

**A Novel Suspended-Load Backpack with
Manually Adjustable Impedance
to Reduce Metabolic Cost of Carrying Loads**

by
Batuhan Toker

**Submitted to
the Graduate School of Engineering and Natural Sciences
in partial fulfillment of
the requirements for the degree of
Master of Science**

SABANCI UNIVERSITY

July 2021



© Batuhan Toker 2021
All Rights Reserved

A Novel Suspended-Load Backpack with Manually Adjustable Impedance to Reduce Metabolic Cost of Carrying Loads

Batuhan Toker

Approved by:



Date of Approval:

ABSTRACT

A Novel Suspended-Load Backpack with Manually Adjustable Impedance
to Reduce Metabolic Cost of Carrying Loads

Batuhan Toker

Mechatronics Engineering, M.Sc. Thesis, July 2021

Thesis Advisor: Prof. Dr. Volkan Patoğlu

Keywords: Biomechanics, wearable assistive devices, suspended load backpack, load carriage, metabolic cost, vibration isolation, inerter, variable impedance devices

Load carriage is an unavoidable part of routine activities in daily life. The additional weight and inertial forces due to carried loads lead to increased metabolic cost during locomotion. More importantly, sustained exposure to repetitive loads on the musculoskeletal system may lead to long-term injuries.

Suspended-load backpacks have been proposed as a means to enhance the energy efficiency of the load carriage during walking. Suspended-load backpacks not only decrease peak reaction forces during load carriage, but also modify their timing within the gait cycle, by establishing an out-of-phase movement between the carried load and the human torso. Typical implementations of suspended-load backpacks rely on a highly compliant suspension with low damping. However, effective implementation of suspended-load backpacks is challenging, as low stiffness suspensions result in excessive deflections under high loads, and the effectiveness of these suspensions is limited to a relatively narrow range of payloads and walking speeds.

In this work, we present the design, implementation, and human subject evaluations of a novel suspended-load backpack that features an inerter as well as a manually adjustable impedance. The adjustable impedance of the underlying passive suspension of the backpack aims to effectively reduce the metabolic cost of load carriage over a large range of payloads and walking speeds. The proposed passive suspension relies on two distinct approaches. On the one hand, the negative transverse stiffness characteristic of axially loaded beams is utilized to decrease the effective suspension

stiffness, as this method can ensure mechanical integrity and load-bearing capabilities of the system, while achieving very low stiffness levels. On the other hand, an inerter is introduced between the load and the frame to increase effective inertia without adding significant weight to the system. The simultaneous use of both strategies results in efficient and compact designs with appropriate filtering characteristics over a large range of payloads and walking speeds. The effectiveness of the proposed backpack design on the metabolic cost reduction during walking is verified through a set of human subject experiments.



ÖZETÇE

Yük Taşımanın Metabolik Maliyetini Azaltmak için Manuel Olarak Ayarlanabilir
Empedansa Sahip Özgün Bir Asılı Yük Sırt Çantası

Batuhan Toker

Mekatronik Mühendisliği, Yüksek Lisans Tezi, Temmuz 2021

Tez Danışmanı: Prof. Dr. Volkan Patoğlu

Anahtar Kelimeler: Biyomekanik, giyilebilir yardımcı cihazlar, asılı yük sırt çantası, yük taşıma, metabolik maliyet, titreşim izolasyonu, inerter, değişken empedans cihazlar

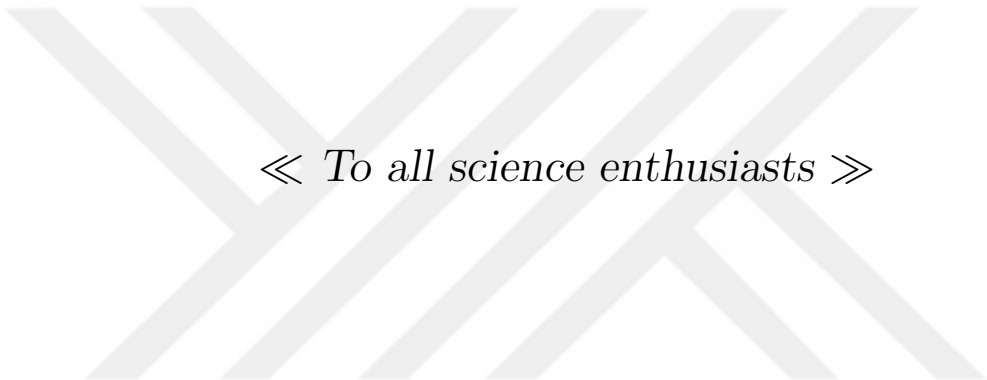
Yük taşıma, günlük hayatta rutin faaliyetlerin kaçınılmaz bir parçasıdır. Taşınan yüklerden kaynaklanan ek ağırlık ve ataletle bağlı kuvvetler, hareket sırasında artan metabolik maliyete yol açar. Daha da önemlisi, kas-iskelet sistemi üzerindeki tekrarlayan yüklere sürekli maruz kalmak uzun süreli yaralanmalara neden olabilir.

Yürüme sırasında yük taşıyan kullanıcının enerji verimliliğini artırmanın bir yolu olarak, asılı yük sırt çantaları önerilmiştir. Asılı yük sırt çantaları, yük taşıma sırasında yalnızca maksimum reaksiyon kuvvetlerini azaltmakla kalmaz, aynı zamanda taşınan yük ile insan gövdesi arasında faz dışı bir hareket oluşturarak kuvvetlerin yürüyüş döngüsü içindeki zamanlamasını da değiştirir. Asılı yük sırt çantalarının tipik uygulamaları, çok esnek ve düşük sönümlemeli bir süspansiyona dayanır. Bununla birlikte, düşük sertlikteki süspansiyonlar, yüksek yükler altında aşırı deplasmanlara maruz kaldığından bu süspansiyonların etkinliği, nispeten dar bir faydalı yük ve yürüme hızı aralığı ile sınırlıdır. Bu nedenle asılı yüklü sırt çantalarının etkili bir şekilde uygulanması zordur.

Bu çalışmada, inerterle donatılmış ve manuel olarak ayarlanabilen empedans özellikleri gösteren özgün bir asılı yük sırt çantasının tasarımını, uygulamasını ve insanlı deneyler ile değerlendirmelerini sunuyoruz. Sırt çantasının temelini oluşturan pasif süspansiyonun ayarlanabilir empedansı, çok çeşitli yüklerde ve yürüme hızlarında yük taşımanın metabolik maliyetini etkin bir şekilde azaltmayı amaçlamaktadır. Önerilen pasif süspansiyon, iki farklı yaklaşıma dayanmaktadır. Bir yandan, süspansiyon esnekliğini arttırmak için eksenel yüklü kirişlerin negatif enine sertlik özelliği kullanılır, çünkü bu yöntem sayesinde çok yüksek esneklik seviyelerine ulaşılabilir. Aynı zamanda sistemin mekanik bütünlüğü ve yük taşıma kapasitesi de korunabilir. Öte yandan, sisteme önemli seviyede ağırlık eklemekten etkin ataleti artırmak için yük ile gerçeve

arasına bir inerter yerleřtirilir. Her iki stratejinin aynı anda kullanılmasıyla, ok eřitli ykler ve yrme hızları zerinde uygun filtreleme zelliklerine sahip verimli ve kompakt bir tasarım elde edilmiřtir. nerilen sırt antası tasarımının yryř sırasında metabolik maliyeti dřrme zerindeki etkinlięi, bir dizi insanlı deneyle doęrulanmıřtır.





« *To all science enthusiasts* »

ACKNOWLEDGEMENTS

First and foremost, I would want to express my gratitude to my supervisor, Prof. Dr. Volkan Patoglu. His informative suggestions encouraged me to improve my understanding and raise the quality of my work. It is a great privilege to work under his supervision.

I'd like to express my gratitude to Prof. Dr. Çağatay Başdoğan and Asst. Prof. Dr. Bekir Bediz for agreeing to be my jurors.

Last but not least, I would like to thank the HMI group, lab staff, friends and my family for their support and company during my master's studies.

This study was partially supported by TÜBİTAK Grant 216M200.

Table of Contents

Abstract	iii
Özetçe	v
Acknowledgements	viii
Table of Contents	ix
List of Figures	xiii
List of Tables	xx
Nomenclature	xxiii
1 Introduction	1
1.1 Contributions	5
1.2 Outline	7

2	Related work	8
2.1	Load Carriage	8
2.1.1	Medical issues related to load carriage	9
2.1.2	Effects of load carriage on the human gait cycle	10
2.1.3	State of the art of the load-carriage devices	10
2.2	Inerter	12
2.3	Variable impedance mechanisms	15
3	Mathematical Modeling	17
3.1	Dynamic model of walking with load	17
3.1.1	Inerter based suspended-load backpack	20
3.2	Transmissibility	34
3.2.1	Displacement transmissibility	34
3.2.2	Force transmissibility	39
4	Design of Inerter-based Suspended Load Backpack	42
4.1	Overall design	42
4.2	Variable stiffness module	43
4.2.1	Analytical model	44
4.2.2	Numerical simulations	48
4.3	Inerter design	49
4.3.1	Analytical model	50
4.3.2	Numerical evaluations	52

5	Implementation and Experimental Characterization	55
5.1	Implementation of the inerter module	55
5.1.1	Experimental setup	56
5.1.1.1	System model	57
5.1.2	Actuator dynamics	57
5.1.3	Characterization results	59
5.1.3.1	System characteristics	59
5.1.3.2	Discussion	64
5.2	Implementation of the variable stiffness module	66
5.2.1	Experimental setup	67
5.2.1.1	System model	67
5.2.2	Actuator Dynamics	68
5.2.3	Characterization Results	69
5.2.3.1	Variable stiffness module	69
5.2.3.2	The inerter module with the variable stiffness module	74
5.2.4	Discussion	79
6	Human Subject Experiments	83
6.1	Participants	84
6.2	Task	84
6.3	Hypotheses	85
6.4	Experimental procedure	85

6.5	Performance Metrics	87
6.5.1	Quantitative Metrics	88
6.5.2	Qualitative Metrics	90
6.6	Analysis of results	91
6.7	Experimental Results	92
6.7.1	Quantitative Metrics	92
6.7.2	Qualitative Metrics	111
6.8	Discussion	113
7	Conclusion	117
	Bibliography	119
	Appendix A	127
	Appendix B	132

List of Figures

1.1	Backpacks for soldiers, hikers, school children, and astronauts	1
2.1	Correspondence between mechanical and electrical networks	13
2.2	Schematic representation of inerter	14
3.1	A single degree of freedom backpack model with a passive suspension	18
3.2	Walking frequency according to Eqn. 3.2 for varying human heights and walking speeds	19
3.3	Additional deflection due to the load	26
3.4	Oscillation force amplitude and phase shift profile for proposed back- pack on a user with a 1.75 m height, 15 kg backpack mass, 1.5 m/s walking speed, for varying spring stiffness and damping values.	27
3.5	Oscillation force amplitude and phase shift profile for proposed back- pack on a user with a 1.75 m height, 15 kg backpack mass, 200 Ns/m damping, 1.5 m/s walking speed, for varying spring stiffness and in- tertance values.	28
3.6	Oscillation load amplitude and phase shift profile for proposed back- pack on a user with a 1.75 m height, 15 kg backpack mass, 50 kg inertance, 200 Ns/m damping, for varying walking speed and spring stiffness values	29

3.7	Oscillation load amplitude and phase shift profile for suspended load backpack on human with weight 1.75m, walking velocity 1.5 m/s, backpack mass 15 kg, for varying spring stiffness and damping	31
3.8	Oscillation load amplitude and phase shift profile for suspended load backpack on a user with a 1.75 m height, 15 kg backpack mass, 200 Ns/m damping, for varying waking speeds and spring stiffness values	32
3.9	Oscillation force contour plot for the proposed backpack suspension on user with an height of 1.75m, for backpack mass 15 kg, stiffness 5000 N/m, damping 200 Ns/m, for varying walking speeds and pack loads	33
3.10	Relative oscillation amplitude contour plot for the proposed backpack suspension on user with an height of 1.75m, for backpack mass 15 kg, stiffness 5000 N/m, damping 200 Ns/m, for varying walking speeds and pack loads	33
3.11	Transfer of motion through the proposed backpack device	34
3.12	System's displacement transmissibility curve for varying values of damping coefficient b and $m= 50$ kg, $M= 25$ kg and $k= 3000\text{N/m}$. .	35
3.13	System's displacement transmissibility curve for varying values of in-ertance factor m and $M=25$ kg, $k= 3000\text{N/m}$, $b= 100\text{Ns/m}$	36
3.14	System's displacement transmissibility curve for varying values of stiffness k and $m= 50$ kg, $M= 25$ kg, $b= 100\text{Ns/m}$	37
3.15	System's displacement transmissibility curve for varying values of stiffness k and in-ertance m for $M= 15$ kg and $b= 100\text{Ns/m}$	38
3.16	System's displacement transmissibility curve for varying values of stiffness k and in-ertance m for $M= 15$ kg	38
3.17	System's displacement transmissibility curve for varying values of stiffness k and in-ertance m for $M= 15$ kg	39

3.18	System's force transmissibility curve for varying values of damping coefficient b and $m=50$ kg, $M=25$ kg and $k=3000$ N/m	40
3.19	System's force transmissibility curve for varying values of inertance factor m and $M=25$ kg, $k=3000$ N/m, $b=100$ Ns/m	41
3.20	System's force transmissibility curve for varying values of stiffness k and $m=50$ kg, $M=25$ kg, $b=100$ Ns/m	41
4.1	Schematic representation of the overall design that consists of as a parallel connection of inerter with a variable stiffness module.	43
4.2	Schematic model of axially loaded buckling beam.	44
4.3	Stiffness change of axially loaded beams under no axial, axial tensile, and compression loadings	48
4.4	Stiffness change of axially loaded beams under no axial, axial tensile, and compression loadings	49
4.5	Schematic representation of mechanical model of an inerter	50
4.6	Contour plot of $m_{\text{effective}}$ for different values of flywheel outer radius and thickness for the parameters given in the Table 4.2	54
5.1	Experimental system identification of the variable negative stiffness module under no axial loading	56
5.2	Schematic representation of the inerter characterization experiment	57
5.3	Schematic representation of the linear actuator	58
5.4	Experimental system identification of the actuator bandwidth	58
5.5	Experimental system identification of the inerter module with low inertance configuration	61
5.6	Experimental system identification of the inerter module with medium inertance configuration	62

5.7	Experimental system identification of the inerter module with high inertance configuration	63
5.8	Experimental system identification of the variable negative stiffness module under no axial loading	65
5.9	Prototype of experimental setup for inerter-variable stiffness based suspended load backpack design	66
5.10	Schematic representation of the experimental setup	67
5.11	Experimental system identification of the variable negative stiffness module under no axial loading	70
5.12	Experimental system identification of the variable negative stiffness module under tensile loading	72
5.13	Experimental system identification of the variable negative stiffness module under compression load.	73
5.14	Experimental system identification of the inerter module with no fly-wheel and variable negative stiffness module under compression. . . .	75
5.15	Experimental system identification of the inerter module with a relatively low inertia flywheel and variable negative stiffness module under compression	76
5.16	Experimental system identification of the inerter module with a relatively high inertia flywheel and variable negative stiffness module under compression	78
5.17	Experimental system identification of the variable stiffness module with a three different configurations	79
5.18	Experimental system identification of the inerter module with varying levels of inertance	81
6.1	Experimental setup with a participant from back and side views . . .	84

6.2	Four different tasks used in experiments: (Trial 0) no load (NL), (Trial 1) suspension-locked load carriage (SL mode), (Trial 2) suspended load with variable stiffness module (VS mode), and (Trial 3) inerter + variable stiffness modules (IVS mode)	86
6.3	Experimental design consists of a single seating, where trials are administered in a randomized order	86
6.4	Flow activity of the System Usability Scale (SUS)	90
6.5	Mean and SD values of net metabolic power for trials. * and o indicate a significant difference of $p < 0.001$ and $p < 0.05$ respectively (NL: No load, SL: suspension-locked, VS: variable stiffness mode, IVS: inerter+variable stiffness mode)	92
6.6	Means and standard deviations of percentage of net metabolic power reduction of each modules with respect to SL condition.	94
6.7	Mean and standard deviation values of normalized heart rate	95
6.8	Mean and standard deviation values of peak accelerative force (N). * indicates a significant difference ($p < 0.001$)	96
6.9	Mean and standard deviation values of percentage change of peak accelerative forces	98
6.10	Mean and standard deviation values of vertical excursion of the torso (m) * and o indicate a significant difference of $p < 0.001$ and $p < 0.05$ respectively	99
6.11	Mean and standard deviation values of vertical excursion of the pack load (m) for the trials of loading conditions of suspension locked (SL) mode, variable stiffness (VS) mode and inerter-variable stiffness (IVS) mode. * and o indicate a significant difference of $p < 0.001$ and $p < 0.05$ respectively	101

6.12	Mean and standard deviation values of the relative displacement between pack and torso (m). * indicates a significant difference ($p < 0.001$) between the conditions.	102
6.13	Mean and standard deviation values phase difference between pack and torso (degree). * and $\circ$ indicate a significant difference of $p < 0.001$ and $p < 0.05$ respectively	104
6.14	Mean and standard deviation values of human torso, pack load and relative displacement between torso and pack load during SL mode. .	106
6.15	Mean and standard deviation values of human torso, pack load and relative displacement between torso and pack load during VS mode. .	107
6.16	Mean and standard deviation values of human torso, pack load and relative displacement between torso and pack load during IVS mode.	108
6.17	Mean and standard deviation values of accelerative forces due to pack load, during trials.	109
6.18	Mean and standard deviation values of accelerative forces due to pack load and $F_{\text{suspension}}$ during VS condition.	110
6.19	Mean and standard deviation values of accelerative forces due to pack load and $F_{\text{suspension}}$ during IVS condition.	110
6.20	Adjective rating scores of trials with load carriage according to the SUS. $\circ$ indicates a significant difference of $p < 0.05$	111
7.1	Different compliance layouts for $Q(s)$ consist of spring-damper-inerter	128
7.2	Backpack performance comparison for layout (a) and (c) given in Figure 7.1 on human with weight 1.78 m, walking speed 5 km/h, backpack mass 18.5 kg, stiffness k 3300 N/m, damping b 96 Ns/m and inertance m 5kg	129

7.3	Backpack performance comparison for layout (c) and (b) given in Figure 7.1 on human with weight 1.78 m, walking speed 5 km/h, backpack mass 18.5 kg, stiffness k 3300 N/m, damping b 96 Ns/m and inertance of m 5 kg	130
7.4	Oscillation load profile for layout(b) on human with weight 1.78m, walking speed 5km/h, backpack mass 18.5 kg, and inertance factor m 10 kg	131



List of Tables

3.1	Description of variables and parameters used in the mathematical model	19
4.1	Parameters used to study buckling beams for numerical analysis . . .	48
4.2	Parameters characterizing the ball-screw inerter.	53
5.1	Inerter characteristics when no flywheel is attached to the nut	61
5.2	Inerter characteristics when the device is configured for medium level inertance	63
5.3	Inerter characteristics when the device is configured for high level inertance	64
5.4	Variable negative stiffness module characteristics under no axial loading.	71
5.5	Variable negative stiffness module characteristics under tension force.	72
5.6	Variable negative stiffness module characteristics under axial com- pression	73
5.7	Inerter with no flywheel and variable negative stiffness module char- acteristics under compression	75
5.8	Inerter with medium level inertia flywheel and variable negative stiff- ness module characteristics under compression	77

5.9	Inerter with high level inertia flywheel and variable negative stiffness module characteristics under compression	78
5.10	Natural frequency estimations for variable stiffness module	80
5.11	Natural frequency estimations when both modules are employed . . .	80
6.1	Backpack properties	83
6.2	Full list of the order of trials for each seating	87
6.3	Percentages of occurrence of trials. Each trial is repeat 24 times and their order is uniformly distributed by using latin square method . . .	87
6.4	Parameters measured during the experiments as quantitative metrics	88
6.5	Original System Usability Scale	90
6.6	System Usability Scale adjective rating	91
6.7	Mauchly's Test of Sphericity for normalized net metabolic power mea- surements	93
6.8	Tests of Within-Subjects Effects for net metabolic power measurements	93
6.9	Pairwise Comparisons for net metabolic power measurements	94
6.10	Mauchly's Test of Sphericity for normalized heart rate measurements	95
6.11	Tests of Within-Subjects Effects for normalized heart rate measurements	96
6.12	Mauchly's Test of Sphericity for peak accelerative force measurements	97
6.13	Tests of Within-Subjects Effects for peak accelerative force measure- ments	97
6.14	Pairwise Comparisons for peak accelerative force measurements . . .	97
6.15	Mauchly's Test of Sphericity for peak-peak torso excursion	99
6.16	Tests of Within-Subjects Effects for peak-peak torso excursion	99

6.17	Pairwise Comparisons for peak-peak torso excursion measurements	100
6.18	Mauchly's Test of Sphericity for peak-peak pack load excursion measurements	101
6.19	Tests of Within-Subjects Effects for peak-peak pack load excursion measurements	101
6.20	Pairwise Comparisons for peak-peak pack load excursion measurements	102
6.21	Mauchly's Test of Sphericity for peak-peak relative displacement measurements	103
6.22	Tests of Within-Subjects Effects for peak-peak relative displacement measurements	103
6.23	Pairwise Comparisons for peak-peak relative displacement measurements	103
6.24	Mauchly's Test of Sphericity for phase difference measurements	105
6.25	Tests of Within-Subjects Effects for phase difference measurements	105
6.26	Pairwise Comparisons for phase difference measurements	105
6.27	Mauchly's Test of Sphericity for adjective rating scores	111
6.28	Tests of Within-Subjects Effects for adjective rating scores	112
6.29	Pairwise Comparisons for adjective rating scores	112
6.30	Comparison of our results with existing suspended load backpack designs. † retrieved from [1].	113
7.1	Questionnaire filled out by participants to evaluate the qualitative metrics with a 5-point Likert scale.	132
7.2	Questionnaire on compliance with the COVID-19 recommended by CORTEX	133

Nomenclature

b	Backpack damping coefficient
F_{osc}	Oscillating backpack force
F_{backpack}	Total backpack force on human.
ω_h	Human walking frequency
ω_n	Natural frequency of backpack
δ	Static deflection of the backpack
g	Gravity
t	Time
v	Walking speed
S	Height of the person
l_0	Leg length of the person
k	Backpack stiffness coefficient
ϕ	Phase angle
m	Backpack inertance coefficient
m_n	Mass of nut
m_f	Mass of flywheel
$m_{\text{effective}}$	Inertance factor per unit mass of inerter
M	Total backpack mass
$M_{\text{inertance}}$	Total mass of inerter
M_{suit}	Total mass of the device

A_h, A_d	Amplitude of oscillation of human and backpack
r_s	Radius of screw
$r_{i,f}$	Inner radius of flywheel
$r_{o,f}$	Outer radius of flywheel
$r_{i,n}$	Inner radius of nut
$r_{o,n}$	Outer radius of nut
k_f	Radius of gyration of the flywheel
$S_{i,j}$	Scaling factor to compare layouts
$k_{\text{performance}}$	Performance stiffness
$k_{\text{resonance}}$	Resonance stiffness
k_{min}	Stiffness value that corresponds to minimum oscillation load
k_{max}	Stiffness value that corresponds to maximum oscillation load
$T_d(s)$	Displacement transmissibility
$T_f(s)$	Force transmissibility
k_s	Stiffness of the tension spring
F_{comp}	Axial loading exerted on beams
$2R$	Transverse loading
D	Transverse deflection
c_1, c_2	Inflection points
δ_e	The axial deflection of the axially loaded beam due to its elasticity
δ_b	The axial deflection due to buckling of the beam
A	Cross sectional area of beam

E	Modulus of elasticity of beam
P_{cr}	The first critical axial load for a clamped-clamped end points configuration



Chapter 1

Introduction

Load carriage is an unavoidable part of routine activities in daily life. The additional weight and inertial forces due carried loads lead to increased metabolic cost during locomotion. More importantly, sustained exposure to repetitive loads on the musculoskeletal system may lead to long term injuries.



FIGURE 1.1: Backpacks for soldiers, hikers, school children, and astronauts

A backpack is a device that is arranged to be attached on the back of human torso to carry a load. Backpacks are generally used to carry loads for different quests, such as military marching and hiking, as shown in the Figure 1.1.

The load carried with a backpack is commonly coupled to user's torso, and causes the weight of the load to act on the human musculoskeletal system. In addition, during dynamic movements, inertial forces due to acceleration of the load are also transmitted to the human. As a result, heavy loads carried by the humans at relatively high walking speeds, quickly reduce mobility and cause muscle fatigue. Walking with high interaction forces escalates the possibility of musculoskeletal injuries on the torso [2–4] and the lower limbs [5, 6]. Moreover, load carriage is also linked to decreased metabolic efficiency and increased muscle fatigue [7–9]. Researchers have investigated physical effects of load carriage [7, 9, 10] and documented transformations in the gait and posture [11], and changes in muscle activities [8] due to the load. Increasing carried load also corresponds to an increase in the joint reaction forces [12].

Reduction of the metabolic cost during load carriage has been a focus of research. Pierrynowski *et al.* [13] proposed a backpack design that maintains an in-phase motion between the backpack and the torso to reduce the energy expenditure during walking with a load. Rome *et al.* [14] showed that out-of-phase suspended-load backpacks reduce the metabolic cost of load carriage during walking by decreasing vertical force applied to the body by the backpack. Mechanical work required to carry a load during walking is dominantly characterized by the amount of vertical movement of the human's center of mass [15–17], and vertical force applied to the human [14, 18]. Ackerman *et al.* [1] showed that metabolic cost of load carriage can be reduced with a load suspension that has lower resonant frequency than the human walking frequency, as it provides a means of vibration isolation with respect to the body. Through the suspension, the movement of the carried load is decoupled

from the body during the load carriage such that they are out-of-phase with each other during peak inertial forces.

Suspended-load backpacks not only decrease peak reaction forces during load carriage, but also modify their timing within gait cycle, by establishing an out-of-phase movement between the carried load and the human torso. Typical implementations of suspended-load backpacks rely on highly compliant suspensions with low damping. However, an effective implementation of suspended-load backpacks is challenging, as low stiffness suspensions result in excessive deflections under high loads, and effectiveness of fixed suspensions is limited to a relatively narrow range of payloads and walking speeds.

Human walking takes place with a step frequency between 1.5 Hz and 3 Hz, depending on the walking speed and leg length. For a suspended-load backpack, the compliance of the mechanism should be selected to prevent resonance, which occurs when the walking frequency is equal to the natural frequency of the suspension. In particular, resonance frequency of the system ω_n ($= \sqrt{k_{eq}/m_{eq}}$, where k_{eq} denotes the equivalent stiffness and m_{eq} is the equivalent mass), should be kept below the step frequency of human walking in order to prevent the resonance with the load's vertical movement. Low resonance frequency can be achieved with a high equivalent mass and low stiffness. In addition to avoidance of the resonance, adjusting the natural frequency of the suspension can be capitalized upon to achieve an out-of-phase movement between the oscillation of the body center-of-mass and the load. A well performing conventional suspended load backpack provides high compliance with low damping characteristics. However, equivalent mass of the backpack changes significantly under different payloads, negatively impacting the effectiveness of suspension.

Relatively recently introduced two-terminal mechanical element, *inertor*, introduces a proportionality named *inertance* between the force applied at the terminals and

the relative acceleration between terminals, filling in the gap in the force-current analogy [19]. Interter allows engineers to directly translate the passive electrical network synthesis theory to a design mechanical devices in a systematic manner [20]. Inerters are useful as they can be designed to feature a high equivalent mass in comparison to their weight. In particular, inerters can provide inertance levels that are of orders of magnitude higher than their mass [19]. Due to the importance of weight in the design of suspended-load backpack designs, inerters can be advantageous, as they can significantly improve performance without adding too much extra weight. In particular, inerters promise performance improvements when they are employed in a suspended-load backpack designs, as they can reduce sensitivity of the suspension to the changes in payload.

The aim of the study is to design a novel suspended-load backpack with an inerter, as well as a manually adjustable impedance to reduce the metabolic cost of load carriage during walking. We study the effect of a properly tuned passive suspension on improving the metabolic efficacy of loaded walking and reducing the reaction forces at the joints, while walking with a loaded backpack. Our design relies on two complementary approaches to adjust the impedance characteristics of the suspension, through (i) a variable stiffness module, and (ii) with an inerter module. Variable stiffness of the backpack is obtained by modulating the transverse stiffness of axially loaded beams. A ball-screw inerter is designed with different levels of inertance to modulate the effective inertia of the system while introducing a low additional mass. We demonstrate the positive effects of using a mechanical filter based on the proposed variable stiffness and inerter modules in contrast to the conventional suspended-load backpacks, by analyzing the dynamic load on the torso and confirming of metabolic cost reduction of load carriage through human subject experiments.

1.1 Contributions

In this thesis, a suspended-load backpack with manually adjustable impedance is proposed to reduce metabolic cost of load carriage. The proposed backpack is implemented as a parallel connection of a variable stiffness module and an inerter module. Human subject experiments are presented demonstrating the beneficial effects of the proposed suspension on the vibrations of the load and the metabolic cost of load carriage.

The contributions of this thesis can be summarized as follows:

- A novel passive suspension device with manually adjustable impedance is proposed. The proposed passive suspension relies on two distinct approaches. On the one hand, the negative transverse stiffness characteristic of axially loaded beams is utilized to decrease the suspension stiffness, as this method can ensure mechanical integrity and load bearing capabilities of the system while achieving very low stiffness levels. On the other hand, an inerter between the load and the frame is implemented to increase effective inertia without adding significant weight to the system. The simultaneous use of both strategies results in efficient and compact designs with limited oscillations and effective filtering characteristics.
- We have derived analytical models of the proposed system and studied filtering characteristics of the suspension through comprehensive simulations.
- We have implemented the proposed system with manually variable impedance characteristics, experimentally characterized the system dynamics and verified the results against our analytical models.
- We have shown that various stiffness and inertance levels can be achieved with the prototype, effectively modulating system's impedance and vibration

characteristics. In particular, the natural frequency of the backpack device can be reduced to 0.90 Hz with variable stiffness module, 0.36 Hz with the inerter module, and 0.34 Hz when both methods are employed simultaneously. These values are adjustable and can be kept lower than the human walking frequency even for slow walking paces.

- The effectiveness of the proposed backpack on the metabolic cost reduction during load carriage is verified through a set of human subject experiments. Our experimental results provide evidence that variable stiffness module can reduce the peak inertial forces by $30 \pm 11.21\%$ to result in a metabolic cost decrease of $6.16 \pm 3.90\%$, while the simultaneous employment of both inerter and variable stiffness modules can reduce the peak inertial forces by $42.17 \pm 15.48\%$ to result in a metabolic cost decrease of $10.24 \pm 3.67\%$.

1.2 Outline

The rest of the thesis is organized as follows:

In Chapter 2, the related work on load carriage, suspended-load backpacks, inerters, and variable impedance mechanisms are presented.

Chapter 3 provides the design objectives, a mathematical model of load carriage, and simulation-based analysis of the proposed suspension.

In Chapter 4, the mechanical designs of inverter and variable stiffness modules are presented together with their analytical models and numerical verification.

Chapter 5 presents the implementation details and the experimental characterization of each module, together with the overall design.

Chapter 6 details the human subject experiments designed to investigate the efficacy of the proposed backpack. Performance evaluations based on quantitative and qualitative metrics and their discussion are provided.

Chapter 7 concludes the thesis and presents future research directions.

Chapter 2

Related work

In this chapter the related work on load carriage, strategies used to reduce the metabolic cost of load carriage, the concept of inerter, and a review of variable impedance devices are presented.

2.1 Load Carriage

Load carriage is a part of daily activities for most of humanity. By definition, load carriage is locomotion while conveying an extraneous mass that is not a part of the body. Backpacks have become the common way to carry loads for ages. For the most part, the load is supported by the shoulders or the back. On average, load carried by school children has been reported to range from 14% up to 38% of their body weight (BW) [21]. This level of load is associated with back pain in the school children [22]. Moreover, soldiers regularly carry heavier loads with an average of 23 kg [23]. For military operations, U.S. Army Infantry School recommends carrying an optimal load of 30% of BW and a maximal load of 45% of BW [24]. In this section,

the effects of load carriage on the human gait cycle and medical issues related to load carriage are presented.

2.1.1 Medical issues related to load carriage

Additional carried mass introduces increased external load to the body during walking. Furthermore, such loads increase the likelihood of injury and long-term disability, due to significantly larger reaction forces induced at the joints. Load-carrying causes reduced metabolic efficiency [7] and increased muscle fatigue [9]. Energy cost of walking with a load is dominated by the vertical motion of the body [25] and force exerted on the person [14]. Besides the additional metabolic cost, injuries may occur on both upper and lower body [2–4, 26].

Increased stress on the spine and lumbar muscles while carrying a load is the main reason for the injuries on the back [27]. The lumbosacral joint load increases by 26% as the backpack load of 15% of BW is carried, and up to 44.5% for loads of 30% of BW [28]. The human body changes the spine curvature and leans forward/backward to handle the increased load on the joint which causes spinal lordosis or kyphosis [29, 30]. These two issues are linked to back pain and damage to the spinal discs by leading to an increase in torsion and shear forces on them [31]. Besides that, as the forward lean increases, shear forces on the lower limbs are accumulated, since the center of mass of the combined body and backpack mass changes [11].

Llyod *et al.* [32] reported a proportional increase in the vertical ground reaction forces as the load increases. Muscle activity of gastrocnemius and vastus lateralis increases enormously when loads of 40% of BW are carried by woman, which leads to an increase in fatigue and risk of lower limb injury for female hikers [32]. Static and dynamic loads due to the carried mass change the gait and posture, thus more

muscular effort is required in the lower limbs. Injuries at the lower limb are commonly caused by a transformed gait cycle when compensating for the static and dynamic load exerted on human [2, 11, 26, 33].

2.1.2 Effects of load carriage on the human gait cycle

The human gait cycle is a pattern of leg repulse to move the body forward in an interval between ground contact of the same foot [11]. An inverted pendulum can be used to model human walking if we neglect the vertical motions of the pelvis and upper body [34, 35].

An ordinary walking gait cycle is made of 3 phases, called double support, single support, and swing phases. In the swing phase, step length and phase time are reduced as the load increases. On the other hand, the step frequency and time are increased in the double support phase while carrying a load [36–38].

2.1.3 State of the art of the load-carriage devices

Carrying loads on a backpack can significantly decrease the metabolic efficiency of walking, not only due to the added weight but also due to the inertial forces induced by the vertical movements of the backpack. The design of a backpack is a complex multi-criteria problem over a wide range of characteristic parameters such as velocity, load, bulkiness, age, gender, etc. For example, relatively heavy loads are typically carried in military and hiking applications at low walking speeds. Carrying heavy loads during walking can lead to high peak forces on the human body and raising the metabolic cost of walking [9, 10]. However, for school children, a range of different loads may be carried at a comparatively low walking pace. There are settings in which extreme loads are carried, for instance, military tactical equipment can weigh up to 57% of body weight [39].

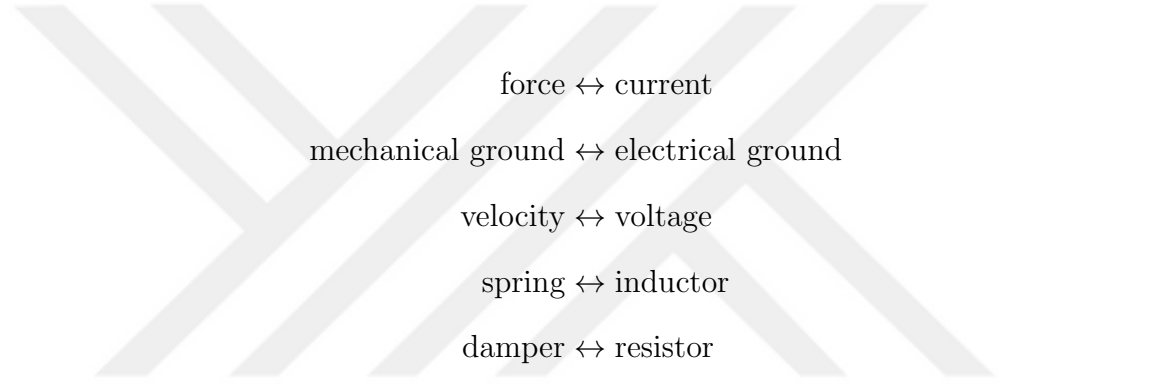
Traditional designs are mainly based on a hip belt and a shoulder support for hiking and military activities. This design framework is beneficial for distributing part of the load on the waist and the torso, by compensating the stress on the shoulders and dorsal [2, 26, 27]. To reduce metabolic cost in situations where significant loads are considered, recent designs feature a direct adjustment of the amount of load transferred to waist [40]. Double-pack backpacks have proposed to locate the carried load close to the human's center of mass (COM) [41]. These type of backpacks make the COM of the carried mass closer to the human's COM and prompt a more natural human gait and posture, with less increase in the muscle activity [42]. Double-packs are not extensively in use due to their drawbacks, such as restricted arm and torso movement, blocked vision, increased heat stress, and ventilatory deterioration [43, 44].

Another approach is to use lower limb or torso exoskeletons to compensate for the issues related to load carriage. With a lower-limb exoskeleton, the mass is rigidly attached to the exoskeleton so that the weight of the payload is transferred directly to the ground [45]. This approach promises a reduction in the net load borne by a human. However, it introduces additional weight due to the exoskeleton attached to the lower limbs. Torso exoskeletons have been proposed to implement dynamic load compensation to reduce the metabolic cost of load carriage during walking [46].

Besides the static load from the backpack, the cyclic vertical acceleration of the torso causes an oscillatory (dynamic) load during walking. This oscillatory force can be in the same magnitude as the static backpack load [14]. The positive effects of walking with a low stiffness backpack have been demonstrated in comparison to existing conventional backpack designs. These advantages include reduction in the oscillating force [47, 48], promising a reduction in the load on the shoulders.

2.2 Inerter

Mechanical and electrical systems have highly similar dynamics, as is generally known. The force and velocity in mechanical systems can be analogized to the current and voltage in electrical systems, respectively, providing the force-current analogy between these systems. The traditional analogies between mechanical and electrical networks have been established by Firestone [49] by introducing the force-current analogy. Force-current (or mobility) analogy between mechanical and electrical networks can be represented by the following accordances:



	force $\leftrightarrow$ current
mechanical ground $\leftrightarrow$ electrical ground	
velocity $\leftrightarrow$ voltage	
spring $\leftrightarrow$ inductor	
damper $\leftrightarrow$ resistor	
potential energy $\leftrightarrow$ magnetic energy	
kinetic energy $\leftrightarrow$ electrical energy	

The spring and damper in mechanical systems can be compared to the inductor and resistor in electrical systems, respectively, in terms of device point of view. However, the mechanical equivalent of the ungrounded electrical capacitor was not identified as a mechanical device. The mechanical equivalent of the grounded capacitor is mass. However ungrounded capacitor cannot be represented by mass since one of its terminals needs to be grounded to an inertial reference frame. Smith introduced a passive mechanical device named *inerter* to complete the force-current analogy [50], allowing for ungrounded capacitors to be represented in the mechanical domain.

According to the definition of inerter, it is an ideal two-terminal device with dynamics analogous to that of a capacitor. The force-current analogy is completed by introducing inerter, and the mechanical networks of spring, damper, and inerter may indeed be directly represented as inductor-resistor-capacitor electrical networks as shown in Figure 2.1.

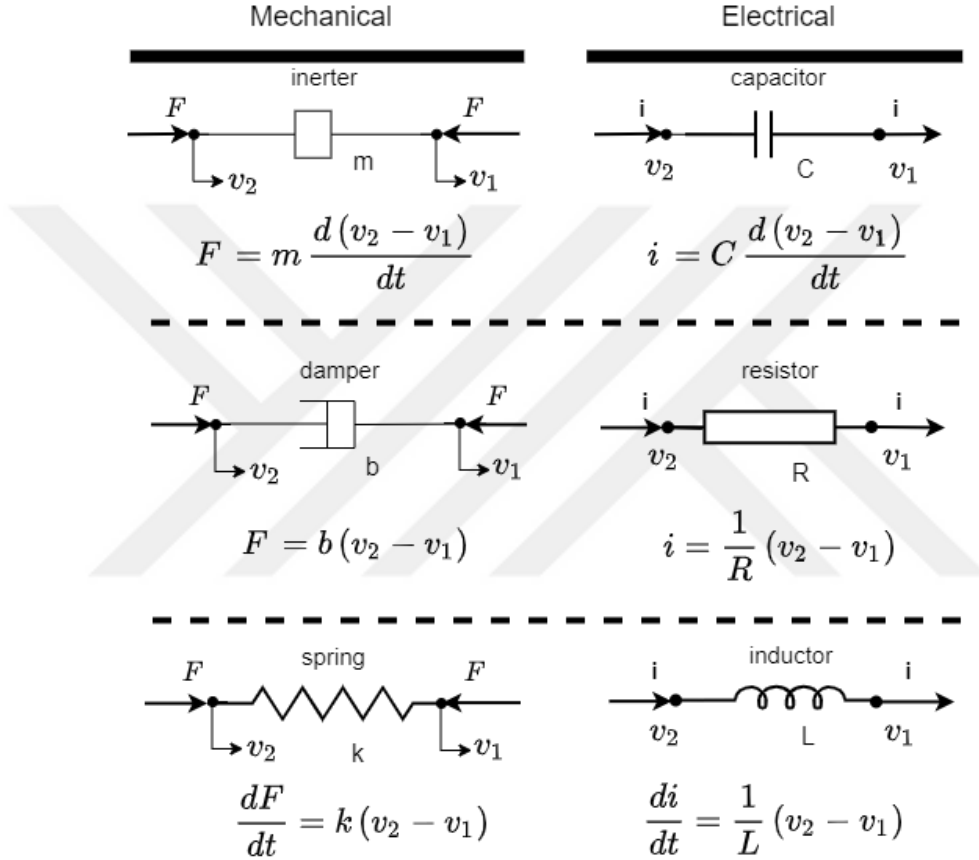


FIGURE 2.1: Correspondence between mechanical and electrical networks

This device can be described as a simulated mass that does not need one terminal to be connected to the ground. Inerters are standard mechanical elements akin to the spring and damper elements. While modeling, controlling and analyzing, its characteristics are not depended on the specific realization of the implementation. By definition, an ideal inerter is a two-terminal, one-port mechanical device with the characteristic that the equal and opposite forces implied from the terminals

are in proportion to the relative acceleration between the terminals. suspension components whose effect on the impedance can be modeled by multiplication of inertance factor and acceleration difference between the terminals as represented in Figure 2.2, where x, v , and a represent displacement, velocity and acceleration of connection points, respectively

$$F = m(a_1 - a_2) \quad (2.1)$$

where $m > 0$, as it denotes the inertance and has unit of kilograms.

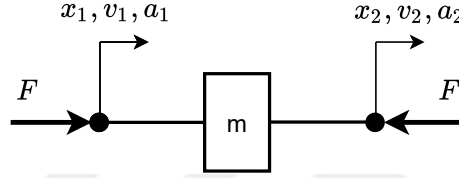


FIGURE 2.2: Schematic representation of inerter

It should be noted that, equal, opposite and co-linear forces are exerted on the terminals. However, the mass of the device is not the same as, and usually much smaller than, its inertance m [50].

In established usage, inerters are realized through fundamental mechanical elements such as rack and pinion [50–52], ball screw [53, 54], gear pump [50], and fluid transmission [55, 56]. In the rack and pinion-based design, the torque is transmitted to the flywheel from the rack through the gears, while the ball-screw inerters directly convert the linear motion into rotary motion. For flywheel based inerters, it has been shown that friction has a large effect on the ideal inertance behaviour of the device [57]. Alternative methods to transmit the power have also been introduced for inerter design. Hydraulic and fluid transmission-based inerter designs promote new applications [58].

Inerters have been employed in a wide range of application areas, such as automotive suspensions [51], railway vehicle suspensions [59], building suspensions [60], vibration absorption [50], and vibration mounts [61].

Theoretically, inerters can have an inertance factor that is orders of magnitude greater than their mass. This motivates us to design, characterize and apply inerter in the passive suspensions to effectively modulate system's effective mass.

2.3 Variable impedance mechanisms

Variable impedance mechanisms are devices whose impedance can be tuned to various grades required by the interactions. These type of mechanisms are beneficial for the devices that handle interactions with various environments. Modulation of the impedance can be obtained in two ways, by designing a passive impedance modulation mechanism or by active control.

In the active control strategy, the control bandwidth limits the frequency range for which modulation is possible, thus the device acts as its uncontrolled dynamics at high frequencies above its control bandwidth. The complexity and need for continued energy source can be considered as the other drawbacks of active control. Increased energy efficiency sets a limit of operation time for the portable device implementations whose performance significantly depends on the weight.

Impedance modulation through the mechanical design is accomplished by adjusting the effective compliance, damping and inertia of the device through the mechanical design, by introducing certain mechanisms to achieve this goal. In this approach, impedance modulation occurs due to mechanical changes in the behaviour of the mechanism. Energy input is only required during modulation and the change of the impedance is physical and pertinent for the whole frequency spectrum.

Variable stiffness devices are commonly employed physical interactions due to their many advantages, such as improved stability, performance, and energy efficiency [62, 63]. Variable stiffness systems provide powerful advancements in vibration isolation applications, for instance, vehicle suspensions [64] and seismic protection systems [65].

Two categories of passive devices have received great attention in the implementation of variable impedance devices: negative stiffness [66] and inerter [67] mechanisms as vibration control mechanisms. These devices have been shown to be effective tools in vibration isolation. The concept of parallel connections between positive and negative stiffness elements has been proposed to provide a low static stiffness and prevent large static deflection [68]. Both of these devices may be included in the design of a variable impedance mechanism and can be employed for vibration isolation.

Chapter 3

Mathematical Modeling

A passive mechanical device is proposed to adjust the impedance characteristics of the suspended-load backpack. The passive design is implemented as a one degree of freedom device with a variable stiffness module, an inerter, and a tension spring.

3.1 Dynamic model of walking with load

In this section, we present the dynamic model of walking with suspended load and the performance metrics used to evaluate suspended-load mechanism designs.

The human pelvis moves up and down periodically during the walking, as the body is moved forward. One can only consider the torso movements in the vertical direction for the analysis of the suspended-load pack, since the backpack is assumed to oscillate only along the vertical direction. A single degree of freedom backpack model with a passive suspension is presented in Figure 3.1, while the variables used in the analytical models are presented in Table 3.1.

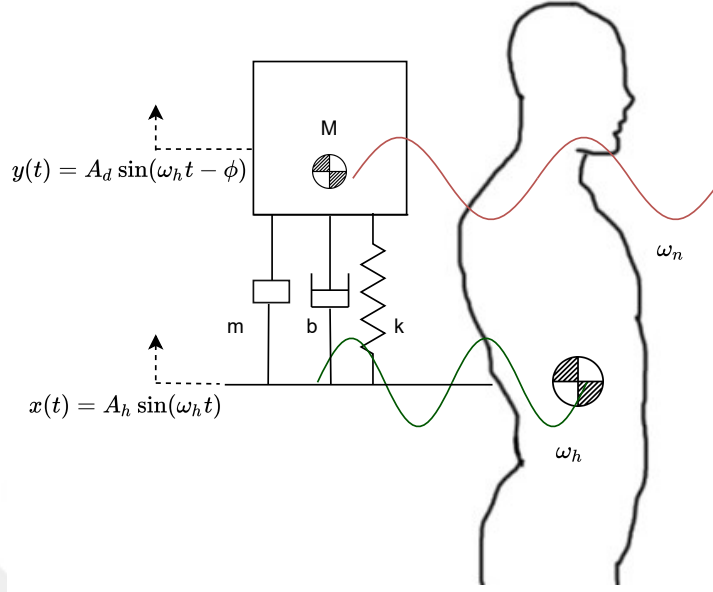


FIGURE 3.1: A single degree of freedom backpack model with a passive suspension

The vertical movements of human torso can be modelled as in Eqn. 3.1, as a sinusoidal profile with an amplitude A_h , radial frequency ω_h , and time t [69].

$$x(t) = A_h \sin(\omega_h t) \quad (3.1)$$

The leg length and lower limb angles can be used to calculate the distance travelled during walking. Grieve et al. have provided an approximate model for the walking frequency ω_h (rad/s) as a function of walking speed v (m/s) and human height S (m) [70] as

$$\omega_h = \frac{4\pi \cdot 64.8 \left(\frac{v}{S}\right)^{0.57}}{60} \text{ [rad/s]} \quad (3.2)$$

where $l_0 = 0.53S$ denotes the approximate leg length of a human with height of S . Figure 3.2 presents human walking frequency varying human height and walking speeds as a colorbar plot.

Variable	Description	Unit
M	Mass of the backpack	kg
m	Inertance of the inerter	kg
b	Damping of the backpack	Ns/m
k	Stiffness of the backpack	N/m
ω_h	Human walking frequency	Hz
ω_n	Passive suspension natural frequency	Hz
S	Height of the human	m
l_0	Leg length of the human	m
A_h	Amplitude of the vertical movement of the human body	m
A_d	Amplitude of the vertical movement of the backpack mass	m
v	Human walking speed	m/s
g	Gravitational acceleration	m/s ²

TABLE 3.1: Description of variables and parameters used in the mathematical model

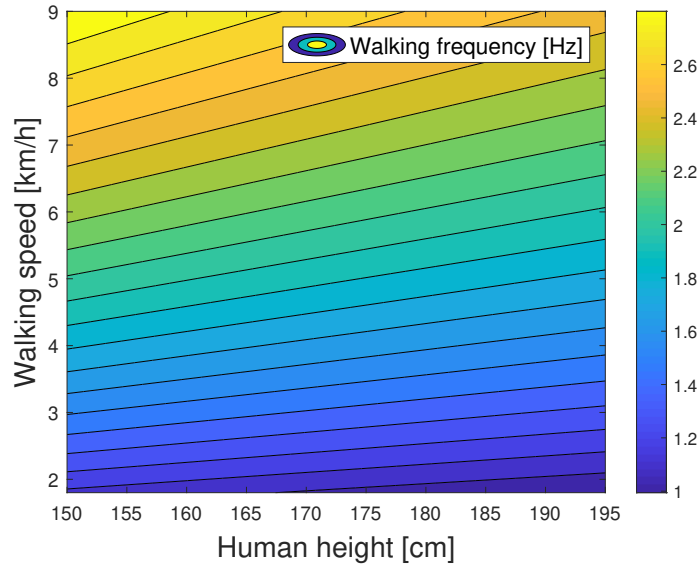


FIGURE 3.2: Walking frequency according to Eqn. 3.2 for varying human heights and walking speeds

An inverted pendulum model can be used to model the amplitude of oscillation of the torso by involving corrections for pelvis rotation, pelvis tilt, knee flexion, ankle position, foot length, and a narrowed walking base, as in Eqn. 3.3 [71].

$$A_h = \frac{l_0}{2} \left(1 - \sqrt{1 - \left(\frac{0.963v}{l_0 \cdot 2 \cdot 1.504 \left(\frac{v}{l_0} \right)^{0.57}} \right)^2} \right) - 0.0157 l_0 \quad (3.3)$$

It is worth to mention that the Eqn. 3.3 does not model the effects of the coupling between the backpack and the torso. It assumes only the walking frequency and human height influence the amplitude of oscillation of the torso.

3.1.1 Inerter based suspended-load backpack

A model of the proposed suspension design is presented in Figure 3.1. The load carried by human is isolated from the torso by a suspension consisting of a spring, a damper, and an inerter connected in parallel to each other. The vertical motion expressed in the Eqn. 3.1 is exerted to the system as shown in the Figure 3.1. These sinusoidal oscillations excite sinusoidal oscillation of the vertical position of the backpack [72] The motion of the backpack under these excitations can be expressed as:

$$y(t) = A_d \sin(\omega_h t - \phi) \quad (3.4)$$

where A_d denotes the amplitude of oscillation, and ϕ is the relative phase shift of the movement between the backpack and the human torso. For a generalized layout, we can write the general equation of motion of suspended backpack with mechanical filter impedance $Q(s)$ as:

$$Ms^2 Y(s) + sQ(s)(Y(s) - X(s)) = 0. \quad (3.5)$$

For this general suspension layout, the transfer function that captures the system response can be expressed as:

$$G(s) = \frac{X(s)}{Y(s)} = \frac{Ms^2 + sQ(s)}{sQ(s)}. \quad (3.6)$$

Our proposed suspension design features a mechanical impedance of the following form: $Q(s) = k/s + b + ms$. Solving for the steady state response of Eqn. 3.5, one can derive the amplitude ratio

$$\left| \frac{X}{Y} \right| = \sqrt{\frac{b^2 \omega_h^2 - (k - \omega_h^2 (M + m))^2}{(k - m \omega_h^2)^2 + b^2 \omega_h^2}} \quad (3.7)$$

as well as the phase shift between human torso and load:

$$\phi = \text{atan} \left(\frac{M b \omega_h^3}{(k - m \omega_h^2) (k - (M + m) \omega_h^2) - b^2 \omega_h^2} \right) \quad (3.8)$$

The undamped natural frequency ω_n of the backpack system is

$$\omega_n = \sqrt{\frac{k}{m + M}}. \quad (3.9)$$

The total force of the pack in the vertical direction acting on the human is equal to the total amount of the gravitational force acting on the load combined with the harness weight and the force driving the load.

$$F_{\text{backpack}} = (M + M_{\text{suit}})g - F_{\text{osc}} \sin(\omega_h t - \phi) \quad (3.10)$$

where F_{osc} denotes the amplitude of the oscillation force acting on the human from the backpack.

$$F_{osc} = M\omega^2 A_h \left| \frac{Y}{X} \right| \quad (3.11)$$

$$F_{osc} = M\omega_h^2 A_h \sqrt{\frac{(k - m\omega_h^2)^2 + b^2\omega_h^2}{b^2\omega_h^2 - (k - \omega_h^2(M + m))^2}} \quad (3.12)$$

The amplitude ratio in Eqn. 3.7 can be used to calculate the oscillation force, which is a function of mass of the backpack, human weight, walking velocity, and stiffness, damping, and inertance of the suspension.

The aim of the suspension design is to decrease the oscillating force, especially the peak force that acts on the human torso, as these forces can induce injuries, cause muscle tiredness, and decrease energy efficiency of load carriage [14]. Backpack performance can be increased by decreasing the oscillation force, while, the main goal is to decrease $F_{backpack}$, as this is the total force acting on the human torso. Hence, proper phasing of F_{osc} may help reduce peak values $F_{backpack}$ and improve energy efficiency of load carriage by shifting the occurrence peak forces to a bio-mechanically more favorable phase of the gait cycle.

Lower $F_{backpack}$ can help user to walk for longer ranges, carry more load, and consume less energy with respect to traditional backpack designs. For an undamped model, F_{osc} can rise to infinity, when the backpack stiffness is equal to the total mass of load and inertance times the walking frequency. This value denotes undamped natural frequency of the system.

$$k_{resonance} = (M + m)\omega_h^2 \quad (3.13)$$

For the stiffness values that are lower than $k_{\text{resonance}}$, the oscillation force can be reduced by increasing the damping coefficient. Higher damping coefficient provides a greater reduction in the oscillating force for the stiffness values near the resonance stiffness. Above the resonance stiffness, F_{osc} converges to a non-zero value of

$$\lim_{k \rightarrow \infty} F_{\text{osc}} = M \omega_h^2 A_h \quad (3.14)$$

For a specific stiffness value of k , called the performance stiffness, the oscillation force remains the same for any value of damping b . For stiffness values lower than the performance stiffness, the corresponding oscillating force is lower than the case for higher stiffness values. Backpack stiffness values below the performance stiffness lead to the minimum oscillating load (with lower stiffness producing lower oscillating loads) and a reduction in the damping coefficient leads to a reduction in the oscillating load for the suspended-load pack.

$$k_{\text{performance}} = \frac{1}{2} M \omega_h^2 + m \omega_h^2 \quad (3.15)$$

Proof of Eqn. 3.15: Consider F_{osc} as a function of k and b , $F_{\text{osc}}(k, b)$. Then, the first derivative of F_{osc} with respect to b , can be solved for k in order to find the performance stiffness. The solutions of the following equation gives the $k_{\text{performance}}$.

$$\frac{\partial}{\partial b} F_{\text{osc}}(k, b) = 0 \quad (3.16)$$

The oscillation force has its maximum and minimum values at the stiffness values of $k_{\text{min,max}} = k_{\text{performance}} \pm \Delta K$, where

$$\Delta K = \frac{\omega_h \sqrt{M^2 \omega_h^2 + 4 b^2}}{2} \quad (3.17)$$

Proof of Eqn. 3.17: Consider F_{osc} as a function of k as $F_{\text{osc}}(k)$. Then, the first derivative of F_{osc} can be solved in order to find the local maximum and minimum. The solutions of the following equation gives the $k_{\min}$ and $k_{\max}$.

$$\frac{d}{dk} F_{\text{osc}}(k) = 0 \quad (3.18)$$

The amplitude of the maximum and minimum of F_{osc} is not dependent on the inertance. Both are function of mass, damping, human walking frequency and amplitude.

$$F_{\text{osc}} \Big|_{k=k_{\min}} = M A_h \omega_h^2 \sqrt{\frac{4b^2 + M^2 \omega_h^2 - M \omega_h \sqrt{M^2 \omega_h^2 + 4b^2}}{4b^2 + M^2 \omega_h^2 + M \omega_h \sqrt{M^2 \omega_h^2 + 4b^2}}} \quad (3.19)$$

$$(3.20)$$

$$F_{\text{osc}} \Big|_{k=k_{\max}} = M A_h \omega_h^2 \sqrt{\frac{4b^2 + M^2 \omega_h^2 + M \omega_h \sqrt{M^2 \omega_h^2 + 4b^2}}{4b^2 + M^2 \omega_h^2 - M \omega_h \sqrt{M^2 \omega_h^2 + 4b^2}}} \quad (3.21)$$

Then, the oscillation force at the performance stiffness can be calculated by substitution as:

$$F_{\text{osc}} \Big|_{k=k_{\text{performance}}} = M \omega_h^2 A_h \quad (3.22)$$

For very low stiffness values, the oscillation force will be:

$$\lim_{k \rightarrow 0} F_{\text{osc}} = M \omega_h^2 A_h \sqrt{\frac{b^2 + m^2 \omega_h^2}{b^2 + (M + m)^2 \omega_h^2}} \quad (3.23)$$

The maximum phase shift occurs at $k = k_{\text{performance}}$ and can be expressed as

$$\phi_{max} = \lim_{k \rightarrow k_{\text{performance}}} \phi = \text{atan}\left(\frac{4 M b \omega_h}{4 b^2 - M^2 \omega_h^2}\right) \quad (3.24)$$

and the minimum and maximum force amplitude occurs at $k = k_{min}$ and $k = k_{max}$ with the phase shift value of:

$$\phi_{min} = \lim_{k \rightarrow k_{min}} \phi = \lim_{k \rightarrow k_{max}} \phi = \text{atan}\left(\frac{M \omega_h}{2 b}\right) \quad (3.25)$$

The relative displacement between the load and human is another important performance metric. This value can be also examined as a criteria in the design of a suspended-load backpack. The relative displacement can be computed as

$$|x - y| = \sqrt{X^2 - 2XY \cos(\phi) + Y^2} \quad (3.26)$$

Suspended-load backpacks perform better with decreased stiffness. However, there exists excessive deflection δ on the springs with very low stiffness values due to gravitational force acting on the load:

$$\delta = \frac{Mg}{k} \quad (3.27)$$

In Figure 3.3, this deflection for the varying backpack payloads is shown for a range of spring stiffness values. Employing a high compliance spring corresponds to high static deflection and very large workspace requirements for the suspension to function properly.

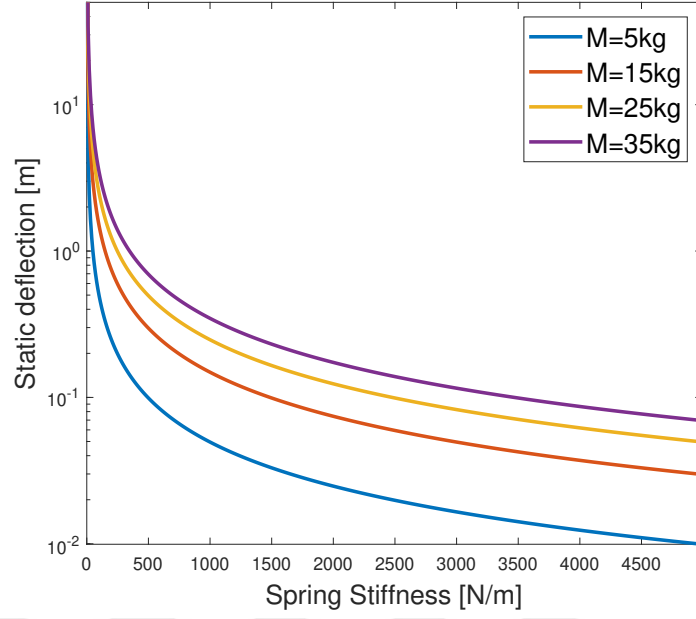


FIGURE 3.3: Additional deflection due to the load

Figure 3.4 presents the oscillation load amplitude and phase shift profiles for varying values of damping b , for an inerter-based suspended load backpack. One can observe from Figure 3.4 that employment of the inerter in the suspended load backpack allows a wide range of stiffness values to be employed while taking advantage of desired filtering characteristics. Adding an inerter to the passive compliance unit, the design can achieve the minimum oscillation amplitude at a positive value of stiffness. Correspondingly, the deflections stay in much smaller ranges, improving feasibility of designs.

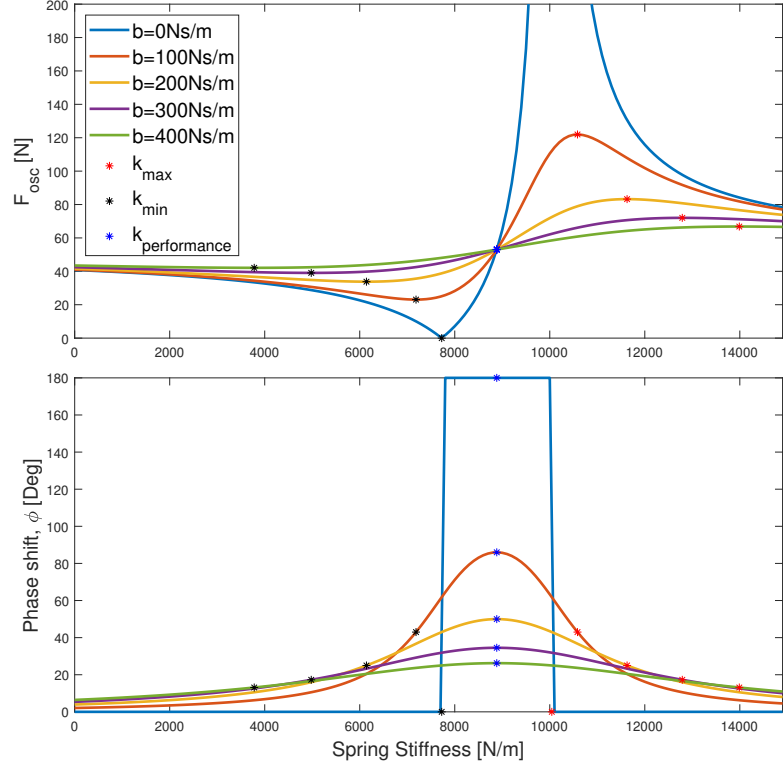


FIGURE 3.4: Oscillation force amplitude and phase shift profile for proposed backpack on a user with a 1.75 m height, 15 kg backpack mass, 1.5 m/s walking speed, for varying spring stiffness and damping values.

The force and phase shift profiles for different values of inertance are represented in Figure 3.5. Since the inerter changes the resonance frequency of the system, the response is also shifted in proportion to the inertance of the system. This relaxation by the inerter allows employment of significantly more stiff springs in the design, resulting in lower deflections under static and dynamic forces.

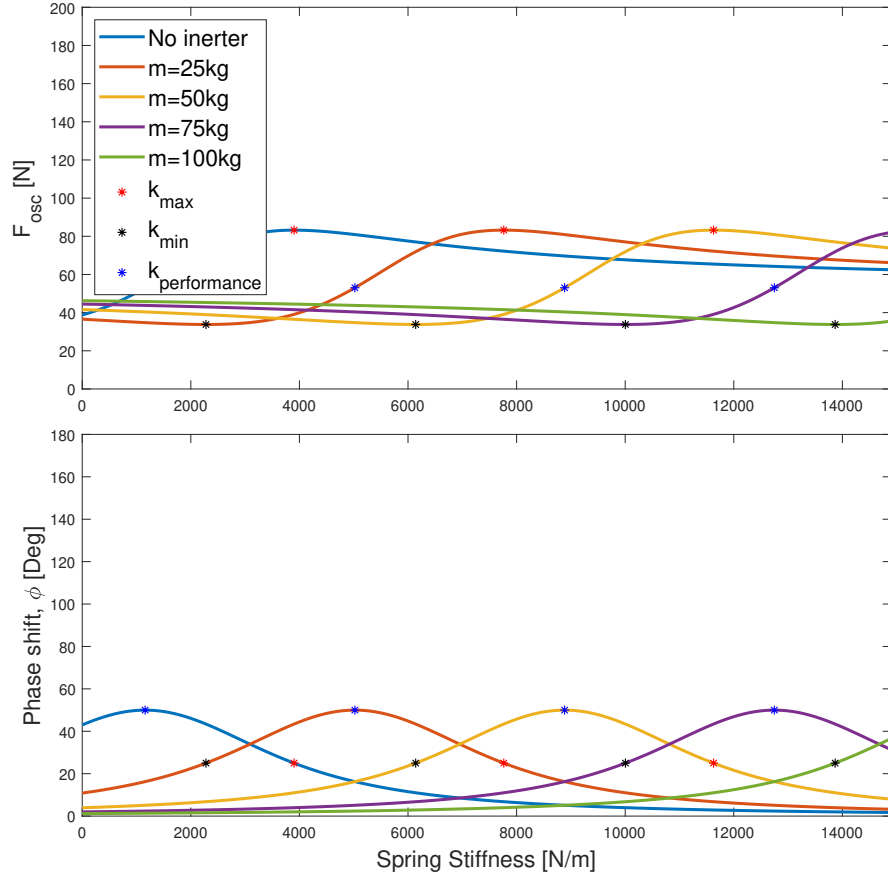


FIGURE 3.5: Oscillation force amplitude and phase shift profile for proposed backpack on a user with a 1.75 m height, 15 kg backpack mass, 200 Ns/m damping, 1.5 m/s walking speed, for varying spring stiffness and intertance values.

In Figure 3.6, the amplitude of oscillation and the phase shift are presented for various human walking speeds. For high walking speeds, the performance stiffness is increased.

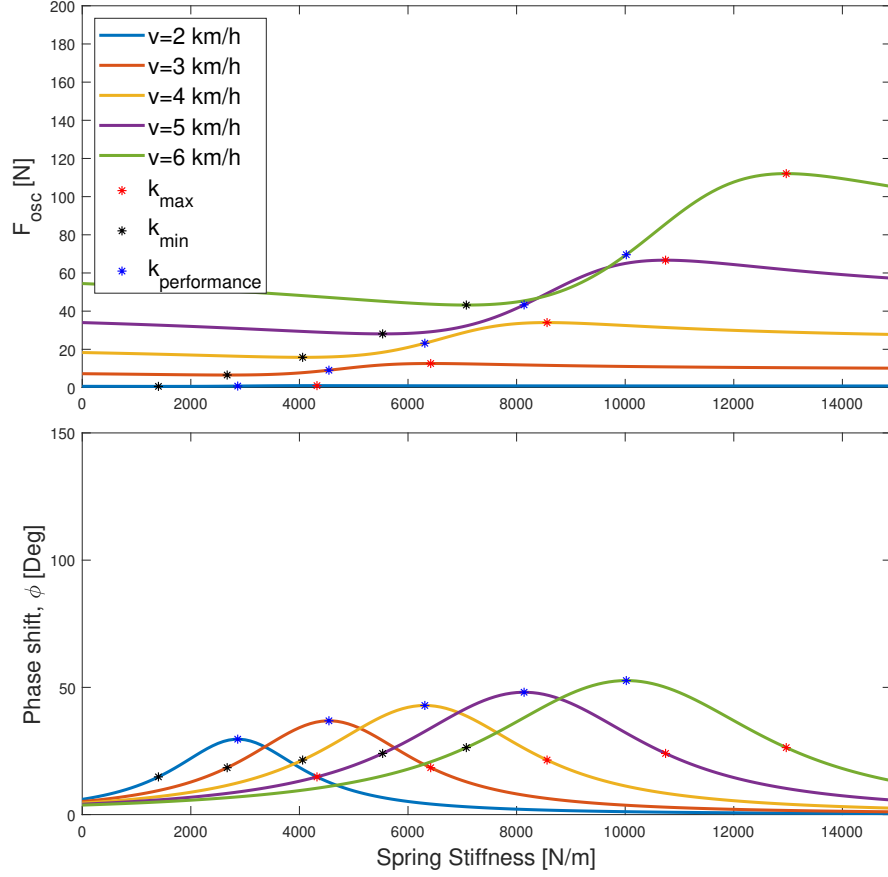


FIGURE 3.6: Oscillation load amplitude and phase shift profile for proposed backpack on a user with a 1.75 m height, 15 kg backpack mass, 50 kg inertance, 200 Ns/m damping, for varying walking speed and spring stiffness values

Special case: Suspended-load backpack

The typical suspended-load backpacks use springs-dampers arranged in parallel or only springs to filter the load dynamics. This serves as a special case for inerter-based suspended load backpack, where inerter is de-activated. For this configuration, the minimum oscillation occurs at very low stiffness values. Using the Eqns. 3.15 and 3.17, one can find k_{min} as

$$k_{min}\Big|_{m=0} = \frac{1}{2}M\omega_h^2 - \frac{\omega_h\sqrt{M^2\omega_h^2 + 4b^2}}{2} \quad (3.28)$$

$$(3.29)$$

$$k_{min}\Big|_{m=0} = \frac{1}{2}M\omega_h^2 + \frac{\omega_h\sqrt{M^2\omega_h^2 + 4b^2}}{2} \quad (3.30)$$

$$(3.31)$$

Note that for this case, k_{min} is negative. Since, the negative stiffness values are not feasible for physical implementation, this value is set to zero. Similarly, k_{max} can be computed as

$$k_{min}\Big|_{m=0} = \frac{1}{2}M\omega_h^2 + \frac{\omega_h\sqrt{M^2\omega_h^2 + 4b^2}}{2} \quad (3.31)$$

Human walking frequency and backpack load are the dominant factors affecting the amplitude of oscillations. Recall that the maximum and minimum values of F_{osc} are not affected by the inertance, thus the formula are still valid. The walking speed and human height define the walking frequency.

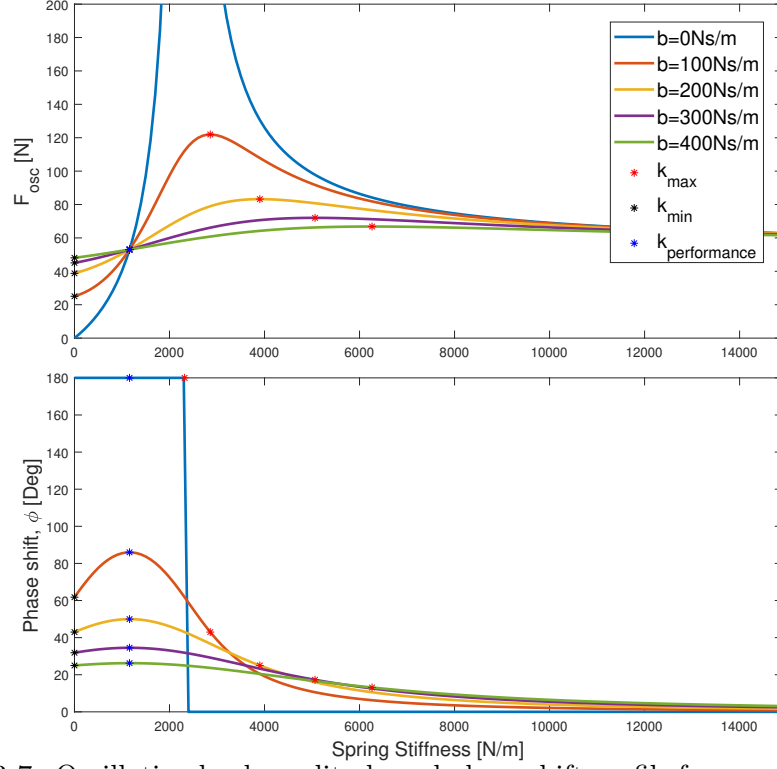


FIGURE 3.7: Oscillation load amplitude and phase shift profile for suspended load backpack on human with weight 1.75m, walking velocity 1.5 m/s, backpack mass 15 kg, for varying spring stiffness and damping

In Figure 3.7, the oscillation force profile and the phase shift are shown for a user with a 1.75 m height, walking at a speed of 1.5 km/h with a backpack mass of 15 kg. It should be noted that above the performance stiffness, higher values of damping corresponds to lower oscillation force. However, at relatively low stiffness values, high values of damping is not beneficial in terms of the amplitude of the oscillation force. The minimum force occurs at zero stiffness. Due to limited deflection range of physical implementations, a zero stiffness is not practical under large dynamic forces. A suspended-load backpack promises an efficient method to reduce the oscillation force for low stiffness and damping values. If no damping is employed in the system, the device is out-of-phase for the stiffness values lower than k_{max} at which the oscillation force is maximized.

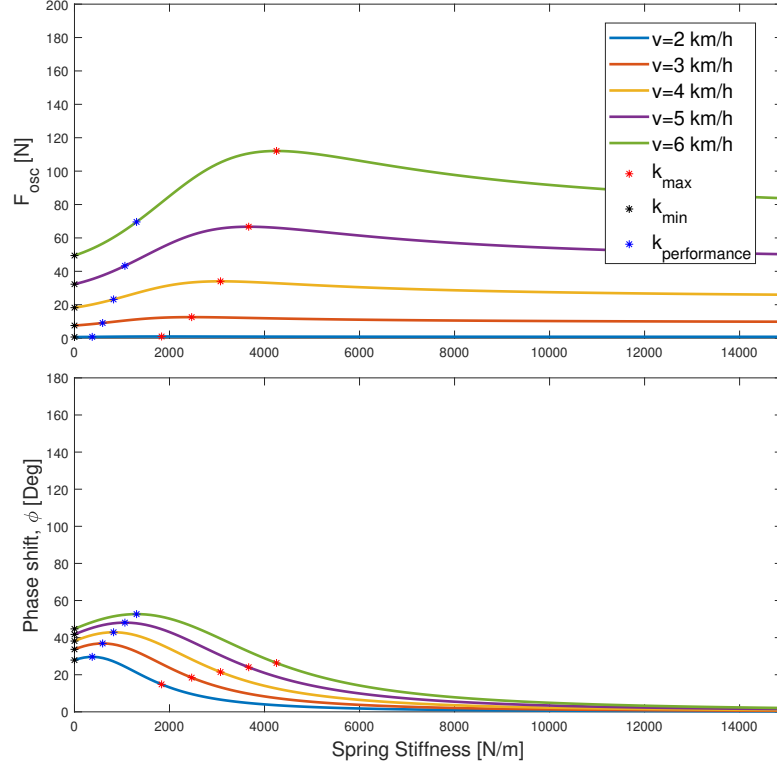


FIGURE 3.8: Oscillation load amplitude and phase shift profile for suspended load backpack on a user with a 1.75 m height, 15 kg backpack mass, 200 Ns/m damping, for varying waking speeds and spring stiffness values

In Figure 3.8, the amplitude of oscillation and the phase shift are represented for various human walking speeds. For high walking speeds, the performance stiffness is increased. Hence, suspended-load backpacks perform better with decreased stiffness. Phase shift is maximized at the $k_{performance}$. Another performance characterization is based on oscillation load over a range of backpack mass and walking speeds. The comparative analysis of spring-damper and spring-damper-inerter layouts in terms of oscillation force amplitude is presented in Figure 3.9.

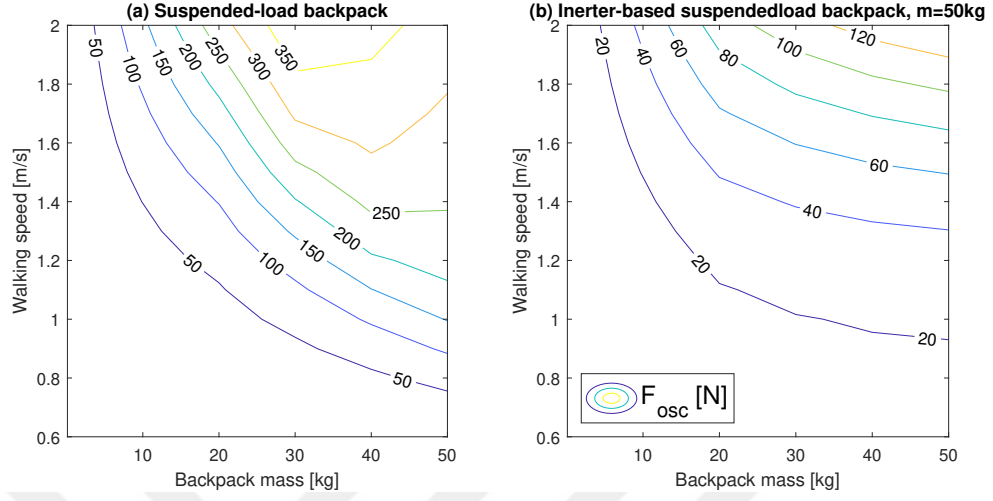


FIGURE 3.9: Oscillation force contour plot for the proposed backpack suspension on user with an height of 1.75m, for backpack mass 15 kg, stiffness 5000 N/m, damping 200 Ns/m, for varying walking speeds and pack loads

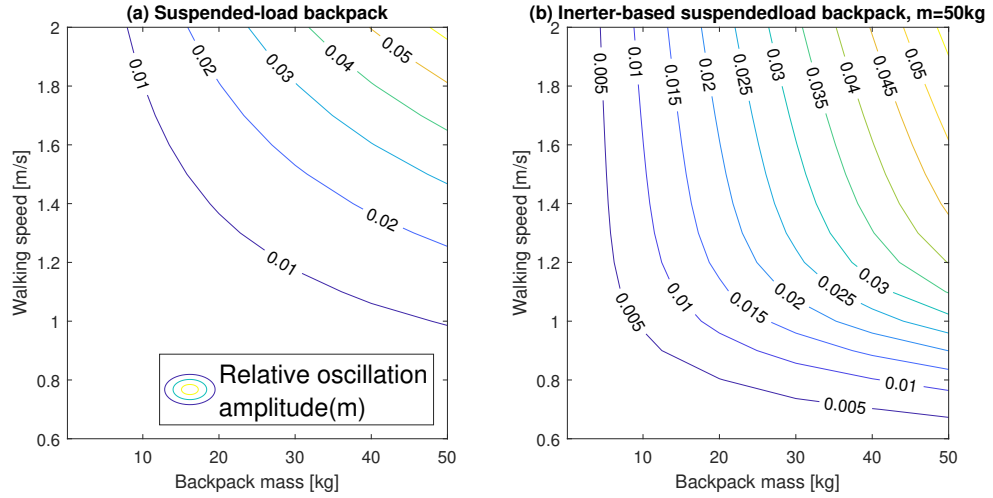


FIGURE 3.10: Relative oscillation amplitude contour plot for the proposed backpack suspension on user with an height of 1.75m, for backpack mass 15 kg, stiffness 5000 N/m, damping 200 Ns/m, for varying walking speeds and pack loads

3.2 Transmissibility

To study the system's response, we also investigate the transmissibility characteristic of the system. Force transmissibility is defined as the ratio of the measured vibrational force to the vibrational force that excites the system. Consider the model in Figure 3.11, where $x(t)$ and $y(t)$ denote time-domain representations of the displacement of the human torso and the load, respectively. A force, F_t is transmitted to the human due to the reactions from the spring k , damping b , and inerter m .

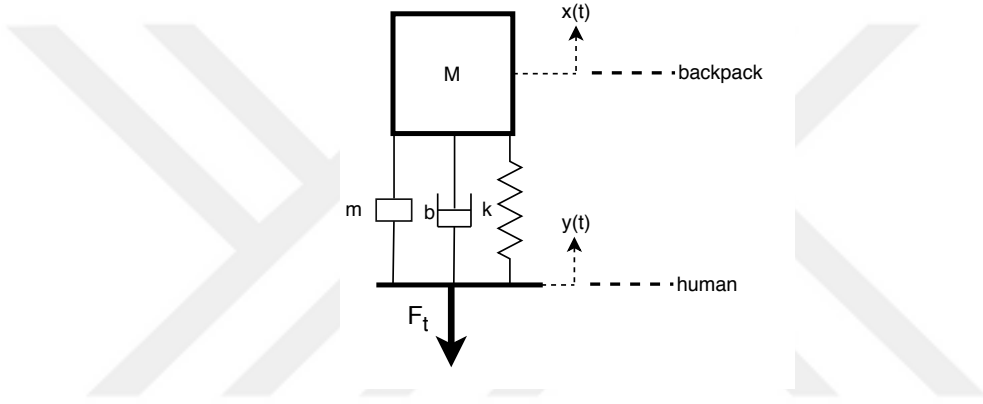


FIGURE 3.11: Transfer of motion through the proposed backpack device

3.2.1 Displacement transmissibility

The displacement transmissibility T_d can be expressed as

$$T_d(s) = \frac{X(s)}{Y(s)} = \frac{m s^2 + b s + k}{(m + M) s^2 + b s + k} \quad (3.32)$$

The equation above can be re-written as

$$T_d(s) = \frac{\frac{s^2}{\omega_0^2} - \frac{M s^2}{K} + 2 \zeta \frac{s}{\omega_0} + 1}{\frac{s^2}{\omega_0^2} + 2 \zeta \frac{s}{\omega_0} + 1} \quad (3.33)$$

where ζ is the damping coefficient given by

$$\zeta = \frac{b}{2\sqrt{(M+m)k}} \quad (3.34)$$

Furthermore, we can simplify the transmissibility given in the Eqn. 3.33 as follows.

$$T_d(s) = 1 - \frac{M}{K} \frac{s^2}{\frac{s^2}{\omega_0^2} + 2\zeta \frac{s}{\omega_0} + 1} \quad (3.35)$$

In Figure 3.12, the change in displacement transmissibility for various values of the damping coefficient b is presented. At a certain frequency, the displacement transmissibility of the system crosses into negative values, independent of the damping coefficient. This frequency is observed between the system's natural frequency $\frac{1}{2\pi} \sqrt{k/(m+M)}$ and $\frac{1}{2\pi} \sqrt{k/(m)}$. At frequencies greater than this mentioned frequency, the system begins to isolate vibrations.

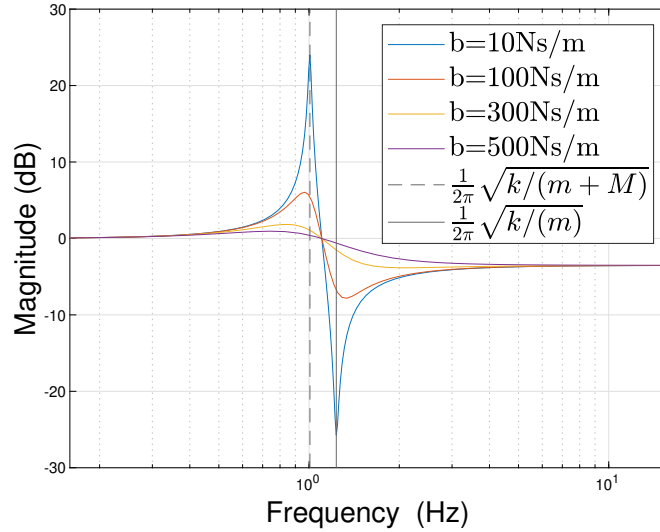


FIGURE 3.12: System's displacement transmissibility curve for varying values of damping coefficient b and $m= 50$ kg, $M= 25$ kg and $k= 3000$ N/m

For the frequencies which correspond to positive magnitudes, the system is in the region of amplification. One can observe that a lower amount of damping corresponds to a higher amplification peak. Higher values of damping coefficient smoothen the transition between the amplification and isolation phases. Figure 3.13 presents the transmissibility of the system for a constant value of the damping coefficient b and stiffness k , for varying values of inertance. As greater values of inertance is introduced to the system, the natural frequency is lowered. Thus, the frequency at which the system starts to isolate vibrations is lowered. Besides an increased range of vibration isolation frequency, a higher amount of inertance corresponds to more robust vibration isolation. It reduces the peak value of amplification so that the relative displacement between the human torso and the carried load decreases.

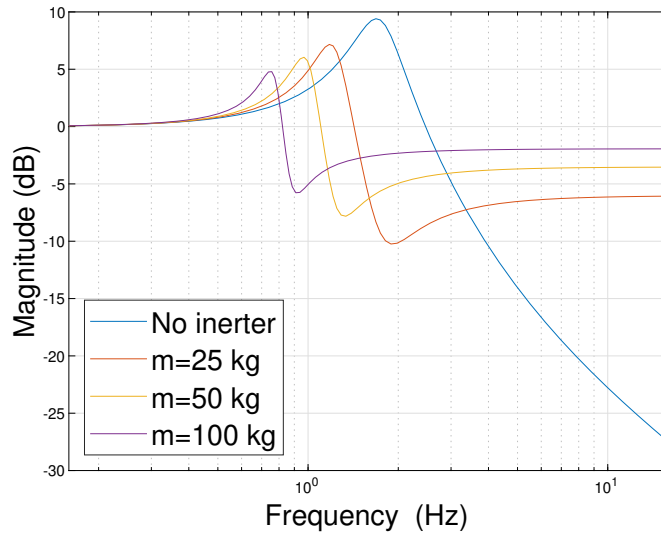


FIGURE 3.13: System's displacement transmissibility curve for varying values of inertance factor m and $M=25$ kg, $k= 3000\text{N/m}$, $b= 100\text{Ns/m}$

In Figure 3.14, the effect of varying stiffness value on the displacement transmissibility is presented. The inertance and damping coefficients are set to constant values. Note that as the stiffness of the system is lowered, the natural frequency of the system is also reduced. More stiff suspended-load backpacks results in a larger

frequency range of amplification. For the low values of stiffness, system crosses to the vibration isolation phase at low frequencies. The peak magnitude during amplification decreases for lower stiffness suspended-load backpacks.

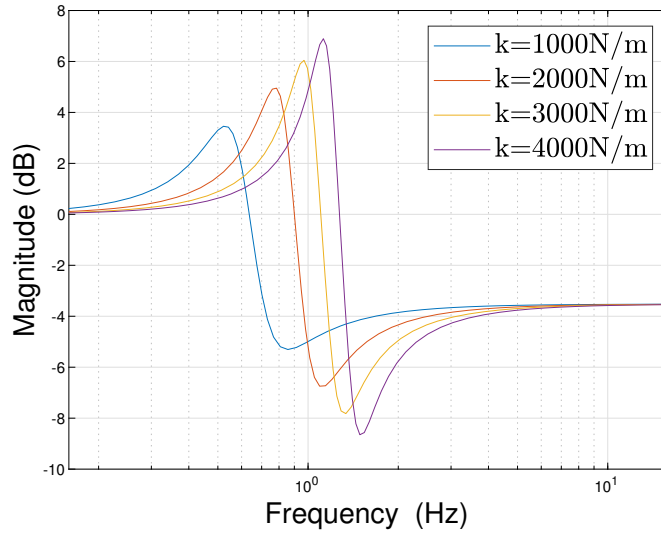


FIGURE 3.14: System's displacement transmissibility curve for varying values of stiffness k and $m= 50$ kg, $M= 25$ kg, $b= 100\text{Ns/m}$

The displacement transmissibility curve for varying stiffness and inertance values are presented in Figure 3.15. Stiffness values of 1000, 2000, 3000 and 4000 N/m are employed with low inertance level of 25 kg and a relatively high inertance level of 100 kg. At relatively low stiffness and high inertance values, the displacement transmissibility curve has its peak at low frequencies. For the configurations with low stiffness and high inertances, system begins to isolate vibrations at relatively low frequencies.

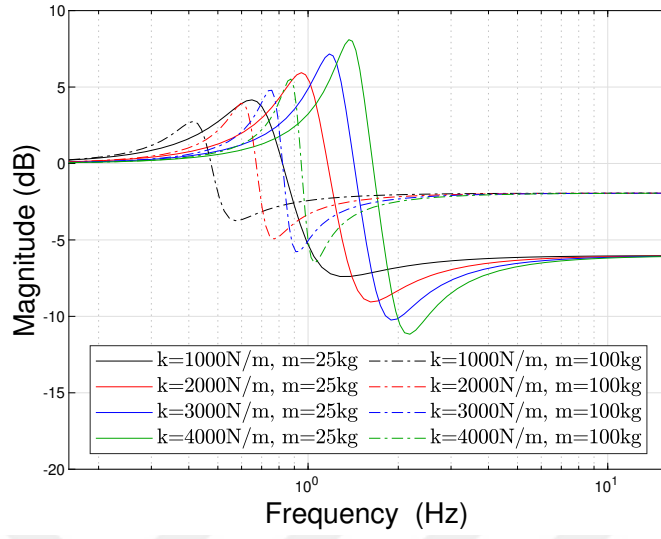


FIGURE 3.15: System's displacement transmissibility curve for varying values of stiffness k and inertia m for $M= 15$ kg and $b= 100\text{Ns/m}$

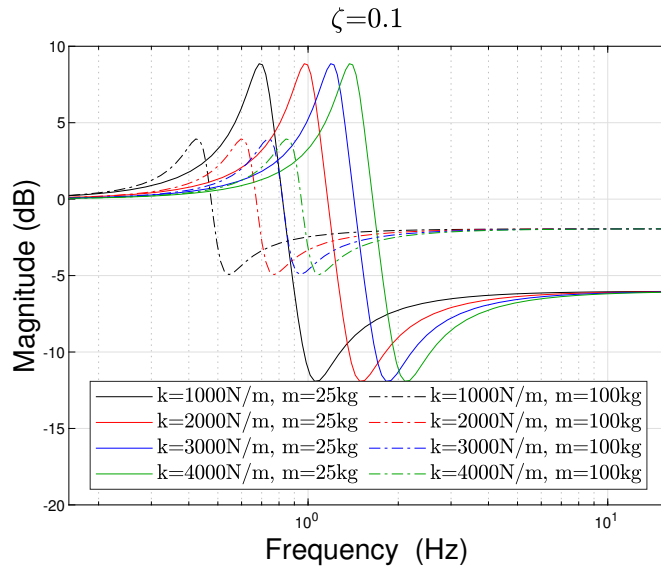


FIGURE 3.16: System's displacement transmissibility curve for varying values of stiffness k and inertia m for $M= 15$ kg

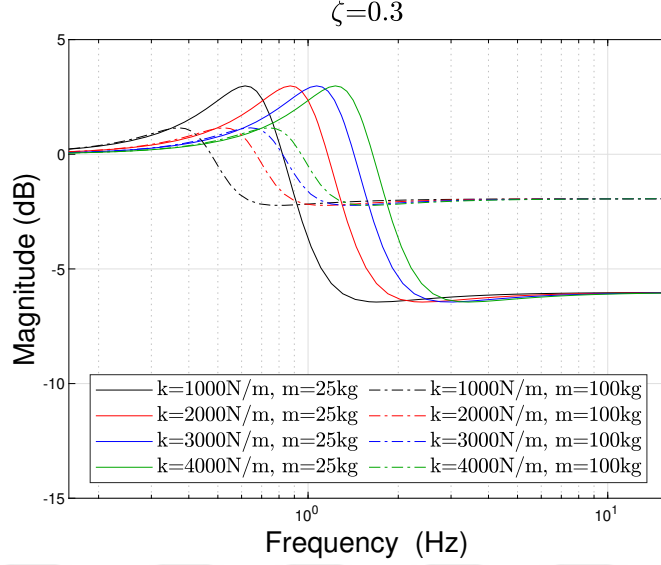


FIGURE 3.17: System's displacement transmissibility curve for varying values of stiffness k and inertance m for $M= 15$ kg

3.2.2 Force transmissibility

Due to the relative movement between the load and the human torso, a force F_t is exerted to the human body. The force transmissibility can be expressed as a function of human walking amplitude and transmitted force due to the reactions from the spring, damper and inerter, by using the following transfer function

$$T_f(s) = \frac{F_t(s)}{Y(s)} = M s^2 \frac{m s^2 + b s + k}{(m + M) s^2 + b s + k} = M s^2 \frac{X(s)}{Y(s)} \quad (3.36)$$

The variation of the force transmitted to the torso with walking frequency is presented in Figure 3.18 for different values of damping coefficient b .

Similar to the displacement transmissibility, at a certain frequency, the force transmissibility of the system does not depend on the damping coefficient b . This frequency appears between the system's natural frequency $\frac{1}{2\pi} \sqrt{k/(m + M)}$ and

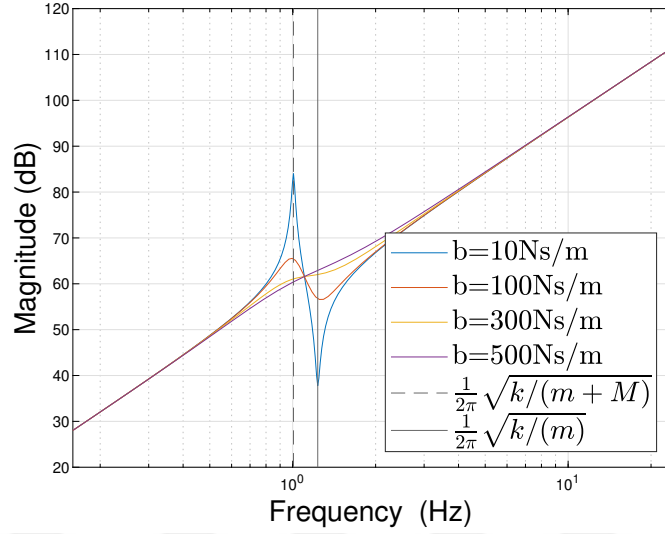


FIGURE 3.18: System's force transmissibility curve for varying values of damping coefficient b and $m= 50$ kg, $M= 25$ kg and $k= 3000$ N/m

$\frac{1}{2\pi} \sqrt{k/(m)}$. Regardless of the damping coefficient a reduction in the force transmissibility is observed for walking frequencies higher than the system's natural frequency. Higher values of damping smooths the transition between the phases, but does not corresponds to lower force transmissibility for the all frequencies.

Force transmissibility of the system for different values of inertance is shown in Figure 3.19. As the inertance increases, the natural frequency of the system is reduced so that the transmitted force is lowered at relatively low frequencies.

However, it can be noticed that more stiff systems transmit higher forces, since their natural frequency is higher than the systems with low stiffness as it can be seen in Figure 3.20.

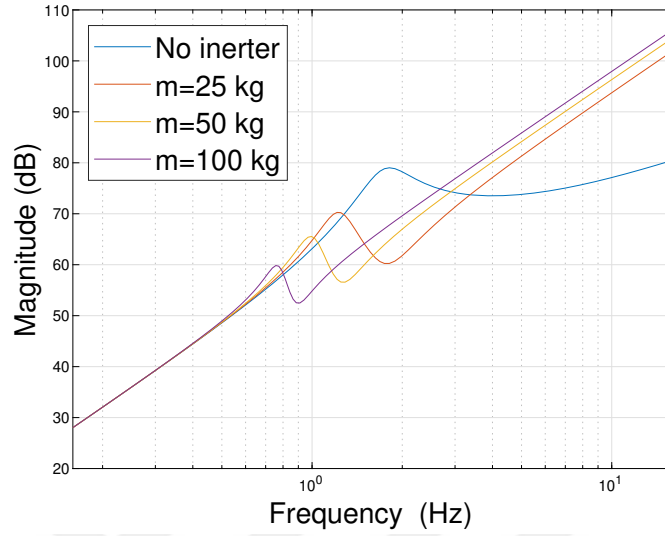


FIGURE 3.19: System's force transmissibility curve for varying values of inertance factor m and $M=25$ kg, $k=3000\text{N/m}$, $b=100\text{Ns/m}$

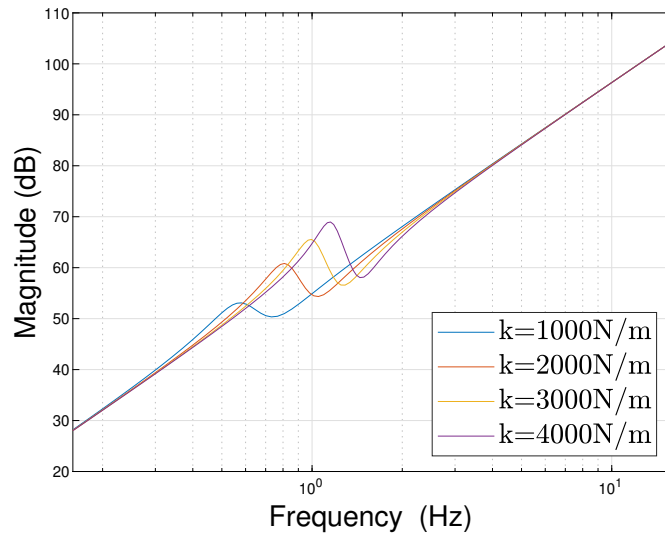


FIGURE 3.20: System's force transmissibility curve for varying values of stiffness k and $m=50$ kg, $M=25$ kg, $b=100\text{Ns/m}$

Chapter 4

Design of Inerter-based Suspended Load Backpack

In this chapter, the mechanical design of the inerter-based suspended load backpack is presented along with its analytical model and numerical evaluations. The proposed design consists of a stiffness module and an inerter module. The stiffness module features a parallel connection of variable stiffness module with a tension spring. The inerter module is attached to the stiffness module in parallel.

4.1 Overall design

The proposed system consists of an inerter module, a variable negative stiffness module, and a tension spring, as shown in the Figure 4.1. It is worth to mention that the bearings used in the implementation of the inerter module also introduce some damping characteristics to the device.

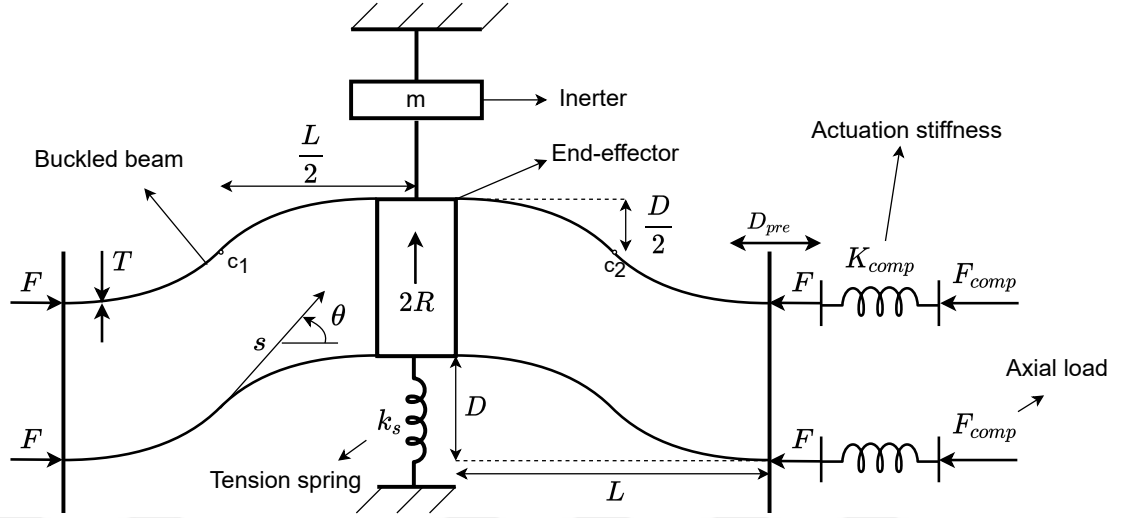


FIGURE 4.1: Schematic representation of the overall design that consists of as a parallel connection of inerter with a variable stiffness module.

A tension spring is used in the design with a stiffness of k_s , and the variable stiffness module modulates effective stiffness k of the system by changing its stiffness characteristics k_n based on the axial loading of the leaf springs. The inerter has an inertance factor of m , and the cumulative damping of the system is modeled with a coefficient of b . The following admittance transfer function captures the steady state frequency response of the system in the frequency domain, relating the force exerted to the end-effector to the end-effector position.

$$Y(s) = \frac{X(s)}{F(s)} = \frac{1}{m s^2 + b s + k} \quad (4.1)$$

where X and F represents position of end-effector and force exerted to the end-effector, respectively.

4.2 Variable stiffness module

A variable stiffness module is employed in the device to modulate its effective stiffness. In this section, we provide an analytical model and numerical evaluations of

the variable stiffness module. The stiffness modulation is based on the transverse stiffness of buckling beams under axial loading.

4.2.1 Analytical model

The analysis in this section closely follows the ones presented in [73, 74].

In Figure 4.2, a schematic model of axially loaded buckling beams are presented. Clamped-clamped boundary conditions for the beams are considered in the figure, while it is possible to handle analogous analysis for other boundary conditions. Furthermore, the deflection pattern is valid under small deflections and when the axial loads are kept below the second critical buckling load.

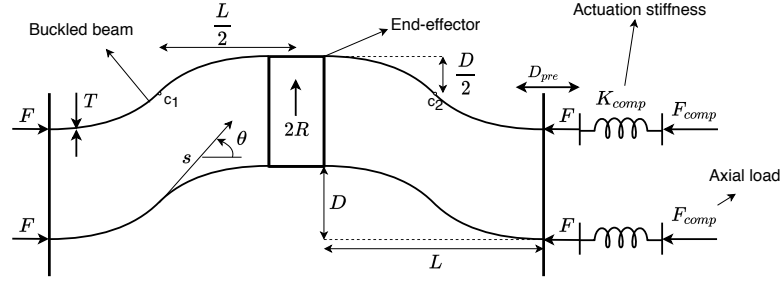


FIGURE 4.2: Schematic model of axially loaded buckling beam.

In Figure 4.2, L denotes the half of the axial length of the beam, and D stands for transverse deflection. Axial loading is represented with F_{comp} , and transverse loading is represented with $2R$. The variable s represents the path length along the deflected beam. The symbol x represents the axial direction and y denotes the transverse direction. Symbols c_1 and c_2 denote the inflection points. Boundary conditions for the analysis are defined as $y(0) = 0$ and $y(\frac{L}{4}) = \frac{D}{2}$. By using the following curvature

$$\frac{d\theta}{ds} = \frac{1}{\sqrt{1-y'^2}} \cdot y'' \quad (4.2)$$

and the moment caused by the axial and transverse loads F_{comp} and $2R$ on infinitesimal beam between points c_1 and c_2 , the moment-curvature relationship from solid

mechanics can be invoked as

$$\frac{-F_{comp}y - Rx}{EI} = \frac{d\theta}{ds} = \frac{y''}{\sqrt{1 - y'^2}} \quad (4.3)$$

in which E represents the modulus of elasticity and I is the area moment of the beam. Horizontal projection of s onto axis x is given as

$$x(s) = \int_0^s \sqrt{1 - y'^2} ds \quad (4.4)$$

Then, Eqn. 4.3 can be approximated for the small deflections of the beam under $|y'| \ll 1$ as

$$-F_{comp}y - R \int_0^s \left(1 - \frac{y'^2}{2}\right) ds = EIy'' \left(1 + \frac{y'^2}{2}\right) \quad (4.5)$$

At this point, an assumption for characterizing the shape of the beam under axial loading is introduced to the analysis. A perfect sinusoid can be used to approximately model the shape of the buckling beam between points c_1 and c_2 for axial loadings below the second critical buckling load [73]. Sinusoidal mode shape is introduced to the system as follows

$$y(s) = A \sin(ws) \quad (4.6)$$

where $w = 2\pi/L$, $A = D/2$, and $D_1 = D/2$. If we introduce Eqn. 4.6 and its derivatives into Eqn. 4.5, then error function $R(D_1, s)$ can be expressed as:

$$\begin{aligned} R(D_1, s) = & -F_{comp} D_1 \sin(ws) - Rs + 0.25RD_1^2w^2s \\ & + 0.125RD_1^2w \sin(2ws) + EID_1w^2 \sin(ws) \left(1 + \frac{D_1^2w^2}{2} \cos^2(ws)\right) \end{aligned} \quad (4.7)$$

Next, we can apply method of weighted residuals in the Galerkin form as

$$\int_0^{L/4} \sin(ws) R(D_1, s) ds = 0 \quad (4.8)$$

The left side of this equation can be expressed as

$$\begin{aligned} & -\frac{1}{8} D_1 F_{comp} \left(L - \frac{2 \sin(\frac{Lw}{2})}{w} \right) - R \frac{\sin(\frac{Lw}{4}) - \frac{1}{4} Lw \cos(\frac{Lw}{4})}{w^2} \\ & - \frac{1}{4} D_1^2 R w^2 \frac{\sin(\frac{Lw}{4}) - \frac{1}{4} Lw \cos(\frac{Lw}{4})}{w^2} + \frac{1}{8} D_1^2 R w \frac{2 \sin^3(\frac{Lw}{4})}{3w} \\ & + \frac{1}{8} E I D_1 w^2 \left(L - \frac{2 \sin(\frac{Lw}{2})}{w} \right) + \frac{1}{64} E I D_1^3 w^4 \left(L - \frac{\sin(Lw)}{w} \right) = 0 \end{aligned} \quad (4.9)$$

Normalizing Eqn. 4.9 by L^2 and P_{cr} to obtain $\frac{D_1}{L}$ and $\frac{F_{comp}}{P_{cr}}$ instead of D_1 and F_{comp} , in which $P_{cr} = \frac{4\pi^2 EI}{L^2}$ is the first critical axial load for a clamped-clamped end points configuration

$$-\frac{1}{8} \left(\frac{F_{comp}}{P_{cr}} \right) \left(\frac{D_1}{L} \right) - \frac{1}{4\pi^2} \left(\frac{R}{P_{cr}} \right) + \frac{1}{3} \left(\frac{R}{P_{cr}} \right) \left(\frac{D_1}{L} \right)^2 + \frac{1}{8} \left(\frac{D_1}{L} \right) + \frac{1}{16} \pi^2 \left(\frac{D_1}{L} \right)^3 = 0 \quad (4.10)$$

The relation between the transverse displacement D_1 of a beam exposed to an axial loading F_{comp} and the transverse force $2R$ can be expressed as

$$\frac{F_{comp}}{P_{cr}} \left(\frac{D_1}{L} \right) + \frac{R}{P_{cr}} \left[\frac{2}{\pi^2} - \frac{8}{3} \left(\frac{D_1}{L} \right)^2 \right] = \frac{D_1}{L} \left[1 + \frac{\pi^2}{2} \left(\frac{D_1}{L} \right)^2 \right] \quad (4.11)$$

In order to describe transverse stiffness in terms of axial loading, we can express F as

$$F_{comp} = F - K_{comp} (\delta_e + \delta_b) \quad (4.12)$$

where δ_e and δ_b denote the axial deflection of the axially loaded beam due to its elasticity and axial deflection due to buckling of the beam respectively. The effects of residual stresses on the axial force are neglected in the analysis because they do

not significantly affect large scale devices. Axial stiffness of the beam is described as $\frac{AE}{L}$, where A is the cross sectional area, $F_{comp} = \frac{AE\delta_e}{L}$. Then, the axial load applied to the beam can be represented as

$$F_{comp} = (F - K\delta_b) \left(\frac{AE/L}{K + AE/L} \right) \quad (4.13)$$

The variables $\xi = \frac{D}{2L}$ and $\gamma = \frac{K}{AE/L}$ and small deflection assumption of $\xi \ll 1$ are defined to the analysis. Then, Eqn. 4.10 and 4.13 are solved to obtain

$$\frac{2R}{P_{cr}} = -K_{b1}\xi + K_{b3}\xi^3 \quad (4.14)$$

where K_{b1} and K_{b3} denote linear and cubic spring constants, respectively, with

$$K_{b1} = \left(\frac{F}{P_{cr}(1 + \gamma)} - 1 \right) \pi^2 \quad (4.15)$$

$$K_{b3} = \left(\left(\frac{AE\gamma}{P_{cr}(1 + \gamma)} - \frac{4}{3} \left(\frac{F}{P_{cr}(1 + \gamma)} - 1 \right) \right) \right) \pi^4 \quad (4.16)$$

K_{b1} is proportional to the axial force F exerted by the compression mechanism as shown in the Eqn. 4.15. For different variations of axial force, negative, positive and zero stiffness can be achieved with this mechanism. Notably, when the compression force applied to the beam is greater than the first critical buckling load, buckling occurs and stiffness of the beam becomes negative. Furthermore, in the tensile loading mode, the beam introduces positive stiffness, and mechanism's stiffness increases as the tension load increases [74].

4.2.2 Numerical simulations

Using the analytical model provided in the previous chapter, we can analyze the model numerically by defining the parameters, as in Table 4.1. High tensile blue tempered (AISI 1095) leaf springs with 0.508 mm thickness and 18 mm width are used with a half-length L of 140 mm. The modulus of elasticity of the beams is 71 GPa and the critical axial load is calculated as 38.28 N.

Property (unit)	Value
K , axial stiffness of the beam (N/mm)	10.82
L , half of axial length of the beam (mm)	120
E , modulus of elasticity of the beam (GPa)	71
P_{cr} , first critical axial load for a clamped-clamped end points configuration (N)	38.28
T , thickness of the axially loaded beam (mm)	0.508
A , cross-sectional area of the axially loaded beam (mm^2)	9.65

TABLE 4.1: Parameters used to study buckling beams for numerical analysis

Figure 4.3 presents theoretical values for transverse forces and deflections of the variable stiffness module under tensile axial loading. As the tensile load increases, the

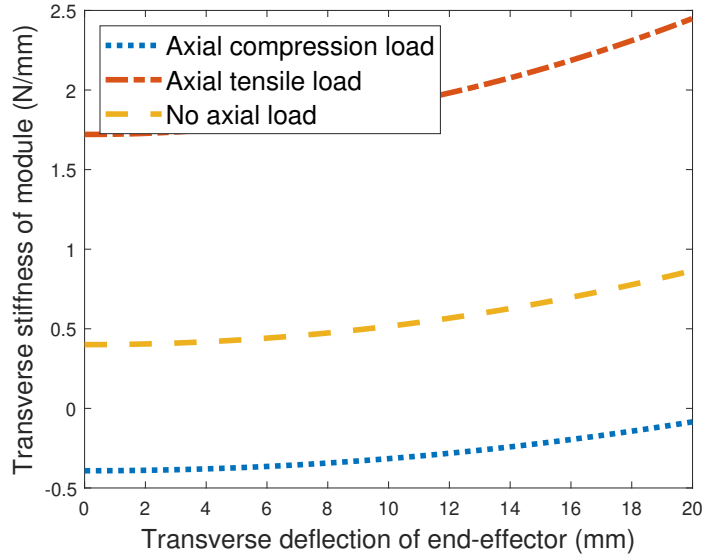


FIGURE 4.3: Stiffness change of axially loaded beams under no axial, axial tensile, and compression loadings

transverse stiffness of the system increases. To obtain a higher stiffness coefficient, one can apply higher tensile axial loading.

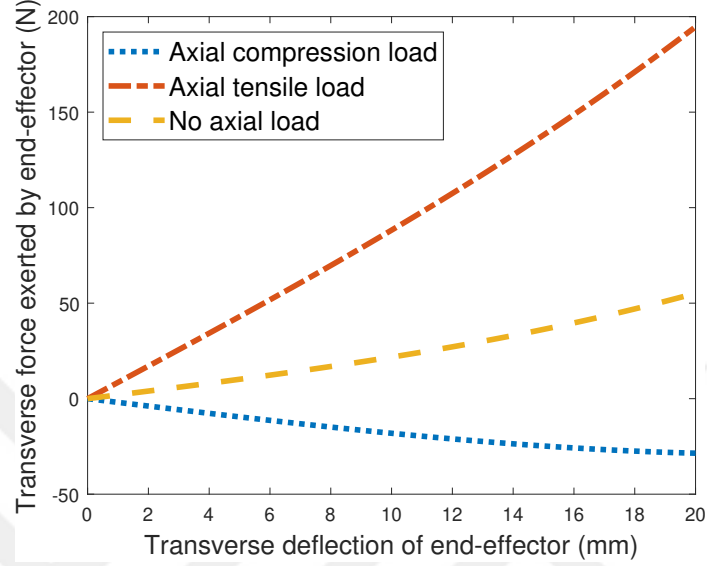


FIGURE 4.4: Stiffness change of axially loaded beams under no axial, axial tensile, and compression loadings

Figure 4.4 presents theoretical values for transverse forces and deflections of the variable stiffness module under compressive axial loading. Transverse stiffness of the leaf springs is zero at the critical buckling load of 38.28 N, and negative stiffness can be achieved compressive forces higher than the buckling load.

4.3 Inerter design

In this section, the analytical model and numerical evaluations of a ball-screw inerter are represented. The following analysis closely follows the one presented in [54].

4.3.1 Analytical model

Considering the system shown in Figure 4.5, the equation of motion for the screw in axial direction can be expressed as

$$F_{\text{applied}} - N_{n,s} = m_s a_s \quad (4.17)$$

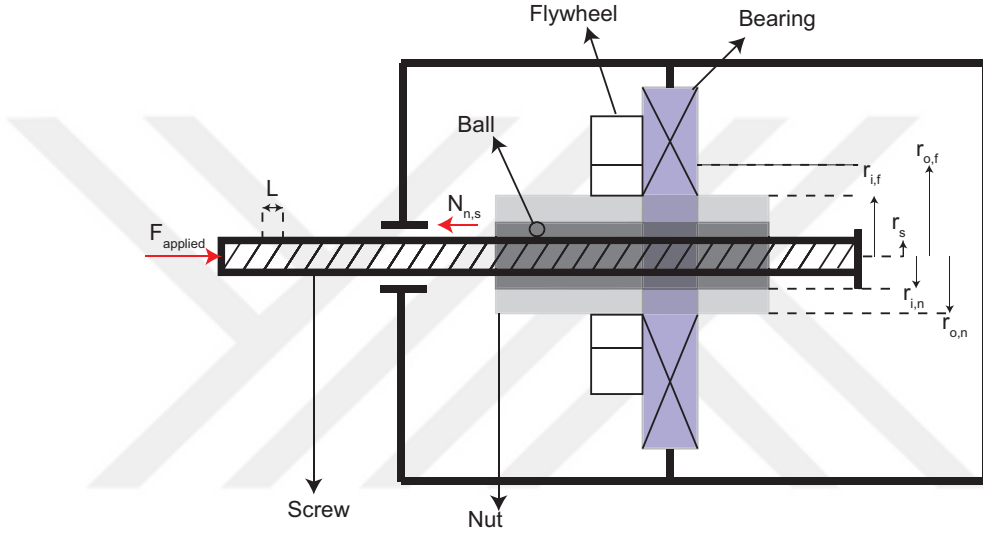


FIGURE 4.5: Schematic representation of mechanical model of an inerter

where F_{applied} is the force applied at the first terminal, $N_{n,s}$ is the reaction force due to the nut on the screw, and a_s is the axial acceleration of the screw. The nut is designed to be rigidly connected to the flywheel. Due to the axial motion of the screw, the rotational equation of motion for the nut and flywheel can be expressed as

$$\frac{N_{s,n} \cdot L}{2\pi} = (I_n + I_f) \alpha_f \quad (4.18)$$

where $N_{s,n}$ is the reaction force due to the screw on the nut, I_n and I_f denote the mass moment of inertia of nut and flywheel, respectively. L is the lead of the screw and α_f represents the common angular acceleration of the flywheel and the nut.

Note that, under the no-slip condition

$$\alpha_f = \frac{2\pi}{L} a_s \quad (4.19)$$

One can arrange the Eqn. 4.18 to solve for the reaction force $N_{s,n}$ as

$$N_{s,n} = (I_n + I_f) \alpha_f \frac{2\pi}{L} \quad (4.20)$$

Then, by using Eqns. 4.17–4.19, the following equality can be derived

$$\begin{aligned} F_{\text{applied}} &= N_{n,s} + m_s a_s \\ &= \left[(I_n + I_f) \left(\frac{2\pi}{L} \right)^2 + m_s \right] a_s \end{aligned}$$

where the mass moment of inertia for the flywheel and the nut read as

$$\begin{aligned} I_f &= \frac{1}{2} m_f (r_{i,f}^2 + r_{o,f}^2) \\ I_n &= \frac{1}{2} m_n (r_{i,n}^2 + r_{o,n}^2) \end{aligned}$$

with m_f and m_n denoting mass, $r_{i,f}$ and $r_{i,n}$ denoting the inner radius, $r_{o,f}$ and $r_{o,n}$ denoting the outer radius of the flywheel and the nut, respectively. For ball-screw inerters, the equation of motion can be approximated as

$$F_{\text{applied}} = \left[\left(\frac{m_n r_n^2}{2} + m_f k_f^2 \right) \left(\frac{2\pi}{L} \right)^2 + m_s \right] a_s \quad (4.21)$$

where k_f is the radius of gyration of the flywheel. Accordingly, the approximated inertance factor for this design can be defined as

$$m = \left(\frac{m_n r_n^2}{2} + m_f k_f^2 \right) \left(\frac{2\pi}{L} \right)^2 + m_s \quad (4.22)$$

It should be noted that Eqn. 4.21 does not include any slip and friction effects and consider the nut and the flywheel as solid cylinders.

4.3.2 Numerical evaluations

Using the analytical model, a case study is performed in this section to numerically study the characteristics of a ball-screw inerter. To achieve an high performance inerter design, we define a new variable called the effective inertance as

$$m_{\text{effective}} = \frac{m}{M_{\text{inertance}}} \quad (4.23)$$

For high performance, one needs to maximize the effective inertance, that is the inertance factor per unit mass. The inertance m and the mass $M_{\text{inertance}}$ of inertance module can be computed as follows:

$$m = 2 \left(m_f(r_{i,f}^2 + r_{o,f}^2) + m_n(r_{i,n}^2 + r_{o,n}^2) \right) \left(\frac{\pi}{L} \right)^2 + m_s \quad (4.24)$$

$$M_{\text{inertance}} = M_{\text{case}} + M_{\text{bearing}} + m_s + m_n + m_f \quad (4.25)$$

For an optimal inertance design, one should compute the maximum inertance factor per unit inertance mass, which is the total mass of all employed physical elements. Defining higher inertia for a flywheel or rigidly connected nut corresponds to a higher inertance factor. However, a high mass of the inerter may also appear.

In Table 4.2, some design parameters are identified. For the backpack design, we need to avoid a high additional mass due to inerter, i.e., avoid high values of $M_{\text{inertance}}$. Inerters may be designed by considering $m_{\text{effective}}$ as a design variable for such applications. It should be noted that increasing the radius of mechanical units brings more mass to the system and may not result in greater $m_{\text{effective}}$ as represented

Component	Parameter	Value
Bearing	Mass	0.26 kg
Casing	Mass	0.35 kg
Ball-Screw	Density	7890 kg/m ³
	Lead	14 mm
	Length	350 mm
	Radius	2 mm
Nut	Density	2700 kg/m ³
	Thickness	60 mm
	Outer radius	25 mm
	Inner radius	5 mm
Flywheel	Density	1240 kg/m ³
	Thickness	$13 < f_t < 25$ mm
	Outer radius	$25\text{mm} < r_{o,f}$
	Inner radius	25 mm

TABLE 4.2: Parameters characterizing the ball-screw inerter.

in the contour plot given in Figure 4.6. Since higher inertia of flywheel corresponds to a high value of inertance, we can conclude that there are only mechanical limitations for the flywheel design. As long as the flywheel is suitable to employ in the mechanism, its inertance factor can be increased with respect to the mass of inertance. Hence, the limitations are due to the mechanical dimensions of the device.

Increasing outer radius and thickness of flywheel corresponds to a higher inertance, but may not satisfy design requirements. It is worth mentioning that designs with higher flywheel inertia require higher volume for the casing, thus the mass of the casing may also increase.

For testing, we have designed three levels of inertia for the flywheel, as low, medium, and high. For a low level of inertia, no flywheel is employed in the inerter, which results in an inertance of 23.4 kg. The mass of inertance for the low-level of inertance configuration is 0.731 kg, so that $m_{\text{effective}}$ is 32.01. For the medium level, a flywheel with $r_{o,f} = 45$ mm and $f_t = 25$ mm, results in an inertance factor 65.6 kg and mass of 0.92 kg. In the high-level mode of inertance, the outer radius is not changed but thickness is increased to 50 mm. This configuration has an inertance of 143.4 kg,

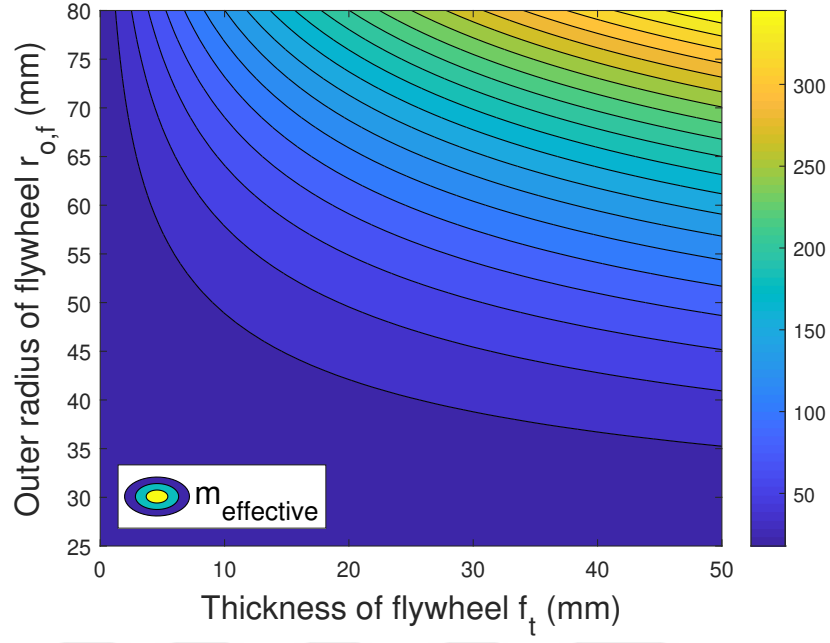


FIGURE 4.6: Contour plot of $m_{\text{effective}}$ for different values of flywheel outer radius and thickness for the parameters given in the Table 4.2

and a mass of 1.014 kg. Comparatively, $m_{\text{effective}}$ values for the medium and high level inertias are 71.3 and 141.42, respectively.

Chapter 5

Implementation and Experimental Characterization

This chapter presents the implementation details of the proposed backpack prototype and the experimental characterization of the variable impedance of the suspended-load backpack under various conditions. The inerter and stiffness modules are examined within a range of frequencies through standalone experiments. These modules are also characterized when they are coupled to each other.

5.1 Implementation of the inerter module

A ball-screw inerter device is designed with three levels (low, medium and high) of inertance. As it is provided in the mathematical modeling section, inertance factor of the inerter module can be adjusted by changing the inertia of the flywheel.

5.1.1 Experimental setup

An accelerometer (VN-100, Vectornav) is attached to the ball-screw type inerter device's moving terminal, as shown in Figure 5.1. A DC motor (Maxon RE 50, Maxon Group) is attached to the end-effector that attaches actuator and one of the terminals of the inerter via a low friction precision slider with 2" stroke. A force sensor (F/T Sensor Mini58, ATI Industrial Automation) is also mounted on the terminal to measure the forces exerted on the inerter.

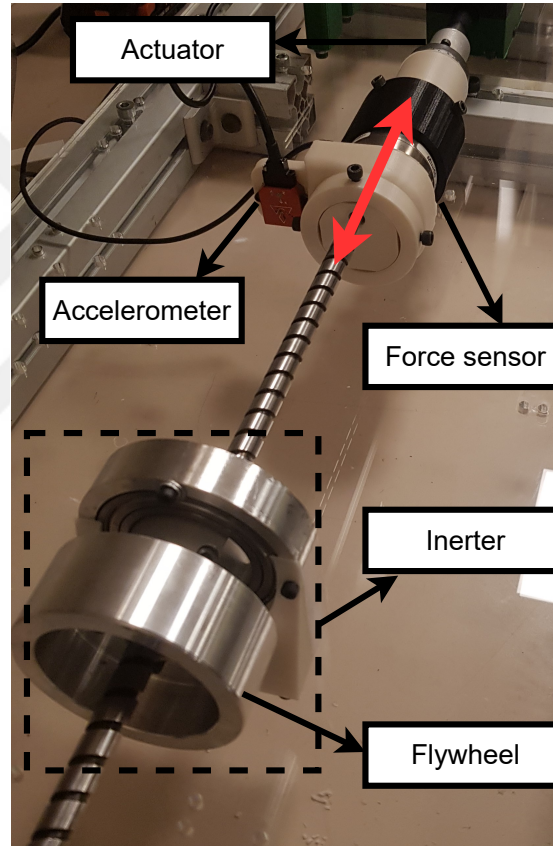


FIGURE 5.1: Experimental system identification of the variable negative stiffness module under no axial loading

The other terminal of the inerter is grounded. The actuator is driven by a motor controller (MAXPOS 50/5, Maxon Group) by using EtherCAT communication protocol. The device is actuated along with the red line as it is shown in Figure 5.1.

5.1.1.1 System model

A motion controlled actuator was employed in the setup to excite the end-effector. The passive mechanical device is actuated and characterized as a mechanical one-port as it represented in Figure 5.2. The accelerometer measures acceleration of the end-effector with respect to ground. Moreover, the force sensor measures the force F_a exerted on the end-effector with respect to ground. Ball-screw inerter is modeled as mechanical one-port device and its frequency domain characteristics are experimentally characterized within a range of frequencies.

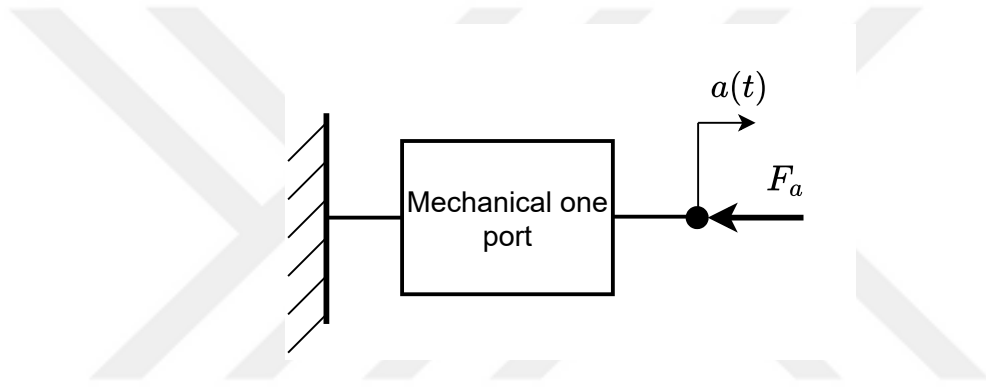


FIGURE 5.2: Schematic representation of the inerter characterization experiment

5.1.2 Actuator dynamics

A DC motor is employed with a ball-screw slider to obtain linear motion. The actuator exhibits its own dynamics with the driver used to control it, as shown in Figure 5.3.

The actuator is excited to follow a sinusoidal position input within a range of different frequencies that range from 0.01 Hz to 50 Hz . The position of the end-effector is measured with an encoder (HEDL 5540, Maxon Group). Actuator's transfer function is shown in the Eqn. 5.1.

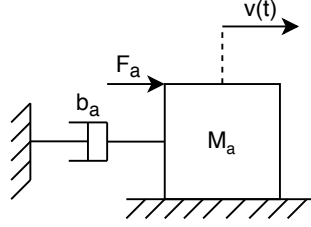


FIGURE 5.3: Schematic representation of the linear actuator

$$\mathcal{L}\left\{\frac{v(t)}{F_a(t)}\right\} \approx T_a(s) = \frac{1}{M_a s + b_a} \quad (5.1)$$

Actuator position control bandwidth is determined through frequency response of the following transfer function:

$$T(s) = \mathcal{L}\left\{\frac{x(t)}{x_d(t)}\right\} \quad (5.2)$$

where $x(t)$ is the position of the actuator end-effector and $x_d(t)$ is the desired position commanded to the controller. The transfer function given in the Eqn. 5.2 is experimentally presented in Figure 5.4.

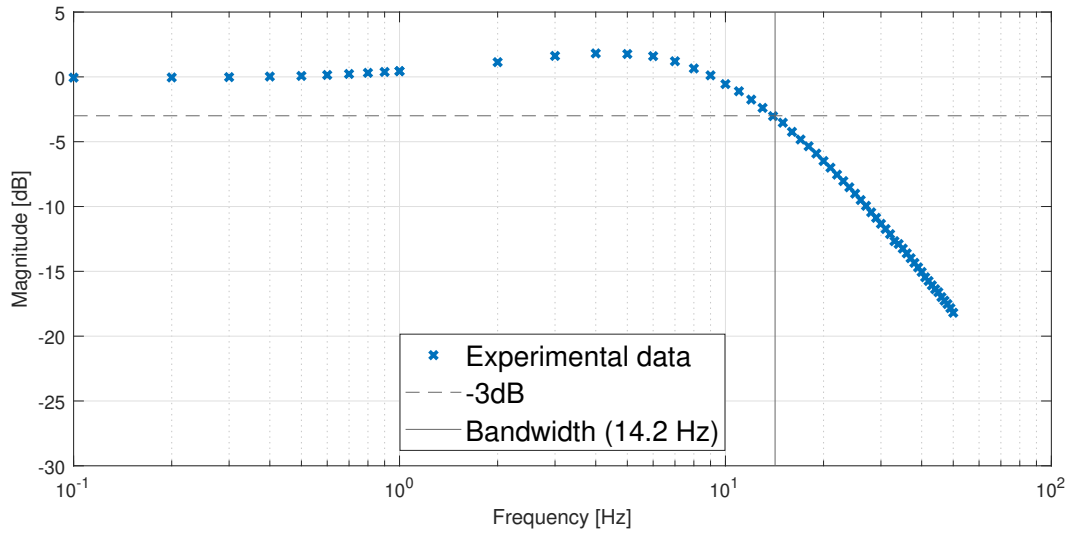


FIGURE 5.4: Experimental system identification of the actuator bandwidth

Using the magnitude response plot of $T(s)$ ($|T(j\omega)|_{dB}$ versus frequency), we can identify the motion control bandwidth as 14.2 Hz , as -3 dB is observed at this frequency. Up to this frequency, the actuator can be closely approximated as an ideal motion source.

5.1.3 Characterization results

In order to characterize the inerter device and verify the results of the analytical models, a set of experiments are implemented. This section presents the experimental characterization results for each configuration of the inerter. The transfer functions are estimated by using Welch's averaged periodogram method, with a sample rate of 500 Hz .

5.1.3.1 System characteristics

Impedance characteristics of the inerter module is experimentally characterized by exciting the device with a chirp signal from 0.1 Hz to 10 Hz . One of the terminals of the device is excited with a motion input of to follow sinusoidal trajectory of $x(t)$

$$x(t) = A_0 \sin(\omega t) \quad (5.3)$$

In this setup, the device characteristics are estimated by using following transfer function

$$\mathcal{L} \left\{ \frac{a(t)}{F_a(t)} \right\} \approx T_1(s) = \frac{s^2}{m_1 s^2 + b_{i,1} s} \quad (5.4)$$

where m_1 and b_1 denote inertance and damping characteristics of inertance. Furthermore, we had also identified inerter with a stiffness characteristics of k_i . Note that m_2 and b_2 denote inertance and damping characteristics captured by the following transfer function

$$\mathcal{L} \left\{ \frac{a(t)}{F_a(t)} \right\} \approx T_2(s) = \frac{s^2}{m_2 s^2 + b_{i,2} s + k_i} \quad (5.5)$$

for both transfer function estimations $a(t)$ and $F_a(t)$ denote the acceleration of the end-effector and the force exerted on it, respectively. Furthermore, m denotes the inertance of the inerter, b_i and k_i stand for the damping and the stiffness characteristics of the inerter respectively. Frequency domain response of the device is identified by fitting the above presented transfer functions. The inertia of the flywheel and nut have a significant impact on the inertance of a ball screw inerter. Consequently, we consider three different levels of inertance, which includes only the nut, the nut and a flywheel with low inertia, and the nut and a high inertia flywheel. This section presents the experimental characterization results for each configuration of the inerter.

Low inertance

To investigate the low inertance configuration of the device, no flywheel is attached to the mechanical one-port device. The device is excited over a range of frequencies, and the impedance characteristics of the device is experimentally evaluated as given in the Figure 5.5. The match between the experimental data and estimation is represented as R^2 value of 79% if we fit $T_1(s)$ to the system. However, $T_2(s)$ is employed in the estimation R^2 value increases to 99%.

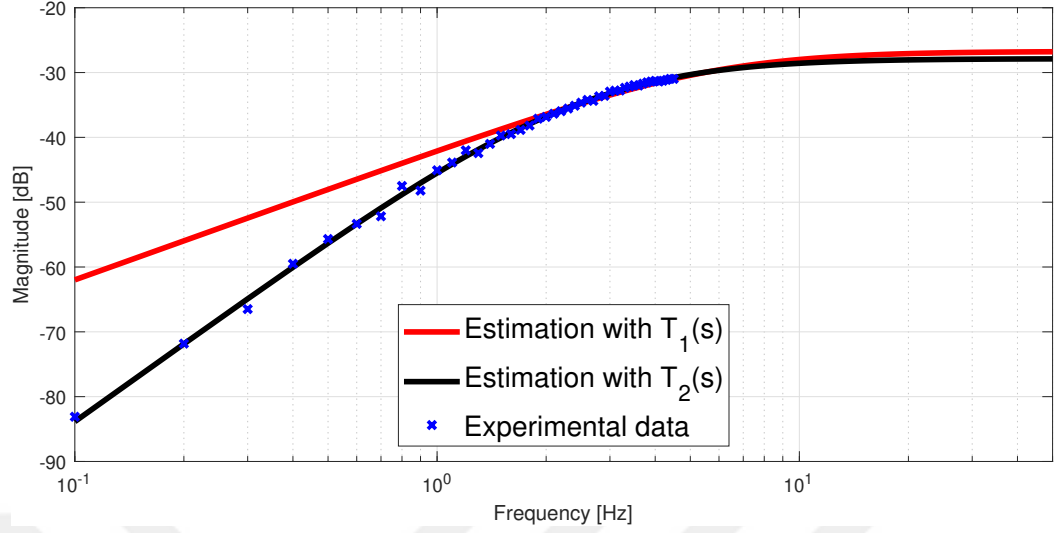


FIGURE 5.5: Experimental system identification of the inerter module with low inertance configuration

In Table 5.1, identified system parameters are presented. The inertance of the inerter is characterized as 24.72 kg , with a damping of 135.4 Ns/m with a low amount of stiffness, 0.155 N/mm using $T_2(s)$. Inertance and damping was estimated 21.71 kg and 125.6 Ns/m by $T_1(s)$. However, match rate is affected by the unmodeled dynamics of the $T_1(s)$ estimation.

Property	Description	Unit	Value
m_1	Inertance	kg	21.71
$b_{i,1}$	Damping	Ns/m	125.6
m_2	Inertance	kg	24.72
$b_{i,2}$	Damping	Ns/m	135.4
k_i	Stiffness	N/mm	0.155
$M_{\text{inertance}}$	Mass	kg	0.731
R_1^2	R-squared value for the $T_1(s)$ estimation	%	79
R_2^2	R-squared value for the $T_2(s)$ estimation	%	99

TABLE 5.1: Inerter characteristics when no flywheel is attached to the nut

Analytical model of inerter for low inertance configuration estimated inertance as 23.4 kg , which is in good agreement with the experimental results.

Medium inertance

A flywheel with a relatively low inertia is attached to the nut of the ball-screw in-
terter to increase the inertia of the rotating part of the device. In Figure 5.6, the
experimental data is presented along with the $T_1(s)$ and $T_2(s)$ estimations.

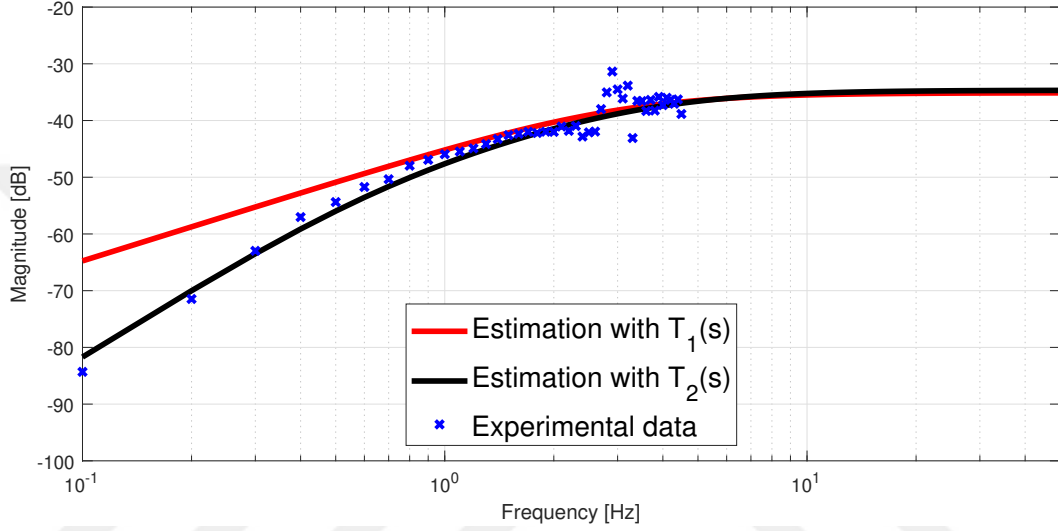


FIGURE 5.6: Experimental system identification of the inerter module with medium inertance configuration

Estimation results are presented in Table 5.2. A good match rate of 95% with the $T_2(s)$, and 80% with the $T_1(s)$ are achieved. The inertance of the device is characterized as 54.08 kg, with a damping of 231.3 Ns/m with $T_2(s)$ estimation. Moreover, a relatively low stiffness characteristics is observed by the estimation as 0.12 N/mm. $T_1(s)$ estimates, 56.97 kg inertance, with a damping of 172.7 Ns/m. Note that, stiffness is dominant on frequency response at low frequencies. Neglecting the stiffness characteristics results in a decreased R-squared for the estimation. Analytical model of inerter for medium inertance configuration estimated as 65.6 kg, which is in a relatively good agreement with the experimental results. Inerter displayed an additional mass characteristics by mounting a flywheel with relatively high inertia when compared with the nut only case.

Property	Description	Unit	Value
m_1	Inertance	kg	56.97
$b_{i,1}$	Damping	Ns/m	172.2
m_2	Inertance	kg	54.08
$b_{i,2}$	Damping	Ns/m	231.3
k_i	Stiffness	N/mm	0.120
$M_{\text{inertance}}$	Mass	kg	0.920
R_1^2	R-squared value for the $T_1(s)$ estimation	%	80
R_2^2	R-squared value for the $T_2(s)$ estimation	%	95

TABLE 5.2: Inerter characteristics when the device is configured for medium level inertance

High inertance

The inerter module is also characterized in the high inertance configuration in which a relatively high inertia flywheel is attached to the device. The experimental data with $T_1(s)$ and $T_2(s)$ transfer function estimations are presented in Figure 5.7.

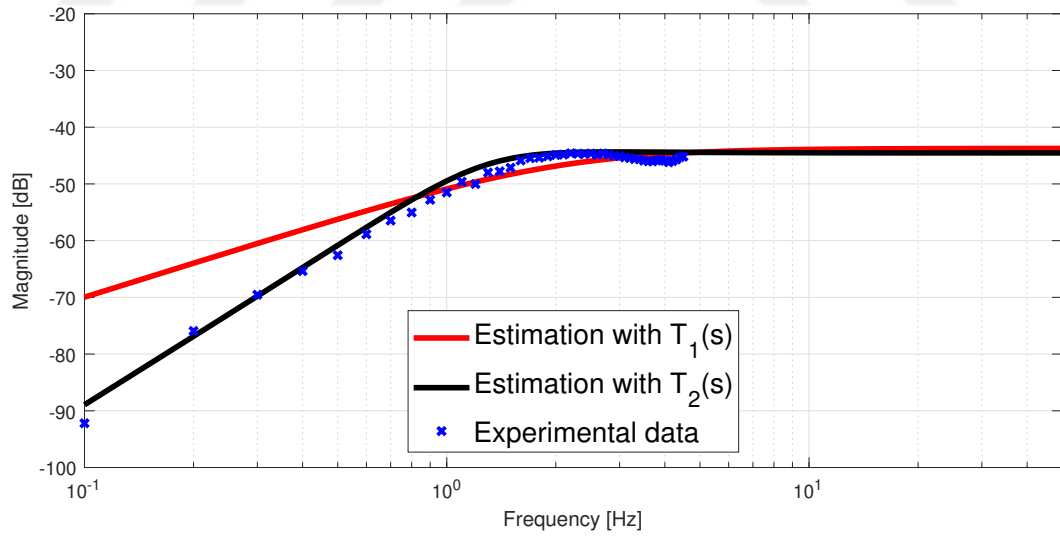


FIGURE 5.7: Experimental system identification of the inerter module with high inertance configuration

Estimated system characteristics are presented in Table 5.3.

Property	Description	Unit	Value
m_1	Inertance	kg	153.8
$b_{i,1}$	Damping	Ns/m	314.6
m_2	Inertance	kg	139.5
$b_{i,2}$	Damping	Ns/m	275.4
k_i	Stiffness	N/mm	0.280
$M_{\text{inertance}}$	Mass	kg	1.014
R_1^2	R-squared value for the $T_1(s)$ estimation	%	77
R_2^2	R-squared value for the $T_2(s)$ estimation	%	98

TABLE 5.3: Inerter characteristics when the device is configured for high level inertance

Using $T_1(s)$, inertance and damping were estimated as 153.8 kg and 314.6 Ns/m respectively. The inertance of the device was identified as 134.7 kg with 286.9 Ns/m damping during the estimation with $T_2(s)$. A low stiffness of 0.120 N/mm was also identified with the second model. The match rate criteria, R-squared value of the estimation, has decreased from 98% to 77% as the stiffness characteristics of 0.306 N/mm is neglected. Analytical model of inerter for low inertance configuration estimated m as 143.4 kg , which is in good agreement with the experimental results. Inerter displayed additional ~ 120 kg of mass characteristics by mounting a flywheel with relatively high inertia when compared with nut only case.

5.1.3.2 Discussion

The impedance characteristics of the inerter module is examined by using the force-acceleration relationship as defined in Eqn. 5.4 and Eqn. 5.5 for different configurations of the inerter. As expected, device's inertance, in other words equivalent inertia characteristics is adjusted by modulating the flywheel inertia. Relatively low inertance for the module is achieved with no flywheel employment. In this case the rotating inertia is smaller than the other configurations.

When the system is configured with a flywheel having medium and high inertias, higher inertances are achieved for a significantly low added mass. During mathematical modeling, we designed three levels of inertance for the inerter by configuring the inertia of the flywheel. We expected 23.4 kg inertance when we did not attach a flywheel to the nut. During characterization, the low inertance configuration resulted in 24.72 kg and 21.71 kg for $T_1(s)$ and $T_2(s)$, respectively. The difference between our model and characterization results may be caused by uncertain inertia of the nut. Then, 54 kg and 139 kg inertances were observed during medium and high inertance configurations which are relatively close to the mathematical model predictions of 65.6 and 143.4 kg, respectively.

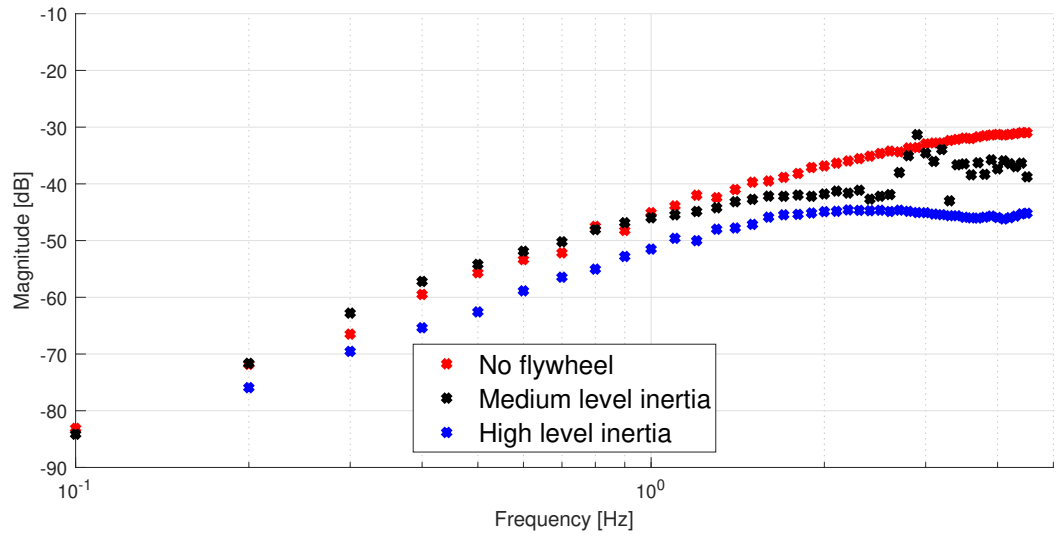


FIGURE 5.8: Experimental system identification of the variable negative stiffness module under no axial loading

Furthermore, some low level of stiffness characteristic (0.185 ± 0.084 N/mm) was observed during the identification due to unmodeled dynamics of the parts employed in the inerter and the experimental setup. Low stiffness may be caused by the small deflections of the ball screw and the compliance of 3D printed parts employed during characterization.

However, the device also displayed relatively high damping characteristics (213 ± 70 Ns/m) due to the bearings and the ball-screw utilized in the design of the inerter. As we employ the high inertia flywheel, the resulting damping was increased, as increasing loading on the nut and bearings may cause increased damping due to the increased friction between moving parts.

5.2 Implementation of the variable stiffness module

Figure 5.9 presents the functional prototype of the proposed backpack. In this prototype, high tensile blue tempered (AISI 1095) leaf springs with 0.508 mm thickness and 18 mm width are used to implement the variable-stiffness module. To axially load the beams, two compression mechanisms are employed with a lead screw with 1.5 mm lead.

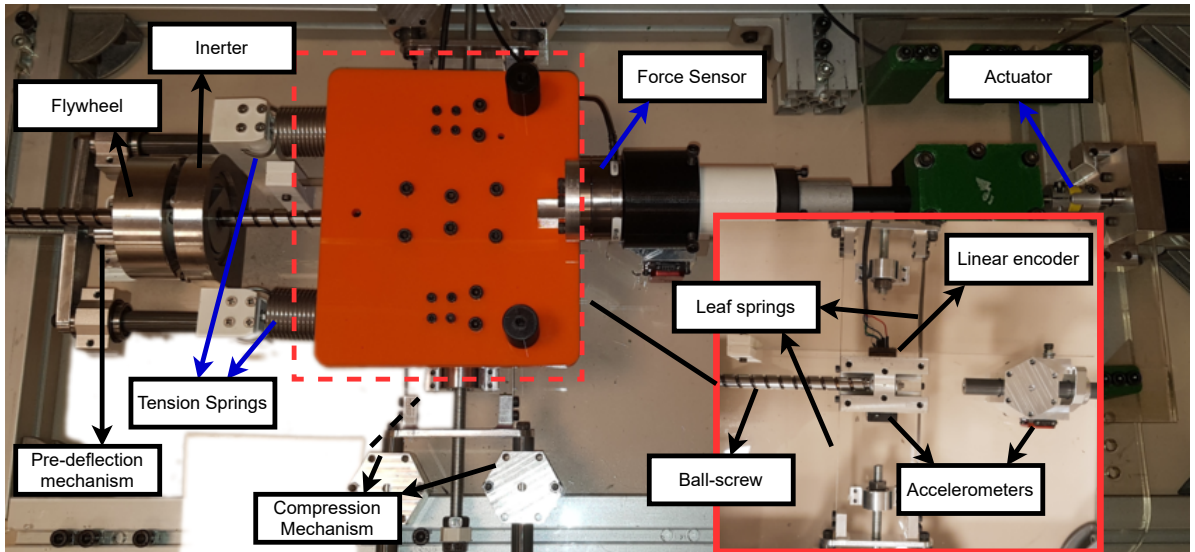


FIGURE 5.9: Prototype of experimental setup for inerter-variable stiffness based suspended load backpack design

The ball-screw inerter and two tension springs (0.60 N/mm) with a pre-tension mechanism are attached to the end-effector of the backpack. The deflection of the end-effector due to the mass of the carried load is compensated via the pre-tension mechanism.

5.2.1 Experimental setup

Accelerometers are employed in the setup to measure the vibrations of the proposed design, as shown in Figure 5.9. Two accelerometers are attached to the end-effector and base of the backpack device in order to measure the accelerations of the human and load. The end-effector of the actuator is attached to the moving base of the backpack device. A DC motor excites the end-effector of the system via a precision low friction slider with 2" stroke. A linear encoder is attached to the end-effector to measure relative position of the end-effector with respect to ground. A force sensor is mounted on the end-effector. Actuator is driven by a motor controller by using EtherCAT protocol and controlled as direct motion source.

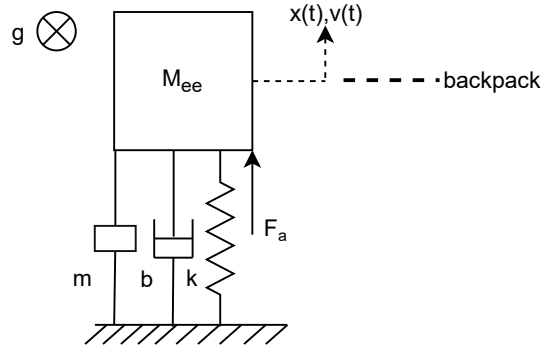


FIGURE 5.10: Schematic representation of the experimental setup

5.2.1.1 System model

The passive mechanical system is actuated by a DC motor and is estimated as a second order low pass filter as shown in Figure 5.10. In this representation, M_{ee}

denotes the mass of end-effector. m is the inertance of the inerter, k is the equivalent stiffness imposed by variable stiffness module and compression spring, and b is the cumulative damping of the mechanism, respectively. Finally, $x(t)$ denotes the position of the end-effector and g is the gravitational acceleration.

The end-effector of the device is excited with a motion input of to follow sinusoidal trajectory of $x(t)$

$$x(t) = A_0 \sin(\omega t) \quad (5.6)$$

Then the parameters of the backpack is estimated by the following transfer function

$$\mathcal{L}\left\{\frac{x(t)}{F_a(t)}\right\} \approx G(s) = \frac{1}{m_{eq}s^2 + b_{eq}s + k_{eq}} \quad (5.7)$$

for which the magnitude of the transfer function is defined as

$$|G(j\omega)|_{dB} = 20 \log_{10} \left| \frac{1}{-m_{eq}\omega^2 + b_{eq}j\omega + k_{eq}} \right| \quad (5.8)$$

5.2.2 Actuator Dynamics

The same actuator employed in the inerter module characterization (see the Section 5.1.1) is employed in the experiments. As it is represented in Section 5.1.2, the actuator dynamics can be neglected through the analysis, since the DC motor can be considered as an ideal motion source up to its bandwidth, which is 14.2 Hz.

5.2.3 Characterization Results

In order to characterize the backpack device and verify the results predicted by the analytical models, a set of experiments is implemented. This section presents the experimental characterization results for the variable stiffness module. The inerter module is attached to the variable stiffness module to identify dynamic characteristics of inerter and variable stiffness modules incorporated. The transfer functions are estimated by using Welch's averaged periodogram method, with a sample rate of 500 Hz .

5.2.3.1 Variable stiffness module

The stiffness characteristics of the variable stiffness module can be manually adjusted by changing the direction and magnitude of the axial load exerted to the beams. A reduction in the stiffness results in decreased natural frequency of the backpack. To validate the adjustability of stiffness characteristics of the variable stiffness module, the inerter module is detached. The variable stiffness module coupled to the tension spring is actuated within a range of frequencies. The frequency response of the system is estimated with the transfer function given in the Eqn. 5.9. The end-effector is excited to follow a sinusoidal motion input given in the Eqn. 5.6, within a range of different frequencies that ranges from 0.1 Hz to 10 Hz . The position of the end-effector and effective actuation force are measured.

$$G_{vs}(s) \approx \frac{1}{m_{eq}^{vs} s^2 + b_{eq}^{vs} s + k_{eq}^{vs}} = \frac{1}{M_{ee} s^2 + b_d s + (k_n + k_s)} \quad (5.9)$$

Depending on the condition an axial load was applied to the mechanism. During no axial load, mechanism stays at the initial condition without any compressive or axial load. Moreover, an axial deflection was provided during the compression and

tension modes. In particular, 3 mm deflection introduced from both sides resulted in 40.5 N axial force during the compression mode. Furthermore, during tension mode, 54 N tension force resulted from 5 mm deflection. The cumulative stiffness of the system is sum of stiffness values of four buckled beams and tension springs (1.2 N/mm). Other parameters used for modeling were presented in Table 4.1.

No axial loading

In the first mode, no axial loading was applied to the variable stiffness module. The system was excited and its response under different frequencies is investigated to characterize dynamics. In Figure 5.11, experimental data and estimated transfer function are presented.

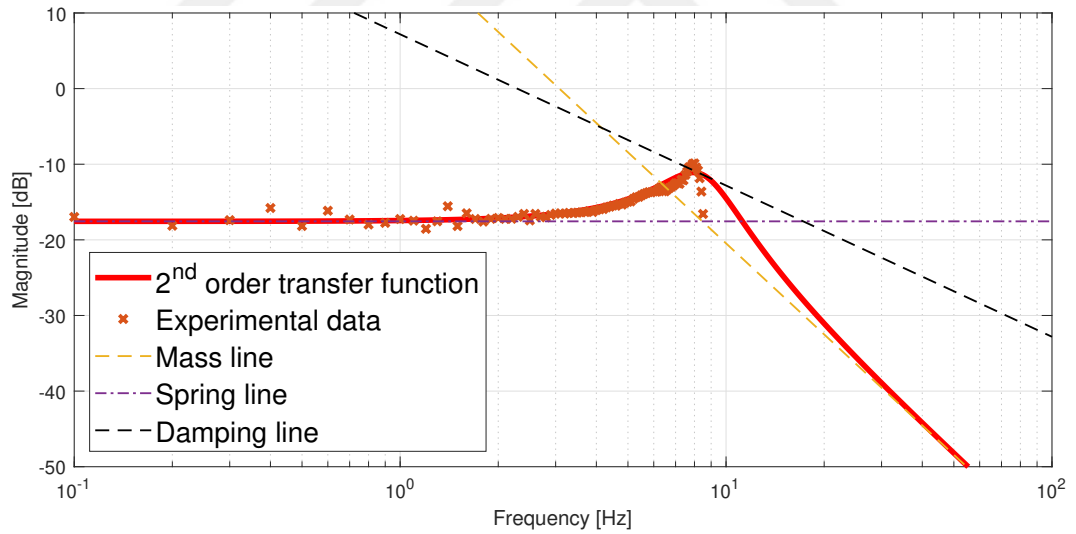


FIGURE 5.11: Experimental system identification of the variable negative stiffness module under no axial loading

Table 5.4 presents the parameters of the variable negative stiffness module under no axial loading. The stiffness introduced by the beams is embedded in the cumulative spring stiffness, and a good match rate is achieved with R^2 of 87%. System's

estimated transfer function indicates that natural frequency of the module is 13.459 Hz.

Property	Description	Unit	Value
M_{ee}	Mass of end-effector	kg	1.055
b_d	Cumulative damping	Ns/m	4.374
$k_s + k_n$	Cumulative stiffness of tension spring and buckled beam	N/mm	7.545
R^2	R-squared value	%	87
RMSE	Root mean squared error	1	0.002096
ω_n	Natural frequency	Hz	13.459

TABLE 5.4: Variable negative stiffness module characteristics under no axial loading.

The mathematical model in Eqn. 4.15 estimates cumulative stiffness of the system as 7.49 N/mm. Moreover, cumulative stiffness of the tension spring and the buckled beams was characterized as 7.545 N/mm, which shows a good match between the mathematical model and experimental results.

Tension mode

Secondly, an axial tension force of 54 N was applied to the variable stiffness module in order to characterize its dynamics in the tension mode.

Again, the system is excited under a range of frequencies to examine its mechanical characteristics. In Figure 5.12, the estimated transfer function and experimental data are represented. Experimental data and estimated transfer function have a match rate R^2 of 96%. Due to increased stiffness, system's natural frequency is increased from 13.459 Hz to 19.639 Hz. The characteristics of the variable negative stiffness module under tension force is presented in Table 5.5. The axial load introduces an increase in the system's overall stiffness.

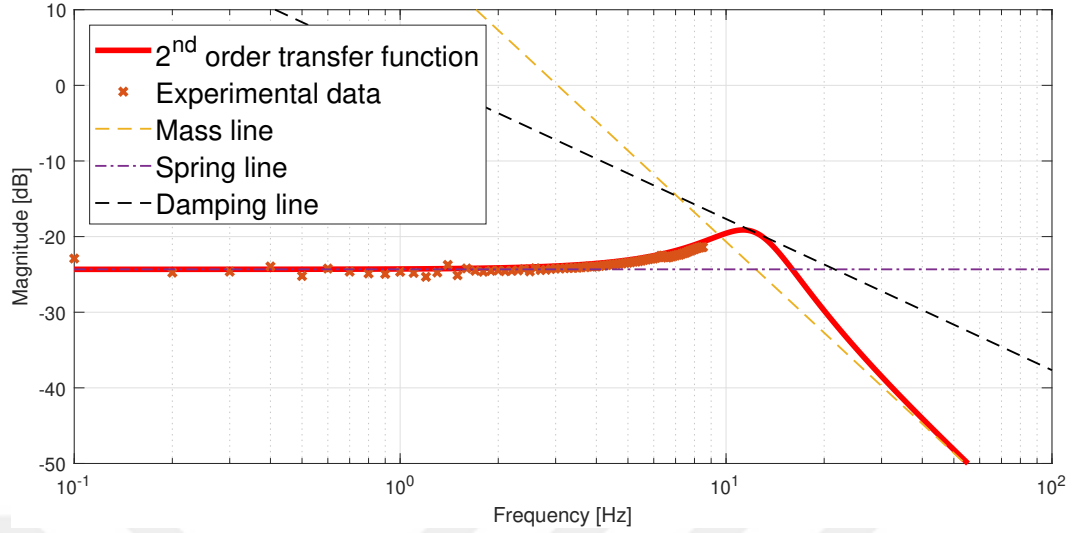


FIGURE 5.12: Experimental system identification of the variable negative stiffness module under tensile loading

The mathematical model in Eqn. 4.15 estimates an increase of 8.88 N/mm stiffness due to tensile axial loading. The cumulative stiffness of the system was characterized as 16.46 N/mm stiffness, which corresponds to a 8.91 N/mm increase when it compared with the no axial load case. Results show a good match between the mathematical model and system identification results.

Property	Description	Unit	Value
M_{ee}	Mass of end-effector	kg	1.081
b_d	Cumulative damping	Ns/m	7.643
$k_s + k_n$	Cumulative stiffness of tension spring and buckled beam	N/mm	16.46
R^2	R-squared value	%	96
RMSE	Root mean squared error	1	0.000169
ω_n	Natural frequency	Hz	19.639

TABLE 5.5: Variable negative stiffness module characteristics under tension force.

Compression mode

Finally, the variable negative stiffness module is characterized under a compression force of 40.5 N. The forced vibration characteristics of the system are estimated as

in Figure 5.13.

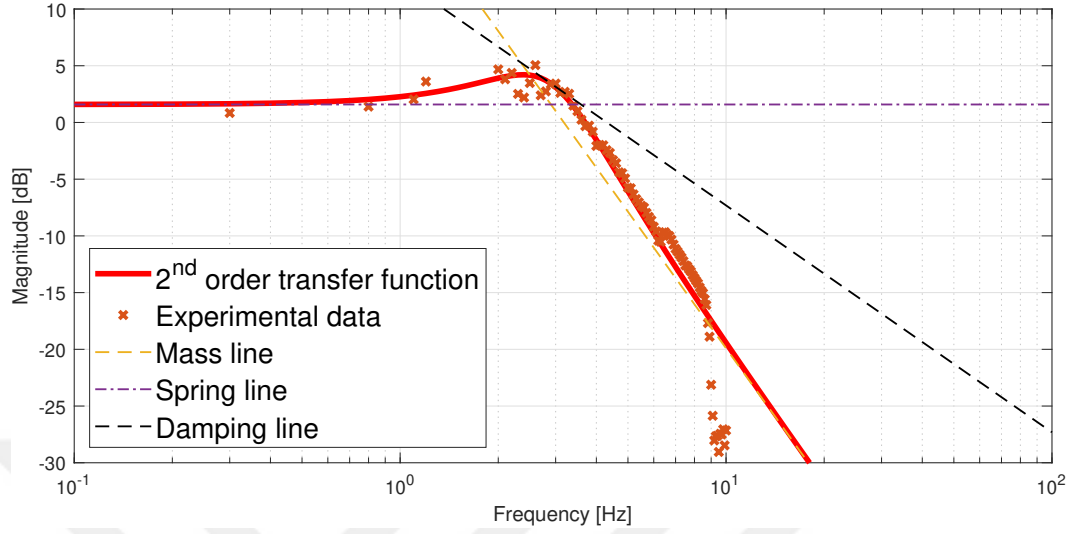


FIGURE 5.13: Experimental system identification of the variable negative stiffness module under compression load.

Experimental data and the estimated transfer function has a good match rate R^2 of 84%. The parameters of the variable negative stiffness module under compression force is shown in Table 5.6.

Property	Description	Unit	Value
M_{ee}	Mass of end-effector	kg	0.9895
b_d	Cumulative damping	Ns/m	2.321
$k_s + k_n$	Cumulative stiffness of tension spring and buckled beam	N/mm	0.8331
R^2	R-squared value	%	84
RMSE	Root mean squared error	1	0.02508
ω_n	Natural frequency	Hz	4.618

TABLE 5.6: Variable negative stiffness module characteristics under axial compression

The equivalent stiffness of the module is decreased due to the property of buckled beam under compression. Natural frequency of the system under compression is estimated as 4.61 Hz, which is smaller than system's estimated natural frequency under no loading. The mathematical model in Eqn. 4.15 estimates a decrease of 6.66

N/mm stiffness. Meanwhile, cumulative stiffness of the system was characterized as 0.8331 N/mm stiffness, which corresponds to 6.71 N/mm decrease compared with identified characteristics. The mathematical model matches with the identified system characteristics.

5.2.3.2 The inerter module with the variable stiffness module

We have further characterized the inerter module attached to the variable negative stiffness module under compression and natural frequency of the system is experimentally characterized by using following transfer function estimation:

$$G_i(s) \approx \frac{s}{m_{eq} s^2 + b_{eq} s + k_{eq}} = \frac{s}{(M_{ee} + m) s^2 + b_d s + (k_n + k_s)} \quad (5.10)$$

For this configuration, cumulative reflected mass is the sum of end-effector mass M_{ee} and inertance factor of m . It should be noted that, the inerter module introduces some stiffness and damping to the system. We hypothesise that inerters can be employed in the backpack designs to reduce the natural frequency of the system without introducing significant level of additional mass. The inertance of a ball-screw inerter is highly influenced by the inertia of the flywheel and the nut. Thus, we have investigated 3 different levels of inertance.

Low inertance

The system is excited within a range of frequencies and the overall response is given in Figure 5.14 with the estimated transfer function.

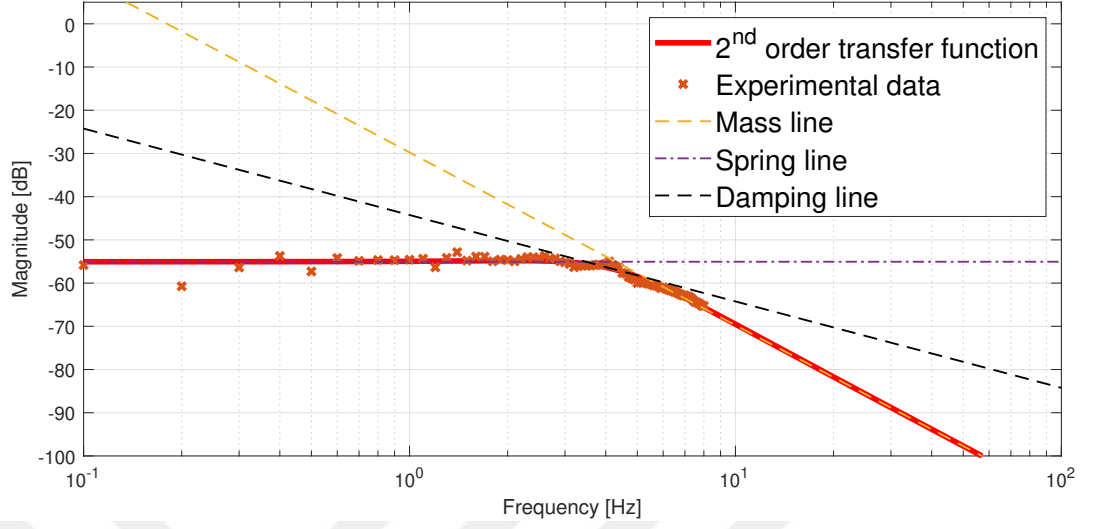


FIGURE 5.14: Experimental system identification of the inerter module with no flywheel and variable negative stiffness module under compression.

Table 5.7 presents the characteristics of the backpack device for the given frequency response in Figure 5.14. The match between the experimental data and the transfer function is calculated as R^2 of 92%. At low inertance, effective mass characteristics of end-effector and inertance is estimated as 30.73 kg. Natural frequency of the system is calculated as 0.69 Hz without a load. This value can be reduced up to 0.51 Hz during walking with a 25 kg load.

Property	Description	Unit	Value
$M_{ee} + m$	Sum of mass of end-effector and Inertance	kg	30.73
b_d	Cumulative damping	Ns/m	163
$k_s + k_n$	Cumulative stiffness of compression spring and buckled beam	N/mm	0.5662
R^2	R-squared value	%	92
RMSE	Root mean squared error	1	0.9842
ω_n	Natural frequency	Hz	1.6191

TABLE 5.7: Inerter with no flywheel and variable negative stiffness module characteristics under compression

Mathematical model given in Eqn. 4.21 estimates 23.4 kg inertance for the inerter module. An axial compression force (40.5 N) was applied to the variable stiffness module and 0.8343 N/mm stiffness is expected by the model given in Eqn. 4.15. Characterization result given in Table 5.7 presents a good match between mathematical model and experimental results. Stiffness of the system was reduced by 6.93 N/mm (6.66 N/mm was expected), and additional effective mass was increased by 29.93 kg which is a relatively similar amount of modeled inertance.

Medium inertance

A relatively low inertia flywheel is rigidly attached to the nut of the ball-screw inerter to increase the inertance of the module. In Figure 5.15, the experimental data and the estimated transfer function are presented.

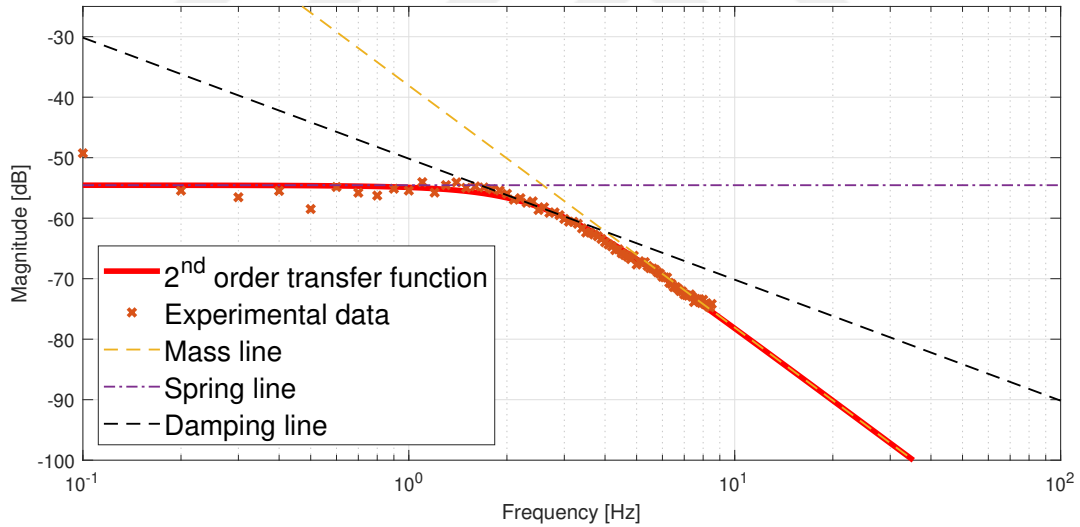


FIGURE 5.15: Experimental system identification of the inerter module with a relatively low inertia flywheel and variable negative stiffness module under compression

Parameters of the inerter with medium inertance and variable negative stiffness module is shown in Table 5.8. We have good match rate R^2 of 98% between the

estimated transfer function and the experimental data. System's cumulative mass characteristics is increased to 80.19 kg with a natural frequency of 0.4106 Hz .

Property	Description	Unit	Value
$M_{ee} + m$	Sum of mass of end-effector and Inertance	kg	80.19
m_f	Mass of flywheel	kg	0.183
b_d	Cumulative damping	Ns/m	322
$k_s + k_n$	Cumulative stiffness of compression spring and buckled beam	N/mm	0.5338
R^2	R-squared value	%	98
RMSE	Root mean squared error	1	0.8744
ω_n	Natural frequency	Hz	0.4106

TABLE 5.8: Inerter with medium level inertia flywheel and variable negative stiffness module characteristics under compression

Mathematical model given in Eqn. 4.21 estimates 65.6 kg inertance for the inerter module. Characterization results given in Table 5.7 present a good match between mathematical model and experimental results. Stiffness of the system was reduced by 6.96 N/mm due to the axial compression. Moreover, effective mass was increased (79.39 kg) by a relatively similar amount of modeled inertance.

High inertance

Furthermore, a relatively high inertia flywheel is rigidly attached to the nut of the ball-screw inerter. The experimental data and estimated transfer function as magnitude plot is shown in Figure 5.16. The characteristics of the system is given in Table 5.8. Estimation of the transfer function has a good match rate of R^2 of 95%. Displayed mass characteristics of the system is increased because of the higher inertia flywheel employment in the ball-screw inerter. In this configuration, the natural frequency of the device is estimated as 0.3949 Hz , which may be reduced up to 0.3667 Hz during load carriage of 25 kg. Increasing flywheel's inertia resulted in 76.61 kg increase in displayed inertance with 0.1 kg increase in mass of the device.

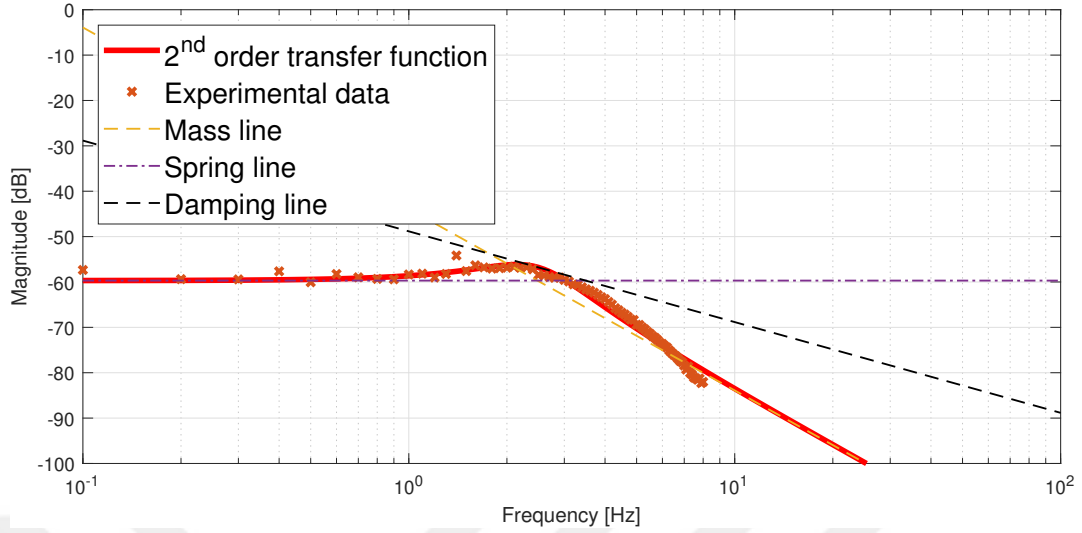


FIGURE 5.16: Experimental system identification of the inerter module with a relatively high inertia flywheel and variable negative stiffness module under compression

Mathematical model given in Eqn. 4.21 estimates 143.4 kg inertance for the inerter module. Moreover, 0.8343 N/mm stiffness is expected by the model given in Eqn. 4.15. Characterization result given in Table 5.7 presents a good match between mathematical models and experimental results. Stiffness of the system was reduced by 6.52 N/mm, and additional effective mass was increased by a relatively similar amount of modeled inertance.

Property	Description	Unit	Value
$M_{ee} + m$	Sum of mass of end-effector and Inertance	kg	156.8
m_f	Mass of flywheel	kg	0.283
b_d	Cumulative damping	Ns/m	276.7
$k_s + k_n$	Cumulative stiffness of compression spring and buckled beam	N/mm	0.9655
R^2	R-squared value	%	95
RMSE	Root mean squared error	1	1.317
ω_n	Natural frequency	Hz	0.3949

TABLE 5.9: Inerter with high level inertia flywheel and variable negative stiffness module characteristics under compression

5.2.4 Discussion

In this chapter, the stiffness and inertance modules are examined for three different stiffness and inertance levels. To evaluate the variable stiffness module, the inerter module is detached. In Figure 5.17, the experimental results for three different axial loading conditions of the variable stiffness module are represented.

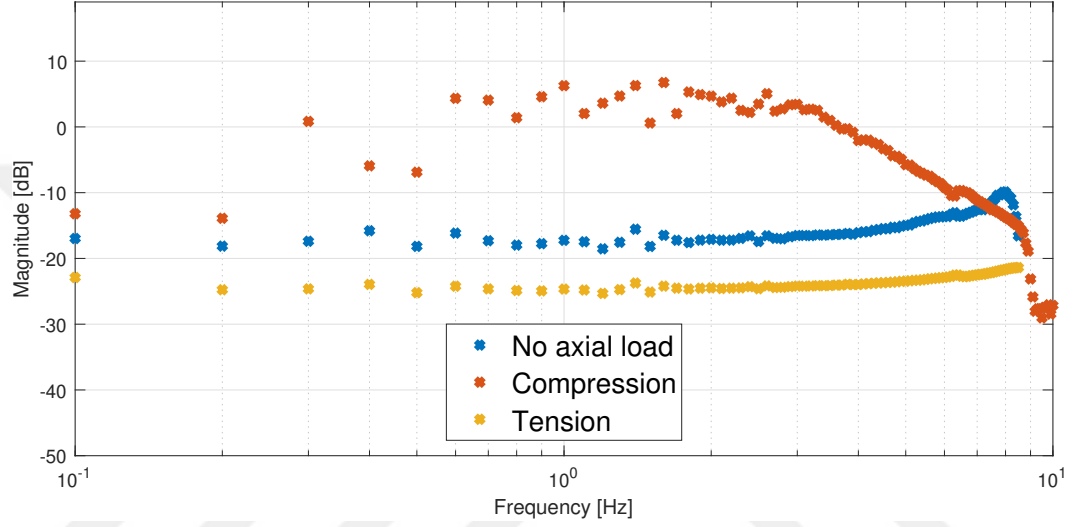


FIGURE 5.17: Experimental system identification of the variable stiffness module with a three different configurations

The variable stiffness module is expected to display different stiffness characteristics depending on the axial loading. First, no axial loading mode is examined. Comparatively, tensile and compressive axial forces are applied to the system through its compression mechanism. In the tension mode, system's effective stiffness is increased. In contrast, the axial compression force results in a decrease in the system's stiffness. In Table 5.10, estimation match rates and observed natural frequency of the system are presented for variable stiffness module. We achieved a good match between our estimation and experimental results with R^2 values of 87, 96 and 84.

Using the model represented in Eqn. 4.15, we expected a reduction or increase in the effective stiffness as we applied different axial loads such as no axial load, compression and tension. The model expected 6.66 N/mm reduction during the

compression mode which fits the mean of experimental results, 6.78 N/mm reduction. We also have good matches during axial tension (16.37 N/mm model expectation, 16.46 identified) and no load cases (7.49 N/mm model expectation, 7.54 identified).

Configuration	ω_n [Hz]	R ²	RMSE
No axial loading	13.459	87	0.002096
Axial tension	19.639	96	0.000169
Axial Compression	4.61	84	0.02508

TABLE 5.10: Natural frequency estimations for variable stiffness module

Moreover a relatively low damping characteristics were reflected by the device at all conditions of axial loading. Mathematical model does not capture the damping characteristics of the module. This damping may be caused by mechanical properties of the parts used in the device.

Furthermore, estimation match rates and observed natural frequency of the system are presented for inerter-based variable stiffness module. We achieved a good match rate at least 92% for all configurations of the backpack. Similar to the standalone characterization of the backpack, there is a consistency between the mathematical model and experimental results. The effective mass of the mechanism was increased as we employ high inertia flywheel.

Configuration	ω_n [Hz]	R ²	RMSE
Low level inertance	1.6191	92	0.4742
Medium level inertance	0.4106	98	0.8744
High level inertance	0.3949	95	1.317

TABLE 5.11: Natural frequency estimations when both modules are employed

Inerter module provides additional inertia characteristics, regardless of its mass. As the inertia of flywheel increases, system shows higher inertial characteristics. This is beneficial during load carriage because system's natural frequency can be reduced with a low mass. In Figure 5.18, experimental results for three configurations of the inerter module are represented.

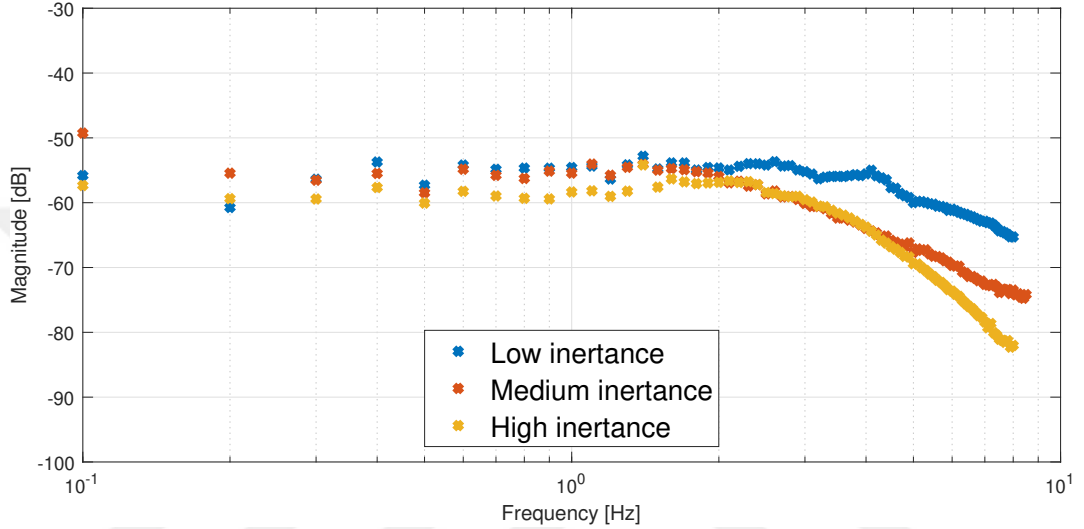


FIGURE 5.18: Experimental system identification of the inerter module with varying levels of inertance

Considering the standalone experimental characterization of the inerter we observed similar inertance values. For low inertance, mathematical model expected 23.4 kg inertance. During standalone experimentation, the inertance of the device was reported 21.71 kg and 24.72 kg, depending on the estimation model. However, the sum of mass of end-effector (0.8 kg) and inertance was estimated as 30.73 kg during inerter-based variable stiffness module characterization. The mismatch between model and results may be caused by unmodeled parameters of the experimental setup. The most important conclusion is that the inerter displays order of magnitude higher mass than its mass. For high inertance configuration, inerter has a mass of 1.014 kg, while the effective mass of backpack device is 148.15 ± 12.23 kg.

Finally, we experimentally characterized variable stiffness module under compression mode with 0.8998 ± 0.0943 N/mm stiffness and 4.779 ± 2.684 Ns/m damping. Furthermore, inertance of the device with high inertia flywheel was characterized as 148.15 ± 12.23 kg. Inerter module introduces a relatively high damping to the mechanism which was measured as 260.67 ± 25.69 Ns/m during high inertance configuration.



Chapter 6

Human Subject Experiments

This chapter presents the human subject experiment using the backpack modules to evaluate the metabolic cost and the interaction force reduction during the load carriage in terms of qualitative and quantitative metrics. The variable stiffness and the inerter modules are examined during the experiments along with the no load and suspension locked conditions. Device characteristics used during experiments are presented in Table 6.1 .

Parameter	Value (Unit)
Mass of backpack	5.2 kg
Pack load	18.3 kg
Stiffness	Variable stiffness condition: 0.8998 ± 0.0943 N/mm Inerter-based variable stiffness condition: 0.8998 ± 0.0943 N/mm
Inertance	Variable stiffness condition: 0 kg Inerter-based variable stiffness condition: 148.15 ± 12.23 kg
Damping	Variable stiffness condition: 4.779 ± 2.684 Ns/m Inerter-based variable stiffness condition: 260.67 ± 25.69 Ns/m

TABLE 6.1: Backpack properties

A backpack base (K-Tek KSHRN3 Stingray Harness (3rd Generation, Standard)) was used with additional straps and support mechanisms to attach the mechanism to the human torso.

6.1 Participants

Eight healthy adult male volunteers ($N=8$, age: 24 ± 2 years, weight= 84 ± 6 kg, height= 185 ± 5 cm) participated in the study. The experimental protocol was approved by IRB of Sabancı University, and informed consent was obtained from all of the participants before conducting the experiments. All subjects were older than 18 years old without any history of back injuries, neurological or physical impairments, cardiac problems, or orthopedic injuries that may affect characteristic motions of the torso. In order to reduce the possibility of musculoskeletal injuries, as an inclusion criteria, participants are required to weigh more than 75 kg, such that payload does not exceed 30% of their body weight.

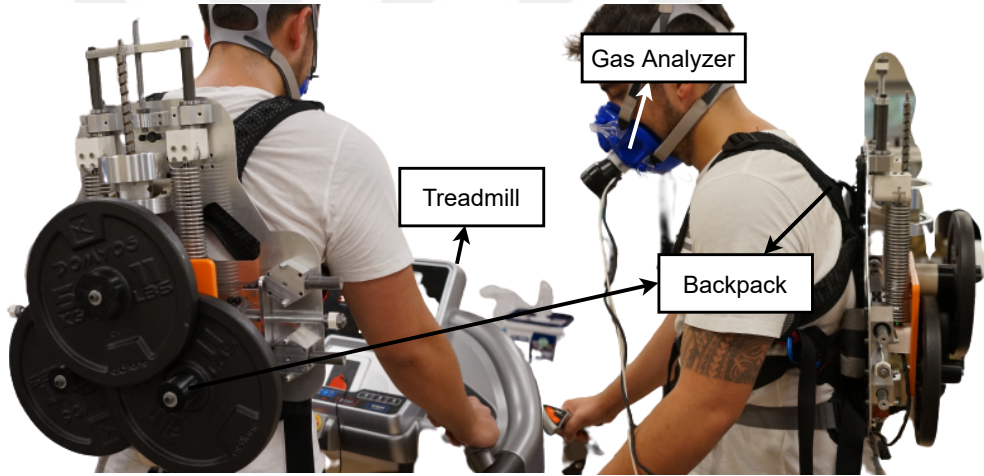


FIGURE 6.1: Experimental setup with a participant from back and side views

6.2 Task

Participants were asked to complete a standard six minute walking test protocol while carrying a 23.5 kg (18.3 kg payload + 5.2 kg device weight, and about $28\pm 1\%$ of their body weight) on their backpack. The experiments were conducted on a treadmill imposing a predetermined walking speed of 1.5 m/s. Participants were

asked to carry the predetermined load while their metabolic consumption was being recorded by a gas analyzer (Cortex Metalyzer 3B, Lehi, UT, USA) and the backpack was configured under suspension-locked (SL mode), lower stiffness (VS mode) and inerter-based low stiffness (IVS mode) suspension conditions. Moreover, the participants were not notified about the particular condition they were trying. They also were asked to fill a questionnaire and rank the conditions they tried.

6.3 Hypotheses

The following hypotheses are tested.

H1 - We hypothesized that the variable stiffness module (VS mode) configured for low stiffness can significantly reduce the metabolic cost of load carriage and peak accelerative force exerted on the user during walking in comparison to the suspension-locked case (SL mode).

H2 - We hypothesized that inerter-based variable stiffness suspension (IVS mode) configured for high inertance and low stiffness can significantly reduce the metabolic cost of load carriage during walking and peak accelerative force exerted on the user during walking in comparison to the suspension-locked case (SL Mode).

6.4 Experimental procedure

Participants were notified about the aims of the study and safety of the subjects were ensured before the experiments.

The experiment consisted of a session of 23.5 kg load carriage at 1.5 m/s (5.4 km/h) walking speed with the variable stiffness (VS mode) and inerter along with variable stiffness (IVS mode) modules, as presented in Figure 6.2.

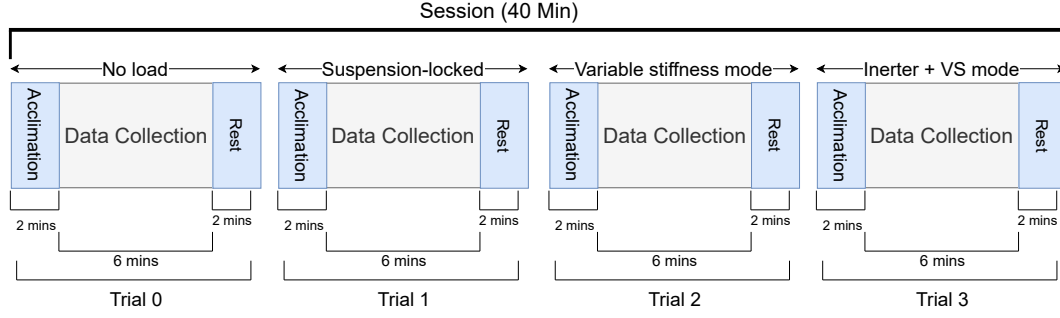


FIGURE 6.2: Four different tasks used in experiments: (Trial 0) no load (NL), (Trial 1) suspension-locked load carriage (SL mode), (Trial 2) suspended load with variable stiffness module (VS mode), and (Trial 3) inerter + variable stiffness modules (IVS mode)

In the VS mode, the inerter module was detached and a compression force were applied to the beams. For IVS mode, inertance was employed with variable stiffness module. For SL mode, suspension of the backpack was locked. During Trial 0, participant did not carry any load. In Trial 1, participants carried the device with a locked suspension (SL mode). The effectiveness of the variable stiffness and inerter modules were tested during Trials 2 and 3, respectively.

During the seating, all of four trials (T0-T3) were completed by the participants within a priorly generated randomized plan. Latin square method was used in order to prevent the selection bias and to avoid the accidental bias [75]. Figure 6.3, presents the experimental plan.



FIGURE 6.3: Experimental design consists of a single seating, where trials are administered in a randomized order

The order of trials varied for all participants. The complete table for the orders of trials are represented in Table 6.2.

Trial order	P1	P2	P3	P4	P5	P6	P7	P8
1 st	T3	T2	T0	T1	T2	T0	T1	T3
2 nd	T1	T3	T2	T0	T1	T2	T3	T0
3 rd	T2	T0	T1	T3	T3	T1	T0	T2
4 th	T0	T1	T3	T2	T0	T3	T2	T1

TABLE 6.2: Full list of the order of trials for each seating

Furthermore, in Table 6.3, the percentages of occurrence orders for all of the trials are represented for all seatings. All of the trials were repeated eight times by each participant in a randomized order for each seatings. The percentages are the same, so that effect of fatigue was equally distributed to all conditions.

Order	T0	T1	T2	T3
1 st	25%	25%	25%	25%
2 nd	25%	25%	25%	25%
3 rd	25%	25%	25%	25%
4 th	25%	25%	25%	25%

TABLE 6.3: Percentages of occurrence of trials. Each trial is repeat 24 times and their order is uniformly distributed by using latin square method

In addition to the randomization, participants were not informed about the case to compare and rank the experiment conditions.

6.5 Performance Metrics

Several variables were measured in order to evaluate the efficiency of the device as quantitative metrics. Besides the measured outputs of participant's effort during the tasks, they were asked to fill in a short questionnaire.

6.5.1 Quantitative Metrics

To evaluate performance quantitatively, metabolic energy expenditure of the volunteers during the experiments were measured by using a high resolution spirometry system. Participants wore a face mask during the experiments, which was connected to an automated analyser (Metalyzer 3B, Cortex Biophysik, Leipzig, Germany). The Metalyzer 3B's validity and reliability have been previously reported as compared to another verified gas analysis system (MetaMax I, Cortex Biophysik, Leipzig, Germany) [76]. The net metabolic power measurements were normalized by the body weight of each participant. Before each session, device were calibrated according to the manufacturer's recommendations. Due to the COVID-19 pandemic, a questionnaire on compliance with the precautions were filled out by the participants and the experiment technician. The COVID-19 related questionnaire is presented in Appendix B.

Parameter	Name	Symbol	Unit
Circulation	Heart Rate	HR	beats per minute (BPM)
	Maximum Heart Rate	HR_{max}	BPM
	Energy expenditure	EE	kcal/h
Gas Exchange	Relative O ₂ uptake	VO ₂ /kg	L/min/kg
	Relative CO ₂ output	VCO ₂ /kg	L/min/kg
Acceleration	Acceleration of human torso in vertical direction	a_h	m/s ²
	Acceleration of pack load in vertical direction	a_b	m/s ²
	Relative displacement between pack load and human torso	x_b	m

TABLE 6.4: Parameters measured during the experiments as quantitative metrics

Table 6.4 presents all the quantitative metrics used to evaluate the experiments. Circulation and gas exchange parameters were used by a software (MetaSoft Studio, Cortex Biophysik, Leipzig, Germany) to measure the energy expenditure of a participant. Energy expenditure (kcal/h) converted to metabolic power (W) and

normalized with participant's weight. Measurements of the last two minutes of walking were averaged for each condition to investigate metabolic cost and heart rate measurements.

Two IMUs (Vectornav VN-100 and CH Robotics UM7) were employed in the backpack to measure acceleration of torso and pack load in vertical direction, respectively. Moreover, a linear encoder (US Digital EM1) was used to measure the relative displacement between the load and the torso. The accelerative force due to the pack load is defined as

$$F_{\text{accelerative}} = m_{\text{pack}} (a_b - g) \quad (6.1)$$

where m_{pack} is the pack load (18.3 kg) and g is the gravitational acceleration. The phase angle between the torso and backpack was calculated using a digital signal processing algorithm based on Discrete Fourier Transform and maximum likelihood estimation of the signals' initial phases [77]. The phase difference between the torso and the pack load was calculated as the difference between the fundamental Discrete Fourier Transform phase spectrum values of the two measured signals [78].

All acceleration and position data were filtered using a zero-lag 6th order low-pass Butterworth filter with a 10 Hz cut-off frequency via MATLAB (ver. 2020a, MathWorks, MA, USA). The acceleration measurements were used to calculate the displacements of human body and pack load by double integrating the signals. We avoided drift using low pass filter on the acceleration measurements to remove noise. The vertical excursions of the torso and pack load were measured as the difference between the positive and negative peaks during load carriage.

Moreover, the percentage change between two conditions were derived by using the following formula:

$$[\%] = \frac{\text{Final value} - \text{Initial value}}{\text{Initial value}} \times 100 \quad (6.2)$$

6.5.2 Qualitative Metrics

In addition to the quantitative metrics, human oriented designs are usually evaluated with subjective evaluations based on a scale that measure the agreement between the physiological and perceived outcomes of the device [79]. A question set was answered by the participants after each trial to classify the device in terms of its usability.

The System Usability Scale (SUS), which is a technique to evaluate design's usability [80], is deployed along with a preference question. The original form of the SUS is presented in Table 6.5.

No.	Question
1	I think that I would like to use this system frequently.
2	I found the system unnecessarily complex.
3	I thought the system was easy to use.
4	I think that I would need the support of a technical person to be able to use this system.
5	I found the various functions in this system were well integrated.
6	I thought there was too much inconsistency in this system.
7	I would imagine that most people would learn to use this system very quickly.
8	I found the system very cumbersome to use.
9	I felt very confident using the system.
10	I needed to learn a lot of things before I could get going with this system.

TABLE 6.5: Original System Usability Scale

The flow activity of the SUS is presented in Figure 6.4.

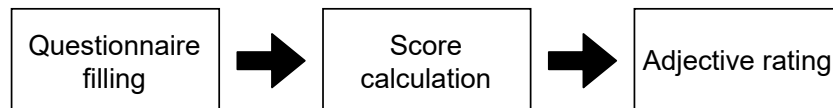


FIGURE 6.4: Flow activity of the System Usability Scale (SUS)

Score Calculation

After the questionnaire deployment, the answers are scored through the calculation stage. For each even numbered questions, take $5 - x$, and for odd numbers, $x - 1$,

where x is the Likert scale answer for the given question. Finally, combining the results and multiply it with 2.5 gives the overall score.

Adjective rating

In the adjective rating process, calculated score value will be used to classification of the system's usability. In Table 6.6, correspondences for the SUS scores are presented.

Grade	SUS score	Adjective rating
A	>80.3	Excellent
B	68-80.2	Good
C	67	Okay
D	51-66	Poor
F	<51	Awful

TABLE 6.6: System Usability Scale adjective rating

The questionnaire based on 5-point Likert scale is provided in Table 7.1. In addition to the questions based on SUS, volunteers ranked the configuration from the best to the worst.

6.6 Analysis of results

Experimental measurements are examined to investigate the effect of backpack configuration on the performance metrics. Box plots of important metrics were presented in order to compare the efficiency of each module. SPSS (ver. 28.0, IBM, NY, USA) was used to conduct the statistical analysis. Mauchly's test of sphericity was implemented on each dependent variable to validate the sphericity assumption of one-way repeated measures ANOVA. If the sphericity assumption was violated, Huynh-Feldt ($\epsilon > 0.75$) or Greenhouse-Geisser ($\epsilon < 0.75$) correction was applied.

One-way repeated measures ANOVA ($\alpha = 0.05$) was applied to test the different loading conditions on each metric. If a significant effect was revealed by ANOVA, pairwise comparisons with Fisher's least significant difference (LSD) test was used to detect the significant differences between the loading conditions.

6.7 Experimental Results

In this section, experimental results are presented.

6.7.1 Quantitative Metrics

Net metabolic power

Net metabolic power measurements are presented in Figure 6.5 with their mean and standard deviation.

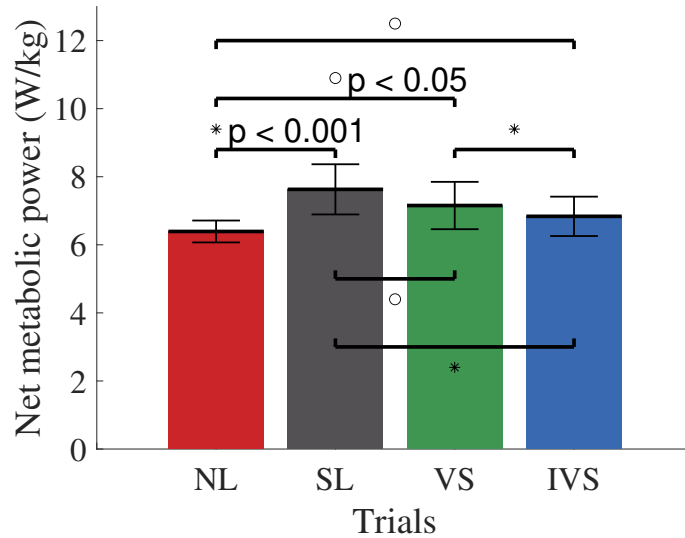


FIGURE 6.5: Mean and SD values of net metabolic power for trials. * and $\circ$ indicate a significant difference of $p < 0.001$ and $p < 0.05$ respectively (NL: No load, SL: suspension-locked, VS: variable stiffness mode, IVS: inerter+variable stiffness mode)

During no loading condition (NL), the gross metabolic cost is measured as $6.39 \pm 0.32 \text{ W/kg}$, which increases to $7.62 \pm 0.74 \text{ W/kg}$ when the backpack was worn by the participant with the SL mode. Proposed modules provided a relief with gross metabolic costs of $7.15 \pm 0.69 \text{ W/kg}$ with VS mode, and $6.83 \pm 0.57 \text{ W/kg}$ with IVS mode respectively. Mauchly's sphericity test indicated that the assumption of sphericity had not been violated ($\chi^2(5) = 6.755$, $p > 0.05$). Measurement data met the sphericity assumption of one-way repeated-measures ANOVA.

					Epsilon		
Within subject	Mauchly's W	Approx. Chi-Square	df	Sig.	Greenhouse-Geisser	Huynh-Feldt	Lower-bound
Loading condition	.307	6.755	5	.245	.630	.857	.33

TABLE 6.7: Mauchly's Test of Sphericity for normalized net metabolic power measurements

A one-way repeated measures ANOVA was performed to evaluate if the net metabolic power was different for the following conditions: NL mode, SL mode, VS mode and IVS mode. The results of one-way repeated measures ANOVA are represented in Table 6.8. The net metabolic power was statistically significantly different among different loading conditions ($F(3,21)=25.523$, $p < 0.001$ and $\eta^2 = 0.785$).

Source		Type III Sum of Squares	df	Mean Square	F	Sig.	Partial Eta Squared
Loading condition	Sphericity Assumed	6.528	3	2.176	25.523	<.001	.785
	Greenhouse-Geisser	6.528	1.891	3.451	25.523	<.001	.785
	Huynh-Feldt	6.528	2.571	2.540	25.523	<.001	.785
	Lower-bound	6.528	1.000	6.528	25.523	.001	.785
Error(Loading condition)	Sphericity Assumed	1.790	21	.085			
	Greenhouse-Geisser	1.790	13.240	.135			
	Huynh-Feldt	1.790	17.994	.100			
	Lower-bound	1.790	7.000	.256			

TABLE 6.8: Tests of Within-Subjects Effects for net metabolic power measurements

A post-hoc pairwise comparison indicates statistically significant differences between the pairs of loading conditions of NL-VS, NL-IVS, and SL-VS at $p < 0.05$ level. Moreover, the group differences between NL-SL, SL-IVS and VS-IVS were statistically significant at $p < 0.001$ level.

(I) Trial	(J) Trial	Mean Difference (I-J)	Std. Error	Sig	95% Confidence Interval for Difference	
					Lower Bound	Upper Bound
T0	T1	-1.237*	.187	<.001	-1.679	-.795
	T2	-.762*	.188	.005	-1.207	-.317
	T3	-.445*	.156	.024	-.812	-.077
T1	T0	1.237*	.187	<.001	.795	1.679
	T2	.475*	.115	.004	.204	.746
	T3	.793*	.121	<.001	.508	1.078
T2	T0	.762*	.188	.005	.317	1.207
	T1	-.475*	.115	.004	-.746	-.204
	T3	.318*	.074	.004	.142	.494
T3	T0	.445*	.156	.024	.077	.812
	T1	-.793*	.121	<.001	-1.078	-.508
	T2	-.318*	.074	.004	-.494	-.142

TABLE 6.9: Pairwise Comparisons for net metabolic power measurements

In Figure 6.6, the percentage change of net metabolic power is presented for VS and IVS conditions with respect to SL condition.

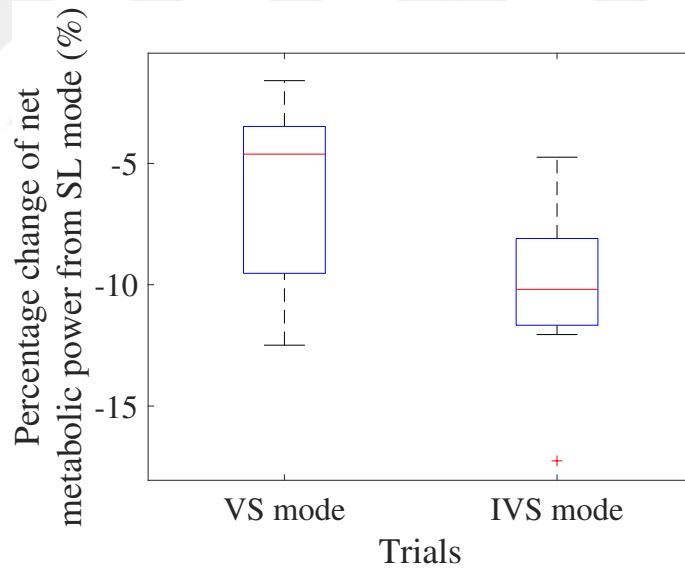


FIGURE 6.6: Means and standard deviations of percentage of net metabolic power reduction of each modules with respect to SL condition.

Comparing VS and IVS conditions with SL condition, the variable stiffness ($M = 6.16$, $SD = 3.90$) and inerter modules ($M = 10.23$, $SD = 3.67$) achieve significant reductions in the net metabolic power.

Normalized heart rate

Heart rate measurements were normalized with respect to no load condition measurements. Normalized heart rate during SL condition ($M = 1.0851$, $SD = 0.0652$) was not significantly different than VS ($M = 1.0660$, $SD = 0.0697$) and IVS ($M = 1.0617$, $SD = 0.0738$) conditions. Normalized heart rate measurements are presented in Figure 6.7 with their mean and standard deviation. However,

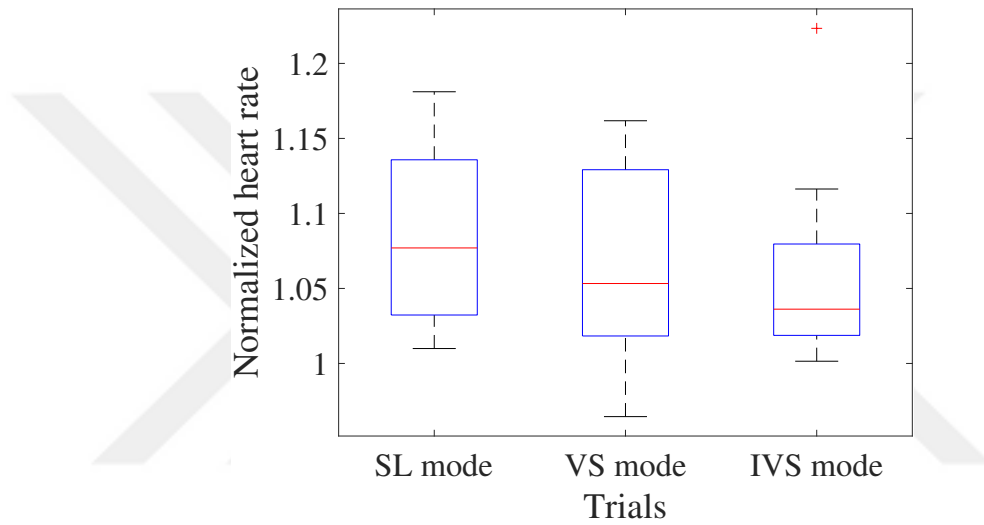


FIGURE 6.7: Mean and standard deviation values of normalized heart rate

Mauchly's sphericity indicated that the assumption of sphericity had been violated ($\chi^2(2) = 6.755$, $p > 0.05$) as presented in Table 6.10.

					Epsilon		
Within subject	Mauchly's W	Approx. Chi-Square	df	Sig.	Greenhouse-Geisser	Huynh-Feldt	Lower-bound
Loading condition	.143	11.673	2	.003	.538	.538	.5

TABLE 6.10: Mauchly's Test of Sphericity for normalized heart rate measurements

A one-way repeated measures ANOVA ($F(1.077,14)=0.368$, $p = 0.577$ and $\eta^2 = 0.05$) was performed with Greenhouse-Geisser correction ($\epsilon=.53$) revealed that measurements were not statistically significant.

Source		Type III Sum of Squares	df	Mean Square	F	Sig.	Partial Eta Squared
Loading condition	Sphericity Assumed	.002	2	.001	.368	.698	.050
	Greenhouse-Geisser	.002	1.077	.002	.368	.577	.050
	Huynh-Feldt	.002	1.117	.002	.368	.584	.050
	Lower-bound	.002	1.000	.002	.368	.563	.050
Error(Loading condition)	Sphericity Assumed	.047	14	.003			
	Greenhouse-Geisser	.047	7.539	.006			
	Huynh-Feldt	.047	7.819	.006			
	Lower-bound	.047	7.000	.007			

TABLE 6.11: Tests of Within-Subjects Effects for normalized heart rate measurements

Accelerative force

The peak accelerative force during the suspension locked condition was 85.50 ± 10.01 N. With variable stiffness module, peak accelerative force was 59.22 ± 7.39 N. As we employ the inerter module with variable stiffness module, the peak force reduced to 48.77 ± 11.08 N.

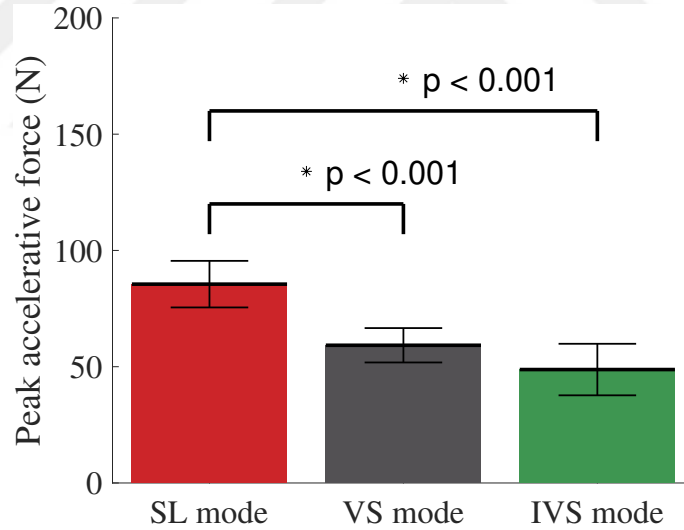


FIGURE 6.8: Mean and standard deviation values of peak accelerative force (N).
* indicates a significant difference ($p < 0.001$)

Mauchly's sphericity test indicated that the assumption of sphericity had not been violated ($\chi^2(2) = 0.361$, $p > 0.05$). Measurement data met sphericity assumption of one-way repeated-measures ANOVA.

Within subject	Mauchly's W	Approx. Chi-Square	df	Sig.	Epsilon		
					Greenhouse-Geisser	Huynh-Feldt	Lower-bound
Loading condition	.942	.361	2	.835	.945	1.00	.50

TABLE 6.12: Mauchly's Test of Sphericity for peak accelerative force measurements

A one-way repeated measures ANOVA was performed to evaluate if the peak accelerative force was different for the following conditions : SL mode, VS mode and IVS mode. Results of one-way repeated measures ANOVA are represented in Table 6.13. There was a significant effect of loading condition on the peak accelerative force ($F(2,14)=28.085$, $p < 0.001$ and $\eta^2 = 0.80$).

Source		Type III Sum of Squares	df	Mean Square	F	Sig.	Partial Eta Squared
Loading condition	Sphericity Assumed	5729.517	2	2864.758	28.085	<.001	.800
	Greenhouse-Geisser	5729.517	1.890	3032.252	28.085	<.001	.800
	Huynh-Feldt	5729.517	2.000	2864.758	28.085	<.001	.800
	Lower-bound	5729.517	1.000	5729.517	28.085	.001	.800
Error(Load condition)	Sphericity Assumed	1428.052	14	102.004			
	Greenhouse-Geisser	1428.052	13.227	107.968			
	Huynh-Feldt	1428.052	14.000	102.004			
	Lower-bound	1428.052	7.000	204.007			

TABLE 6.13: Tests of Within-Subjects Effects for peak accelerative force measurements

Post hoc pairwise comparison revealed that SL-VS and SL-IVS mode differences were statistically significant at $p < 0.001$ level. There was, however, no significant difference between VS and IVS modes ($p=0.075$).

(I) Trial	(J) Trial	Mean Difference (I-J)	Std. Error	Sig.	95% Confidence Interval for Difference	
					Lower Bound	Upper Bound
T1	T2	26.282*	4.518	<.001	15.599	36.965
	T3	36.726*	5.573	<.001	23.547	49.904
T2	T1	-26.282*	4.518	<.001	-36.965	-15.599
	T3	10.444	5.003	.075	-1.387	22.274
T3	T1	-36.726*	5.573	<.001	-49.904	-23.547
	T2	-10.444	5.003	.075	-22.274	1.387

TABLE 6.14: Pairwise Comparisons for peak accelerative force measurements

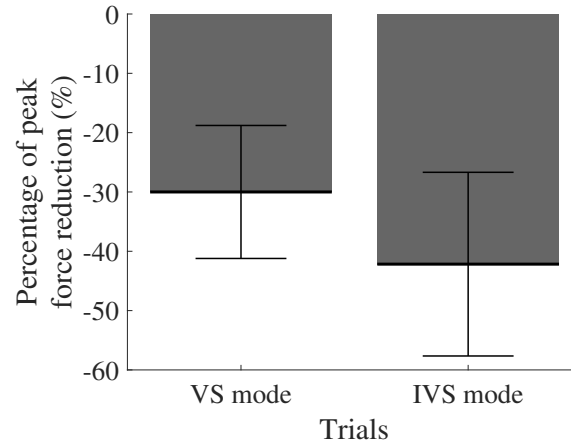


FIGURE 6.9: Mean and standard deviation values of percentage change of peak accelerative forces

VS and IVS conditions result in a significant reduction in the peak accelerative force exerted on the human. The dynamic force is reduced with the variable stiffness ($M = 30.00$, $SD = 11.21$) and inerter+variable stiffness ($M = 42.17$, $SD = 15.48$) modules, as it is represented in Figure 6.9.

Torso excursion

The vertical excursions of the torso are presented in Figure 6.10. All data are means with standard deviations for SL, VS and IVS conditions.

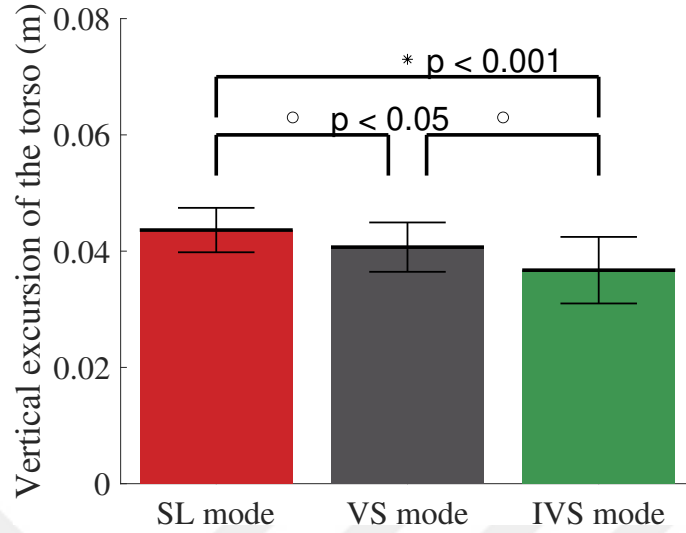


FIGURE 6.10: Mean and standard deviation values of vertical excursion of the torso (m) * and $\circ$ indicate a significant difference of $p < 0.001$ and $p < 0.05$ respectively

Mauchly's sphericity indicated that the assumption of sphericity had not been violated ($\chi^2(2) = 0.654$, $p > 0.05$). Measurement data met sphericity assumption of one-way repeated-measures ANOVA.

Within subject	Mauchly's W	Approx. Chi-Square	df	Sig.	Epsilon		
					Greenhouse-Geisser	Huynh-Feldt	Lower-bound
Loading condition	.654	2.460	2	.292	.748	0.906	0.5

TABLE 6.15: Mauchly's Test of Sphericity for peak-peak torso excursion

The results of one-way repeated measures ANOVA ($F(2,14)=37.611$, $p < 0.001$ and $\eta^2=0.843$) are presented in Table 6.16.

Source		Type III Sum of Squares	df	Mean Square	F	Sig.	Partial Eta Squared
Loading condition	Sphericity Assumed	191.936	2	95.968	37.611	<.001	.843
	Greenhouse-Geisser	191.936	1.497	128.246	37.611	<.001	.843
	Huynh-Feldt	191.936	1.812	105.915	37.611	<.001	.843
	Lower-bound	191.936	1.000	191.936	37.611	<.001	.843
Error>Loading condition)	Sphericity Assumed	35.723	14	2.552			
	Greenhouse-Geisser	35.723	10.476	3.410			
	Huynh-Feldt	35.723	12.685	2.816			
	Lower-bound	35.723	7.000	5.103			

TABLE 6.16: Tests of Within-Subjects Effects for peak-peak torso excursion

Post-hoc analysis of pairwise comparisons revealed that during suspension locked condition ($M = 0.0436$, $SD = 0.0038$), the vertical excursion of the torso is significantly higher than VS ($M = 0.0407$, $SD = 0.0042$) at $p < 0.05$ level. Post-hoc analysis also indicates statistically significant differences between the pairs of loading conditions of SL and IVS ($M = 0.0367$, $SD = 0.0057$) modes at $p < 0.001$ level. Moreover, during VS condition, the vertical excursion of the torso is significantly higher than IVS condition at $p < 0.05$ level. Pairwise comparison table is represented in the Table 6.17.

(I) Trial	(J) Trial	Mean Difference (I-J)	Std. Error	Sig	95% Confidence Interval for Difference	
					Lower Bound	Upper Bound
T1	T2	2.937*	.581	.001	1.562	4.311
	T3	6.902*	.988	<.001	4.566	9.237
T2	T1	-2.937*	.581	.001	-4.311	-1.562
	T3	3.965*	.775	.001	2.133	5.797
T3	T1	-6.902*	.988	<.001	-9.237	-4.566
	T2	-3.965*	.775	.001	-5.797	-2.133

TABLE 6.17: Pairwise Comparisons for peak-peak torso excursion measurements

Pack load excursion

The vertical excursions of the pack load are presented in Figure 6.11. All data are means with standard deviations for loaded conditions of SL, VS and IVS.

For suspension locked condition ($M = 0.0362$, $SD = 0.0059$) the vertical excursion of the torso was higher than VS ($M = 0.0322$, $SD = 0.0027$) and IVS ($M = 0.0289$, $SD = 0.0025$) modes. Moreover, Mauchly's sphericity indicated that the assumption of sphericity had been violated ($\chi^2(5) = 11.668$, $p < 0.05$). Hence, Greenhouse-Geisser correction ($\epsilon = 0.53$) was applied to meet sphericity assumption of one-way repeated measures ANOVA.

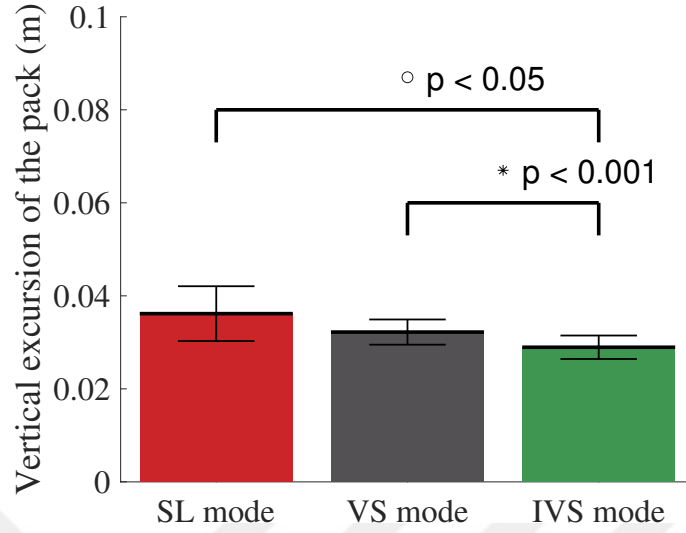


FIGURE 6.11: Mean and standard deviation values of vertical excursion of the pack load (m) for the trials of loading conditions of suspension locked (SL) mode, variable stiffness (VS) mode and inerter-variable stiffness (IVS) mode. * and $\circ$ indicate a significant difference of $p < 0.001$ and $p < 0.05$ respectively

					Epsilon		
Within subject	Mauchly's W	Approx. Chi-Square	df	Sig.	Greenhouse-Geisser	Huynh-Feldt	Lower-bound
Loading condition	.143	11.668	2	.003	.539	.559	.5

TABLE 6.18: Mauchly's Test of Sphericity for peak-peak pack load excursion measurements

Results of one-way repeated measures ANOVA ($F(1.077, 7.539) = 13.978$, $p < 0.05$ and $\eta^2 = 0.666$) for the vertical excursion of the pack load measurements are presented in Table 6.19.

Source		Type III Sum of Squares	df	Mean Square	F	Sig.	Partial Eta Squared
Loading condition	Sphericity Assumed	209.588	2	104.794	13.978	<.001	.666
	Greenhouse-Geisser	209.588	1.077	194.598	13.978	.006	.666
	Huynh-Feldt	209.588	1.117	187.627	13.978	.005	.666
	Lower-bound	209.588	1.000	209.588	13.978	.007	.666
Error(Load condition)	Sphericity Assumed	104.959	14	7.497			
	Greenhouse-Geisser	104.959	7.539	13.922			
	Huynh-Feldt	104.959	7.819	13.423			
	Lower-bound	104.959	7.000	14.994			

TABLE 6.19: Tests of Within-Subjects Effects for peak-peak pack load excursion measurements

Post-hoc analysis indicates statistically significant differences between the pairs of

loading conditions of SL and IVS modes at $p < 0.05$ level. Moreover, there is statistically significant difference between VS and IVS conditions at $p < 0.001$ level.

(I) Trial	(J) Trial	Mean Difference (I-J)	Std. Error	Sig.	95% Confidence Interval for Difference	
					Lower Bound	Upper Bound
T1	T2	3.964	1.782	.062	-.251	8.178
	T3	7.227*	1.489	.002	3.707	10.748
T2	T1	-3.964	1.782	.062	-8.178	.251
	T3	3.264*	.480	<.001	2.129	4.398
T3	T1	-7.227*	1.489	.002	-10.748	-3.707
	T2	-3.264*	.480	<.001	-4.398	-2.129

TABLE 6.20: Pairwise Comparisons for peak-peak pack load excursion measurements

Relative displacement

The measurements of the peak-peak relative displacement between pack load and human torso are presented in Figure 6.12. All data are means with standard deviations for loading conditions of SL, VS and IVS.

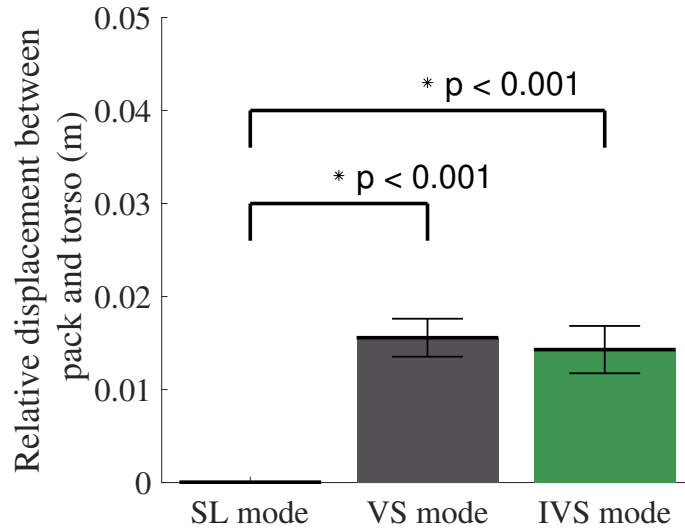


FIGURE 6.12: Mean and standard deviation values of the relative displacement between pack and torso (m). * indicates a significant difference ($p < 0.001$) between the conditions.

Mauchly's sphericity test indicated that the assumption of sphericity had not been violated ($\chi^2(2) = .474, p > 0.05$).

					Epsilon		
Within subject	Mauchly's W	Approx. Chi-Square	df	Sig.	Greenhouse-Geisser	Huynh-Feldt	Lower-bound
Loading condition	.924	.474	2	.789	.929	1	.5

TABLE 6.21: Mauchly's Test of Sphericity for peak-peak relative displacement measurements

A one-way repeated measures ANOVA is presented in Table 6.22. Results one-way repeated measure ANOVA ($F(2,14)=207.708, p<0.001$ and $\eta^2=0.967$) revealed that relative displacement was statistically significantly different among different loading conditions.

Source		Type III Sum of Squares	df	Mean Square	F	Sig.	Partial Eta Squared
Loading condition	Sphericity Assumed	1189.450	2	594.725	207.708	<..001	.967
	Greenhouse-Geisser	1189.450	1.859	639.918	207.708	<..001	.967
	Huynh-Feldt	1189.450	2.000	594.725	207.708	<..001	.967
	Lower-bound	1189.450	1.000	1189.450	207.708	<..001	.967
Error(Load condition)	Sphericity Assumed	40.086	14	2.863			
	Greenhouse-Geisser	40.086	13.011	3.081			
	Huynh-Feldt	40.086	14.000	2.863			
	Lower-bound	40.086	7.000	5.727			

TABLE 6.22: Tests of Within-Subjects Effects for peak-peak relative displacement measurements

The pairwise comparison table is presented in the Table 6.23.

(I) Trial	(J) Trial	Mean Difference (I-J)	Std. Error	Sig.	95% Confidence Interval for Difference	
					Lower Bound	Upper Bound
T1	T2	-15.533*	.721	<..001	-17.236	-13.829
	T3	-14.253*	.895	<..001	-16.368	-12.137
T2	T1	15.533*	.721	<..001	13.829	17.236
	T3	1.280	.910	.202	-.872	3.432
T3	T1	14.253*	.895	<..001	12.137	16.368
	T2	-1.280	.910	.202	-3.432	.872

TABLE 6.23: Pairwise Comparisons for peak-peak relative displacement measurements

The post-hoc analysis revealed statistically significant differences between the pairs of loading conditions of SL ($M = 0.00004, SD = 0.00002$) and VS ($M = 0.01557,$

$SD = 0.00204$) conditions at $p < 0.001$ level. The difference between SL condition to IVS ($M = 0.01429$, $SD = 0.00254$) condition was also statistically significant at the same level $p < 0.001$. VS and IVS conditions were not significantly different according to the post-hoc analysis ($p=0.202$).

Phase difference

The phase difference between pack load and human torso are presented in the Figure 6.13.

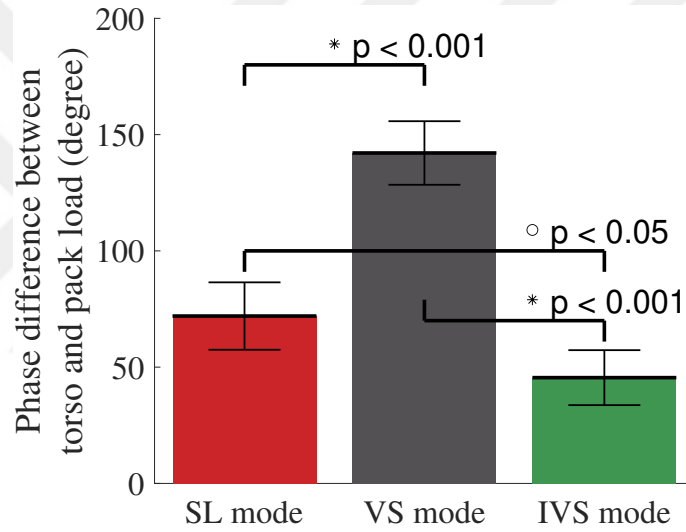


FIGURE 6.13: Mean and standard deviation values phase difference between pack and torso (degree). * and o indicate a significant difference of $p < 0.001$ and $p < 0.05$ respectively

All data are means with standard deviations for loaded conditions of SL ($M = 71.97$, $SD = 14.49$), VS ($M = 142.09$, $SD = 13.67$) and IVS ($M = 45.48$, $SD = 11.81$). Mauchly's sphericity indicated that the assumption of sphericity had not been violated ($\chi^2(2) = 2.573$, $p > 0.05$).

					Epsilon		
Within subject	Mauchly's W	Approx. Chi-Square	df	Sig.	Greenhouse-Geisser	Huynh-Feldt	Lower-bound
Loading condition	.651	2.573	2	.276	.741	.894	.5

TABLE 6.24: Mauchly's Test of Sphericity for phase difference measurements

Results of one-way repeated measures ANOVA ($F(2,14)=92.839$, $p<0.001$ and $\eta^2=0.93$) are presented in Table 6.25.

Source		Type III Sum of Squares	df	Mean Square	F	Sig.	Partial Eta Squared
Loading condition	Sphericity Assumed	39872.106	2	19936.053	92.839	<.001	.930
	Greenhouse-Geisser	39872.106	1.483	26888.267	92.839	<.001	.930
	Huynh-Feldt	39872.106	1.788	22303.359	92.839	<.001	.930
	Lower-bound	39872.106	1.000	39872.106	92.839	<.001	.930
Error(Loadng condition)	Sphericity Assumed	3006.323	14	214.737			
	Greenhouse-Geisser	3006.323	10.380	289.622			
	Huynh-Feldt	3006.323	12.514	240.236			
	Lower-bound	3006.323	7.000	429.475			

TABLE 6.25: Tests of Within-Subjects Effects for phase difference measurements

Post-hoc analysis indicates statistical significance at $p < 0.001$ level between SL-VS and VS-IVS conditions. Moreover, post-hoc analyses also revealed that the decrease from SL condition to IVS condition was statistically significant at $p < 0.05$ level. The pairwise comparisons for phase difference measurements are presented in the Table 6.29

(I) Trial	(J) Trial	Mean Difference (I-J)	Std. Error	Sig.	95% Confidence Interval for Difference	
					Lower Bound	Upper Bound
T1	T2	-70.120*	9.240	<.001	-91.970	-48.270
	T3	26.490*	6.136	.003	11.981	41.000
T2	T1	70.120*	9.240	<.001	48.270	91.970
	T3	96.610*	6.166	<.001	82.030	111.190
T3	T1	-26.490*	6.136	.003	-41.000	-11.981
	T2	-96.610*	6.166	<.001	-111.190	-82.030

TABLE 6.26: Pairwise Comparisons for phase difference measurements

The torso and pack load displacements and the relative displacement between them during a gait cycle for SL condition are presented in Figure 6.14. The torso and load displacement follow similar motion profiles since the pack is not allowed to swing. However, the phase difference between torso and pack load results in a shifted motion trajectory for human torso, compared with vertical excursion of pack load. However, the device was not rigidly attached to the wearer's body.

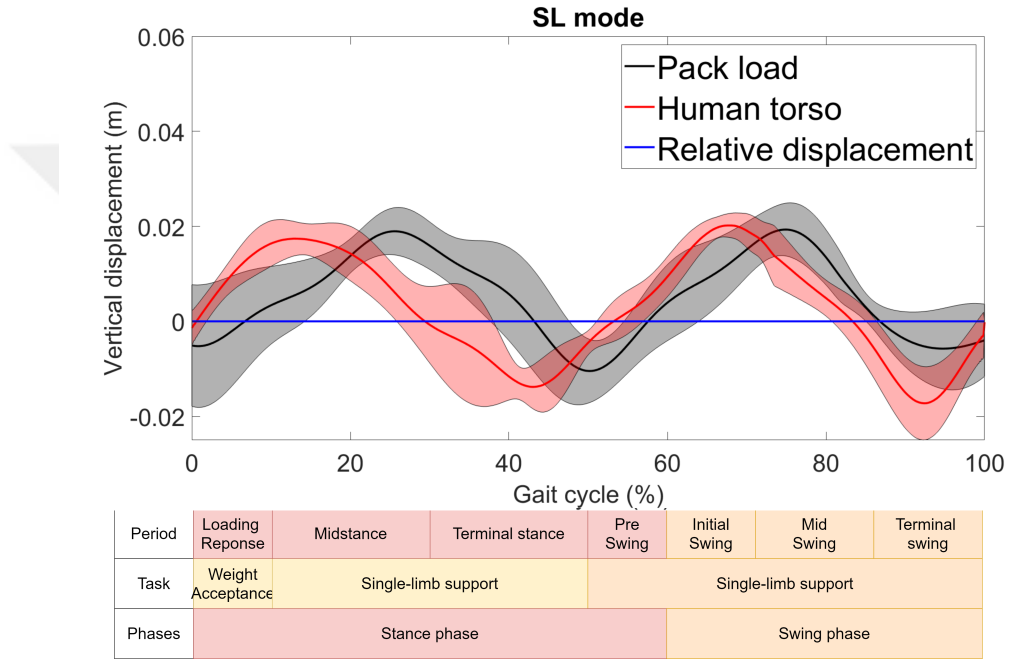


FIGURE 6.14: Mean and standard deviation values of human torso, pack load and relative displacement between torso and pack load during SL mode.

During VS condition, there was a phase difference ($142.09 \pm 13.67^\circ$) between the pack load and human torso as presented in Figure 6.15. The phase angle was significantly higher than the suspension locked case (-49.35% mean difference, $p < 0.001$). Vertical excursion of the load is reduced since the load is enabled to swing as the torso moves in the vertical direction. However, reduced vertical excursion of the torso during VS mode was increased compared with the SL mode (-6.65% mean difference, $p < 0.05$).

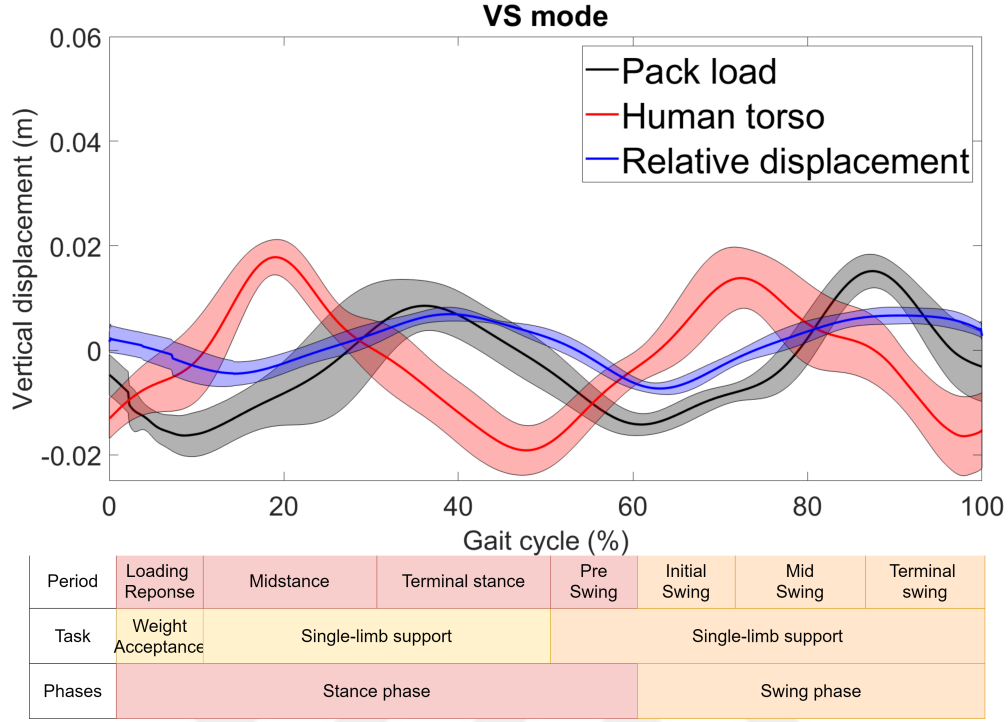


FIGURE 6.15: Mean and standard deviation values of human torso, pack load and relative displacement between torso and pack load during VS mode.

In Figure 6.16, human torso and pack load displacement and relative displacement trajectories during a gait cycle during IVS condition are presented. The phase difference ($M = 45.48$, $SD = 11.81$) was significantly less than SL ($p < 0.05$) and VS ($p < 0.001$) conditions. Inerter-based variable stiffness module reduced vertical excursion of the pack and torso when compared with the other conditions, due to the high damping the inerter module introduces to the system.

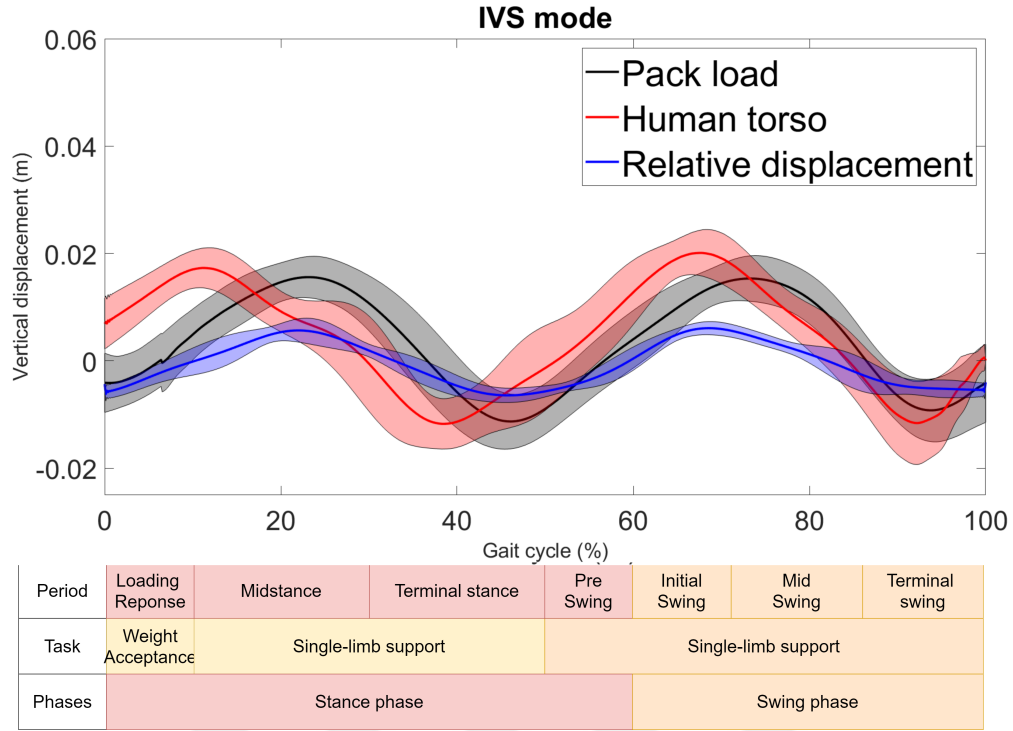


FIGURE 6.16: Mean and standard deviation values of human torso, pack load and relative displacement between torso and pack load during IVS mode.

Finally, the accelerative force trajectories during a gait cycle are presented with mean and standard deviations for SL, VS and IVS conditions in Figure 6.17. The peak accelerative force was reduced by $30 \pm 11.21\%$ during the VS condition. The accelerative forces during the SL condition was significantly higher ($42.17 \pm 15.48\%$, $p < 0.001$) than the IVS condition. The VS condition caused higher accelerative forces during loading response period with respect to the IVS condition. The pack load exerts higher force at suspension-locked case compared to the VS and IVS conditions during all phases of a gait cycle.

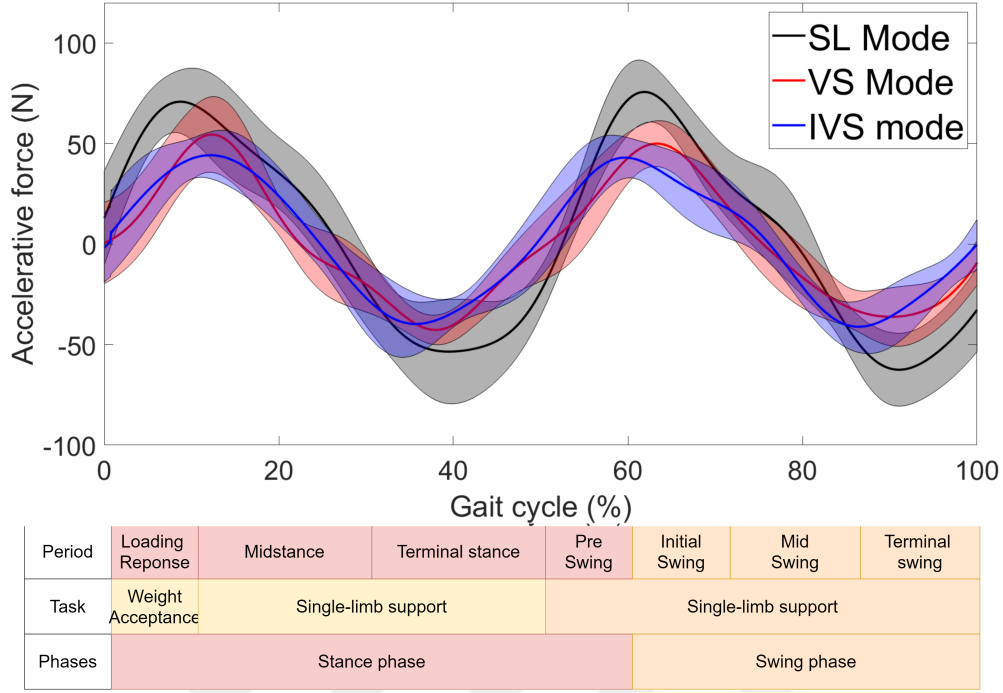


FIGURE 6.17: Mean and standard deviation values of accelerative forces due to pack load, during trials.

One can define force on the suspensions using the relative displacements measurements $(x-y, \dot{x}-\dot{y}, \ddot{x}-\ddot{y})$, identified system parameters (m, b, k) and mathematical model using the schematic given in Figure 3.1

$$F_{\text{suspension}} = m(\ddot{x} - \ddot{y}) + b(\dot{x} - \dot{y}) + k(x - y) \quad (6.3)$$

Identified parameters ($k = 0.8998 \text{ N/mm}$, $b = 4.779 \text{ Ns/m}$) were used along with the corresponding relative displacement measurements to calculate $F_{\text{suspension}}$ presented in Figure 6.18. There is a mismatch between the mathematical model and identified parameters during terminal stance and terminal swing of the gait cycle.

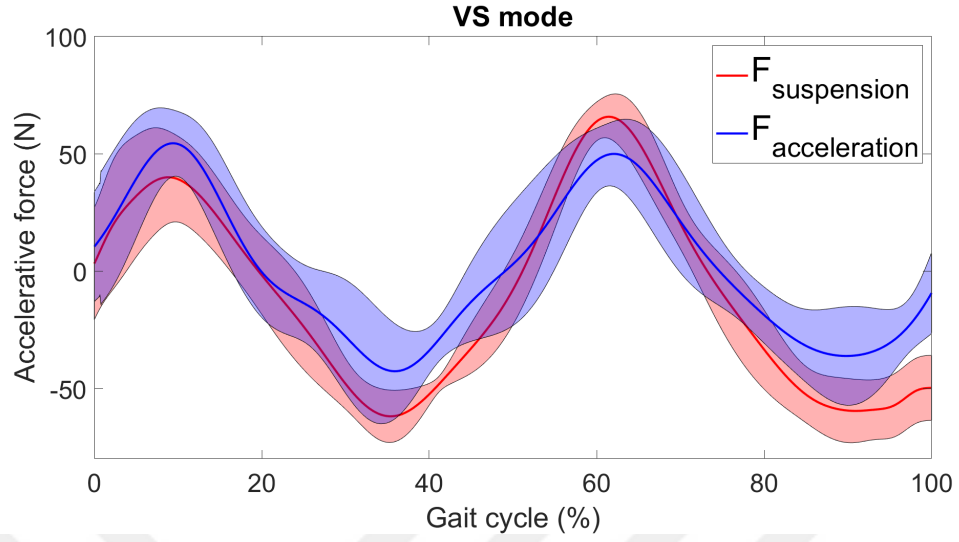


FIGURE 6.18: Mean and standard deviation values of accelerative forces due to pack load and $F_{\text{suspension}}$ during VS condition.

Identified parameters ($k = 0.8998 \text{ N/mm}$, $b = 260.67 \text{ Ns/m}$, $m = 148.15 \text{ kg}$) were used along with the corresponding relative displacement measurements to calculate $F_{\text{suspension}}$ presented in Figure 6.19. There is a match between the mathematical model and identified parameters during the gait cycle.

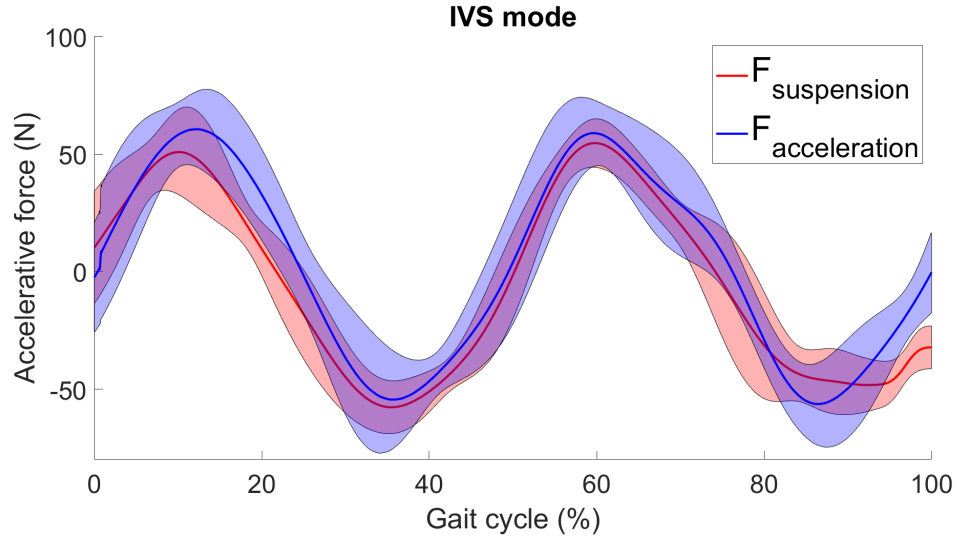


FIGURE 6.19: Mean and standard deviation values of accelerative forces due to pack load and $F_{\text{suspension}}$ during IVS condition.

6.7.2 Qualitative Metrics

By using the System Usability Scale, adjective rating of SL, VS and IVS conditions are calculated. Adjective rating scores of each condition are represented in Figure 6.20. Mauchly's sphericity test indicated that the assumption of sphericity had not been violated ($\chi^2(2) = .981, p > 0.05$).

					Epsilon		
Within subject	Mauchly's W	Approx. Chi-Square	df	Sig.	Greenhouse-Geisser	Huynh-Feldt	Lower-bound
Loading condition	.981	.114	2	.945	.982	1.00	.5

TABLE 6.27: Mauchly's Test of Sphericity for adjective rating scores

Adjective ratings were calculated as 63.75 ± 2.31 for SL, 73.12 ± 3.95 for VS and 73.43 ± 3.52 for IVS. Device was rated as "poor" during SL condition. However participants rated device as "good" during VS and IVS conditions.

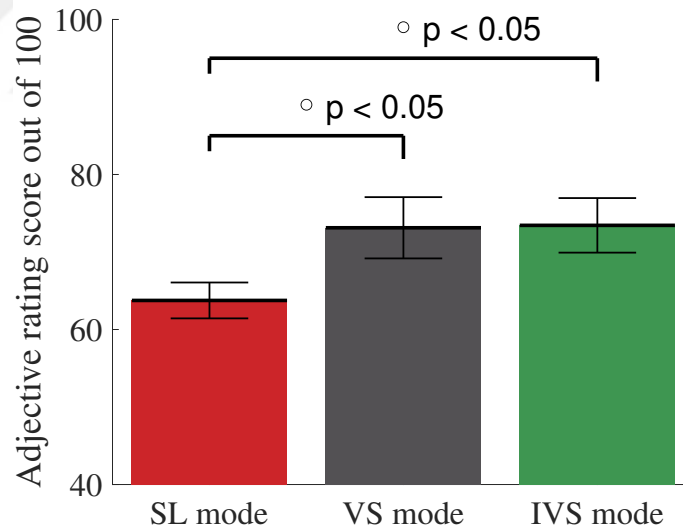


FIGURE 6.20: Adjective rating scores of trials with load carriage according to the SUS. $\circ$ indicates a significant difference of $p < 0.05$.

Results of one-way repeated measures ANOVA ($F(2,14)=17.472, p<0.001$ and $\eta^2=0.714$) are presented in Table 6.28.

Source		Type III Sum of Squares	df	Mean Square	F	Sig.	Partial Eta Squared
Loading condition	Sphericity Assumed	484.896	2	242.448	17.472	<.001	.714
	Greenhouse-Geisser	484.896	1.963	247.008	17.472	<.001	.714
	Huynh-Feldt	484.896	2.000	242.448	17.472	<.001	.714
	Lower-bound	484.896	1.000	484.896	17.472	.004	.714
Error(Loadng condition)	Sphericity Assumed	194.271	14	13.876			
	Greenhouse-Geisser	194.271	13.742	14.138			
	Huynh-Feldt	194.271	14.000	13.876			
	Lower-bound	194.271	7.000	27.753			

TABLE 6.28: Tests of Within-Subjects Effects for adjective rating scores

Post-hoc analyses revealed that the adjective rating score change from SL to VS and IVS conditions are statistically significant at $p < 0.001$ level. However, there was no significant difference between adjective rating scores of VS and IVS modes ($p=0.879$). Pairwise comparison table is presented in Table 6.29.

Participants were also asked to rank the trials they had tried, from best to worst (1 to 3), VS and IVS conditions were not ranked third by any participant. The SL condition was reported as the worst by all participants ($M = 3$, $SD = 0$). The IVS condition ($M = 1.37$, $SD = 0.51$) was evaluated to be better than VS condition ($M = 1.62$, $SD = 0.51$).

(I) Trial	(J) Trial	Mean Difference (I-J)	Std. Error	Sig.	95% Confidence Interval for Difference	
					Lower Bound	Upper Bound
T1	T2	-9.375*	1.752	.001	-13.518	-5.232
	T3	-9.687*	1.856	.001	-14.077	-5.298
T2	T1	9.375*	1.752	.001	5.232	13.518
	T3	-.312	1.973	.879	-4.978	4.353
T3	T1	9.688*	1.856	.001	5.298	14.077
	T2	.313	1.973	.879	-4.353	4.978

TABLE 6.29: Pairwise Comparisons for adjective rating scores

6.8 Discussion

To evaluate the performance of the backpack modules, 6 qualitative 2 quantitative metrics were investigated. The experimental results validate both of the hypotheses. Metabolic cost reduction is the key aspect for improved suspended-load backpack devices as it is the measure of energy expenditure during the load carriage. A reduction will correspond to reduced risk for medical issues and increased duration of load carriage and decreased fatigue. In Table 6.30, our backpack conditions are compared with the existing ones in literature.

Backpack	Inertance	Stiffness	Damping	Walking speed	Pack load	Metabolic cost reduction
Rome <i>et. al.</i> [14]	$\emptyset$	$\sim 0.53\text{N/mm}^\dagger$	$\sim 100\text{Ns/m}^\dagger$	5.6 km/h	27 kg	6.5%
Foissac <i>et. al.</i> [48]	$\emptyset$	$5.14 \pm 0.159\text{N/mm}$	$10.4 \pm 2.1\text{Ns/m}$	4.5 km/h	18 kg	3.8%
Huang <i>et. al.</i> [81]	$\emptyset$	0.89 N/mm	48 Ns/m	5 km/h	15 kg	15.2%
VS condition	$\emptyset$	$0.8998 \pm 0.0943\text{N/mm}$	$4.779 \pm 2.684\text{Ns/m}$	5.4 km/h	23.5 kg	6.16%
IVS condition	$148.15 \pm 12.23\text{kg}$	$0.8998 \pm 0.0943\text{N/mm}$	$260.67 \pm 25.69\text{Ns/m}$	5.4 km/h	23.5 kg	10.23%

TABLE 6.30: Comparison of our results with existing suspended load backpack designs. $\dagger$ retrieved from [1].

The aim of the study was to validate the metabolic cost reduction provided by the proposed backpack device. It was hypothesized that metabolic cost of walking will be decreased as the pack load is allowed to oscillate through a passive suspension.

The metabolic cost was decreased by 6.16% during the VS condition and 10.23% during the IVS condition when compared to the suspension locked condition. At 5.4 km/h, the VS condition provides comparable improvement compared to the Rome's backpack at 5.6 km/h (6.5%) [14]. Note that as the walking speed increases, the suspended backpacks promise improved performance since the excitation frequency is increased. VS condition provides better performance than Foissac's study at 4.5 km/h (3.8%) [48]. Lastly, Huang *et. al.* [81] reported 15.2% metabolic cost reduction with a suspended-load backpack at 5 km/h. This variation in findings could be due to the backpack's different impedance characteristics, experimental methods, as well as

the properties of frame straps, belt, etc. that attaches the mechanism to the human torso.

Previous research has shown that a suspended-load backpack decreases user's metabolic cost if its resonance frequency is significantly lower than the human walking frequency [1, 47, 48]. As we employed the inerter module along with the variable stiffness module, during the IVS condition, the metabolic cost were significantly less the VS and SL conditions. Employing inerter had decreased the resonance frequency of the pack, which resulted in decreased metabolic cost, accelerative force, pack and load excursions in the vertical direction.

The highest phase differences were measured during the VS condition, and the lowest were during IVS condition. As expected from the mathematical model (see Figure 3.5), the phase shift is maximized for low stiffness-damping configuration of the backpack. Inerter employment resulted in decreased phase shift and oscillation force amplitude as this module introduces relatively high amount of damping to the system. Foissac *et. al.* [48], observed lower phase difference for suspended-load backpacks when compared with rigid backpacks. However, Rome *et. al.* [14], estimated an out of phase (180°) movement between pack load and human torso in their design.

The current findings reveal that the torso's amplitude is influenced by the pack's movement. Vertical excursion of the torso is decreased as the pack load is allowed to swing. This difference probably took place to adjust to the pack's increased vertical motion and the mechanical forces exerted on the torso by the pack load during SL condition. Participants may have adopted to reduced vertical excursion to adopt the pack load's movement in vertical direction.

During the suspension locked condition, the peak accelerative forces were significantly higher than VS and IVS modes ($p < 0.001$). Note that, damping of the inerter module was also dominant on the torso and pack load. Absence of inerter module's

damping may result in improved design and better performance. Mathematical model reveals that relative displacement during IVS mode should be significantly less than VS mode. The mismatch between the model and experimental results can be attributed to additional parasite damping, simplified model of complex backpack attachment dynamics and nonlinear deflection characteristics of axially loaded buckling beams.

Proposed backpack's characteristics are adjusted using a hardware-based impedance modulation approach. Even, the prototype met the goal of lowering the vertical excursion of the load as well as the peak accelerative forces by comparing the peak-to-peak vertical excursion of the load to the vertical excursion of the torso of a walking person, there are some limitations that may be improved in future designs. First, the mechanism was 5.2 kg, which may be reduced with use of alternative design and manufacturing methods. Additional 5.2 kg during load carriage results in 10.5 % increase in net metabolic rate [82]. Metabolic cost reduction may be improved with reduced mass of mechanism. Limited work-space for the pack load due to the variable stiffness module may cause the prototype to under-perform at relatively high walking speeds due to increased vertical excursion of the torso. Reducing the overall damping may improve the design to be more efficient under various conditions of pack load and walking speed. Finally, the pack load was farther away from the body, which is a fundamental trade-off for suspended load backpacks. This distance may cause balance problems and decreased efficiency.

Backpacks were also evaluated using System Usability Scale. Adjective rating scores of suspended backpack conditions were significantly ($p < 0.05$) higher than the suspension locked condition. According the adjective rating, wearers considered device as "poor" during the suspension locked condition. Moreover, backpacks were examined as "good" during VS and IVS conditions. IVS module was preferred as the best by 62.5% of participants. Moreover, 37.5% of participants preferred VS

mode as the best. All of the participants identified suspension-locked case as the worst trial among the others.

These results indicate that a passive device can reduce the metabolic cost of load carriage (23.5 kg at 5.4 km/h speed). Moreover, decreased peak accelerative forces can help user to walk more, carry more load, and consume less energy with respect to the traditional backpack designs. Although this study found some significant differences across the conditions, bigger sample sizes, longer testing times, more repetitions with the same participants and various pack loads and walking speed evaluations may be required to determine whether variable stiffness based or inter-variable stiffness based suspended load backpacks can reduce fatigue and enhance endurance under various conditions.

Chapter 7

Conclusion

A novel passive suspended load backpack with manually adjustable impedance has been proposed. The proposed passive suspension relies on two distinct approaches. On the one hand, the negative transverse stiffness characteristics of axially loaded beams is utilized to decrease the suspension stiffness, as this method can ensure mechanical integrity and load bearing capabilities of the system while achieving very low stiffness levels. On the other hand, an inerter between the load and the frame is implemented to increase effective inertia without adding significant weight to the system. The simultaneous use of both strategies is possible with efficient and compact designs featuring effective filtering characteristics.

Analytical models of the proposed system have been derived and filtering characteristics of the suspension has been studied through comprehensive simulations. A prototype of the system has been implemented and experimentally characterized. These results have also been verified against the analytical models. Effective stiffness and inertia characteristics of the backpack is shown to be manually adjustable using variable stiffness and inerter modules.

Experimental results indicates statistically significant difference between variable stiffness and inerter-based variable stiffness conditions of backpack for net metabolic power, phase difference, vertical excursion of torso and vertical excursion of pack load measurements.

The effectiveness of the proposed backpack on the metabolic cost and peak force reduction during load carriage has been verified through a set of human subject experiments. It has been shown that the variable stiffness module can reduce the peak inertial forces by $30 \pm 11.21\%$ to result in a metabolic cost decrease of $6.16 \pm 3.90\%$, while the simultaneous employment of both inerter and variable stiffness modules can reduce the peak inertial forces by $42.17 \pm 15.48\%$ to result in a metabolic cost decrease of $10.24 \pm 3.67\%$.

Future works include a more comprehensive set of experiments that include motion capture with ground reaction forces and sEMG measurements of the participants to further analyse bio-mechanics of load carriage with suspended load backpacks.

Bibliography

- [1] Jeffrey Ackerman and Justin Seipel. A model of human walking energetics with an elastically-suspended load. *Journal of Biomechanics*, 47(8):1922 – 1927, 2014.
- [2] Joseph Knapik, Everett Harman, and Katy Reynolds. Load carriage using packs: A review of physiological, biomechanical and medical aspects. *Applied Ergonomics*, 27(3):207 – 216, 1996.
- [3] Joseph Knapik, Katy Reynolds, and Everett Harman. Soldier load carriage: Historical, physiological, biomechanical, and medical aspects. *Military medicine*, 169:45–56, 02 2004.
- [4] Michael Holewijn. Physiological strain due to load carrying. *European journal of applied physiology and occupational physiology*, 61:237–45, 02 1990.
- [5] L. Stewart Anderson Jr., Casey M. Rebholz, Laura F. White, Patricia Mitchell, Edward P. Curcio III, James A. Feldman, and Joseph H. Kahn. The Impact of Footwear and Packweight on Injury and Illness Among Long-Distance Hikers. *Wilderness Environmental Medicine*, 20(3):250 – 256, 2009.
- [6] Ghori G.M.U. and R Luckwill. Responses of the lower limb to load carrying in walking man. *European journal of applied physiology and occupational physiology*, 54:145–50, 02 1985.
- [7] Peter Quesada, Larry Mengelkoch, Robert Hale, and Sheldon Simon. Biomechanical and metabolic effects of varying backpack loading on simulated marching. *Ergonomics*, 43:293–309, 03 2000.

- [8] Youlian Hong, Jing-Xian Li, and Daniel Fong. Effect of prolonged walking with backpack loads on trunk muscle activity and fatigue in children. *Journal of electromyography and kinesiology : official journal of the International Society of Electrophysiological Kinesiology*, 18:990–6, 09 2007.
- [9] Daijiro Abe, Satoshi Muraki, and Akira Yasukouchi. Ergonomic effects of load carriage on energy cost of gradient walking. *Applied Ergonomics*, 39(2):144 – 149, 2008.
- [10] K. B. Pandolf, B. Givoni, and R. F. Goldman. Predicting energy expenditure with loads while standing or walking very slowly. *Journal of Applied Physiology*, 43(4):577–581, 1977. PMID: 908672.
- [11] Renee Attwells, Stewart Birrell, Robin Hooper, and Neil Mansfield. Influence of carrying heavy loads on soldiers’ posture, movements and gait. *Ergonomics*, 49:1527–37, 12 2006.
- [12] E. Harman, Ki-Hoon Han, P. Frykman, M. A. Johnson, Franklin Russell, and M. Rosenstein. The effects on gait timing, kinetics, and muscle activity of various loads carried on the back: 774. *Medicine and Science in Sports and Exercise*, 24, 1992.
- [13] M. R. Pierrynowski, R. Norman, and D. Winter. Mechanical energy analyses of the human during local carriage on a treadmill. *Ergonomics*, 24 1:1–14, 1981.
- [14] Lawrence Rome, Louis Flynn, and Taeseung Yoo. Biomechanics: Rubber bands reduce the cost of carrying loads. *Nature*, 444:1023–4, 01 2007.
- [15] K. Holt, J. Hamill, and Robert O. Andres. Predicting the minimal energy costs of human walking. *Medicine and science in sports and exercise*, 23 4:491–8, 1991.
- [16] Justus Ortega and Claire Farley. Minimizing center of mass vertical movement increases metabolic cost in walking. *Journal of applied physiology (Bethesda, Md. : 1985)*, 99:2099–107, 01 2006.

- [17] RR Neptune, FE Zajac, and SA Kautz. Muscle mechanical work requirements during normal walking: the energetic cost of raising the body's center-of-mass is significant. *Journal of biomechanics*, 37(6):817—825, June 2004.
- [18] Michael Lafiandra and Everett Harman. The distribution of forces between the upper and lower back during load carriage. *Medicine and science in sports and exercise*, 36:460–7, 04 2004.
- [19] M. C. Smith. Synthesis of mechanical networks: the inerter. *IEEE Transactions on Automatic Control*, 47(10):1648–1662, Oct 2002.
- [20] M. Z. Q. Chen, K. Wang, Y. Zou, and G. Chen. Realization of three-port spring networks with inerter for effective mechanical control. *IEEE Transactions on Automatic Control*, 60(10):2722–2727, 2015.
- [21] S N Forjuoh, D Little, J A Schuchmann, and B L Lane. Parental knowledge of school backpack weight and contents. *Archives of Disease in Childhood*, 88(1):18–19, 2003.
- [22] F.C. Trevelyan and S.J. Legg. Back pain in school children—where to from here? *Applied Ergonomics*, 37(1):45 – 54, 2006. Special Issue: Fundamental Reviews.
- [23] J. Knapik. Loads carried by soldiers: Historical, physiological, biomechanical and medical aspects. 1989.
- [24] EH. Burba. The soldier's load. *Infantry*, 1986.
- [25] Justus Ortega and Claire Farley. Minimizing center of mass vertical movement increases metabolic cost in walking. *Journal of applied physiology (Bethesda, Md. : 1985)*, 99:2099–107, 01 2006.
- [26] Katy Reynolds, John Kaszuba, Robert Mello, and John Patton. Prolonged treadmill load carriage: Acute injuries and changes in foot anthropometry. page 45, 06 1990.
- [27] Joon-Hyuk Park, Paul Stegall, Sunil Agrawal, S. Yarlagadda, John Tierney, Shashank Sharma, and John Gillespie Jr. Wearable upper body suit for assisting human load carriage. 08 2015.

- [28] J-H Goh, A Thambyah, and K Bose. Effects of varying backpack loads on peak forces in the lumbosacral spine during walking. *Clinical Biomechanics*, 13(1, Supplement 1):S26 – S31, 1998.
- [29] Kimberly D. Dahl, He Wang, Jennifer K. Popp, and D. Clark Dickin. Load distribution and postural changes in young adults when wearing a traditional backpack versus the backpack. *Gait Posture*, 45:90 – 96, 2016.
- [30] Heidi Orloff and Catherine Rapp. The effects of load carriage on spinal curvature and posture. *Spine*, 29:1325–9, 07 2004.
- [31] S. Kumar. Cumulative load as a risk factor for back pain. *Spine*, 15:1311–1316, 1990.
- [32] Katrina Simpson, Bridget Munro, and Julie Steele. Backpack load affects lower limb muscle activity patterns of female hikers during prolonged load carriage. *Journal of electromyography and kinesiology : official journal of the International Society of Electrophysiological Kinesiology*, 21:782–8, 06 2011.
- [33] Joon-Hyuk Park. *Wearable Torso Exoskeletons for Human Load Carriage and Correction of Spinal Deformities*. PhD thesis, Columbia University, 2016.
- [34] G. A. Cavagna, N. C. Heglund, and C. R. Taylor. Mechanical work in terrestrial locomotion: two basic mechanisms for minimizing energy expenditure. *American Journal of Physiology-Regulatory, Integrative and Comparative Physiology*, 233(5):R243–R261, 1977. PMID: 411381.
- [35] RM Alexander. Energy-saving mechanisms in walking and running. *The Journal of experimental biology*, 160:55—69, October 1991.
- [36] Stewart Birrell and Roger Haslam. The effect of military load carriage on 3-d lower limb kinematics and spatiotemporal parameters. *Ergonomics*, 52:1298–304, 10 2009.
- [37] Everett Harman, Ki Hoon, Peter Frykman, and Clay Pandorf. The effects of backpack weight on the biomechanics of load carriage. page 72, 05 2000.

- [38] Philip Martin and Richard Nelson. The effect of carried loads on the walking patterns of men and women. *Ergonomics*, 29:1191–202, 11 1986.
- [39] J. Bachkosky, M. Andrews, R. Douglass, J. Feigley, L. Felton, F. Fernandez, P. Fratarangelo, A. Johnson-Winegar, R. Kohn, N. Polmar, R. Rumpf, J. Sommerer, and W. Williamson. Lightening the load. page 90, 09 2007.
- [40] Joon-Hyuk Park, Xin Jin, and Sunil Agrawal. Second spine: Upper body assistive device for human load carriage. *Journal of Mechanisms and Robotics*, 7:1–10, 02 2015.
- [41] Ray Lloyd and Carlton Cooke. The oxygen consumption with unloaded walking and load carriage using two different backpack designs. *European Journal of Applied Physiology - EUR J APPL PHYSIOL*, 81:486–492, 03 2000.
- [42] Stephen Legg and A Mahanty. Comparison of five modes of carrying a load close to the trunk†. *Ergonomics*, 28:1653–60, 01 1986.
- [43] Christa Devroey, Ilse Jonkers, An Becker, Gerlinde Lenaerts, and Arthur Spaepen. Evaluation of the effect of backpack load and position during standing and walking using biomechanical, physiological and subjective measures. *Ergonomics*, 50:728–42, 06 2007.
- [44] R. F. Johnson, J. Knapik, and D. Merullo. Symptoms during load carrying: Effects of mass and load distribution during a 20-km road march. *Perceptual and Motor Skills*, 81:331 – 338, 1995.
- [45] A. B. Zoss, H. Kazerooni, and A. Chu. Biomechanical design of the berkeley lower extremity exoskeleton (bleex). *IEEE/ASME Transactions on Mechatronics*, 11(2):128–138, 2006.
- [46] Joon-Hyuk Park, Paul Stegall, Haohan Zhang, and Sunil Agrawal. Walking with a backpack using load distribution and dynamic load compensation reduces metabolic cost and adaptations to loads. *IEEE Transactions on Neural Systems and Rehabilitation Engineering*, 25:1419–1430, 09 2017.

- [47] J. Hoover and S. Meguid. Performance assessment of the suspended-load backpack. *International Journal of Mechanics and Materials in Design*, 7:111–121, 06 2011.
- [48] Matthieu Foissac, Guillaume Millet, Andre Geyssant, Philippe Freychat, and Alain Belli. Characterization of the mechanical properties of backpacks and their influence on the energetics of walking. *Journal of biomechanics*, 42:125–30, 01 2009.
- [49] F. A. Firestone. A new analogy between mechanical and electrical systems. *The Journal of the Acoustical Society of America*, 4(1A):5–5, 1932.
- [50] M. C. Smith. Synthesis of mechanical networks: the inerter. *IEEE Transactions on Automatic Control*, 47(10):1648–1662, 2002.
- [51] M. C. Smith and Fu-Cheng Wang. Performance benefits in passive vehicle suspensions employing inerters. In *42nd IEEE International Conference on Decision and Control (IEEE Cat. No.03CH37475)*, volume 3, pages 2258–2263 Vol.3, 2003.
- [52] Malcolm Clive Smith. Force-controlling mechanical device, patent no: Us7316303b2.
- [53] Yujie Shen, Long Chen, Xiaofeng Yang, Dehua Shi, and Jun Yang. Improved design of dynamic vibration absorber by using the inerter and its application in vehicle suspension. *Journal of Sound and Vibration*, 361:148 – 158, 2016.
- [54] Karthik Madhamshetty and James Manimala. Low-rate characterization of a mechanical inerter. *Machines*, 6:32, 07 2018.
- [55] S. Swift, M. Smith, A. Glover, C. Papageorgiou, B. Gartner, and N. Houghton. Design and modelling of a fluid inerter. *International Journal of Control*, 86, 11 2013.
- [56] Glover and Malcolm Clive Smith. Force-controlling hydraulic device. patent no: Us8881876b2.
- [57] Fu-Cheng Wang and Wei-Jiun Su. Impact of inerter nonlinearities on vehicle suspension control. *Vehicle System Dynamics - VEH SYST DYN*, 46:575–595, 07 2008.

- [58] S. J. Swift, M. C. Smith, A. R. Glover, C. Papageorgiou, B. Gartner, and N. E. Houghton. Design and modelling of a fluid inerter. *International Journal of Control*, 86(11):2035–2051, 2013.
- [59] Fu-Cheng Wang, Min-Kai Liao, Bo-Huai Liao, Wei-Jiun Su, and Hsiang-An Chan. The performance improvements of train suspension systems with mechanical networks employing inerters. *Vehicle System Dynamics - VEH SYST DYN*, 47:805–830, 06 2009.
- [60] Fu-Cheng Wang, M-F Hong, and C-W Chen. Building suspensions with inerters. *Proceedings of The Institution of Mechanical Engineers Part C-journal of Mechanical Engineering Science - PROC INST MECH ENG C-J MECH E*, 224:1605–1616, 08 2010.
- [61] Ali Siami, H.R. Karimi, Alfredo Cigada, Emanuele Zappa, and Edoardo Sabbioni. Parameter optimization of an inerter-based isolator for passive vibration control of michelangelo’s rondanini pietà. *Mechanical Systems and Signal Processing*, 98:667–683, 01 2018.
- [62] O. C. Kara and V. Patoglu. Vnstylus: A haptic stylus with variable tip compliance. *IEEE Transactions on Haptics*, pages 1–1, 2020.
- [63] M. Garabini, A. Passaglia, F. Belo, P. Salaris, and A. Bicchi. Optimality principles in variable stiffness control: The vsa hammer. In *2011 IEEE/RSJ International Conference on Intelligent Robots and Systems*, pages 3770–3775, 2011.
- [64] I. Youn and A. Hać. Semi-active suspensions with adaptive capability. *Journal of Sound and Vibration*, 180:475–492, 1995.
- [65] Takuji Kobori, Motoichi Takahashi, Tadashi Nasu, Naoki Niwa, and Katsura Ogasawara. Seismic response controlled structure with active variable stiffness system. *Earthquake Engineering & Structural Dynamics*, 22(11):925–941, 1993.
- [66] Huan Li, Yancheng Li, and Jianchun Li. Negative stiffness devices for vibration isolation applications: A review. *Advances in Structural Engineering*, 23, 12 2019.

- [67] Michael Chen and Yinlong Hu. *Inerter and Its Application in Vibration Control Systems*. 04 2019.
- [68] Nanhui Yu, Huajiang Ouyang, Jiafan Zhang, Jianping Ye, Qinghua Yan, and Xiao Zheng. Performance analyses of passive vibration isolator with parallel connection of quasi-zero stiffness and inerter dampers. *Applied Sciences*, 10(19), 2020.
- [69] Charalambos Charalambous. The major determinants in normal and pathological gait. *Classic Papers in Orthopaedics*, pages 403–405, 12 2014.
- [70] D Grieve and Ruth Gear. The relationships between length of stride, step frequency, time of swing and speed of walking for children and adults. *Ergonomics*, 9:379–99, 10 1966.
- [71] Xu Xu. *An Investigation on the Interactivity between Suspended-load Backpack and Human Gait*. PhD thesis, North Carolina State University, 2008.
- [72] S. S. Rao and Philip Griffin. *Mechanical vibrations*. Pearson Education, 2018.
- [73] M. T. A. Saif. On a tunable bistable mems-theory and experiment. *Journal of Microelectromechanical Systems*, 9(2):157–170, 2000.
- [74] M. Yalcin, B. Uzunoglu, E. Altintepe, and V. Patoglu. Vnsa: Variable negative stiffness actuation based on nonlinear deflection characteristics of buckling beams. In *2013 IEEE/RSJ International Conference on Intelligent Robots and Systems*, pages 5418–5424, 2013.
- [75] Suresh Kp. An overview of randomization techniques: An unbiased assessment of outcome in clinical research. *Journal of human reproductive sciences*, 4:8–11, 03 2011.
- [76] Tim Meyer, T Georg, C Becker, and W Kindermann. Reliability of gas exchange measurements from two different spiroergometry systems. *International journal of sports medicine*, 22:593–7, 12 2001.

- [77] M. Sedlacek and M. Krumpholtz. Digital measurement of phase difference - a comparative study of dsp algorithms. *Metrology and Measurement Systems*, 12:427–448, 2005.
- [78] Hristo Zhivomirov. Phase difference measurement with matlab (2021). (matlab central file exchange),. retrieved september 4, 2021.
- [79] Pauline Maurice, Jernej Čamernik, Dasa Gorjan, Benjamin Schirrmeister, Jonas Bornmann, Luca Tagliapietra, Claudia Latella, Daniele Pucci, Lars Fritzsche, Serena Ivaldi, and Jan Babic. Objective and subjective effects of a passive exoskeleton on overhead work. *IEEE Transactions on Neural Systems and Rehabilitation Engineering*, PP:1–1, 10 2019.
- [80] John Brooke. Sus: A quick and dirty usability scale. *Usability Eval. Ind.*, 189, 11 1995.
- [81] Ledeng Huang, Zhenhua Yang, Ruishi Wang, and Longhan Xie. Physiological and biomechanical effects on the human musculoskeletal system while carrying a suspended-load backpack. *Journal of Biomechanics*, 108:109894, 06 2020.
- [82] Raymond Browning, Jesse Modica, Rodger Kram, and Ambarish Goswami. The effects of adding mass to the legs on the energetics and biomechanics of walking. *Medicine and science in sports and exercise*, 39:515–25, 04 2007.

Appendix A

Analysis of possible layouts for inerter based suspended-load backpack

To implement inerters in suspended backpack models, we will investigate three different layout that are consisting of spring, damper and inerter. Figure 7.1, shows three different layouts. Proposed layouts are characterized with different impedances. Layout (a) is

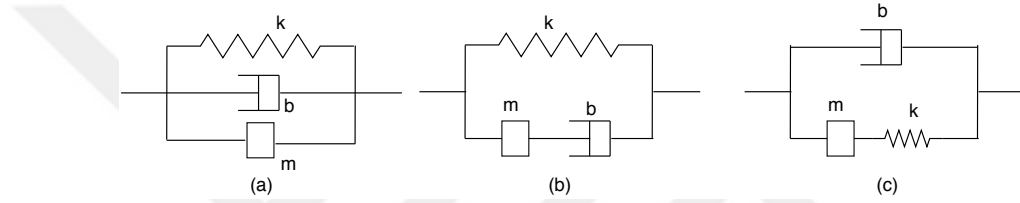


FIGURE 7.1: Different compliance layouts for $Q(s)$ consist of spring-damper-inerter

parallel connection of spring, damper and inerter. In layout (b), damper and inerter are in a series combination with parallel connected to the spring. Then finally, spring and damper in series at parallel combination with spring. Their impedance characteristics can be shown as below

$$\begin{aligned}
 Q_a(s) &= \frac{k}{s} + b + m s \\
 Q_b(s) &= \frac{k}{s} + \frac{b m s}{b + m s} \\
 Q_c(s) &= b + \frac{k m s}{m s^2 + k}
 \end{aligned}$$

In order to identify the most efficient layout, a scaling factor is defined. The scaling factor S is defined as the ratio of the $|F_{\text{osc}}|$ for two different layouts. The scalar as can be calculated for $(i,j=a,b,c)$ by using following equation

$$S_{i,j} = \frac{|F_{\text{osc}}|_{Q=Q_i}}{|F_{\text{osc}}|_{Q=Q_j}} \quad (7.1)$$

For example the scalar factor $S_{a,b}$ can be used in order to compare the layouts (a) and (b) in Figure 7.1. The only difference is $|F_{osc}|$ in comparison of the layouts. For the values of S lower than 0, oscillation force is lower for the first indice. In Figure 7.2, the scaling factor $S_{a,c}$ is shown on a contour plot on a wide range of walking speed (v) and backpack mass (M). For any velocity and backpack mass, layout (a) does not provide better performance when it is compared with layout (c) since the scaling factor is greater than 1. Even, layout (a) and layout (c) consist of spring, damper and inerter elements, performance is worse in the parallel connection of all elements. Series combination of inerter and spring with parallel damper corresponds to an improved performance by a reduction in the oscillating backpack force.

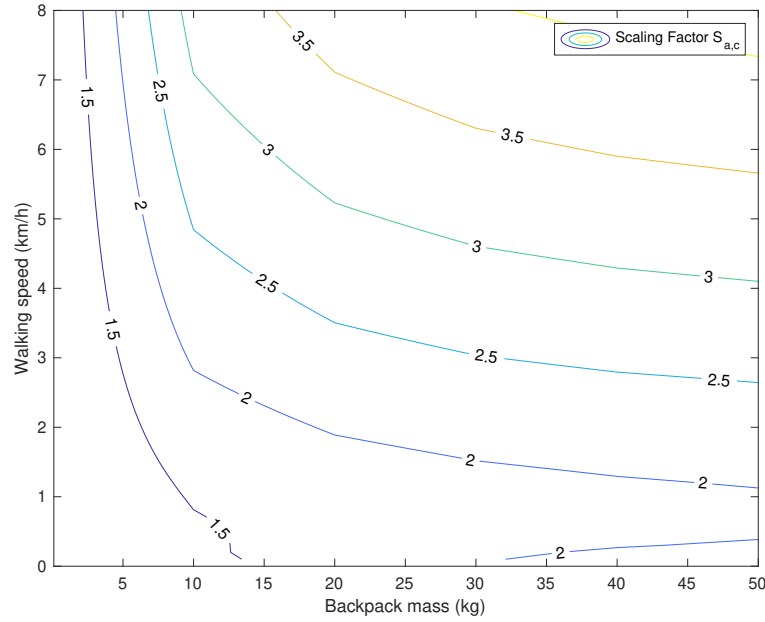


FIGURE 7.2: Backpack performance comparison for layout (a) and (c) given in Figure 7.1 on human with weight 1.78 m, walking speed 5 km/h, backpack mass 18.5 kg, stiffness k 3300 N/m, damping b 96 Ns/m and inertance m 5kg

Scalar factor $S_{c,b}$ characterizes the performance trade-off of placement of inerter in series

of one of spring and damper. For a particular walking speed, both of the arrangements provide similar performance. In the designs for high walking speeds, layout (b) promises lower oscillating force with comparison to layout(c). However, at low speeds, layout (c) provides lower amount of oscillating force and provides improved performance.

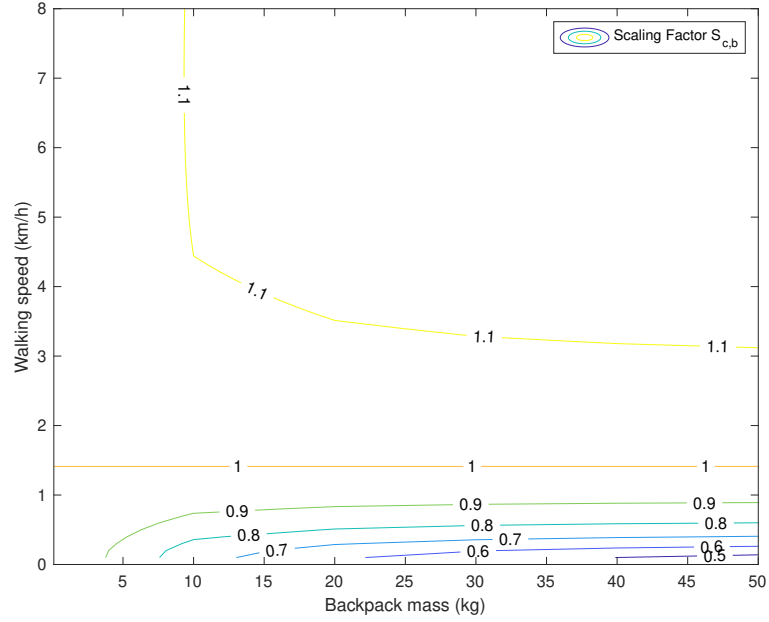


FIGURE 7.3: Backpack performance comparison for layout (c) and (b) given in Figure 7.1 on human with weight 1.78 m, walking speed 5 km/h, backpack mass 18.5 kg, stiffness k 3300 N/m, damping b 96 Ns/m and inertance of m 5 kg

Another difference between the proposed layouts is $k_{\text{performance}}$. Performance stiffness for the layouts are given as following

$$k_{\text{perf},b} = \frac{1}{2}\omega^2(M + m \pm \sqrt{M^2 + m^2})$$

$$k_{\text{perf},c} = \frac{M m \omega^2}{M + 2 m}$$

Layout (b) results in two values for performance stiffness, which provides a relaxation in design. In Figure 7.4, the oscillation load profile is shown for layout (b) in a relatively high walking speed and small inertance factor.

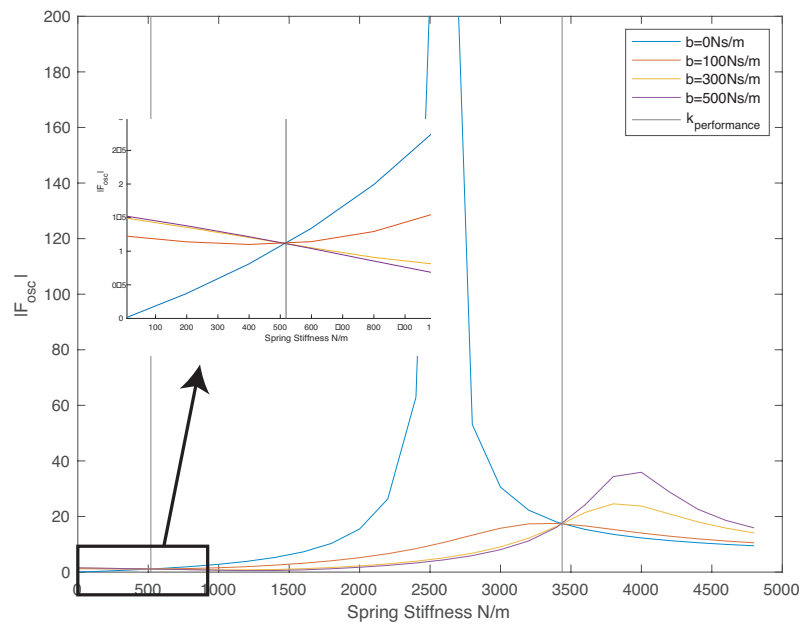


FIGURE 7.4: Oscillation load profile for layout(b) on human with weight 1.78m, walking speed 5km/h, backpack mass 18.5 kg, and inertance factor m 10 kg

Appendix B

In this section two of the questionnaires used during human-subject experiments are presented.

TABLE 7.1: Questionnaire filled out by participants to evaluate the qualitative metrics with a 5-point Likert scale.

	Strongly disagree	Strongly agree
1. I think that I would like to use this backpack frequently.	1 2 3 4 5	
2. I found the backpack unnecessarily complex.	1 2 3 4 5	
3. I thought the backpack was easy to use.	1 2 3 4 5	
4. I think that I would need the support of a technical person to be able to use this backpack.	1 2 3 4 5	
5. I found the various functions in this backpack were well integrated.	1 2 3 4 5	
6. I thought there was too much inconsistency in this backpack.	1 2 3 4 5	
7. I would imagine that most people would learn to use this backpack very quickly.	1 2 3 4 5	
8. I found the backpack very cumbersome to use.	1 2 3 4 5	
9. I felt very confident using the backpack.	1 2 3 4 5	
10. I needed to learn a lot of things before I could get going with this backpack.	1 2 3 4 5	
	Best	Worst
11. Preference - Please rank all backpack devices you have tried so far, from the best to the worst. (Answer may be updated after each trial) .	1 2 3	

TABLE 7.2: Questionnaire on compliance with the COVID-19 recommended by CORTEX

Question No.	Participant:	Date:	
Prior to testing (questions to subject via phone)			
1)	Have you had fever, chills, shortness of breath, cough, loss of taste or smell within the last 7 days? If yes, please reschedule the experiment.	Yes ☐	No ☐
2)	Have you been exposed to individuals that tested positive to COVID-19 within the past 14 days? If yes, please reschedule the experiment.	Yes ☐	No ☐
3)	Have you been tested positive with COVID-19 within the past 14 days?	Yes ☐	No ☐
Prior to testing when subject enters facility			
4)	Is the subject's body temperature above 37.4°C? If yes please reschedule test!	Yes ☐	No ☐
5)	Has the subject sanitized his/her hands? Does the subject wear a face mask?	Yes ☐	No ☐
6)	Do you as a technician/physician wear additional protection such as face shield, examination gloves and disposable gowns?	Yes ☐	No ☐
7)	Have you ventilated the room?	Yes ☐	No ☐
8)	Has the subject placed all his/her belongings in a prepared (disinfected) plastic box?	Yes ☐	No ☐
During Cardiopulmonary Exercise Testing (CPET)			
9)	Please keep your distance with the subject.	Yes ☐	No ☐
10)	Use a barrier between you and the subject.	Yes ☐	No ☐
After testing			
11)	Did you leave the laboratory equipment for 20 minutes before you disinfected?	Yes ☐	No ☐
12)	Did you use reusable turbine? If yes please disinfect!	Yes ☐	No ☐
13)	Did you disinfect the reusable turbine afterwards?	Yes ☐	No ☐
14)	Did you use reusable CPET mask? If yes please disinfect!	Yes ☐	No ☐
15)	Did you disinfect the used CPET mask afterwards?	Yes ☐	No ☐
16)	Did you use additional bacterial and viral filter?	Yes ☐	No ☐
17)	Have you ventilated the room for at least 10 min. or do you use other means to improve the air quality?	Yes ☐	No ☐
18)	Have you disinfected the plastic box?	Yes ☐	No ☐