



THE DEVELOPMENT OF AMPLIFIED VIBRATION-ABSORBING ISOLATORS FOR TONAL TIME-VARYING EXCITATION

by

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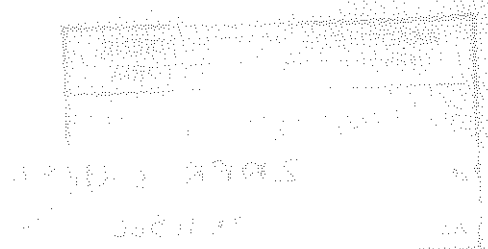
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Summary

The development of amplified vibration-absorbing isolators for tonal time-varying excitation

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Vibration isolation is a procedure through which the transmission of oscillating disturbances or forces is reduced. The ideal isolator is one that will support the equipment being isolated without transmitting any dynamic forces. An isolator with infinite static stiffness and zero dynamic stiffness will achieve this goal. Although this ideal isolation cannot be obtained in practice, it can be approximated through a wide range of devices. This approximation occurs over a limited frequency band and methods of increasing this band were investigated. The goal of this thesis was to further our understanding of mechanical systems that can approximate the ideal isolator behaviour.

To compare the various devices the blocked transfer dynamic stiffness was defined. This value was found to represent the isolator properties without the additional complication of the equipment being isolated as happens in traditional transmissibility methods. Three classes of devices were distinguished namely isolators, vibration-absorbing isolators (VAI) and amplified vibration-absorbing isolators (AVAI). The last two types exploit nodalisation to reduce the dynamic stiffness over a limited frequency range. The focus of this work is the broadening of the effective low stiffness bandwidth of amplified vibration-absorbing isolators by adapting system characteristics. If the excitation is tonal time-varying these devices can be used successfully.

Two novel adaptive amplified vibration-absorbing isolators were introduced and studied in the time and frequency domains. The type I AVAI uses flexible reservoir walls to vary the isolation frequency. The type II device incorporates a heavy metal slug. Both devices use variable pressure air springs to change their stiffness. The use of air springs are convenient, offers low damping and can be used in an application such as a pneumatic rock drill handle to eliminate the need for a control system. Conceptual design methodologies for both damped and undamped fixed and adaptive isolation frequency AVAIs are presented. To determine the effects of tuning the equations were transformed in terms of constant frequency ratios and the variable stiffness ratio. The devices can be controlled using an optimisation approach, but care should be taken since the method could be unsuccessful in some cases.

The design was then applied to a pneumatic rock drill. This application was particularly demanding because the stiffness had to be large enough for the operator to remain in control of the drill, yet low enough to offer isolation. Extensive measurements of drill vibration at a test facility found that the

maximum acceleration values were 18.72 m/s^2 . The maximum allowed under the proposed European Union legislation is 10 m/s^2 for short durations. The excitation consisted of a large tonal component and wide-band noise. The tonal component contributed $\sim 50\%$ of the total weighted equivalent acceleration experienced by the operator and a vibration absorbing isolator should therefore be an ideal solution. The measurements also showed that the excitation frequency is a function of the supply air pressure. By using the supply air pressure to feed the air spring the device could be made self-tuning. Numerical simulation showed that there is only a slight difference between using the supply pressure and forcing coincidence of the excitation and isolation frequencies. It was also found that the vibration levels could be reduced to below 10 m/s^2 in some cases.

Keywords: vibration, absorber, isolator, tonal time-varying, control, optimisation

Opsomming

Die ontwikkeling van 'n vibrasie-absorberende isolator met hefboomversterking vir enkelfrekwensie tyd-variante opwekking

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Vibrasie-isolasie is 'n prosedure wat ten doel het om die transmissie van ossillerende verplasinge en kragte te verminder. Die ideale isolator sal toerusting staties ondersteun sonder dat dinamiese kragte oorgedra word. 'n Isolator met oneindige statiese styfheid en geen dinamiese styfheid nie sal aan dié vereiste voldoen. Alhoewel ideale isolasie nie in die praktyk haalbaar is nie, kan dit benader word deur 'n verskeidenheid van toestelle. Hierdie benadering vind slegs oor 'n beperkte frekwensieband plaas en metodes wat dié band kan vergroot is bestudeer. Die doelwit van die proefskrif is om die bestaande kennis van meganiese stelsels wat die ideale isolator kan benader, uit te brei.

Om die verskillende toestelle met mekaar te vergelyk is die geblokkeerde-oordrag dinamiese styfheid gedefinieër. Daar is gevind dat hierdie waarde die isolator se eienskappe weerspieël sonder die komplikasie van die toerusting wat geïsoleer word, soos wat gebeur met die tradisionele transmissiemetodes. Drie groepe toestelle word onderskei, naamlik isolators, vibrasie-absorberende isolators (VAI) en vibrasie-absorberende isolators met hefboomversterking (AVAI). Die laaste twee tipes maak gebruik van nodalisering om die dinamiese styfheid oor 'n beperkte frekwensieband te verminder. Die fokus van die werk is om die effektiewe lae-styfheid bandwydte te verlaag deur die isolator se eienskappe te wysig. In gevalle waar die opwekking gedomineer word deur 'n enkele frekwensie wat tyd-variant is, kan die toestelle met groot sukses gebruik word.

Twee unieke vibrasie-absorberende isolators met hefboomversterking is bestudeer in die tyd- en frekwensiedomeins. Die tipe I AVAI verander die styfheid van die reservoier wand om sodoende die isolasiefrekwensie te verander. Die tipe II AVAI maak gebruik van 'n swaar metaalprop om die absorbeerdermassa te vermeerder. Beide toestelle maak gebruik van veranderbare lugdrukvere om hul styfheid te verander. Die gebruik van lugvere is gerieflik, dra min demping by en kan saam met 'n lugdrukboor gebruik word op so 'n manier dat 'n beheerstelsel onnodig is. Konseptuele ontwerpmetodologie is voorgestel vir beide ongedemp en gedempte, veranderbare en vaste frekwensie toestelle. Om die effek van instemming beter te bestudeer is die vergelykings getransformeer in terme van konstante frekwensieverhoudings en die veranderbare-styfheidverhouding. Die toestelle is beheer deur van optimalisering gebruik te maak, maar hierdie metode moet omsigtig benader word aangesien die tegniek in sekere gevalle nie sal werk nie.



Die ontwerpmetodologië is vervolgens toegepas op 'n lugdrukrotsboor. Die toepassing is uitdagend aangesien die styfheid enersyds hoog genoeg moet wees sodat die operateur die boor effektief kan gebruik en andersyds laag genoeg moet wees sodat genoegsame isolasie verskaf word. 'n Groot aantal metings van boorvibrasie is gedoen by 'n toetsfasiliteit. Die maksimum geweegde ekwivalente versnelling wat gemeet is, was 18.72 m/s^2 . Die maksimum wat toegelaat word deur voorgestelde wetgewing in die Europese Unie is 10 m/s^2 , en dit net vir kort periodes. Daar is gevind dat hoë, noubandopwekking ~50% bydra tot die geweegde ekwivalente versnelling en 'n vibrasie-absorberende isolator blyk dus die ideale oplossing te wees. Die metings het ook getoon dat die opwekfrekwensie 'n liniêre funksie is van die toevoerdruk. Die toevoerdruk kan dus direk gebruik word om die lugveerstyfheid te bepaal en sodoende kan die behoefte vir 'n beheerstelsel uitgeskakel word. Numeriese simulاسie het getoon dat deur dit te doen daar slegs 'n klein verskil in effektiwiteit is teenoor 'n situasie waar die beheerstelsel die opwek- en isolasie frekwensies ekwivalent geforseer het. Daar is ook gevind dat die vibrasie tot onder 10 m/s^2 verminder kan word in sekere gevalle.

Sleutelwoorde: vibrasie, absorbeerder, isolator, enkelfrekwensie tyd-variant, beheer, optimering

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Table of contents

1	Introduction to vibration-absorbing isolators	1
1.1	Background	3
1.2	Isolators	8
1.2.1	Passive isolators	8
1.2.2	Adaptive isolators	10
1.2.3	Active isolators	10
1.3	Vibration-absorbing isolators	13
1.3.1	Passive vibration-absorbing isolators	14
1.3.2	Adaptive vibration-absorbing isolators	18
1.3.3	Active vibration-absorbing isolators	23
1.4	Amplified vibration-absorbing isolators	26
1.4.1	Passive amplified vibration-absorbing isolators	26
1.4.2	Adaptive amplified vibration-absorbing isolators	34
1.4.3	Active amplified vibration-absorbing isolators	41
1.5	Thesis objectives	44
1.6	Thesis description	45
2	Novel adaptive amplified vibration-absorbing isolators	47
2.1	Adaptive AVAI with variable reservoir wall flexibility (Type I)	48
2.1.1	Reservoir flexibility covering full wall	48
2.1.2	Reduced-area reservoir wall flexibility	64
2.1.3	Single flexible wall	66
2.2	Adaptive AVAI with slug (Type II)	68
2.2.1	Slug springs	69
2.2.2	Slug stops	77
2.2.3	Leakage	79
2.2.4	Slug with diaphragm seal	83
2.3	Conclusion	84
3	Adaptive control methods	85
3.1	Type I AVAI	85
3.1.1	Tuning at start-up	91
3.1.2	Sudden excitation frequency change	93
3.1.3	Slow change in excitation frequency or AVAI properties	94
3.1.4	General tuning method	95
3.2	Type II AVAI	97
3.3	Conclusion	98
4	Design methodologies of AVAIs for a pneumatic rock drill handle	100
4.1	Introduction	100
4.2	Vibration reduction of tool handles	100
4.3	Vibration measurement of a Boart Longyear S250 pneumatic rock drill	104
4.4	Type I amplified vibration-absorbing isolator	109
4.4.1	Narrow-band excitation with noise	109
4.4.2	Type I AVAI design	114
4.5	Type II amplified vibration-absorbing isolator	134

4.5.1	Narrow-band excitation with noise	134
4.5.2	Type II AVAI design	137
4.6	Conclusion	145
5	Experimental results	147
5.1	Introduction	147
5.2	Type I AVAI	148
5.2.1	Experimental results	148
5.2.2	Parameter estimation	151
5.3	Type II AVAI	155
5.3.1	Experimental results	155
5.3.2	Parameter estimation	157
5.4	Control	160
5.5	Conclusion	164
6	Conclusions	165
7	References	168
Appendix A		172
A.1	Background (dynamic stiffness of the relaxation model)	173
A.2	Isolators	174
A.2.1	Passive isolator (intermediate mass isolator)	174
A.2.2	Active isolator (absolute velocity feedback isolator)	175
A.2.3	Active isolator (general feedforward active isolator)	176
A.3	Vibration-absorbing isolators	177
A.3.1	Passive vibration-absorbing isolator	177
A.3.2	Passive vibration-absorbing isolator (multiple-absorber VAI)	180
A.3.3	Passive vibration-absorbing isolator (non-linear VAI)	182
A.3.4	Adaptive vibration-absorbing isolator	183
A.3.5	Active vibration-absorbing isolator (acceleration and displacement feedback)	186
A.3.6	Active vibration-absorbing isolator (relative velocity feedback)	188
A.3.7	Active vibration-absorbing isolator (absolute velocity feedback)	190
A.4	Amplified vibration-absorbing isolators	192
A.4.1	Passive amplified vibration-absorbing isolator	192
A.4.2	Passive amplified vibration-absorbing isolator (multiple absorbers fitted)	197
A.4.3	Passive amplified vibration-absorbing isolator (non-linear)	200
A.4.4	Passive amplified vibration-absorbing isolator (motion transformation system)	200
A.4.5	Adaptive amplified vibration-absorbing isolator	202
A.4.6	Active AVAI (acceleration and displacement feedback)	205
A.4.7	Active AVAI (relative velocity feedback)	206
A.4.8	Active AVAI (absolute velocity feedback)	208
Appendix B		209
B.1	Adaptive AVAI with variable reservoir wall flexibility (Type I)	210
B.1.1	Reservoir flexibility covering full wall	210
B.1.2	Reduced-area reservoir wall stiffness	217
B.2	Adaptive AVAI with slug (Type II)	220
B.2.1	Slug springs	220
B.2.2	Slug stops	227



B.2.3 Leakage	230
B.2.4 Slug with diaphragm seal	235
Appendix C	238
C.1 Type I AVAI (equation of motion)	239
C.2 Type I AVAI (quadrature objective function)	240
C.3 Type II AVAI (equation of motion)	240
Appendix D	241
D.1 Vibration measurement of a Boart Longyear S250 rock drill (calibration factors)	242
D.2 Type I AVAI design (air spring stiffness)	242
D.3 Type I AVAI design (heavy liquid properties)	243
D.4 Type I AVAI design (forces acting on the drill)	244
D.5 Type I AVAI design (forces acting on the handle)	245
D.6 Type II AVAI design (effective area calculation)	246
D.7 Type II AVAI design (damped design method)	247
Appendix E	249
E.1 Refined model for a type I AVAI	250



Nomenclature

Symbol	Description	Unit
a	Acceleration	m/s^2
A_0	Orifice area	m^2
A_a	Port area	m^2
A_b	Reservoir area	m^2
A_e	Effective area	m^2
$a_{h,j}$	j^{th} $1/3$ octave-band acceleration	m/s^2
$a_{h,w}$	Frequency-weighted hand-transmitted acceleration	m/s^2
A_r	Area ratio	
c	Damping coefficient	Ns/m
C	Discharge coefficient	
C_c	Constant discharge coefficient	
c_n	Complex Fourier coefficient	
d_0	Orifice diameter	m
d_a	Port diameter	m
d_b	Reservoir diameter	m
d_i	Inner diameter	m
d_o	Outer diameter	m
E	Young's modulus	Pa
e_i	i^{th} Unit vector	
F	Force amplitude	N
f	Frequency	Hz
f	Objective function value	
f_c	Control force	N
F_T	Transmitted force amplitude	N
G	Shear modulus	Pa
h	Hysteretic damping coefficient	N/m
h	Height	m
H	Transfer function	
i	$\sqrt{-1}$	
I	Moment of inertia	m^4
I_G	Mass moment of inertia about the centre of gravity	$kg.m^2$
k	Stiffness	N/m
k_G^r	Radial geometric stiffness	N/m
k_G	Axial geometric stiffness	N/m
k_{Gh}	Axial geometric stiffness excluding height effects	N/m
K_j	j^{th} $1/3$ octave band weighting factor	
l	Port length (type I)	m
l	Slug length (type II)	m
l_c	Slug spring compressed length	m
l_e	Protrusion length	m



l_i	Inner length	m
l_o	Outer length	m
l_p	Port length (type II)	m
l_r	Reservoir length	m
l_T	Total length	m
\dot{m}	Mass flow rate	kg/s
m	Mass	kg
M	Narrow-band excitation amplitude	
n	Ratio of specific heats	
N	Normal load	N
p	Power	W
P, p	Pressure	Pa
P_o, P_i	Initial pressure	Pa
R	Rayleigh energy	J
R	Gas constant	J/kg.K
r	Radius	m
r_i	Inner radius	m
R_i	Reaction force i	N
r_o	Outer radius	m
S_{xx}	Power spectral density	
S_{xy}	Cross spectral density	
T	Kinetic energy	J
T_o	Initial temperature	K
T_r	Transmissibility	
T_s	Supply air temperature	K
V	Potential energy	J
V	Volume	m ³
W	One-sided power spectral density	Unit ² /Hz
X, Y, U	Harmonic excitation amplitude	m
x, u, y	Time-dependent displacement	m
α	Non-linearity parameter	
α	Lead screw helix angle	rad
α, β, γ	Gain	
δ_{st}	Static deflection	m
ΔW	Energy loss per cycle	J
$\Delta\omega_e$	Excitation bandwidth	rad/s
$\Delta\Omega_e$	Noise bandwidth	rad/s
ε	Noise amplitude	
ε	Radial clearance	m
ζ	Damping ratio	
η	Loss factor	
θ	Angle	rad
λ	Reduced area ratio	
μ_A	Area ratio	
μ_k	Stiffness ratio	



μ_m	Mass ratio	
ν	Poisson's ratio	
ρ	Density	kg/m ³
φ, θ	Rotation angle	rad
ω	Circular frequency	rad/s
ω_e	Excitation frequency	rad/s
ω_i	Isolation frequency	rad/s
Ω_i	Damped isolation frequency	rad/s
ω_n	Natural frequency	rad/s
Ω_n	Damped natural frequency or natural frequency of a two degree of freedom system	rad/s