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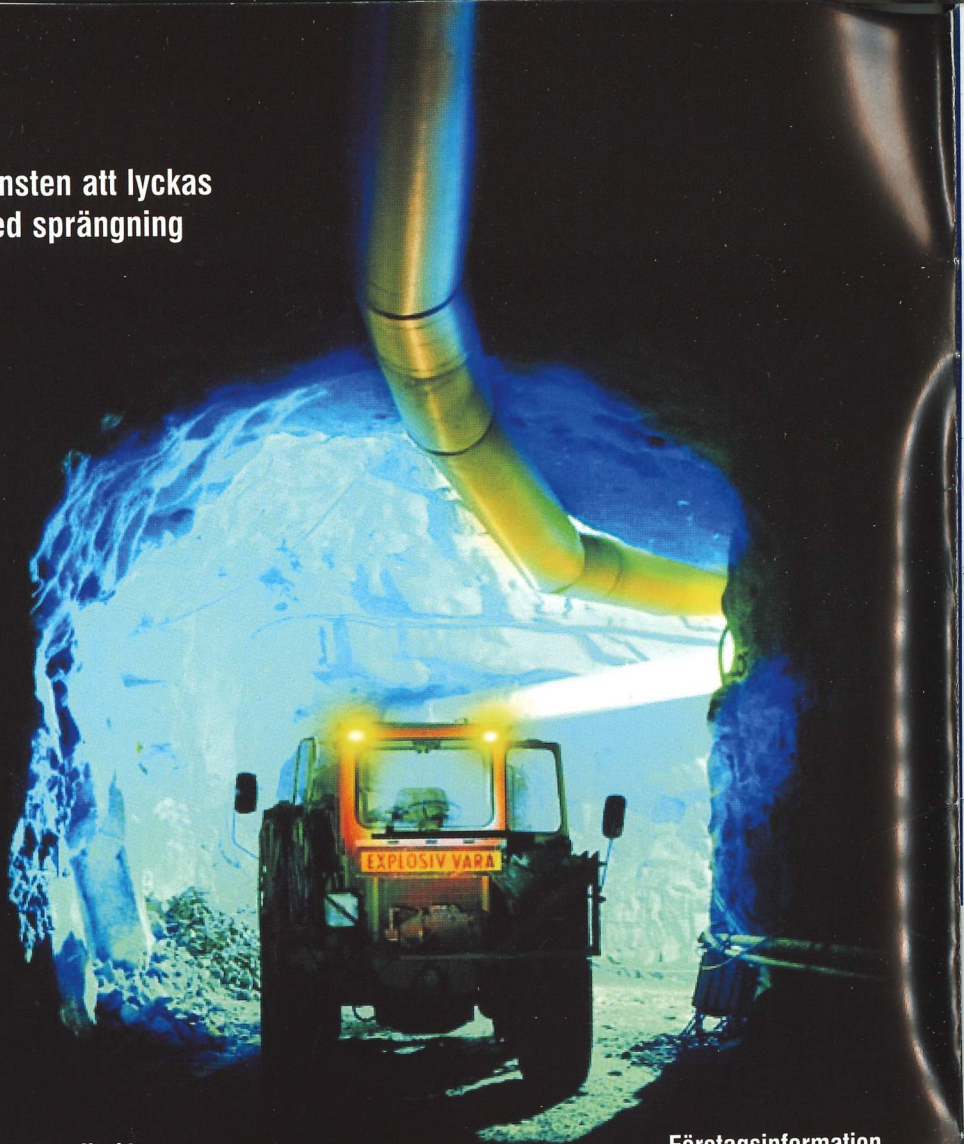


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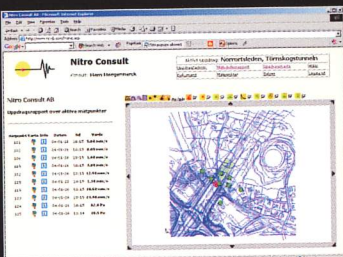


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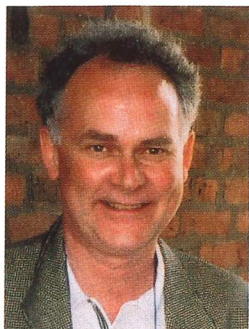
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Hur märks vibrationer?



Vibrationer i strukturer kan kännas och kan höras indirekt. Vibrationer med stort utslag kan ses. Om de är långsamma kan man se en fram-och-återgående rörelse. Med hjälp av mätinstrument kan man indirekt registrera vibrationer.

Ibland kan man se utmattningsskador på maskiner och byggnader som kan misstänkas vara orsakade av vibrationer. "Vita fingrar" kan vara orsakade av vibrationer.

Skandinaviska Vibrationsföreningen erbjuder i höst tre olika symposier eller temadagar som belyser några av ovanstående aspekter av vibrationer.

Arbetsutskott 4, som ansvarar för vibrationers inverkan på människan, anordnar 27 oktober en temadag om "Hälsorisker från vibrationer i arbetsmiljön". Detta ämne är synnerligen aktuellt med tanke på de kommande föreskrifterna om vibrationer från Arbetsmiljöverket.

På Vibrationsdagen 23 november är temat "Ljud ger vibrationer – vibrationer ger ljud".

I anslutning till Vibrationsdagen anordnar arbetsutskott 3, som ansvarar för information om trafik- mark och byggnadsvibrationer, 24 november en temadag om "Byggnadsvibrationer".

Innehållet i temadagarna presenteras på annan plats i tidningen och på www.svib.se.

Jag hoppas att innehållet lockar många och kommer att ge nyttig kunskap att bygga vidare på. ■

Martin Almgren

Ordförande,

Skandinaviska Vibrationsföreningen,

e-post: martin.almgren@ingemansson.se

Förslag till föreskrifter om Vibrationer samt allmänna råd om tillämpningen av föreskrifterna

Arbetsmiljöverket har tagit fram förslag till föreskrifter om vibrationer baserade på Europeiska unionens (EU:s) direktiv 2002/44/EG om arbetstagares exponering för risker i samband med vibrationer i arbetet.

Förslaget med konsekvensutredning finns att tillgå på verkets webbplats, www.av.se under rubriken *Regler*. Där finns även förslag på föreskrifter om Buller.

Den intresserade kan lämna synpunkter till mig så kan jag sammanställa och skicka till Arbetsmiljöverket. Det går också bra att lämna synpunkter direkt till Arbetsmiljöverket senast 4 oktober 2004.

Jag vill påminna om SVIBs temadag 27 oktober. Den kommer att behandla föreskrifterna.

Martin Almgren

SVIB styrelse

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AU 4 Vibrationers inverkan på människan

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AU 5 Forskning och utbildning inom vibrationsområdet

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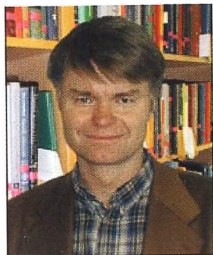
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Olof Friberg

Olof Friberg, Göteborg, har hastigt avlidit 50 år gammal. Närmast anhöriga är maken Åsa och barnen Gustaf, Kristina och Erik.



Olof, som var göteborgare, utbildade sig efter studentexamen till marinofficer i reserven. Han började att studera skeppsbyggnadsteknik på Chalmers 1974, vilket sedan ledde till civilingenjörsoch dok-

torsexamen. Han tjänstgjorde därefter vid Institutionen för Byggnadsmekanik som forskare och universitetslektor, samt från 1998 som biträdande professor. Han utnämndes 2002 till professor i fordonsmekanik.

Olof var en internationellt erkänd och framstående forskare inom området fordonsdynamik, speciellt datorsimulering av fordonsvibrationer. "Fordonen" kan till exempel vara personbilar, lastbilar, satelliter eller flygplan. Olofs forskning utmärktes av att han bemästrade kombination av matematiska metoder, data programmering och praktisk simulering inom industrin.

Även utanför Chalmers hade Olof en omfattande verksamhet. Han samarbetade med forskare vid University of IOWA, USA, för utveckling av simuleringsprogram, som i dag används i industriprojekt i många länder. Han fungerade sedan 1997 som akademisk handledare för projektet "Optimering av komfort i bussar" vid Scania Buses & Coaches i Katrineholm. Han har arbetat med ett flertal industriprojekt vid Volvo AB.

Som lärare var Olof mycket skicklig, engagerad och uppskattad. Han hade stor förmåga att för studenterna tala om vad som var väsentligt,

och vad man förväntades lära sig. Många examensarbetare kan vittna om hans positiva och engagerade sätt att handleda och examinera.

Vid sidan om jobbet hade Olof segling och musik som stora hobbies. Alla vi vid Byggnadsmekanik fick ta del av dessa då Olof omtänksamt ordnade segelutflykter för personalen och skrev humoristiska hyllningsvisor till oss vid jämna födelsedagar. Han behärskade många instrument och spelade bland annat keyboard i blues- och jazzgrupper. Vi minns Olof som en varm och omtänksam arbetskamrat som lämnat oss alltför tidigt.

MATS ANDER, PETER ERIKSSON

KENNETH RUNESSON, ALF SAMUELSSON

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Study of Crossbar Tyres in Rolling Conditions

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Abstract

Prediction of tyre/road noise requires a good understanding of the radiation properties of a vibrating tyre, but necessitates first of all an accurate prediction of the structural vibrations of a rolling tyre. In order to achieve this, a three-dimensional contact model was previously proposed and validated for the case of a tyre having only circumferential grooves [1].

The question arising in this work is whether the model is adequate for the prediction of the structural vibrations of a crossbar tyre, for which the tread pattern is made of regularly spaced bars in the width direction.

Acceleration measurements were made on crossbar tyres in rolling conditions in order to understand the effect of the crossbar impact on the vibration behaviour of the tyre. The measured acceleration signals were then

compared to the ones calculated with the three-dimensional contact model and an acceptable agreement was found for the position of the pitching frequencies resulting from the periodicity of the crossbars on the tyre and for the amplitude of the acceleration signals in the normal direction to the tyre structure.

Introduction

The study of crossbar tyres is a relevant one in the sense that crossbar tyres represent an extreme case of patterned tyres. In this work, accelerations signals measured in the normal and tangential directions on crossbar tyres are studied in rolling conditions and are compared with the acceleration signals calculated with a 3D contact model, previously validated for the case of an airplane tyre [1]. The model is good at predicting pitching frequencies induced by the periodicity of the crossbars. In terms of acceleration amplitude, the agreement is reasonably good with regard to the limited number of generation mechanisms implemented in the model

Measurements of acceleration

Measurements on crossbar tyres in rolling conditions were made in laboratory-controlled conditions on a test drum located in a semi-anechoic room. The measurements were made in the frame of the EC research project RATIN at Goodyear Technical Centre, Luxemburg. The measurement procedure concerns the recording of acceleration signals on the tyre structure in Till exempel har geofonen kapacitet att beräkna responspektra i realtid enligt förslaget ISO/WD 18431-4 Mechanical vibration and shock – Signal processing – Part 4: Shock response

spectrum analysis the normal and in the tangential direction while rolling. The crossbar tyre used for the measurements has 64 bars that are equally spaced. The tyre is fixed on an axle and is loaded by a static force of 3000 N on the drum before the drum is rolled.

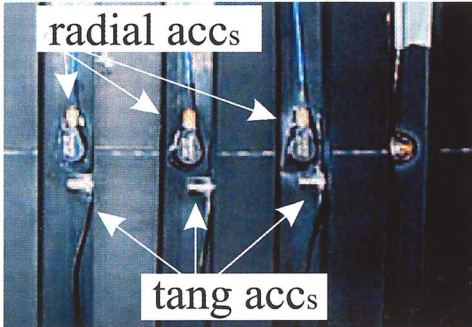


Figure 1: Mounting of the accelerometers

In order to perform acceleration measurements, accelerometers were glued on the tyre. To measure normal acceleration, accelerometers are glued on the tyre structure inside the grooves made by the bars, see Figure 1. For the tangential acceleration, the accelerometers are glued on the bars, in the tangential direction, which means that they mostly measure the tangential motion of the bars due to tangential forces and some rotational motion that can be due to the bending of the tyre structure.

Calculations

The acceleration from the vibrating tyre structure can be calculated in the normal direction with the time-domain formulated contact model developed by the authors [1]. The contact model needs the tread profile (obtained by scanning), the road roughness (laser-measured), the tyre structure response and some material data as input in order to solve the 3D contact problem for each time step. As an example, Figure 2 shows a snapshot of the tyre/road interaction at a certain time step.

Analysis and comparison

Figure 3 shows a typical time acceleration signal measured in the normal and tangential directions for the case of the tyre rolling at 80 km/h on the ISO road. The figure shows the time averaged acceleration over all tyre loops completed during measurements and the time period corresponds to a fully completed tyre loop. At $0 = t$ s, the accelerometer is on the top of the tyre ($0 = t$ s) and moves through the front part of the tyre, approaching the leading edge (0.04 s). From 0.04 s to 0.05 s, the accelerometer travels through the contact patch (region 2). After 0.05 s, it travels through the back of the tyre towards the top again.

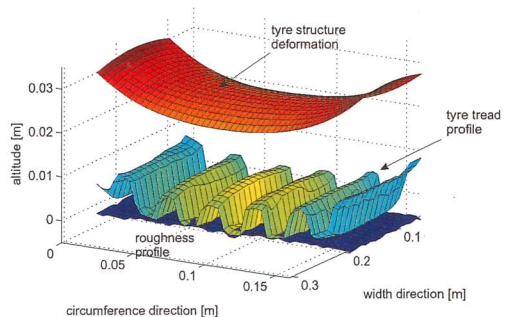


Figure 2: Snapshot of the tyre/road interaction between the crossbar tyre and the ISO road at 80 km/h

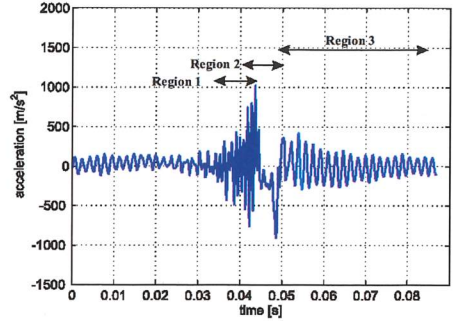
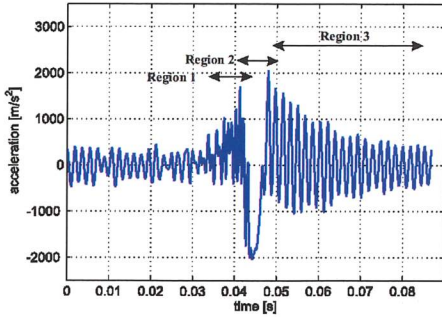


Figure 3: Normal (left) and tangential (right) acceleration time signals recorded for the tyre rolling at 80 km/h on ISO road

While completing an entire loop, the normal accelerometer records an acceleration that is composed of three parts. The first part is the acceleration in stationary conditions, which is the shape of the acceleration signal that would be obtained in the case of the smooth tyre rolling on a smooth road (see [2]). The second part is composed of the waves produced ahead of the accelerometer by the tyre/road contact, which propagate in the opposite direction of the tyre rotation. This signal is the compressed wave visible in the part of the signal called region 1.

Finally the third part is composed of the waves produced behind the accelerometer, propagating in the same direction as the tyre rotation. These waves have biggest amplitude from the trailing up to the top of the tyre, in the part of the signal called region 3.

Figure 4 shows the comparison between measured and calculated power spectrum density of the normal acceleration signals for several rolling speeds. At 60 km/h, a group of two peaks, one at n, lf below 500 Hz and one at n, uf around 700 Hz is visible. These two peaks are representative for the pitching frequency n, pf due to the periodicity of the

crossbar tyre influenced by the Doppler effect as

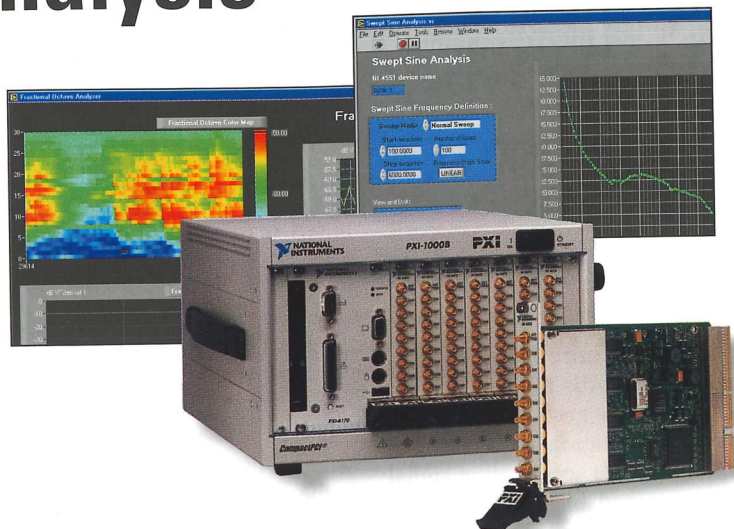
$$(1) \quad (f_{l,n}; f_{u,n}) = \frac{f_{p,n}}{1 \pm \frac{u}{c(f_{p,n})}}$$

where u is the rolling velocity and $() n, p f c$ is the wave group velocity. n, lf is characteristic for the wave propagating in region 3 whereas n, uf represents the wave present in region 1.

The spectrum below 250 Hz is characteristic for the acceleration in stationary conditions and is therefore not influenced by the Doppler effect. For higher frequencies, the signal is influenced by the higher harmonics of the pitching frequency.

The agreement between measurements and calculations is good at 60 km/h in terms of shape and amplitude. At all speeds, the low frequency region is well captured, meaning that the contact time and length are well predicted. At 80 km/h, the model fails in predicting the acceleration peak at n, uf . A reason for this can be that the measurement peak is built of two peaks, the upper one of the pitching frequency plus the lower one of the first harmonic of the pitching frequency, present at about the same frequency whereas

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the model predicts the peaks at different frequencies. At 100 km/h, the agreement is globally better than 80 km/h.

Generally, as rolling velocity increases, the acceleration peak corresponding to n , lf remains constant whereas the peak corresponding to n , uf decreases. This means that the vibration amplitude at the leading edge decreases. An explanation for the actual decrease could be the importance of damping effects as the excitation frequency increases. Additionally, the fact that this trend is both observed in the measurements and predicted by the model talks against the effect of tangential forces on normal tyre structure vibra-

tions since tangential force excitation is not implemented in the model.

The amplitude of the acceleration spectrum measured in the tangential direction (not shown here) decreases at n , uf for increasing rolling speed as for the normal direction, whereas the peak at n , lf increases. At this point of the study, it seems that the implementation of normal structure vibrations due to tangential excitation is not necessary since the model captures the right trends with normal force excitation only. A further step is the validation of the model for commercial tyres in order to verify this hypothesis.

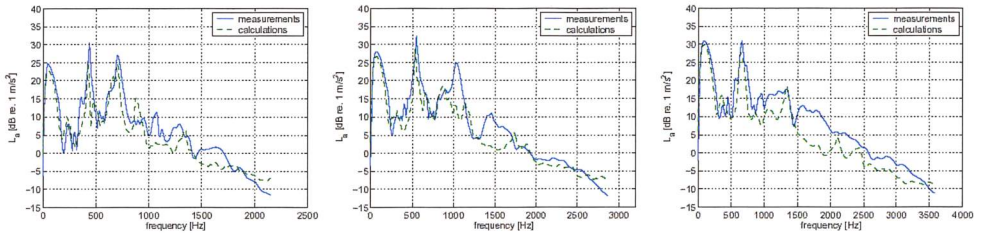


Figure 4: Comparison between the measured and calculated normal accelerations at 60 (left), 80 (middle) and 100 km/h (right)

Acknowledgements

The work presented in this paper has been financially supported by the EC research project RATIN, G3RD-CT 2000-00097.

Nonlinear Stochastic Response of a Shallow Cable

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The paper considers the stochastic response of geometrical nonlinear shallow cables. The response of the cable is investigated for a reduced two-degree-of-freedom system with one modal coordinate for the in-plane displacement and one for the out-of-plane displacement. At first harmonic varying chord elongation at excitation frequencies close to the corresponding eigenfrequencies of the cable is considered in order to identify stable modes of vibration. Dependent on the start conditions the system may enter one of two states of vibration in the static equilibrium plane with the out-of-plane displacement equal to zero, or a whirling state with the out-of-plane displacement different from zero. Possible solutions are found both analytical and numerical. Next, the chord elongation is modelled as a narrow-banded Gaussian stochastic process, and it is shown that all the indicated harmonic solutions now become unstable with probability one. Instead the cable jumps randomly forth and back between the two in-plane and the whirling mode of vibration. A theory for determining the probability of occupying either of these modes at a certain time is derived based on a homogeneous, continuous time three state Markov chain model. It is shown that the transitional probability rates can be determined by first-passage crossing rates of the envelope process of the chord wise component of the support point motion relative to a safe domain determined from the harmonic analysis of the problem.

Introduction

Cables used as structural support elements of masts, towers and cable-stayed bridges are characterized by a sag-to-chord-length ratio below 0.01, which means that natural frequencies for the vertical and the horizontal vibrations are pair-wise close. Typically the supported structure is performing narrow-banded random vibrations, with the implication that the cables will be subjected to narrow-banded support-point motions. Especially the component of the relative support-point motions along the chord is of importance for the cable dynamics. Primarily, the said displacement component is exciting the cable as an additive load term in the in-plane equation of motion. Additionally, the component appears as parametric (multiplicati-

ve) load terms in both the in-plane and the out-of-plane equations of motion. If the center frequency of this excitation is close to a pair of cable eigenfrequencies strong 1:1 internal resonance vibrations in the involved modes take place. Since, the eigenfrequencies of the symmetric in-plane modes is increasing with the Irvine stiffness parameter, similar internal resonance couplings involving the 1st symmetric in-plane mode, the 2nd antisymmetric in-plane and the 2nd antisymmetric out-of-plane modes may occur at a somewhat higher sag-to-chord-length ratio corresponding to the so called cross-over eigenfrequency of the cable. At a still higher sag-to-chord-length ratio nonlinear interactions between the 1st symmetric mode and

the 2nd symmetric out-of-plane modes may occur. Similar interaction patterns exist for the 2nd and higher order symmetric in-plane modes.

For shallow cables additional internal resonances caused by non-linear or parametric coupling terms occur, if the excitation frequency is a multiple of the eigenfrequencies of the considered pair of modes (superharmonic response), or a fraction of these frequencies (subharmonic response). Normally, the subharmonic response is more pronounced. Also coupled or uncoupled harmonic response with the frequency fraction $2/3$ (combinatorial harmonic response) may occur.

In this paper a Galerkin based reduction of the nonlinear shallow cable equations to a two-degrees-of-freedom system using the fundamental in-plane and out-of-plane linear modal modes as a functional basis. At first possible stable periodic motions are identified under 1:1 internal resonance due to harmonically varying support points. These turn out to be a whirling and two in-plane modes of vibration. The in-plane modes are characterized with large and small amplitudes, respectively. Next, the support point motion is changed to a narrow banded excitation with center frequency W and envelope process comparable to the frequency and amplitude of the harmonic excitation. During the stochastic excitation the modal coordinates perform stochastic vibrations around the identified stable periodic motions, which form a kind of mean value responses. Although stochastic, these motions are distinguishable by qualitatively different levels of amplitudes and phases, for which reason they will be referred to as "states". During narrow banded stochastic excitation it turns out that the motion in all states is unstable with probability 1 with the implication that jumps takes place to one of the two other possible states. Correspondingly, the response of the

cable is modelled as a three state homogeneous continuous time Markov chain, [1]. The transitional probability rate for jumps from state i to state j is denoted v_{ij} , where the index values $i = 1, 2, 3$ denote the plane mode with small amplitude, the plane mode with large amplitude and the whirling mode, respectively. Hence, v_{13} represent the probability rate of jumping from the plane mode with small amplitude to the whirling mode.

The mechanisms causing jumps between the described states are identified. It is shown that jumps are related to out-crossings of the envelope of the nondimensional chord elongation at barrier levels, which can be determined from the analytic solutions to the harmonic excitation.

The vibrations of the cable is caused by elongations of the chord due to forced motions of the support point. Assuming a parabolic approximation to the equilibrium suspension, and retaining up to cubic geometric nonlinear terms, the equation of motion have been derived in [2].

Harmonic Analysis

Fig. 1 shows a cable with the chord placed along the horizontal x -axis and the static equilibrium state $y(x)$ in the (x, y) -plane. The cable is supported by springs k_1 and k_2 modelling the flexibility of the support point. The material point of the initial cable configuration with the coordinates $(x, y, 0)$ undergoes the displacement vector $[q_1(t), q_2(t)]^T$ in the coordinate directions at the midpoint of the cable with $q_1(t)$ and $q_2(t)$ being the out-of-plane and in-plane components, respectively. The cable is elastic with the elasticity modulus E and cross-sectional area A . The sag, the chord length, the total weight of the cable and the horizontal cable force in the equilibrium state are denoted as f , l , W and H , respectively.

The data of the considered cable refers to the longest stay in the cable-stayed bridge across the Øresund between Denmark and Sweden. The supports are assumed fixed, corresponding to $k_1 = k_2 = \infty$. The stiffness of the cable is $EA = 2.17 \cdot 10^9$ N, and the equilibrium force $H = 5.5 \cdot 10^6$ N. The chord length is 260 m, and the cable mass per unit length is $m = 81.05$ kg/m, corresponding to a total weight of $W = 2.038 \cdot 10^5$ N. The referential amplitude of the chord elongation is $\Delta l_0 = 0.5$ m corresponding to $e_0 = 0.759$.

First a deterministic harmonic excitation with the circular frequency $\Omega \equiv \omega_1 \equiv \omega_2$ is considered, which means that

$$e(t) = e_0 \cos \Omega t + a. \quad (1)$$

Fig. 2 shows possible solutions of the amplitudes Q_2 of $q_2(t)$ as function of the excitation frequency Ω . Analytical solutions, which are stable in 3D, i.e. where the out-of-plane component $q_1(t)$ is free to move, have been indicated by an unbroken bold line, whereas stable analytical solutions in 2D, where the out-of-plane component is suppressed so $q_1(t) \equiv 0$, are marked with a thin unbroken line. Finally, unstable solutions are indicated by a dashed signature. The quality of the analytic solutions have been checked by numerical integration of the equations of motion for the stable branches, and have been indicated by a (∞) on the plot. In this case Q_2 have been obtained from rms values obtained as time average over one period $T = 2\pi/\Omega$. The stability analysis of the various solutions have been carried out by means of Floquet theory.

The branches indicated by p_1, p_2 and p_3 indicate the harmonic response in 2D. p_1 is stable up to a certain maximum excitation frequency Ω , where a fold instability takes place. p_3 is stable in certain disjoint sub intervals of the Ω -axis.

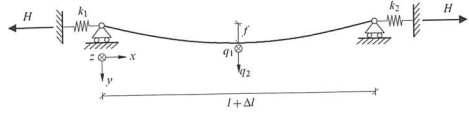


Figure 1: Cable in equilibrium configuration.

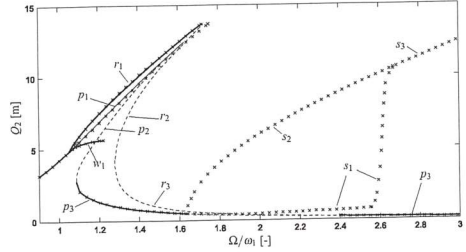


Figure 2: Q_2 during harmonic excitation as a function of excitation frequency Ω , $e_0 = 0.759$. Stable solutions in 3D ($\equiv$). Stable solutions in 2D ($\equiv$). Unstable solutions ($---$). Numerical solutions (∞).

In 3D the stable plane motion (i.e. $q_1(t) \dots 0$, but is free to move) is indicated by r_1 , whereas r_2 and r_3 signify corresponding unstable solutions. r_3 is formed by bifurcation from p_1 at a critical frequency and remain stable up to a certain maximum excitation frequency Ω , where a fold instability takes place. Attraction to r_1 takes place if the system is started with very small initial values $q_1(0) \mid 0$ or $'q_1(0) \mid 0$ along with relatively large values of $q_2(0)$ and $'q_2(0)$. If $q_1(0) = 'q_1(0) \dots 0$ the system is attracted to p_1 . The amplitudes of the mode of vibration p_1 and r_1 are comparable, whereas the phases are completely different.

w_1 indicate a whirling motion. This motion is also formed by bifurcation from r_1 which takes place at a slightly larger frequency than the bifurcation into the branch p_1 . The whirling motion remains stable up to a certain maximum frequency.

At the frequency in the vicinity of $\Omega \sim 2 \omega_1$, where p_3 is unstable, a stable branch s_2 exist. In this mode $q_2(t)$ is harmonic with the frequency $\Omega \cong 2 \omega_1$, whereas $q_1(t)$ is a subharmonic of the order $n = 2$, resulting in an eightshaped trajectory. s_1 indicates the corresponding plane subharmonic in 2D. No analytic solutions have been indicated for the subharmonic responses.

Narrow-banded Excitation

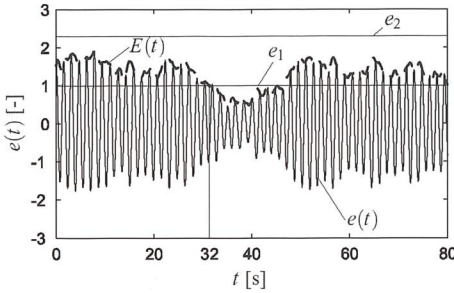


Figure 3: Narrow banded chord elongation process, $\mu=0.003$, $e_0 = 1.5$, $\Omega = 1.25 \omega_1$. Underlying process $e(t)$ (----). Envelope process $E(t)$ (-----).

The stochastic model for the non-dimensional elongation of the chord-length $e(t)$ is obtained as a second order filtration of Gaussian white noise

$$\ddot{e}(t) + 2\mu\Omega\dot{e}(t) + \Omega^2 e(t) = \sqrt{2\mu\Omega^3} e_0 W(t), \quad (2)$$

where μ is a bandwidth parameter, Ω is a circular center frequency and $W(t)$ is a unit Gaussian white noise process. A realization of $e(t)$ and the related envelope process $E(t)$ are shown on Fig. 3. e_1 and e_2 represent the minimum and maximum value of the amplitude e_0 where a certain periodic motion may exist under harmonic excitation with the circular frequency Ω . As seen at $t \sim 32$ s the upper envelope process performs a first passage out of the interval $[e_1, e_2]$ at

the lower boundary e_1 . We shall later see that such an out-crossing cause a transition from one periodic motion to another.

Homogeneous Continues Time Markov Chain

Let T_{ij} be the first passage time interval until transition to state j takes place on condition that the system starts in state i . Similarly T_i denotes the first passage time

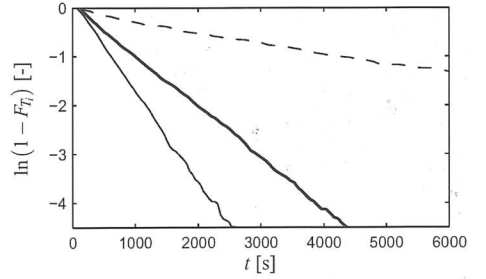


Figure 4: Monte Carlo simulation of distribution function for the time interval in a given state, $\Omega = 1.25 \omega_1$, $\mu = 0.0003$, $e_0 = 1.5$. Low-amplitude in-plane mode p_3 (—). High-amplitude in-plane mode r_1 (---). Whirling mode ω_1 (····).

interval of occupancy in state i until transition to any of the other states.

It is assumed that the transition from one state to another may be described by a three state homogeneous continuous time Markov chain.

Let j_1 and j_2 denote the possible transitions states from state i . Then the distribution function of T_i follows from

$$P(T_i > t) = P(T_{ij_1} > t \wedge T_{ij_2} > t) = \exp(-v_{ij_1}t) \exp(-v_{ij_2}t) \Rightarrow \quad (3)$$

$$\left. \begin{aligned} F_{T_1}(t) &= 1 - \exp(-(v_{12} + v_{13})t) \\ F_{T_2}(t) &= 1 - \exp(-(v_{21} + v_{23})t) \\ F_{T_3}(t) &= 1 - \exp(-(v_{31} + v_{32})t) \end{aligned} \right\}.$$

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The transition rates is then found from

$$v_{ij} = \frac{1}{E[T_{ij}]}, \quad (4)$$

where $E[T_{ij}]$ denotes the expected first passage time interval until transition. The results (3) have been verified by Monte Carlo simulation using the following procedure.

At first the system (??) is integrated with an equivalent harmonic excitation until stationarity is achieved in state i . Next the excitation is changed to a stochastic excitation by integrating (2) with initial values $e(0)$ and $\dot{e}(0)$ determined by the terminal values of the harmonic excitation. In the numerical procedure the unit white noise process is replaced by an equivalent broad banded Gaussian process (broken-line process). Fig. 4 shows plots of the resulting histograms based on 2500 sample values. $\ln(1-F_T(t))$ should map into a straight line as a function of time for an exponentially distributed random variable.

Triggering Mechanisms for Transition Between States

Fig. 2 shows the harmonic amplitude response in the resonance area as a function of the excitation frequency Ω at a fixed amplitude $e_0 = 0.759$. On Fig. 5 the corresponding amplitude plots of Q_2 have been shown, when the amplitude e_0 is varied, and the frequency Ω is fixed at $\Omega = 1.25 \omega_1$.

The response branch r_1 (state $i = 2$) only exists for excitation amplitudes e_0 in the interval $[e_{21}, e_{23}]$. As e_0 is decreased towards the limit e_{21} the maximum point (fold instability point) on Fig. 2 is achieved at smaller and smaller frequencies. For $e_0 = e_{21}$ the maximum value take place at a frequency equal to the excitation frequency Ω . If e_0 is decreased below e_{21} the branch r_1 cannot be achieved

at all at harmonic excitation with the frequency Ω , for which reason a jump to another state must take place, which is to the state p_3 ($i=1$) as shown on the figures. In the same way, as e_0 is increasing towards the limit e_{23} the bifurcation of the branch r_1 from p_3 will take place at increasingly larger frequencies. For $e_0 = e_{23}$ the bifurcation takes place at the excitation frequency Ω . If e_0 is increased above e_{23} no bifurcation to r_1 takes place at harmonic excitation with the frequency Ω , which means that r_1 does not exist at the frequency Ω for $e_0 > e_{23}$. Instead a jump to another state takes place, which in the present case is to the whirling mode w_1 ($i = 3$).

As seen the whirling mode w_1 ceases to exist at sufficient large values of e_0 , where the curve is continued into p_1 . The limit point, noticeable as a bend on the w_1 - p_1 curve, represents the highest value of e_0 , where bifurcation of w_1 from p_1 is possible. Due to the continues extension of the branches w_1 and p_1 we shall consider the combination of w_1 and p_1 as a single branch, which exists for e_0 in the interval $[e_{32}, \infty[$. As e_0 is decreased towards e_{32} the instability point of w_1 (the right-most point on Fig. 2) is achieved at smaller and smaller frequencies. For $e_0 = e_{32}$ the instability point is achieved at a frequency equal to the excitation frequency Ω . Consequently, if e_0 is decreased below e_{32} the branch w_1 cannot be achieved at all under harmonic excitation with the frequency Ω , for which reason a jump to another state takes place, which happens to the state r_1 at the amplitude e_{32} .

Similarly, The branch p_3 exists for excitation amplitudes e_0 in the interval $[0, e_2]$. As e_0 is increased towards the limit e_2 the instability point of p_3 (the leftmost point with a vertical tangent on Fig. 2) is achieved at larger and larger frequencies. For $e_0 = e_2$ the instability point is achieved at the excitation frequency. Consequently, if e_0 is increased

above e_2 the branch p_3 cannot be achieved at all under harmonic excitation with the frequency Ω , for which reason a jump to another state takes place. In the present case the jump takes place to w_1 with $e_2 = e_{13}$. As e_0 is increased the bifurcation point between p_3 and the subharmonic solution s_1 moves towards the left on Fig. 2.

Under the assumption of narrow banded excitation, the results show that the main triggering mechanism of transition from one state to another is the outcrossing of the envelope process of the non-dimensional chord elongations. If broad banded excitation is considered other triggering mechanisms may arise. One is the phase lag between the excitation and the response which also may trigger jumps between vibration modes.

Numerical Example

In this example the mean first passage time $E[T_{ij}]$ until transition to state j , given the process starts in states i is evaluated by two methods. In one case the transition is defined to take place, when the envelope makes a first-passage at the barrier e_{ij} . In the other case transition is determined from the qualitative behavior of the responses $q_1(t)$ and $q_2(t)$. Because the transition of states is only visible in the responses after rather long transition intervals the latter method will predict larger mean values than the former. In the numerical simulation based on the response the following procedure for identification of state transitions were used. A transition

from w_1 to p_3 is registered when Q_1 and Q_2 is decreased below $Q_1 < 1.0$ and $Q_2 < 2.2$ for $\Omega = 1.1 \omega_1$.

A transition from w_1 to r_1 is registered, either when $Q_1 < 1.0$ and $E(t) < 1.8$ for $\Omega = 1.25 \omega_1$, or $Q_1 < 1.0$ and $E(t) < 2.5$ for $\Omega = 1.4 \omega_1$. The reason for using the envelope levels in the criteria is to exclude transitions from ω_1 to p_1 in the sample, since these states are

grouped as explained above. Correspondingly, a transition from r_1 to p_3 is registered when $Q_2 < 2.2$ and a transition from r_1 to w_1 is registered when $Q_1 > 1.0$.

Finally, a transition from p_3 to w_1 is registered when $Q_2 > 3.5$. In all cases at least 500 sample values are used in the estimate of $E[T_{ij}]$. Based on this estimate v_{ij} is next determined from (4).

The transition rates are listed in Table 1. The results of the two numerical methods agree qualitatively, but the transitions rates determined from the response are 100%-300% smaller than those determined from the expected first passage time of the envelope process of the non-dimensional chord elongation. The reason for this discrepancy is the difficulty in observing the state transition from the response due to the long transition interval. Since it has been demonstrated that the triggering mechanism unambiguously is related to the crossing of the envelope process, it is concluded that state observations based on the response may be erroneous.

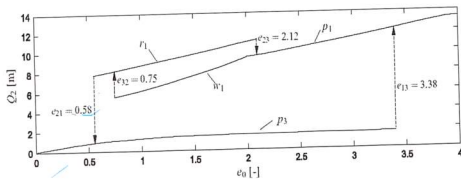


Figure 5: Q_2 during harmonic excitation as a function of amplitude e_0 , $\Omega = 1.25 \omega_1$.

Analytical Solutions		Simulation
$e_0 = 1.5, \mu = 0.0003$		
$\Omega = 1.25\omega_1$		
v_{12}	0	0
v_{13}	0.000495	0.000491
v_{21}	0.00298	0.00189
v_{23}	0.00408	0.00134
v_{31}	0	0
v_{32}	0.00235	0.00121

Table 1: Transition rates found from the envelope process of the chord elongation and from the response using Monte Carlo simulation.

Concluding Remarks

In order to analyze nonlinear resonance phenomena in the lowest modes of vibration of a shallow cable a reduced two-degree-of-freedom model has been formulated involving a mode representing motion in the static equilibrium plane, and a mode representing out-of-plane motion. Due to the shallowness of the cable the retained modes have closely separated eigenfrequencies. At first possible periodic solutions to the reduced equations of motion under harmonically varying support point motion has been identified based on truncated Fourier series, and the stability has been checked by Floquet theory.

It is shown that up to three stable periodic motions may co-exist: An in-plane low amplitude motion, an in-plane large amplitude motion and a whirling motion involving the in-plane and out-of-plane modal coordinates.

Above a certain frequency the in-plane large amplitude motion has a slightly different amplitude and a completely different phase from the corresponding motion with the out-of-plane component suppressed, which turns out to be unstable.

Next, the excitation is changed to narrow-banded Gaussian process with center fre-

quency and mean envelope equal to the frequency and amplitude of the harmonic excitation. It turns out that vibrations in any of

the three state sooner or later becomes unstable, followed by a jump to another state. Based on Monte Carlo simulations it is demonstrated that the transition between the states may be described by a homogeneous,

continuous time three state Markov chain. Additionally, it is demonstrated that the jump is related to critical values of the envelope process of the non-dimensional chord elongation, which represent limit values for excitation of the harmonic motions at the frequency Ω . A unique correspondence between out-crossings of the envelope process and the initiation of state transitions. Hence, the transitional probability rates of the Markov chain may be calculated from the mean first-passage times of the envelope process relative to the indicated limit values. Finally, due to very long transition time intervals between the states, it is demonstrated that estimates of the transitional probability rates from the response may lead to significant errors.

Acknowledgement

This work has been supported by the Danish Technical Research Council through the project Daming Mechanisms in Dynamics of Structures and Materials.

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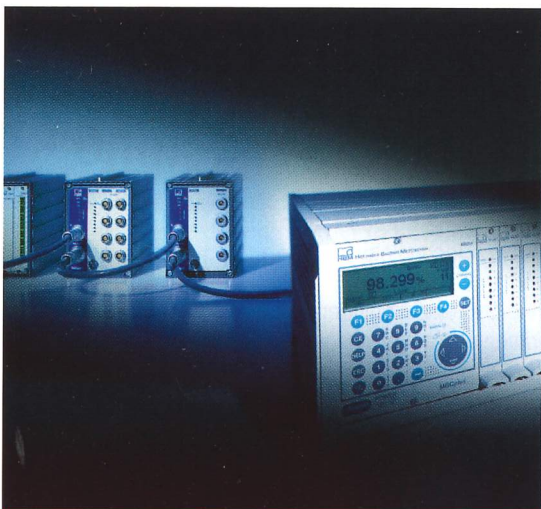
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Non-contact measurements of tool vibrations in a milling machine

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Abstract

Empirical knowledge of the machining process is now being complemented by more scientific methods to determine machine tool vibration properties that are associated with stable machining. Current methods used to predict the behaviour of a milling process (e.g. "impact-test") use measurements made on non-rotating systems. However, such measurements do not detect the effects of gyroscopic and centrifugal forces present in a rotating system. To be able to fully investigate the behaviour of a high speed rotating system one need to use non-contact measurement methods e.g. the Laser Doppler Vibrometry (LDV) method, which is becoming common for vibration measurements; although the method is not without its problems when applied to rotating objects. The work presented here consists of an investigation into the use of LDV and inductive position sensors with magnetic excitation to measure vibrations of a tool in a milling machine. The ability to perform non intrusive measurements should make it possible to analyse changes in machine tool dynamics during the cutting process and in the long term find ways to stable machining.

1 Introduction

Manufacturers of modern machine tools are increasingly implementing advanced process monitoring and supervisory process control [1] to complement the basic functionality of the machine tool control system. At their simplest, process monitoring systems are used to help prevent or limit the effects of catastrophic events such as tool breakage [2] or spindle failure. Such events can be detected by monitoring the current drawn by axis drives and spindle motor [3], or by more advanced techniques such as cutting force monitoring or measurements of vibrations using accelerometers or acoustic emission using sensitive transducers and signal conditioning software [4-8]. By setting safe limits for the monitored parameter(s) based on experience or trials, unusual or unexpected events which may indicate a catastrophic failure can be used as a trigger to stop the machine. To be able to fully investigate the behaviour of a high-speed rotating system, such as a machine tool spindle, it is necessary to use non-contact measurement methods. Several approaches to the non-contact measurement of rotating objects have been developed. These include optical techniques such as Pulsed Laser TV-Holography [9] and Laser Doppler Vibrometer techniques (LDV) [10] which

all have their difficulties when applied to rotating objects. [11] Another way of measuring vibrations is with the use of inductive position sensors which is commonly used for displacement measurement. E.g. in SKF's (www.skf.se) magnetic bearing product; position sensors are used in order to keep track of the axle position while it is being controlled by magnets keeping it hovering inside the bearing free of mechanical contact. This paper describes an investigation in the possibilities of measuring vibrations and Frequency Response Functions (FRF) of a tool in a milling machine using LDV or inductive position sensors. The two methods have been tested separately and no comparison between the two has been made.

2 LDV measurements of a polished dummy tool

A dummy tool with a radius of 10 mm and a length of 100 mm was manufactured from a solid stainless steel tool blank. The shaft was mounted in a lathe and polished using emery paper, diamond paste with particles ranging from 9 μm to 0.25 μm and with a chemical polishing fluid. In the actual experiment a spray which is normally used for crack detection was used to create a removable diffuse (optically rough) surface on the polished dummy tool. Both the polished tool surface and the sprayed surface were measured by the optical profiler, and a representative area of the tool (of 304 x 199 μm) was sampled in steps of 414 nm. The measurements showed a normally distributed surface structure with $R_a = 11,29$ nm, implying that the polished surface was optically smooth compared to the laser wavelength of 633 nm. The sprayed surface showed substantially higher values, $R_a = 21,02\mu\text{m}$, giving an optically raw surface.

For the vibration measurements, a PSV 300 LDV system from Polytec GmbH

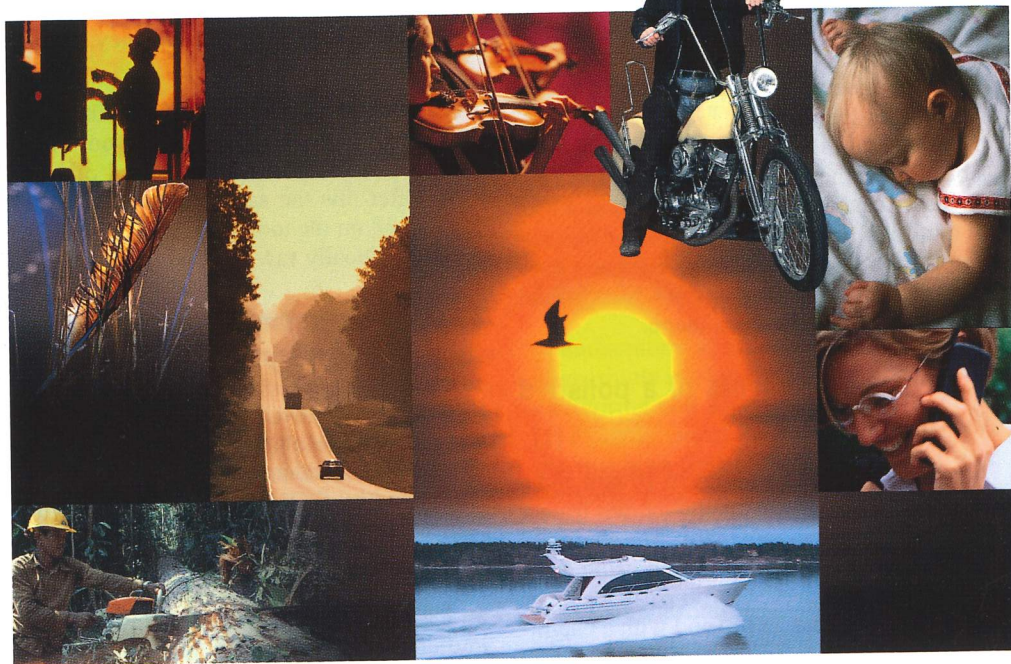
(www.polytec.com) including a displacement decoder was used. The LDV scanning head was mounted on a sturdy tripod and placed approximately 2 m from the tip of the dummy tool on a soft damped material to reduce the influence of structural floor vibrations. The maximum detectable frequency was set to 16 kHz.

The LDV was used to measure the vibrations at the tip of the polished dummy tool in the radial direction at a spindle speed of 6000 rpm. After this the same measurement was carried out on the tool after being sprayed to give an optically raw finish.

3 FRF measurements using inductive sensors and magnetic excitation

FRF measurements using inductive sensors and magnetic excitation were carried out by the use of a standard magnetic bearing manufactured by SKF Revolve and a specially made dummy tool with magnetic properties. (The inductive sensors in the bearing have 1% linearity, a sensitivity of 50 $\mu\text{m}/\text{V}$, a range of 1.5 mm, a resolution of 30 nm and are immune to shifts in temperature). The bearing was mounted in a milling machine on the machine table with its W axis along the y direction of the machine (Figure 1). A modal test of the mounted bearing was performed which revealed no significant modal properties in the frequency band of interest, implying that the FRF measurements of the spindle could be performed without unwanted influences from the mounted bearing structure. A dummy tool with magnetic properties was mounted in the machine spindle and lowered down into the bearing. The FRF of the mounted dummy tool was then calculated by measuring the response from the inductive position sensors while exciting the structure with the magnets. The magnetic bearing control software MBscope was set up to perform an analysis of the system in

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the interval 50-1000 Hz with 600 points/decade and 30 s/decade using a sine sweep with an amplitude of 1A. Measurements were made while exciting the structure in the W13 direction for 21 different spindle speeds from 0 to 7000 rpm in steps of 350 rpm. The measurements started with 0 rpm and with a machine structure temperature of approximately 18° C.

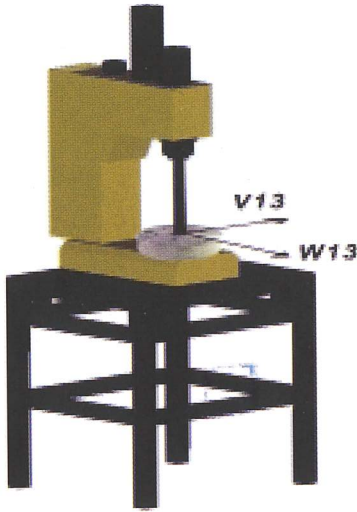
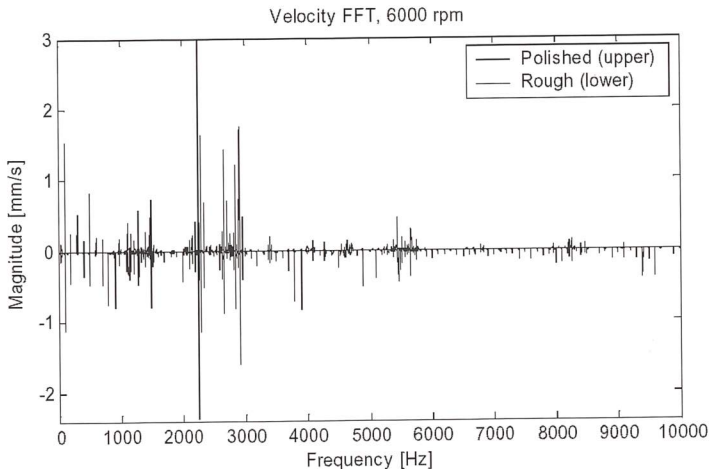


Figure 1 Milling machine with a magnetic bearing mounted on the machine table for spindle FRF measurements. The bearing W direction is along the normal y direction for the milling machine.

4 Results

4.1 LDV

The results from the LDV measurements at 6000 rpm are presented in a velocity spectrum chart where the polished and rough dummy tool measurements are displayed together in Figure 2. The spectrum of the rough surface has been flipped down to the negative side in the chart to simplify comparison of the two spectra. In the charts it can be seen that the spectrum of the rough dummy tool, in opposite to the polished one contains peaks at $f * n$ Hz where f is the rotational speed of 100 Hz (6000 rpm) and $n = 1, 2, 3 \dots$. These peaks are expected due to the presence of a speckle noise repeated for each dummy tool revolution in the sampled data.



*Figure 2 Spectra of the polished and the sprayed surface at a spindle speed of 6000 rpm. The spectrum of the rough dummy tool has been mirrored along the frequency axis down to the negative side to simplify comparison between the two. Multiple harmonics of $n * 100$ Hz where $n = 1, 2, 3 \dots$ can be seen in the spectrum of the sprayed surface.*

4.2 Inductive sensors

The results from the FRF measurements with the magnetic bearing are presented in Figure 3 and Figure 4. Figure 3 displays the magnitude response and the phase response for a non rotating spindle. Figure 4 shows the measured magnitude response functions for all spindle speeds where the response functions are displayed along the x axis from 50-1000

Hz. (High response values are represented by bright pixels in the image). A resonance frequency can be seen at 500 Hz in Figure 3 and Figure 4 for the non rotating case. The difference between the FRF of the non rotating spindle and the rotating ones is obvious especially regarding the resonance frequency at 500 Hz. On the upper left hand side in Figure 4 the spindle speed can be seen as a disturbance in the measurement.

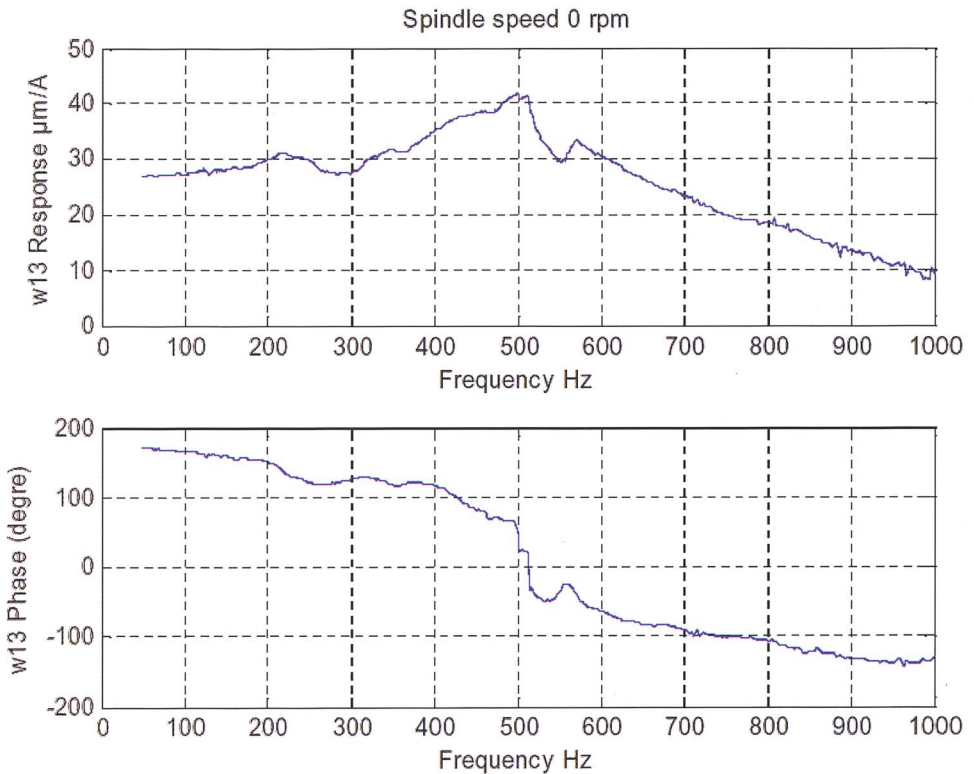


Figure 3 Frequency response function for 0 rpm.

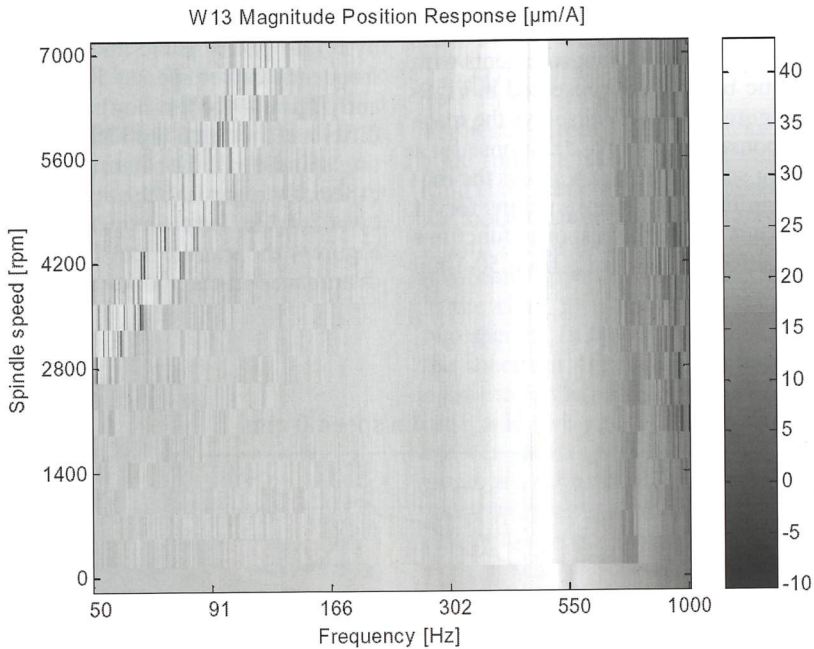


Figure 4 Plot of the frequency response functions in μm per ampere in to the magnets [$\mu\text{m}/\text{A}$] with the spindle speeds along the y axis and the frequency between 50-1000Hz on the x axis (log scale).

5 Discussion and concluding remarks

By polishing the tool to a level where the surface could be considered as optically smooth; the speckles noise is removed and the measurements of axial vibrations can be performed. If this method is to be used for vibration measurements of a rotating tool when it is cutting, several other difficulties must be tackled. The laser beam must have a clear path to the tool with no interfering metal chip, oil or cooling fluid from the cutting process. The polished part of the tool surface, where the measurement will be conducted, has to be free of particles such as dust or pro-

cess fluids. Any disturbances on the surface will cause speckle noise and unwanted peaks in the spectrum. The test with the magnetic bearing showed that it is possible to measure the FRF during different spindle speeds with a magnetic bearing, and therefore making it possible to detect e.g. speed dependent components in the FRF. The difference detected in the measurements of the non rotating spindle and the rotating ones can have its origin in temperature or speed dependent components in the spindle structure. More tests will be conducted with other test sign-

als and during other machine conditions. The influence of the unknown excitation force from the machine itself must also be considered in future measurements. Comparisons between LDV and inductive position sensors will also be carried out.

6 Acknowledgements

The financial and technical support for this work was provided by the Swedish Agency of Innovation Systems (Vinnova) and SKF Nova, respectively. The LDV system and auxiliary equipment have been financed by the Kempe Foundations.

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.



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Hälsorisker med vibrationer i arbetsmiljön

Temadag om Vibrationsdirektivet arrangerad av SVIB AU 4

på Arbetsmiljöverket 27 oktober

Programmet och anmälningsformuläret för AU 4s Temadag har funnits på SVIBs Hemsida sen i Juni. Ett del anmälningar har kommit in men det finns fortfarande plats för er som är intresserade. Det förslag som jag hörde Van der Wenne tala om i början av 90-talet har nu långt efter hans frånfälle blivit direktiv och lag. Men hur skall lagen tolkas och tillämpas? Författaren och några av de närmast berörda deltar i Temadagen. Låt dom förklara och svara på frågor.

Alf Olsson

Om Temadagen

Höga intensiteter och stora doser vibration och stöt förekommer i arbetsmiljön.

Ett stort antal arbetstagare drabbas av olyckshändelser och långvarig ohälsa till följd av vibrationsexponering på jobbet, till exempel inom transportindustrin. Effekten av påverkan är ibland uppenbar och kan avläsas direkt exempelvis genom brott, skador och svullnader.

Den kan också ske gradvis eller varierande över längre tid och blir då mindre uppenbar, särskilt som effekten av denna påverkan kan vara förknippad med tidsfördröjning (latens), tröskelnivåer, tillvänjning eller eftereffekter. Som om denna komplexitet inte vore nog.

Till och med icke-vibrationsexponerade människors prestationsförmåga, och därmed

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Förhandsintresset har visat att Temadagen kommer att locka deltagare med skilda intressen,

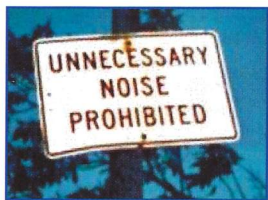
bakgrund och profession. En bidragande faktor till detta är de nationella föreskrifter om vibrationsgränser i arbetsmiljön som nu införs genom lagstiftning.

Utöver att sprida värdefull information kommer Temadagen att bidra till nya kontakter och förhoppningsvis fruktsamma möten mellan nyckelpersoner inom skandinaviskt näringsliv, forskning och utbildningsväsende samt myndigheter med flera.

Hjärtligt välkomna till Temadagen!

Johan Granlund, ordförande SVIB AU4

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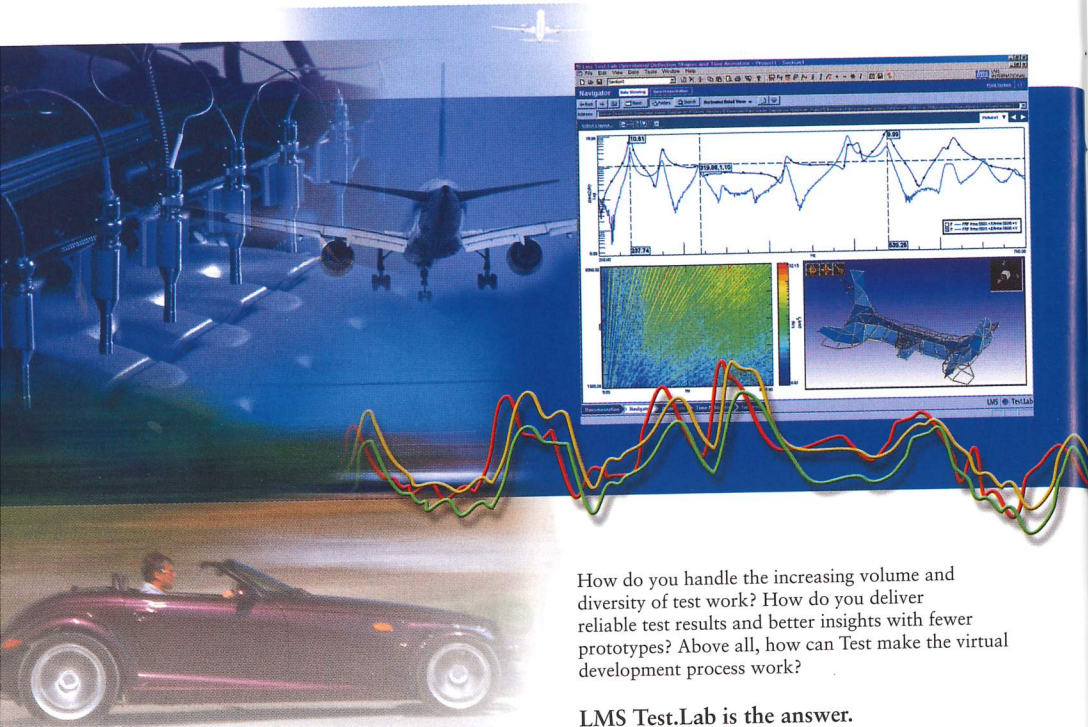
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Vibrationsdagen 23 november 2004 i Stockholm

"Vibrationer ger ljud och ljud ger vibrationer"

Temat denna gång är mitt favoritämne säger Martin Almgren i ett mail han skrev i mitten av juli. Sannolikt regnade det den dagen eftersom han kunde se så ljust på något som kommer att ske i slutet av november. SVIB har haft som tradition att låtta på sinnesstämningen i denna ruggiga årstid med de vanligtvis mycket uppskattade Vibrationsdagarna. Efter att ha varit på turné i Köpenhamn, Göteborg och Oslo de tre senaste åren kommer vi under 2 dagar, Vibrationsdagen den 23 och AU3s Temadag den 24 november, att återvända till Rotundan och Operakällaren i Stockholm. Programmen som följer är i det närmaste klara vid det här laget. Så förbered er på en faktaspäckad men angenäm ljuspunkt i november.

Alf Olsson

"Jag vill ha många olika aspekter på detta ämne och hoppas på ett stimulerande utbyte av erfarenheter, metoder och lösningar" säger Martin vidare innan han redovisar vilka föredragshållare och titlar som är klara eller nästan klara:

Erik Jansson & Anders Askenfelt, KTH -
The physics of the violin

Sten Ljunggren KTH -
Models for structural vibrations coupling with sound in fluids with applications for gypsum boards

Ole Dössing Rovsing Dynamics -
Sound and vibration t s v

Sven Tyrland, Ingemansson -
Loudspeaker vibrations and sound radiation

Claes Fredö Ingemansson -
Acoustic vibration excitation of mechanical systems – practical solutions

Torben Licht B&K -
Acoustic sensitivity and other influences from environmental parameters of accelerometers

Olav Skille, Norge -
Low frequency sound and vibration massage

Wolfgang Kropp, CTH -
Sound radiation from vibrations in car tyres

Urban Emborg, A2 Acoustics -
Propeller noise excitation of fuselage vibrations

Därtill hoppades han få några föredrag klara innan programmet spikas, trycks och distribueras i mitten av oktober.

"Shockwave excitation of ground waves and building vibrations"

"Vibrations and sound radiation in wind power turbines"

Det finns plats för några till. Välkommen med förslag avslutar han med:

Martin Almgren

E-post: martin.almgren@ingemansson.se

Temadag den 24 november

Byggnadsvibrationer

SVIB AU 3 inbjuder i samband med Vibrationsdagen till en temadag den 24 november med några aktuella föredrag från trafik- mark och byggnadsvibrationsområdet. Tidigare AU3-dagar har haft tyngdpunkten på trafikvibrationer. Denna gång ägnar vi oss åt sprängning och annan verksamhet och den påverkan det kan ha på byggnader, utrustning och människor. Hjärtligt välkomna!

*Tomas Odebrant
Ordförande AU 3*

A. Vibrationer från sprängning och annan byggverksamhet

Nya möjligheter att övervaka och styra sprängningar i vibrationskänsliga miljöer
Mikael Creütz, Golder Associates, Stockholm

Bergschakt vid Södra Länken Nynäsvägen, Stockholm. Planering, utförande och kontroll av komplicerad pallsprängning
Kjell- Arne Fredin, K-A Fredin Bergkonsult, Stockholm

Skadeorsakande resonanser i betongbjälklag i villa
Ingvar Bogdanoff , WSP Samhällsbyggnad, Geo, Göteborg

Vibrations- och stomljudsmätningar med GSM-stöd
Sven-Erik Johansson, NitroConsult, Stockholm

B. Vibrationskänslig verksamhet

Ingvar Bogdanoff , WSP Samhällsbyggnad, Geo, Göteborg håller två föredrag dels om
Skaderisk på datorer och kringutrustning dels
Skadeorsakande resonanser i betongbjälklag i villa.

Ingmar Holmström, Husdoktorn, talar om
Hur och varför det uppstår skador på fastigheter och gör jämförelser mellan gamla och nya konstruktioner.

Jon Nyberg, Länsförsäkringar, redovisar
Juridiska aspekter vid skadereglering

Jörgen Svensson, Ingemanson Technology AB, Göteborg beskriver
Forskarhuset, Chalmers, Göteborg. "Vibrationsfritt" bjälklag.

Sen följer

Comfort and vibrations on floors due to walking loads

Svend Ole Hansen Aps, Köbenhavn. Bernt Suikkanen, COWI A/S, Århus. Frederic Sebastien Collette, COWI A/S, Kongens Lyngby.

Building dynamics - vibration isolation of a two-storey laboratory

Kjell Vågfelt, Ingemansson Technology AB, Göteborg

C.

Mät- och analysteknik

Äntligen en standardiserad metod att beskriva vibrationsmiljön för känslig utrustning!

Kjell Ahlin, Blekinge Tekniska Högskola, Karlskrona

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Increasing the Stiffness of a Light Weight Laminate Structure Utilising Shape Memory Alloy Actuators

Lauri Kantola*, Patrik Söderström** and Merja Sippola***

* University of Oulu, Department of Mechanical Engineering, Mechatronics and Machine Diagnostic Laboratory, P.O.BOX 4200, FIN-90014 University of Oulu, Finland. ** Helsinki University of Technology, Laboratory of Lightweight Structures, P.O.Box 4300, FIN-02015 HUT, Finland. *** VTT Technical Research Centre of Finland, Industrial Systems, Reinforced Plastic Structures and Small Craft, Otakaari 7B, P.O.Box 1705, FIN-02044-VTT, Finland. * Lkantola@paju.oulu.fi, ** Patrik.Soderstrom@hut.fi, *** Merja.Sippola@vtt.fi

Abstract

This paper describes the results of the active shape and stiffness control tests of a light-weight laminate structure which utilises shape memory alloy (SMA) wires. The objective of the study was to actively change the shape and stiffness of the structure. The structure was a simply-supported laminate beam, which - at two separately controllable points - is actively

co-supported by a truss-like active structure consisting of vertical laminate supports and diagonal SMA wires. Four SMA wires were attached to the structure so that one pair of wires would affect one vertical support. The force generated by the wires was guided to the beam. The current for heating each pair of wires was controlled independently. To measure the strains of the structure four strain gauges were bonded to the beam.

An eccentric rotational mass-motor system was placed to the mid-point of the beam. To monitor the rotational speed of the motor, an optical incremental encoder was used whereas a function generator controlled the motor speed. The range of frequency was set from 0 to 25 Hz. The motor speed varied in accordance with a slow triangular wave. The

signals of the strain gauges were measured and analysed.

The results clearly show how the shape of the beam changes when the wires are heated. The stiffness of the structure also changes, and by heating the wires, natural frequencies change considerably. The conclusion is that SMA wires can be used to actively control the shape and stiffness of light weight structures. However, the original (non-deformed) shape of the structure should be designed properly. Learning how to optimise the design process of smart structures is also a future challenge.

Keywords: Shape memory alloy (SMA), laminate structure, stiffness, shape, vibration.

1. Shape memory alloy actuated smart structures

Smart material refers to a material that can sense changes in its surroundings and then react in a useful way. Therefore, smart materials are responsive. Smart structure refers to a structure that has actuation, sensing, processing and communication capabilities. Smart structures often utilise smart materi-

als and have the capability of actively responding to a changing external or internal environment [1]. More advanced smart structures may also have a self-monitoring or even self-repair capability [2]. Smart structures usually consist of a lightweight mechanical structure and they utilise embedded or external sensors and smart material actuators. The control system may also be embedded in the structure. The functionality of the structure can either be related to quasi-static shape management or vibration control. The dynamical properties strongly depend on the actuator technology used [3]. Shape memory alloy (SMA) actuated smart structures can have a high power-to-weight ratio. The other advantages SMA offers are silence, cleanliness, low drive voltage, and the solid state nature of the actuator [4].

2. Lightweight laminate structure and actuation mechanism

The lightweight mechanical structure used in the shape and stiffness control tests was a simply-supported laminate beam actively co-supported at two separately controllable points by a truss-like active structure consisting of vertical laminate supports and diagonal SMA wires (see figure 1). The span of the beam was 500 mm, width 70 mm and thickness 3 mm.

Actuators were 0,4 mm NiTi shape memory alloy wires from Memory-Metalle GmbH. Structure was designed for bending loads, and wires act in pairs on each vertical support. Figure 2 shows the principle of the actuation mechanism. The shape memory actuation effect (SME) is caused by thermoelelastic martensitic transformation of the alloy [5].

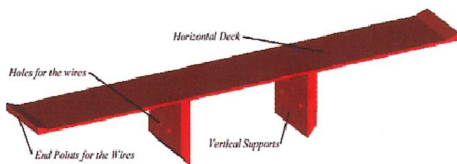


Figure 1. Mechanical structure of the beam without and with the actuators.

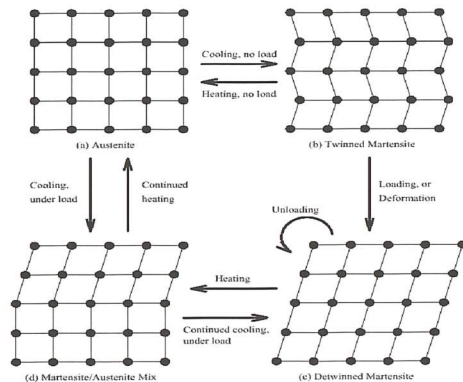


Figure 2. Basic principle of shape memory effect (SME). When the material is in the high temperature austenitic (parent) phase and it is cooled down, the twinned martensitic structure appears. Loading the material causes twinned structure to change into detwinned martensite that can be reverted to the parent phase by heating the material. This cycle can be repeated and used as an actuator.

3. Testing the single SMA wire

As the SMA wires are not fully standardised products and the properties vary between batches the thermoelastic behaviour of the wires under cyclic heating and cooling was tested experimentally. Tests were done by using the materials testing machine (Instron). Stress vs. strain curves for different phases of the material were measured together with the heat-induced force and displacement. The

temperature of the wire was monitored by using a single thermocouple. The temperature sensor was made of thin sheet metal, and it could be bent so that the sensor tightly spiralled round the actuator. The sensor IC amplified signal and a continuous voltage of 10 mV/C° were measured. Figure 3 shows the stress vs. strain curves for martensitic and austenitic (parent) phases. Temperature curves are also shown.

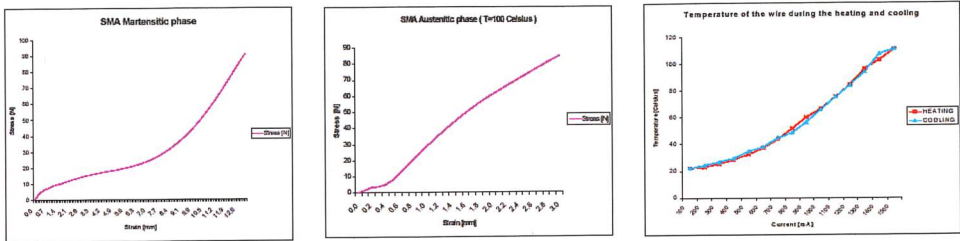


Figure 3. Stress vs. strain curves for the martensitic and austenitic (parent) phases. Also, the temperature curves during the resistive heating and free cooling are shown.

The force generated by the wire was measured using the load cell of the testing machine. A SMA sample was attached between the two plastic clamps. Clamps were installed in the testing machine which was programmed to stretch the wire by 5 % and then hold the position. The wire was then resistively heated. The electrical resistance of the wire was measured to be 5,3 . Heating current was supplied in 100 mA steps. The force between the clamps was measured. Figure 4 shows

the force curves when the wire was being stretched and heated for the first, the second and the subsequent times. According to the figure, it takes quite a high force (45 N) to stretch the cold wire. Due the heating, the force level raises when the temperature reaches 70°C. When the wire is cooled down the force level drops below the starting level. The remanent force is now ca. 10 N. If the wire is heated and cooled again, the force curve looks different. The starting and final levels stay at 10 N while the maximum is at 90 N. Slight hysteresis is also seen in the force curve.

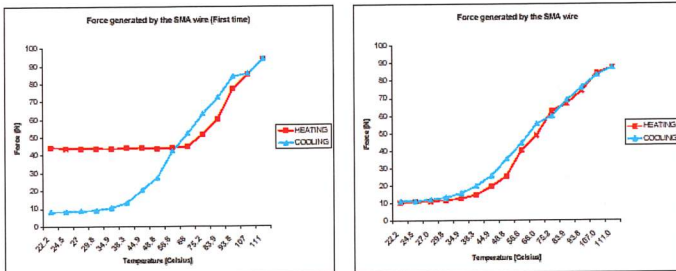
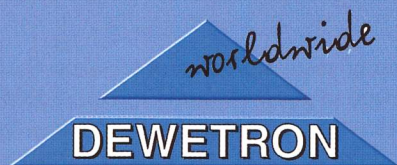


Figure 4. Actuator is stretched and heated for the first and the subsequent times to generate force.



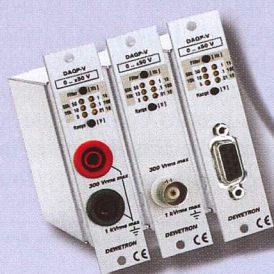
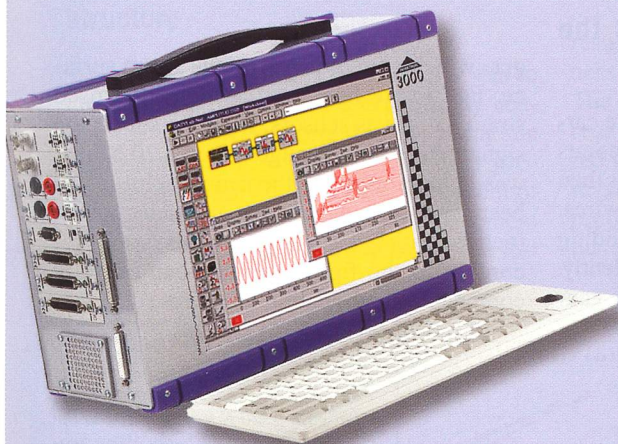
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Similar results are achieved when the wire is used to generate motion. Also in this case the wire was attached to the testing machine. The machine was programmed to load the wire with a constant force of 50 N. The wire was then resistively heated, and clamp displacement was measured. Figure 5 shows how the clamp distance changes when the wire is heated for the first time and for the subsequent times. It is seen that when the

wire is martensitic and loaded with 50 N force it stretches almost by 10 %. Due to heating the shape recovery starts when the temperature reaches 60°C. When the wire is cooled down it tends to stretch more than at the first time. If the wire is heated and cooled again the displacement curve shows hysteretic behaviour. However, the starting point and the final point of the motion curve stay fairly unchanged.

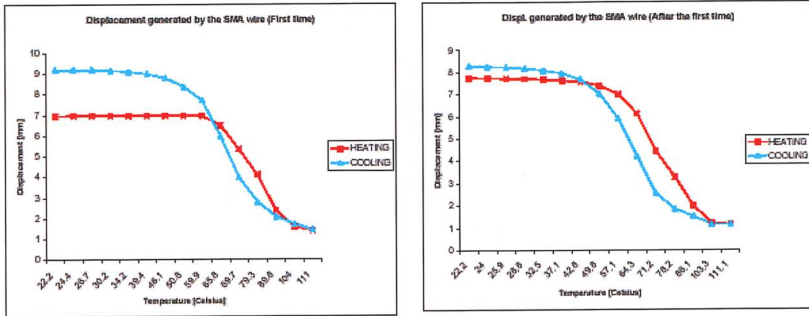


Figure 5. Actuator is loaded and heated for the first time and for the subsequent times to generate motion.

4. Controlling the shape of the structure

Shape control tests of the structure were carried out by applying a 20 N load to five points on the beam. Load-induced bending was then compensated by actively heating the wires. The aim was to compensate local bending by heating the actuator pairs independently. Bending was measured by four strain gauges

ges, which were bonded on the beam surface. For active heating, the algorithm used strain gauges as sensors. Figure 6 shows the signals from strain gauges when a 20 N load was being applied to the centre point of the beam and wires were being gradually heated (active heating was not ON in this case). According to the figure bending can be fully compensated by heating the wires.

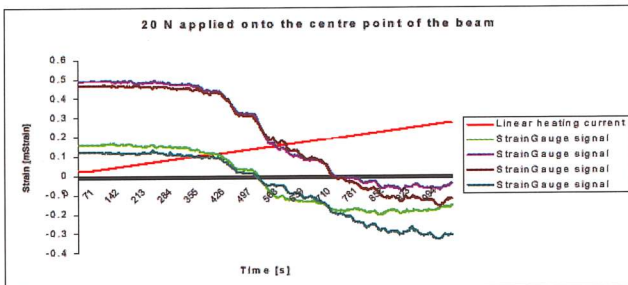


Figure 6. 20 N load applied to the centre point of the beam. Wires heated gradually. Strain gauge signals show how the shape of the structure changes due to the heating. Bending displacement can be fully compensated by heating the wires.

The effect of active heating is seen in Figure 7. A 20 N load was applied to the centre point of the beam and wires were actively heated. The aim was to drive the strain gauge signals to the zero-strain condition so that the bending of the beam is locally and globally compensated. The figure shows how the heating current reaches the maximum level when

load is being applied. Seventy seconds later the beam reaches a stable condition up to a point where the loading condition suddenly changes at 320 seconds. The load now moves to the point that is located near the right hand vertical support. It is seen that the heating current for wire pair 2 rises while that for the others drops. The zero-strain condition is well achieved.

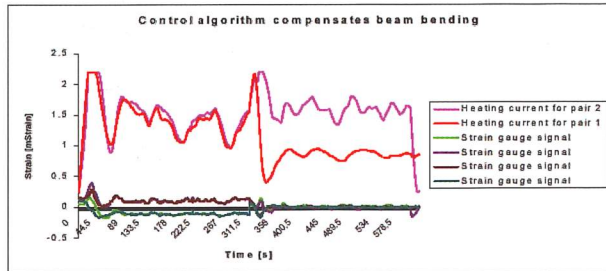


Figure 7. 20 N load applied to the beam. Wires heated automatically. Heating currents show how the active heating algorithm responds to loading conditions. The aim is to drive the strain gauge signals to the zero-strain condition so that the bending of the beam is locally and globally compensated.

5. Controlling the stiffness of the structure

The aim of the stiffness control tests was to change the stiffness of the structure by heating the wires. Tests were done by using the materials testing machine which was programmed to apply load to the centre point of the beam. The corresponding displacement was then measured. Figure 8 shows the test

results. It can be clearly seen how the stiffness of the structure changes due to heating. The spring constant of the "heated" structure is more than doubled (3,7 @ 8,0) compared to that of a non-heated one. The structure can now be assumed to be a spring whose properties can be changed by heating the actuator wires.

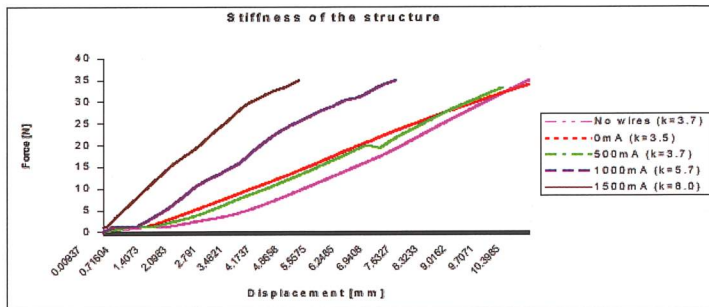


Figure 8. Curves above show how the stiffness of the structure changes due to heating. Stiffness change is significant and the spring constant changes from 3,7 to 8,0.

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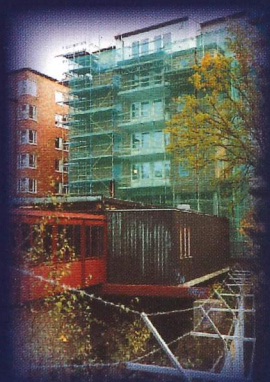
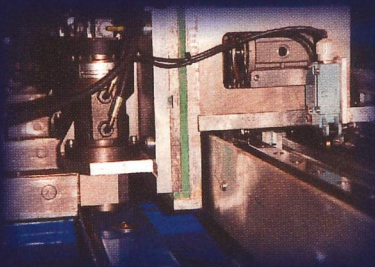
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6. The structure as a motor bed

Based on the results gained from the stiffness control tests the structure was used as a motor bed. Figure 9 shows the motor bed test arrangement. The main components are the controlled smart structure, four channel strain gauge amplifiers, PCbased controller and the motor control circuitry. SMA con-

trol electronics consist of voltage-controlled amplifiers and load resistances. A DAQ-card was used to collect the data, and the shaker was driven by a DC-motor. Two rotational masses were attached to the motor axle. There was a belt drive between the incremental encoder and the motor. Rotational speed dependency on the control voltage is shown in the figure.

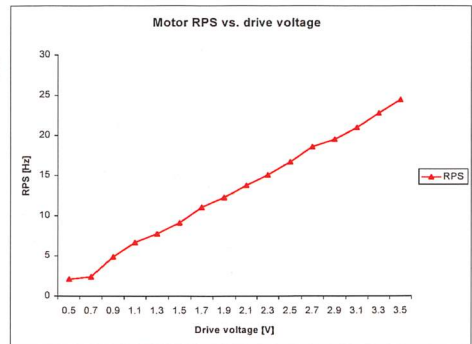
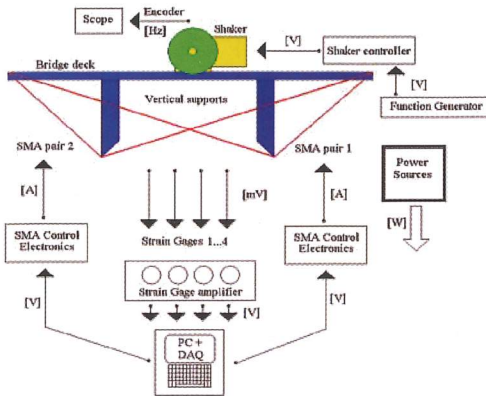
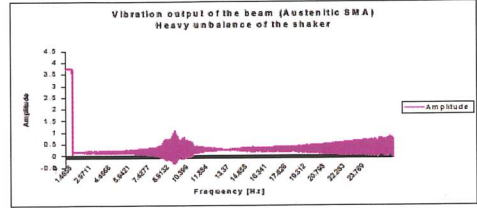
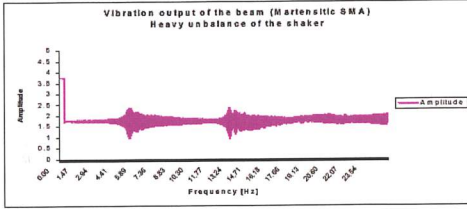
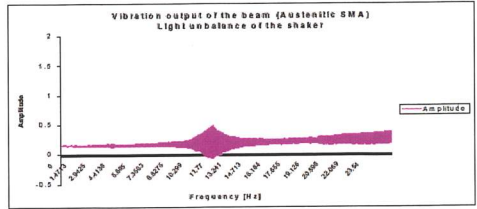
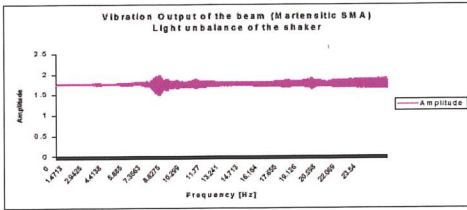


Figure 9. Test arrangement and the motor speed.

Figure 10 shows the vibration output of the structure when actuators were in fully martensitic and austenitic phases, respectively. For the light vibration input, the natural frequency of the former is around 8,8 Hz while that of the latter is to be seen around 12,0

Hz. For the heavy vibration input, the frequencies are 6,5 Hz and 9,5 Hz, respectively. If the change in natural frequency is transformed to a change in spring stiffness, the result indicates that the stiffness of the "heated" structure more than doubles as compared to the non-heated one. This is consistent with the results gained from the stiffness con-



trol tests.

Figure 10. Vibration outputs of the structure. SMA actuators are used to change the stiffness of the structure. In both cases natural frequencies have changed. The change in shape is also visible.

7. Discussion

Test results show that the developed SMA-actuated smart structure is capable of changing both its shape and stiffness. The changes are well controllable. These characteristics can be exploited in many ways. Therefore, the Technical Research Centre of Finland has patent pending on the structure. As actuators the shape memory alloys have properties that make them suitable for the quasi-static property control of lightweight structures. Resistive heating of actuator wires is possible in a certain thickness range. Dynamical properties of the structure are limited due to the slow cooling process of the wires. Naturally, forced cooling raises the operation

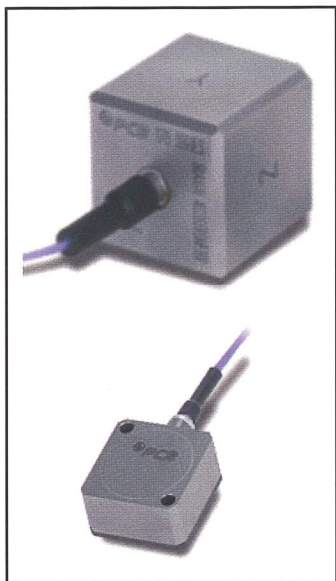
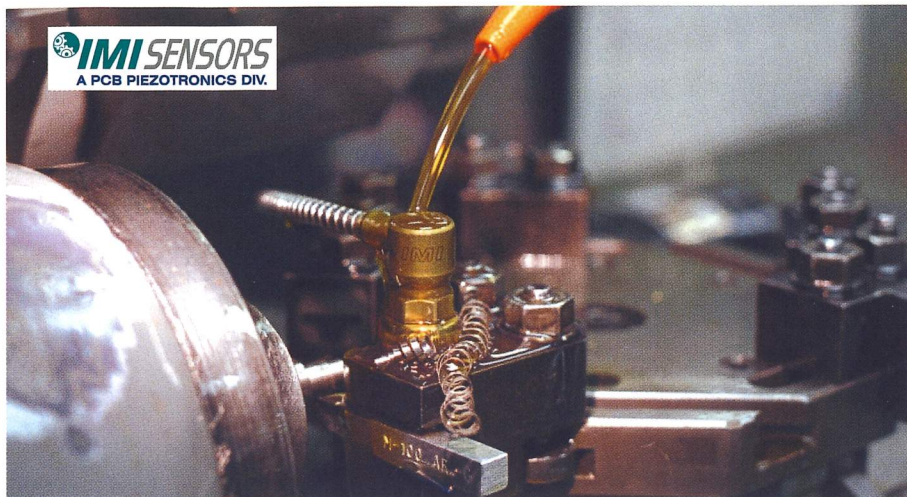
frequencies.

Simulation of structural behaviour requires finite element models of both the structure itself and the actuator material. Advanced control algorithms require simple and mathematically efficient models which take both hysteresis and environmental changes into account. Several models of the actuator material are proposed in the literature. The form of a model structure used depends on the case and the accuracy specifications.

The future work consists of investigating the shape memory material models. The aim is to find the models that suit well the controller design. Distributed fiber optic sensors and plate structures are also under investigation. More complex systems will require more simulated data to ensure operation in changing environments.

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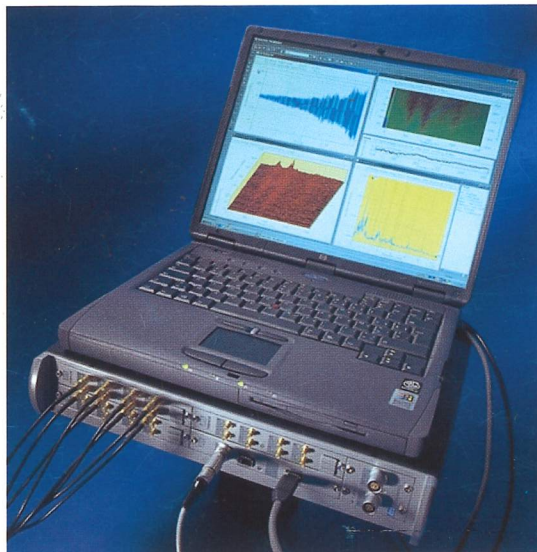
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