxnivå? Är det det högsta värdet som inträffar på 100 år, medelvärdet av de fem högsta passagerarna, 95 %-percentilen eller något annat?

I Norge har man en definition och mätmetod. Nordic Vibration Research 2004 ska äga rum 3-4 juni i Stockholm. Passa på att anmäla Dig via www.svib.se för att få reda på det senaste inom vibrationsforskningen. ■

Martin Almgren

Ordförande,

Skandinaviska Vibrationsföreningen,

e-post: martin.almgren@ingemansson.se

Nästa nummer av

SVIB

V I B R A T I O N S

NYTT

kommer i Maj

Kontakta Saven Hitech



för råd om lämplig utrustning för
mätning och provning av
ljud och vibrationer


SAVEN HITECH

- | | |
|------------------------|----------------------------|
| ✓ ljudnivåmätare | ✓ PC-baserade mätsystem |
| ✓ laddningsförstärkare | ✓ vibratorer |
| ✓ kalibratorer | ✓ vibrationskontrollsystem |
| ✓ impulshammare | ✓ consulting |
| ✓ FFT-analysatorer | ✓ support |
| ✓ mätbandspelare | ✓ service |

SAVEN HITECH AB

TEL 08-505 641 00

E-MAIL info@savenhitech.se

www.savenhitech.se

FINLAND

TEL +358 9 72 55 49 60

E-MAIL info@saven.fi

NORGE

TEL +47 33 05 66 88

E-MAIL info@saven.no

SKANDINAVISKA VIBRATIONSFÖRENINGEN, SVIB

2003-11-25

PROTOKOLL FRÅN ÅRSMÖTE

Närvarande: Se separat deltagarförteckning

1

Fastställande av dagordning

Det i enlighet med stadgarna och med kallelsen utsända förslaget till dagordning godkändes.

2

Val av mötesordförande och mötessekreterare

Till mötesordförande valdes Martin Almgren och till mötessekreterare Alf Olsson.

3

Val av justeringsmän för dagens protokoll

Gabor Csaba och Elisabeth Blom utsågs att jämte ordföranden justera protokollet från årsmötet.

4

Fråga om stämman blivit behörigen kallad

SVIBs verksamhetsberättelse, balans- och resultaträkningar samt budget för tiden från den 1 september 2002 till den 31 augusti 2003 hade sänds ut per e-post tillsammans med kallelse och dagordning den 2 november 2003.

Stadgarnas krav på kallelse högst sex och minst två veckor före årsmötet var således uppfyllt varför stämmans behörighet kunde godkännas.

5

Avgående styrelsens redogörelse för verksamheten under det föregående verksamhetsåret

Ordföranden föredrog sin och styrelsens berättelse för verksamheten för tiden 1 september 2002 - 31 augusti 2003.

6

Revisionsberättelse

Revisorernas redogörelse över SVIBs räkenskaper utfärdad av Torbjörn Kloow och Bent Simonsson lästes upp av sekreteraren. Revisorerna hade föreslagit att årets underskott kronor 27 967,94 skulle föras i ny räkning.

7

Fastställande av resultat- och balansräkningar

Sekreteraren redovisade balans- och resultaträkningarna. Ordförande hade tidigare redogjort för det underskott som uppstått till följd av att den sista delen av de medel för utförda Standardiseringsprojekt inom CEN/TC 231 och som parkerats i SVIB nu hade betalats ut till SIS. Detta ärende hade föredragits redan föregående år.

8

Beslut rörande ansvarsfrihet för den avgående styrelsen

Med hänvisning till styrelsens verksamhetsberättelse samt det som sagts i samband med avgivna redovisningar enligt ovan beslutade stämman att bevilja styrelsen ansvarsfrihet för det gångna verksamhetsåret.

9

Fastställande av budget för kommande verksamhetsår

Medlemsavgifter och behållningen från Vibrationsdagen samt specialsymposier beräknades inbringa ca 47 000 kronor. Kostnader till totalt samma belopp hade upptagits för Föreningens korrespondens, serviceavgifter för pensionärer och hedersledamöter samt för revision och försök med webbaserade diskussionsgrupper samt för styrelsen och arbetsutskotten. Iris Turunen-Rise föreslog att SVIB skulle kontakta Europeiska Akustiska Sällskapet när det gällde utformningen av de webbaserade diskussionsgrupperna.

10

Fastställande av föreningens årsavgift

Medlemsavgiften för 2003/2004 fastställdes till oförändrat 450:- kronor per år.

11

Val av ordförande för tiden t o m nästkommande årsmöte

Valberedningen hade genom ett e-post från Nils-Åke Nilsson informerat sekreteraren om sitt förslag till styrelsen för kommande verksamhetsperiod.

Till ordförande för SVIB: omval av Martin Almgren.

Till vice ordförande: omval av Ole Døssing.

Till sekreterare: omval av Alf Olsson.

Till övriga ledamöter: omval av Ylva Vidhög, Christian Madshus och Thomas Abrahamsson.

Stämman förrättade valet och fastställde i tur och ordning Valberedningens förslag.

12

Val av två revisorer och revisorssuppleant

Valberedningen föreslog omval av Bengt Simonsson och Torbjörn Kloow som revisorer för nästkommande verksamhetsår. Som revisorssuppleanter föreslogs omval av Knut Bertelsen
Valberedningens förslag godkändes.

13

Tillsättande av en valberedning

Nils-Åke Nilsson erbjöd sig att kvarstå som sammankallande i valberedningen. Vidare omvaldes Leif Nielsen, DK och som ny ledamot i valberedningen invaldes Mats Backholm, FI.

14

Övriga ärenden

Några övriga frågor förekom inte.

Vid protokollet

Justeras:

Alf Olsson

Martin Almgren

Elistabeth Blom

Närvarande:

Kjell	Ahlin
Martin	Almgren
Mats	Backholm
Ole	Døssing
Christian	Madshus
Tomas	Odebrant
Alf	Olsson
Gabor	Chaba
Elisabeth	Blom
Iirs	Turunen-Rise
Ylva	Vidhög
Johan	Granlund

Gabor Csaba

Blekinge Tekniska Högskola
Ingemansson Technology AB
Vibkon AB
Rovsing AS, Köpenhamn
Norges Geotekniske Institutt, Oslo
Ingemansson Technology AB
SNT, Säftele
Alstom Power Sweden
ÅF Energi & Miljö
Standard Norge
Forsmark Kraftgrupp AB
Vägverket Konsult

Allt inom Givare från Dytran för Acceleration, Tryck & Kraft



Kraftgivare

- Område: 0 - 11 kN
- Känslighet:
0.1– 110 mV/N

Impulshammare

Tillbehör

- Kablar, adaptrar

Accelerometrar

- Industrigivare
- Lab-givare
- Miniatur
- Tri-axiella
- Hög Temperatur
- Shock
- Airborne

Tryckgivare

- Område: 0 - 1000 bar
- Känslighet:
0.05 - 140 mV/bar
0.05 pC/bar

Signalkonditionering

- Power Supplies
- Laddningsförstärkare

Acoutronic AB

Box 1180
SE-181 23 LIDINGÖ

Stockholm: Tel: 08-765 02 80

acoutronic@acoutronic.se
<http://www.acoutronic.se>

Göteborg: Tel: 031-744 34 50



Digital Vibrationssensor

Jag ringde häromdagen till Christer Svensson på Sigicom AB för att kontrollera att vi hade rätt uppgifter i SVIB registret. Det hade vi och det gladdde mig mycket eftersom jag tidigare nåtts av uppgifter om att företaget haft det svårt en tid. Man hade tappat en stor kund o s v. Nu berättade Christer att man tagit fram en ny digital givare och det tyckte jag borde intressera fler än mig. Därför bad jag honom att skriva en kort artikel om produkten. Tack för bidraget som följer.

Alf Olsson

Vid mätningar på byggnader i samband med byggverksamhet som sprängning, pålning, schaktning samt vid påverkan från väg- och järnvägstrafik används vanligen geofoner för att mäta vibrationsnivån.

Hittills har dessa varit analoga. En traditionell geofon lämnar en analog spänning ut som sedan tas om hand av ett instrument som digitaliserar och lagrar informationen. Bland nackdelarna finns störningskänslighet, geofon och instrument måste samkalibreras, och att automatisk överföring av identiteten på geofonen saknas.

Nu har Sigicom utvecklat en digital geofon som ger många fördelar.

Den lämnar ifrån sig mätsignalen i digital form varför efterföljande kablar och kommunikationsutrustning inte påverkar signalkvaliteten.

Den digitala lösningen innebär också att endast geofonen behöver kalibreras, eftersom den efterföljande utrustningen bara hanterar digital kommunikation. Geofonen lämnar också ifrån sig sitt serienummer, kalibreringsdatum och annan väsentlig information för full spårbarhet.

Sigicoms digitala geofon har dessutom en inbyggd digital signalprocessor som utför alla filtreringar och detekteringar som behövs för mätning enligt olika normer.

I praktiken innebär detta att man bara behöver ställa in vilken standard man vill mäta efter, sedan utförs den aktuella signalbehandlingen.

Den digitala geofonen klarar idag följande svenska standarder:

SS 460 48 66 Vibration och stöt – Riktvärden för sprängningsinducerade vibrationer i byggnader. *Svängningshastighetens maxvärde inom 5-300 Hz*

SS 02 52 11 Vibration och stöt - Riktvärden för vibrationer i byggnader härrörande från pålning, spontning, schaktning och packning för bedömning av skaderisk

Svängningshastighetens maxvärde inom 5-150 Hz eller 2-150 Hz

SS-ISO 8569 Vibration och stöt – Vibrationskänslig elektronisk utrustning- Metod för mätning och rapportering av vibrationsdata i byggnader.

Accelerationens maxvärde inom 5-300 Hz

SS 460 48 61 Vibration och stöt – Mätning och riktvärden för bedömning av komfort i byggnader.

Maximal RMS-nivå för den vägda svängningshastigheten inom 1-80 Hz

Eftersom filtrering och detektering ligger i signalprocessorns programvara går det lätt att ta fram och installera nya standarder.

Till exempel har geofonen kapacitet att beräkna responsspektra i realtid enligt förslaget

ISO/WD 18431-4 Mechanical vibration and shock – Signal processing – Part 4: Shock response spectrum analysis

Digital kommunikation och strömförsörjning sker via en busskabel som också tillåter att många geofoner samtidigt kan vara inkopplade i samma nätverk dvs. på samma kabel. Geofonen kan samtidigt ta ut maxvärden inom ett ställbart tidintervall och vid trigg-

ning sampla ett långt tidförlopp med en hög samplingfrekvens.

Genom att effektförbrukningen är extremt låg (20 mW) vid kontinuerlig mätning så lämpar sig den digitala geofonen utmärkt för långtidsmätning med batteridrivna mätsystem.

Geofonen ingår i ett system där även andra typer av sensorer, både statiska och dynamiska, kan kopplas in ■

/Christer Svensson, Sigicom AB



ansvars besiktning ab

... har allt sedan etableringen 1972 varit verksam i konsultbranschen, huvudsakligen inom områdena som rör miljöstörningar.

... skall vara den självklara samarbetspartnern genom att utföra sina åtaganden i rätt tid till rätt kvalitet med ett högt miljötänkande och ekonomiskt motsvara kundens förväntningar. Detta åstadkommer vi genom ständig utveckling av spetskompetens inom våra teknik-områden och stödsystem.

... är idag etablerade i Göteborg, Uddevalla, Malmö/Lund med huvudkontor i Stockholm.

... tillhandahåller kvalificerade tjänster inom:

- Riskanalys
- Besiktning
- Mätning
- Rådgivning
- 3:e manshantering – Juridik och skadehantering
- IT/IS databassystemet VIPNET™

Vipnet™ är ett Internetbaserat databassystem som hjälper till att hålla ordning och reda på miljöstörningar som buller, vibrationer, grundvatten, lutningar/sättningar, trafik, m.m.

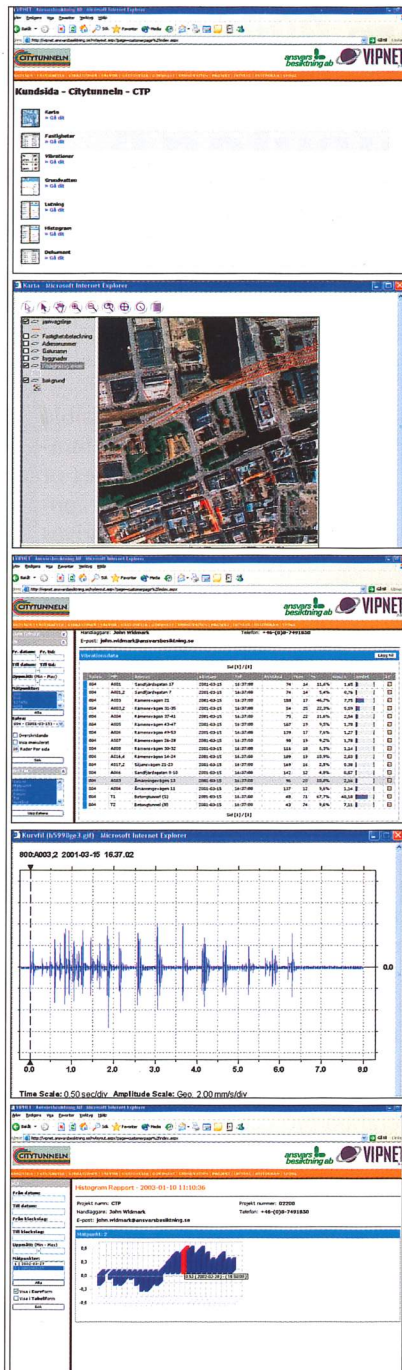
Vipnet™ och våra andra tjänster används idag på Citytunnelprojektet, Nordlänken, E4:an Uppsala – Mehedeby m

Ansvarsbesiktning AB

Besöksadress: Bisittargatan 24, Box 4105, 129 04 HÄGERSTEN
Tel. 08-749 18 50 | Fax 08-647 11 37

Address: Herkulesgatan 3 A, 417 01 GÖTEBORG
Tel. 031-510 900 | Fax 031-510 985

E-post: info@ansvarsbesiktning.se | www.ansvarsbesiktning.se



Guldaccelerometern 2003

The "SVIB Guldaccelerometern" was 2003 awarded to *Docent Leif Kari*.



Professor Leif Kari, Dr. Tech.

He has at the Marcus Wallenberg Laboratory, MWL, for Sound and Vibration Research at the Royal Institute of Technology, KTH, in Stockholm produced valuable research results on vibration isolation, in particular vibration isolator characterization, including material acoustics, in the audible frequency range.

His findings have thus improved the possibilities to understand and predict noise abatement with vibration isolators. His experience covers both measurements and theoretical modelling.

He started his research activities with the aim to develop a method for measuring the dy-

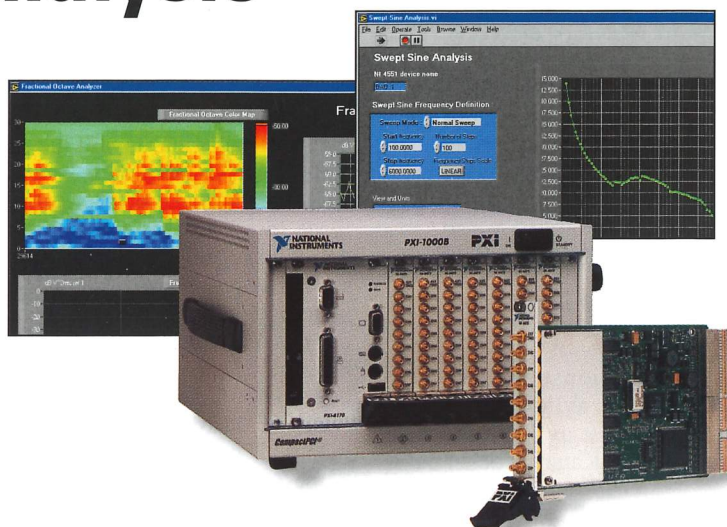
namic stiffness of vibration isolators. For this he designed a test rig which gave him the measurement results of vibration, also in the audible frequency range, that he needed for the next step which was to develop theoretical dynamic stiffness models.

In the latest reports he explains the possible use of magneto-rheological rubber for vibration isolators. The mechanical properties of MR-material can be rapidly changed by applying an external magnetic field. This gives good opportunities for manufacturing of equipment for active vibration control also in the audio frequency range.

The Chairman of the Scandinavian Vibration Society said while giving him the "SVIB Guldaccelerometern" that he hoped that the award would encourage Leif to continue his research with his team at KTH and increase the long list of publications, that he started to produce in 1988, amongst which 20 papers have been published in internationally reputed periodicals after assessment by referees'. His work has made him well known, not only to the members of the Vibration Society, after participation in several Vibration Days and Special Symposia but also internationally following participation in several cooperation projects.

The "SVIB Guldaccelerometern" was this year donated by PCB Piezotronics, Inc. and at the time of award presented by Anders Åhlström, PCB Scandinavia AB ■

Sound and Vibrations Analysis



With LabVIEW™ and PXI

Perform FFT, octave, THD, SINAD, level analysis, and more with LabVIEW and the new NI 4472 dynamic signal acquisition module. The NI 4472 features:

- 120 dB dynamic range (24 bit)
- Eight analog inputs
- 45 kHz bandwidth
- Multiple module expansion and synchronization

[ni.com/
soundandvibration](http://ni.com/soundandvibration)

Get more information
at [www.ni.com/
soundandvibration](http://www.ni.com/soundandvibration)



Sweden: +46 8 587 895 00
ni.sweden@ni.com

Norway: +47 66 90 76 60
ni.norway@ni.com

© 2004 National Instruments Corporation. All rights reserved. Product and company names listed are trademarks or trade names of their respective companies.

✚ 5/1-04 OlleBackteman- minnesord



Olle Backteman - En praktisk vibrations- tekniker

Akustikern och vibrationsteknikern Olle Backteman, Ekerö, har avlidit 68 år gammal. Han sörjes närmast av hustrun Lena, sonen Peter med hustru Katarina och 4 barnbarn.

Olle föddes och växte upp i Gustav Vasa församling i Stockholm. Han gick i Vasa Real och i Högre Tekniska Läroverket i Stockholm.

Efter examen arbetade Olle på Atlas Copco AB där han var ansvarig för ljud- och vibrationsmätningar på företagets produkter, därefter på Billman Regulator AB där Olle var ansvarig för provning av nya produkter som en länk mellan utveckling och konstruktion.

Olle anställdes 1964 vid Akustikbyrån och utvecklades snabbt till en mycket kunnig mätingenjör. Han klarade snart allt från komplicerade impedansmätningar i ubåtssektioner till mätningar av vibrationer i berg. Olles omfattande mätningar av punktexciterat ljud i byggnader ledde till en byggforskningsrapport som blivit ett referensverk.

1983 lämnade Olle Akustikbyrån för att arbeta i egen regi med bland annat kvalificerade vibrationsmätningar. Han deltog i forsk-

ningsprojekt inom sten- och glasindustrin och var en mycket uppskattad lärare för utbildning av skyddsingenjörer vid Arbetarskyddsstyrelsen.

Från 1987 var Olle verksam som säljare och demonstratör av avancerade mätinstrument och system för tillståndskontroll på Acoustic AB. Med sin gedigna tekniska bakgrund vann han respekt hos kunderna.

1994 började Olle på Ingemansson Technology AB och fick då tillfälle att utöva sina både praktiska och teoretiska kunskaper inom hela akustik- och vibrationsområdet. Han tillförde företaget stor erfarenhet och kunskap och kom att bli läromästare i mätteknik för många unga ingenjörer. Olle delade också generöst med sig av sitt kunnande till övriga inom det internationella vibrationsområdet. Hans stora engagemang i Skandinaviska Vibrationsföreningen, SVIB, vittnar om detta. Han var utskottsordförande i SVIB under många år och ordnade bland annat ett antal välbesökta och uppskattade seminarier med mätanknytning. För sina insatser belönades Olle 1993 med SVIB's guldnål.

Arbetet var Olles stora intresse men det hindrade honom inte från att engagera sig inom idrottsrörelsen. Han var idrottsledare på Ekerö där han i över 17 år tränade pojkar i fotboll och ishockey samt var ledare för flera lag. På senare tid var golf hans stora sportintresse.

Olle var en mycket godhjärtad och glad person. Han var alltid positiv, ställde helt upp för företaget och sina arbetskamrater. Inget var omöjligt för honom. Vi som på olika sätt har haft förmånen att få utnyttja Olles kunnande kommer att sakna hans praktiska kunnande för tekniken och hans intresse för sina medmänniskor. Vi kommer att minnas hans pedagogiska skicklighet och hans lugna och trygga sätt

För vänner och kollegor inom SVIB och på Ingemansson Technology AB

Erik Rudolphi

Vibrations from the demolition of a large offshore platform

Presented at SVIB 2003 – The Vibration Day, 25 Nov. 2003 Norwegian Geotechnical Institute, OSLO

H. Heyerdahl, C. Madshus, K. Tronstad & E. Lied

Norwegian Geotechnical Institute, Oslo, Norway

ABSTRACT: Demolition of a steel offshore platform at Aker Stord yard in western Norway involved toppling of large structure parts by explosive blasting. The weight of each part, called lattice frames, was approximately 3000 tons, with a centre of gravity 35 m above the ground when placed upright onshore. NGI was asked to predict ground vibrations from ground impact and to evaluate possible damage to nearby infrastructures. Limit values for damage of these constructions were suggested. Vibration predictions were based on literature regarding demolition of tall chimneys, dynamic ground compaction and data from rock blasting. A cube root scaling scheme, also accounting for varying ground conditions, was found suitable. Results from drop-tests at the yard area were performed to improve ground vibration predictions. Measurements during the toppling operation verified that ground vibrations agreed well with predicted values. No damage on constructions was reported.

1 Introduction

Scrapping of the steel offshore platform "Maureen" owned by Phillips Petroleum Company has been performed at Aker Stord yard. The project has been undertaken by Aker Offshore Partner AS. Several large platform parts were lifted onshore at Eldøy Quay to be toppled in the scrapping area. A large lifting barge was used to lift the platform parts onshore at Eldøy Quay. The yard area after the onshore lifting operation is shown on Figure 1.

The largest platform parts to be toppled were the two lattice frame structures shown in Figure 1, each weighing close to 3000 tons, and having a centre of gravity ~35 m above ground when placed upright onshore. The total mass was reduced before toppling to

2300 tons by removal of parts of the lattice frames while the structure was still standing in upright position. The supporting legs then were cut off by explosive blasting to initiate the toppling. NGI was hired for evaluation of ground vibrations caused by the impact from the toppling operations, and for defining limit values for objects assumed sensitive to vibrations resulting from the ground shock from the impact of the structures hitting the ground. In this paper, predicted and measured vibration is discussed.

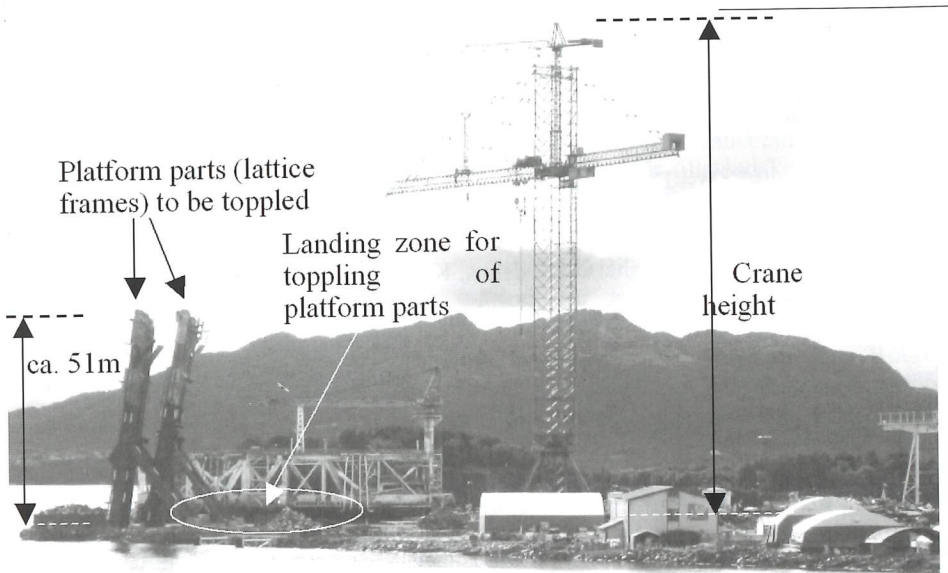


Figure 1. Two large platform parts placed onshore at Eldöy Quay. The giant crane may be moved on rails from current position to the wharfside on left side of picture.

2 Acceptance criteria and analysis approach

2.1 Acceptance criteria for ground vibrations

A giant crane in the yard area was assumed to be the most sensitive nearby construction. No specifications of acceptance criteria for such cranes related to ground vibrations of this kind were found. Tentative criteria had to be established, and were based on two criteria: 1) information from the crane company that the crane is designed to resist a sideways deflection of 0.5 in the top, and 2) general designing criteria, namely that the crane is able to resist a sudden loss of load, for instance in case of a sling break, which will produce a considerable dynamic load on the crane. This information has been used to esti-

mate corresponding loads in the lower part of the crane, based on crane geometry and structural information. The natural frequency of the crane, which was unknown, is important in such a calculation. A natural frequency of 0.5 Hz was assumed for the relatively flexible crane. Thus, ground vibrations would expectedly be "high-frequent" when compared to the natural frequency of the crane. Therefore, the dynamic load from ground vibration would be concentrated in the lower part of the crane.

Criterion 1) indicates that vibration velocity of the ground below the crane's foundations of ca. 300 mm/s is acceptable. Since deflection criteria are normally based on serviceability load (SLS), this value is assumed to have a large safety margin. Criterion 2) gives much higher acceptable vibration (2-3

times), which is not unreasonable, since this is an accidental load criterion (ULS). Vibration acceptance criteria for nearby buildings have been estimated from the Norwegian Standard NS 8141 (2001).

2.2 Vibration from object with large extension

Vibration prediction has been performed assuming that vibration is generated by a point source. When the extent of a vibration source is large, definition of such a point source may be difficult if the distance to the vibration source is small. In our case, both height and width of the platform part hitting the ground exceed 50 m, while the shortest distance from the of the platform part when hitting the ground to the giant crane is less than 50 m. Definition of the correct distance to an assumed point source may therefore be doubtful. In addition, the structure is discontinuous, consisting of three large main pipes

connected by smaller pipes. It may be questioned whether all of the energy from the impact is relevant for predicting ground vibration in a point.

To overcome this problem, predictions of ground vibrations for the giant crane have been made for various cases, A, B and C. In case A, only the upper part of the structure, having most of the potential energy (ca. 1550 tons with gravity centre 43 m above the ground) was considered. In Case B, the upper part of the main pipe closest to the giant crane was studied, taking the distance from the local mass centre of this structure part (ca. 650 tons, gravity centre 43 m above ground) into the prediction model. In Case C, all of the mass (ca. 2300 tons, gravity centre 35 m above ground) and the distance to the global mass centre of the structure when hitting the ground were used. The three cases are illustrated in Figure 3 together with the situation after toppling of the first lattice frame.

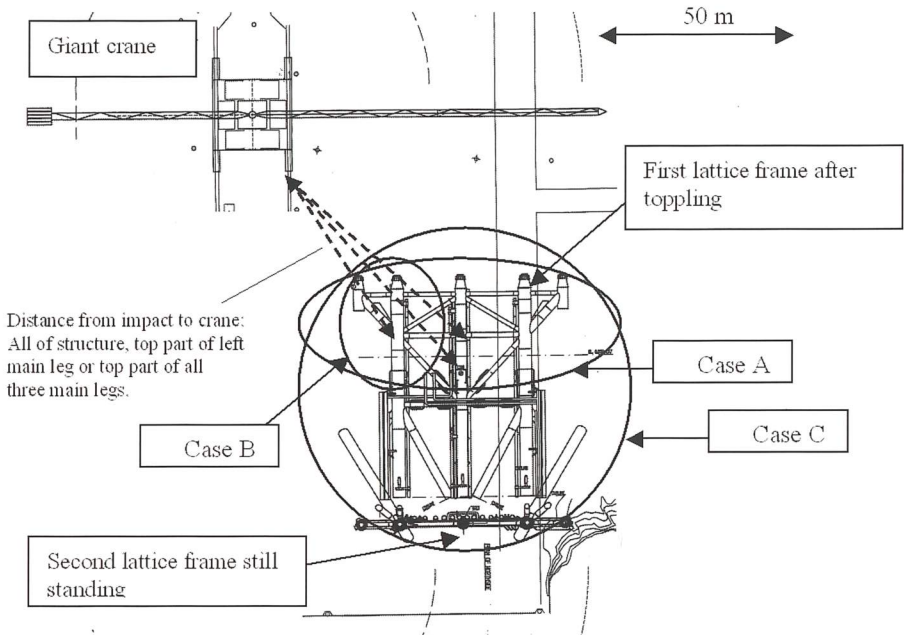


Figure 2. Analysis of ground vibration. Distance from assumed point source to giant crane for three cases A, B and C. Situation after toppling of first lattice frame.

3 Vibration predictions

3.1 Case records from literature

Generation of ground vibration from dynamic impact is a complicated process, involving several coupling and loss mechanisms. Numeric simulations therefore may easily lead to wrong conclusions. Empirical methods yield more reliable results. Published data from measured vibration from demoli-

tion of chimneys, compaction of rock fill by heavy drop weights from large height ("Dynamic deep compaction") and from tests involving blasting of large charges in rock (military tests) have been reviewed and systematized. The studied case records are summarized in Table 1. The records mainly originate from tests performed on deep rock fill or other soils of various types, in addition to blasting in rock.

Reference	Type of test	Ground	C_s	ρ
			m/s	kg/m ³
Madshus and Langberg (1999)	Rock blasting	Rock	2000	2700
Mayne (1985)	Dynamic deep compaction	Not spec.	300	2000
Skip (1997)	Demolition, concrete chimneys	Hard soil	600	2100
Skip (1997)	Drop weights on ground	Soil	300	1900
Skip (1997)	Drop weights on concrete slab	Soil	300	1900
Madshus et al. (1993)	Drop weights on rock fill	Rock fill	700	2200
Skip and Buckley (1977)	Rigid drops on ground	Soil	300	1900

Table 1. Back calculated case records from literature. Type of test and ground conditions.

No empirical data for kinetic energies as large as involved in the platform toppling operation (ca. 10^8 J) was found. However, by use of a scaling law, it is possible to upscale data from tests involving smaller energies to events involving far higher energies. On the site at Eldøy there is ca. 1-2 m of rock fill above the rock surface, which represents a stiffer system than the sites from which data has been collected. On one hand, this should imply that vibrations relatively speaking should be smaller. On the other hand, loss mechanisms are less prominent than for soils, and this pulls in the opposite direction. To be able to transfer empirical data from soil to rock, a dimensionless scaling scheme has been used, taking into consideration the mass density and shear wave velocity of the ground and, thus, implicitly also the strength of the ground. Based on evaluation of data from

literature and drop tests on the site, it has been concluded that maximum ground vibration velocity (PPV – Peak Particle Velocity) follows a cube root scaling law, i.e. it is related to $\sqrt[3]{W}$ in which W is energy that "hits the ground" in the collision or blasting. Mayne (1985) and Skip and Buckley (1977) use square root scaling to study vibration decay with distance, while cube root scaling is used in the other references in Table 1. The recorded peak particle velocity in m/s from the studied case records is assumed to follow the relation in Equation 1 below:

$$V = C_s \cdot v_o \cdot r^n \quad (1)$$

where C_s is the shear wave velocity in the ground in m/s, v_o is the dimensionless velocity, n is an empirical exponent and r is the dimensionless distance, calculated from real

distance R in m, using the expression in Equation 2 below:

$$r = R / \left(\frac{W}{C_s^2} \right)^{\frac{1}{3}} \quad (2)$$

where W is the energy in Joule for the case considered, and r is the mass density in kg/m^3 .

Dimensionless scaling of peak particle velocity and distance from the vibration source has been applied to compare data from varying drop energies and ground conditions. The value of C_s at the different sites is taken from the published case records, or, if not given in the literature, has been assumed based on experience. In Figure 3, dimensionless peak particle velocity (V/C_s) is plotted versus dimensionless distance, based on case records listed in Table 1. Predicted ground vibration

for the giant crane based on the various vibration models deduced from literature is given for the three different cases A, B and C, as defined in Section 2.2 and Figure 2.

The scatter in the studied empirical data is relatively large also after dimensionless scaling. This may be explained by the differences in vibration source and coupling to the ground. For instance, the energy released in an explosion may not necessarily be comparable to kinetic energy when it comes to prediction of vibration. In addition, differences in site characteristics (ground type, depth of soil above rock i.e.) as well as scarcity in the published literature with respect to soil conditions are of importance.

In Figure 3, predicted dimensionless peak particle velocity for the three cases A, B and C is included for all curves. Case A gives the

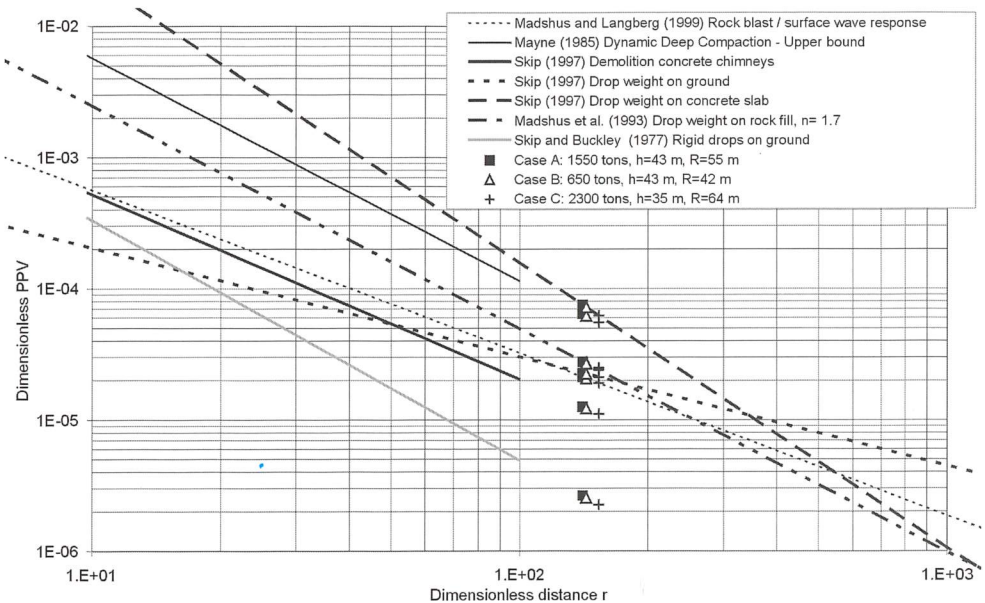


Figure 3. Dimensionless scaling of case records. Predicted vibration from toppling of a lattice frame with mass 2300 tons and centre of gravity 35 m above the ground is shown in diagram.

highest predicted vibration. The highest estimate for ground vibration close to the giant crane corresponds to peak particle velocity 200 mm/s, assuming a shear wave velocity of 2000 m/s at the site. The lowest estimate corresponds to peak particle velocity 5 mm/s.

3.2 Drop tests

The predicted vibration of 200 mm/s for the giant crane from the "worst case" line in Figure 3 is below the limit value of 300 mm/s for the giant crane. However, taking uncertainty into account, it was decided to calibrate the vibration predictions by performing drop tests on the site at Eldøy Quay to ensure minimal risk of damage on the giant crane or other nearby structures.

A total of six drop tests were performed, using drop weights made of steel plates with mass 1.8 (two tests), 4.8, 9.4 and 15.6 (two tests) tons. Ball shaped weights would have been preferable, but were not available. The weights were dropped from 18 m height. A total of 8 SM6 geophones were placed along a line in measurement points 5, 10, 20, 40 and 80 m from the drop site (some small changes in distances for some of the tests). Such an exponential increase in distance between the sensors is assumed to be suitable for studying impact events, because peak particle velocity tends to decrease linearly in a log-log plot. In each measurement point one vertical sensor was installed. In addition, horizontal sensors were installed at distances 5, 10 and 80 m, measuring in the direction pointing away from the vibration source. The first drop test, using a drop weight of 1.8 tons, resulted in much higher vibrations than the other tests, relatively speaking, particularly for sensors at short distances (5 and 10 m). The reason for this peculiarity is not fully understood. However, the weights had to be dropped from the same point for all six tests. It may be that coupling to the ground was better for the first tests, before

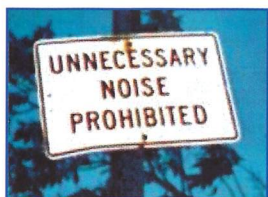
the ground was disturbed. (After each test, the drop weight was removed by an excavator, and the ground on the landing point was prepared for the next test. No compaction was performed between the tests).

Results from the drop tests (horizontal and vertical sensors) are plotted in Figure 4 as dimensionless peak particle velocity versus dimensionless distance, together with literature curves. Based on normal distribution of measured peak particle velocity-values from the drop tests, the 95% and 50% confidence lines have been calculated and are also included in Figure 4. The position of the upper confidence line is very much influenced by the results from the first drop test, as discussed above. It is common practice to use 95% confidence line to predict upper estimates of ground vibration. The upper estimate is then assumed to be a conservative (high) prediction for the vibrations, the mean value estimate being our "best guess". The predicted mean value estimate for ground vibration is not reduced from the drop tests, but the 95% estimate is considerably lower than the "worst case" curve based on the literature records.

4 Measurements during toppling

For maintenance of the giant crane it was of importance to document ground vibrations during the toppling of the platform structures. Out of two large such drops, the first one was assumed to be most critical, due to smaller distance to the crane. The ground was instrumented by accelerometers and geophones along two axes; see Figure 5. Accelerometers were used close to the impact zone; while geophones were used at larger distances. Accelerometers were also placed at the crane foundations, and at selected points in the crane tower. Strain gauges were placed in the crane tower to monitor dynamic loading in selected girders.

Huller om Buller?



WSP Akustik ingår i WSP Group med ca 80 akustiker i koncernen. Det gör akustikheten inom WSP till en av Europas största aktörer i branschen. I Sverige är vi ett 40-tal medarbetare med över 40 års erfarenhet av specialisttjänster inom områdena akustik, buller och vibrationer. Vi har välutrustade ljudlab och programvaror som täcker de flesta behov inom alla branscher.

WSP Akustik har kontor i Stockholm, Göteborg, Malmö, Helsingfors, Oslo, Bergen, Kristiansand, Stavanger, Tønsberg, London, Birmingham och Bristol.

Vi är ackrediterade av Swedac att utföra mätningar inom akustik- och vibrationsområdet. Vi tillhandahåller kundanpassad utbildning och är också aktiva inom standardiseringsorgan för att bistå med vår expertis och för att vara med och styra utvecklingen åt rätt håll.

Välkommen att kontakta oss!

Stockholm: 08-555 233 00, Göteborg 031-72 72 634, Malmö 040-699 63 33

www.wspgroup.se

WSP Akustik



Vilka givare behöver Ni?



Piezelektriska

Accelerometrar 1-ax, 3 ax
Kraftgivare 1-ax, 3-ax
Tryckgivare

Piezoresistiva

Tryckgivare

Kapacitiva

Accelerometrar

Total Customer Satisfaction Guarantee!

 **PCB SCANDINAVIA AB**

Ring 08-444 38 70

E-post: info@pcbscandinavia.se

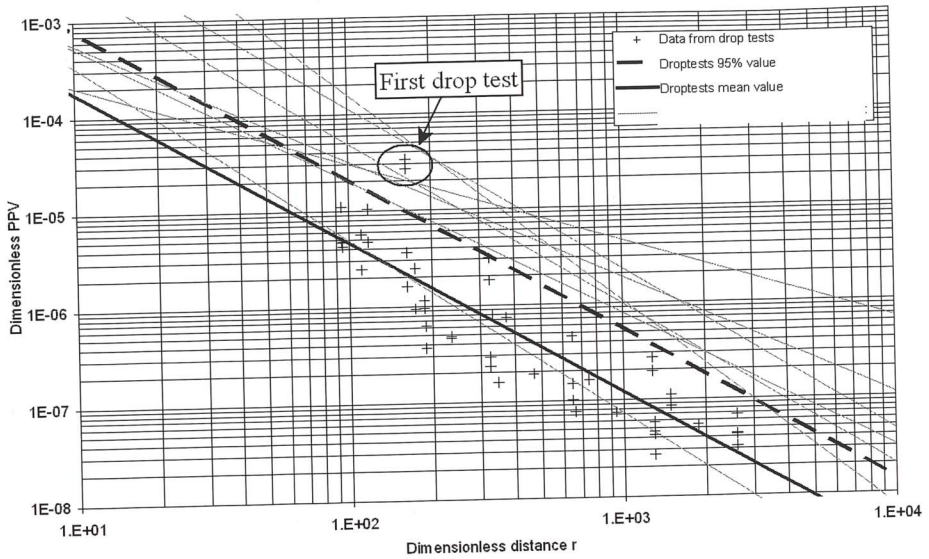


Figure 4. Results from drop tests. Reference for curves from literature is given in Figure 2.

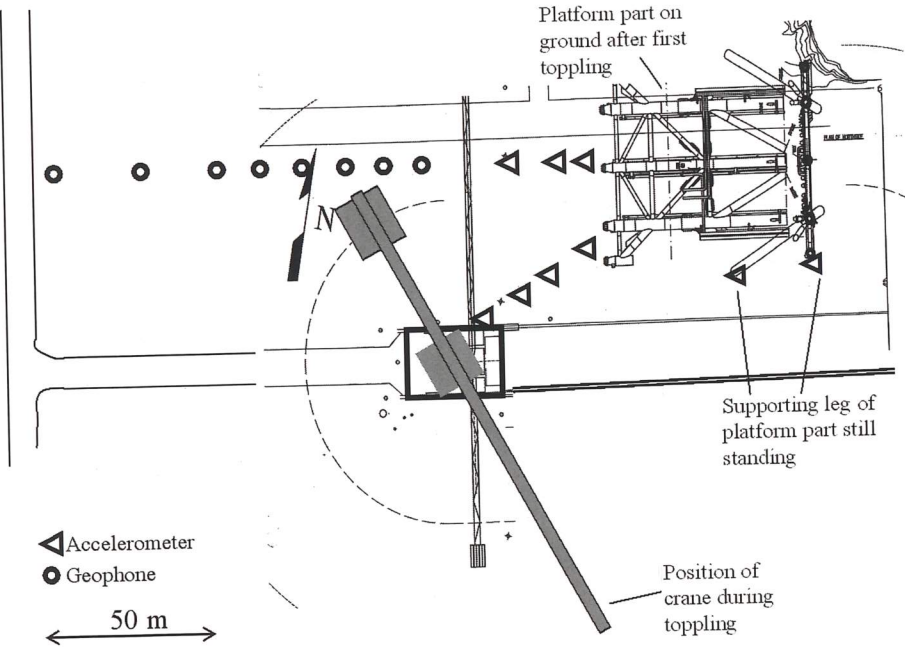


Figure 5. Sensor array for measurement of ground vibration during toppling.

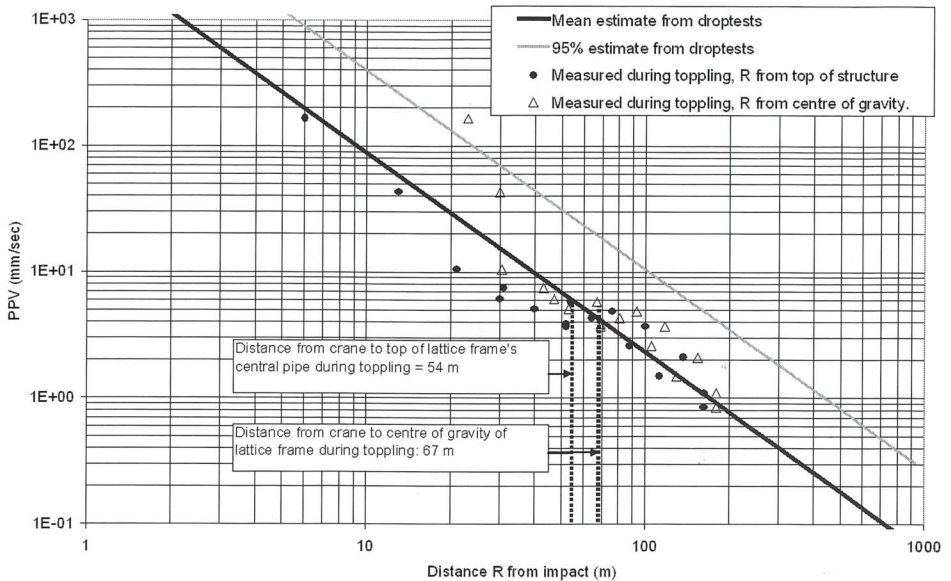


Figure 6. Ground vibration measured during toppling of lattice frame (2300 tons).

The peak particle velocities from each sensor are plotted in Figure 5 together with the mean value estimate and the 95% estimate from drop tests. For this plot, no scaling is used. Measured PPV-values are given for two different assumptions: In one case, the distance to the sensors is calculated from the global centre of gravity of the entire lattice frame. In the other case, the distance is measured from the top of the central main pipe.

The measured ground vibration generally agrees very well with the mean value prediction of ground vibrations. At small distances using the gravity centre as the reference does not seem to be appropriate. Here, using the top of the structure as reference for the point source gives a much better match with the mean value prediction. This may be related to protruding objects on the upper part of lattice frame.

No damage on constructions or buildings was reported from the toppling operations.

Stapled lines indicate position of the giant crane for the two alternatives.

5 Conclusions

Comparison of ground vibrations from published case records from different sites is possible introducing using dimensionless cube root scaling, and taking site characteristics as shear wave velocity of the ground and soil density into account. Predictions have been improved by site specific drop testing. Measurements during toppling confirmed that predictions gave reliable values for ground vibration from impact of a structure involving large energies, when based on tests involving energies less than 0.5% of the kinetic energy in the actual impacts from the toppling of the platform parts ■

Vi kan vibrationsprovning...



Nu levererar vi vibratorer från välkända Gearing & Watson. De har fokus på driftsäkerhet, storlekar från 9 N till 53 kN, alla luftkylda.

Våra PC baserade styrsystem kommer från m+p international, är i absolut världsklass med många system installerade i Skandinavien.

Vi integrerar vibratorer med glidbord, headexpanders, fixturer, accelerometrar och våra klimatkammare från Thermotron Industries till innovativa, nyckelfärdiga lösningar.

Med mer än 20 års erfarenhet och egen serviceavdelning vill vi vara din självklara partner inom miljötest.



PROXITRON
Mät & Test

Motala 0141-580 00
Stockholm 08-10 15 80
www.proxitron.se



First Announcement!

CEEES international conference

**Fatigue caused by mechanical
and corrosive stresses**

September 17, 2004

The conference takes place at
The Army Museum, Stockholm

We look forward to contributions also from
colleagues within CEEES national societies.

You are most welcome to Stockholm!

This conference, organised by SEES, is held in connection with the CEEES
assembly in Stockholm on the 16th of Sept. 2004.

SEES

**Swedish Environmental Engineering Society
The Swedish Institute for Product
Resistance Science and Technology
Dpt of Trade Service in The Association of
Swedish Engineering Industries
Box 5510, SE 114 85 Stockholm
www.sees.se**

Din komplette kalibreringsleverantör

Låg frekvens

Vår lågfrekvensvibrator har en slaglängd på upp till 15 cm.

Hög acceleration

Vi kan kalibrera accelerometrar upp till 50 000 m/s².

Hög precision

Med vår laserinterferometer kalibrerar vi referensaccelerometrar.

Impedansgivare

En impedansgivare är en kombinerad accelerometer och kraftgivare.

Komplett verifiering

Förutom givare kalibrerar vi även kompletta vibrationsmätinstrument. Vi kontrollerar då filter, tidsvägning, känslighetsomkopplare, mm.



Fullservice

SP kan kalibrera de flesta slags instrument, t ex inom områdena ljud och ljus, elektricitet, klimat, mått och vikt, mm.

Varför kalibrera hos SP?

SP är Sveriges centrum för mätteknik. Här finns landets bästa mättekniska resurser och största sakkunskap inom området. Vi har lång erfarenhet av att inventera mätidon, kravnivåer och mätbehov för bl a industrin. Vi genomför kalibrering för hela företaget.

Som riksmätplats upprätthåller SP referensnormalerna för samtliga centrala mätstorheter. Vi har följaktligen hela den spårbarhetskedja som moderna kvalitetssystem fordrar vid kalibrering av mätidon. Forskning och utveckling bedrivs kontinuerligt för att ytterligare förbättra noggrannheten.

Kurser och konferenser



Vibrationsdagen 2004

Vibrationsdagen blir 23 november 2004 i Stockholm. Plats meddelas senare: Opera terrassen eller IVA på Grev-Turegatan har föreslagits.

Temat denna gång är:

Vibrationers växelverkan med ljud och buller

Vi vill gärna få förslag på föredrag.
Kontakta Martin Almgren
(E-post: martin.almgren@ingemansson.se)
Titel och sammandrag önskas senast 1 maj 2004.

The ICSV11 Congress & Exhibition

St. Petersburg, Russia 5-8 July 2004, in at the four-star Hotel Pribaltiyskaya.
First rate scientific several Keynote Paper presentations and an extensive social programme.
Deadline for papers expired 2004-02-20.
See further <http://www.monomax.org/icsv11> or mail to
ICSV11 Congress Secretariat (temporary address) [ijav@rol.ru]

AU3-dag 2004

I samband med SVIB-dagen hösten 2004 kommer en AU3-dag att anordnas på liknande sätt som vi gjorde 2002 i Göteborg.

Temat denna gång är:

Vibrationer i byggnader

Dagen är tänkt att ägnas åt tre delområden:

- 1. Vibrationer från sprängning och annan byggverksamhet.***
- 2. Vibrationskänslig utrustning***
- 3. Vibrationsalstrande utrustning***

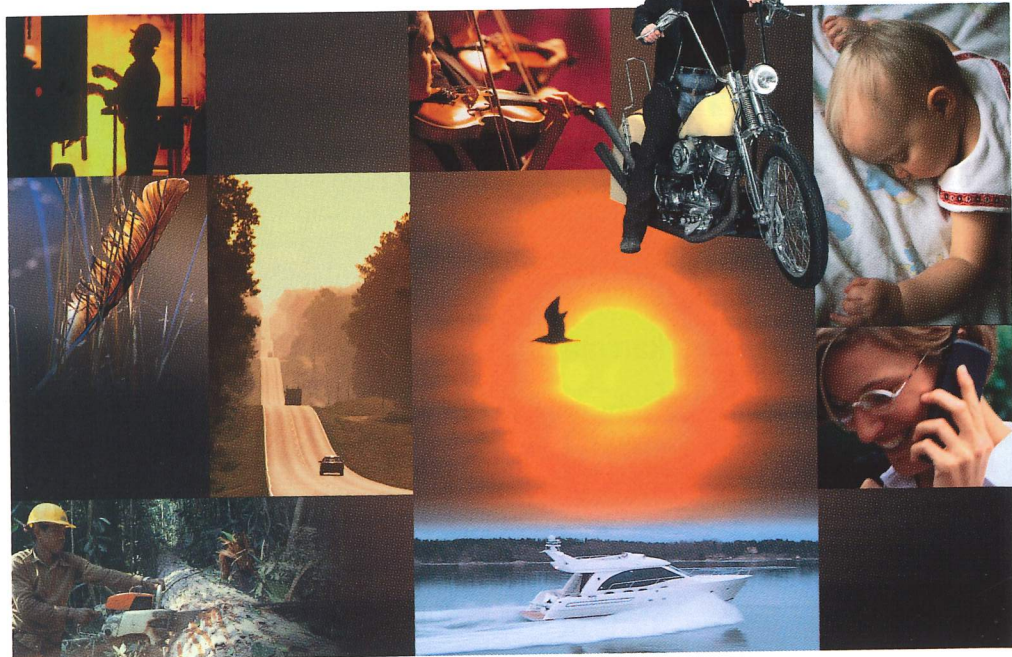
Vill ni bidra med ett föredrag,
kontakta Tomas Odebrant
E-post: tomas.odebrant@ingemansson.se

Titel och sammandrag önskas senast
30 april. 2004.

Human Response to Vibration, HRV

September 2004
Tid och plats inte bestämt,
men det blir i Englad.

Nu ska du lyssna väldigt noga!



THP Communication

Gör så här. Luta dig tillbaka och blunda. Vad hör du? Människor, en radio, ventilationssus, en industri? Du kanske känner vibrationer i huset från bilarna på gatan eller motorvägen?

Själva har vi lyssnat noga i 40 år och hört en hel del. Vårt jobb är nämligen att skapa tysta, vibrationsfria miljöer och produkter, och så bra förutsättningar som möjligt för tal, sång och musik.

Ingemansson är Europas ledande expert inom ljud och vibrationer. Vi hjälper konsulter, byggtreprenörer, industriföretag och myndigheter överallt i samhället.

Vi tar fram tysta bostäder, arbetsplatser, vägar, fordon, maskiner och offentliga miljöer som ökar allas livskvalitet.

Dessutom säljer vi mät- och analyssystem, och mjukvara för simulering. Vi erbjuder också ett komplett utbildningsprogram.

Hur låter det runt omkring dig just nu? Är det tyst och behagligt har vi antagligen varit med och utvecklat miljön i både stort och smått. Har du ett eget ljudproblem som du vill lösa, är det bara att ringa och prata med oss.

Vi är bra på att lyssna.



INGEMANSSON

LMS Pimento

The Noise & Vibration Analyzer With Portable Power



LMS Pimento

Looking for a solution for multichannel testing and signal processing, noise and vibration measurements, acoustic testing, rotating machinery and structural analysis,...? A portable solution with the power of a bigger lab system? A solution that fits-in 20 years of LMS application experience? **You've just found it ...**

LMS Pimento combines a complete noise and vibration tool box with unrivaled hardware performance:

- 4 to 24 channels in real-time
- 24-bit ADC and DAC for maximum dynamic range
- 40kHz real-time bandwidth on all channels
- Microphone and ICP signal conditioning
- AC/DC/battery powered
- Ultra portable - only 4 kg
- Compact and rugged design
- Ultra silent - no fan
- Test, analyze and report on one system
- Rotating machinery testing
- Multi-channel spectrum analyzer
- High speed throughput acquisition and processing
- Acoustic analyzer with ANSI real-time 1/3 octaves
- Structural testing, ODS, and modal analysis
- Advanced post-processing

Want to see it for yourself? Visit www.lmsintl.com for more information on **LMS Pimento**



INGEMANSSON

Ingemansson Automotive AB • Box 276, 401 24 GÖTEBORG SWEDEN

Tel +46 (0)31 774 74 00 • Fax +46 (0)31 774 74 74 • e-mail: lms@ingemansson.se



LMS
INTERNATIONAL

Empowering Engineering Innovation

Nordic Vibration Research 2004

Call for papers before 20 March
KTH in Stockholm, June 3-4
More information can be provided by AU
5's chairman,
Prof. Thomas Abrahamsson e-mail:
thab@solid.chalmers.se
Registration via the SVIB homepage:
www.svib.se

10th International Conference on Hand-Arm Vibration

June 7 -10, 2004 Las Vegas, Nevada,
USA
Kontaktperson: Douglas D Reynolds
E-mail reynolds@clark.nscce.edu
Phone: +1 702 895 3807
Fax: +1 702 895 4677



Anders Nöremark

Bilder från NGI i Oslo

Temadag om Vibrationsdirektivet

Arrangör: SVIB AU 4, Johan Granlund,
Johan.Granlund@vv.se
Arbetsmiljöverket 27 oktober



Kjell Ahlin



Christian Madshus



Ylva Vidhög



Ole Dössing



På bilden förekommer Martin Almgren, ordförande
i SVIB, Tomas Odebrant, ordörande i AU 3 och i blå
"skjorte" chefen för Sintef, Anners Hanneborg f

Nyheter från Acoutronic

Ljud & Vibrationsanalysatorer från OROS OR3x-serien, Mindre, Lättare, Kraftfullare...

Specifikationer:

- 2-32 kanaler
- 40 kHz bandbredd/kanal
- 24 bitars ADC
- Inspelning till hårddisk
- Parallell 1/n-oktavanalys, FFT & ORDER-Tracking
- ICP- & LEMO-ingångar
- Trigg- & Tachoingångar
- Signalgenerator, medger även lyssning på mätdata
- Nätverksanslutning
- Portabelt

Programvara:

- ODS
- Modalanalys
- Ljudeffekt
- Ljudintensitet
- Ljudkvalitet
- Balansering

my OROS
NOISE AND VIBRATION ANALYZERS
www.oros-signal.com



OR34, vikt 1.4 kg



OR35, vikt 2.8 kg
batteridrivet



OR38 & OR36,
vikt 8 resp. 5 kg



Acoutronic AB

Box 1180
SE-181 23 LIDINGÖ

Stockholm: Tel: 08-765 02 80

acoutronic@acoutronic.se
<http://www.acoutronic.se>

Göteborg: Tel: 031-744 34 50

Steam turbine vibration characteristics

by
Mike McGuire
Dynamics Consultant Rugby, UK.

1 Introduction

Large steam turbine rotor lines consist of a number of coupled rotors supported by fluid film bearings which are generally mounted on concrete foundations.

(Fig.1). In the past, particularly in Europe, steel foundations were also used but their use has virtually disappeared in recent years. Typically the overall length of the shaft line may be in the region of 150 feet (50 metres) with a total rotor mass of several hundred tons.

The blade heights range from less than 1 inch (25mm) in the early high pressure stages up to in excess of 3 feet (1 metre) for the low pressure, last stage blades (even longer for half speed applications).

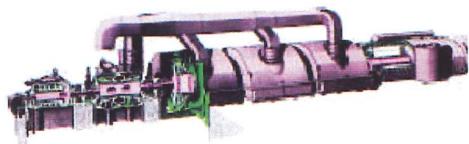


Fig.1 Shaft-line of a turbine-generator unit.

Operators of steam turbine plant require optimum efficiency without prejudice to overall plant reliability and availability. A major factor in the safe and reliable operation is to ensure that the vibration characteristics of the coupled rotor/blading system are entirely satisfactory. The avoidance of resonance conditions and the limitation of vibration are therefore a major priority when designing

rotor systems for both new and retrofit applications. ALSTOM and its predecessor companies have developed well proven dynamic analysis techniques, which are an integral part of the design process and ensure that vibration experienced at site is well within internationally recognised criteria. These techniques, which have been validated and supported by a wealth of detailed experimental data obtained from service experience and test rigs, enable new designs to be developed with total confidence. The accuracy of the methods is such that on site validation and prototype testing to confirm vibration integrity is nowadays only occasionally carried out to provide routine benchmark confirmation.

This paper provides an overview of the main factors considered and the approach adopted by ALSTOM to ensure vibrational reliability for both new and retrofit rotor systems. The following three areas are addressed:

- Lateral vibration - critical speeds, unbalance excitation & rotor running stability
- Torsional vibration - torsional frequencies, electrical disturbances & coupled shaft/blade vibration Blade vibration.
- It is not intended to cover the fundamental theory and analytical techniques but more detailed information can be found in the selected papers listed in the bibliography.

Låt oss visa vägen till rätt testutrustning

Den rätta och trygga vägen går till de ledande tillverkarnas produkter. Dessa har år efter år köpts av krävande svenska företag, av dina kollegor, sedan de gjort professionella bedömningar av *alla* aspekter.

Därför vet du att vi visar vägen till rätt testutrustning.



Elektrodynamiska vibratorer: LDS

LDS har urvalet, resurserna och det oskattbara kunnandet. Från 8,9 N till 289 kN. Combobaser och glidbord. Förstärkare. Fixturer. Vägen till rätt vibrationstest.



Klimatkammare: Vötsch

(generalagent är Axeb AB)

Kyla, värme och fukt. "Walk-in".

Vägen till rätt klimattest.



Kombinationer

Kombinera klimattest med vibrationstest.

Vägen till rätt, heltäckande test, om det är vad du behöver.



Styrsystem: Dactron

Enastående bra, lättlärt användargränssnitt. Välj mellan Sinus (S), Resonance search track och dwell, Random (R), SoR, RoR, SoRoR, classical shock, SRS, TTH. Road simulation. Vägen till rätt styrning av vibrationstest.



Datainsamling: Dactron, MSL

Dactron: Portabla FFT-analysatorer, 2 – 32 kanaler.

Analys av vibration, ljud, roterande maskiner. Modal analys.

MSL: Distribuerad datainsamling. Web-baserad övervakning.

Vägen till rätt datainsamling.

Kontakta oss för en diskussion, eller begär mer information!

Siddis AB • Box 182 • 184 22 Åkersberga

tel 08-544 019 56 • fax 08-540 212 65 • info@siddis.se • www.siddis.se

2 Lateral Vibration

The starting point for any assessment of the rotordynamic behaviour of the coupled shaft line is to set up an analytical model. For lateral vibrations the model can be subdivided into four distinct parts (Fig. 2) which can be analysed separately.

These are:

- a) the rotor.
- b) the bearing oil films.
- c) the bearing pedestals.
- d) the foundation.

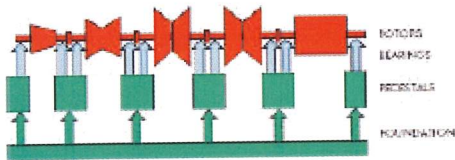


Fig.2 Schematic arrangement of rotors and supports.

The dynamic characteristics of each of these parts are analysed separately to the required degree and then combined together to determine the vibration response of the complete coupled system to the various forms of excitation which are inevitably present in service (e.g. mechanical unbalance, coupled concentricity errors, rotor bends, rotor asymmetry - for two pole generators, steam excitation forces in seals).

In practice, the rotor and oil films are the most important elements of the model. Only in exceptional cases does the influence of the bearing pedestals and foundation significantly affect the overall rotordynamic characteristics. By adhering to established design guidelines, it is normally possible for the tur-

bine designer to design a specific rotor system that will operate successfully when installed on the different foundation types which are commonly used.

The converse of this is that, only in exceptional circumstances, is it possible to make significant changes to the rotordynamic behaviour by changing the dynamic characteristics of the foundation. However, it is important to recognise that, although the dynamic characteristics of the foundation may not be significant, there are circumstances where the static characteristics of the foundation can have a greater influence on the dynamic behaviour of the shaft line, particularly for those cases where two bearings per rotor are employed. For example, it is inevitable that some deflection of the foundation may occur as a consequence of changes in foundation loading due to vacuum loads, friction of sliding surfaces, pipe thrusts etc. The potential impact of these effects on for example; bearing loading, critical speeds or seal clearances is therefore a factor which has to be considered. These effects are minimised by adopting the concept of a single bearing between rotors.

2.1 Rotors

The rotor is characterised by its inertia and stiffness matrices, which are calculated using techniques, which are well established throughout the industry. These take into account the detail variations of mass distribution and stiffness changes along the rotor line, any additional coupling flexibility (e.g. the influence of clutches for some combined cycle applications) and the influence of temperature on material properties. In the past special care had to be taken to replicate correctly the stiffening effect of shrunk on discs together with their manufacturing tolerances but ALSTOM have long since abandoned this form of design in favour of monobloc and welded rotors which offer superior mechanical integrity (Fig.3).

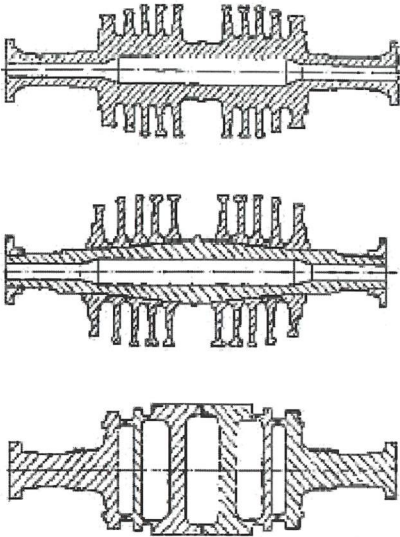


Fig.3 Different rotor constructions.

The bearing span of the rotor is the most significant parameter which the rotordynamic designer can use to influence the running behaviour. For new machines the choice of the optimum bearing span is therefore an important part of the design process. However, for retrofit applications this parameter is already fixed and this provides an additional challenge to the designer.

For lateral vibrations material damping is ignored because for most practical purposes the vibratory stress is sufficiently low to render it insignificant, particularly when compared with the damping provided by the bearing oil films.

2.2 Bearing Oil Films

The journal bearing oil films are important elements of

the coupled system and their influence is a major

contributory factor in controlling the vibratory response of the rotor line. The dynamic characteristics of the oil films are characterised by the well known four stiffness and four damping coefficients. They are anisotropic with cross coupling between horizontal and vertical motion that results in non circular shaft orbits. Most important, the oil films provide substantial damping which contributes to the relatively modest dynamic magnification when passing through critical speeds.

However, under certain conditions (e.g. if the bearing is lightly loaded) they can induce unstable, self excited vibrations.

The actual values of the coefficients are strongly influenced by the bearing geometry (length, diameter, bearing clearance - or to be more specific the difference in radius of curvature between the shaft and the bearing), rotational speed, lubricant properties, bearing type (e.g. fixed arc, tilting pad, number of arcs/pads, degree of pre-load) and bearing load. In service changes such as foundation movements may introduce misalignment that, particularly for shaft lines employing two bearings per rotor, can influence the bearing characteristics and hence the overall dynamic behaviour of the rotor line.

ALSTOM has been at the forefront of experimental and analytical research carried out over many years to identify accurately the bearing coefficients for different bearing types and a database of the dynamic properties of different types of bearing has been established. In addition, a full thermo-hydrodynamic bearing analysis method has been developed which has been validated using test

rigs (Fig. 4) capable of testing full size bearings over a wide range of operating conditions. This was particularly important in demonstrating for large bearings the influence of local thermal distortions of the bearing profile on the dynamic characteristics.

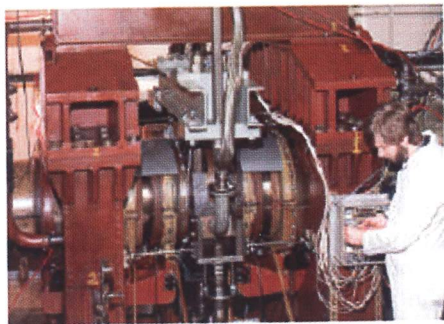


Fig.4 Dynamic test rig for 20 inch bearings.

The bearing analysis method is an important design tool that is used to calculate the contract specific bearing characteristics both for ALSTOM bearing designs and those supplied by others. This is particularly relevant for third party retrofit applications where there is often a preference to retain the original bearings without change but for which accurate bearing coefficients may not be available. However, in such cases care must be taken to ensure that all features of the bearing are correctly modelled and that there are no physical constraints which prevent correct functionality. For example with tilting pad bearings it is important to ensure that the influence of the pad pivot is fully taken into account.

2.3 Bearing Pedestals

In most cases bearing pedestals are stiff relative to the oil film and therefore the rotor-dynamic behaviour is less sensitive to the pedestal dynamic characteristics. It is normal therefore to represent the pedestal as simple horizontal and vertical mass spring systems.

In those cases where the bearings are an integral part of the turbine frame/foundation the pedestal characteristics are modelled together with the foundation.

2.4 Foundation

In the past detailed mass elastic studies were carried out to produce sophisticated analytical models of complex foundation structures from which the cross coupled foundation impedances at the bearings could be determined. Such analyses predict large numbers of natural frequencies below and in the operating speed range, which the subsequent response analysis showed were insensitive to excitation from the rotor system. Further studies led to the realisation that the influence on the rotor behaviour of the complex structural modes predicted by the mass elastic analysis were strongly inhibited by the influence of the large inertia and damping (due for example, to friction at sliding and bolted interfaces and internal hysteresis) inherent in the foundation. It was concluded that much simpler representations of the foundation support characteristics were entirely satisfactory for the accurate prediction of rotor vibration behaviour. This clearly facilitates the analysis process and is particularly advantageous for third party retrofit applications in that it is only necessary to have relatively simple details of the foundation in order to predict the rotor vibration characteristics.

2.5 Design Objectives

It is inevitable that there are inherent steady state forces present for all rotor lines that will excite vibration. For steam turbines the most common excitation source is residual unbalance which is minimised as far as possible by the adoption of systematic assembly processes prior to the application of high speed modal balancing of each individual rotor (Fig. 5).



In general, the quality of factory balance achieved and the adoption of careful assembly processes when coupling individual rotors together at site is such that vibration levels within zone A of ISO 7919/2 and ISO 10816/2 (see Tables 1, 2 & 3) are readily achieved without the need for in situ balancing.

Zone	Shaft Rotational Speed (rpm)			
	1500	1800	3000	3600
	Peak to peak relative displacement of shaft (μm)			
Newly commissioned machines	100	90	80	75
Unrestricted long term operation	200	185	165	150
Unsatisfactory for long term operation (Remedial action required)	320	290	260	240

Table 1: Recommended Values for maximum relative shaft displacement for steam turbines and generators

Zone	Shaft Rotational Speed (rpm)			
	1500	1800	3000	3600
	Peak to peak absolute displacement of shaft (μm)			
Newly commissioned machines	120	110	100	90
Unrestricted long term operation	240	220	200	180
Unsatisfactory for long term operation (Remedial action required)	385	350	320	290

Table 2: Recommended Values for Maximum Absolute Shaft Displacement for Steam Turbines and Generators

Zone	Shaft Rotational Speed (rpm)	
	1500/1800	3000/3600
	R.m.s. vibration velocity of bearing (mm/sec)	
Newly commissioned machines	2.8	3.8
Unrestricted long term operation	5.3	7.5
Unsatisfactory for long term operation (Remedial action required)	8.5	11.8

Table 3: Recommended Values for Bearing Housing/ Pedestal Vibration Velocity for Steam Turbines and Generators

The first objective of the rotordynamic process is to design a shaft line which operates satisfactorily with low steady state vibration levels at rated speed, is relatively insensitive to changes in operational parameters and has satisfactory margins between rated speed and any critical speeds.

Steady state synchronous (one per rev) vibration excited by unbalance is the main source of vibration experienced on large steam turbines. However, it is not the only source and a particularly important area of concern is the potential for self excited, sub synchronous rotor instability which if allowed to develop fully can result in significant load reductions or, in the worst case, tripping of the unit. The second objective therefore is to de-

sign stable rotor systems. This is discussed further in the next section.

The analysis models described above are the cornerstone on which successful designs are achieved to meet the declared objectives.

2.6 Shaft Line Stability

Sub synchronous, unstable vibrations may be encountered in service due to the inherent instability of the hydrodynamic journal bearings (classic 'oil whirl'). Alternatively, the overall damping provided by the rotor/bearing system may be insufficient to contain the effect of external destabilising forces which are generated as a consequence of steam flow variations within the turbine (often referred to as 'steam whirl'). Unlike rotor unbalance this is a transient vibration, which locks on to a fundamental, rotor system natural frequency - normally the first critical speed of the relevant rotor. In the worst case such vibrations increase rapidly and may cause the vibration trip limits to be exceeded. For other cases, the non-linear nature of the bearing oil films results in increased damping as the vibration amplitude increases and a 'pseudo unstable' limit cycle behaviour is observed. Such behaviour often provides an indication that the stability limit is being approached and provides a warning that remedial actions may need to be initiated.

'Steam whirl' arises due to the creation of dynamic forces in steam sealing passages at blade tips and rotor hubs. These forces are generated by lateral movements of the rotor within the steam seals, which act at right angles to the rotor displacement. They are represented in the analysis model as skew symmetric stiffness coefficients which are inherently destabilising but are offset by additional direct stiffness and damping terms. The combined effect of these terms coupled with the bearing damping determines whether the

overall system will be stable or unstable.

The magnitude of the 'steam whirl' excitation is proportional to steam flow, dependent on the detail geometry of the seals and is strongly influenced by both the lateral movement of the shaft (the 'Thomas' effect) and the circumferential ('swirl') component of the leakage flow. Its effects are most significant in high pressure (HP) cylinders where the ratio of steam density to rotor mass is greatest. HP rotors of high power supercritical units are therefore potentially the most vulnerable. Conversely this phenomenon is less significant for nuclear units (except AGR's) and combined cycle plants.

There are a number of ways of improving the stability margin including:

- avoidance of unnecessarily low rotor flexibility (i.e. increased critical speed). avoidance of lightly loaded bearings.
- adoption of alternative bearing designs (e.g. Pocket

bearings, multi arc fixed pad bearings, tilting pads etc.)

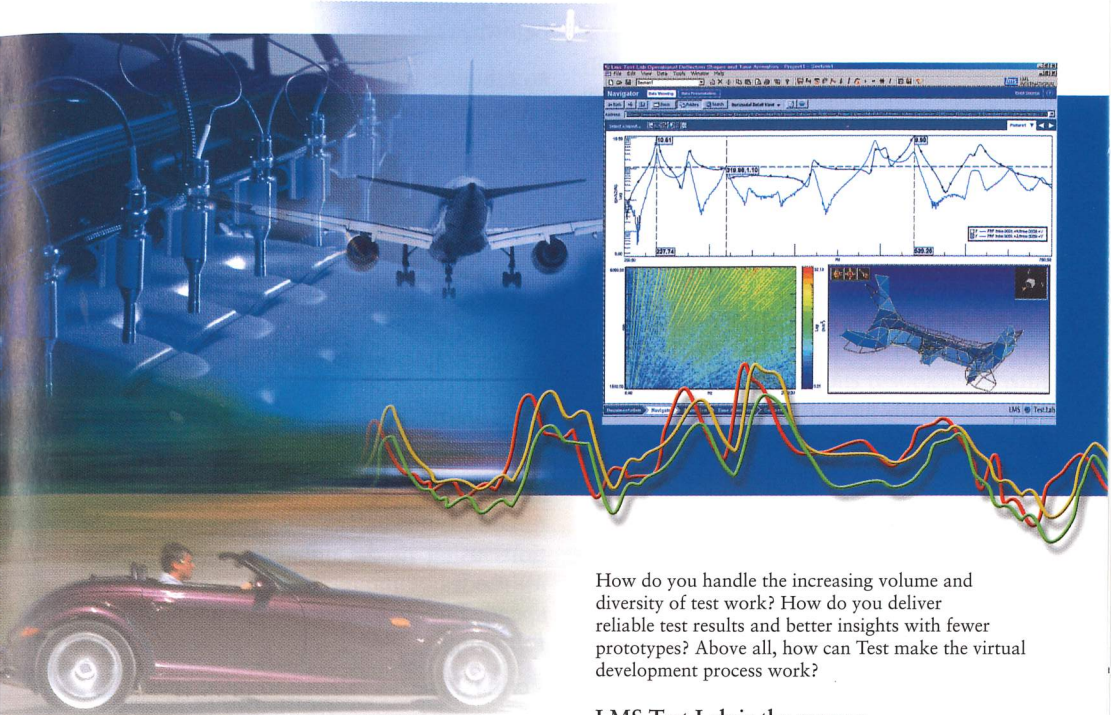
With improved damping capacity and/or reduced sensitivity to external load changes.

- optimised seal geometry.
- reduction of inlet swirl to seals.

ALSTOM's first significant experience of 'steam whirl' in the UK was in the early 1970's on a class of 500MW units that had a flexible HP rotor. A detailed collaborative exercise was carried out in conjunction with the national utility (CEGB) which included experimental investigations of seal and bearing characteristics. Remedial actions involved seal and bearing modifications and the introduction of swirl breaks (Fig. 6) to reduce the magnitude of the circumferential flow component entry to the seals. Since that time

LMS Test.Lab

The Revolution for the Integrated Test Laboratory



How do you handle the increasing volume and diversity of test work? How do you deliver reliable test results and better insights with fewer prototypes? Above all, how can Test make the virtual development process work?

LMS Test.Lab is the answer.

LMS Test.Lab is a complete and integrated solution for test-based noise and vibration, durability and environmental engineering. It builds upon the blistering performance of the LMS SCADAS III front-end and incorporates over 20 years of experience.

Test.Lab makes testing easier and more productive – without compromising data confidence. It gives better insights through measurement and analysis innovation. Insights that drive the complete development process, from target setting, to making virtual simulation work.

Visit www.lmsintl.com to request a free LMS Test.Lab CD.



INGEMANSSON

Ingemansson Automotive AB • Box 276, 401 24 GÖTEBORG SWEDEN
Tel +46 (0)31 774 74 00 • Fax +46 (0)31 774 74 74 • e-mail: lms@ingemansson.se

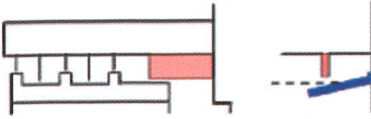


LMS
INTERNATIONAL

Empowering Engineering Innovation

experience from different parts of the Company has been consolidated together with service experience feedback to develop and refine the advanced aerodynamic methods which are used to provide a basis for the systematic assessment of steam excitation forces for specific seal geometries. The assessment of steam excitation is routinely carried out as part of the rotor design process.

Swirl breaks at tip-seals...



... and in diaphragm gland segments.



Fig.6 Swirl breaks at a tip-seal and in a diaphragm gland.

2.7 Lateral Vibration Design Process

ALSTOM have amassed substantial experience over many years in understanding rotordynamic vibration behaviour and resolving associated service problems.

Swirl breaks at tip-seals...

This has led to the adoption of a standard process for the dynamic design of large steam turbine shaft lines which applies generally to both new and retrofit applications - the only difference is that for third party retrofits more conservative assumptions are adopted

in some areas to compensate for any uncertainty in the detail description of other parts of the system. This process for full speed units is summarised in the guidelines listed below:

- set up shaft line mass inertia and stiffness matrices using standard method.
- determine bearing oil film stiffness and damping coefficients for each bearing (note: for third party retrofit applications in which the original bearings are to be reused make an assessment of any special features which may need additional attention).
- use standard bearing pedestal and foundation support impedance values - subject to there being no significant features which might require a more detailed analysis to be performed.
- carry out standard rotor response calculations using modal unbalance sets to excite all modes of interest up to maximum over-speed. Targets for acceptance are set for the following:

- critical speeds relative to rated speed.
- modal response at critical speeds.
- modal response at rated speed.

- for third party retrofit applications it is common to target a first critical speed, close to or slightly higher than that of the rotor which is being replaced.

Variations from this are only permitted after confirmation by more detailed analysis.

- perform initial stability analysis with steam excitation at centre of rotor to enable the basic stability to be assessed before proceeding with detailed design.

In addition, for HP and combined HP/IP rotors:

- calculate steam excitation coefficients for all shaft and blade seals.
- complete audit of seal and bearing damping and assess overall impact on stability.
- carry out complex eigenvalue calculation for complete coupled shaft line. Results require minimum stability margins to be achieved at overspeed (i.e. avoidance of oil whirl), at full load with standard seals and at full load when all stabilising features have been included.
- for third party retrofit applications more conservative margins are set to compensate for any uncertainty about bearing supports and adjacent rotors.

The adoption of this process ensures that the vibration characteristics of the shaft line are entirely satisfactory for unrestricted long term operation throughout the lifetime of the turbine. Its effectiveness is amply demonstrated by well established operational feedback from a wide range of both new and retrofit applications.

3 Torsional Vibration

Torsional vibrations can be excited in large steam turbine shaft trains by electrical disturbances occurring in the active part of the generator. The relevant excitation frequencies correspond to the first and second harmonic of the electrical grid system frequency (note: for half speed turbines with four pole generators the system frequency is twice the rotational frequency).

Torsional vibrations are not coupled to either the turbine casings or lateral shaft vibration, hence there is no transmission of external forces which give a physical indication to operators that such vibration is present.

Torsional excitation can be subdivided into:

- short term disturbances (fractions of a second) at 1 and 2 times the system frequency caused by short circuits (in the electrical grid, transformer or generator stator windings) or synchronisation failure. The magnitude of the disturbance torques produced depends on the electrical properties of the electrical system and in the worst case, which is usually a line to line fault at the generator terminals, may be in the range of 6 to 8 times the normal torque of the unit.
- low, long term electrical disturbance torques. These are negative sequence current at 2 times the electrical system frequency due to unbalanced loading of the three electrical phases or subsynchronous resonance arising from interactions with long transmission lines which may excite the lower torsional modes.

In all cases the applied excitation acts uniformly along the active length of the generator and therefore the magnitude of the effective excitation depends on the modal shape of the generator rotor. For example, maximum excitation will occur if all parts of the generator are in phase. However, if the mode shape involves antiphase motion with a node at or close to the centre of the generator there will be substantial energy cancellation (Fig. 7) and the influence of the electrical disturbances will be significantly reduced - this is often the situation for full speed (3000/3600 rpm) shaft lines.

As for lateral vibrations the starting point is to model the torsional characteristics of the shaft line but in this case there is no influence whatsoever from the bearings or stationary parts.

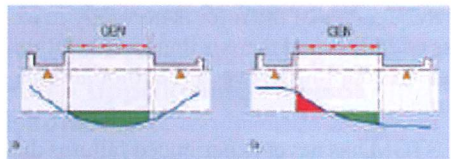


Fig.7 Comparison of the effective excitation of different torsional mode-shapes.

3.1 Rotor Model

The characterisation of the rotor model is carried out using similar assumptions to those made for lateral vibration. As stated above there is no influence from the bearings or stationary parts and therefore the potential for uncertainty in the calculated results is much reduced. The only damping is that provided by the shaft material which is low ($\sim 0.1\%$ of the critical damping ratio for the modes of interest) at the level of torsional dynamic stresses experienced under normal operation. It should be noted that as local stresses increase under fault conditions such damping increases substantially. No advantage is taken of this in the assessment of torsional integrity which are therefore inherently conservative.

3.2 Influence of Blades

The fundamental natural frequency of most blade rows is such that they can be considered as rigid relative to the torsional frequencies of interest and it is sufficient to model them only by their additional torsional inertia. However, this is not the case for the long last (L-0) and penultimate (L-1) blade rows in the low pressure cylinder and they are represented as branched systems which fully replicate the 'all in phase' (zero nodal diameter) modes of these blades - often referred to as umbrella modes.

The coupling of the rotor and blades introduces additional modes in which the blades vibrate in phase and antiphase with the rotor and which, under adverse conditions, could result in unacceptably high dynamic stresses in the blades and their roots. For 'stiff' welded drum rotors the torsional natural frequencies of the rotor is normally in excess of 200Hz and there is therefore relatively small coupling between the rotor and blade modes.

ALSTOM has never experienced failures due

to this phenomenon but there are a number of examples in the industry where catastrophic failures have occurred and great care is therefore taken to avoid this risk.

3.3 Design Objectives

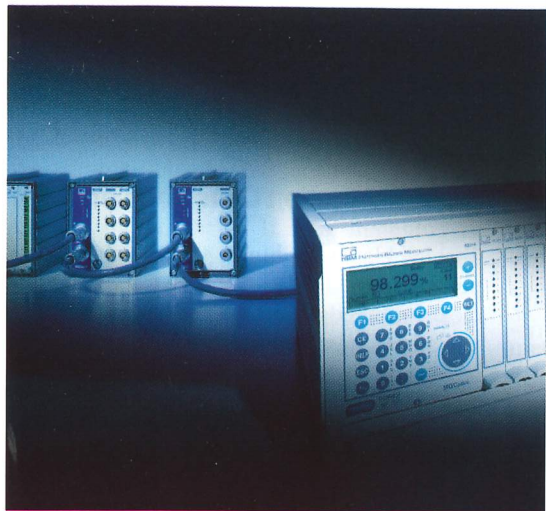
The absence of significant damping under normal conditions in the torsional system is such that it is not possible to rely on attenuation of the vibration peaks to avoid unacceptable dynamic stresses. Therefore, in order to eliminate any risk, ALSTOM has adopted a simple but safe design philosophy in which the torsional natural frequencies must be clear of both 1x (50/60 Hz) and 2x (100/120 Hz) of the electrical system frequency.

Generally, the variations of the grid system frequency are such that a relatively small margin is more than adequate to avoid any potential problems. However, although there are few uncertainties associated with torsional modelling, it is inevitable that there may be differences between calculated and actual values. For this reason a conservative approach is adopted and the design criterion requires a larger margin thereby rendering unnecessary the need for any experimental confirmation of the torsional frequencies.

By ensuring that such frequency margins are achieved, the transient stresses during short term disturbances or the steady state stresses in long blades due to negative sequence currents remain within normal acceptance standards without having to carry out more detailed response calculations.

3.4 Experimental Validation of Torsional Frequencies

As stated above the accuracy of the torsional frequency analysis methods are such that, provided the design margins are satisfied, there is no practical requirement to carry out validation tests on the complete rotor line in situ. However, some utilities may request



www.hbm.com

MGCplus ...

Data Acquisition Systems

... the ultimate modular, scalable and configurable system. Any component can be combined to find individual solutions that fit specific applications and, at the same time, ensure the reliability and price advantages provided by serial products. The spectrum ranges from stand-alone measurement devices through complex, PC-controlled measurement systems with several thousand measurement channels.

One platform for many applications:

- Research and Development
- Test Rigs
- Calibration
- Manufacturing
- Weighing
- Experimental Stress Analysis
- Mobile Data Acquisition

METROLINK

Metrolink Scandinavia AB • Reprovägen 6, 183 77 TÄBY
Tel 08 756 23 33 • Fax 08 756 23 47
Email info@metrolink.se • www.metrolink.se



measurement with confidence

Industri Accelerometer Prisvärda Livstids garanti



Rondsystem Totalnivå Lagerkondition

Komplett



Från SEK 28.000:- med program!

**Allt inom maskinvibrationer! Utbildning • Övervakning med FFT •
Balansering • Vibrometrar • Konsult • Uppriktning • Felsökning
och Tillståndskontroll.**



Vibrationsteknik AB • Taborsbergsvägen 20, 602 13 Norrköping
Tel. 011-16 62 10 • Fax 011-16 62 20 • URL www.vtab.se • Email sales@vtab.se

verification and ALSTOM has experience of different techniques for doing this. The two most commonly used methods are:

- a) sensitive strain gauges installed at the shaft journals to measure torsional strains (Fig. 8).
- b) toothed wheels to measure the torsional oscillation.

Both have been successfully used in the past but the strain gauge approach is now the preferred method.

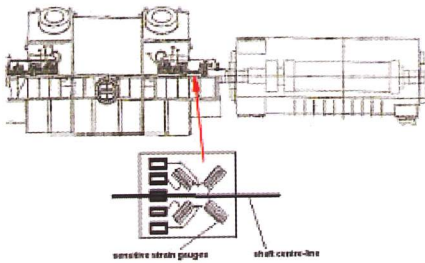


Fig.8 Torsion measurement using strain gauges.

The sensitivity is such that by careful selection of the measurement planes and with the use of appropriate signal conditioning techniques, it is possible to excite all frequencies of interest by the minor random electrical network disturbances, without impact on the normal operation of the power plant.

An alternative to carrying out measurements at site is to perform tests on a bladed rotor in the spin pit prior to dispatch to site. This approach is particularly advantageous for the identification of coupled blade modes directly by means of a non contacting optical measurement system or strain gauges mounted on the blades. The results obtained are compared with the equivalent calculation for the same rotor configuration and provide an accurate benchmark for the calculation and the coupled rotor system. The benefit of this from the customer's viewpoint is that it provides

confirmation of the calculation and the important frequencies prior to installation and avoids the need for further torsional testing at site.

4 Blade Vibration Considerations

Turbine blades are highly tuned, lightly damped components and it is necessary to have a clear knowledge and understanding of their vibration characteristics. The consequence of fatigue damage occurring during the life-time of the plant are serious and ALSTOM has invested considerable analytical and experimental effort into ensuring that the vibration characteristics are optimised and the integrity of the blading is assured. The success of these efforts is demonstrated by the relatively insignificant number of blading failures in modern turbines that can be attributed to fundamental blade vibration as the root cause. This has been achieved by building on long established experience feedback and a programme of continuous improvement of prediction and experimental techniques. The present design methods provide a consistent and reliable basis for ensuring that all modes of potentially damaging vibration are carefully tuned clear of the operating range and that the dynamic stress levels are within acceptable limits for unrestricted long term operation.

4.1 Excitation of Blade Vibration

The presence of natural frequencies in any structure is only of consequence if there are excitation sources available to feed energy into the corresponding mode of vibration. For steam turbine blades there are three main sources of excitation:

- (i) The steam flow which, for each blade row, can be considered as a series of steady state jets. It can be shown by Fourier analysis that these steady forces, when coupled with the rotation of the turbine, produce sinusoidal

excitation components at exact multiples (engine orders) of the shaft rotational speed. For such excitation to be of concern it is necessary to have coincidence between both the excitation and blade frequencies AND matching of the excitation force pattern and the relevant modal shape. If such matching is not present then energy cancellation occurs and it is not possible to excite the relevant mode of vibration even if a resonant condition exists. This is the single most important factor in avoiding damaging vibration of turbine blade rows in service.

(ii) Vibration transmitted from the rotor system that could feed energy into certain modes of vibration. This can be due either to torsional or lateral vibration of the shaft line. This is particularly relevant for torsional vibration as discussed in more detail in the preceding sections. However, the control of shaft and bearing vibration is such that the levels of stress induced in the blades as a consequence of lateral shaft vibration is negligibly small. Nevertheless, care is taken to avoid any such potential resonance condition occurring.

(iii) Non engine order, random excitation ('buffeting') produced by unsteady aerodynamic effects. Such forces are generally only of significance for last LP stages operating under off design conditions when the normal orderly flow patterns have broken down. This is particularly relevant if the turbine is operated for extended periods with high back pressures or low flow. This is a non resonant, transient excitation of the blading, which causes the blades to vibrate at their natural frequencies. This situation cannot be avoided by tuning and it is necessary to ensure that the blades are capable of withstanding the excitation forces. For this reason the safe operation of long LP blades is protected by a back pressure limitation diagram together with related 'alarm' and 'trip' values which is specific to each turbine generator.

In limiting cases, where extended operation with elevated back pressure is required more robust blades have to be adopted. All of the above have to be considered but the low engine order excitations are of the greatest importance. They arise as a result of the unavoidable non-uniformity in the flow and pressure distribution due to the wakes from the stationary blade (and to a much lesser degree the moving blades) trailing edges, geometric variations within normal manufacturing tolerances, the effects of discontinuities such as half joints and bled steam extraction points, and any other effects which the shaft rotation will convert into cyclic excitation forces.

In general, the single most powerful excitation is that due to the wakes of the stationary blades which produce a dominant excitation force at the blade passing frequency (i.e. number of fixed blades x shaft rotational frequency which is often referred to as nozzle passing frequency) and at harmonics thereof, albeit with lower excitation levels. The magnitude of the nozzle passing frequency excitation reduces as the stage reaction increases. The excitation strength at other engine orders is generally less severe. (Fig. 9).

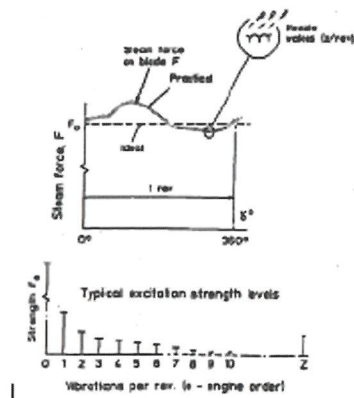


Fig.9 Wake excitation of moving blades.

4.2 Design Objectives

In the previous section some of the background principles have been discussed which, when applied correctly, enable an assessment to be made of the potential for excitation of blade vibration in service. In practice the blade designer embarks on an integrated design process to achieve an optimum compromise between aerodynamic efficiency and mechanical integrity.

All stages are designed to ensure that the natural frequencies of practical importance are clear of resonance with the significant low order excitation frequencies. It is therefore general practice to tune blades so that there are no resonant frequencies, which could excite damaging vibration at or close to rated speed by any of the first eight engine orders. In addition, dependent on the stage loading, tuning is also carried out for any relevant modes that could be resonant at nozzle passing frequency. Other modes are generally not significant but it is important to ensure that there are no particular features (for example, stay bars in the exhaust space) which could cause enhanced excitation of other engine orders.

A key prerequisite is to maintain circumferential symmetry and thereby eliminate any mistuning effects for each blade row. This is achieved on early stages by the use of pretwisted integral shrouds (Fig. 10) to provide a continuous blade to blade interconnection and, for penultimate and last stages, by adopting either free standing blades or a combination of integral shrouds and/or part height snubbers. In the past when riveted shrouds were used the blades were assembled as discrete packets.

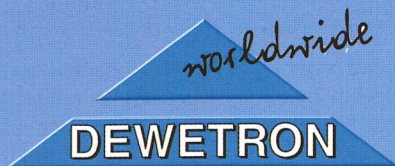


Fig.10 Turbine with integrally shrouded blades.

The use of integral shrouds eliminates packets and therefore there is no risk of exciting packet vibration modes.

Additional measures have to be taken for the first HP stages of turbines that are required to operate under the more onerous conditions associated with partial arc admission. In such cases both the steady and dynamic loading when operating at part load with some nozzle sectors closed is substantially greater than for a full admission turbine. Furthermore, the number of potential excitation frequencies is increased so that in general tuning alone is not sufficient to avoid potentially damaging vibration. More detailed response calculations are therefore carried out to evaluate the dynamic stress levels for such stages. Factors that are taken into account include the magnitude and circumferential variation of the steady steam forces due to the distribution of the open and closed nozzle sectors, the relevant modal shapes and the damping. The consequence of this is that significantly more robust blades are required than would be necessary if full arc admission were used.

For last stage blades the area of most difficulty is to determine the resistance to 'buffeting' excitation when operating under abnormal flow conditions. The application of ad-



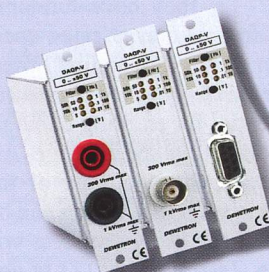
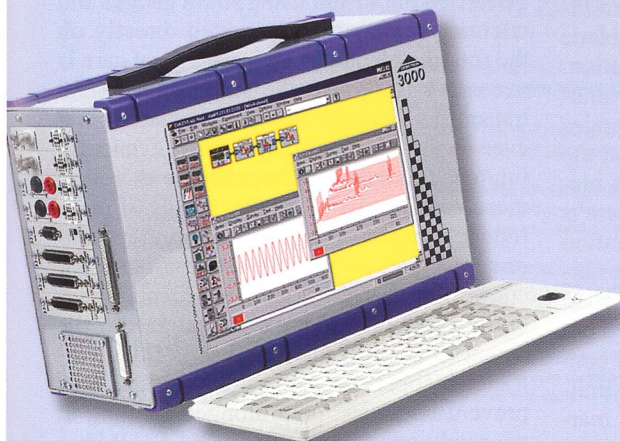
DEWE-DSA

Dynamic Signal Analyzer

Det expanderbara vibrationsmätssystemet

- 16-64 samtidigt samplande ingångar
- 200 kHz 24-bitars A/D-omvandlare
- 110 dB dynamik
- Inbyggda anti-vikningsfilter
- Menystyrd mjukvara med:

- FFT
- Oktavband
- FRF
- Orbit m.m.



**SYSTEM
TECHNOLOGY**
SWEDEN AB



www.systemtech.se
e-post: svib@systemtech.se

Box 304 • 581 02 LINKÖPING • Tel: 013-35 70 30 • Fax: 013-14 05 30

vanced unsteady CFD aerodynamic methods have advanced significantly in recent years but it is still not possible to predict reliably the excitation forces under such conditions. Empirical design rules based on past experience and the adoption of new designs, which have evolved from those which have demonstrated satisfactory behaviour, provide a basis for avoiding gross deficiencies but there is, as yet, no substitute for experimental verification.

4.3 Experimental Validation

In the past extensive experimental validation was carried out for all aspects of blade vibration. However, the advances in analytical methods have reached a state of sophistication such that routine blade vibration testing is no longer necessary for all but the last two LP stages. Testing of the early stages, which are customised for each individual application, is now only carried out occasionally to provide a benchmark for the analysis tools.

In general, unlike the early stages, the last two LP stages are developed as standard families. The most appropriate standard blade is then selected for each specific application. These families are designated as 'flexible' blades - i.e. they have fundamental modes whose frequencies at rated speed are in the domain of the low engine orders. Such blades are carefully tuned to ensure that there is no potential for exciting damaging vibration in the operating speed range. As for the earlier stages, the accuracy of the analytical models is such that tuning can be carried out as part of the design process with total confidence. Testing (Fig. 11) is generally limited to confirmation of the operational margins on the first production rotor during the final balancing runs in the factory. For such blades, once the design has been finalised there is no further customisation for individual contracts and therefore further testing for subsequent contracts is not required.



Figure 11 Measured stress ranges at different loads

The behaviour of last stage blades operating under 'buffeting' conditions is not so well defined and there is therefore a greater reliance on experimental testing to establish the magnitude of the dynamic stresses and to define practical but safe operational envelopes. Such measurements can, in principle, be carried out in situ using strain gauges and telemetry transmitters mounted directly on the rotating elements. However, great care is necessary to protect the strain gauges and transmitters from the harsh environment and even so the life of such systems is extremely limited. In situ measurements are therefore more usually carried out by means of more robust, non contacting, tip vibration measurement systems. These provide a basis for confirming that there is no unacceptable vibration within the operating range but the extent of the test programmes is often restricted by operational constraints which may prevent operation or prolonged running at the conditions of most interest. ALSTOM has therefore focussed such testing on multi-stage, model turbines (Fig. 12) which by correct scaling simulate both the aerodynamic and mechanical properties of the turbine whilst not suffering from any of the constraints present in the power station.

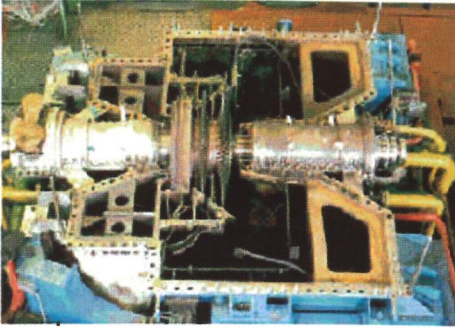


Fig. 12 Model steam turbine.

5 Summary/Concluding Remarks

The development of proven dynamic analysis techniques and their integration within the design process has ensured that signifi-

cant vibration related problems of modern steam turbine shaft lines and coupled blade systems are rare. This achievement is based on a clear understanding of the physical concepts together with the application of past experience, advanced calculation methods and experimental test programmes. The continued use and further development of these techniques will ensure that the ever increasing challenge to optimise power density and efficiency with reduced first and lifetime costs for both new and retrofit equipment will continue to be achieved without prejudice to overall plant reliability and availability. ■

Behöver din verktygslåda förnyas?

INSUL är ett program för beräkning av ljudisolering hos:

- Enkeltväggar
- Dubbelväggar med många typer av stommar
- Bjälklag med eller utan nedpendlade undertak
- Tvåglasfönster

Beräkningarna är baserade på analytiska beräkningar som bara behöver enkla indata för konstruktionen. Programmet kan beräkna reduktionstalet i tersband samt det vägda reduktionstalet, R_w och $R_{w+C_{50-500}}$ samt reduktionstalet mot trafikbuller, R_{Atr} . Valfri tjocklek och typ av isoleringsmaterial i kaviteten. Öppen databas för inläggning av egna material.

Olika typer av stommar kan hanteras, t ex trästommar, stålstommar, separerade och saxade stommar samt stommar med akustikregler eller vibrationsisolering. Bjälklag med undertak med olika typer av infästningar, t ex nedpendlade punktfästningar, kan beräknas. Valfri glastyper hanteras.

ENC är ett program i 7 moduler uppbyggt av en mängd delprogram som behandlar:

- Nivåer och bedömningsgrunder
- Ljudeffekter och ljudutbredning utomhus
- Rumsakustik och ljudabsorption
- Ljudisolering, huvar och skärmar
- Ljuddämpare, ljud i kanaler och breakout ljud
- Vibrationsisolering och dämpning
- Omfattande semiempirisk databas för uppskattning av ljudeffekter hos källor.

INSUL och **ENC** är till stor hjälp för akustiker, produktutvecklare, arkitekter och konstruktörer som snabbt vill ta fram konstruktioner som uppfyller ställda ljudkrav.

WSP Akustik ingår i WSP Group med ca 80 akustiker i koncernen. Det gör akustikerkraften inom WSP till en av Europas största aktörer i branschen. I Sverige är vi ett 40-tal medarbetare med över 40 års erfarenhet av specialisttjänster

inom områdena akustik, buller och vibrationer. Vi har välutrustade ljudlab och programvaror som täcker de flesta behov inom alla branscher.

WSP Akustik har kontor i Stockholm, Göteborg, Malmö, Helsingfors, Oslo, Bergen, Kristiansand, Stavanger, Tønsberg, London, Birmingham och Bristol.

Mer info på www.wspgroup.se eller **INSUL**:s hemsida www.insul.co.nz **ENC**:s hemsida www.causalsystems.com **INSUL** och **ENC** distribueras i Skandinavien av **WSP Akustik**, Bengt Johansson, Telefon 08-555 232 54 bengt.johansson@wspgroup.se

- Hör av dig för demoversion!

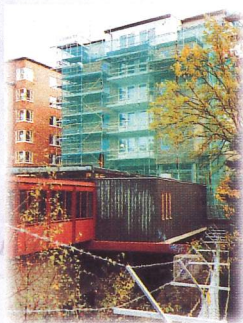
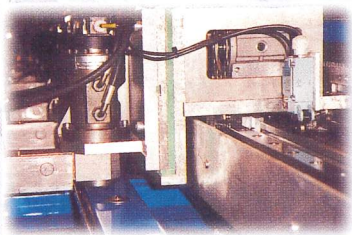
WSP Akustik, Lumaparksvägen 7
Box 92093, 120 07 Stockholm.
Telefon 08-555 232 00
Fax 08-555 232 50



MINIMAL VIBRATION FÖR DIN APPLIKATION™

Christian Berner **Vibrationsteknik** är en erfaren och kompetent samarbetspartner inom vibrationer och stömljud. Vi erbjuder optimala lösningar för Din applikation inom

- järnväg
- bygg
- industri
- produkter



sylomer®
SYLODYN®

GERB

Vi representerar marknadsledande tillverkare av både PUR-elastomerer och fjäderelement.

Du är välkommen med Din applikation.

Teknisk rådgivning • Pålitlig partner
Flexibilitet • Marknadsledande produkter



CHRISTIAN BERNER AB

BOX 207 • SE-433 24 PARTILLE • TEL 031-33 66 900 • FAX 031-33 66 999

E-post: info@cbab.se • Hemsida: www.cbab.se

Jernbaneverket i Norge -

utfører fullskalaforøk for utprøving av metoder for reduksjon av strukturstøy (stomljød) fra jernbanetrafikk i fjelltunneler.

av Christian Madshus, NGI

Jernbaneverket i Norge rustet for tiden opp banen som går ut fra Oslo i sydvestlig retning; Drammensbanen. På strekningen mellom Sandvika og Asker bygges en ny dobbeltsporet tunnel som går under flere bebygde områder. Mot Asker er det liten fjelloverdekning og dårlig fjellkvalitet. Her har forundersøkelser /1/ vist at konvensjonelle løsninger med matter under ballasten ikke vil gi tilstrekkelig reduksjon av strukturstøyen i boligene over tunnelen. For å gi plass for ekstra tiltak er tunnelen på dette partiet sprengt ut 2m dypere enn vanlig.

I denne forbindelse har Jernbaneverket satt i gang et utviklingsprosjekt for å finne den teknisk og økonomisk beste løsningen for å få strukturstøyen ned på et akseptabelt nivå. Etter nylig fullførte studier /2/ setter norske myndigheter en grensen for strukturstøy fra jernbane på 32dBA i oppholdsrom, målt som statistisk maksimalverdi ved togpasseringer. Løsningen som prosjektet kommer frem til vil bli anvendt både i den aktuelle tunnelen, og på en rekke andre tunnelstrekninger som vil bli bygget i fremtiden.

Første fase i prosjektet har vært å etablere en numerisk regnemodell for å simulere overføring av dynamisk last og vibrasjon fra toghjul, via forskjellige typer overbygning og underbygning, og med i fjell. Ved hjelp av modellen er det valgt ut fire alternative baneoppbygginger som blir prøvet ut i full skala. Som referanse blir det også prøvet ut tre mer konvensjonelle baneoppbygginger. Oppbyggingene består i forskjellige kombinasjoner av

ballast, steinfylling, lettklinker, ballastmatter og skinneinnfestinger.

Fullskalforsøkene utføres i den aktuelle delen av tunnelen, som nå er ferdig sprengt ut. De syv forskjellige baneoppbyggingene er lagt ut etter hverandre i ca. 8 m lengder. En godsvogn er bygget om og forsynt med servo-hydrauliske aktuatorer for å simulere de dynami-

ske lastene fra passerende tog. For hver baneoppbygging blir det målt dynamiske krefter og akselerasjon på sviller, i forskjellige nivåer i baneoppbyggingen og på fjell i tunnelsålen og på tunnelvegg. I tillegg blir det målt støy og akselerasjon på gulv i utvalgte boliger over tunnelen. Under opplastning av statisk vekt på vognen måles skinnenedbøyning. Forsøkene vil gi unike data for fastlegge strukturstøyrducerende effekt av baneoppbygginger og vil også gi verdifulle data for "kalibrering" og videreutvikling av beregningsmetoder for strukturstøy.

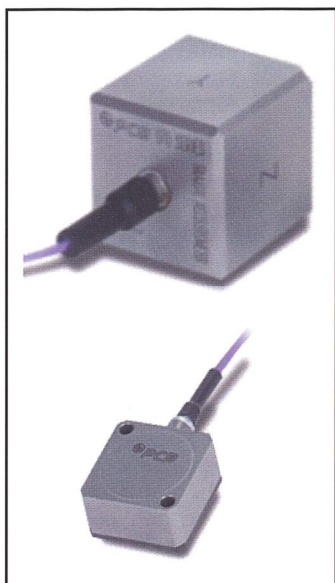
Prosjektet er initiert av, og utføres for Jernbaneverket Utbygging – Nytt dobbeltspor Sandvika – Asker. Prosjektet gjennomføres i fellesskap av Norconsult, Brekke og Stand Akustikk og Norges Geotekniske Institutt. Kungliga Tekniska Högskolan / MWL bistår med det servo-hydrauliske belastningsutstyret. Oppbygging av prøvestrekningene er utført av entreprenør for tunnelarbeidene; Mica AS. Sporet er lagt av BaneService.

Fullskalaforøkene gjennomføres tidlig i mars 2004. Konklusjoner skal foreligge omkring påsketider. ■



SPINDLER®

Industriaccelerometer för vibrationsövervakning av
högstighets bearbetningsmaskiner



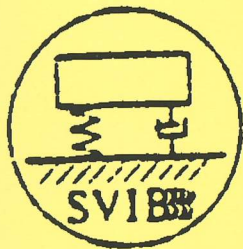
 **PCB PIEZOTRONICS** INC.
VIBRATION DIVISION

Kapacitiva Accelerometrar

En- och triax accelerometrar för
typiska miljömätningar. Givarna
har frekvensområden från DC
till 500 Hz och finns i
mätområden från 3g till 20g
med 100 eller 1000 mV/g i
känslighet.

 **PCB SCANDINAVIA AB**

Ring 08-444 38 70 eller
E-post: info@pcbscandinavia.se



Vibrationsdagen

2004

Vibrationsdagen blir 23 november 2004 i Stockholm. Plats meddelas senare: Opera terrassen eller IVA på Grev-Turegatan har föreslagits.

Temat denna gång är:

Vibrationers växelverkan med ljud och buller

Vi vill gärna få förslag på föredrag. Kontakta Martin Almgren

E-post: martin.almgren@ingemansson.se

Titel och sammandrag önskas senast **1 maj 2004**.

AU3-dag 2004

I samband med SVIB-dagen hösten 2004 kommer en AU3-dag att anordnas den 24 november på liknande sätt som vi gjorde 2002 i Göteborg.

Temat denna gång är:

Vibrationer i byggnader

Dagen är tänkt att ägnas åt tre delområden:

1. Vibrationer från sprängning och annan byggverksamhet.

2. Vibrationskänslig utrustning

3. Vibrationsalstrande utrustning

Vill ni bidra med ett föredrag,

kontakta Tomas Odebrant

E-post:

tomas.odebrant@ingemansson.se

Titel och sammandrag önskas senast

30 april. 2004.

Utbildning Konsulttjänster Programvaror

Vi är Nordens ledande utbildningsföretag inom ljud- och vibrationsområdet.

Vi erbjuder även kvalificerade konsulttjänster och Matlab-baserade programvaror inom området.



The toolbox for spectrum analysis, statistics and more



The toolbox for modal analysis, mechanical simulation, and system identification



The toolbox for experimental fatigue and durability analysis, and environmental engineering computations



Axiom EduTech AB

P.O. Box 2822

SE-187 28, TÄBY

SWEDEN

PHONE: +46-708-57 50 38

FAX: +46-708-57 50 10

info@axiom-edutech.com

www.axiom-edutech.com

Saven EduTech har bytt namn och heter nu Axiom EduTech

The screenshot shows a web browser window displaying the homepage of Nitro Consult AB. The browser's address bar shows the URL 'http://www.nitroconsult.se/'. The website has a blue header with the company name 'Nitro Consult AB' and a navigation menu. The main content area is titled 'Välkommen!' (Welcome!) and contains text about the company's services, including 'Nitro Consult är en av Sveriges största företag inom...' and 'Nitro Consult är en av Sveriges största företag inom...'. There are three images on the right side: a yellow sun, a red heart, and a group of people in green uniforms. The footer contains the text '© 2008 Nitro Consult'.

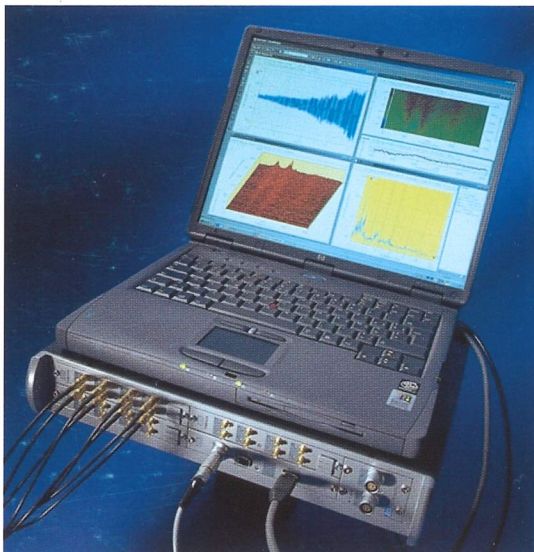

Nitro Consult

www.nitroconsult.se

Nitro Consult

LMS Pimento

The Multichannel Analyzer with Portable Power



- Compact, ruggedized design powered by ac, dc, or internal battery
- Expandable from 4 to 24 channels with full signal conditioning and ICP® supply
- 24-bit ADC, >1 MegaSample/sec throughput to host laptop/PC
- Up to 40kHz bandwidth/channel, with multiband and real-time octaves
- 2 tach inputs for tracked analysis
- Auto industry CAN-bus support
- 24-bit DAC signal source for structural testing
- Based on familiar Windows 2000 operating system

Empowering Engineering Innovation



INGEMANSSON AUTOMOTIVE AB
P.O. Box 276, S-401 24 GÖTEBORG
Kämpegatan 7
tel.: +46 (0)31 774 74 00
fax.: +46 (0)31 774 74 74
lms@ingemansson.se
www.ingemansson.com



LMS
INTERNATIONAL

Empowering Engineering Innovation

Interleuvenlaan 68
B-3001 Leuven
<http://www.lmsintl.com>