

**Development of a Dynamic Biomechanical Model for
Load Carriage: Phase III Part C2**

**Development of a Dynamic Biomechanical Model Version 1 of
Human Load Carriage**

by
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Abstract

The overall purpose of the DRDC research program on dynamic biomechanical modeling is to improve the understanding of human load carriage capabilities and to understand the benefits of load carriage system design features to human health and mobility. Earlier phases of the dynamic biomechanical model have led to a new modeling approach that treats the pack-person interface as a dynamic suspension system. In the current study, both 2D and 3D dynamic modeling software packages were selected to permit multiple models of the pack person suspension characteristics. The selected software both permit full user control of model geometry, inertial properties, have extensive libraries of existing dynamic elements for modeling constraints, allow the user to construct complex constraint equations and allow the user to input complex forcing functions. For both the 2D and 3D models, two types of dynamic tests were conducted to determine the impulse response and the natural frequencies. For the 2D model, the impulse response test showed typical results for a mildly under-damped system with the amplitude ratio plot showing a modest peak at approximately 8 Hz, higher than the estimated natural frequency of 4.8 Hz. On the other hand, the impulse response test for the 3D model gave a vertical displacement typical of an over-damped system and an amplitude ratio plot with several resonant frequencies at approximately 2.5 Hz and again at 5 Hz. With the damping reduced by a factor of 100, there were some initial oscillations of the system followed by a slow decay in the vertical position and as expected, the minimally damped 3D model displayed a dominant natural frequency at approximately 5 Hz. Overall, the 2D model required much higher damping coefficients to bring about a pack displacement pattern similar to that of the 3D model. In addition, the 3D model behaviour was more consistent with the physical system. The next stage in model development is to integrate a waist belt model (Hadcock, 2002) being developed separately into the 3D model.

Résumé

L'objectif global du programme de recherche sur la modélisation biomécanique dynamique menée par RDDC est d'améliorer la compréhension des capacités humaines de transport de charge et de comprendre les avantages des caractéristiques de conception de systèmes de transport de charge pour la santé humaine et la mobilité. Les phases antérieures de la modélisation biomécanique dynamique ont servi de fondement à la création d'une nouvelle méthode de modélisation dans laquelle l'interface sac-personne est traitée comme un système de suspension dynamique. Durant l'étude en cours, on a choisi des progiciels de modélisation dynamique en deux et en trois dimensions afin de pouvoir créer plusieurs modèles de l'interface sac-personne. Les progiciels choisis permettent à l'utilisateur de définir entièrement la géométrie du modèle et les propriétés d'inertie, comprennent des bibliothèques exhaustives d'éléments dynamiques existants pour la modélisation des contraintes et permettent à l'utilisateur de construire des équations de contraintes complexes et de saisir des fonctions de forçage complexes. Pour les deux modèles (2D et 3D), on a mené deux types d'essais dynamiques : l'un pour déterminer la réponse impulsionnelle et l'autre, les fréquences naturelles. Pour le modèle 2D, les résultats de l'essai de réponse impulsionnelle obtenus sont typiques pour un système légèrement sous-amorti, et le tracé du rapport d'amplitude affiche une crête modeste à environ 8 Hz, ce qui est supérieure à la fréquence naturelle estimative de 4,8 Hz. Par contre, pour le modèle 3D, le déplacement vertical obtenu lors de l'essai de réponse impulsionnelle est représentatif d'un système sur-amorti, et le tracé du rapport d'amplitude montre plusieurs fréquences de résonance à environ 2,5 Hz, et d'autres à 5 Hz. Lorsque l'amortissement a été réduit par un facteur de 100, le système a montré quelques oscillations initiales, suivi d'une lente décroissance, et comme prévu, le modèle 3D légèrement amorti a affiché une fréquence naturelle dominante d'environ 5 Hz. Globalement, il a fallu utiliser des coefficients d'amortissement plus élevés avec le modèle 2D qu'avec le modèle 3D, pour obtenir une séquence de déplacement du sac à dos similaire à celle du modèle 3D. De plus, le comportement du modèle 3D ressemblait davantage à celui du système matériel. La prochaine étape de développement du modèle consiste en l'intégration d'un modèle de ceinture¹, qui est élaboré séparément, au modèle 3D.

¹ Hadcock, L., Master of Science Thesis for the School of Physical and Health Education, Université Queen's, Kingston, Canada. Tous droits réservés. Juillet 2002.

Executive Summary

The overall purpose of the DRDC research program on dynamic biomechanical modeling is to improve the understanding of human load carriage capabilities and to understand the benefits of load carriage system design features. By increasing our understanding of how dynamic loads are best borne by the body, we can learn what strategies that are used by the human organism to minimize the effect of load bearing on its health and mobility. Earlier static modeling work (Stevenson et al, 1995) identified several useful biomechanical load carriage limits, which has led to a recommendation of a maximum of 290 N compressive load on the shoulders and a maximum acceptable shear load in the lumbar area of 135N. The current work begins this process of integrating mathematical modeling of human load carriage with the knowledge previously gained about human biomechanical and discomfort tolerances.

Two dynamic modeling software packages were selected to permit multiple attempts at modeling the pack person interface. MSC Working Model is a two-dimensional (2D) planar analysis package and MSC Visual Nastran performs three-dimensional (3D) motion and stress analysis. The selected software both permit full user control of model geometry, inertial properties, have extensive libraries of existing dynamic elements for modeling constraints, allow the user to construct complex constraint equations and allow the user to input complex forcing functions.

2D Model Description

The 2D software, Working Model (WM), was used extensively in the initial stages of modeling to evaluate potential suspension configurations for the shoulder straps.

MSC WM software provides a limited drawing tablet to create the physical shape of objects. Creation of a WM object with the correct profile required a two-stage process. This process resulted in a consistent body shape across the 2 and 3D analyses.

A simple rectangular pack shape was created based on the geometry of the Clothe the Soldier (CTS) pack tested in the December 2001 trial held at Queen's University. The mass and inertial properties of the 'medium' load condition were modeled. Thus, the mass of the pack model was 25 kg, while the body model had a mass of 70 kg.

Initially the shoulder straps were modeled as a Voigt-Kelvin visco-elastic model with a stiff linear spring in series with a linear damper. Rope elements were added later between the springs and the attachment points on the body to cause

the strap model to only exert a force when in tension. In addition, the torso was constrained to move along a path fixed in the vertical (+/- Z) direction.

3D Model Description

A 3 dimensional surface scan of the 50 percentile male mannequin torso created previously, was imported into Visual Nastran 4D® (VN4D). For this analysis, a rectangular representative pack was created based on the geometry of the CTS pack and consistent with the pack model used in the 2D analyses.

Displacement of the torso model is a controlled variable in these analyses; therefore the torso mass and inertial properties do not enter into the dynamic analysis. The pack and torso were defined as frictionless objects with a coefficient of restitution of 0.5.

Consistent with the 2D model, shoulder straps in the 3D model were modeled as a Voigt-Kelvin visco-elastic material with a stiff linear spring in series with a linear damper. For each step of the analysis, the 3D strap length was evaluated and a strap force was applied only if the current length indicated the strap was in tension. The 3D torso was modeled as rigidly fixed to an actuator constrained to move along the Z-axis.

Results and Discussion

In an undamped oscillating system, the resonant natural frequency of oscillation is determined by the following relationship:

$$\omega = \sqrt{\frac{k}{m}}$$

For both the 2D and 3D models, two types of dynamic tests were conducted to determine both the impulse response and the natural frequencies.

The 2D planer model had spring stiffness of 20000 N/m in the upper and lower shoulder straps and the mass of the backpack was modelled as 25 kg. Treating the vertical force components of the upper and lower straps as the springs controlling the vertical oscillations of the pack allows calculation of an effective spring stiffness that can be substituted into the equation for the natural resonant frequency. In this case, the estimated natural frequency was 4.8 Hz.

When released from 2.5 cm above the rest position, the displacement response of the pack showed results typical for a mildly under damped system. A range of forcing function frequencies were also analysed using the 2D model and the ratio of the pack motion amplitude to the amplitude of the forcing function was calculated and plotted as a function of the forcing frequency. A modest peak was observed at approximately 8 Hz, higher than the estimated natural frequency of 4.8 Hz.

The 3D planer model had 4 springs (upper and lower, left and right side) each with spring stiffness of 20000 N/m, and the mass of the backpack was modelled as 24.5 kg. The vertical force components of the upper and lower straps were treated similarly to the method described for the 2D model, with the exception that there were four springs to be included in effective spring stiffness calculation. This $K_{\text{Equivalent}}$ was substituted into the equation for the natural resonant frequency to estimate the natural frequency of the 3D model, which was calculated to be 7.3 Hz.

The damping coefficients (4 and 5 N.s/cm) that were determined to be appropriate from the results of the 2D model were used in the 3D model. When the 3D pack model was displaced 2.5 cm vertically from the rest position and then released, the results shown were typical for an over-damped system.

When a range of forcing function frequencies were analysed using the 3D model, the ratio of the pack motion amplitude to the amplitude of the forcing function was calculated and plotted as a function of the forcing frequency. The results indicated resonant frequencies at approximately 2.5 Hz and again at 5 Hz.

Damping was reduced from 400 and 500 Ns/m in the upper and lower shoulder strap models to 4 and 5 Ns/m respectively. With the damping reduced by a factor of 100, there were some initial oscillations of the system, followed by a slow decay in the vertical position.

As expected, the minimally damped 3D model displayed a dominant natural frequency at approximately 5 Hz. The other lower resonant frequencies were more easily identified as occurring at approximately 2 and 2.5 Hz.

Conclusions

Two shoulder carried dynamic load carriage models have been developed. The 2D model required much higher damping coefficients to bring about a pack displacement pattern similar to that of the 3D model. In addition, the 3D model behaviour was more consistent with the physical system.

The next stage in model development is to integrate the waist belt model being developed separately into the 3D model.

Sommaire

L'objectif global du programme de recherche sur la modélisation biomécanique dynamique menée par RDDC consiste à améliorer la compréhension des capacités humaines de transport de charge et de comprendre les avantages des caractéristiques nominales de systèmes de transport de charge. En améliorant notre compréhension du transport de charge dynamique par le corps, nous pouvons apprendre les stratégies que l'humain utilise pour minimiser les effets du transport d'une charge sur sa santé et sa mobilité. Lors des travaux de modélisation statique antérieurs², on a établi plusieurs limites biomécaniques de transport de charge qui ont abouti à la recommandation des limites maximales suivantes : 290 N pour la charge de compression aux épaules et 135 N pour la charge de cisaillement dans la région lombaire. Les travaux en cours visent à intégrer le modèle mathématique de transport humain de charge et les connaissances recueillies antérieurement sur les tolérances biomécaniques et l'inconfort ressenti par les humains.

Deux logiciels de modélisation dynamique ont été sélectionnés afin d'élaborer plusieurs modèles de l'interface sac-personne. MSC Working Model est un logiciel d'analyse plane en deux dimensions (2D), et MSC Visual Nastran effectue des analyses des mouvements et des contraintes en trois dimensions (3D). Les logiciels choisis permettent à l'utilisateur de définir entièrement la géométrie du modèle et les propriétés d'inertie, comprennent des bibliothèques exhaustives d'éléments dynamiques existants pour la modélisation des contraintes et permettent à l'utilisateur de construire des équations de contraintes complexes et de saisir des fonctions de forçage complexes.

Description du modèle 2D

Le logiciel de modélisation en 2D, Working Model (WM), a été utilisé considérablement durant les étapes initiales de la modélisation afin d'évaluer les configurations potentielles de suspension des sangles d'épaules.

Le logiciel WM de MSC est offert avec une tablette graphique limitée qui sert à créer les formes matérielles des objets. La création d'un objet WM et de son profil correct se fait en deux étapes. Le processus suivi permet de créer un modèle de corps ayant une forme cohérente pour les analyses en 2D et en 3D.

La forme d'un sac à dos rectangulaire simple a été créée en fonction de la géométrie du sac à dos utilisé lors des essais du programme Habillez le soldat (HLS), tenus en décembre 2001 à l'Université Queen's. La masse et les propriétés d'inertie d'une charge « moyenne » ont été modélisées. Donc, la masse du sac

² Stevenson, J.M., Bryant, J.T., DePencier, R.P., Pelot, R.P. et Reid, J.G. Research and Development of an Advanced Personal Load Carriage System: Section A, B, C (Phase I). Contrat MAS n° W7711-4-7225/01-XSE 29 (350 pp), 1995

utilisé pour le modèle était de 25 kg, et la masse du corps utilisé pour le modèle était de 70 kg.

Initialement, les sangles d'épaules ont été modélisées à l'aide de matériau visco-élastique Voigt-Kelvin et d'un ressort linéaire rigide monté en série avec un amortisseur linéaire. Plus tard, des éléments de câble ont été ajoutés entre les ressorts et les points de fixation au corps afin que la sangle modélisée applique une force seulement lorsqu'elle est sous traction. De plus, le mouvement du torse ne pouvait se faire que le long d'un trajet vertical fixe (+/- Z).

Description du modèle 3D

On a balayé la surface du torse du mannequin masculin du 50^e centile, qui avait été fabriqué précédemment, pour créer une image tridimensionnelle et l'importer dans le logiciel Visual Nastran 4D® (VN4D). Pour cette analyse, un sac à dos rectangulaire représentatif a été produit en fonction de la géométrie du sac HLS, qui est cohérent avec le modèle de sac utilisé pour les analyses 2D.

Dans ces analyses, le déplacement du modèle du torse était une variable contrôlée; par conséquent, la masse et les propriétés d'inertie du torse n'ont pas été prises en compte dans l'analyse dynamique. Le sac et le torse ont été définis comme étant des objets sans frottement ayant un coefficient de restitution de 0,5.

Tout comme dans le modèle 2D, les sangles d'épaules ont été modélisés à l'aide de matériau visco-élastique Voigt-Kelvin et d'un ressort linéaire rigide monté en série avec un amortisseur linéaire. À chaque étape de l'analyse, on a évalué la longueur de la sangle du modèle 3D, et une force était appliquée à la sangle seulement lorsque la longueur indiquait que celle-ci était sous traction. Le modèle 3D du torse supposait que celui-ci était fixé rigidement à un actionneur ne pouvant se déplacer que dans l'axe Z.

Résultats et observations

Dans un système produisant des oscillations non amorties, la fréquence de résonance naturelle d'une oscillation est calculée au moyen du rapport suivant :

$$\omega = \sqrt{\frac{k}{m}}$$

Pour les modèles 2D et 3D, on a mené deux types d'essais dynamiques : l'un pour déterminer la réponse impulsionnelle et l'autre, les fréquences naturelles.

Dans le modèle 2D planaire, la rigidité des ressorts était de 20 000 N/m dans les parties supérieure et inférieure des sangles d'épaules, et la masse du sac utilisé était de 25 kg. En considérant les composantes de force verticales des parties supérieure et inférieure des sangles comme des ressorts qui commandent les oscillations verticales du sac à dos, on a pu calculer la rigidité effective des

ressorts, pour la substituer à la fréquence de résonance naturelle dans l'équation. Dans ce cas, la fréquence naturelle estimative était de 4,8 Hz.

Lorsque le sac était relâché à partir de 2,5 cm au-dessus de sa position de repos, les résultats obtenus pour son déplacement vertical étaient typiques pour un système légèrement sous-amorti. On a également analysé une plage de fréquences de forçage à l'aide du modèle 2D, puis on a calculé le rapport entre l'amplitude de mouvement du sac et l'amplitude de la fonction de forçage, qu'on a représenté comme une fonction de la fréquence de forçage. Une crête modeste a été observée à une fréquence d'environ 8 Hz, qui est supérieure à la fréquence naturelle estimative de 4,8 Hz.

Le modèle 3D planaire comportait 4 ressorts (parties supérieure et inférieure, et gauche et droite). La rigidité de chaque ressort était de 20 000 N/m, et la masse du sac à dos utilisé pour le modèle était de 24,5 kg. Les composantes de force verticales des parties supérieure et inférieure des sangles ont été traitées selon une méthode similaire à celle utilisée pour le modèle 2D, sauf que les quatre ressorts devaient être inclus dans le calcul de la rigidité effective des ressorts. La valeur $K_{\text{Equivalent}}$ a été substituée dans l'équation à la fréquence de résonance naturelle afin d'estimer la fréquence naturelle du modèle 3D, qui était de 7,3 Hz.

Les coefficients d'amortissement de 4 et 5 Ns/cm, que l'on a jugé appropriés d'après les résultats de la modélisation 2D, ont été utilisés dans le modèle 3D. Lorsque, dans le modèle 3D, le sac était relâché à partir de 2,5 cm au-dessus de sa position de repos, les résultats obtenus pour son déplacement vertical étaient typiques pour un système sur-amorti.

On a analysé une plage de fréquence de forçage à l'aide du modèle 3D, puis on a calculé le rapport entre l'amplitude de mouvement du sac et l'amplitude de la fonction de forçage, qu'on a représenté comme une fonction de la fréquence de forçage. Les résultats indiquent que les fréquences de résonance étaient d'environ 2,5 Hz, puis de 5 Hz.

L'amortissement pour les modèles de sangle d'épaules supérieure et inférieure, de 4 et de 5 Ns/cm a été réduit à 4 et à 5 Ns/m respectivement. Lorsque l'amortissement a été réduit par un facteur de 100, le système a montré quelques oscillations initiales, suivi d'une lente décroissance dans la position verticale.

Tel que prévu, le modèle 3D légèrement amorti a affiché une fréquence naturelle dominante d'environ 5 Hz. Les autres fréquences de résonance plus basses ont été plus faciles à identifier, soit à 2 et à 2,5 Hz environ.

Conclusions

Deux modèles de sangle d'épaules pour le transport dynamique de charge ont été élaborés. Dans le modèle 2D, les coefficients d'amortissement devaient être beaucoup plus élevés que ceux du modèle 3D pour obtenir une séquence de déplacement du sac similaire. De plus, le comportement du modèle 3D ressemblait davantage à celui du système matériel.

La prochaine étape de développement du modèle consiste à intégrer un modèle de ceinture, qui est élaboré séparément³, au modèle 3D.

³ Hadcock, L., Master of Science Thesis for the School of Physical and Health Education, Université Queen's, Kingston, Canada. Tous droits réservés. Juillet 2002

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1.0 Introduction

The Phase I static biomechanical model used only pack geometry, a cylindrical shoulder shape, strap angles, tension and the friction ratio of T1/T2 straps to calculate the shoulder reaction forces (Stevenson et al., 1995). Lumbar reaction force was determined by the lean angle and the force needed to maintain static equilibrium. A waist belt was added in Phase II. This belt was modeled as two half cones where the slope of the cone represented the anatomical slope of the hips. The hoop stress equation was used to determine the net compressive force generated by tension in the waist belt. Net compressive force was then divided into lift and friction components. Rigby W.A. (2000) attempted to include additional pack elements (i.e. load lifter straps) in an improved model and validate measured waist reaction forces with a load cell. He found a wide range of frictional forces and variable suspension system compliance based on the measured strap tensions. His conclusion was that proper modeling of the pack-person interface forces was essential. This thesis served to point the way to a new modeling approach that treats the pack-person interface as a dynamic suspension system. It also pointed to the need to find better ways to determine the nature of the pack-person interface.

2.0 Purpose

The overall purpose of the DRDC dynamic biomechanical modeling research program is to improve the understanding of human load carriage capabilities and to understand the benefits of load carriage system design features. This body of work is specifically directed at developing a biomechanical analysis and load carriage system design tool for rapid iterative development of future load carriage concepts and prototypes.

By increasing our understanding of how dynamic loads are best borne by the body, we can learn what strategies are used by the human organism to minimize the effect of load-bearing on its health and mobility. It is known that different strategies dominate under different conditions. In short duration activities requiring high mobility and large upper body motions, a stiff suspension system and load carriage on the shoulder girdle are preferred. For long distances over flat terrain, loads are better supported by the pelvic girdle. Earlier static modeling work (Stevens et al, 1995) identified several useful biomechanical load carriage limits. This has lead to a recommendation of a maximum of 290 N compressive load on the shoulders and a maximum acceptable shear load in the lumbar area of 135N. This and similar knowledge can be applied by designers of load carriage equipment to evaluate the effect and utility of load carriage design features.

The current work begins this process of integrating mathematical modeling of human load carriage with the knowledge previously gained about human biomechanical and discomfort tolerances.

3.0 Methodology

Two dynamic modeling software packages were selected to permit multiple attempts at modeling the pack person interface. MSC Working Model is a two-dimensional (2D) planar analysis package and MSC Visual Nastran performs three-dimensional (3D) motion and stress analysis. The selected software both permit full user control of model geometry, inertial properties, have extensive libraries of existing dynamic elements for modeling constraints, allow the user to construct complex constraint equations and allow the user to input complex forcing functions. Both software packages are products of MSC Working Knowledge and share considerable overlap in the user interface and structure.

Features of the selected software are summarized in Table 1 below.

Table 1 Features of Selected Software

Software	Type	Additional Analysis	Web address
MSC Working Model®	Dynamic, Rigid Body Motion Planar – 2D modeler	No	http://www.krev.com/2002/welcome.html
MSC Visual Nastran 4D®	Dynamic, Deformable Bodies, 3D modeler	Linear Elastic Finite Element and Thermal Analyses	http://www.krev.com/products/wm2d_f01.html

4.0 2D Model Description

The 2D software, Working Model (WM), was used extensively in the initial stages of modeling to evaluate potential suspension characteristics for the shoulder straps. This provided a series of initial estimates for constraint values used subsequently in the 3D model.

4.1 Geometry – 2D

MSC WM software provides a limited drawing tablet to create the physical shape of objects. This is supplemented with the ability to import images from any CAD software that can export *.DFX file formats. These imported images have no physical properties and cannot interact in the analysis with other objects. Creation of a WM object with the correct profile that could be assigned physical properties

required a two step process. Previously, a 3 dimensional surface scan of the 50 percentile male mannequin torso had been made and the resulting shape file (IGES format) was imported into AutoCad®. A sagittal silhouette of this was created in AutoCad® and then exported back to the WM program as an image. The silhouette shape was then used as a drawing template within WM to create an object polygon. Finally, this body shaped polygon could be assigned material properties and could now interact with objects in the analysis. This process resulted in a consistent body profile across the 2 and 3D analyses.

4.2 Mass and Inertial Properties – 2D

A simple rectangular pack shape was created based in the geometry of the Clothe the Soldier (CTS) Pack tested in the December 2001 in a field trial held at Queen’s University, Kingston. Mass and inertial properties of a typical medium load condition (24.56 kg payload) were modeled. Figure 1 gives a summary of the CTS geometry used in the definition of the 2D and 3D pack models. The exact location of the center of gravity was controlled by modeling the bag portion of the rucksack as two objects. The first object had the correct geometry but was given a mass of only 1 kg. The second object was a 25mm diameter circle located at the desired COG position with a mass of 23.56 kg.

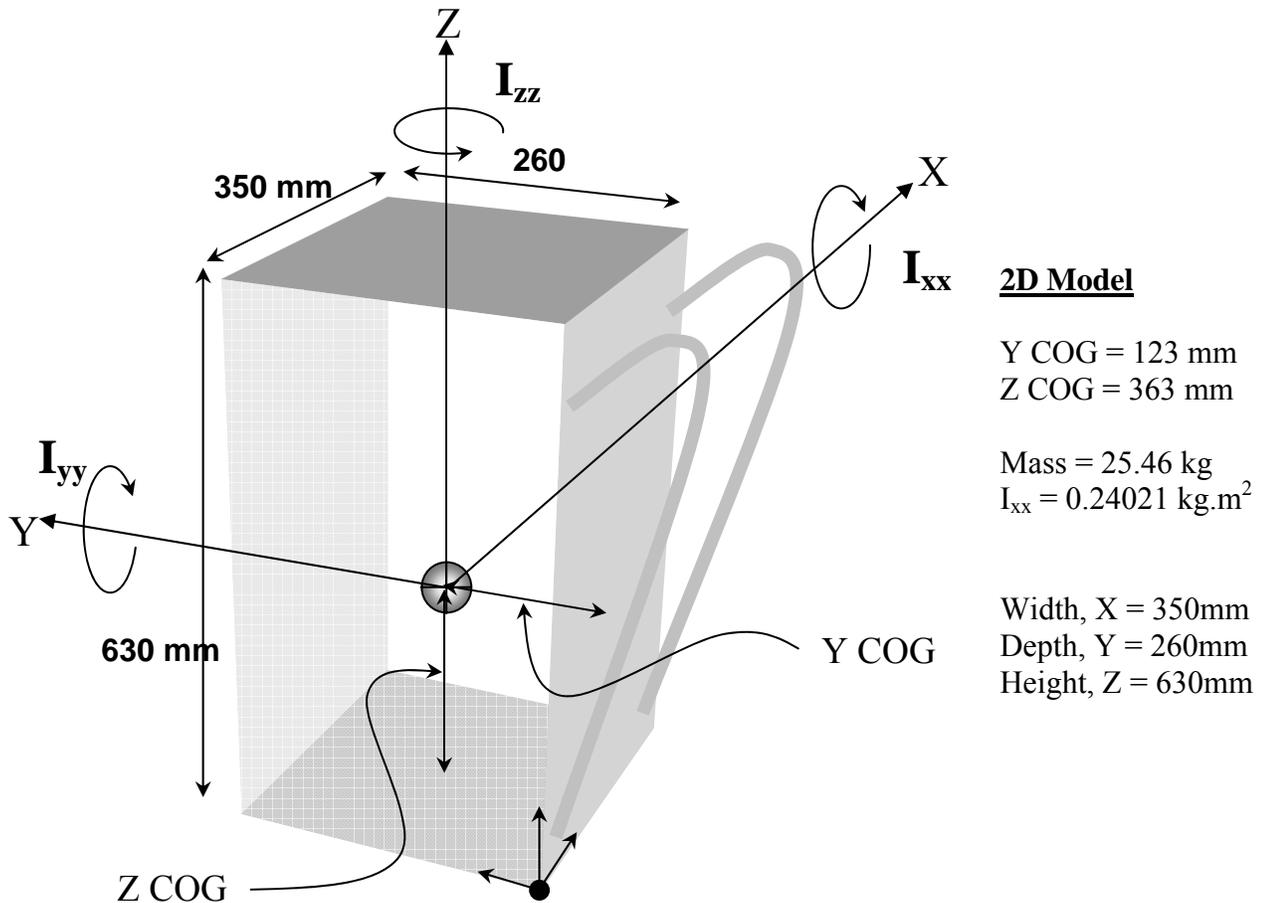


Figure 1 Clothe the Soldier Rucksack Parameters

WM allows an explicit definition of the moment of inertia and so this was defined to match those measured during the field trial. The centre of gravity (CoG) and moments of inertia of the pack were determined using the trifilar pendulum developed under PWSC Contract W7711-0-7632-02. The COG was located on the midline of the pack at $x = 175\text{mm}$, $y = 123\text{mm}$, $z = 363\text{mm}$, measured from the lower right corner indicated in Figure 1.

4.3 Material Properties – 2D

With the exception of the mass properties, both the pack and torso were given nominal material properties to reduce the complexity of the system interactions. Some of the interaction factors, such as friction between the body and the pack, are non-linear and discontinuous in nature resulting in complex interaction dynamics. These factors can obscure the effect of varying the suspension element parameters and so the material properties were nominally modeled as having no friction and a moderate coefficient of restitution (0.5). A summary of the 2D model properties appears in Table 2.

Table 2 Summary of 2D Model Material Properties

Property	Pack Model	Body Model
Mass	25 kg	70 kg
Planar moment of Inertia - I_{xx}	17772.3 N.m.s ²	Default
Friction coefficient Static	0	0
Friction coefficient Dynamic	0	0
Coefficient of restitution	0.5	0.5

4.4 Shoulder Straps - 2D

Initially the shoulder straps were modeled as a Voigt-Kelvin visco-elastic model with a stiff linear spring in series with a linear damper. Rope elements were added later between the springs and the attachment points on the body to cause the strap model to only exert a force when in tension. An illustration of the resulting strap model is shown in Figure 2.

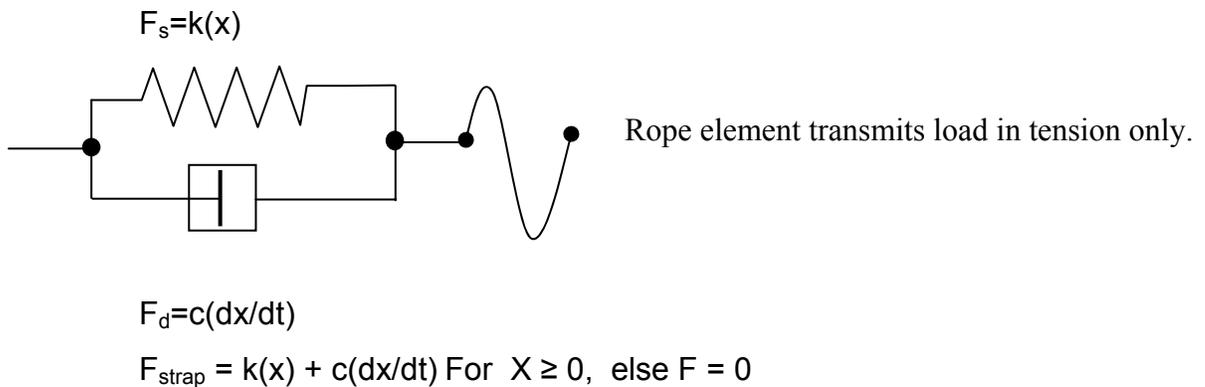


Figure 2 Shoulder strap mathematical model

Table 3 2D Model Shoulder Strap Parameters

Location	Spring Constant K	Damping Coefficient C
Upper Shoulder strap	20 000 N/m	5 N s/m
Lower Shoulder strap	20 000 N/m	4 N s/m

4.5 Forcing Function – 2D

The torso was constrained to move along a path fixed in the vertical (+/- Z) direction. Model motion is applied in two stages. In stage one, the model is allowed to achieve an equilibrium condition under the effects of gravity, material

properties and contact interface constraints. During this phase, the torso model is held in position (using a fixed length actuator element) and the pack model shifts to an equilibrium position in contact with the back. This phase is completed by $t = 1.2$ seconds.

At $t \geq 1.2$ seconds, a second position actuator element takes over and ramps up to a vertical sinusoid of ± 25 mm amplitude. This forcing function has the form:

$$((1-1/\exp(t))*A*\sin(\omega*t) + L$$

where:

$((1-1/\exp(t)) =$ exponential ramp up

A = amplitude of vertical displacement

$\omega =$ frequency of oscillation

L = initial neutral position

4.6 Final Model – 2D

The final 2D model is shown below in Figure 3.

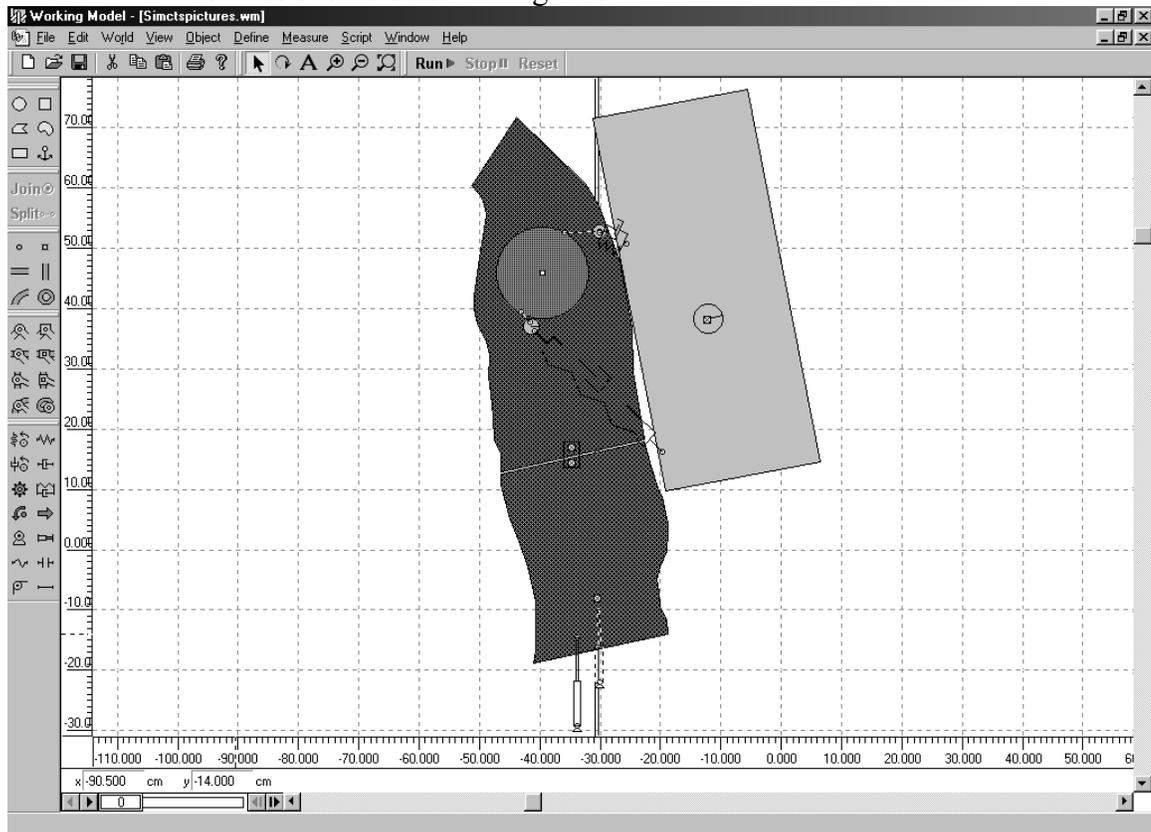


Figure 3 2D Model of the CTS pack in Working Model 2D

5.0 3D model description

The simple 2D model readily solved multiple combinations of spring parameters and damping coefficients and was used to determine the appropriate range of values to be used for these parameters in the 3D model.

5.1 Geometry – 3D

A 3 dimensional surface scan of the 50 percentile male mannequin torso created previously (Reid et al., 2002) was imported into Visual Nastran 4D® (VN4D). The torso was treated as a solid object by VN4D for the purposes of this analysis. The image file can also be imported into CAD programs such as SolidWorks® and AutoCad® which will allow us to divide the torso into components to determine such things as shear load between the upper and lower torso or the contact load in a particular region.

The left and right sides of the mannequin body are not perfect mirror images, which created a slight asymmetry in the 3D model. This resulted in a small asymmetry in the exact locations of the right and left shoulder strap attachment points. The effect of this on the numerical solutions is negligible.

VN4D has a limited capacity to create simple solid objects in the drawing pallet. Most complex objects must be created in a CAD program and imported into VN4D. For this analysis, a rectangular representative pack was created based on the geometry of the CTS pack and consistent with the pack model used in the 2D analyses.

5.2 Mass and Inertial Properties – 3D

Displacement of the torso model is a controlled variable in these analyses; therefore torso mass and inertial properties do not enter into the dynamic analysis. Mass of the torso was set at 70 kg, which is the mass of a 50 percentile male, and was assumed to be evenly distributed throughout the body. Moments of inertia were calculated by VN4D from the geometry and mass distribution.

Methodology for measuring these parameters is described in Section 4.2. VN4D permits user specification of the location of the centre of gravity and all moments of inertia for objects, without requiring a mass distribution definition. This allowed results for I_{xx} , I_{yy} and I_{zz} measured with the trifilar pendulum to be entered directly into the property definition of the pack model.

5.3 Friction and Material Properties – 3D

Specific values for the coefficient of friction and coefficient of restitution are not readily available for the pack and clothing over skin conditions. The 3D model was kept consistent with the 2D model. This approach reduced the complexity of the interface interactions which allowed controlled characterization of the shoulder suspension system model. The pack and torso were defined as frictionless objects with a coefficient of restitution of 0.5.

A summary of the 3D model properties appears in Table 4.

Table 4 Summary of 3D Model Material Properties

Property	Pack Model	Body Model
Mass	25 kg	70 kg
Planar moment of Inertia - I_{xx}	0.240205 kg.m ²	Default
Planar moment of Inertia - I_{yy}	0.223097 kg.m ²	Default
Planar moment of Inertia - I_{zz}	0.125468 kg.m ²	Default
Friction coefficient Static	0	0
Friction coefficient Dynamic	0	0
Coefficient of restitution	0.5	0.5

5.4 Shoulder Straps 3-D

Consistent with the 2D model, shoulder straps in the 3D model were modeled as a Voigt-Kelvin visco-elastic material with a stiff linear spring in series with a linear damper. VN4D permits a user to change the governing equations of constraints and thereby create any desired spring/damper combination. As well, a user can incorporate logical tests into the analysis and apply different equations under different conditions. Shoulder straps do not apply a force when their current length is shorter than their natural length. Therefore, for each step of the analysis, the 3D strap length was evaluated and a strap force was applied only if the current length indicated the strap was in tension.

The final form of the constraint equation was:

$$\text{If } |\vec{s}| \geq \text{resting length}$$

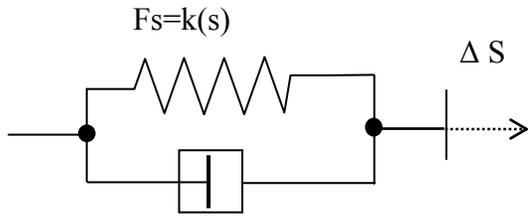
$$\text{Where } |\vec{s}| = \sqrt{(x^2 + y^2 + z^2)}$$

$$\text{Then } F(t) = ks + c\dot{s}$$

$$\text{Else } F(t) = 0$$

$$k = \text{spring stiffness (N / m)}$$

$$c = \text{damping coefficient } t \text{ (N.s / m)}$$



$$F_d = c(ds/dt)$$

$$F_{\text{strap}} = k(s) + c(ds/dt) \text{ For } \Delta S \geq 0, \text{ else } F = 0$$

Figure 4 3D Shoulder Strap Model

Table 5 3D Model Shoulder Strap Parameters

Location	Spring Constant K	Damping Coefficient C
Upper Shoulder strap	20 000 N/m	500 N.s/m
Lower Shoulder strap	20 000 N/m	400 N.s/m

5.5 Forcing Function – 3D

For this study, motion of the torso was constrained to the vertical (Z) axis. Torso displacement was governed by the sinusoidal forcing function used in the 2D model. The attachment points for the shoulder straps were positioned equidistant from the torso's centre line to preclude inducing out of plane forces. Some asymmetry was present in the 3D mannequin form at the underside of the shoulder, which meant that the lower portion of the strap could not be positioned at the same vertical position on the left and right sides of the body. This resulted in slightly different initial lengths of the springs, which in turn required that logical tests for the "in Tension condition" be written individually to match these different initial lengths.

The 3D torso was modeled as rigidly fixed to an actuator constrained to move along the Z axis. All applied forces in this model were in the X-Y plane, therefore pack motion was constrained to move in the vertical (+/- Z) direction. As in the 2D analysis, the analysis proceeded in two stages. In stage one, the model was allowed to achieve equilibrium under the effects of gravity, material property responses and contact interface constraints. During this phase, the torso rests in position and the pack is allowed to settle into an equilibrium position on the back.

At $t \geq 1.2$ seconds, the actuator becomes active and the motion ramps up to a vertical sinusoid of +/- 25 mm amplitude. This forcing function has the form:

$$((1-1/\exp(t))*A*\sin(\omega*t))$$

where:

$((1-1/\exp(t)) =$ exponential ramp up

A = amplitude of vertical displacement

$\omega =$ frequency of oscillation

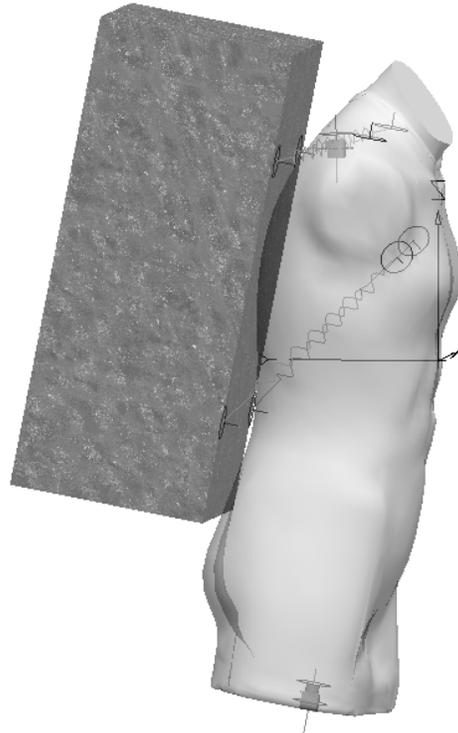


Figure 5 3D model of the CTS Pack in Visual Nastran 4D

6.0 Results and Discussion

In an undamped oscillating system, the resonant natural frequency of oscillation is determined by the following relationship:

$$\omega = \sqrt{\frac{k}{m}}$$

Determining this frequency for the model (and ultimately the physical system is important since:

- i. It is relatively easy to get an object to vibrate at its' natural frequency and hard to force it to oscillate at other frequencies.
- ii. Oscillating bodies will select their natural frequencies when excited by complex forcing functions and will respond to them, effectively filtering out other frequencies.

iii. Most complex objects have multiple resonant frequencies.

In a damped system, there are three possible conditions:

- i. Under damped, when the oscillations gradually decrease in amplitude over time.
- ii. Critically damped, when the damping coefficient equals the undamped natural frequency of the system. In this case there is no overshoot and the system reaches the rest position without oscillating.
- iii. Over damped, when the system does not oscillate but reached the rest position more slowly than when critically damped.

For the 2D and 3D models, two types of dynamic tests were conducted to determine the impulse response and the natural frequency of the models. An impulse response test demonstrates the damping level. The amplitude ratio analysis test demonstrates the number and value of a system's natural frequencies.

6.1 2D Model - Estimate of Natural Resonant Frequency

The 2 D planer model had a spring stiffness of 20000 N/m in the upper and lower shoulder straps and the mass of the backpack was modelled as 25 kg. The model is not a simple oscillator since the strap springs are acting at an angle to the vertical motion we are tracking. The vertical force components of the upper and lower straps, shown in Figure 6, can be treated as the vertical springs controlling the vertical oscillations of the pack. This allows calculation of an effective spring stiffness that can be substituted into the equation for the natural resonant frequency.

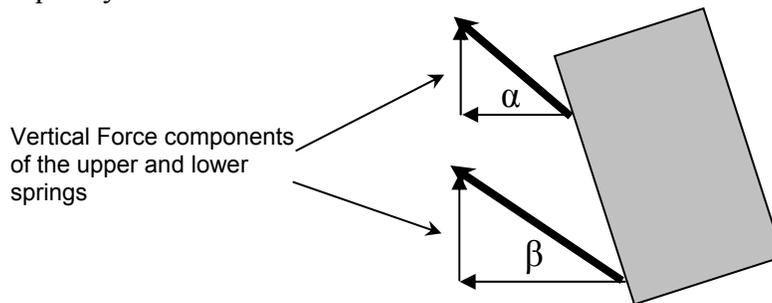


Figure 6 Vertical Force on the Pack Generated by the upper and lower straps

An estimate of the effective stiffness was made as follows:

$$K = 20000 \text{ N/m}$$

$$\alpha = 24 \text{ degrees}$$

$$\beta = 48 \text{ degrees}$$

$$K_{\text{Equivalent Upper}} = K \sin\alpha, K_{\text{Equivalent Lower}} = K \sin\beta$$

The springs act in parallel therefore they are additive:

$$K_{\text{Equivalent}} = K (\sin\alpha + \sin\beta)$$

$$K_{\text{Equivalent}} = 20000(0.40514 + 0.73728) = 22848 \text{ N/m}$$

Substituting this into the equation for natural frequency (in cycles/second) gives;

$$\omega_{2D} = \sqrt{\frac{k}{m}} = \sqrt{\frac{22848 \text{ N/m}}{25 \text{ kg}}} \div 2\pi \text{ rad/s} = 4.8 \text{ c/s}$$

6.2 2D Model - Impulse Response Test

The pack was lifted 2.5 cm vertically from the rest position and released under the influence of gravity. This supplied an impulse to the pack and the vertical displacement was recorded over time. The result shown in Figure 7 is typical for a mildly under damped system that oscillates 3-4 times before coming to rest.

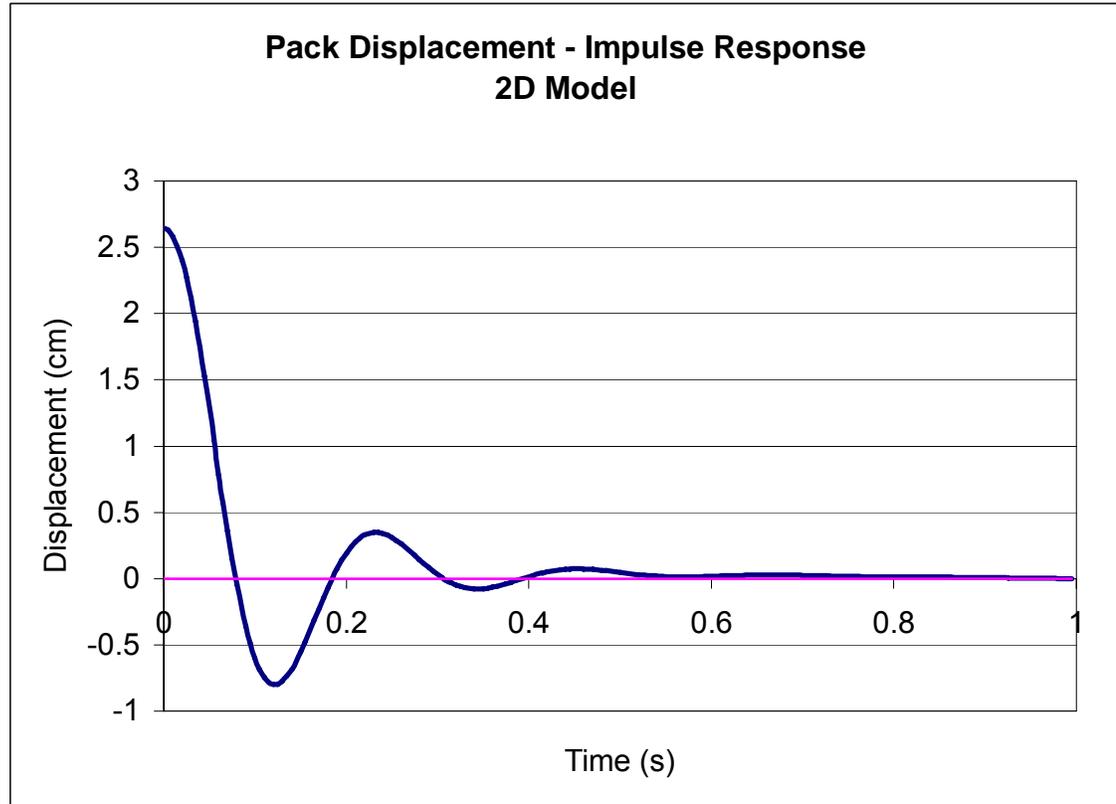


Figure 7 Impulse Response 2D Model

6.3 2D Model - Amplitude Ratio plot

A range of forcing function frequencies was analysed with the 2D model. From these analyses, the ratio of the pack motion amplitude to the amplitude of the

forcing function was calculated and plotted as a function of the forcing frequency in Figure 8. A modest peak is observed at approximately 8 Hz, higher than the estimated natural frequency of 4.8 Hz.

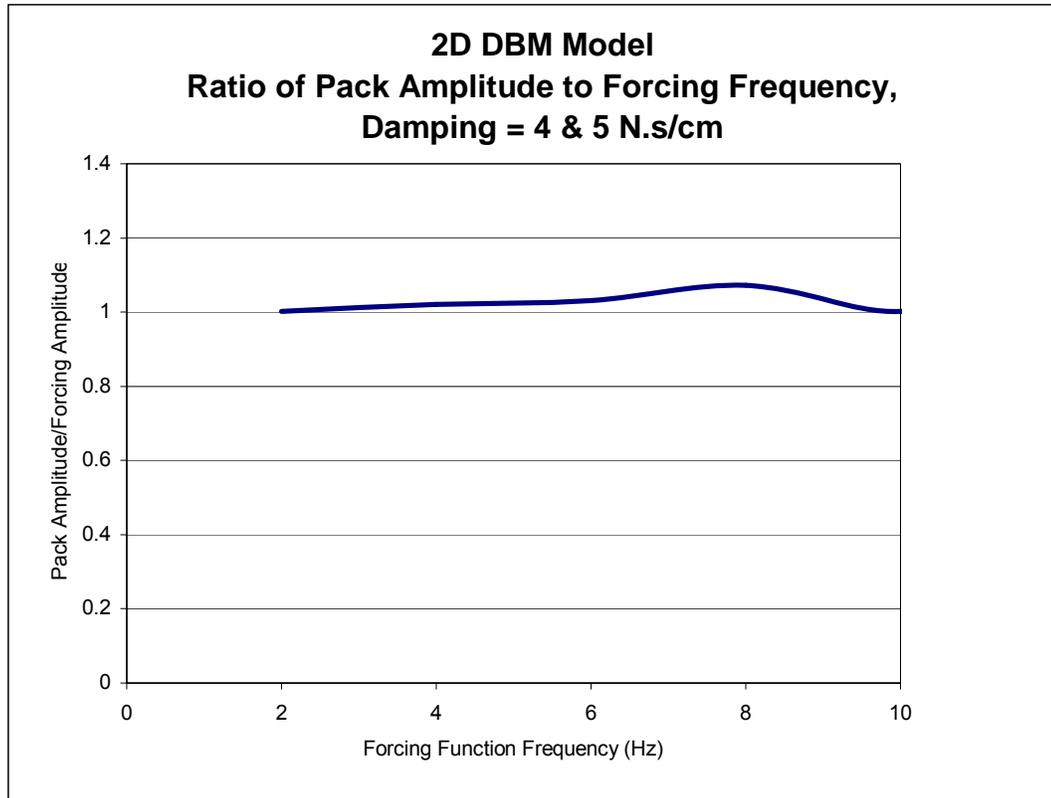


Figure 8 Ratio of Pack Amplitude to Forcing Frequency 2D Model

6.4 3D Model - Estimate of Natural Resonant Frequency

The 3 D planer model had 4 springs (upper and lower, left and right side) each with spring stiffness of 20000 N/m. The mass of the backpack was modelled as 24.5 kg. Again, this model is not a simple oscillator since the strap springs are acting at an angle to the vertical motion. The vertical force components of the upper and lower straps were treated similarly to the method described in Section 6.1, with the exception that there are four springs to be included in effective spring stiffness calculation. This $K_{Equivalent}$ was substituted into the equation for the natural resonant frequency to estimate the 3D model natural frequency.

An estimate of the effective stiffness was made as follows:

$$K = 20000 \text{ N/m}$$

$$\alpha = 27.3 \text{ degrees}$$

$\beta = 60$ degrees

$K_{\text{Equivalent Upper}} = K \sin\alpha$, Number of upper springs = 2

$K_{\text{Equivalent Lower}} = K \sin\beta$, Number of upper springs = 2

The 4 springs act in parallel therefore they are additive:

$K_{\text{Equivalent}} = 2K (\sin\alpha + \sin\beta)$

$K_{\text{Equivalent}} = 40000(0.4587 + 0.8712) = 53194 \text{ N/m}$

Substituting this into the equation for the natural frequency gives;

$$\omega_{2D} = \sqrt{\frac{k}{m}} = \sqrt{\frac{53194 \text{Ns} / \text{m}}{24.5 \text{kg}}} \div 2\pi \text{ rad} / \text{s} = 7.3 \text{ c} / \text{s}$$

6.5 3D Model - Impulse Response Test

The damping coefficients (400 and 500 N.s/m) that were determined to be appropriate from the results of the 2D model were used in the 3D model. The pack model was displaced 2.5 cm vertically from the rest position and then released. This supplied an impulse and the vertical displacement was recorded. Results are shown in Figure 9 and are typical for an over-damped system. In this type of system, the time constant of the positional decay is function of the damping coefficient and system mass.

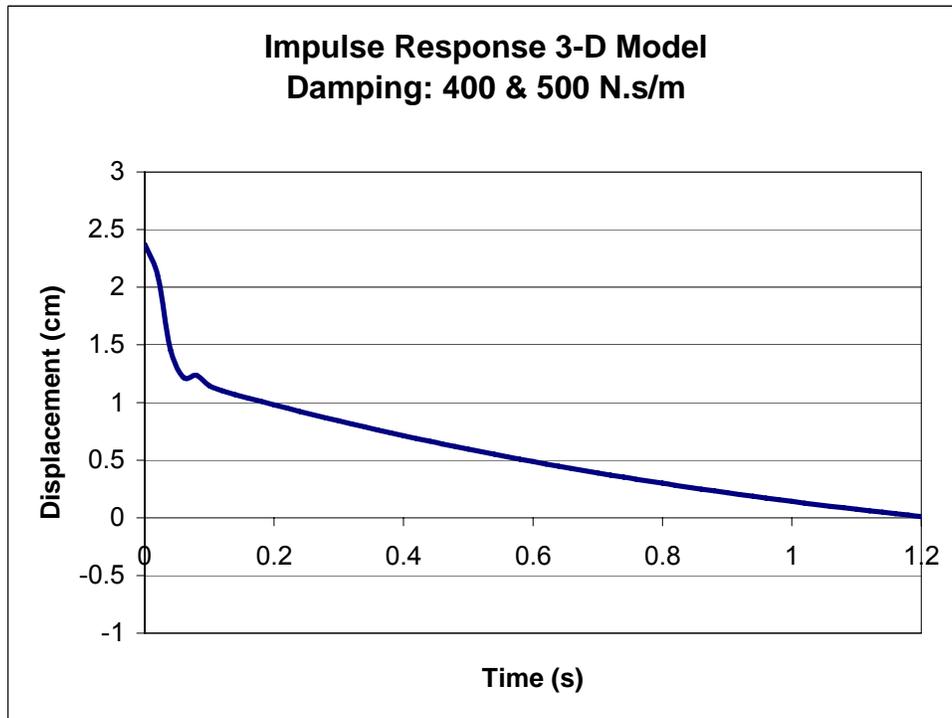


Figure 9 Impulse Response - 3D Model

6.6 3D Model - Amplitude Ratio plot

A range of forcing function frequencies was analysed with the 3D model. From these analyses, the ratio of the pack motion amplitude to the amplitude of the forcing function was calculated and plotted as a function of the forcing frequency in Figure 10.

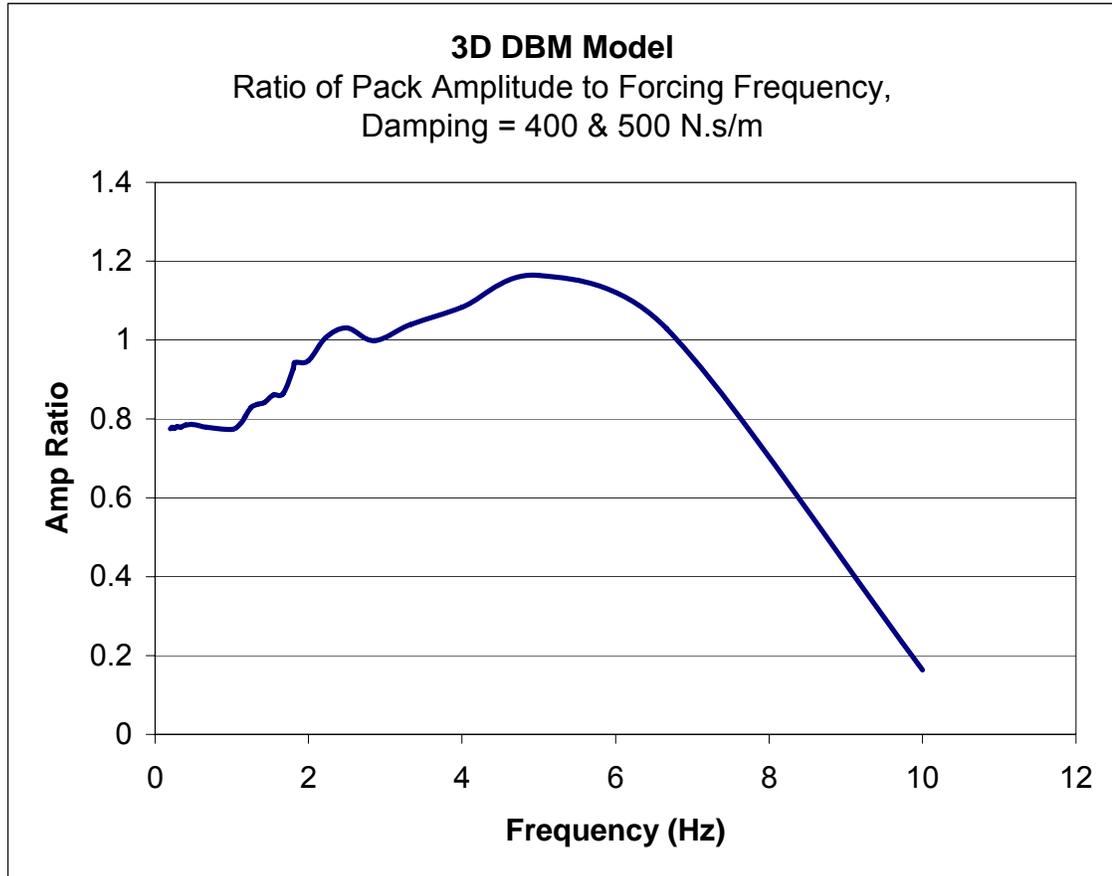


Figure 10 Ratio of Pack Amplitude to Forcing Frequency 2D Model

These results indicate several resonant frequencies at approximately 2.5 Hz and again at 5 Hz.

6.7 3D Model with minimal damping - Impulse Response Test

Due to the over-damped response that the 3D model exhibited in the impulse response test, a second analysis was undertaken with reduced damping coefficients. Damping was reduced by a factor of 100 from 400 and 500 Ns/m in the upper and lower shoulder strap models to 4 and 5 Ns/m respectively. Results from this work are shown in Figure 11.

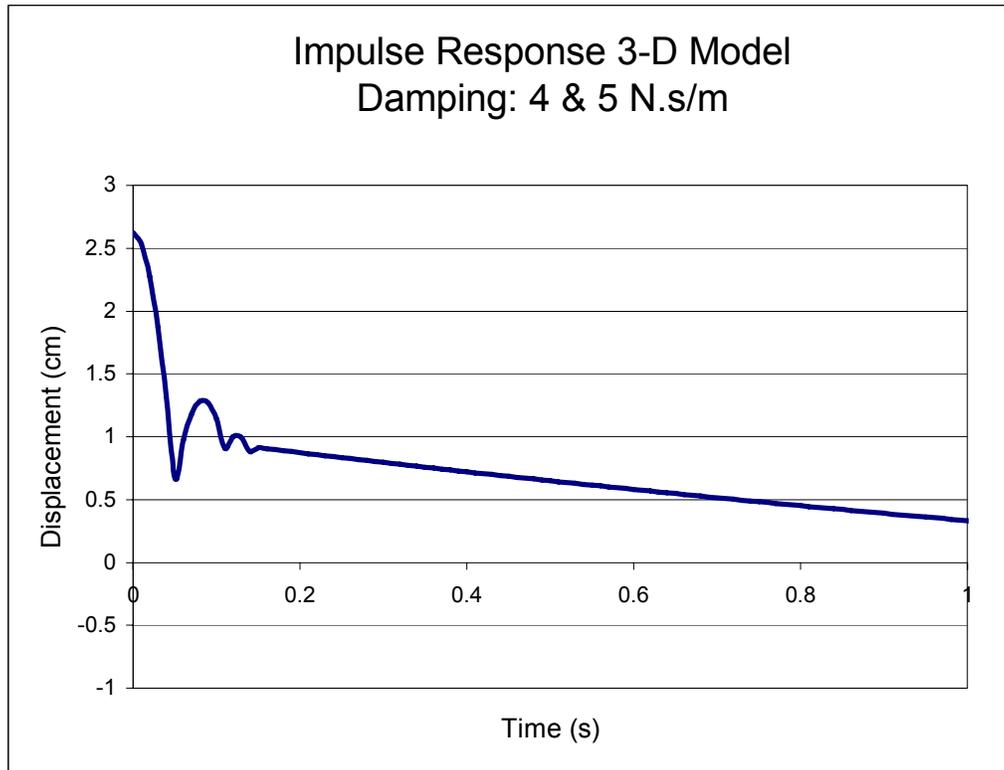


Figure 11 3D Model with minimal damping - Impulse Response

With the damping reduced by a factor of 100, there are some initial oscillations of the system followed by a slow decay in the vertical position. This decay behaviour should be explored to determine the cause, as it is not typical of a simple damped oscillator.

6.8 3D Model with minimal damping - Amplitude Ratio plot

The ratio of the pack motion amplitude plotted against the frequency of the forcing function for the minimally damped system is shown in Figure 12. As expected, the model displays a dominant natural frequency at approximately 5 Hz. The other lower resonant frequencies are more easily identified as occurring at approximately 2 and 2.5 Hz.

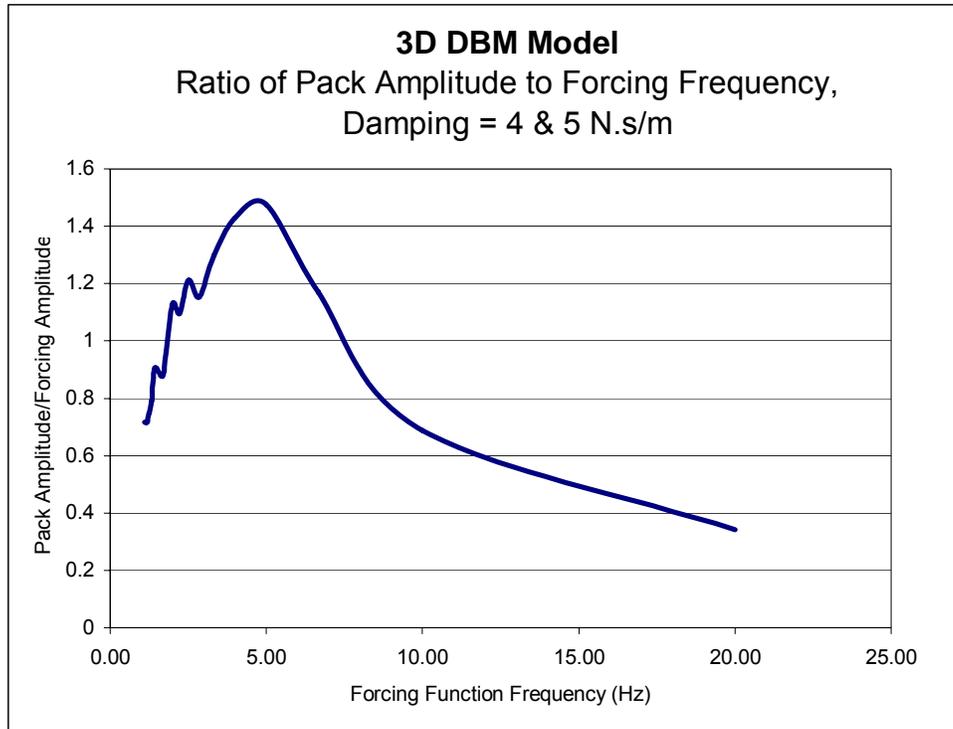


Figure 12 3D Model with minimal damping – Amplitude Ratio

7.0 Conclusions

Two shoulder carried dynamic load carriage models have been developed. Both the 2D and 3D models are based on the geometry and mass distribution of a loaded Canadian CTS rucksack and interact with a body shape representing a 50 percentile male Canadian soldier.

Although the two models presented were developed to be similar; they have the same interface interaction properties, similar geometric parameters, the motion was constrained to the same plane, the shoulder straps were mathematically analogous and the forcing functions were identical. The behaviour of the models did not prove to be interchangeable. The 2D model required much higher damping coefficients to bring about a pack displacement pattern similar to that of the 3D model. Nylon webbing and its attachment points on a pack are expected to provide relatively small amounts of damping to the system. The 3D model behaviour was more consistent with the physical system.

This leads to the conclusions that 2D modelling of this three dimensional dynamic system would be limited and some of the interactions of the human body with the pack suspension system could not be captured with a 2D model.

8.0 Next Steps

The next stage in model development is to integrate the waist belt model being developed separately (Hadcock, L., 2002) into the 3D model. Both the 2 and 3D pack models developed to date are based on the loaded packs carried in a human load carriage trial. During these trials, torso and pack accelerations, as well as strap force data, was recorded during a series of high mobility activities. A wide range of activities was recorded which created a data bank of different complex excitation functions. VN4D is able to import data and can be made to use this 3D torso accelerometer data as a forcing function for the torso in the model. As the real-time acceleration history of the pack motion was captured simultaneously during the human trials, model response can be benchmarked against this. A portion of this data will be used to refine the model response while a portion will be reserved to use as a validation tool.

9.0 References

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- Stevenson, J.M. Bryant, J.T. dePencier, R.D., Pelot, R.P., and Reid, J.G. (1995) Research and Development of an Advanced Personal Load Carriage System: Phase I. PWGSC Contract# W7711-4-7225/01-XSE. DCIEM-CR-2001-093
- Reid S.A., Bryant, J.T, Stevenson, J.M. (2002) User's Manual V 3.0: Dynamic Load Carriage Compliance Tester Automated Cell: Phase II C&D. PWGSC Contract #W7711-0-7632-02. Report to DRDC by Queen's University, 28 pages.

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(U) The overall purpose of the DRDC research program on dynamic biomechanical modeling is to improve the understanding of human load carriage capabilities and to understand the benefits of load carriage system design features to human health and mobility. Earlier phases of the dynamic biomechanical model have led to a new modeling approach that treats the pack-person interface as a dynamic suspension system. In the current study, both 2D and 3D dynamic modeling software packages were selected to permit multiple models of the pack person suspension characteristics. The selected software both permit full user control of model geometry, inertial properties, have extensive libraries of existing dynamic elements for modeling constraints, allow the user to construct complex constraint equations and allow the user to input complex forcing functions. For both the 2D and 3D models, two types of dynamic tests were conducted to determine the impulse response and the natural frequencies. For the 2D model, the impulse response test showed typical results for a mildly under-damped system with the amplitude ratio plot showing a modest peak at approximately 8 Hz, higher than the estimated natural frequency of 4.8 Hz. On the other hand, the impulse response test for the 3D model gave a vertical displacement typical of an over-damped system and an amplitude ratio plot with several resonant frequencies at approximately 2.5 Hz and again at 5 Hz. With the damping reduced by a factor of 100, there were some initial oscillations of the system followed by a slow decay in the vertical position and as expected, the minimally damped 3D model displayed a dominant natural frequency at approximately 5 Hz. Overall, the 2D model required much higher damping coefficients to bring about a pack displacement pattern similar to that of the 3D model. In addition, the 3D model behaviour was more consistent with the physical system. The next stage in model development is to integrate a waist belt model (Hadcock, 2002) being developed separately into the 3D model.

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(U) Load carriage; Dynamic Biomechanical Model; MSC Working Model; 2D Model; 3D Model; Dynamic Modelling Software

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