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FACULTY OF ENGINEERING

AN INVESTIGATION INTO SOME DYNAMIC EXCITATION TECHNIQUES

THESIS

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BY

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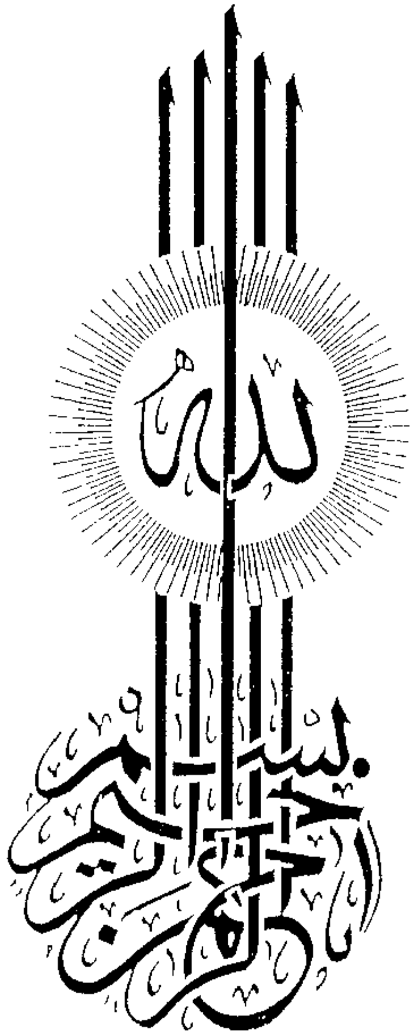
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SUMMARY

The solution of the vibrations and noise problems, implies
ˑetermination of the dynamic characteristics of the system
.er consideration. Several excitation methods have been used
- this purpose. The aim of the present investigation is to
.dy the different excitation methods to select the most proper
-d for the dynamic testing of machine tools.

For the sake of comparison, the different excitation
-chniques were used for the determination of the dynamic
racteristics of a boring bar ϕ 25 x 267 mm length, fixed in
special fixture mounted on a center lathe. The different
methods which have been studied in the present work are harmonic-,
ndom-, micro tremor-and impulse techniques.

rmonic excitation test: harmonic excitation is the oldest
hod used for the determination of the system characteristics.
t allows the observation of the vibration amplitude and phase
ngle at each exciting frequency. The response is obtained with
rood degree of certainty and no computation is required.
ˑer it requires a lot of expensive test equipment such as
wave generators, power amplifiers, shakers, phase meters,
s filters and x-y plotters. The exciting frequency can be
either manually or automatically. The harmonic excitation
requires a relativly long time for preparation as well as
unning the test.

m excitation test: The system under test is subjected to
dom force spectrum which covers the frequency range of
rest. The random excitation force is usually obtained from
ndom wave generator and a shaker. Both of the force and the

response signals are recorded simultaneously and analysed using the cross correlation tecnnique. The random excitation method is much faster than the harmonic exeitation method. It requires almost the same instrumentation used in the harmonic method except the phase meter. However the theoretical analysis of random vibrations is rather complicated. Data processing using the computer needs a considerably long period due to the complexity of the used computer program.

<u>Micro tremor technique</u>: Machine tools can be excited by vibrations with small amplitude resulting from the idle running of the machine, which have usually a random nature and which are known as micro tremor.

The test comprises the measurement of the vibration amplitude at a position which is near to the source of excitation (Input signal), and at the position at which the response is to be determined (Output signal). This method can be used for the determination of the natural frequencies and the damping ratio only. Data analysis and -processing are carried out in exactly the same manner as in the case of the random excitation test.

<u>The impulse test:</u> The system is subjected to an impulsive force using a specially designed hammer. The exciting force and the system transient response are recorded simultaneously as time functions. Both signals are Fourier transformed to get the system transfer function using a special computer program prepared for this purpose. This method is very fast, the test requires only few seconds.The computer program is simple and needs only few minutes for computation. The whole dynamic characteristics can be obtained by a single pulse. However, this method is inconvenient

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NOMENCLATURE

Symbol	Computer Symbol	Units	Definition
$A_f\ (f_m)$	AF (FM)	N	Real part of the Fourier transform of the impulse force signal.
$A_y\ (f_m)$	AY (FM)	mm	Real part of the Fourier transform of the system response.
$B_f\ (f_m)$	BF (FM)	N	Imaginary part of the Fourier transform of the impulse force signal.
$B_y\ (f_m)$	BY (FM)	mm	Imaginary part of the Fourier transform of the response signal.
CC1 (k)	CC1 (K)	-	Sum of the positive and the negative shifts for the cross correlation functions.
CC2 (k)	CC2 (K)	-	Difference between positive and negative shifts of the cross correlation function.
c	-	$\frac{Ns}{m}$	The damping coefficient of the system under test.
c_c	-	$\frac{Ns}{m}$	Critical damping of the system under test.
D	-	-	Damping ratio.
F (t)	-	N	Exciting force signal.
F (I)	F (I)	N	Discrete force values obtained by sampling the continous function F (t).
$F_i\ (t)$	-	N	Impulsive force.

Symbol	Computer Symbol	Units	Definition
F_i (I)	FI (I)	N	Discrete force values obtained by sampling the continuous function $F_i(t)$.
F_m	-	N	Maximum magnitude of the impulsive force.
Fo (f_m)	FO (FM)	N	Force magnitude at the discrete freauency (f_m).
f_c	-	Hz	Frequency content of the impulse force spectrum.
f_m	FM	Hz	Discrete frequency coordinate, fm = 1, 2,...,f_u
f_{nq}	-	Hz	Nyquist frequency.
f_s	-	Hz	Sampling frequency.
f_u	FU	Hz	The upper boundary frequency of the signals.
H (f_m)	-	um/N	Frequency response function.
h	-	mm	Height of the hammer above the object.
I	I	-	Order of samples, I = 1, 2, 3, ..., N.
Imag(f_m)	IMAG (FM)	-	Imaginary part of the frequency response.
K	K	-	The correlation coefficients order where (K = 0, 1, 2, ..., Z).
K_{dy}	-	N/um	Dynamic stiffness.
K_{st}	-	N/um	Static stiffness.

Symbol	Computer Symbol	Units	Definition
M	-	Kg	Hammer mass.
M_f	MF	N	Mean value of the force signal.
M_y	MY	m/s^2	Mean value of the response signal.
me	-	Kg	Equivalent mass of the vibrating system.
$mag(f_m)$	MAG (FM)	um/N	Magnitude of the frequency response functions.
N	-	-	Number of samples.
$P_{fy}(f_m)$	PFY (FM)	Nm/s^2	Real part of the cross spectral density at the discrete frequency (f_m).
$q_{fy}(f_m)$	QFY (FM)	Nm/s^2	Imaginary part of the cross spectral density at the discrete frequency (f_m).
R_f (k)	RF (K)	-	The normalized auto correlation function of the force signal.
$R_{fy}(k)$	RCR (K)	-	The normalized cross correlation for positive shift.
$R_{fy}(-k)$	LCR (K)	-	The normalized cross correlation for negative shift.
R_y (k)	RY (K)	-	The normalized auto correlation function of the response signal.
$Rmag(f_m)$	RMAG(FM)	um/N	Real part of the frequency response.

Symbol	Computer Symbol	Units	Definition
$S_f\ (f_m)$	SF (FM)	N^2/Hz	Power spectral density of the force record at the discrete frequency (f_m).
$S_y\ (f_m)$	SY (FM)	$m/s^2/Hz$	Power spectral density of the response at the discrete frequency (f_m).
$S_{fy}(f_m)$	-	Nm/s	cross spectral density between input and output.
T	-	sec	Observation time (record time length).
T_r	-	sec	Truncation time for the impulse response.
T_z	-	sec	Time length of the correlation function.
t_i	-	ms	Pulse duration for the impulse force.
U	-	-	Magnification factor .
v	V	m/s	Impact velocity.
W	-	N/um	Calibration factor.
y(t)	-	m/s^2	System response (accelerat-ion).
$y_i(t)$	-	m	System response of the impulsive force.
y(I)	Y (I)	m/s^2	Discrete response values obtained by sampling the continuous time function y (t).
y_i (I)	YI (I)	m/s^2	Discrete response values obtained by sampling the continuous time function y_i (t).

Symbol	Computer Symbol	Units	Definition
$Y(f_m)$	Y (FM)	m/s^2	Acceleration magnitude.
Z	Z	-	Maximum number of correlation coefficients.
$\theta_f(f_m)$	ANGF (FM)	(°)	Phase angle between the real and the imaginary parts of the impulsive force.
$\theta_y(f_m)$	ANGY (FM)	(°)	Phase angle between the real and the imaginary parts of the Fourier transform of the acceleration signal.
$\phi(f_m)$	ANG (FM)	(°)	Phase angle between the exciting force and the system response.
σ_f^2	RVF	N^2	Variance of the force record.
σ_f	RVSF	N	Standard deviation of the force record.
σ_y^2	RVY	$(m/s^2)^2$	Variance of the response record.
σ_y	RVSY	m/s^2	Standard deviation of the response record.
τ	-	sec	Time shift.
w	-	rad/s	Angular frequency.
Δt	DT	sec	Sampling time interval.
Δf	DF	Hz	Frequency resolution.
$\gamma^2(f_m)$	COH (FM)	-	Coherency between input and output calculated at the discrete frequency (f_m).

INTRODUCTION

Mechanical structures are usually subjected to vibrations due to the fluctuation of the acting forces or the unbalance of rotating parts. Vibrations may have detrimental effect on the structure. Vibrations at resonance or the occurence of chatter may cause excessive wear and fatigue of machine tool components, destruction of the cutting tools and of the workpiece surface beside the troublesome chatter noise.

In order to solve vibration-and noise problems of mechanical systems their dynamic characteristics must be determined experimentally. In fact the dynamics of any system is completely defined by its frequency response function.

There are different excitation methods which can be used for the determination of the frequency response of mechanical structures such as; harmonic-, random-, micro tremor-, and impulse techniques. These methods differ widely with respect to the equipment needed, effort and time required in preparation and in carrying out the test.

There is still however insufficient information about the advantages and limitations of the different excitation techniques which enables the selection of one method for a given system.