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Värdefulla mätresultat

Mätresultat kan vara värdefulla på flera olika sätt. Föreställ Dig följande orsakssamband:

- Maskin – krafter i rörelse – vibrationer – slitage eller utmattningsbrott – haveri – kostnad för produktionsbortfall och reparation.

- Maskin – krafter i rörelse – vibrationer – begränsad produktionshastighet – begränsade intäkter.

- Maskin – krafter i rörelse – vibrationer – vibrationsskador på människor – försäkringskostnad eller skadeersättning.

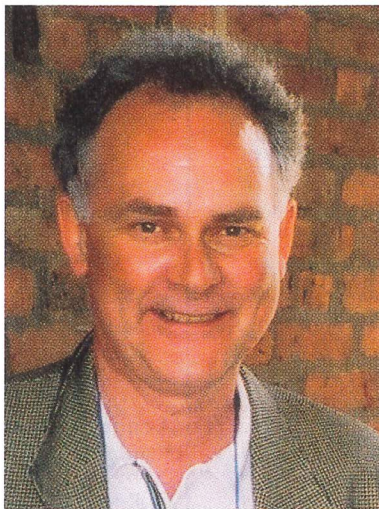
- Maskin – krafter i rörelse – vibrationer – ljudutstrålning – bullerskador eller bullerstörning – försäkringskostnad eller skadeersättning.

I alla dessa fall ger vibrationer upphov till kostnader eller begränsning av intäkter. Genom att mäta vibrationer kan man i samtliga dessa fall minska risken för onödiga kostnader och öka intäkterna. Att mäta onödigt mycket kan också vara dyrt. Mätningar i kombination med beräkning eller simulering ger ofta ett totalt sett billigare och bättre svar på hur man ska åtgärda problemet. En kollega berättade för mig om en anläggning där en maskin gick sönder med jämna mellanrum. Varje gång detta hände gjorde man brandkårsinsatser med vibrationsmätningar.

Problemet upprepades i mer än ett decennium. När min kollega blev inkopplad kunde han, med hjälp av 1) sin kunskap om hur

delarna i en sådan maskin kan röra sig och vibrera, 2) kundens gamla mätresultat och 3) en enkel teoretisk modell, hitta orsaken och föreslå åtgärder som sparar mycket pengar åt kunden. Heureka, utropade han när han funnit sambandet.

I ovanstående exempel hade onödigt många mätningar utförts, men de fanns i alla fall dokumenterade och tillgängliga för min kollega. Som beställare av mätningar skulle jag vara väldigt nogga med att planera mätningen noga. Jag skulle tänka igenom varför mätningen ska göras – vilket svar vill jag få? Jag kan också hoppas på att den som ska utföra mätningen vägleder mig som beställare till rätt problemlösning.



En dokumentation av mätningen ska man kräva, och betala för. Ibland vill kunder komma billigt undan för stunden och betalar bara för att få ett mätvärde som kan användas direkt för en tillfällig lösning. Senare användning av mätresultatet omöjliggörs då. Eventuella fel i den gamla mätningen bli omöjliga att upptäcka utan dokumentation. Jag har själv varit med om att misslyckas med att sälja in rapportering av en mätning där kunden var väldigt envis och kortsiktigt sparsam.

Mätresultatets värde varar om de är väl-dokumenterade och om de görs tillgängliga i öppna beständiga format. Det är bra om mätningar kan göras enligt en standard.

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Innehåll:

Värdefulla mätresultat - <i>Martin Almgren, Ordförande, SVIB</i>	1
Encyclopedia of vibrations, 2002 - <i>B. R. Mace</i>	4
The Tenth International Congress on Sound and Vibration - <i>Hans Bodén and Anders Nilsson</i>	10
EU-direktiv 2002/44/EG om risker till följd av vibrationsexponering i arbetet - <i>Pär Friberg</i>	12
Obituary Bo Lindqvist - <i>Jenny Buck</i>	15
The UK HRV conference - <i>Alf Olsson</i>	17
Vibration exposure on agricultural machines - <i>Richard M Stayner and Dr Andrew J Scarlett</i>	18
Active Magnetic Bearings International Standardization - <i>Osami Matsushita</i>	30
Kurser och konferenser -	44
Ride Comfort and ability to work in tilting trains - <i>Johan Forstberg</i>	45

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Ibland måste man göra avvikelser mot standard. Då ska avvikelserna stå tydligt i mät-rapporten. Om mätningen inte sker enligt standard ska mätmetoden beskrivas mycket noggrant. Jag tycker det är viktigt att göra diagrammen i rapporten så fullständiga som möjligt. Det är inte ovanligt att ett diagram tas ur sitt sammanhang. Man har då inte rätt förutsättningar om inte diagrammet innehåller nödvändiga upplysningar. Om vibrations-hastigheten anges som nivå i dB är det t ex nödvändigt att ange referenshastigheten dvs 1 nm/s eller 50 nm/s. Det skiljer 34 dB mellan dessa olika sätt att ange vibrationshas-tighetsnivå. Ytterligare ett exempel är redo-visning av spektrum. Det är viktigt att ange analysbandbredden. För brusartade vibrationer skiljer det 10 dB för analysbandbredden 10 Hz och 1 Hz.

Det blir alltmer vanligt med konditions-övervakningssystem (condition monitoring) för maskiner och anläggningar. Allt från pappersmaskiner till broar övervakas. Företagens skatt i form av mätsignaler blir mera värd om mätsystemet har ett öppet format, som inte är låst till en viss tillverkares system. Jag ber den som ska upphandla sådana mätsystem att tänka framåt och specificera mätsignalernas och mätresultatens använd-barhet. ■

Martin Almgren

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Encyclopedia of vibrations, 2002

by

S. G. Braun (editor-in-chief), D. J. Ewins and S. S. Rao (editors). London: Academic Press, 1595 pp. price £620, \$925 (three volumes, hardbound). ISBN 0-12-227085-1

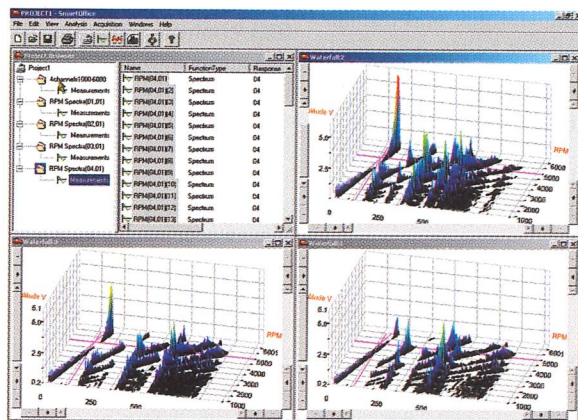
With its three volumes and 1660 pages, the editors of the Encyclopedia of Vibrations have produced a substantial and important reference work for practicing engineers and researchers. With its numerous contributors, the Encyclopedia provides comprehensive coverage of topics concerning mechanical vibrations. The stated aim of the Encyclopedia is to «focus on aspects of interest to practicing and research engineers», and these aspects cover a broad spectrum of traditional, modern and interdisciplinary topics. In achieving this aim, the Encyclopedia goes further than previous references works.

The general scope of the Encyclopedia concerns mechanical vibrations interpreted in a wide sense. It covers aspects of more traditional vibration analysis, application and engineering. However, emphasis is also placed on more modern, interdisciplinary topics, such as active control, smart structures and signal processing, and on the use of computers for vibration modelling, identification and control. These topics are at the forefront of current research and developments in the field and have become important because of the enormous impact of computer-related technologies in recent decades. With this enlarged scope, the Encyclopedia offers a modern reference work for the vibrations engineer or for those with interests in the field. Encyclopedias come in a variety of styles, depending on the form and length of the entries.

Here, there are 180 or so articles, arranged alphabetically, the length of an article typically being 5–15 pages. This style seems very sensible, the length of a typical article allowing the topic to be developed in some depth, as is necessary. The reader is encouraged to navigate the Encyclopedia by use of internal cross-referencing, as well as the contents, and index pages. A glossary of terms is provided, together with seven appendices covering nomenclature, standards and so on. There are 55 colour plates. The strength of the Encyclopedia can be judged by the quality of those who have been involved: the three editors, the 20 expert members of the editorial advisory board and the 140 contributors to the Encyclopedia, all of whom are acknowledged authorities.

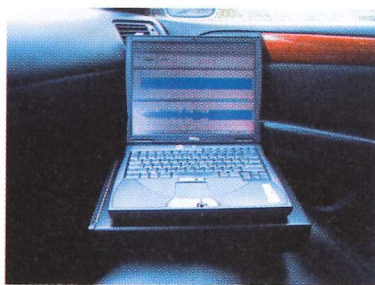
There are too many articles to list the titles individually. However, to give some favour of the content of the Encyclopedia, the articles can broadly be grouped as follows. First, there is a group of 30 or so articles concerning the fundamentals of vibration, briefly covering material that is contained in a first vibrations textbook. A further 20-odd articles concern more advanced, but still traditional, material. Together, these articles summarize various aspects of the fundamentals of vibration theory, analysis and modelling, the behaviour of structural elements such as beams, plates and membranes, random vibrations, non-linear systems and so on. The remaining articles in general con-

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tain brief, review-type material, on more advanced and/more specific topics. It is here that the main value of the Encyclopedia lies, in the opinion of this reviewer. These articles fall broadly into the following groups:

Vibrations: specific applications (56): Many specific topics are considered, ranging from transport vehicles to bridges, packaging to tyres. A number concern machinery bearings, gears, rotors, etc. together with condition monitoring and diagnostics. Other articles concern fluids and structures, noise, earthquake engineering, fatigue and human effects.

More general issues include modal analysis, model identification and model updating. Experimental issues (18): Vibration instrumentation, shock and vibration measurement and testing, environmental testing. Experi-

mental modal analysis, NDT, ultrasonics. Numerical analysis and methods (12): Linear algebra and eigenvalue analysis. Finite element, boundary element and finite difference methods; software. Signal processing (17): with a slant towards the processing of vibration signals. Digital signal processing: averaging, windowing, integration, etc. Spectral analysis; cepstral analysis; time-frequency methods. Digital filtering, optimal and adaptive filters. Transform methods (FFT, Hilbert, etc.). Neural networks. Active control and smart structures (18): Active vibration control: isolation, damping, absorbers and applications to civil structures and vehicles. Hybrid control. Smart materials (piezo-electric, electrorheological, etc.), structures, sensors, actuators and applications; MEMS. In general, the articles are of high quality and reflect the state of the art. This is especi-

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ally true of self-contained articles or those that form part of a co-ordinated series concerning different aspects of a single, broader topic. However, there is some unevenness in style and depth, suggested «further reading» and so on. This particularly applies to those articles that present the more basic fundamentals of vibration theory, the stuff of textbooks, perhaps. One is left with a feeling of a lack of co-ordination of detailed content, with some repetition and omission of core material. Those most likely to refer to these sections would be the inexperienced reader or an expert working outside his field perhaps introductory, overview articles would have helped to organize this material for their benefit. The range of topics covered is broad, but there is some unevenness in content. Issues covered only modestly at best include audio-frequency vibrational behaviour and structure-borne sound, NVH-related issues, active structural acoustic control and the dynamics of structures with uncertain properties (stochastic FE, reliability methods, fuzzy structures, fuzzy FE, etc.).

The use of colour is welcome and more of the figures would certainly benefit from being in colour (e.g., time/frequency representation, mode shapes, etc.). All books have a shelf-life, of course, and one can envisage significant advances in this field being made in the coming years. However, one can expect most of the content of the Encyclopedia to remain of value for 10–20 years or more. Purchase also includes access to an on-line version of the Encyclopedia for a limited period, including permission to download material for classroom and other uses. The facility will not be available until 2002/3, however.

The acid test concerns who would use the Encyclopedia and whether it would meet their needs. Practicing vibrations engineers, experts looking outside their immediate specialism and new and established researchers

with some background knowledge of vibrations will all find much valuable material in these volumes. The Encyclopedia is enjoyable for casual browsing, and valuable for deeper studies.

In summary, the editors, co-editors and contributors have performed a valuable service and should be congratulated for assembling a work of such substance. At £620/\$925 the Encyclopedia is not cheap, but provides 1600 pages in a two-column format: a substantial quantity of diverse, high-quality material in one source, that reflects the current state of the science and engineering of mechanical vibrations. The Encyclopedia deserves a place in all academic and specialist libraries ■

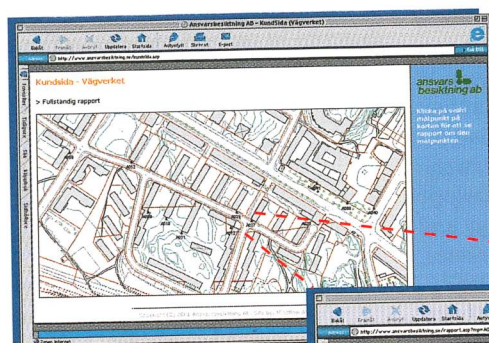
B. R. Mace

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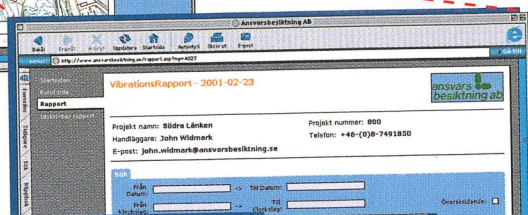
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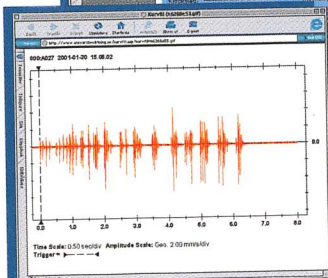


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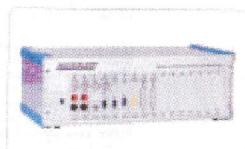
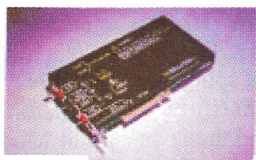
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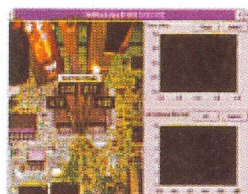
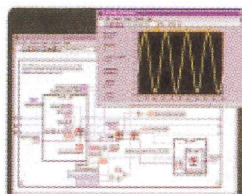
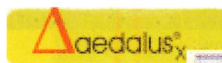
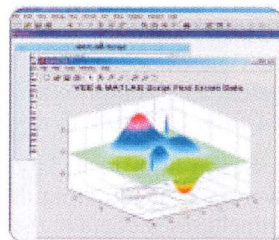
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The Tenth International Congress on Sound and Vibration

The Tenth International Congress on Sound and Vibration, ICSV 10, was held at the Royal Institute of Technology, KTH in Stockholm from the 7th to 10th July this year. The Congress was sponsored by KTH, the International Institute of Acoustics and Vibration, IIAV and the Scandinavian Vibration Society, SVIB. It was also supported by the American Society of Mechanical Engineering, ASME and additionally sponsored by Ingemansson Technology, PCB and Larson Davis.

A total of 620 papers by authors from 52 countries were published at the Congress. Proceedings distributed in paper and electronic formats. 584 papers were finally presented in 13 parallel sessions lead by 88 technical session chairmen. There were 676 delegates and 73 accompanying persons at the ICSV10 which made it the largest in the ICSV series of congresses so far. 126 delegates came from Sweden, 48 came from UK, 44 from USA and 42 from Russia.

Seven high level keynote presentations were presented during the congress by prominent researchers in the field of sound and vibration. The IIAV Honorary Fellow Lecturer was given by Heinrich Kuttruff, Germany "Room acoustics - Art or Science, Ann Dowling, Cambridge University, UK "Singing Flames - The Coupling of Acoustics and Combustion", Stephen J. Elliot, ISVR UK "Distributed control of Sound and Vibration, Yuri Bobrovnskii, Mechanical Engineering Research Institute of Moscow RU, "Measurement of the Vibration Energy Characteristics on Structures", Otto Gartmeier, Daimler-Chrysler, Germany, "Noise and Vibration in Vehicle Comfort Design",

Jean-Louise Guyader, INSA de Lyon, France, "Energy Residual; A Tool to study the Dispersion of Vibroacoustic Performances of Structures and David Burnett, NATO SACLANT, Italy "Finite-Element Methods for Structural Acoustics: Physics, Mathematics and Modelling".

The congress registration opened in the Nobel Museum located in the old town of Stockholm on Sunday afternoon the 6th of July. There was a welcome reception sponsored by PCB and later in the evening there was a dinner in restaurant Berns for the session chairs.



The 18th century king and queen of Sweden opening the Congress.

The opening ceremony started early Monday morning with some welcoming words from Congress Co-Chair Professor Anders Nilsson. He gave a brief historical expose of Stockholm in science and art as an introduction to a couple of songs by the Swedish 18th century poet Carl-Michael Bellman sung by

members of the Säfte Opera Company. The 18th century king and queen of Sweden finally entered the auditorium and officially opened the Congress.

The IIAV President Professor Barry Gibbs proceeded to give the IIAV fellow award to Professor Malecki from Poland. He was however unable to come to Stockholm and the award was accepted on his behalf by Rector of AGH in Krakow, Professor Richard Ta-deusiewicz.

The IIAV Honorary Fellow award was then presented to Professor Heinrich Kufruff.

*Hans Bodén and Anders Nilsson,
Congress co-chairs*



Martin Almgren, Malkolm Crokker och Tomas Lagö.

EU-direktiv 2002/44/EG om risker till följd av vibrationsexponering i arbetet

Under sommaren 2002 antogs inom EU ett särdirektiv om arbetstagares hälsa och säkerhet vid exponering för risker i samband med vibrationer i arbetet, även kallat *Physical agents vibration directive* (2002/44/EG). Medlemsländerna i EU har fram till den 6 juli 2005 på sig att överföra direktivet till nationell lagstiftning. Medlemsländerna har i sina nationella lagar och föreskrifter möjlighet att skärpa kraven i direktivet men får inte sänka dem, då detta är ett s.k. minimidirektiv som beskriver den lägsta nivån av krav på hälsa och säkerhet som får förekomma inom EU.

Kraven i direktivet riktar sig till arbetsgivare inom verksamheter i vilka arbetstagare utsätts eller kan utsättas för risker till följd av vibrationsexponering i arbetet. Direktivet omfattar både exponering för handarmvibrationer och helkroppsvibrationer.

För att uppfylla de krav som ställs i direktivet så skall arbetsgivaren planera och bedriva arbetet på ett sådant sätt att arbetstagarnas vibrationsexponering nedbringas till lägsta möjliga nivå. I denna planering skall arbetsgivaren bl.a. se över möjligheterna till alternativa arbetsmetoder, val av lämplig arbetsutrustning, underhållsprogram, utformning av arbetsplatser m.m.

Vidare skall arbetsgivaren säkerställa att arbetstagare som utsätts för risker till följd av vibrationsexponering ges information och utbildning rörande dessa risker.

Arbetsgivaren skall även regelbundet bedöma riskerna som arbetstagarna utsätts för till följd av exponering för vibrationer i ar-

betet. Denna riskbedömning skall innehålla en uppskattning av arbetstagarnas vibrationsexponering. Denna uppskattning kan genomföras på två sätt, antingen genom mätning eller genom hänvisning till relevant information om använd arbetsutrustnings vibrationsnivå under de aktuella förhållandena (t.ex. från tillverkaren av arbetsutrustningen).

Med uppskattning av arbetstagarnas vibrationsexponering menas i direktivet den dagliga vibrationsexponeringen, $A(8)$ d.v.s. en slags medelnivå för den vibrationsexponering som arbetstagaren utsätts för under en 8-timmars arbetsdag. Medlemsländerna har som alternativ till $A(8)$ möjlighet att ange den dagliga vibrationsexponeringen som VDV , men detta är för svensk del inte aktuellt.

Den dagliga vibrationsexponeringen, $A(8)$ bestäms för **handarmvibrationer** som vektorsumman för effektivvärdena av de frekvensvägda accelerationerna i tre rätvinkliga riktningar, under en 8-timmarsperiod, i enlighet med *ISO 5349-1 (2001), kapitel 4-5 och dess bilaga A*. Om utrustning hålls av båda händerna, så skall en uppskattning göras på båda händerna och det högre av värdena skall ligga till grund för riskbedömningen.

För **helkroppsvibrationer** bestäms den dagliga vibrationsexponeringen, $A(8)$ av den högsta av effektivvärdena för de frekvensvägda accelerationerna, inklusive multiplikation med k-faktor, i tre rätvinkliga riktningar, i enlighet med *ISO 2631-1 (1997), kapitel 5-7 samt dess bilagor A och B*.

Om det i riskbedömningen framkommer att en arbetstagares vibrationsexponering, $A(8)$ överstiger insatsvärdena (*action value*) i direktivet, skall orsaken till detta utredas och ett tidsplanerat åtgärdsprogram utarbetas och genomföras. Åtgärdsprogrammet kan bestå av tekniska och organisatoriska åtgärder och skall ha som mål att minimera riskerna för ohälsa. Åtgärdsprogrammet skall dokumenteras och kunna visas upp för tillsynsmyndigheten i medlemslandet.

Vidare så säger direktivet att vibrationsexponeringen, $A(8)$ inte under några omständigheter får överskrida gränsvärdena (*limit value*). Om det i riskbedömningen framkommer att detta sker skall arbetsgivaren omedelbart vidta åtgärder för att minska vibrationsexponeringen så att den ligger under gränsvärdena. Arbetsgivaren skall även fastställa orsakerna till att gränsvärdena överskridits samt förbättra skyddsåtgärderna och de förebyggande åtgärderna för att undvika att överskridandet upprepas.

Av direktivet framgår det även att arbetsgivaren i sin riskbedömning skall uppmärksamma om vibrationsexponeringen har ett stort stötnnehåll, inverkan av särskilda arbetsförhållanden, om arbetstagare tillhör särskilt utsatta riskgrupper m.m.

Arbetsgivaren skall anordna hälsokontroller när arbetstagare exponerats för sådan vibration att ett samband kan fastställas mellan denna exponering och skadliga hälsoeffekter, när det är sannolikt att sjukdom eller hälsoeffekter kommer att inträffa p.g.a. arbetstagarens särskilda arbetsmetoder och när det finns utprovade metoder för att upptäcka sjukdom eller de skadliga hälsoeffekterna. Arbetsgivaren skall under alla omständigheter ordna med lämpliga hälsokontroller till arbetstagare som exponeras för vibrations-exponering över insatsvärdet.

I direktivet anges följande minimikrav för insats- och gränsvärden:

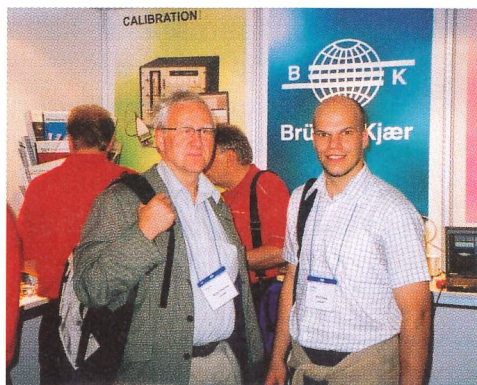
- Insatsvärde (*action value*) Gränsvärde (*limit value*)
- Handarmvibration 2.5 m/s^2 5.0 m/s^2
- Helkroppsvibration 0.5 m/s^2 1.15 m/s^2

Direktivet ger även medlemsländerna möjlighet att, under vederbörligen motiverade omständigheter, införa undantag från gränsvärdet avseende helkroppsvibrationer när det gäller luft- och sjöfartsområdet. Medlemsländerna kan även införa möjligheten att beräkna vibrationsexponeringen under en 40-timmarsperiod i de fall då arbetstagarnas exponering varierar märkbart från den ena stunden till den andra. Vidare så kan medlemsländerna, efter samråd med arbetsmarknadens parter, införa övergångsperioder när det gäller arbetsutrustning vars användning innebär att gränsvärdena inte går att uppfylla trots de senaste tekniska framstegen och organisatoriska åtgärder.

Arbetsmiljöverket kommer att med anledning av detta direktiv ge ut nya föreskrifter om vibrationer. Dessa föreskrifter kommer att träda i kraft senast den 6 juli 2005 ■

Pär Friberg

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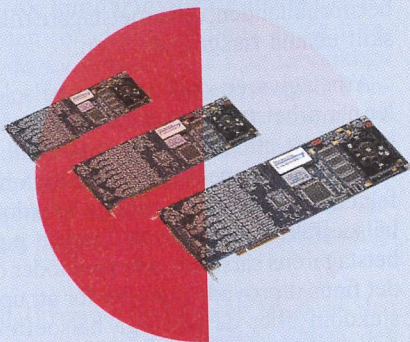
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Obituary Bo Lindqvist



† Bo Lindqvist, 1st September 2003

Just a month ago PN3 was pleased to circulate the minutes of the last meeting which included this picture of Bo Lindqvist and the notification that he was to retire from a successful career not only with his company Atlas Copco but his involvements in PNEUROP, ISO and CEN.

It is with much regret and sadness that we have to report that Bo passed away after a short illness on Monday of this week. This will be most deeply felt by Maire his wife and also his children who were only able to share a very short part of his retirement.

We are sure that all PNEUROP members would like to join in and send condolences to Bo's family at this distressing time.

Bo began his career with Atlas Copco in the late 1960's to begin pioneering work in tool ergonomics which involved him in investigations into the effects of vibration on the human body. His involvement in PNEUROP then began in the 1970's when PN17 "Vibration" committee was formed and he chaired this for the whole of the committee's existence.

Not satisfied with keeping the work in a manufacturer forum he was invited to Convene a working group in ISO TC118. So up until his retirement in June 2003 he chaired ISO TC118 SC3 WG3 "Vibrations in hand held tools" which was responsible for the key standards for vibration measurement in the ISO 8662 series. Many of these involved seminal research into test methodology for the range of tools requiring vibration measurement. Bo was also responsible for a number of publications and many papers on the subject of ergonomics and became a well recognised and respected figure globally in this area.

Bo the man was a likable person who always promoted discussion, debate and cooperation from all sides in his working life. He had wide interests outside of his work that included his "eco-house" which embodied many of his and his wife's thoughts on 'green technology'.

He will be greatly missed.

Chair PN3 – Jenny Buck

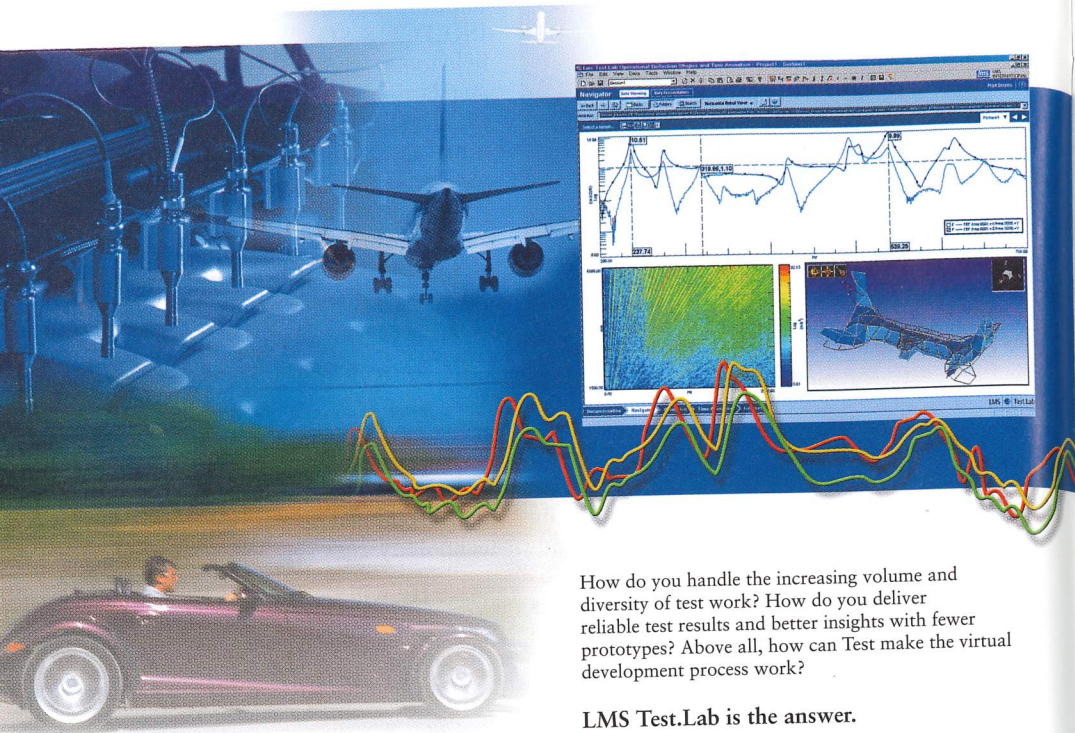
Secretary – Greg Bordiak

Bo Lindqvist var ordförande 1991 – 1996 och utsågs därefter till hedersledamot i SVIB. Bo var en positiv och inspirerande ordförande som visste att inte förvänta sig för mycket. När han i slutet av 80-talet återkom till kommittéarbetet efter några års frånvaro konstaterade han att allting var sig likt. Hans favorituttryck var att "nu har vi 10 minuter på oss om 10 minuter har vi inget på oss". Det är med djupt beklagande jag konstaterar att Bos tid har runnit ut och att allt inte blir sig riktigt likt efter detta.

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The UK HRV conference



The UK HRV (Human response to vibration) conference was this year organised by the Institute of Naval Medicine, INM, in Gosport. During the 3 days conference that had gathered just above 70 participants who presented over 10 countries. 45 papers were presented.

The recently approved EU directive on "Physical Agents" was still a matter of discussion. Several engineering and operational efforts will be needed in order to reduce the time and amount of vibration that people are still exposed to in several industries and workshops in order to meet the EU requirements. There is never the less an important difference.

Hand Arm vibration has been known to be harmful to people being exposed and several well documented symptoms of illness exists. The action- and limit values for hand arm vibration given in the directive are in general considered to be realistic

The situation regarding Whole Body vibration is more confusing. There are two dif-

ferent measures "A8" and "VDV" identified by the EU directive. The "VDV" value is defined in ISO 2631-1:1997 but the A8 value in fact is not defined anywhere in a standard. The measure A(8) is inferred. This was identified as one of the reasons the UK would have supported a revision of either 2631-1 or of the vocabulary for human vibration so that it could be defined somewhere and subsequently referenced.

They A(8) and the VDV values give furthermore different permissible exposure times for identically measured magnitudes of vibration due to the fact that the operator is not only exposed to pure vibration but also to a certain amount of shocks or bumps. Those countries that choose the "VDV" method will find that this method is discriminating since the permissible time for using several transport-, farming- and construction machines will be drastically reduced. Read the following article about "quad bikes" by Richard Styner and you will understand the dilemma

■
Alf Olsson

Vibration exposure on agricultural machines

CASE OF ATVS ("QUAD BIKES")

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Abstract

Vibration, both whole-body and hand-arm, was measured on four models of ATV under controlled test conditions, and for the whole daily exposure of workers on three farms that used one of those models. Exposure is compared with the Exposure Action Level and Exposure Limit Level of the Physical Agents (Vibration) Directive, using both VDV and rms A(8). Large differences between VDV and rms A(8) assessments are discussed. The suitability of the ISO 5008 test track for ATVs is discussed.

1. Introduction

This study of ATVs is a small part of a larger project concerned with the exposure of farm workers to whole-body vibration when operating tractors and other self-propelled machines (Scarlett & Stayner, 2001). The aims of that project were:

- To gather data from working farms.
- To assess the relevance of standard tests of whole-body vibration "emission".
- For tractors, to assess the benefits of recently introduced suspensions.
- To assess the impact of the Physical Agents (Vibration) Directive on the farming industry.
- And to provide data for discussion of the alternative exposure measures of VDV and A(8) rms acceleration.

A short study of ATVs was added because these machines are becoming increasingly

common on farms, and particularly on hill farms and others with sheep. Also, ATVs had not been included in earlier studies (LINES, Stiles & Whyte, 1995).

ATVs, or "Quad Bikes" are lightweight four wheeled vehicles originally developed from motor-bikes. The driver sits astride a saddle-type seat, in the open air, steers by means of handle-bars, and has clutch, brake and other controls similar to those of a motor-bike. The machines are fitted with low pressure tyres, and are marketed as "go anywhere" vehicles. With the addition of load racks front and rear, they are very useful for farmers wishing to visit stock that is dispersed around large areas of rough terrain and to take out feed to remote locations.

Although the main project is concerned with whole-body vibration, it was opportune also to gather data on the handle-bar vibration of these machines.

2. Machines tested

The four machines tested were all in the lower power range, having engine capacities of 350cc to 400cc. Some details are given in Table 1.

Make	Honda	Kawasaki	Suzuki	Yamaha
Model	TRX 350 TM (Four Trax)	KLF300 (-C)4x4	LT-F-300 (King Quad)	400 Big Bear
Weight kg	253	290	312	301
Wheelbase m	1.25	1.2	1.16	1.23
Track width m	0.85	0.84F / 0.86R	0.86	0.82
Front tyres	AT24 8-12	AT24 8-11	AT24x8-11	AT25 8-12
Pressure lb/in2	2.8	5	4.4	3.6
Rear tyres	AT24 9-11	AT24 10-11	AT25x10-12	AT25 10-12
Pressure lb/in2	2.8	4	4	3.6
Front suspension	Independent	Independent	Independent	Independent
Rear suspension	Live axle	Live axle	Independent	Live axle

Table 1: Details of ATVs tested

3. Measurement conditions

In order of reality, the test conditions were: Test track, simulated field conditions, visits to farms. Measurements were made under the con-

trolled conditions at Silsoe Research Institute before any farm visits. This was to ensure that the data acquisition and analysis were sufficiently robust and appropriately scaled.

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3.1. Test track:

The ISO 5008 100m (smoother) artificial track was used (ISO, 2002), as it had been for tractors. The standard range of speeds included 10, 12, 14 km/h with, in the case of the ATVs extra speeds up to 20 km/h. The machines were tested unladen except for the instrumentation pack and driver. The same driver (body weight approximately 70kg) drove the machines for all track and controlled field tests to improve comparability. It should be noted that this track is made of spaced wooden slats.

In addition, for the hand-arm vibration measurements, the machines were driven on a smooth concrete surface. Speeds for this were the maximum in each forward gear. These were chosen only because they were the most easily repeatable, and may not be adequately representative of working conditions.

3.2. Simulated (controlled) Field Conditions:

These were based on components of the daily stock round of a fairly typical sheep farmer in the Welsh Borders. They thus comprised a length of fairly smooth road; a length

of farm track; and a circuit around the headlands of a field. Total time for all components was about 10 minutes.

3.3. Farm Use:

These measurements were restricted to only one of the models of ATV tested at Silsoe. Examples of the same model were found on 3 farms. Two of these were Welsh hill farms raising sheep, and the third was in the Wye valley, and had been converted to a game shoot. On the hill farms a sack of feed or a dog was carried. On the gamekeeper's machine the load alternated between a feed spreader mounted on the rear carrier for ground feeding, and sacks of feed for topping up standing feeders (hoppers). In each case the daily round, took between 30 minutes and an hour, and was made either at the start of the day (0630hrs) or in the middle of the morning.

The object was to make two measurement visits to each farm, one for WBV, and one for HAV. However we achieved only 2 HAV visits because of not wishing to outstay our welcome at one of the farms, but repeated the WBV on one very helpful farm. Durations of the stock rounds and vehicle speeds are shown in Table 2, below.

<i>Farm</i>	<i>Duration min</i>	<i>Average km/h</i>	<i>Max km/h</i>
<i>Hill Farm#1 (1)</i>	55	10.8	46
<i>Hill Farm#1 (2)</i>	70	10.1	53
<i>Hill Farm#2</i>	35	9.4	34
<i>Gamekeeper</i>	140*	7.5	41

* Includes 20 minute break for removing the feed hopper

Table 2: ATV whole-body records, duration and speed range

An 8-channel digital recorder (TEAC PC-Card Recorder) was used as the front end of a PC based HvLab vibration analysis package. Six channels were used for vibration data,

either whole-body vibration, sampled at 200Hz, or hand-arm vibration sampled at 2000Hz. The seventh channel was used for vehicle travel speed, measured with a Doppler radar speed sensor of the type commonly used on farm tractors. This was clamped to

the rear load-carrying rack. The eighth channel was used to record a signal indicating whether the driver was on the seat or not. This last was later used to eliminate sections of recordings when the driver either got off the machine altogether, or stood up on the footrests to look over a hedge or to take the shock of some particularly rough ground through his legs. For the test track measurements, the eighth channel was used for a marker to indicate the start and stop of each run, for later analysis.

The vibration data was pre-processed through a pair of Larson Davis HVM100 "Human Vibration Meters", each of which was connected to a tri-axial accelerometer. For whole-body vibration, one of these was in the form of a seat pad (PCB 356B40), taped to the saddle. The other was a set of 3 Senso-nics piezo-resistive transducers clamped to the right footrest. For handlebar measurements, tri-axial units (PCB 356A24) were strapped to the centre of each hand-grip, beneath light-weight palm protectors.

The Larson Davis meters were set to store frequency weighted data as rms values at one minute intervals, and as VDV's at 15 minute intervals. The signals passed through to the digital recorder were not weighted.

The meters, recorder and associated electronics were fitted in a tractor tool box, used

for earlier tests, which was mounted on the front load rack. Power was taken from the vehicle battery to enable extended measurement periods.

All the recorded data was scanned visually in HvLab to check for obvious defects. The test track and controlled field data was analysed in HvLab between marked start and stop points, to give frequency weighted rms values and VDV's. For horizontal (X- and Y-axes) components of WBV, magnitudes were multiplied by 1.4. For the farm data, the time records of acceleration in HvLab were also multiplied by the driver sensor signal (x1 or x0) before calculating an alternative set of values for the seat vibration. The facility was available for plotting travel speed, driver presence and one-second running rms accelerations.

5. Farm exposures

As indicated above, the WBV records were for stock rounds that lasted between half and one hour, with the exception of the gamekeeper who had held back some work from the previous day in order to provide a long example. The stockman on Hill Farm#1 drove his ATV rather harder than the other two, with the result that he returned higher values of both rms acceleration and VDV. Tables 3(a,b) show the rms and VDV results for seat vibration, Tables 4(a,b) for the footrest, and Table 5 for the hand-grips.

<i>Farm</i>		<i>X</i>	<i>Y</i>	<i>Z</i>
Hill Farm#1 (1)	Overall rms	1.06	1.39	1.39
	Driver on seat	1.00	1.32	1.26
	Difference %	6	5	9
Hill Farm#1 (2)	Overall rms	0.90	1.29	1.21
	Driver on seat	0.84	1.20	1.09
	Difference %	7	7	10
Hill Farm#2	Overall rms	0.72	0.65	0.87
	Driver on seat	0.69	0.62	0.77
	Difference %	4	4	12
Gamekeeper	Overall rms	0.55	0.76	0.85
	Driver on seat	0.53	0.74	0.66
	Difference %	4	3	22

Table 3: (a) Whole-body vibration exposures, ATVs on farms, weighted rms acceleration on seat

Farm		X	Y	Z
Hill Farm#1 (1)	Overall VDV	12.6	16.1	17.0
	Driver on seat	11.7	15.1	15.5
	Difference %	7	6	9
Hill Farm#1 (2)	Overall VDV	12.4	17.5	18.4
	Driver on seat	11.5	16.1	16.0
	Difference %	7	8	13
Hill Farm#2	Overall VDV	10.0	8.1	10.9
	Driver on seat	9.5	7.9	10.2
	Difference %	5	3	6
Gamekeeper	Overall VDV	11.3	11.9	15.6
	Driver on seat	11.0	11.7	14.7
	Difference %	3	2	6

Table 3: (b) Whole-body vibration exposures, ATVs on farms, VDV on seat

The effect on measured exposure values of excluding times when the driver was not in contact with the seat was generally less than 10%. One notable exception was the case of the Gamekeeper, for whom Z-axis r.m.s. changed by over 20%. This was attributable to periods when the driver was off the machine, either to reload the feed sacks, or to fill the feed hoppers. At those times he left the engine idling, which transmitted strong and clearly visible vibration to the seat surface. However, the magnitude was not so great as to affect also the VDV, which was determined by larger transients during travel.

Farm	X	Y	Z
Hill Farm#1 (1)	0.71	0.76	3.62
Hill Farm#1 (2)	0.64	0.74	3.05
Hill Farm#2	0.49	0.43	2.07
Gamekeeper	0.57	0.53	3.30

Table 4a: ATV footrest vibration exposures on farms, weighted rms acceleration

Farm	X	Y	Z
Hill Farm#1 (1)	9.3	8.9	38.7
Hill Farm#1 (2)	9.6	9.8	37.8
Hill Farm#2	7.7	5.2	23.5
Gamekeeper	10.1	9.0	47.5

Table 4b: ATV footrest vibration exposures on farms, VDV

Farm	Left hand	Right hand
Hill Farm#1	8.1	9.3
Gamekeeper	3.9	4.8

Table 5: ATV hand-transmitted vibration exposure in farm trials, m/s^2 rms (weighted sum)

6. "Safe" working time

The measured data can be used to estimate how much daily use could be made of this type of machine before reaching the Action or Limit values prescribed in the Physical Agents Directive. This has been attempted for each farm, using the data for driver on

the seat, with the interesting results shown in Table 6 below. Table 7 shows equivalent times for exposure to hand-arm vibration. One very obvious point to emerge is the wide disparity between estimated times of "safe" use according to which measure is used for whole-body vibration. Then, if the rms measure is used for whole-body vibration, it seems that the limits would be reached first for hand-arm vibration. Thirdly, farm use of between 30 minutes and one hour a day would be below the Exposure Limit Value in all cases. Such use would be around the EAV for hand-

arm vibration. And it would exceed the EAV for whole-body vibration only when assessed according to the VDV.

Although the daily use patterns of the farmers found in this project did not result in exposure that is excessive when assessed according to the Directive, the same cannot be said with confidence for recreational users of these machines. It is likely that such users would actively seek conditions with more or greater shocks, and professional instructors may also be exposed for longer daily durations.

Farm	Estimated time to Action ValueMinutes		Estimated time to Limit ValueHours	
	VDV	A8	VDV	A8
Hill Farm#1 (1)	5	69	2.5	6.1
Hill Farm#1 (2)	8	83	3.6	7.3
Hill Farm#2	29	202	13.5	17.8
Gamekeeper	20	219	9.4	19.3

Table 6: Estimates of time to PA(V)D action and Limit Values for whole-body vibration (based on on Z-axis magnitudes)

Farm	Estimated time to Action Value Minutes	Estimated time to Limit Value Hours
Hill Farm #1	36	2.3
Gamekeeper	132	8.7

Table 7: Estimates of time to PA(V)D action and Limit Values for hand-arm vibration (based on weighted rms sum for worst hand)



Kawasaki KLF300 4x4

7. Levels of vibration, track and field

ATV Seat Vibration (1.4 multiplier): Machine B

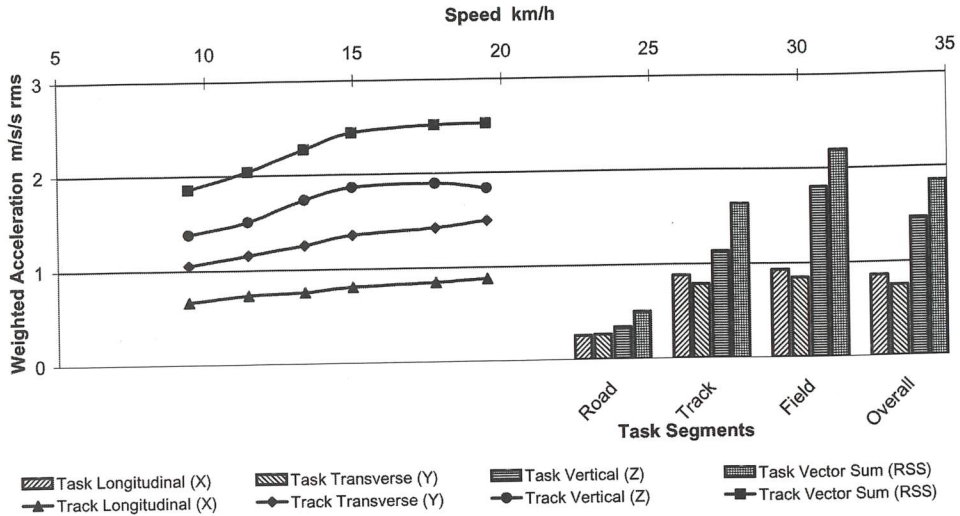


Figure 1: Typical results from controlled vibration tests at Silsoe, showing values measured on test track and over simulated field circuit.

The composite graph above shows typical results of the controlled tests for the seat vibration of machine B (note: the models described in Table 1 are not to be identified individually with machines "A", "B", etc). Variations of the components of vibration with travel speed for the ISO 5008 track, are shown on the left, with on the right the com-

ponents for each section of the controlled "Field" circuit and for that circuit as a whole. These results are typical for half of the machines, showing reasonable increases with increasing forward speed, and between road, track and field surfaces. They also show the predominance of the vertical (Z-axis) component in all circumstances. For the other two machines, and for the footrests of all machines, the vertical component was very much more dominant.

8. Comparison field (controlled) and farm

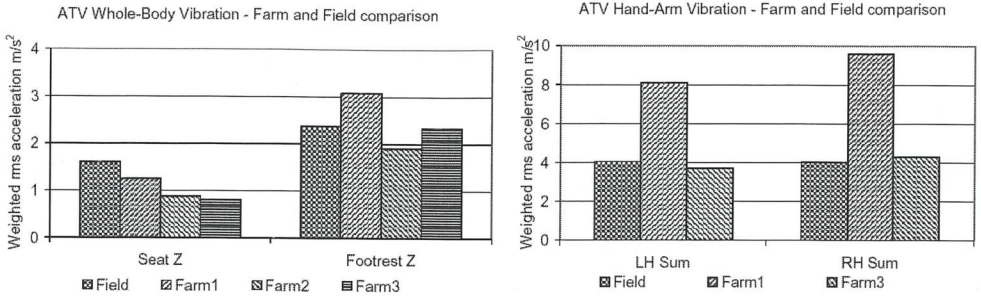


Figure 2: Vibration exposures on farms (average rms levels) compared with emission levels measured on Silsoe field circuit, WBV on left (Z-axis), HAV on right (Sum).

Results from the simulated field circuit at Silsoe (Figure 2, above) are of the same order of magnitude as the farm data. However, there is such a wide spread of the farm data that it would be imprudent to draw more comprehensive conclusions.

There is a lack of correlation between results for seat and for feet, which could be attributable to different driver characteristics,

differences between the machines, or differences between the combinations of speed and ground surface.

What is apparent is a similarity between the foot-rest and handle-bar vibration, despite the different frequency weightings. This supports the suggestion that there is a significant component of engine vibration at the footrest, together with some vehicle low frequency motion feeding through to the handle-bars.

9. Machine comparisons, track and field

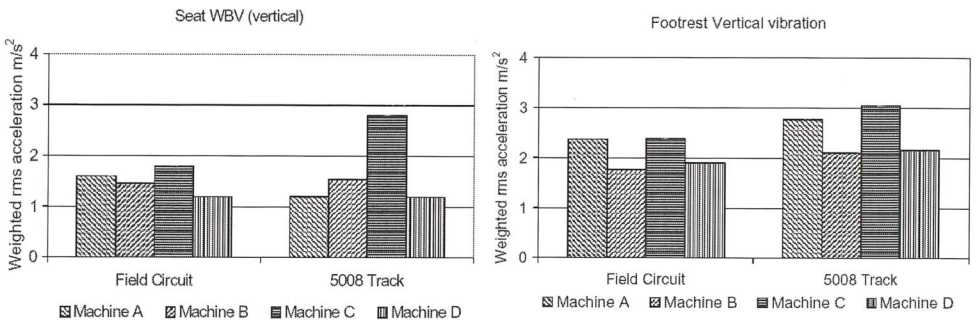
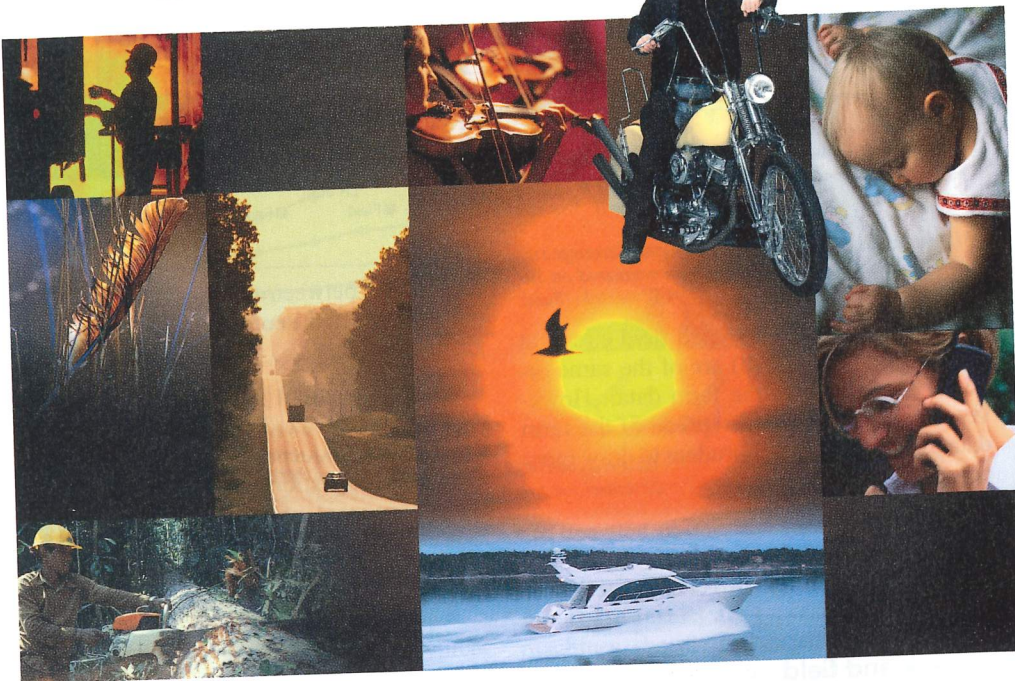


Figure 3: Comparison of different ATVs according to whole-body vibration from test track and field tests, seat on left, footrests on right.

The test track results, which are shown above as the average of the 3 standard test

speeds of 10, 12 and 14 km/h, do not rate the machines at all similarly according to seat vibration. It is possible that the effect of the

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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.

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slatted construction feeds through to the saddle differently on each machine. The results for foot-rest vibration are much more consistent, but again it is probable that the arti-

ficial component from the slat frequency is elevating the test track results. Some further analysis is needed to confirm this.

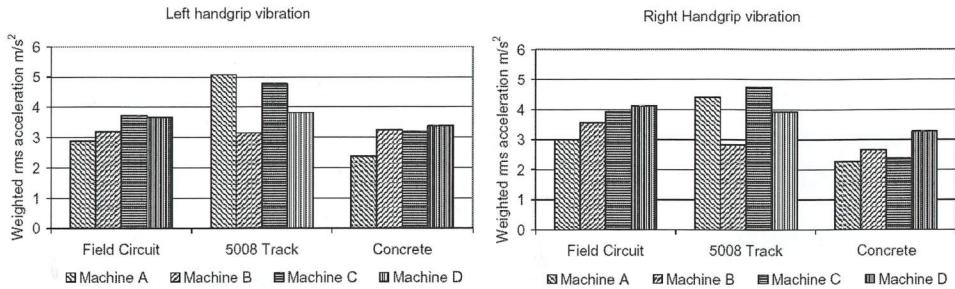


Figure 4: Comparison of different ATVs according to hand-arm vibration from test track and field tests, left hand and right hand shown separately.

Left and right hand results are quite similar. As for the WBV comparisons, the slatted test track does not provide a good predictor of the field results. However, the results from the smooth concrete surface do compare quite closely. The field results were consistently higher than those from the smooth surface. It is tempting to suggest that this was because of the ground surface feeding through. However, the engine speed range for the field was also much wider than for the track test, and could have passed through various structural resonances. Further analysis would be needed to explain the source of the difference.

10. Comment on A_{eq} and VDV

The estimation of "safe exposure" times, in section 6 above, highlights two features relating to the comparability of VDV and rms measures as indicators. One of these should be obvious from the definitions of the measures themselves, the other could have been anticipated, but required real field data to make it obvious.

First, when estimating how long an operator can use a machine before reaching either the Action Value or the Limit Value, it should be clear that a small change in VDV magnitude is associated with a large change in the time. This is a natural result of acceleration being raised to the 4th power and integrated over time. VDV is thus much more sensitive to acceleration magnitude than is rms, and intentionally so.

Secondly, the equivalence used in the Directive, based on the exposure for 8 hours, assumes that **estimated** or eVDV is a good representation of actual VDV. This form of estimate was only ever intended to apply to vibration that is more or less continuous and devoid of transient high acceleration events or shocks. Most conditions in which whole-body vibration exposure is of concern do not conform to these limitations. Thus a vibration exposure that yields an rms ($A(8)$) value of 1.1 m/s² is likely to result in a VDV very different from 21 m/s^{1.75}.

Taken together, these probably explain the disparity between the two sets of estimates

11. Conclusions

- Stock farmers who use ATVs for about an hour a day are likely to be exposed to vibration approaching the Exposure Action Level, for hand-arm vibration, or for whole-body vibration when measured as rms A(8).
- Recreational users of similar machines could be exposed to considerably higher daily levels.
- The ISO 5008 test track, in the allowable slatted construction, does not provide a good indication of field or farm vibration magnitudes.
- Controlled field tests can provide a good indication of farm exposure.
- The equivalence of VDV and rms A(8) as indicators of vibration exposure, as prescribed in the Directive, is not appropriate for the conditions found on this type of vehicle.

12. References

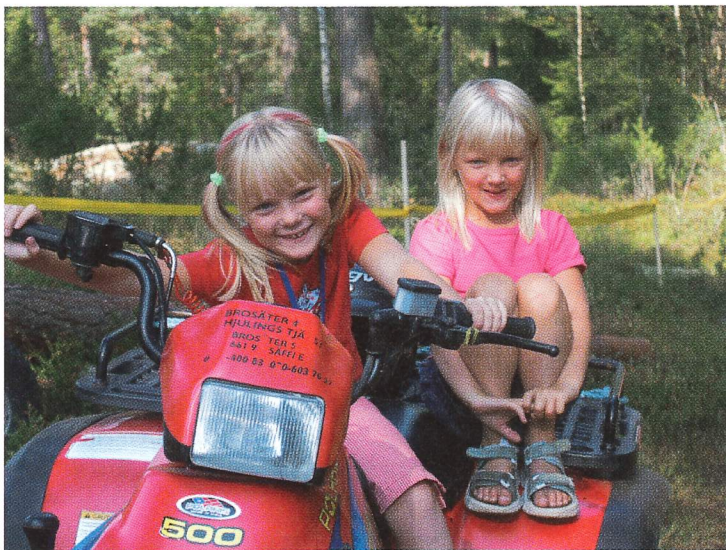
International Organisation for Standardisation (2002) Agricultural wheeled tractors and field machinery – Measurement of whole-body vibration of the operator, ISO 5008.

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Acknowledgements

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Scientists of the future?

Active Magnetic Bearings International Standardization

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1. Introduction

Several types of AMB equipped rotating machinery are currently being successfully implemented by commercial businesses^[1]. Some successful examples include turbo-molecular pumps (TMP), expanders in chemical plants, spindles in machine tool and centrifugal process compressors. The new concept of AMB installation is indeed welcome owing to its features of being contact free, maintenance free, without mechanical losses, etc.

When considering centrifugal compressors, the stiffness of conventional oil-film bearings is relatively high compared with AMB. The rotor vibration is thus suppressed within low levels as regulated by ISO 7919^[2] and/or API617^[3]. On the other hand, the stiffness of the AMB is so weak that the rotor vibrations exhibit large amplitudes, but it is quite normal. As estimated in these comparison between these two types of bearing dynamics, the existing ISO standards are valid only for conventional oil-film bearings, but not suitable for AMB rotors.

Generally speaking, customers require low vibration level for any machine which they want to purchase. By their maintenance experience of conventional turbomachines equipped with oil-film bearings, they learn that the rotor vibration level should be less than 10 mmpp. Then, these users seem to be reluctant to accept relatively large vibration

level of AMB rotors. In addition to the vibration level, the stability margin related to servo feedback control is required concerning the power electronics unit and the robustness against noises. To some customers, how feedback control works is still a mystery.

Manufactures must convince customers to accept this new design and concept of AMB applied to rotating machinery. They need extra energy to erase this mysterious fog covering customers.

One of the most powerful solution to educate proper AMB knowledge and to accelerate the application is the ISO standardization of this technology. The objectives of the standardization on AMB are summarized as follows:

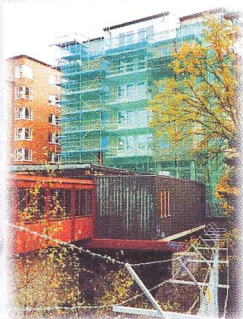
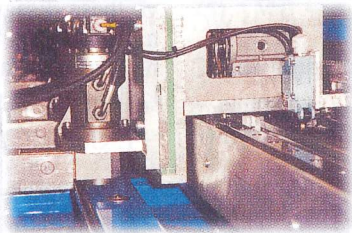
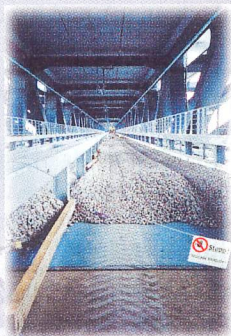
- (1) to provide proper information for mutual understanding between vendors and users, mechanical engineers and electrical engineers, etc.
- (2) to provide useful means in simplifying contract, commission and maintenance
- (3) to serve industry consensus and guidelines on the requirement concerning vibration level and system stability for safe and reliable operation of AMB equipped rotors

In this paper, we describe the history of our ISO activity and introduce ISO 14839 series treating AMB rotors.

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2. ISO TC108/SC2/WG7

In August, 1993, we proposed the necessity of the standardization of AMB rotors to ISO TC108 (Technical Committee: Vibration and Shock) / SC2 (Sub Committee: Mechanical vibration), and it was adopted as a new work item. After collecting experts from each countries, ISO allocated this team in ISO TC108/SC2/WG7 "Working Group: AMB project" in 1997 with the chairmanship of Mr. Matsushita, Japan. At the same time, a domestic committee was rearranged as JSME/ATS10-25 in order to prepare drafting for ISO standards.

Every ISO standard is developed step by step through following 4 stages before the publication of International Standard:

WD.....Working Draft: discussed inside WG

CD.....Committee Draft: handled in SC matters

DIS.....Draft International Standard : TC matters

FDIS...Final DIS: the final circulation and check prior to the publication.

Each draft is circulated to the member bodies for voting with commenting. Comments are fed back to WG to revise the draft and to prepare the step up to the next stage. Publication as an International Standard requires approval by at least 75 % of the member bodies casting a vote.

Since the establishment of ISO TC108/SC2/ WG7, we have been organizing the WG7 international meeting approximately twice a year; so far, the following outcomes have been obtained:

ISO 14839-1 Vocabulary, issued in 2002-05

ISO FDIS 14839-2 Evaluation of vibration

ISO CD 14839-3 Evaluation of stability margin

3. ISO 14839-1 Vocabulary ^[4]

As we know, a new technology tends to produce new commercial words for the business propaganda. We also experience the confusion with dialects defined by themselves in companies and academies. For instance, someone says "I have developed a sensor-less bearing. My AMB don't need any sensing pick-up." But, this is alternatively to be corrected as a self-sensing bearing. There are so many cases of this nature in business talks. In addition, AMB is a combined technology with mechanical and electrical engineers. Since both parties use their own words in certain terminology, we need the unification for their mutual understanding.

Part 1 titled Vocabulary is prepared to solve these disagreements. It contains mechanical and electrical hardware, electromagnets, rotordynamics, control and monitoring laid out in six chapters as follows:

1. General, 2. rotors, 3. stators, 4. sensors
5. dynamics, control and electronic and
6. Auxiliary equipment

Several definitions to be noticed are introduced from part 1as follows:

(1) Symbol of AMB

For rotating machinery equipped with active magnetic bearings, the graphical symbols for bearing are shown in Fig.1.

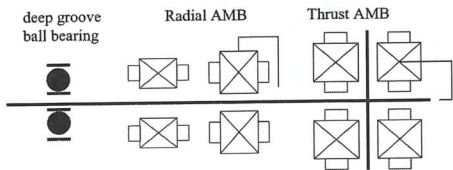


Fig. 1 Graphical symbols for bearings

(2) Active magnetic bearing, AMB

means to support a rotor, without mechanical contact, using only attractive mag-

netic forces based upon servo feedback technology which normally consists of sensors, electromagnet, power amplifiers, power supplies and controllers, as shown in Fig.2.

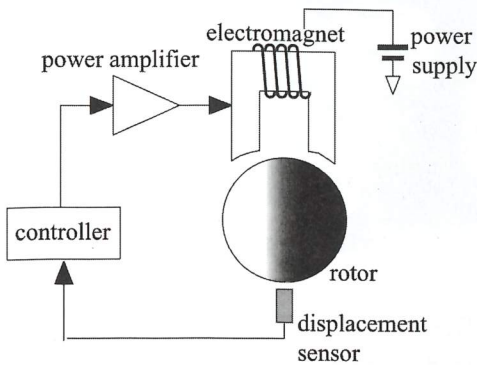


Fig. 2 Principle of active magnetic bearing

(3) Number of poles

in a radial AMB, sum of the south and north poles of an AMB electromagnet actuator, e.g., heteropolar type and homopolar type, as shown in Fig.3.

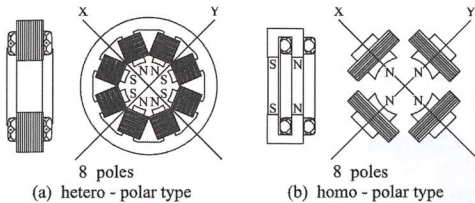


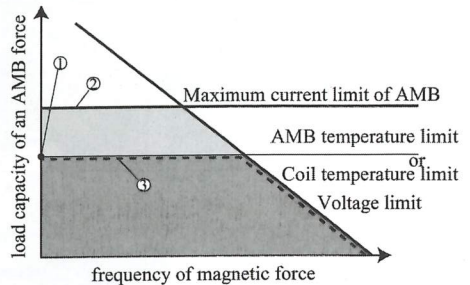
Fig. 3 Number of poles of radial AMB

(4) Load capacity of an AMB

maximum load of a magnetic bearing acting on the rotor at its fixed middle position.

Note: This is usually limited by the magnetic saturation of the ferromagnetic material of stator and rotor core, the maximum coil current available from the power amplifier which drives the magneti-

zing coil and the maximum driving voltage of the power amplifier, as shown in Fig.4.



- ① static load capacity
- ② peak transient load capacity
- ③ dynamic load capacity (maximum continuous operation)

Fig. 4 Load capacity of an AMB

(5) AMB system

system consisting of a rotor, position sensor or other means to detect rotor position, controller(s), power amplifier and electromagnets to levitate and support the rotor by attractive magnetic force, as shown in Fig.2 and 5.

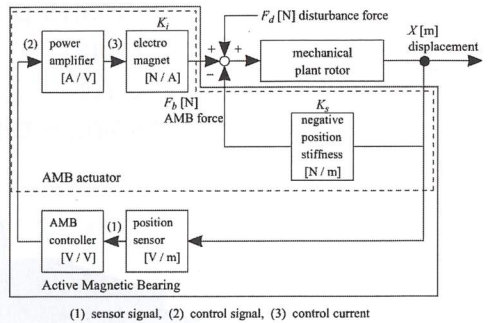


Fig. 5 Block diagram of an AMB system

(6) Unbalance force rejection control

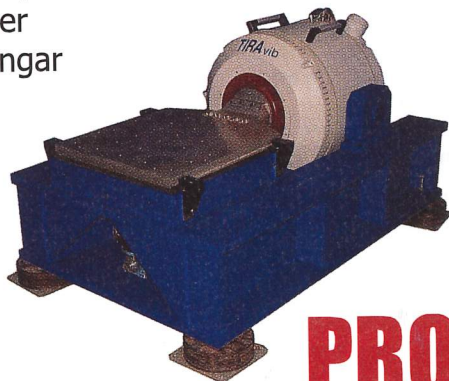
special control method which allows the rotor to rotate around its principal axis of inertia while the transmitted unbalance force through the AMB is minimized, which leads to resulting vibration of the bearing casing, as shown in Fig.6.

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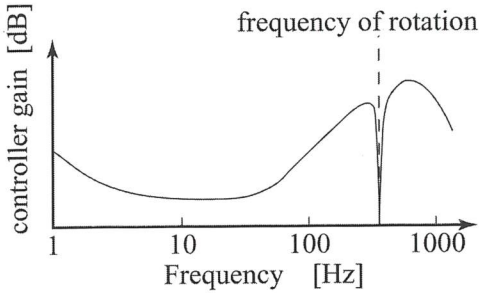


Fig.6 Example of a controller transfer function for unbalance force rejection control

4. ISO FDIS 14839-2 :Vibration [5,6]

4.1 Background

The main part of turbomachinery is still supported by oil film bearing being. Because of the stiff support of oil-film bearings with small clearances, e.g., $C/R = (\text{bearing radial clearance}) / (\text{journal radius}) \gg 1/1000$, shaft vibration should be regulated within low levels to avoid oil-film rupture of the lubricant and metal contact inside the bearing, as stated in ISO 7919 series for turbines, generators, pumps, compressors, etc.

On the other hand, since AMBs support the rotor softly in large bearing clearance, e.g. $C/R \gg 5/1000$, large motion should be normal, unless no rubbing.

In addition, the design of AMB for effective vibration control requires the placement of AMB and position sensor near the most vibrating portion of eigen modes, as shown in Fig.7. Thus, the detection of large vibration is inevitable and should be allowable. This is the reason why AMB rotors require their own ISO standards.

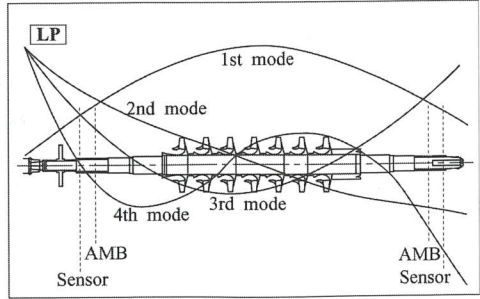


Fig.7 Eigen mode shape of LP rotor

4.2 Existing Standards

In the case of process compressor equipped with AMB, a customer can comply with API617 regulation^[3] which requests a low

$$L_v = 25.4 \sqrt{\frac{12000}{N_{MCS}}} \text{ and } L_v \leq 25.4 \mu\text{mpp}$$

where $N_{MCS} = \text{max. continuous speed, rpm.}$

For instance, if $N_{MCS} = 9500 \text{ rpm}$,

$$L_v = 25.4 \mu\text{mpp}$$

vibration level as follow:

If the existing ISO standard concerning the journal vibration of turbomachines equipped with the oil-film bearing is applied, the vibration zone limit values would be as follows:

$$\text{Zone A } S_{pp} = 4800 / \sqrt{N}$$

ISO 7919-3 Coupled industrial machine^[2]

where $N = \text{the maximum (trip) speed}$

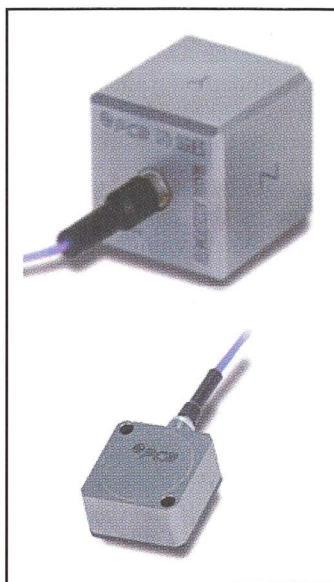
For instance, if $N = 12500 \text{ rpm}$,

$$S_{pp} = 43 \mu\text{mpp for Zone A.}$$

These existing values are too small compared with bearing clearance, e.g., $2C = 500 \text{ mm}$. Existing standards for oil-film bearings are too strict for AMB rotors.

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4.3 Scope of Part 2

This part 2 provides general guidelines for measuring and evaluating AMB rotor with respect to the following two indices:

- shaft vibratory displacement measured at or close to the AMBs, and
- working current and voltage measured in magnetic coils or power supply amplifiers.

Both indices are measured under nominal operating steady-state conditions in house and/or on site. Since AMB equipment has its own position sensor within the feedback control system, the detected displacement value is subject to this standard and no additional sensor required.

This part applies to industrial rotating machines generating or consuming nominal power greater than 15 kW, and is not limited by size or operational rated speed. The large turbomachine like compressors, pumps, turbines, generators and other rotors supported by AMBs are included in this standard, and small AMB rotors such as TMPs, spindles, etc are excluded.

4.4 Evaluation Criteria

One of the features of the AMB is large clearance. If the rotor does not contact with the stator, even large vibration is still normal. Therefore, the AMB vibration criteria are derived from the following parameters:

- No contact with bearings, adjacent displacement sensors and touch bearing,
- No contact with stationary labyrinth seals and
- limit of current compared to power amplifier capability

Then the minimum radial clearance, noted $C_{\min}$, can be defined by the minimum gap when statically moving the rotor in any radial direction. The touch down bearing gap is generally set to be $C_{\min}$ by design.

The index $D_{\max}$ is the maximum value of shaft vibratory displacement measured by AMB sensors under the steady-state operation. This index should be compared with the minimum clearance $C_{\min}$ for evaluating space margin to prevent the rotor/stator contact.

Finally, the corresponding zone limit table for AMB, established from international experience, is given in Table 1.

Concerning the evaluation of coil current and voltage margin, as of this writing, there is not enough data to define a standard that guarantees the reliable operation of the electrical part of the AMB systems. These current and voltage values are thus not put into the present evaluation.

5. ISO CD14839-3 Stability Margin

5.1 Background

The AMB feedback control provides positive stiffness and positive damping so that the AMB operates to maintain the rotor at a centered position. The AMB rotor should be thus evaluated for damping and stability properties for all relevant operating modes. There are two parts to this assessment.

First, the run up-behavior of the system is to be evaluated by the damping effect concerning critical speeds within the rotational speed range. It corresponds to the sensitivity to unbalance excitation at each critical speed measured in a rotation test. For example, when the unbalance vibration response is measured as shown in Fig. 8, the sharpness of each peak amplitude is evaluated, as called Q-factor evaluation. These damping requirements for an AMB system during run-up are covered by ISO 10814 (Q-factors)^[7,8], and are not the subject of this part 3.

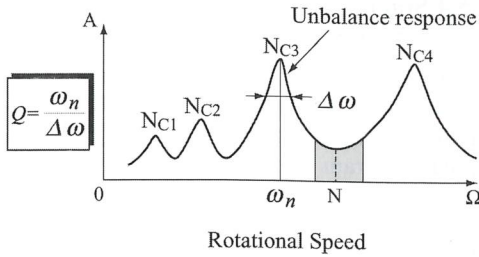


Fig.8 Unbalance response curve

The second part, which is covered by this part 3, deals with the stability of the system for safe and reliable operation of the AMB rotor system from the viewpoint of the feedback control. This analysis is critical since it ensures a minimum level of robustness with respect to system variations (e.g. system variations caused by high temperature) and disturbance acting on the rotor (e.g. higher frequency noise). This standard specifies a particular index to evaluate the stability margin and delineates measurement of this index.

Zone Limit	Displacement $D_{\max}$	Case study if $C_{\min}=230 \mu\text{m}$	
A/B	$< 0.3 C_{\min}$	$< 69 \mu\text{m}$	$< 138 \mu\text{mpp}$
B/C	$< 0.4 C_{\min}$	$< 92 \mu\text{m}$	$< 184 \mu\text{mpp}$
C/D	$< 0.5 C_{\min}$	$< 115 \mu\text{m}$	$< 230 \mu\text{mpp}$

Table 1 Proposed criteria of Zone limits

Note: the definition of each zone

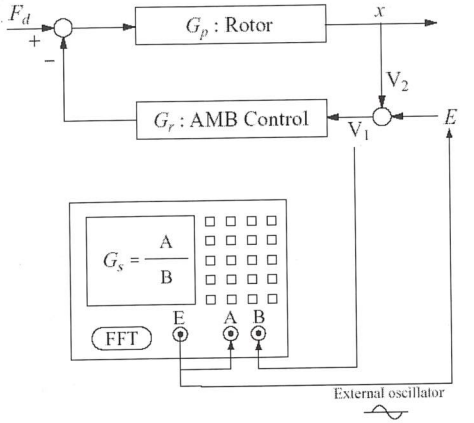
- Zone A: The vibration of newly commissioned machines would normally fall within the zone.
- Zone B: Machines with vibration within this zone are normally considered acceptable for unrestricted long-term operation.
- Zone C: Machines with vibration within this zone are normally considered unsatisfactory for long-term continuous operation. Generally, the machine may be operated for a limited period in this condition until a suitable opportunity arises for remedial action.
- Zone D: Vibration, eccentricity and current within this zone are normally considered to be sufficient ly severe to cause damage to the machine.

5.2 Scope of Part 3

The category of rotating machines to be subject to follows part 2. This standard concerns the system stability measured during normal steady-state operation in house and/ or on site. The in-house evaluation is an absolute requirement for shipping of the equipment, while the execution of on-site evaluation depends upon mutual agreement between the purchaser and vendor.

5.3 Transfer Functions

The closed loop of AMB-rotor system of Fig. 5 is simplified, as shown in Fig.9, using the notation of the transfer function G_r of the AMB control part and the transfer function G_p of the plant (rotor). At a certain point of this closed loop network, we can inject excitation as harmonic or random signals form E_1 and measure the response signals V_1 and V_2 directly after and before the injection point, respectively. By using a 2-channel FFT analyzer, we can measure the ratio among these three signals in the frequency domain to obtain following transfer functions.



$$G_s = \frac{1}{1 + G_o} = + \frac{V_1}{E}$$

Sensitivity function

$$G_o = - \frac{V_2}{V_1}$$

Open loop transfer function

$$G_c = \frac{G_o}{1 + G_o} = - \frac{V_2}{E}$$

Closed loop transfer function

Fig. 9 Measurement of transfer functions

The ratio of these V_1 and V_2 signals provides an open loop transfer function G_o :

$$G_o(s) = -V_2 / V_1(s) \quad (1)$$

At the same time, we can measure the closed loop frequency transfer function G_c and the sensitivity function G_s by the ratio of the following formulas:

$$G_c(s) = -V_2(s) / E(s) \quad (2)$$

$$G_s(s) = V_1(s) / E(s) \quad (3)$$

These transfer functions are mutually compatible with the open loop G_o in the following mathematical expression:

$$G_c = \frac{G_o}{1 + G_o} \Leftrightarrow G_o \Leftrightarrow G_s = \frac{1}{1 + G_o} \quad (4)$$

5.4 Stability index for sensitivity function

A typical example of the open loop transfer function $G_o(j\omega)$ is displayed on Bode diagram in Fig.10. It can be rearranged on a polar diagram with the form of amplitude $|G_o(j\omega)|$ and phase $\angle G_o(j\omega)$ as shown in Fig.11. Such a diagram is called Nyquist plot of the open loop transfer function. Since the characteristic equation is provided by $1 + G_o(s) = 0$, the distance between the locus and the critical point $(-1,0)$ on the Nyquist plot is directly related to the stability margin. Generally it can be stated that the larger the curve's minimum distance from the critical point, the greater is the system stability margin.

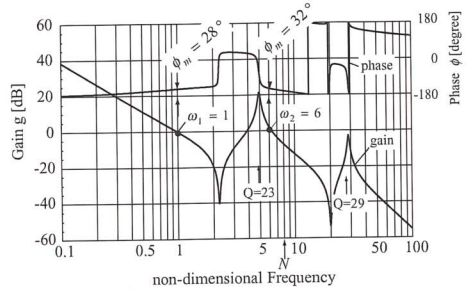


Fig. 10 Open loop transfer function

The shortest distance measured from the critical point is indicated by $AB = D_{\min}$, where a circle of radius $D_{\min}$ centered at $(-1,0)$ is tangent to the locus. For this example, the gain margin is the distance AG (an intersection between the locus and the real axis) and the phase margin is the angle $\angle AOP$ (between the real axis and a line extending from the origin to the intersection between the locus and the unit circle).

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

6. Case Study [5,6]

As stated in the literature [3], an AMB centrifugal compressor was installed at a refinery plant as shown in Fig.13. The vibration level measured at the commission is shown in Fig.14. This machine with the minimum clearance $C_{\min} = 230$ mm is judged by Zone A, referring to Table 1 of ISO FDIS 14839-2.

One of the open loop transfer functions is shown in Fig.15. By the rearrangement of the open loop transfer function to the sensitivity function according to Eq.(4), we find the maximum value of sensitivity gain of 6dB. The stability margin of this machine is then categorized in Zone B, referring to Table 2 of ISO CD 14839-3.

This machine has normally been operating without major trouble in approximate 10 years since the installation in December of 1992.

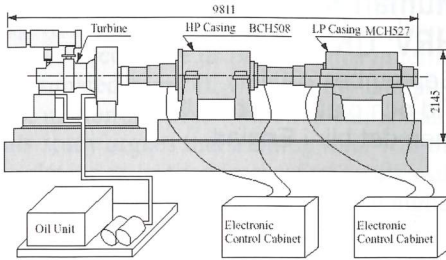


Fig. 13 arrangement of centrifugal compressor with magnetic bearings

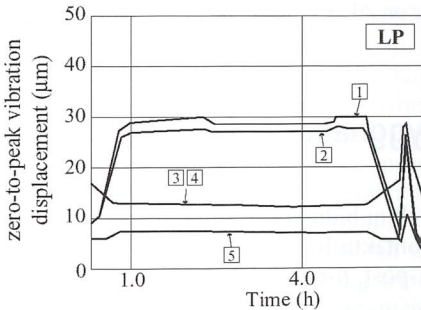


Fig. 14 Field data of the LP rotor (during start-stop)

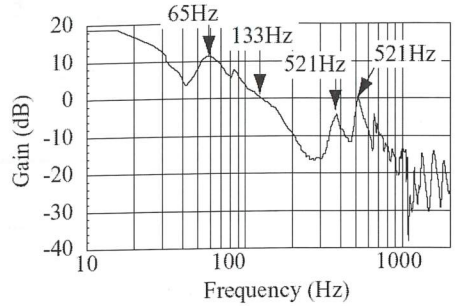


Fig. 15 Open loop transfer function (0 rpm, in-house test)

7. Conclusion

ISO TC108/SC2/WG7 AMB project and the corresponding domestic committee JSME/ATS 10-25 has been developing ISO standards of ISO 14839 series. The part 1 "Vocabulary" was already published and the part 2 "Evaluation of vibration" is reaching at the final stage prior to the publication. Part 3 "Evaluation of stability margin" has gained international consensus on the sensitivity index to propose the draft for ISO circulation and voting.

We hope that our effort to develop the international standards encourages the AMB application to the wide range in turbomachine and our information contributes to each phase of designing, operation and maintenance among AMB turbomachinery business so that all mystery will be erased. ■

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Ride Comfort and ability to work in tilting trains

Report from a train study carried out with NSB, Norway

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ABSTRACT

Short train journeys, good ride comfort and good ability to work in a train are some of the main requirements for the potential train passenger. High speed, especially in curves, often means decreased ride comfort and ability to work. Before introducing a new tilting train, the Norwegian State Railways wanted to compare its performance in terms of estimated comfort, work ability and possible motion sickness with a non-tilting train. This study argues that a tilting train will offer almost as good ride comfort levels estimated by subjects, while maintaining higher speed in curves.

1 Introduction

Throughout the world, railway companies are searching for ways of increasing train performance, for example in terms of speed and comfort. However, most countries have a significant mileage of curved track which limits speeds, and thus certain measures must be taken to shorten travel times. High speeds in curves will give uncomfortably high lateral accelerations and jerks. One solution is to build straight lines, another is to tilt the car body inwards in curves. Tilting trains have also been shown to fulfil requirements of track forces and stability and therefore they are coming into use in Europe and elsewhere

Early tests have shown that this has a favourable influence on ride comfort. Other tests have estimated ride comfort in curve transitions and full curves. This study is concentrated on how test subjects are judging or estimating their overall comfort and ability to work. Another issue of interest is what influence tilting trains have on nausea and symptoms of motion sickness.

2 Methods and material

The main hypothesis of the test was that tilting trains with their higher speed in curves are not degrading their passenger estimation of ride comfort and ability to work compared with a conventional train. The design idea was to create conditions where lateral accelerations and roll motions perceived by the test subjects could be separated. Normally these two quantities are coupled with the selected tilt compensation. By selecting different speeds in curves with different tilt compensations, it is possible to find conditions where high tilt velocities are combined with low perceived lateral acceleration by the subjects and vice versa. Therefore, three different speed levels were selected with a corresponding lateral acceleration in track

level of 1.0, 1.4 and 1.8 m/s², respectively. The condition *I* and *II* used the 1.8 speed level with 53 % and 45 % effective compensation in the tilt system, respectively. The condition *III* and *IV* used 1.4 speed level with 53 % and 62 % compensation. The reference condition

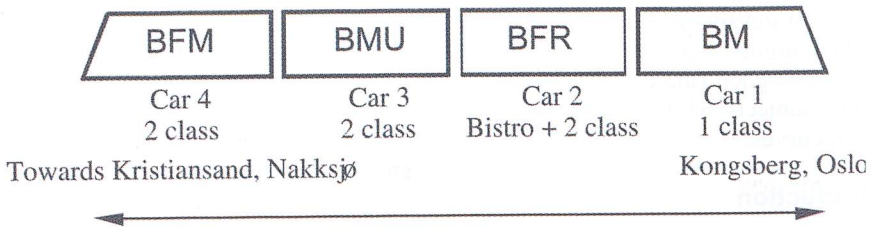
(*V*) was the 1.0 speed level without compensation, i.e. non-tilting. Table 1 gives an overview of the test conditions and the corresponding level of the some specific quantities.

Figure 1 Principal layout of the test train

Condition	<i>I</i>	<i>II</i>	<i>III</i>	<i>IV</i>	<i>V</i> (reference)	Units
Lateral acc. in track plane	1.8		1.4		1.0	[m/s ²]
Cant deficiency	280		220		150	[mm]
Tilt compensation ratio	53†%	45†%	53†%0	62†%	0†%	-
Tilt angle ⁱ	5.6	4.7	4.7	5.4	0	[°]
Tilt angle velocity ⁱ	2.8	2.35	2.1	2.5	0	[°/s]
Roll velocity ⁱ	5.7	5.2	4.8	5.1	2.1	[°/s]
Lateral acc. in car body ⁱ	0.85	1.0	0.7	0.6	1.0	[m/s ²]

Remark: i Maximum values

Table 1 Compilation of the conditions *I* - *V*. For the different conditions the following quantities are noted: lateral acceleration in the track level and used tilt compensation ratio as well as typical maximum tilt angle, lateral acceleration in car body and roll velocity



2.1 Tilting train

The newly delivered tilting train, class BM73, belonging to NSB was used in this test, see Figure 2. This train consists of four cars, see Figure 1. The train allows a lateral acceleration at the track plane of 1,8 m/s² corresponding to a cant deficiency of 280 mm. The main focus was on the two middle cars: one bistro car and a second class car, since the end cars might have less ride comfort due to tilt motions. These cars also used different tilt compensation ratios in order to

test two conditions at the same time.



Figure 2 NSB test train of class BM73 "Signatur"

2.2 Test runs

Tests were conducted with five test runs, one run per day except for one occasion with two runs on one day. Test runs lasted altogether somewhat more than 2 hours plus stop time at the reversing station. The test took place in October 1999. During the test runs, the conditions *I* and *II* were combined with different compensation in car 2 and 3, condition *III* and *IV* combined in the same way and condition *V* was tested alone.

2.3 Test line section

The chosen test line section (part of the south-west coast main line between Oslo and Kristiansand) was the line between the stations Kongsberg and Nakksjøe, which is about 94 km long and extremely curved. About 20 % of the track length has curve radii less than 400 m, another 8 % of the track length has curves with radii between 400 – 600 m and 30 % of the track length consists of transition curves. About 18% of the length is straight track. Reverse curves are typical. Typical lengths of transition curves and curve elements are between 60 – 80 m corresponding to a time of 2 – 3 s for passing the element.

2.4 Test subjects

Test subjects were from a nearby college with 32 to 60 subjects per test run. Their mean age was 22.4 years and there was about equal distribution between females and males in each test run.

2.5 Scales

Ride comfort and ability to read and write was estimated on a five point scale from *very bad* (1) to *very good* (5). Subjects were also asked to mark if they had any comfort complaints for lateral or vertical vibrations, large lateral forces or rocking motions on a scale of *no complaints* (0), *complaints* (1) and

strong complaints (2). Motion sickness was estimated at the same time as nausea on a five point scale from *no symptoms* (0) to *very nauseated* (4).

Subjects were asked to estimate their comfort, ability to work and degree of nausea four times during the test run: twice at the intermediate station (Norda≤gutu), at the reversing station (Nakksjøe) and at the end station (Kongsberg).

2.6 Measurements

Car nr 2 and 3 were equipped with a box containing a lateral and a vertical accelerometer together with a roll velocity gyroscope. These signals, together with tilt angles for the middle cars, lateral acceleration at bogie level and speed, were tape-recorded for later analysis.

3 Results

The overall results are that the subjects estimated their ride comfort levels as good or better, see Figure 3. However, the standard deviation of ride comfort levels are relatively high (0.4 - 0.8 comfort units) and most differences between the conditions are to be found within one standard deviation. The mean comfort levels seemed to increase and culminated on the third test part as the subjects were getting used to this situation.

In order to explain any possible influence between different factors on comfort, results from the first test part is used, where the largest differences in comfort levels exist. Possible factors are different speed levels, tilt compensation, lateral acceleration in car body etc. An analysis of variance (ANOVA) reveals significant effects, if any. The best significant model is to use tilt compensation ($p < 0.02$) but the coefficient of determination is very low ($r^2 = 0.04$).

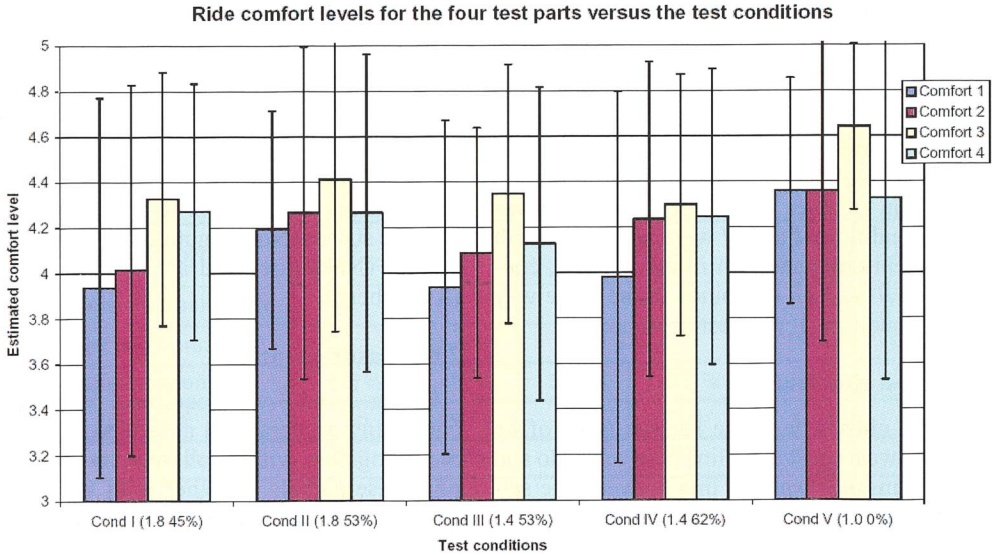


Figure 3 Estimated mean ride comfort level for the five test conditions together with standard deviation for the four test parts. Comfort levels are 3 = Neither good nor bad comfort, 4 = good comfort and 5 = very good comfort.

The non-tilting train condition *V* (compensation 0 %) gives the best estimated ride comfort, followed by compensation values of 45 % and then 53 % and 62 % at the same ride comfort level, see Table 2.

are high for non-tilting condition *V* and for condition *II*. For rolling motion, the complaints levels are the lowest for the non-tilting condition but 0.5 – 1 times more for tilting conditions.

Tilt compensation	Mean ride comfort ± standard error
0	4.36 ± 0.13
45	4.19 ± 0.13
53	3.94 ± 0.06
62	3.98 ± 0.10

Table 2 Mean ride comfort levels with the corresponding standard error for the different tilt compensation levels in the first test part.

3.1 Comfort disturbances

A more complex pattern is the result from comfort disturbances, see Table 3. Here the subjects complain more about lateral vibrations in tilting conditions at higher speeds, complaints about lateral (centrifugal) forces

3.2 Ability to work

For the ability to work the mean value for the test parts are 3.6 for condition *I* and 3.8 for condition *II – IV* and 4.1 for condition *V*. It is not possible to find any statistically significant differences between the conditions.

Condition	Lateral vibrations	Lateral forces	Rocking motions
<i>I</i> (1.8, 45%)	0.58	0.31	0.30
<i>II</i> (1.8, 53%)	0.38	0.54	0.29
<i>III</i> (1.4, 53%)	0.64	0.34	0.41
<i>IV</i> (1.4, 62%)	0.70	0.20	0.33
<i>V</i> (1.0, 0%)	0.48	0.48	0.19

Table 3 Mean level of comfort disturbances for each condition. (0 = no complaints, 1 = complaints & 2 = strong complaints)

4 Discussion and Conclusions

When discussing the estimated ride comfort for this test in Norway, one should have in mind the extreme curved track on this line. Typical time in a transition curve is 2-3 s, then a full curve with 2-10 s, another transition curve with 2-3 s, then into a reverse curve and a repeated sequence again. This means that the tilting system is working almost all the time, and at its full tilt velocity.

In this case, the tilt motions affect the estimated ride comfort as well as motion sickness and nausea. The increasing estimated comfort levels are probably reflecting that subjects are getting used to the tilt/roll motions and the decline on the part 4 reflect tiredness.

From a strict point of view, discomfort in each transition curve according to P_{CT} is less with a tilting train compared with a non-tilting train, in spite of higher curve speed, i.e. the local ride comfort is better with a tilting train.

A comparison with a Swedish test on X2000 done on much less curved lines and at higher speeds (180 – 200 km/h) shows a similar pattern. This test used two compensation values 70 % and 55 %. Two conditions used the 55 % compensation, one with a limited tilt acceleration and the other a limited tilt velocity. In general, a reduced compensation increases both perceived lateral acceleration for the passengers and lateral jerk but decreases tilt/roll motions. This has the overall effect that the P_{CT} values will be increased with a reduced compensation value. In this study (X2000), the subjects' estimated ride comfort for the whole test part were higher with a lower compensation, see Table 4.

Cond.	Estimated ride comfort mean (Std. dev.)	Lateral acc. [%]	Estimated working ability mean (Std. dev.)
A	4.08 (± 0.20)	58	4.05 (± 0.23)
F	4.22 (± 0.32)	54	4.12 (± 0.22)
G	4.19 (± 0.27)	54	4.19 (± 0.23)

Remark: Condition A had 70% compensation and condition F had 55% compensation with a limited tilt acceleration and condition G also 55% compensation but limited tilt velocity.
Std.dev. = standard deviation

Table 4 Estimated ride comfort and ability to work as well as comfort complaints from lateral acceleration

4.1 Conclusions

Estimated ride comfort from subjects for a whole test part or journey may not be the same as comfort disturbances or complaints from transition curves. The reduction of P_{CT} values are usually large when introducing a tilting train, which corresponds to a better curving performance. However, this enhanced performance may not be fully utilized due to effects probably from tilt motions. Lower tilt compensations may solve this problem with the benefit of some reductions in symptoms of motion sickness. A careful choice of tilt compensation due to differences in lines and curves may be necessary.

4.2 Further research

This kind of general or overall ride comfort estimations have not been much investigated. The relationship between ride comfort estimations and nausea should be investigated and may give new insight of the construction of a better tilt control system and high performance trains.

A similar research project at KTH and VTI is looking into the problem of comfort and ability to work/write/read caused by low-frequency lateral vibrations. Some earlier results are available. ■

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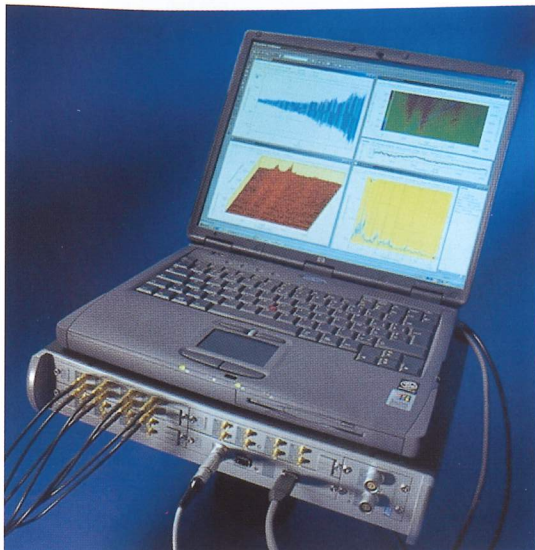
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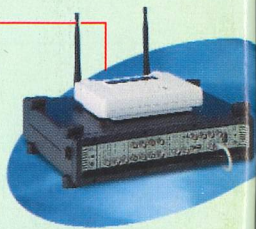


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