STUDY OF A NOVEL MATERIAL SOLUTION FOR VIBRATION ISOLATION

Thesis submitted in fulfilment of the requirement for the degree of
Doctor of Philosophy

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School of Engineering, Cardiff University- October 2016

PAPERS PRODUCED FROM THIS THESIS

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ABSTRACT

Vibration isolation is an important requirement for many engineering systems. In particular, in the context of vibration isolation for light-weight automotive vehicles that exhibit wide variation in sprung mass, several limitations are associated with passive isolation systems. Such passive systems cannot obtain wide variations in the suspension parameters which required for reliable performance. While these technical drawbacks can be overcome by implementing active systems, these are associated with an increase in complexity, cost and potentially negative impact on reliability. In this context, composite fluid materials, which combine different components in a way that enhances an isolator's performance, could represent a possible alternative approach with promising potential. However, the application of composite fluid materials for vibration isolations is still an underdeveloped area. The composite fluid material that is the subject of this research is referred to as Foam Filled Fluid (FFFluid). It is composed of three components, namely compressible (foam) particles, a viscous carrier fluid and a packaging material. This composite material has recently been investigated for applications in impact energy management but is not understood in anti-vibration application.

Thus, the objective of this research was to understand the mechanisms, to characterise design parameters and to predict the responses of such composite material when used for vibration isolation systems. A theoretical understanding of the working principle for a FFFluid-based isolator was first achieved. Then, experimental work was conducted to assess the performance of such a device. The characterisation of the composite material was carried out via a systematic study; this study was then validated by an experimental

programme based on a Design of Experiments approach. Finally, empirical prediction models of the system were extracted by analysing the obtained data statistically.

The conducted research shows that a FFFluid-based isolator possesses several advantages over commonly used existing solutions. Its main benefit is the potential capability of adjusting stiffness and damping coefficients by changing one component or more of this composite material. It was shown that increasing the volume of the composite material led to increased stiffness and damping coefficients. Besides, increasing the ratio of fluid in the mixture caused to increase the stiffness coefficient. The most important parameters that have an influence on the response of FFFluid were: the size of foam, the ratio of foam to fluid, volume of the material and fluid viscosity. Therefore, empirical models were established based on these parameters, the accuracy of these models were 85%

Through the study of this novel material, the application of the FFFluid concept as a vibration isolator solution was studied. In practice, the design parameters of such a system could be adjusted through a control mechanism, to provide an adaptive solution. This could represent a suitable means to bridge the gap between passive and active suspension system in the context of vibration isolation for light-weight vehicles.

LIST OF PUBLICATIONS

1- Foam Filled Fluid (FFFluid) Isolator. Haithem Elderrat, PGR conference, Engineering School, Cardiff University- 11/11/2013. [Poster].

2- Investigation of the Foam Filled Fluid Technology for Anti-Vibration Devices. Haithem Elderrat, Huw Davies, Emmanuel Brousseau. *International Journal of Structural Analysis & Design – IJSAD*- Volume 1: Issue 3, ISSN: 2372-4102, Publication Date: 30 September 2014.

3- Improving the Exploitation of Fluid in Elastomeric Polymeric Isolator- Haithem Elderrat, Huw Davies, Emmanuel Brousseau, *International Journal of Mechanical, Aerospace, Industrial, Mechatronic and Manufacturing Engineering* Vol:9, No:8, 2015.

4- The Characterisation of a Foam Filled Fluid Vibration Isolator- Haithem Elderrat, Huw Davies, Emmanuel Brousseau. *International Journal of Earthquake Engineering– IJE,* Volume 2: Issue 1. Publication Date: 30 April 2015.

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NOTATIONS

Nomenclature	Meaning	Unit
A	Effective area of the system.	m^2
$\overline{A}_1$	Average value of factor A at L level.	-
$\alpha_0, \alpha_1, \ldots, \alpha_n$	Linear coefficients.	-
$\alpha_{11}, \ldots \alpha_{1n}$	Quadratic coefficients.	-
A_{opt}	Average performance for parameter A at optimum level.	-
B_{opt}	Average performance for parameter B at optimum level.	-
C_{opt}	Average performance for parameter C at optimum level.	-
D_{opt}	Average performance for parameter D at optimum level.	-
c	Damping coefficient.	$N.s/m$
d	Diameter.	m
D	Absorbed energy.	$N.m$
E_s	Young's modulus of material.	N/m^2
f	Frequency.	Hz
f_A	Degree of freedom of factor A.	-
f_e	Degree of freedom of error.	-
f_T	Total degree of freedom.	-
F_t	Transmitted force.	N
F_0	Applied force.	N
F	Variance ratio.	-

F_Y	The value of the F-test.	-
F_{Fr}	Friction force.	N
g	Gravitational acceleration.	m/s^2
k	Stiffness coefficient	N/m
L	Level of factors.	-
m	Mass.	kg
M	Number of factors.	-
MS_E	The residual mean squares.	-
MS_Y	Regression mean squares.	-
N	Total number of experiments.	-
P_1	Pressure.	N/m^2
P_0	Initial pressure.	N/m^2
p_{at}	Atmosphere pressure.	N/m^2
P_Y	Significance F.	-
r	Frequency ratio.	-
R^2	Coefficient of determination.	-
S_A, S_B, S_C, S_D	Variance of each factor.	-
S_ε	Standard error.	-
$S_{\varepsilon i}$	Standard error for coefficient.	-
SS_E	Sum of the squared error.	-
SS_Y	The regression sum of squares	-
S_T	Total variation.	-
t	Time.	sec
T	Grand total of all results.	-

T_d	Motion response.	-
T_f	Force transmissibility.	-
T_r	Thickness.	M
T_Stat	The value of the T-test.	-
u	Displacement.	M
V_0	Initial volume.	m^3
V_1	Volume.	m^3
X	Amplitude of the response.	m
x	Displacement.	m
$\dot{x}$	Velocity.	m/sec
$\ddot{x}$	Acceleration.	m/sec^2
Y	Output parameter.	-
Y_N	Output of N trial.	-
Y_{opt}	Performance at optimal condition.	-
Z	Number of independent variables in a regression model.	-
β	Significance level of regression model.	-
φ	Phase angle.	$Rad.$
ε	Residual.	-
σ_{el}^*	Elastic stress.	N/m^2
ρ^* /ρ_s	Relative density of foam.	-
ρ^*	Density of the foam.	kg/m^3
ρ_s	Density of material that made foam.	kg/m^3
δ_{st}	Static deflection under constant force.	m
ω_n	Natural frequency of structure.	Rad/sec

ω	Frequency of excitation.	Rad/sec
γ	Constant of gas.	-
ζ	Damping ratio.	-

ABBREVIATIONS

Abbreviation	Meaning
ANOM	Analysis of mean.
ANOVA	Analysis of variance.
C.F	Correction factor.
CUSP	Cardiff University Structural Performance.
DoE	Designs of Experiments.
DoF	Degree of Freedom.
EV	Electric vehicle.
ER	Electrorheological fluid.
F2MC	Fluidic flexible matric composite.
FFFluid	Foam Filled Fluid.
FFFoam	Fluid Filled Foam.
GFRP	Glass fibre reinforced polymer.
SALi	Shock absorbing liquid.
SDOF	Single degree of freedom.
OA	Orthogonal Array.
OFAT	One-Factor-At-Time.
PS	Polystyrene.
PE	Polyethylene.
MR	Magnetorheological fluid.

CHAPTER 1

INTRODUCTION

1.1 Background

One of the first characteristics that is noted by riding a vehicle is how the vehicle reacts to road excitations and vibrations. A vehicle's reaction is mainly influenced by the suspension system. Such a system is designed to maintain contact between vehicle's tyres and the road (road handling) and to isolate the frame of the vehicle from road disturbances (ride comfort). Also, it has to carry a static load of the vehicle. The suspension system mainly consists of stiffness and damping elements that connect a vehicle to its wheels. These elements react in opposition to the deflection caused by vibrations in order to control such deflection. Hence, the vehicle is isolated from road disturbances.

Generally, a softer damper provides a more comfortable ride compared to a stiffer one which provides better stability and thus, better road-handling [1]. Classical automotive suspension systems are passive, which means stiffness and damping properties are fixed during the design stage to achieve optimum performance for the intended application. Therefore, a compromise is always made between designing for two such opposing goals.

The passive suspension system for quads lightweight vehicles (L-category vehicle), whose their weights are less than 1000 pound (450kg) [2, 3], shows another limitation due to high variations in the payload. The ratio between the laden and unladen weight of lightweight vehicles could reach up to 50%. In such vehicle, softer spring is required during unladen weight of lightweight vehicle compared to stiffer one that is need during full laden of this vehicle.

Suspensions system of lightweight vehicles subject to a broad range of excitation conditions due to the relatively large variation in static load (mass) as well as variation in road disturbance. Hence, it becomes more challenging for passive systems to provide a high comfort for various static loads, and handling for various road disturbance. This means that ideally both variable spring and variable damping settings are desired for isolating the body mass. Therefore, there are a greater challenges in selecting suitable suspension characteristics [4]. In particular, Sharp and Hassan [1] showed that wide variations in the suspension parameters are required to obtain reliable performance over many different road surfaces.

At the same time, increased competition in the automotive market has forced companies to research alternative solutions to passive suspension systems. In order to improve handling and comfort performance, active, semi-active and adaptive systems are being developed instead of a conventional fixed spring and damper system.

In the case of active systems, actuators and sensors are used with stiffness and damping elements to react against unwanted vibration. Therefore, stiffness and damping coefficients of such isolation system could be adjusted. This approach can usually lead to a high level of performances for isolating vibration. However, active systems are not preferred solution especially for lightweight vehicles; this is due complexities in design and the high cost and weight [5, 6]. For example, it noted that the Nissan Infinity active suspension was 90kg heavier than a conventional suspension [7].

Semi-active systems have been developed to take advantage of the some of the positive aspects of both passive and active control systems – for example, the reduction in cost and complexity exhibited by passive systems, but also the greater control afforded by active systems. This may be achieved by operating semi-active springs or dampers to alter the force when there is a necessity for vibration suppression, and switching off this force when it is not required [8]. Semi-active system is also achieved by employing different solutions such as smart materials. Various semi-active dampers have been proposed in the literature. For examples Karnopp et al.[8] proposed an electro-hydraulic device, which enables the separate control of compressive and rebound forces using solenoid operated poppet valves. Giorgetti et al. [9] and Yao [10] use the magnetorheological (MR) principle to increase damping when required for the shock absorber in an automotive application. Despite the significant advantages associated with reduced power consumption, typical limitations in the current solutions of a semi-active system have to adjust either the stiffness value or the damping value of isolation system. Although, few studies were introduced semi-active systems which are able to adjust both stiffness and damping coefficient via semi-active [11], but such solutions were associated with high cost and weight. Hence, semi-active isolation systems are not able to solve all conflicts in the suspension system of lightweight vehicles.

The last system is an adaptive system. It is able to adjust stiffness or damping value of the system. This system is similar to active and semi-active systems but there is no time control in this system. For instance, the adaptive air suspension system for Audi A8 car [12]. Such system is used air springs to improve the handling during loading/unloading conditions. However, the disadvantages of such system, it is required another control system to modify the damping value of the suspension system [12], which cause to increase the cost and weight of the system.

One of the greatest challenges for the future generation of lightweight vehicles is to develop a novel suspension concept that enables the variation of both its stiffness and damping coefficients without adding weight and cost associated with active systems.

Such properties can be a benefit for other applications such as vehicle engine mounting systems [13]. For this reason, in this thesis a new type of vibration isolator, which may be considered as an adaptive system, and based on compressible flexible particles and incompressible viscous liquid was investigated. Such a composite material is dissipating energy due to the viscoelastic properties of the flexible foam particles, and the shearing of fluid around these particles [14]. It is called Foam Filled Fluid and termed as FFFluid.

The FFFluid concept combines different isolation mechanisms (stiffness and damping elements). This combination can be altered by changing the properties of the constituent components. Therefore, changing the physical characteristics of FFFluid material such as the packaging size/shape, the amount of fluid or flexible foam material lead to modifying both stiffness and damping coefficients of the system. This provides a significant opportunity for tailoring the response of the system to meet the different demands placed upon it [15]. Hence, FFFluid is a promising solution to meet the performance requirements of suspension systems of lightweight vehicles.

The main challenges to achieve this is the relative lack of relevant research underpinning this concept. This mixture is comparatively recent, and only a few academic articles have been published related to this technology [15-17]. Although the initial concept of FFFluid has been studied for impact energy management applications, its potential as vibration isolation system is still unexplored. The performance of such a concept as a suspension system or vibration isolator is still not investigated.

In addition, at present the limited understanding of the behaviour of the FFFluid, computer simulations cannot be used to reflect the FFFluid's behaviour. This is due to the absence of a mathematical model of the mixture. Therefore, a designer cannot predict the performance of such a system at an early stage, and thus, more than one prototype may need to be designed and tested in this case. This leads to increasing the cost of assessment for the isolator.

1.2 Aim and Objectives of the Research

This research aims to explore the characterisation of the Foam Filled Fluid mixture with a view of utilising it as an adaptive vibration isolation system in lightweight vehicles. Interestingly, such systems could have the capability for the off-line tuning of both the stiffness and damping coefficients.

To achieve this overall aim, the specific objectives which have been identified for this study, are:

i. To present a comprehensive assessment of developments of vibration isolators, both active and passive ones, for automotive application. The review should include the basic concept and features of isolators, the main criteria that are used to assess isolators, and the advantages and limitations of each type of these systems.

ii. To examine the characteristic of the FFFluid concept as a viable approach for designing vibration isolators with reliable performance for light-weight vehicles.

iii. To systematically study the influence of different parameters on the performance of an FFFluid vibration isolator.

iv. To develop a mathematical model that can predict the behaviour of an FFFluid isolation system.

1.3 Structure of the Thesis

This study is continuing previous research to explore and examine the Foam Filled Fluid mixture [15, 18]. These earlier studies concentrated on the development of passive FFFluid shock absorbers while this research is aimed to understanding the mechanism and behaviour of a vibration isolator using the FFFluid technology. The remaining sections of the thesis are organised as follows.

Chapter 2 presents a classical theoretical understanding of vibrations from a physical perspective. The different types of vibrations are described. Also, methods for reducing or eliminating unwanted vibrations, with a focus on vibration isolators as one of these methods, are reported. Next, a review of vibration isolators, which serves to highlight the novelty of this research is given. This begins with the concept of a vibration isolator and a description of the various configurations of isolators. The influence of stiffness and damping factors that affect the selection of vibration isolators are then described. The review ends by presenting the different designs of existing vibration isolators, paying particular attention to the advantages and limitations of each solution.

Chapter 3 presents experiments to assess the performance of a FFFluid vibration isolator. This chapter begins with a background of Foam Filled Fluid (FFFluid) concept. A description of the components, mechanism, and previous applications of FFFluid devices are reported. Then experimental procedures are carried out to determine the performance of such mixture. In addition, the advantages of such a composite material are studied, with a comparison offered between a FFFluid and other existing isolation system.

Chapter 4 exhibits a systematic study for characterising the FFFluid mixture. Then, it is followed by an experimental investigation of the system to underpin the study. The focus of the experiments is to identify the critical parameters of the FFFluid isolator using a design of experiments approach. This includes the influence of these parameters on the performance of the experimental FFFluid isolator. The contribution of each parameter is also found using statistical tools.

In Chapter 5 reports another set of FFFluid isolator experiments. The aim of these experiments, in this case, is to develop an empirical model of the system. The validation of the developed model is also achieved.

Chapter 6 presents the key conclusions and contributions of this research and offers suggestions for further work.

CHAPTER 2

LITERATURE REVIEW

2.1 Introduction

Although different types of vibration isolators and suspension systems are used, it is still necessary to develop new vibration isolation technology to meet more challenging engineering requirements [19]. Thus, the aim of this thesis was to study a composite material that can adjust its stiffness and damping coefficients via modification of its parameters. Therefore, the first step toward achieving this aim was reviewing the background and literature relevant to the research topic, which is the purpose of this chapter.

The background literature starts by introducing the concept of vibration. This includes the physical meaning of vibrations, the types of vibration excitations, and the methods to reduce vibrations. Then, the focus is moved to vibration isolation systems as one of the most preferred solutions to minimise the effect of vibrations. Criteria for evaluating the performance of isolators and the main parameters that affect the performance of vibration isolators are presented. The following section considered various technologies that are used to design a suspension system for a passenger car. The relative merits of each technology are also discussed. The chapter concludes by presenting the FFFluid concept, which is the technology investigated and reported in this thesis.

2.2 Concept of Vibration

Any repetitive, periodic, or oscillatory response of a mechanical system is called vibration. The repetition rate is referred to as the frequency while the value of this motion is called the amplitude. A vibratory system involves the transfer of its kinetic energy to its potential energy and vice versa, with some energy dissipating in each cycle. The movement of a pendulum is an example of a vibratory system.

Vibrations could be classified into two main groups as shown in Figure 2-1. Vibrations can naturally occur in an engineering system and may represent its free and natural dynamic behaviour, which is called free vibration. Also, vibration may be forced onto a system through some form of excitation. This is known as forced vibration. Forced vibrations can also be divided into two broad categories according to their predictability. If the response of the system is predicted, it is known as deterministic vibrations. The opposite situation is called random vibrations, which is when the value of motion acting on the system is not known at any given time.

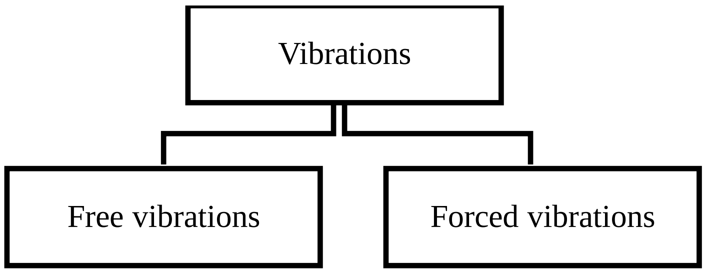


Figure 2-1: Classification of vibration

The science and engineering of vibrations involve two broad categories of applications: generation of a necessary form of useful vibration, or suppression of unwanted vibrations. Vibration could be utilised in some applications. In such cases, the presence of vibration is required to increase the efficiency of systems; for instance, vibration-powered generators [20], and whole body vibration training [21]. However, the presence of vibration in most structures and equipment is considered to be a problem. Unwanted vibration causes a reduction in the performance of machines or even may cause machine failure. It can also lead to a resonant condition for a given structure that can result in the destruction of the system. The Tacoma Narrows Bridge is an example of a structure that was destroyed due to resonance three months after its construction in 1940 [22, 23]. Furthermore, on a large number of applications, human beings act as an integral part of the system. For example, various types of transportations [24, 25], buildings [26], and machines [27, 28]. If the vibration is transmitted to humans, it may cause discomfort, loss of efficiency, and possibly health problems. Therefore, it is important to review the method of controlling unwanted vibration in the systems.

2.3 Methods to Reduce Vibrations

When vibrations play a negative effect on mechanical systems, it is important to reduce them. The problem is depicted schematically in Figure 2-2. The vibrations that are generated by a source (e.g. car, power-tool, train and washing machine) are inevitably transmitted to a receiver (e.g. passenger, hands, building) that is linked to it. The disturbance reaches the receiver through the transmission path as shown schematically in Figure 2-2. The methods of shock and vibration control may be grouped into three broad categories: (1) reduction of the vibration at its source, (2) reduction of the vibration at its response, and (3) reduction of the vibration by the insertion of anti-vibration devices. These methods are reviewed in detail below.

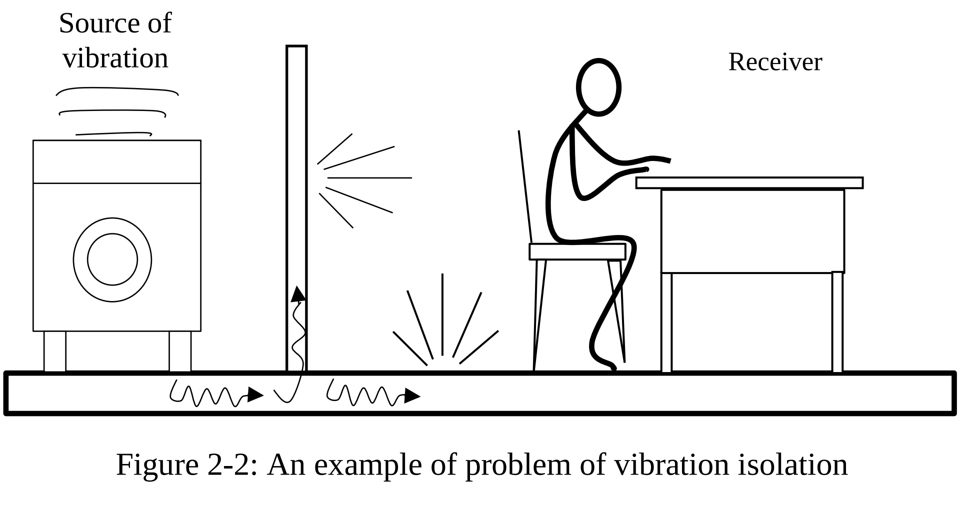


Figure 2-2: An example of problem of vibration isolation

2.3.1 Reducing the Source of Vibration

The first method of limiting the effect of vibrations on mechanical systems and structures involves reducing them at the source. To achieve this, several techniques can be used, such as improving the design of a system, balancing the moving components, reducing the clearance between elements of the mechanical system and choosing appropriate bearings.

A common source of forced vibrations in machines involves any moving part or rotating element. The level of vibration is raised by increasing the imbalance between the moving parts. An imbalance in a rotating machine implies that the axis of rotation does not coincide with the centre of mass of the whole system. It is a widespread source of forced vibration in rotating elements, such as a fan and propeller. Therefore, to attenuate vibrations in rotating machine, it is important to balance moving masses [29, 30]. Moreover, the source of vibration could increase when using excessive clearances in bearings. The bearing is a vital component of rotating machinery, and it is used to allow constrained relative motion between two or more parts, typically rotation or linear movement. The vibration that is produced by rolling bearings can be complex and can result from defects on the rolling surface or geometrical errors in associated components. Therefore, vibrations could be reduced by choosing appropriate components and bearing [31].

However, reducing the vibrations at their sources may not always be feasible. Indeed, many sources of vibration cannot be altered, such as road conditions and natural events like earthquakes. Also, the level of vibration may still be unacceptable despite improving the design of the system by balancing moving parts [22]. Therefore, vibration could be reduced at their response, this solution is introduced in the following section.

2.3.2 Reducing the Vibration at its Response

Reducing the vibration at its response can be achieved by redesigning the system and in particular by changing its natural frequency. When the natural frequency of the structure or equipment coincides with the forcing frequency, the amplitude of the system's response will be increased due to resonance. Under such conditions, changing the natural frequency of a structure is one of the solutions to prevent resonance. The following expression represents the natural frequency of structure:

$$\omega_n = \sqrt{\frac{k}{m}} \tag{2-1}$$

where: m is the mass of the structure and k is its stiffness. Therefore, to modify the natural frequency of a system, the stiffness or mass of the structure should be changed. Another method of reducing the vibration at its response is an auxiliary mass, this could be achieved by attaching an auxiliary mass and stiffness element to the original system. With proper tuning, the auxiliary mass vibrates and reduces the vibration of the system to which it is attached. Such a system, called also a vibration absorber, is effective over a narrow band of frequencies only. Therefore, it is usually used for systems that operate at a constant speed. High voltage transmission lines are a common form of application for auxiliary masses.

Similar to reducing the vibrations at their sources, solving the vibration problem by reducing the vibration level at their responses is not always be applicable. For example, attaching an auxiliary mass to the suspension system of the quarter car was studied by Hassan [32]. He developed three mass system of quarter car model by connecting an auxiliary mass to the wheel through a spring and a damper. Although there was a reduction of the vibration level at its response, this solution was not suitable in such application due to increasing the size and the weight of such suspension system [1, 30].

This leads to the introduction of the third method of vibration control, which relies on isolating the system with anti-vibration devices.

2.3.3 Insertion of Vibration Isolation Devices

A vibration isolator is a device that is inserted between the source of vibration and the system of interest (receiver). Vibration isolation devices can isolate any equipment or structure from vibratory loads by breaking the vibration transmission path from the source to the equipment. This enables a reduction in the vibratory effect. Such system usually consists of a rigid body representing equipment that is connected to a foundation by an isolator that has resilience and energy-dissipating means. The model of a single-degree-of-freedom (SDOF) that is often used to represent a vibration isolation system is shown in Figure 2-3. A mass m is suspended on a parallel combination of a spring of coefficient k and a dashpot c (which are both considered massless), F_0 is the applied force, and F_t is the transmitted force. If the vibration isolator was designed properly, the transmitted force F_t is smaller than the applied force F_0.

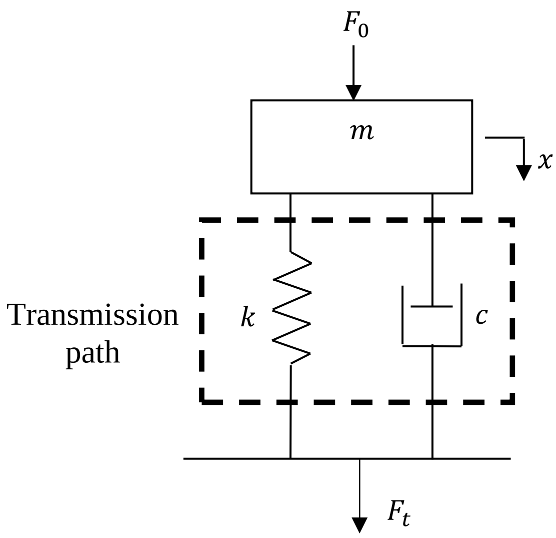


Figure 2-3: Single-degree-of-freedom vibration isolator model.

The techniques reported in the previous sections to minimise the adverse effect of vibrations may not always be applicable. Thus, cutting the transmission path between the sources of vibration and the receiver is the most common solution [19]. Therefore, the next section covers the state of the art in vibration isolation systems.

2.4 Vibration Isolators

A vibration isolator is defined as a device that reflects and absorbs waves of oscillating energy. This has the desired effect of providing vibration insulation. The basic constituents of an isolator are the resilient load-supporting means (the stiffness) and the energy dissipation means (the damping). The next section presents these concepts.

The stiffness of the system: This is the rigidity of a system or an object. When an external force is applied to the object, it usually resists deformation by a restoring force. This resistance is referred as stiffness. It is calculated by finding the ratio of the generalized force to the generalized displacement. Quantifying the level of stiffness in a system is determined by using the following:

$$k = \lim_{\Delta x - 0} \frac{\Delta F}{\Delta x} = \frac{\partial F}{\partial x} \tag{2-2}$$

where: k is the stiffness, F is the applied force, and x is the displacement. Figure 2-4 illustrates these parameters. The most important types of stiffness systems are: metal spring, elastomer mount, and air spring. Each type has different advantages and limitations. More details on these types are presented in section 2.7.

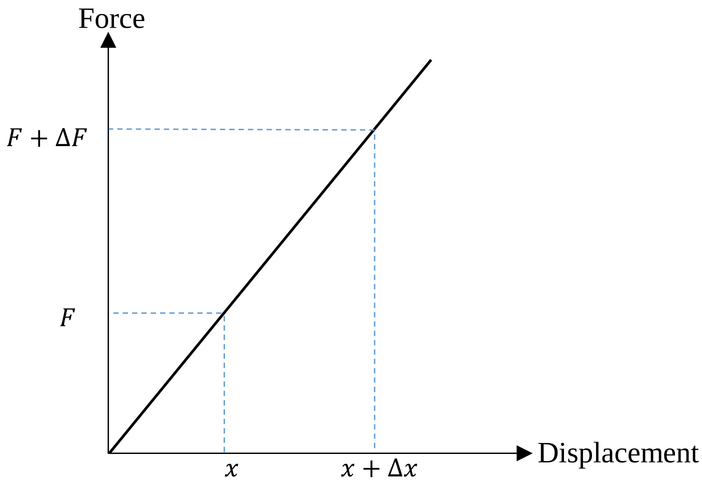


Figure 2-4: Force vs. displacement curve for a linear elastic structure.

Damping of the system: When a structure is subject to oscillating deformations or forces, its state can be described by the combination of kinetic and potential energy. In the case of real structures, some of the energy is lost every cycle. This is called structural (material) damping. Damping is the mechanism by which the vibrational energy is gradually converted into heat energy. Damping force exists if there is a relative velocity between two ends of the damper. It refers to three fundamental types:

- Coulomb damping: it is a type of constant mechanical damping in which energy is absorbed via sliding friction (friction between rubbing surfaces that are either dry or have insufficient lubrication). The damping force is constant in magnitude but opposite in direction to that of the motion of the vibrating body.

- Viscous damping: it is an energy loss that occurs in liquid lubrication between moving parts or in a fluid forced through a small opening by a piston. The viscous hydraulic damper is an example of such damping. The viscous damping force is directly proportional to the relative velocity between the two ends of the damping device.

- Material damping: when the material is deformed due to an applied force, some energy is absorbed and dissipated by the material due to the friction between the its internal planes.

To quantify the level of damping in a system, the absorbed energy per cycle is determined by plotting the displacement versus force of the system during the loading. For a given cycle of motion, a hysteresis curve is generated as shown in Figure 2-5. The energy lost per cycle in a damper in a harmonically forced system, is the area captured within the hysteresis loop D, may be expressed as:

$$D = \oint F.dx \tag{2-3}$$

Which is equal to [20]:

$$D = \pi \omega c X^2 \tag{2-4}$$

Where: F is the applied force, X is the displacement, ω is the frequency, and c is damping coefficient.

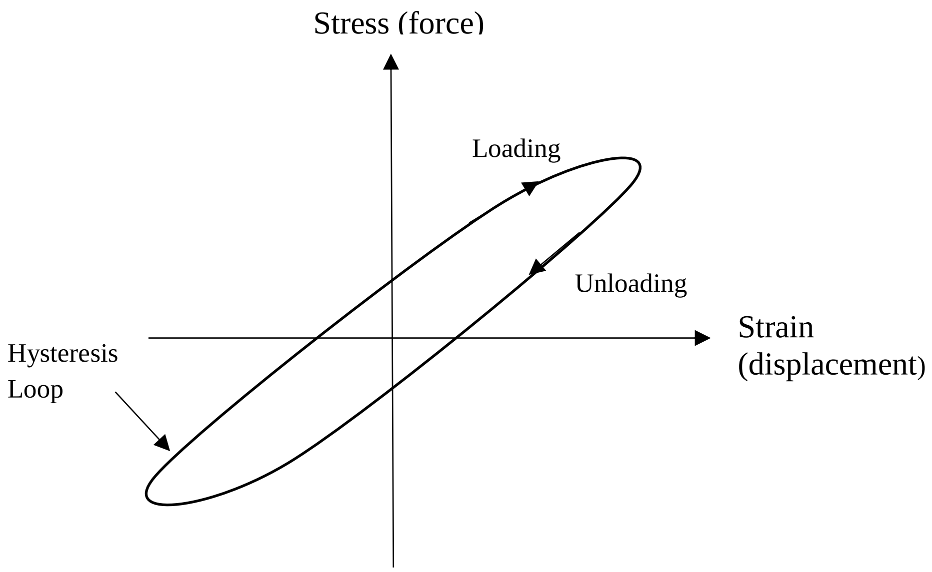


Figure 2-5: Hysteresis loop for materials

2.5 Evaluating the Effectiveness of an Isolator

There are many types of vibration problems, with many different kinds of functional requirements and constraints. However, the most common requirements that one encounters are transmissibility, motion response and position alignment. More explanations for each of these criteria are given below:

1-Transmissibility: This concept is usually employed to measure the performance of a vibration isolator. The transmissibility is the ratio of the vibration amplitude of the foundation (transmitted force through the isolator, F_t) to the vibration amplitude of the equipment (force applied to the isolator, F_o). For a linear system under harmonic force, the equation of motion is:

$$F = m\ddot{x} + c\dot{x} + kx \tag{2-5}$$

where: m is the mass of the system, k is the stiffness coefficient, c is the damping, x is the displacement, $\dot{x}$ is the velocity and $\ddot{x}$ is the acceleration. There are two solutions to this equation, the transient solution which dies out after some time, and the steady state solution. The steady state solution is expected to be harmonic. So, it is assumed as:

$$x_p(t) = X\cos(\omega t - \varphi) \tag{2-6}$$

where: X is the amplitude of the response, φ is the phase angle of the response, and ω is the frequency of the response. By substituting this equation into 2-5, the following expression is derived.

$$F_0 \cos\omega t = X[(k - m\omega^2)\cos(\omega t - \varphi) - c\omega\sin(\omega t - \varphi)] \tag{2-7}$$

By using the trigonometric relations

$$\cos(\omega t - \varphi) = \cos\omega t \cos\varphi + \sin\omega t \sin\varphi \tag{2-8}$$

$$\sin(\omega t - \varphi) = \sin\omega t \cos\varphi - \cos\omega t \sin\varphi \tag{2-9}$$

The solution of this equation is:

$$X = \frac{F_0}{\sqrt{[(k-m\omega^2)^2+\omega^2c^2]}} \tag{2-10}$$

The force transmitted to the system through the isolator (spring and damper) is given by:

$$F_t(t) = kx(t) + c\dot{x}(t) \tag{2-11}$$

$$F_t = \frac{F_0\sqrt{[k^2+\omega^2c^2]}}{\sqrt{(k-m\omega^2)^2+\omega^2c^2}} \tag{2-12}$$

Under the harmonic motion, the transmissibility is expressed:

$$T_f = \frac{F_t}{F_0} = \sqrt{\left(\frac{k^2+\omega^2c^2}{[(k-m\omega^2)^2+\omega^2c^2]}\right)} \tag{2-13}$$

The transmissibility could be expressed in term of frequency ratio r and damping ratio ζ, by using the next equation

$$T_f = \sqrt{\frac{1+(2\zeta r)^2}{(1-r^2)^2+(2\zeta r)^2}} \tag{2-14}$$

where: the frequency ratio r and the damping ratio ζ are calculated as follows:

$$r = \frac{\omega}{\omega_n} \tag{2-15}$$

$$\zeta = \frac{c}{2m\omega_n} \tag{2-16}$$

2- *Motion Response*. In some systems and machines, it is important to minimise the movement of mass. Some types of isolators are designed particularly to protect against the motion of the foundation, as in the case of protecting of a delicate instrument from the motion of its container. In this case, the performance of the isolator can also measure the motion response. In particular, it is the ratio of the displacement amplitude of the equipment to the quotient obtained by dividing the excitation force amplitude by the static stiffness of the isolator. The motion response for single degree of freedom (SDOF) is expressed as follows:

$$T_d = \frac{X}{\delta_{st}} = \frac{1}{\sqrt{(1-r^2)^2+(2\zeta r)^2}} \tag{2-17}$$

where: T_d is the motion response, X is the amplitude of the displacement of the mass, and δ_{st} is the static deflection under constant force.

Static deflection is related to the distance the sprung mass (essentially the body) moves downward in response to weight. The static deflection rate of the system determines its natural frequency. It can be determined by a simple formula expressed as follows:

$$\delta_{st} = \frac{g}{4\pi^2 \omega_n^2} \tag{2-18}$$

where: g is the acceleration due to gravity (9.8 m/s^2).

3-Position alignment: the position of an isolated platform relative to some reference must often be regulated. For example, a suspension system is required to level the vehicle body for different loads and maintain the vehicle attitude during operation. Therefore, suspension displacement must be determined. The suspension deflection is determined by subtracting displacement of the base (road) from the displacement of the platform (body of the vehicle).

2.6 Effect of Stiffness and Damping on the Performance of Isolator

As reported earlier, the main components of a vibration isolator are the stiffness and damping elements. These elements play a significant role in the performance of an isolator for a single degree of freedom. Under the harmonic motion, the transmissibility and motion response are expressed by Equation 2-13. Figure 2-6 shows the evolution of motion response as a function of the frequency ratio (r). Several points can be highlighted from this Figure.

I. Two regions can be identified: the isolation region, where the motion response is less than 1, and the amplification region where the value of motion response is more than one.

II. For un-damped system ($\zeta = 0$). The motion response is infinite at $r = 1$, which is its resonance state.

III. The value of motion response can be changed by changing the stiffness of the isolator. Therefore, the value of force transmitted to the receivers can be reduced by reducing the natural frequency of the system

IV. A higher value of damping improves the performance of the system at any frequency.

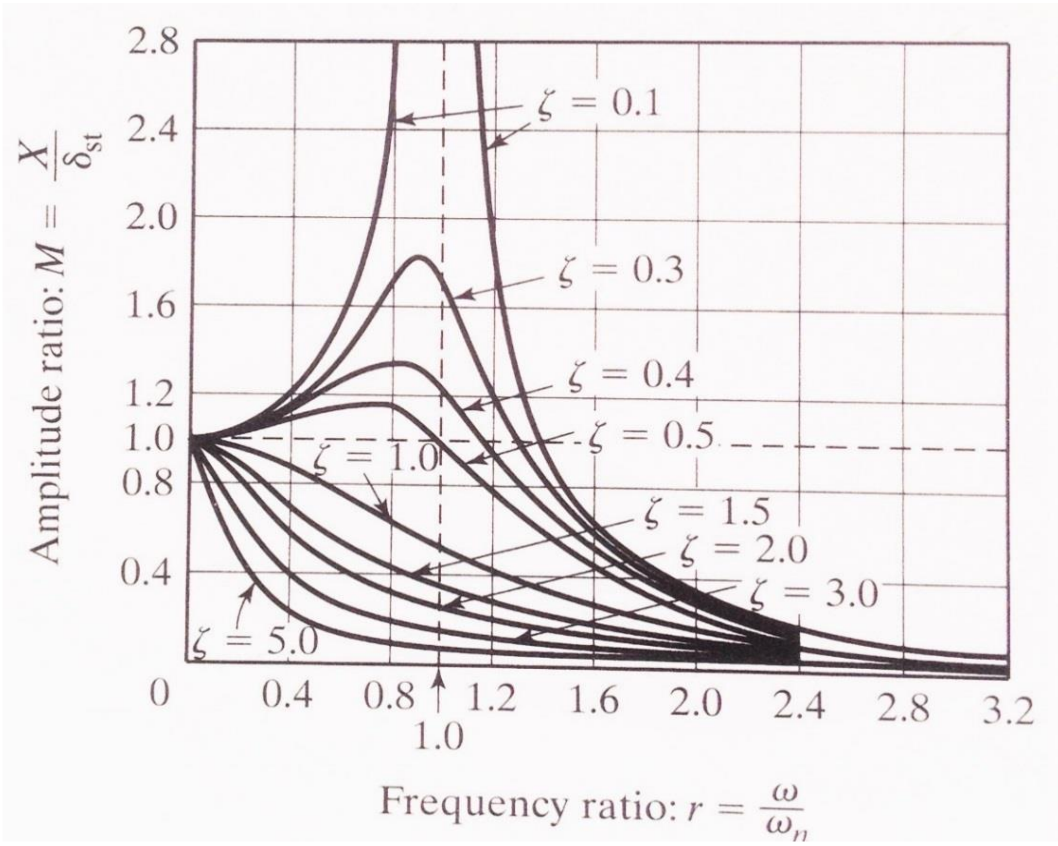


Figure 2-6: Relationship between the frequency ratio and motion response [22].

Vibration isolators are usually installed for many applications. Typical applications include vehicle suspensions, aircraft landing gears, optical tables and seismic protection. Generally, vibration isolators can be implemented in two types of situations as described below.

<u>*Vibration isolation system with the rigid foundation*</u>: This type of isolator is used to reduce the magnitude of the force transmitted from the vibrating machine to its foundation. In this case, the vibrating machine is fixed directly to the floor or the base. This base will be subjected to a static load and vibrating load due to the running machine. Therefore, the isolation system is placed to minimise the transmitted force to the foundation as shown in Figure 2-7. An example of such isolator: the vehicle engine mounts that are used to

isolate the driver and passenger from both noise and vibration which are generated by the engine [33], isolation of structures and buildings [34, 35].

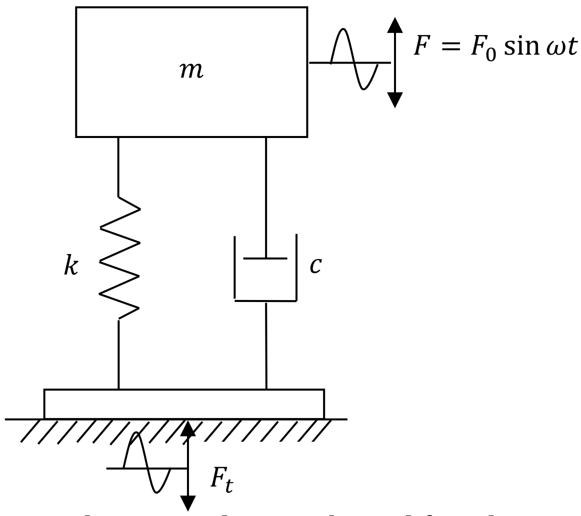


Figure 2-7: Vibration isolator with rigid foundation

Vibration Isolation with Base Motion: In the second type, the function of a vibration isolation system is to minimise the magnitude of motion transmitted from a vibrating foundation to the equipment as shown in Figure 2-8. For example, the suspension system of a vehicle is used to protect passengers on rough road conditions [36, 37].

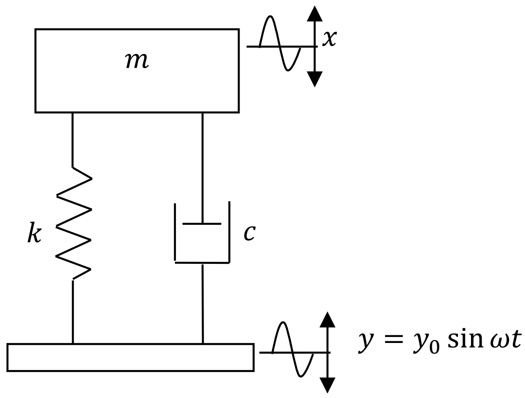


Figure 2-8: Vibration isolator with base motion.

A key focus of the present thesis is to characterise a novel a material that is suitable for meeting the demands concurrent with a lightweight vehicle. Therefore, the next section covers the different types of vehicle suspension systems.

2.7 Types and Characteristics of Vehicle Suspension Systems

As one of the most important features in a vehicle, the suspension is a major focus for automotive engineers. The suspension is the term given to the system of damper and springs as well as linkages, which connect a vehicle to its wheels. The functions of a suspension system are:

- To isolate the vehicle body from external disturbances coming from irregular road surfaces to achieve ride comfort;
- To carry the weight of the vehicle body;
- To react to variations in loading conditions, which are generated by changes in the number of passengers and transported loads;
- To keep a firm contact between the road and the tyres, and thus, to ensure good handling performance.

However, to achieve these tasks optimally leads to design conflicts. Softer suspension system provides a more comfortable ride while stiffer ones provide better stability and thus better road-handling.

Another difficulty when designing suspension systems is the need for different stiffness coefficients depending on load variation of vehicles. Increasing the load of the vehicle modifies its natural frequency and increases the static deflection of the system. The most comfortable ride occurs when the natural frequency of the vehicle is in the range of 60 to 75 cycle/min [38]. Therefore, in practice, suspension frequencies for passenger cars fall in the range 1 to 1.2 Hz (60 to 72 cycle/min) with corresponding static deflections of 225 to 175 mm (8.8 to 6.9 in) [38]. It is important to keep these values (natural frequency and height ride) within predetermined limits. To achieve this, different values of stiffness coefficients to support such different loads may be required especially for lightweight vehicles.

This may not be an important issue for standard family vehicles because their percentage change in total mass does not vary widely for different loading conditions. For example, two passengers (total average weight 175kg) cause an increase of 10% in the total weight of a family car (average weight 1750kg). However, the same weight of two passengers causes an increase of more than 30% in the mass of the lightweight vehicle class a L7e (average weight 450kg). Therefore, this is an important issue for the suspension system of lightweight vehicles.

To clarify a family vehicle requires a stiffness of $92kN/m$ to provide a natural frequency of 1.15Hz in unladen case. This natural frequency changes to 1.1Hz with the weight of two occupants aboard. However, the natural frequency of a L7e vehicle changes from 1.15Hz in the unladen case to 0.95Hz with an equal two-occupant load. Such a frequency is less than that considered comfortable for humans [39]. Moreover, 15mm is only the difference of height for a family vehicle between the empty case and the case of two people inside it, while this difference is 85mm for a L7e vehicle. Such a deflection of the vehicle will cause significant changes in the geometric relationship of the suspension components. Consequently, a variable stiffness element is required to improve the suspension system of lightweight vehicles.

Suspensions can be divided into passive, active systems, semi-active or adaptive system [40, 41]. In what follows, the relative merits of these methodologies are described. This will serve to highlight the significant benefits that can be gained by employing Foam Filled Fluids in such systems.

2.7.1 Passive Suspension System

In passive suspension system, the stiffness and damping elements are attached to the mass of the structure to form this system as shown in Figure 2-9. These elements can store and dissipate energy associated with local relative motions. However, the stiffness and damping properties remain fixed for all time. This system is characterised by the simplicity of design and reliability of the performance. For such these reasons, passive suspensions are the most frequently adopted solution to vibration control, and it is used in many varieties of engineering systems [42]. Various types of damping and stiffness elements are used as passive elements. These types are studied in more detail.

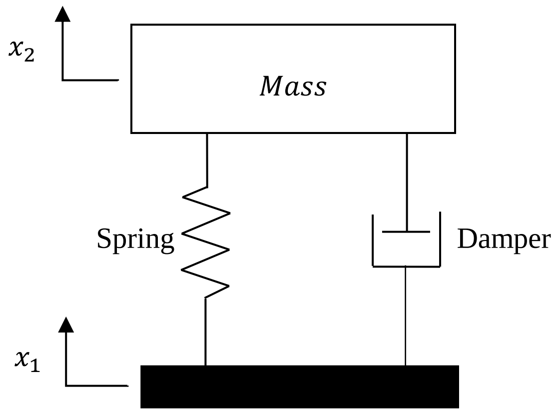


Figure 2-9: passive suspension system.

For the damping element, most conventional vehicle suspension systems utilise viscous dampers [43]. This damper typically suppresses vibration by a piston moving through oil. Holes in the piston are covered by valves, which resist the flow of the oil through the holes in a controlled manner to produce the damping effect. There are three categories of passive dampers: monotube damper, twin tube damper, and double-ended damper. Figure 2-10 shows these types of viscous dampers. The mono tube damper has one fluid reservoir, and consists of a single piston moving inside a cylinder; at the end of the cylinder there is an accumulator to accommodate the change in volume. The volume enclosed between the housing is called a reservoir of fluid, as shown in Figure 2-10a. There are two fluid reservoirs in a twin tube damper, one inside the other. There are also two housings, an inner and outer housing. The inner housing drives the piston rod assembly, and the volume enclosed is named as the inner reservoir in the same way as a mono tube damper. Also, the volume that is defined by the space between the inner housing and the outer housing is referred to as the outer reservoir, as shown in Figure 2-10b. The last type of damper is called a double-ended damper; the double-ended damper does not require a reservoir for changes in volume. There is a piston which is double ended inside the cylinder, and during movement of the piston in this cylinder, an effective fluid orifice is produced. Figure 2-10c illustrates this type. In spite of using these dampers widely in automotive applications, these passive dampers have constant damping coefficients. Therefore, it is not able to obtain good performance over many different road surfaces [1].

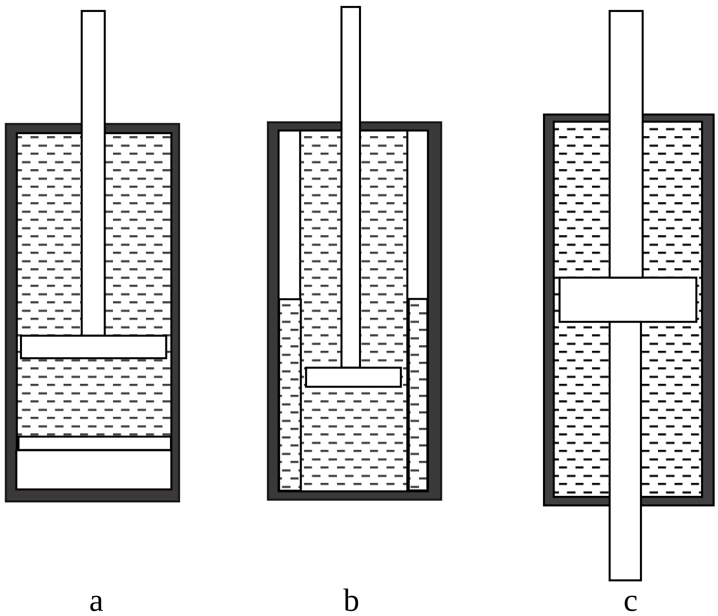


Figure 2-10: Viscous dampers: (a) End tube damper, (b) Twin-tube dampers, (c) Mono-tube.

Stiffness elements for passive suspensions can be made in many different resilient materials. The most common materials are metallic, pneumatic and elastomeric material. Therefore, these types are described in the following section.

I- Metallic spring: Such springs can support the load of the system and store the resulting energy. There are different types of metal springs: coil springs, ring spring, Belleville spring, involute springs, wire-mesh springs [44]. Three types of steel springs are found in automobiles: coil springs, leaf springs, and torsion bars. There are several advantages associated with such springs, such as their ability to operate in a broad range of temperatures and other difficult environmental conditions. Also, they can combat the effect of corrosion [44-46]. Such springs are also able to carry a high static load [45, 47, 48]. For such these reasons, the coil is the most common type of stiffness element for commercial vehicles [49-51].

Although such isolators offer high reliability and excellent performance, some negative points are holding back the development of these systems, and which may lead to them not being employed [52]. Metal springs are more expensive and heavier in comparison to other resilient materials [44, 53]. Hence, this spring may be made of different materials. Several designs were introduced that attempted to reduce the weight of such system. For example, Audi introduced new lightweight springs made of glass fibre-reinforced polymer (GFRP) instead of steel springs to be used for mid-size models. This design is 40% lighter than the steel one [54]. Also, vehicle leaf springs with a light composite material are also developed [51, 55]. Such springs have high strength, high ductility, excellent manufacturability, and also low weight.

Although the weight of metallic spring could be reduced by using composite material, manufacturing spring using these materials is still associated with high costs [56]. The other negative point of such spring, it is hard to change its stiffness value. So, if a variable stiffness element is needed such as for the suspensions systems of lightweight vehicles [4], it is required to use active or adaptive systems to improve the suspension characteristics, but it may at higher cost and increased mass.

II- Pneumatic Spring: The resilient element of this system is a gas (often air). The stiffness of the air spring is derived from the gas laws governing the relationship between pressure and volume. By assuming an adiabatic compression, the stiffness of the air system can be obtained by using the following expression:

$$k = \frac{\gamma P A^2}{V} \tag{2-19}$$

where P is the pressure of the sealed air in working conditions; V is the volume; A is the effective area of the system; and γ is a constant, which is equal to 1.4 when using air [57]. The theory and mathematical models of a pneumatic systems has been developed by Bachrach and Rivin [58]. A pneumatic system also has an inherent damping, this is due to the flexible diaphragm, friction in the package (usually piston and cylinder), and irreversibility in the compression and expansion of the gas [59]. Therefore, Bachrach and Rivin's model was further refined by Erin et al. [60] and Ryaboy [61]. These authors considered an equivalent mechanical analog and gave detailed consideration to the role of the sealing diaphragm. Such model predicts the performance more accurate than the previous one. Although these types of inherent damping for the pneumatic system, an extra damper may still be added to such system [62]. The main attraction of using this type of spring is the ability to change the stiffness value. Indeed, the stiffness of the system could be adjusted by changing the pressure of the gas. Another advantage of such spring is that they have a greater capacity for energy storage than others [53]. They also provide a high load capacity that enables the isolation of large setups including heavy elements [61].

However, limitations of pneumatic systems are related to their cost. In addition to the cost of manufacturing such systems [63], a pressure regulator and compressor to pumps the air into the bellows are required. The other issue is the cost of maintenance. Indeed, these systems require more maintenance in comparison with others [64, 65]. This is due to the leakage or failure in one of these components; bellow, pipes (lines), valves or compressors.

III- Elastomeric Spring: An elastomer is a natural or synthetic rubber material that has viscoelastic properties. This material demonstrates both stiffness and damping properties, and therefore, it is used widely in anti-vibration devices [45, 66]. There are several advantages when using this material: elastomer isolators have higher internal damping than metal springs, and therefore, there is no need for an external damper [45]. Low cost, weight and less maintenance are other advantages of elastomer devices. Therefore, it is extensively used as a means to mitigate resonant vibration responses.

However, some aspects of their mechanical behaviour require more research, such as their limited operational temperature range [67]. Indeed, the mechanical properties of elastomers can change significantly due to temperature variations. In particular, prolonged exposure to low temperatures can result in rubber crystallization, with a considerable increase in stiffness of the material. Young’s modulus changes widely at glass temperature, which is around 120℃ in some polymers. This means these devices may not be applicable for applications where the temperature is high [66]. Also, similar to metallic spring, it is not easy changing the stiffness coefficient of the elastomer system.

Therefore, it is not the preferred solution in the applications which are required a variable stiffness such as lightweight vehicles.

The other limitiation in elastomer material usually fails to distribute a load uniformly under compression load, resulting local areas of stress concentration in the material; the area that directly below the loads are deformed more than other areas [45, 67]. To minimise such issue, the elastomer material may be use with other materials in different configurations, for example, a bonded elastomer structure or mixing shredded elastomer particles with soil material.

In bonded elastomer structure design, the whole load-carrying surfaces of elastomer are bonded by a metal plate to distribute the load across its surface [45]. Multilayers of elastomer that are bonded to fibre or thin steel-reinforcing sheets can also be introduced [68, 69]. Such bonded elastomer configuration is providing a preferred distribution in the surface load, and it may prevent side bulging of the elastomer material. This design was applied in BMC Mini vehicle [39]. Also, this design has been used as a spring in rail application instead of steel springs. It could be produced with sufficient strength and durability without the problem of rust or wear and also less weight [70]. But, the high cost is the main disadvantage of this configuration. Such issue prevents of introducing such configuration in wider applications [42, 66].

Another configuration of using an elastmor with another material is mixing shredded rubber particles with soil material. Such a mixture was proposed as a base of buildings and structures [71]. The preliminary research worked for the protection of low-to-medium-rise buildings by Tsang et al. [72], their researchers found that it can reduce the shaking level of vertical ground motion significantly [72]. Xiong and Li reach to similar results [73]. However, such solution may effect on environmental matters such as ground water contamination and the impact on local ecology [71]. Therefore, it is still requiring further investigation on these environmental issues. Also, this configuration may be feasible in the bases of buildings where the soil is readily available at the medium, but it is not the applicable in automotive suspensions systems where the soil is not readily available in such systems.

To conclude, the passive system is the most common type used of isolator, especially in automotive suspensions. Generally, traditional passive suspension systems are still lower cost and relatively simple to manufacture in comparison with other designs. However, achieving acceptable performance over a broad range of excitation conditions is acknowledged as a problem. Hassan [32] showed that wide variations in the suspension parameters are required to obtain good performance over many different road surfaces. Clearly, conflicts in the suspension system of lightweight vehicles cannot be completely overcome with passive isolation systems [4].

2.7.2 Active Suspension System

In this case, the coefficients of the system are changeable by adding an active power source. This power source uses to actuate the isolator. In an active vibration control, a counteracting dynamic force is created by one or more actuators to suppress the transmission of the system disturbance force, this actuator is connected to a controller and a sensor as shown in Figure 2-11. According to signal processed by the sensor and the controller, the actuator is providing a dynamic force which should be suppressed vibrations [74].

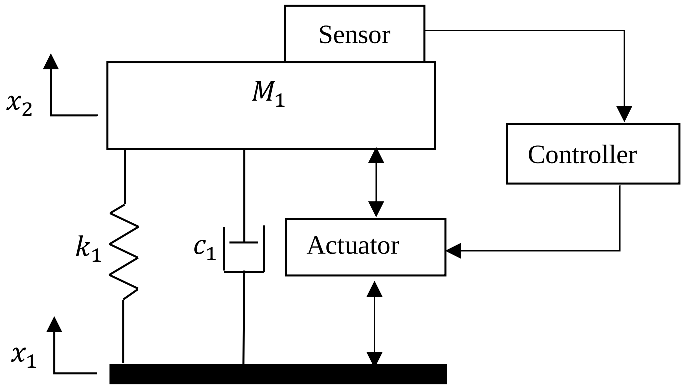


Figure 2-11: Active suspension system.

The main benefit of such suspension systems is that it addresses the trade-off between the conflicting requirements of ride comfort and handling. Also, variations of static loads of vehicle can also be dealt with using such an approach. In this case, variable spring stiffness is provided. In particular, such stiffness can be adjusted proportionally to the change in mass. As a consequence, there is no change in the natural frequency of the vehicle. Furthermore, an active suspension system can be used to eliminate body roll during cornering. Hence, the wheels can be oriented optimally on the road during cornering.

Leegwater [75] developed a vehicle vibration control system using active control. In his study, an active suspension is investigated the capable of levelling the car during cornering to improving handle of vehicles and self-levelling car with various mass. The author showed that the active part of the suspension was able to achieve good cornering behaviour and to adapt to static load variations. Stamm [76] investigated an active suspension solution for military vehicles to solve the conflicts in such vehicles. When protection of military vehicles is improved, this improvement could impair mobility and payload of such vehicle. For example, as armor is upgraded for increased protection, the mobility and payload can be decreased. Since the suspension is designed to carry a finite amount of weight, the added armor forces a reduction in payload, and the heavier vehicle is not as manoeuvrable as previous. In Stamm research, an active suspensions system that is able to improve mobility, protection, and payload simultaneously was studied. Stein and Ballo [77] dealt with an electrohydraulic active vibration control system for the driver's seat for off-road vehicles. The experimental results showed that 10dB improvement was achieved by the vibration absorption as compared with the passive seat. Baker et al. [78] presented the application of active suspensions of electric vehicle (EV). The active system always provided considerable advantages for comfort and safety ride of the vehicle.

Active systems are also used to solve the design trade-off for various applications. Howell [79] and McGehee et al. [80] investigated a hydraulic active control technology for the landing gears of the airplane. Results of experimental investigations showed that a controlled gear is effective in reducing the loads transmitted by the gear to the airframe during ground operations. Hagino et al. [81] also gave an example of active vibration control for an engine mount system. An electromagnetic actuator generated the dynamic force to counteract engine disturbance and reduce the frame vibration at engine idle speeds.

One of the most importance features in active suspension systems is designing an effective controller. Many control methods have been proposed to overcome suspension problems. For example, PID controllers [82] and fuzzy control schemes [83, 84]. Athas and Falb [85] proposed the active suspension system by using the concept of linear-quadratic (LQ) control. They showed that active systems gave better performance regarding comfort ride compared to the passive suspension. While Yoshimura et al. [86] constructed active suspension systems by using the concept of sliding mode control. According to this author, the sliding mode control is much better than LQ control concept and passive suspension system. Although various types of control systems are investigated, there is currently no consensus on how best to perform automatic control.

Despite their attractive performance and their ability to solve the conflicts in vehicles, the practical implementation of the active system is still limited for the majority of vehicles including lightweight vehicles. This is due to the increase in weight, cost and power consumption associated with them. Thus, complexities in design, high cost and energy consumption are significant disadvantages of active vibration control [72, 74]. For instance, Crolla designed an active vehicle isolation (suspension) solution that could enhance ride comfort (vehicle body acceleration) by 35% [87]. However, the development of such a system was not pursued, due to the associated high cost, weight and complexity, which significantly outweighed the potential performance enhancements. As a result, active isolation systems have not been widely introduced into production despite the significant research developments achieved [88].

2.7.3 Semi-active Suspension System

The third type is semi active suspension system; such system is capable of changing one or more properties in response to a command signal. The key difference between semi-active and active isolation solution is that an active system applies an external force to the system either in an upward or downward direction, regardless of the absolute input excitation. In contrast, semi-active systems do not employ such an external force. Instead, they employ other solutions which lead to vary the stiffness or damping coefficient of the system, such as using solenoid valves, MR damper or a variable orifice. Figure 2-12 show a semi-active system with the variable damping element. Semi-active systems are less expensive to design and consume less energy.

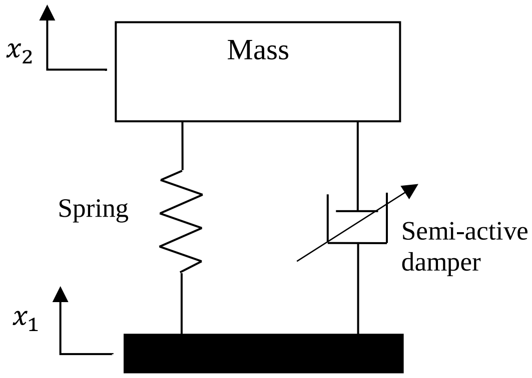


Figure 2-12: Semi-active suspension system

Variable damping in semi-active dampers can be achieved by a variety of technologies, such as solenoid valves and smart fluids. The first semi-active suspension systems were proposed by Crosby and Karnopp [89], and Karnopp et al.[8]. These authors used an electro-hydraulic device, which enabled the separate control of compressive and rebound forces using solenoid operated poppet valves. The studies showed that performance comparable to that of fully active vibration control systems could be achieved with the semi-active solution. Cebon [90] investigated a valve adjustable semi-active damper for lorry suspensions. This researcher found that body acceleration and tyre force could be reduced simultaneously by 28% and 21% of their values for optimal passive damping [90].

Semi-active suspensions continue to gain popularity in vehicle applications due to their advantageous characteristics over passive suspension systems. Nowadays, the most common type of semi-active approach is to use a smart fluid. This is due to the cost-effectiveness and simplicity design. A smart fluid is a fluid that changes its properties in response to an applied electric or magnetic field. It consists of micron sized particles suspended in an inert carrier liquid. There are two types of smart fluid; Electrorheological fluid (ER) and Magnetorheological fluid (MR). Numerous dampers have been designed to utilize the benefits of smart fluids [91-94]. Jolly [94] introduced a mono damper as a heavy duty vehicle seat suspension system while Lee [91] studied an MR suspension on a heavy truck application, designed as twin tube MR damper. An example of an such isolator is shown in Figure 2-13. Semi-active dampers utilising MR-fluid technology were also introduced for commercial purpose. General Motors (GM) introduced the first primary MR suspension on the Cadillac Seville STS in 2002 [95]. Such a damper is now appearing on more than a dozen models from a wide variety of vehicles, including Audi TT, Audi R8, Ferrari 599GTB, and Holden HSV Commodore [96].

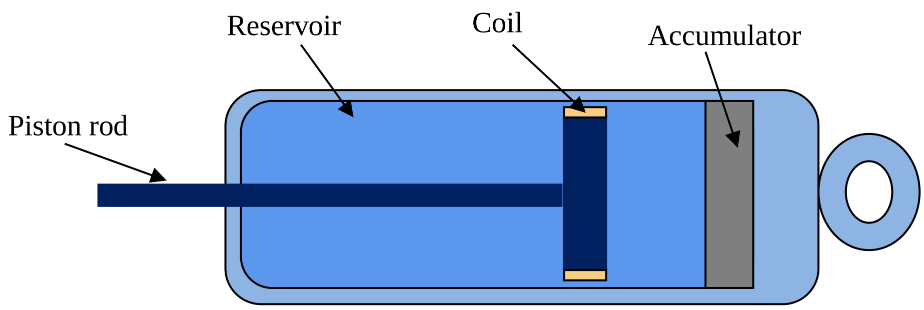


Figure 2-13: Monotube MR damper section view [93].

In the last ten years, researchers also concentrated their efforts on developing dampers with energy recovery from vehicle suspensions. Segal et al.[97] demonstrated that roughly 200watts of power are dissipated in an ordinary Sedan traversing a poor road at 13.4m/s. Self-powered semi-active control systems using regenerative dampers have been proposed [98, 99]. One of the main solutions is to use electromagnetic dampers. Such suspensions are composed of electromechanical elements, and the kinetic energy of vehicle body vibration can be regenerated as useful electrical energy [100]. Researchers have also been carrying out a proof-of-concept study to evaluate the feasibility of obtaining significant energy savings by using optimised regenerative magnetic shock absorber in vehicles [101].

The suspension system of lightweight is required to modify both stiffness and damping coefficient. To date, few studies have investigated solutions that are able to modify both stiffness and damping coefficients via the semi-active approach. Anubi et al. [102] considered a practical implementation of a variable stiffness mechanism, in conjunction with an existing variable damping device made of MR dampers. Figure 2-14 illustrates the concept put forward to allow variable stiffness of the structure. The variable stiffness mechanism concept is achieved through the control of the force. The result is a new variable stiffness semi-active suspension system. However, such a concept is hard to design. Also, it is associated with an increase in weight and size that is not suitable for lightweight vehicles.

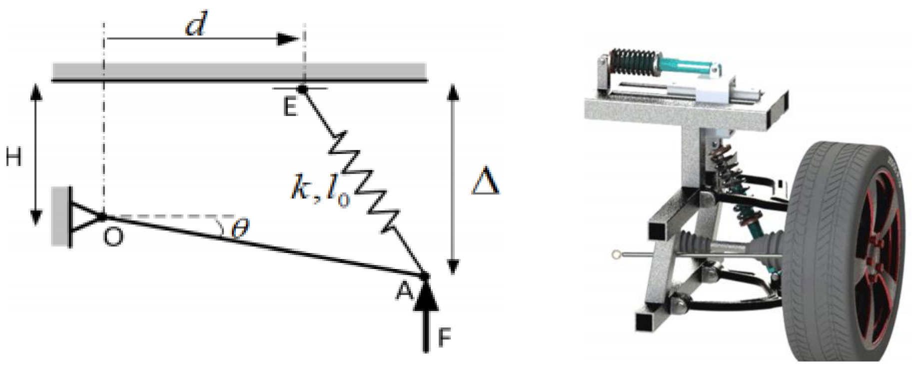


Figure 2-14: Mechanism for variable stiffness [103]

Liu et al.[11] proposed a variable damping and stiffness solution using two MR dampers and two springs. The developed structure used two Voigt elements (each one composed of a controllable damper and a constant spring) in series to realize variable stiffness and damping as shown in Figure 2-15. Due to the difficulty to modify the stiffness coefficient of the structure, these authors used MR damper to control the stiffness and damping value of the structure. Zhang et al.[104] and Liu et al. [105] also proposed other configurations using a similar concept. Although there are a fast of response to change the stiffness and damping coefficients. Such design is not applicable for lightweight application; this is due to the complexity of design two controller, big sizing of the system and high cost.

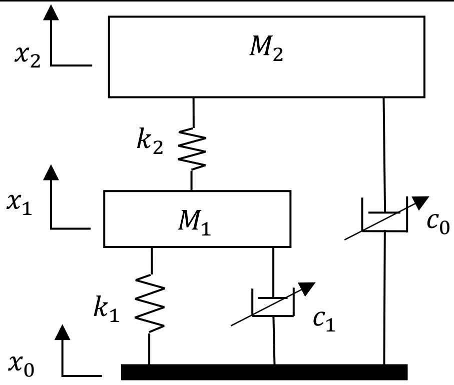


Figure 2-15: Schematic model of variable stiffness and damping.

Although the actuators and elements used to implement semi-active control techniques are quite diverse, but these solutions modify the stiffness and damping coefficients separately, which cause to increase the complexity, the cost and the weight of the system [104, 105]. Hence, the current solutions of semi-active system may not sufficient to solve all conflict in suspension system of the lightweight vehicles.

2.7.4 Adaptive Suspension System

This system is capable of changing one or more properties of the system. It is similar to the active and semi-active system, but there is no real-time control in an adaptive system. The most common concept used for the adaptive system is a pneumatic suspension system. The static deflection of the pneumatic spring could be reduced by compressing the gas to a higher pressure. Therefore, the stiffness of the system could be adjusted by changing the pressure of the gas. As a result; the pneumatic system can provide different load supports for vehicles. The gas pressure is adjustable for a particular load, increasing the pressure of compressed gas when riding loaded, and deflate the air when riding unloaded for optimal ride quality. Air spring is common in heavy vehicles; it is also found in designs some of the luxury cars such as Audi adaptive air suspension system, Porsche air suspension management system, and AIRMATIC air suspension system for Mercedes-Benz [12, 41]. Shan et al. [106] proposed another adaptive structure with variable stiffness structures utilizing Fluidic Flexible Matrix Composites (F2MC) and working fluid. Significant changes in the stiffness can be achieved by simply opening or closing an inlet valve of the F2MC tube. A closed/open modulus ratio as high as 56 times is achieved experimentally.

Variable spring and variable damping settings are desired for suspensions system of a lightweight vehicle. However, the current types of adaptive systems modify only the stiffness value of the system. To modify the damping element in an adaptive suspension system, it is required to use another control system. For example Renno et al.[107] used MR damper with a pneumatic spring to adjust of the damping and the stiffness coefficients of the system respectively. However, there are increasing in the cost, the size and the complexity of such system [108]. MR damper with pneumatic system usually uses for very high-end cars only such as the Mercedes S-Class; Also, this suspension system can be an expensive option for other vehicles; for example, there is an extra cost around to £3000 to use such a suspension for Audi SQ7[109].

It is still challenging to design an adaptive suspension system that is able to adjusting the stiffness and damping coefficient of the system without adding weight and cost associated with current systems. In this thesis, a novel composite material, which based on compressible flexible particles and the incompressible liquid was studied. FFFluid technology involves mixing particles of a flexible foam material and liquid, with the mixture being retained in a strong package as shown in Figure 2-16. When applying forces or vibrations, the hydraulic characteristics of the matrix fluid distribute loading over the surface of flexible particles. Therefore, the particles of flexible material absorb energy when it occur under bulk compression during loading. Also, some energy is damped due to the contraction of the package. This technology creates a composite material that shares the impact protection and vibration isolation characteristics of flexible particles, fluid, and packages.

Such a composite material has been used previously to design a passive anti-vibration device. However, changing the physical characteristics of FFFluid material cause changes both stiffness and damping coefficients of the system. Such changes to properties can be benefit for suspension system of lightweight vehicles and other applications such as vehicle engine mounting systems [13].

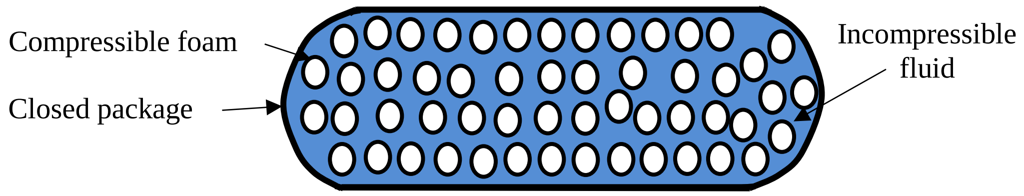


Figure 2-16: FFFluid sample

Several experiments were conducted to validate the shock-absorbing properties of FFFluid, and the concept of this mixture was originally demonstrated by Courtney [14, 110]. The inventor of this mixture patented devices, such as impact energy capsules, which are registered in the Intellectual Property Office under number GB2470180 [111]; and GB1996/003243 [109]. Some of the proposed applications are body armour; weight reduction using FFFluid plus foam; and soft pedestrian-friendly car bumpers.

The FFFluid concept has some unique properties, such as low cost and weight and simple design. This technology has been investigated in recent years by researchers at Cardiff University in order to design a vehicle bumper system to improve the safety of pedestrians [18, 112, 113]. This system enables the bumper to be stiffer when bearing an impact between vehicles, but less stiff in the case of an impact between a vehicle and a pedestrian, thereby potentially reducing the degree of personal injury during accidents [18].

Also, a project at Nanjing University investigated the effects of mixing liquid and elastomer capsules. In particular, Teng and Chen designed a vibration isolator using same mixture [16, 114]. These authors mixed rubber balls and oils to design such a device. The performance of the device was assessed, but the contribution of the viscous effect of the fluid in eliminating the vibrations was not accounted for. Furthermore, this group investigate the nonlinear force – displacement relation of the isolator (hardening stiffness element of the system), which could be utilised in design passive vibration isolator for a heavy machine.

However, FFFluid structure that is able to modify its stiffness and damping coefficients by changing the volume fraction of any component of the mixture could be designed. Davies [15] found that FFFluid exhibit a combination of energy absorption mechanisms, and this combination can be altered by changes to the properties of the constituent components. This provides a significant opportunity for tailoring the response of the device to meet the different demands placed upon it. Hence, this material also has the potential ability to be used as suspension systems for lightweight vehicles.

2.8 Summary of Chapter 2

This chapter started with presented background knowledge of vibration. This included the classification of vibrations and the importance of studying vibrations. This chapter also presented the different ways that are used to suppress unwanted vibration. These methods of shock and vibration control may be grouped into three broad categories: reduce the vibration at its source, reduce the vibration at its response, and reduce the vibration by inserting anti-vibration devices. The first two ways may not be applied in many situations. This has led the researchers to focus on the third way, which is design vibration isolators.

A key focus of the present thesis is to study a novel material that is suitable for meeting the demands concurrent with a lightweight vehicle. Therefore, the types of vehicle suspension system were also reviewed, and the most important point can be summarised as follows:

- ✓ The functions of the suspension system are: (1) to carry the static load of the vehicles, (2) to isolate the frame of the vehicle from road disturbances and (3) to maintain contact between a vehicle’s tyres and the road. The ideal suspensions system of the vehicle has variable stiffness and damping to solve these conflicting requirements.

- ✓ The performance conflicts cannot be overcome with passive isolation systems. However, reliability, cost-effectiveness and low maintenance make passive suspension systems still attractive.

- ✓ Active suspension systems have an actuation mechanism to generate a force that can modify their stiffness and damping coefficients. Hence, the trade-off between the conflicting requirements of ride comfort and handling could be achieved. Also, the variation of static loads of vehicles could be solved. Since the active system uses sensors, control units and an energy source, they are typically associated with increased weight, cost and energy consumption and reduced reliability.

- ✓ Semi-active and adaptive systems have been developed to take advantage of the best features of both passive and active control suspensions. These approaches can be used to tackle the high cost of active systems while maintaining acceptable performances. This is achieved by operating active springs or dampers to produce force when there is a necessity for vibration suppression, and switching off this force when energy input is required.

- ✓ Recently, research in semi-active and adaptive suspensions have continued to narrow the gap between semi-active and fully active suspension systems by developing variable stiffness and variable damping structures without add weight and cost associated with active system. Although there are improvements in the performance of these new devices, these display complex designs and are still not suitable to be used as suspension systems for lightweight vehicles.

- ✓ Foam Filled Fluid (FFFluid) is a mixture with interesting potential for anti-vibration devices. It was found that FFFluid exhibit a combination of isolating mechanisms, and this combination can be altered by changing one or more components of the mixture. Therefore, their stiffness and damping coefficient could be modified. Hence such mixture is promising to be used as a suspension system for lightweight vehicles.

CHAPTER 3

FOAM FILLED FLUID (FFFluid) TECHNOLOGY

3.1 Introduction

As discussed in Chapters 1 and 2, it remains a challenge to design adaptive suspension systems or vibration isolators in which both the stiffness and damping coefficients are adjustable. The development of new material for use in anti-vibration systems is still an active research area. Foam Filled Fluid (FFFluid) is a promising composite material for addressing such an issue. Hence, the objectives of this chapter are to (1) present the concept of the FFFluid mixture; (2) present initial experiments carried out to assess the performance of an FFFluid isolator; and (3) study the advantages of FFFluid isolators over existing alternatives.

This chapter is organised as follows. First, the methodology of this chapter is introduced, then FFFluid technology is presented, including the components of the system and its mechanisms. This is followed by a study of the FFFluid isolator. This study contained a description of the mechanism of FFFluid in isolating dynamic systems and initial experimental work to examine the isolating properties of FFFluid. The chapter finishes by studying the advantages of such a system and the comparisons are made between FFFluid and existing isolators.

3.2 The Methodology of the Chapter

The objectives of the research in this chapter are: exploring the mechanisms, assessing the performance and studying the advantages of FFFluid isolation system. Different methodologies were carried out to fulfil these objectives. Figure 3-1 shows the steps that were taken in this chapter.

Studying the mechanisms of FFFluid was achieved via a theoretical investigation. This investigation was on the components of the FFFluid system and the previous research of this mixture. Initial experiments which supported this study were also carried out. Such experiments aimed to observe the behaviour of FFFluid's components and to study the function of fluid with foam particles by using transparency packages.

The second objective was assessment the performance of the FFFluid system. As there are no mathematical models to predict the performance of the FFFluid system, the performance of FFFluid was achieved via experimental investigations. This investigation was started by designing a test rig of a FFFluid, then it was examined by using a hydraulic actuator system. Such actuator is able to apply static and dynamic loads, also it is able to record the suitable input and output data via DAQ of the system.

The final objective was studying the advantages of the FFFluid system. This was achieved by comparing the performance of FFFluid rig and the performance of existing isolators. Such comparisons were obtained by implementing different methodologies. The Matlab Simulink was used to compare the performance of FFFluid rig and spring-damper unit. While experiments were carried out to comparing the performance of a dry foam and FFFluid under a static load. Moreover, theoretical investigations were carried out to explore other advantages of FFFluid.

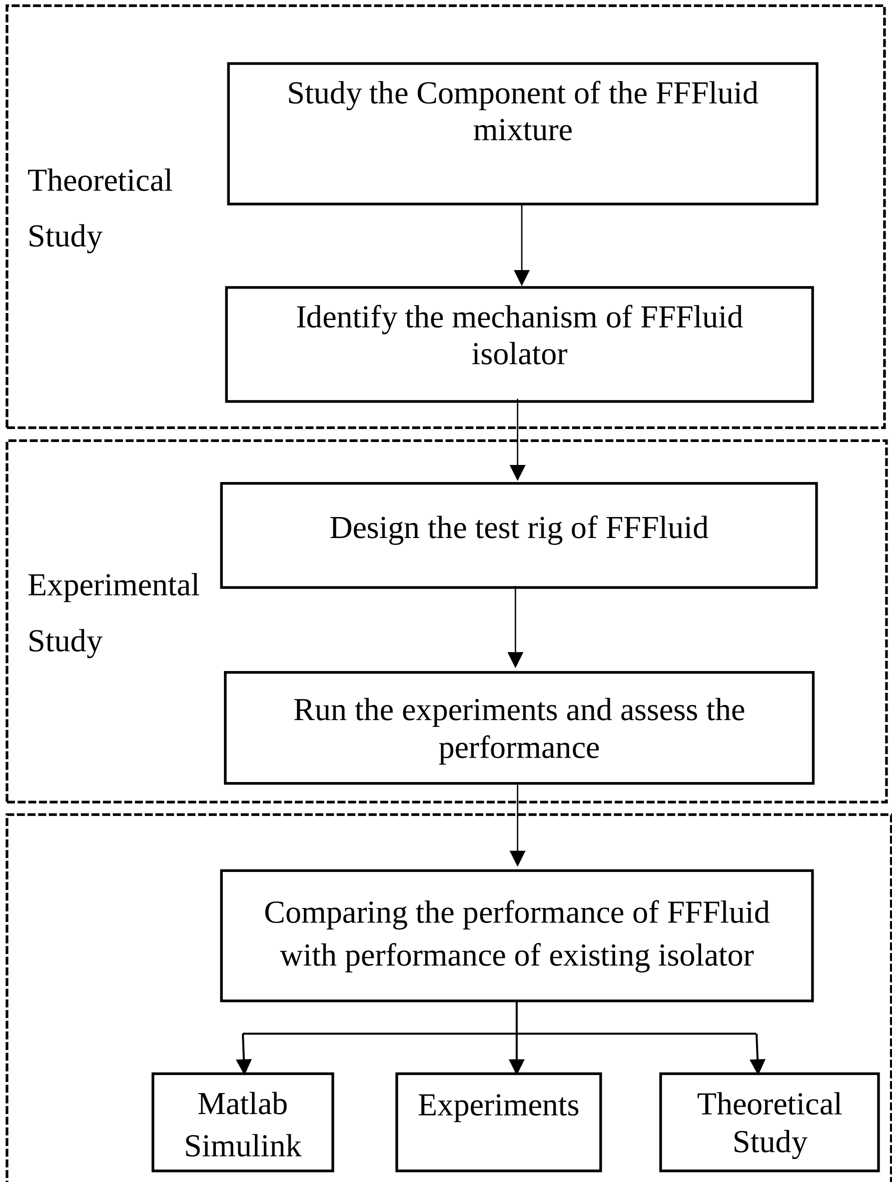


Figure 3-1: Steps (Methods) executed in the chapter

3.3 Loading Machine

Experiments were performed on a Losenhausen hydraulic testing machine as shown in Figure 3-2. This machine is able to apply displacement up to ± 75mm and loads up to 100kN. Losenhausen machine is equipped with a load cell (made by Interface company, model number 1220ACK-250KN-B), displacement transducer (made by RDPE company, part number: ACT3000), and software for continuous data acquisition (MTS 793 software and MTS FlexTest controller). So, there is a possibility to record the input displacement against the output force of the test rig. Subsequently, the displacement vs force diagram could be obtained for both static and dynamic load. Hence, stiffness and damping coefficient could be determined from such diagrams.

Losenhausen hydraulic testing machine has calibrated recently by Denison Mayes Group. The error in this machine is around % 0.25 and uncertainty is % 0.33. Therefore, this machine was selected to examine the performance of FFFluid test rig for all experiments in this research.

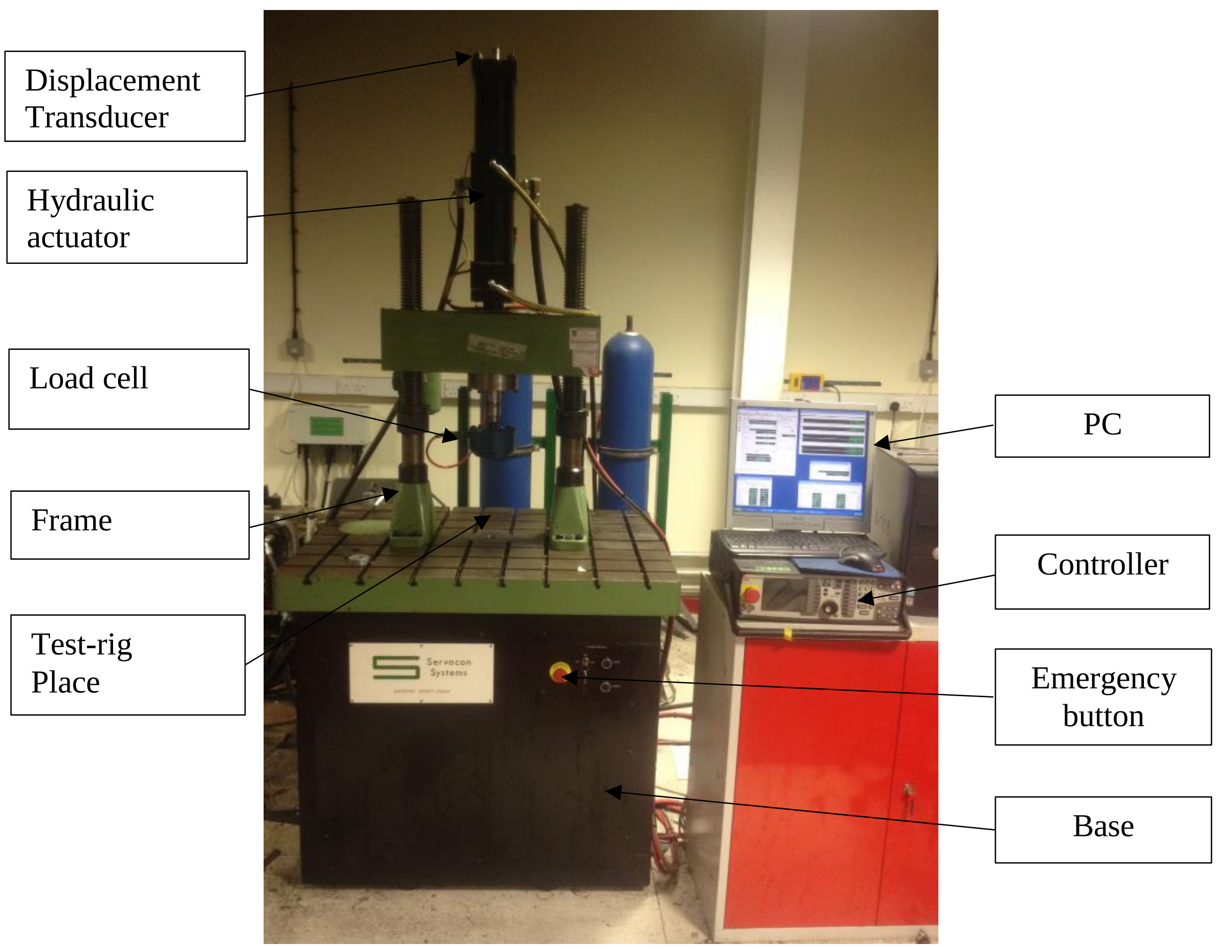


Figure 3-2: Losenhausen hydraulic testing machine.

3.4 Foam Filled Fluid Technology

Foam Filled Fluid consists of large numbers of small flexible capsules mixed within an incompressible matrix liquid. The mixture (known as FFFluid) is contained in a closed vessel. Such closed vessel should be able to change its volume (shrink and stretch) during the loading. This composite material enhances the performance of flexible foam material for eliminating unwanted vibration. Such a mixture exhibits bulk compression characteristics and acts like an elastic fluid when subjected to impact or loading. The three primary components FFFluid are:

- The particles of flexible foam material.

- The matrix fluid.

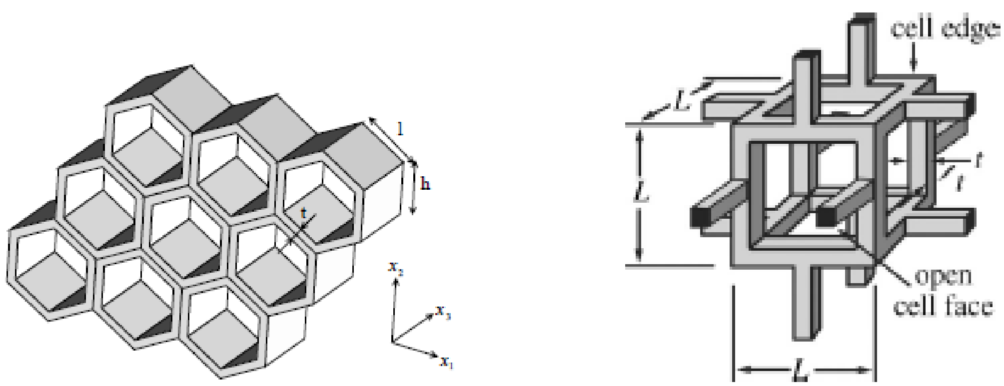
- The package.

In the following section, the components of the FFFluid mixture are presented.

3.4.1 Flexible Foam Material

This material has both viscous and elastic properties, allowing it to reduce or eliminate kinetic energy if it occurs under a cycle load. The foam material is made up of an interconnected network of solid plates or struts, which form the faces and edges of individual cells (pores). The foams can be divided into two different types: open-cell foams with continuous edges; and closed-cell foams, where the faces are closed and each cell is sealed. In the FFFluid mixture, flexible foam capsules can take any form. Several types of foams have been used in designing FFFluid samples, such as expanded polystyrene beads, polyethylene, hollow tubes, hollow rubber balls, and polymeric microspheres [18, 115]. FFFluid could be designed by using both flexible dense solid and foam. However, the concentration in this Thesis is on using flexible foams only, because their low cost and ready availability in small particles.

Flexible foam materials are described by the geometric structure of the cells, which includes both the shape and size of the cells and the way the cells are distributed. There are two types of geometric foam as shown in Figure 3-3. Figure 3-3a shows a honeycomb structure (two-dimensional) while Figure 3-3b shows the structure of foam (three dimensional). The response of such materials also depends on the direction of loading. For example, the behaviour of the honeycomb structure when loaded in the direction of the X1 or X2 axes is different to the conduct of the structure during loading in the X3 direction. Furthermore, the behaviour of a foam cell when loaded along one of the principal axes (perpendicular to the cell face and edge) is different to the behaviour of the cell during loading in other axes (any axes that are not perpendicular to the face of the foam).



a b

Figure 3-3: Structure of cellular solid: (a) Honeycomb structure (b) Cell foam structure.

Figure 3-4 illustrates a schematic of the compressive stress-strain curves for foam material by applying load along principal axis. The elastomeric foam of the cell walls bends first, and a linear elastic deformation is obtained. The linear elastic region is developed at low stresses until a critical stress is reached; then, if the compression increases beyond this point, the cell walls undergo collapse by elastic buckling or plastic yielding depending on the type of material the cell wall is made of. This results in either (i) a plateau of elastic buckling (recoverable deformation) for soft elastic foam materials; or (ii) a plateau of plastic buckling (non-recoverable deformation) for hard elastic foam materials. The last stage of deformation is densification; the foam essentially becomes a solid as the gases inside cells are all evacuated, which is composed of the solid material from which it is made.

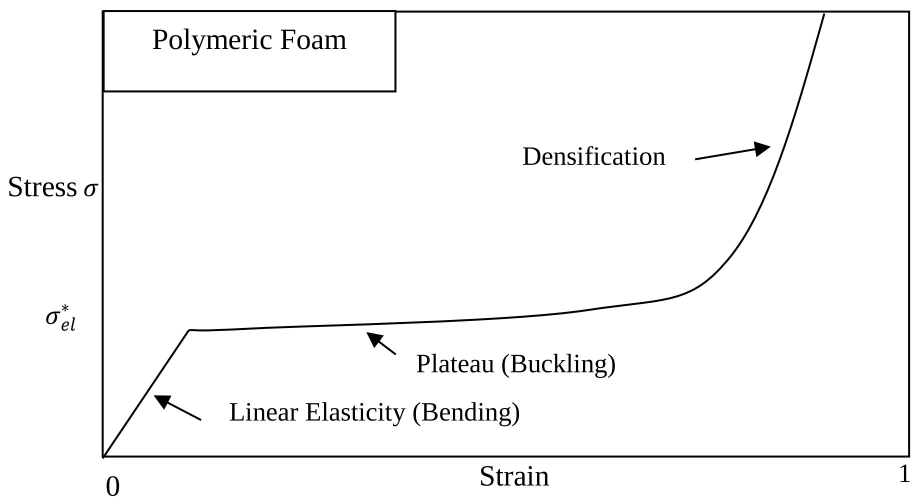


Figure 3-4: compressive stress-strain curves for foam material [116]

If designing a low-cost packaging that is only to be used once, either elastic-buckling or plastic-buckling material could be used. However, for designing an anti-vibration device that needs to withstand the vibrations produced by repetitive motion, it is important for the designer to ensure that the applied stress does not reach the critical elastic stress. Therefore, it is essential to determine the maximum elastic stress of the foam material, which is calculated by using the following expression [66]:

$$\sigma_{el}^{*} = 0.05E_s\left(\frac{\rho^*}{\rho_s}\right)^2 + p_0 - p_{at} \tag{3-1}$$

where σ_{el}^{*} is the elastic stress, E_s is the Young's modulus of the material that the foam is made of, ρ^* is the density of the cellular solid structure, ρ_s is the density of the solid material, p_0 is the initial pressure of the fluid inside the cells, and p_{at} is the atmosphere pressure.

In FFFluid devices, the particles of the foam material are subjected to multi-axis loading (multi-loading). In this case the stress is shared on these axes, therefore the elastic region could be larger during multi-loading [66]. During compression of the foam in FFFluid, it is assumed that there are three principal stresses and that these stresses are all equal. This means there is no bending, and the plastic collapse occurs at a much higher stress than with one axis loading. This stress is three times higher than the stress during compression in uni-axial loading [66]. Hence, the maximum elastic stress of the foam material in an FFFluid sample is determined by using:

$$\sigma_{el}^{*} = 3\left[0.05E_s\left(\frac{\rho^*}{\rho_s}\right)^2 + p_0 - p_{at}\right] \tag{3-2}$$

3.4.2 The Matrix Fluid

Any fluid that is able to transfer hydraulic pressure changes is potentially viable. However, the fluid must cause no chemical reaction with the flexible foam materials. Various types of fluid have been used previously in FFFluid applications such as grease, silicone oil, glycol anti-freeze, wall-paper paste, Vaseline, and aviation hydraulic oil [110].

Using the fluid in a foam material appears in a previous concept known as Fluid Filled Foam (FFFoam). The standard foam is impregnated with air that flows out of the structure during loading. The air has an extremely low viscosity such that its effects are usually neglected. In FFFoam concept, water or any other liquids are used instead of gases. When using water or other liquids instead of air; Therefore, the viscous contribution of liquid to the dynamic response of the structure can be significant. It was found that there is a possibility to reduce the weight of motorcycle helmet by replacing part of the dry polystyrene layer with an open cell polyurethane impregnated with glycerol (as shown in Figure 3-5), and maintaining the same performance [117]. A variety of engineering applications could be benefit greatly from the presented FFFoam, particularly shock absorbers and protective equipment [118].

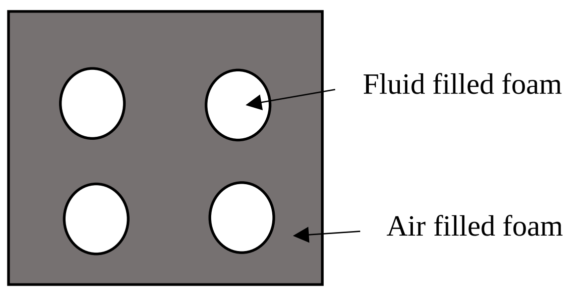


Figure 3-5: Cross sectional view of FFFoam

Although there are improvements in the performance of anti-vibration devices by using FFFoam structures instead of dry foam material, FFFoam concept does not solve the issue of poor stress distribution on foam material during the loading. The stress in the surface under the load directly is higher than stress in other areas. Also, FFFoam requires another layer of dry foam to prevent the impregnation liquid from squeezing-out of the structure [113].

In the new concept of Foam Filled fluid (FFFluid technology), the fluid is mixed with particles of flexible foam material in an enclosed package. The viscous fluid flows between (around) the particles as shown in Figure 3-6. Such movements distribute the stress around all particles of foams; therefore, these particles are deformed equally. Also, during the compression, the fluid shears due to the fact the particles shrink and the ratio of fluid volume to material volume changes, which provide another mechanism for isolating the level of vibrations.

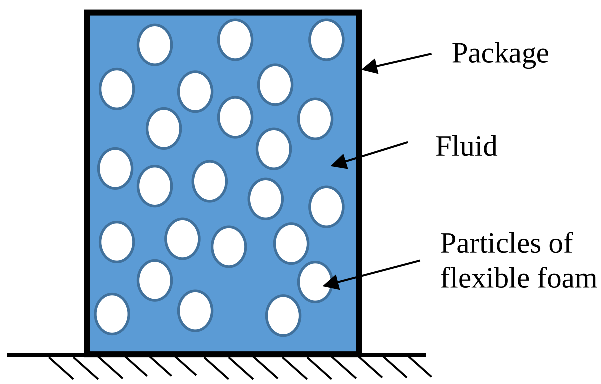


Figure 3-6: Movement of fluid in FFFluids

The three functions of the fluid in the FFFluid concept are:

- To transfer pressure throughout the whole of the interior of the container.
- To provide viscous damping during a dynamic loading, due to movement of fluid between particles or through them.
- To act as a lubricant, allowing relative movement between adjacent particles of foam.

To illustrate the function of the fluid, an experiment was carried out to examine the differences between the behaviour of dry foam particles and that of foam particles surrounded by fluid in enclosed containers.

Two containers were filled with particles of flexible foam, the first vessel contained 55ml of dry polystyrene (PS) foam (average diameter 1.5mm), while the second vessel contained a

55ml blend of the same type and amount of foam particles with water. A load equal to 5kN was applied to both containers for 5 minutes before being removed. The volumes of the containers were recorded against time, and the results are shown in Figure 3-7.

The dry foam was compressed by about 12% of its volume with the load applied, while the blend of fluid/foam was only compressed by 8% of its volume. In the dry foam package, the foam pieces remained locked together and only made a small recovery in volume after removing the load. Figure 3-8a illustrates the volume of foam before applying a load (55ml) while Figure 3-8b shows the container after applying the load. The volume is seen to become 49ml even after removing the load. For the foam/fluid blend, the original volume was 55ml, and the sample returned to the same volume after the load was removed as shown in Figure 3-9a, and Figure 3-9b. During compression, the matrix liquid allows the particles of foam to move relative to each other and provides hydraulic pressure equalisation, so that the particles are uniformly compressed on all sides. In the case of dry foam, the frictional force between foams prevents relative movement and allows the development of small, high-stress zones at the points of contact between spheres. Moreover, the particles located adjacent to the surface experience higher stress than particles situated in the middle of the container.

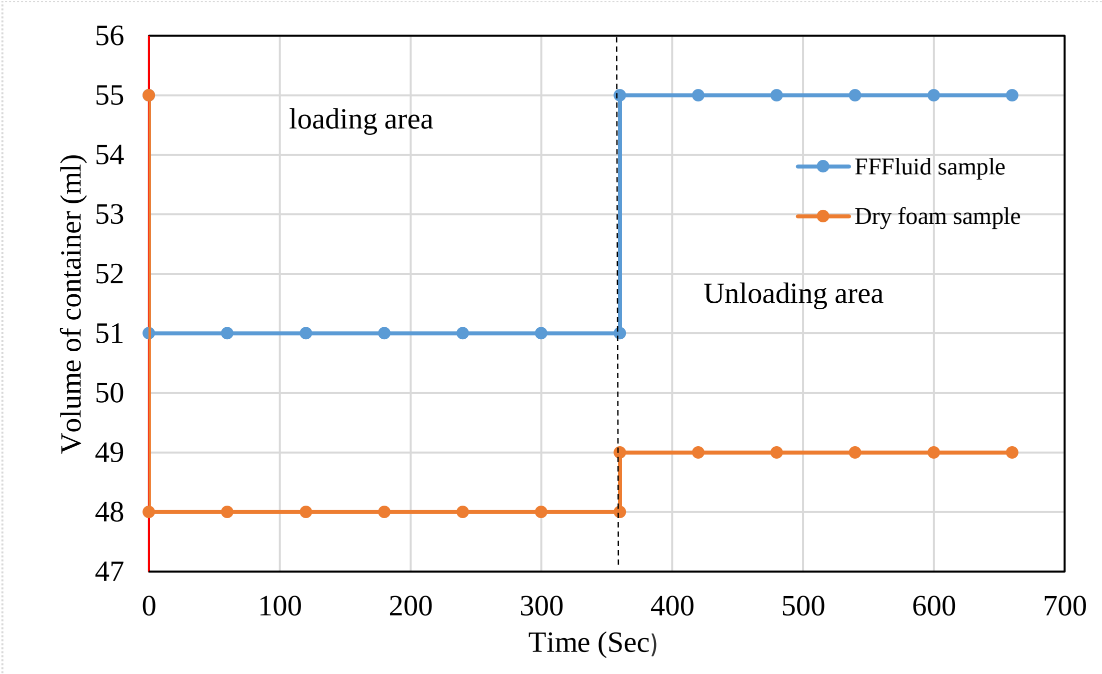
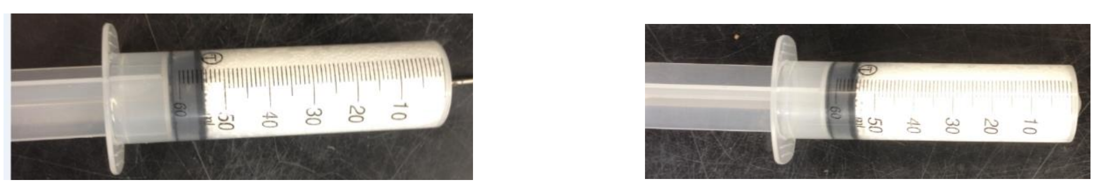


Figure 3-7: Step loading/unloading of columns of dry PS and FFFluid

a b

Figure 3-8: loading/unloading of container of dry PS a) before loading, b) after loading



a b

Figure 3-9: Step loading/unloading of FFFluid: a) before loading, b) after loading

3.4.3 The Package

The last component of the FFFluid concept is the package, an enclosed container used to maintain the mixture of foam particles and fluid so that the pressure is transferred from the fluid to the foam particles. The package is the first element to receive the loads or impacts, which are then transferred to the fluid and flexible foams. Therefore, the package must be flexible (able to expand and contract), and hence the total volume of the package is changeable. Furthermore, the package is used to eliminate some sort of unwanted vibration during expansion and contraction of the package, hence improving the overall dampening mechanism of the system.

Several packages were designed in previous FFFluid devices, such as a piston and cylinder arrangement and a rubber cylinder as shown in Figure 3-10 [15, 119, 120]. A variety of other packages have been proposed, as shown in Figure 3-11.

a

b

Figure 3-10: Design package of FFFluid devices: a) piston and cylinder arrangement, b) rubber cylinder and metal cylinder.

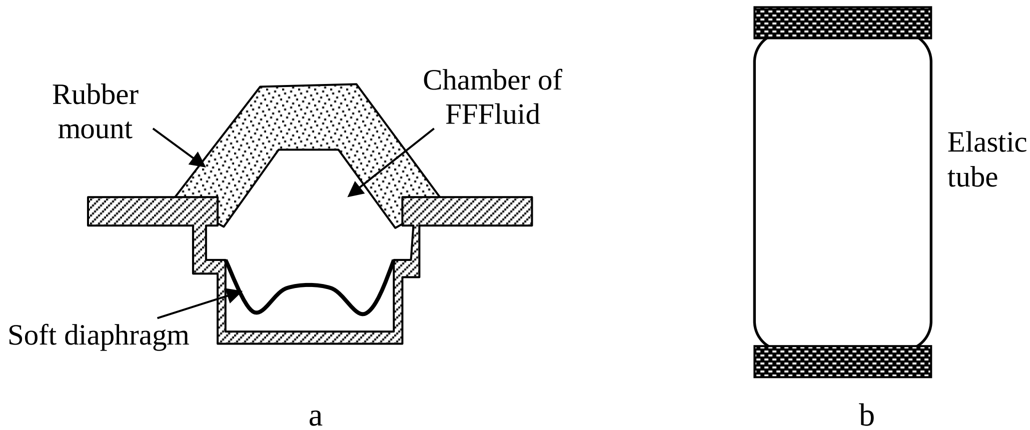


Figure 3-11: Proposed package of FFFluid devices: a) Elastomer mount, b) elastic tube.

3.5 FFFluid Isolator (FFFluid Suspension System)

Since Courtney [110, 121] first describes the technology of FFFluid, several FFFluid devices were proposed and explored. The majority of the proposed applications were introduced by Cheshire Innovation [122]. The main objective of these devices was to minimise the effects of impacts and shocks on equipment or the human body. These applications varied from the protection of the human body [123] to the protection of buildings [14] and vehicle components [15]. The majority of previous FFFluid devices were designed to eliminate unwanted shocks and forces occurring over small time durations such as vehicle bumper device. This research aimed to investigate the potential of FFFluid devices to reduce vibrations in applications where the fluctuating forces are applied continually.

Vehicle suspension systems are one type of vibration isolator system. The function of a suspension system is to support the vehicle weight and to provide comfort and a safe ride. Such systems must have a means of energy dissipation and elastic properties to achieve their functions. Foam Filled Fluids offer several contributions of elastic properties and energy dissipation mechanisms. These mechanisms were not all considered in the previous studies of passive isolator for such a mixture [14]. Therefore, these mechanisms are presented in this section:

- Elastic and damping properties of flexible foam particles: When FFFluid is subjected to a load, the flexible foam particle will be deformed. Therefore, it is dissipating energy via damping properties of foam material. Such material also provides elastic properties to isolating unwanted forces.
- The contribution of gases inside the cells: Besides the energy absorbing mechanism of the foam deformation, there is another mechanism for isolating unwanted vibration in the foam that is related to the fluid (gas) inside the cells. In closed-cell foams, the gas is compressed as the foam deforms, storing energy that is subsequently recovered when the foam is unloaded. The gas is able to increase the stiffness of the structure by increasing its pressure inside the cell. The pressure inside closed capsules changes in response to pressure changes in the surrounding fluid; it is changed from an initial pressure (P_0) to loading pressure (P_1) during loading. The ratio of fluid to capsule volume also increases as the capsules are compressed from initial volume (V_0) to the smaller volume (V_1) as shown in Figure 3-12.

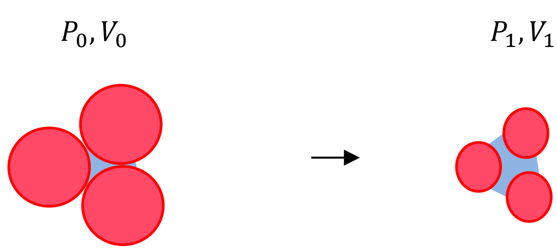


Figure 3-12: Compression air inside cells from (P_0) to (P_1).

- The viscosity of the fluid: The purpose of the fluid in the FFFluid sample is to lubricate the device and pass pressure around all of the foam particles. When the

FFFluid system is subjected to loads, the incompressible liquid moves around all the particles of foam to distribute the pressure, and as such this movement converts some energy into heat through the viscous effect of the fluid. Before loading, the flexible capsules are in close contact and incompressible matrix fluid fills all of the void space between the capsules as shown in Figure 3-13a. Figure 3-13b shows the movement of the fluid around the compressible foam particles by applying the load.

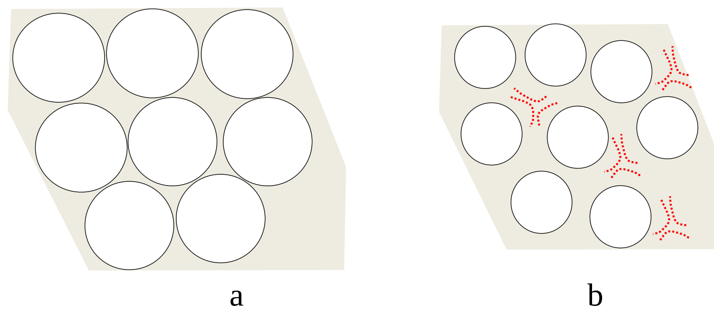


Figure 3-13: The viscous liquid around compressing capsules: a) before loading, B) during loading.

❖ The contribution of the Package: The final contribution of the system is due to the effect of the package. The package can absorb energy due to the viscous impact of the package if it is made of elastic material, or due to the friction effect if it is a piston and cylinder arrangement.

To summarise the energy isolation process, energy is damped by a FFFluid composite material via the following mechanisms: (i) The work done in compressing the flexible material; (ii) The work done by the matrix fluid as it shears; (iii) The package contribution; (iv) The elastic behaviour of the FFFluid due to the compressed flexible particles and cell gases.

3.6 Evaluation the Performance of FFFluid Isolator

FFFluid devices consist of three different components. The interaction between these components leads to complex mechanical behaviour which is why the performance of previous applications of FFFluid was determined experimentally [15, 121]. Hence, the evaluation of the FFFluid isolator performance is also achieved via experimental procedures. First, the test rig is designed, and then it is examined experimentally. The following presented these procedures.

3.6.1 Design the Experimental Rig

Various types of the FFFluid package could be used, but because this investigation was focused on perfomance the FFFluid mixture under loads, a controlled volume package (piston and cylinder arrangement) was selected. The FFFluid isolator could be employed in applications from small devices to the large structures. However, in this research, the focus was on medium sized applications, such as the suspension systems of lightweight vehicles (category L7e). Hence, the test rig was designed to be used for such applications. The maximum static load for the quarter this vehicle is 1.2kN (120 kg) [52]. The rig had a length of 300mm, a radius of 60mm, and the volume of the FFFluid sample was 1litre. These dimensions were chosen based on the space available for suspension systems in lightweight vehicles [52].

The cylinder of the test-rig was designed as an open-ended tube with two covers as shown in Figure 3-14a,b and c. The length of the cylinder is 300mm, the outer radius is 60mm and the inner radius is 57.5mm. Such a design was chosen to simplify the preparation process of the FFFluid mixture. Both the top and base covers were designed to connect to the open-ended tube via screws and nuts as shown in Figure 3-14e. The head of the piston had a diameter of

56.5 mm and two rubber rings (O-rings) were fitted to minimise the column friction between the piston and cylinder while maintaining a seal as shown in Figure 3-14d. All of these parts are presented in Figure 3-14, and the dimensions shown are in millimetres.

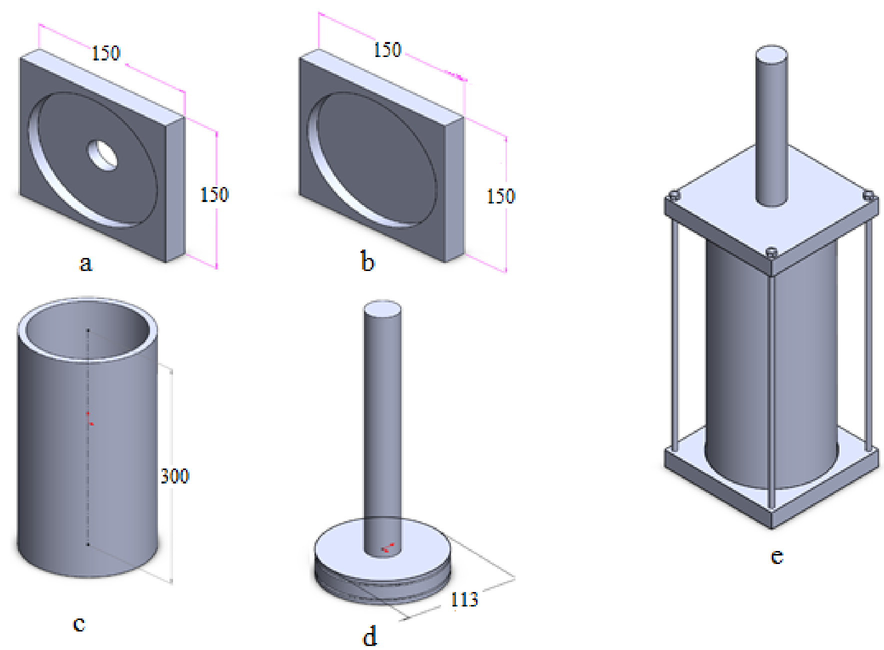


Figure 3-14: Parts: a) Upper Cover, b) Bottom Cover, c) Body of Cylinder, d) Head of Piston, e) assembly parts of test rig

3.6.2 Experimental Procedures

The mixture was made of polyethylene elements (with a density of 70kg/m^3), and silicon oil 1000cSt. This type of foam was chosen because its ability to carry load 1.2kN (120kg) without deformed plastically. As mentioned previously, the flexible foams are deformed elastically under low loads; then it is deformed plastically when the load is increased. Hence, the maximum load applied on the test rig must not reach the plastic region. The elastic stress of polymeric foam under loading (σ_{el}^{*}) could be determined by the using Equation 3-2 [66].

The silicon oil was chosen because its ability to transfer the pressure around all particles of the foams. Also, it does not have any chemical effect on foam material as it already used in previous FFFluid devices [17, 121].

The sample of FFFluid was prepared by adding the small particles of foam to the piston firstly until it was full. The particles have cubic shapes (length 2-3mm). Then, the silicone fluid was poured to fill the voids between the particles of foam. This was followed by mixing the blend properly to ensure evacuated the air. Finally, the cylinder was closed tidily. Figure 3-15 shows the test rig was fixed onto the loading machine. Experiments were performed on a Losenhausen hydraulic testing machine, which was equipped with a load cell, a transducer and software for continuous data acquisition. It is controlled by an MTS FlexTest controller.

Figure 3-15: Set-up test rig on Test System.

The experimental procedures adopted are below:

Test 1: A dynamic displacement (displacement ± 10, frequency1Hz) was applied to the test rig without the FFFluid blend. This test was conducted to characterize the friction between the cylinder and the piston. A lubricant oil was used on the inner surface of the cylinder and the head of the piston of the testrig to minimize such friction. The input displacement and the output force were recorded during the loading, therefore displacement–force graph could be obtained from the recorded data. The graph of this test is shown in Figure 3-16. It noticed some energy was damped due to this friction force.

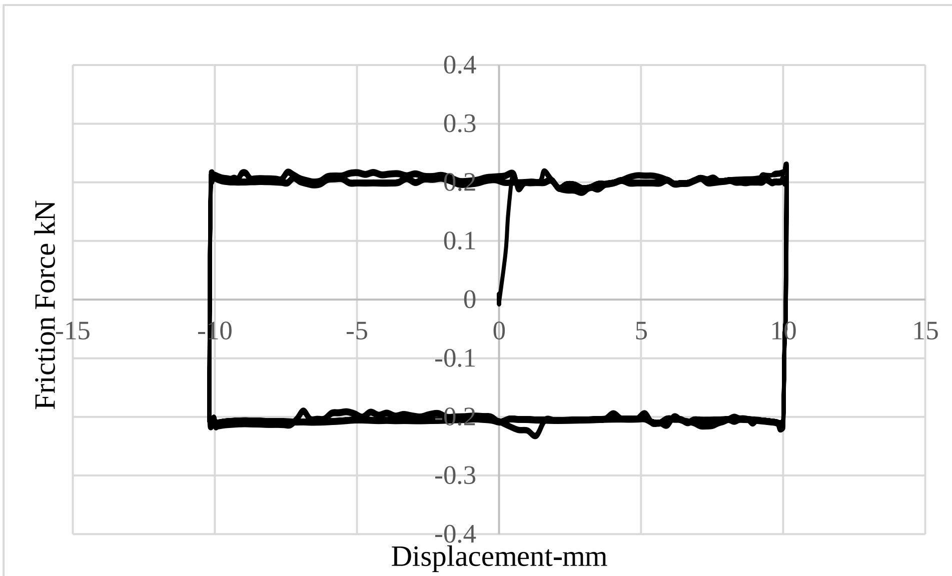


Figure 3-16: Displacement- Friction force graph of the test rig.

Test 2: In this set, the cylinder was filled with FFFluid. Then, a quasi-static load was applied to the test rig. In quasi-static loading, the loading is applied so slowly (very low time) that basically the mixture deforms in a static manner and inertia effects are neglected. The applied displacements ranged from 0 to 50mm in total time equal to 50sec. So, the accelaration was equal to $0.00002 m/s^2$. Hence, the inertia effect could be neglected in such test. This series of test was conducted to check the presence of oil leakage from the test rig under maximum applied displacement. Also, the stiffness coefficient of the test rig could be calculated from this test. The input displacement and the output force were recorded. Therefore, displacement

– force curve could be plotted. Figure 3-17 shows the graph that is obtained from the recorded data. The initial observation based on this finding is that the system has an almost linear relationship between force and displacements at low displacements. The gradient from 0 to 30mm equal to m=0.02. While the gradient is increased at displacements higher than 35mm. The gradient between 35mm to 50mm is about m=0.045. This behaviour of FFFluid resulted from strained flexible foam and from compressed the trapped gas, which they have nonlinear behaviours [66]. Overall, this characteristic makes FFFluid mixture useful for designing vibration isolators.

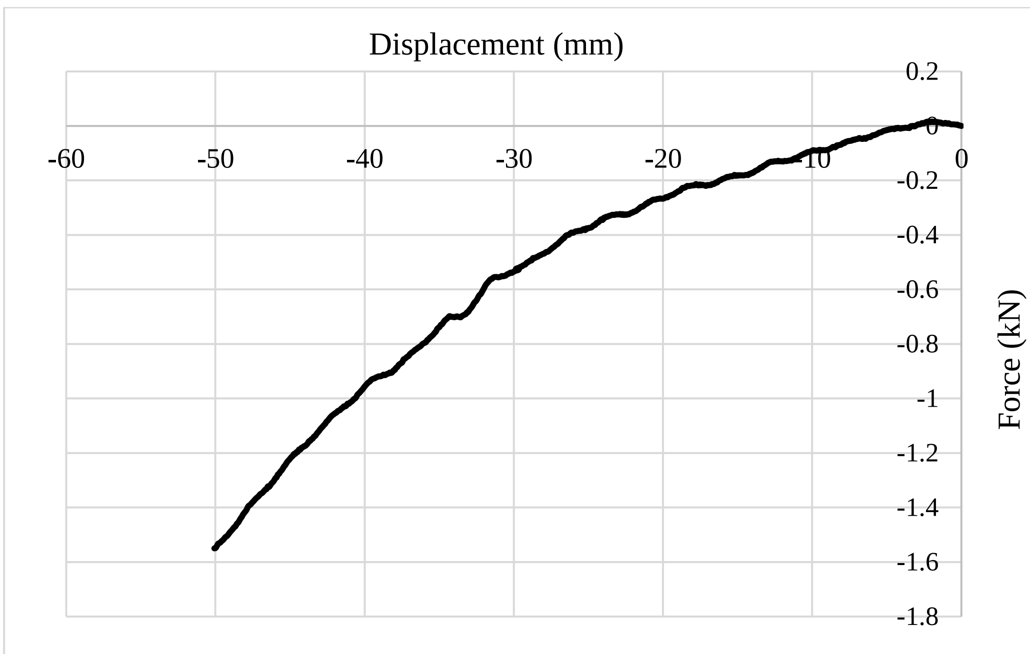


Figure 3-17: Displacement- Force graph of FFFluid under static load.

Test 3: A dynamic displacement (sine wave) was applied to the FFFluid rig. In particular, displacement ± 10 mm were applied at frequency 1Hz. This type of input was chosen due to the ease of analysing such harmonic motion. This series of tests aimed to investigate the behaviour of an FFFluid rig under dynamic loads. The DAQ of the loading machine is recoded the input displacement and output force for this test. Figure 3-18 shows the results of applying the dynamic load for the first five loops, this Figure was plotted from the data recoded of this test. The hysteresis loop due to absorbing energy is clearly observed in this Figure.

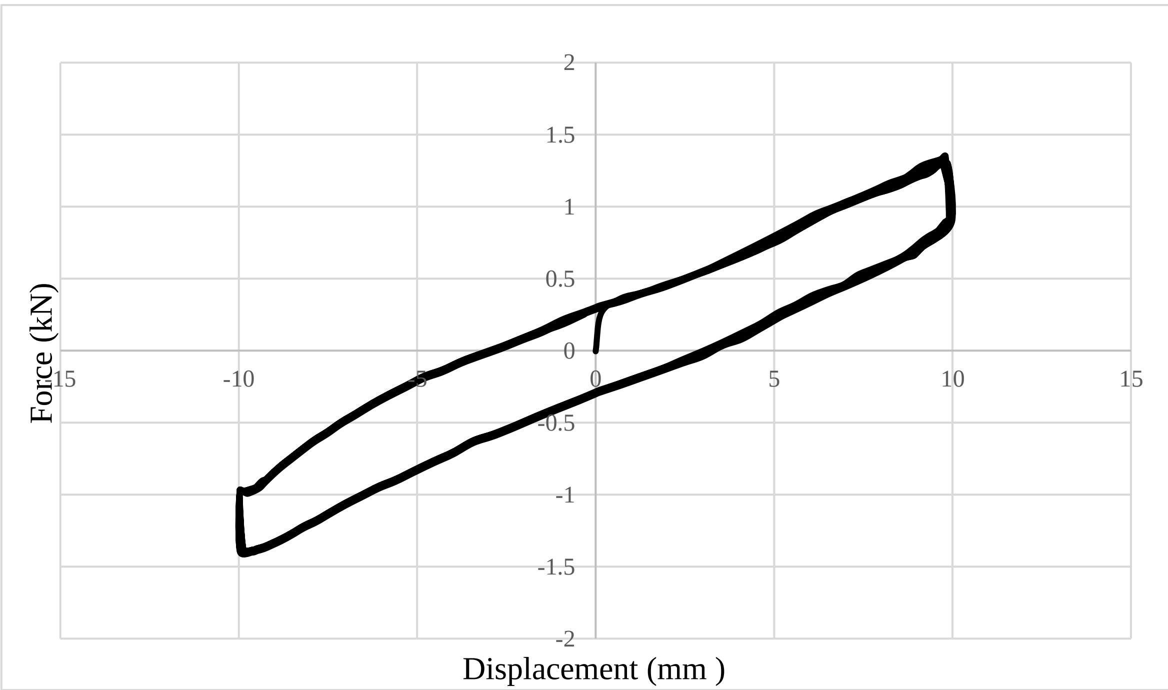


Figure 3-18: Displacement- Force graph of FFFluid under dynamic load.

3.6.3 Determine Coefficients of FFFluid System

In this rig, in addition to the stiffness and damping force, the friction force between the piston and the cylinder also contributes to absorbing unwanted vibrations. The equation of motion of single degree of freedom is:

$$F = m\ddot{x} + c\dot{x} + kx \mp F_{Fr} \tag{3-3}$$

Where: m in the mass of the system, k is the stiffness coefficient, c is the damping, x is the displacement, $\dot{x}$ is the velocity and $\ddot{x}$ is the acceleration. The value of friction is determined from test 1. As illustrated in Figure 3-16, it was determined that the friction force was $F_{Fr} \cong \mp 0.2\text{kN}$.

The stiffness value of the test rig was determined from Figure 3-17. The constant value of the stiffness was calculated experimentally by using Trendline function in Microsoft Excel, and it is equal to $k = 30\text{kN/m}$. However, the test rig behaved nonlinearly at higher amplitude of displacements, and the value of stiffness as function of displacement was also calculated experimentally by using also Trendline function in the Microsoft Excel, it is equal to $k = 600x + 0.4\ \text{kN/m}$.

The last parameter is the damping coefficient. This factor can be determined by plotting the graph of displacement against damping force. Referring to equation 3-3, by knowing the friction force and stiffness force, the damping force could be determined. Figure 3-19 shows the relationship between the displacements and damping force graph by using a constant value of stiffness coefficient.

The vertical displacement of the loop in the damping force gives $2c\dot{x}$ according to theory of damping [22, 124]. Form Figure 3-19, the differences between the upper and lower loops at the middle position of the loops was $2c\dot{x} = 0.19kN$.

The dynamic testing started from the middle position, and it is equal to

$$x = X \sin \omega t \tag{3-4}$$

The velocity and acceleration are:

$$\dot{x} = X\omega \cos \omega t \tag{3-5}$$

$$\ddot{x} = -X\omega^2 \sin \omega t \tag{3-6}$$

Where: X is displacement, and it is equal to ±10mm, and ω in the frequency (red/sec), the value of frequency is $f = 1.00Hz$. To convert the frequency to rad/sec, the next expression is used.

$$\omega = 2\pi f \tag{3-7}$$

Moreover, the total time of one cycle can be determined by

$$T = {}^{1}/_{f} \tag{3-8}$$

At the quarter total time of one cycle $t = T/4$, the position of piston reached to the maximum amplitude, and the velocity at this point $\dot{x} = 0$ (since it was a sine waveform and the dynamic testing started from the middle position).

To determine the velocity at the middle position where $t = 0.5T$. Hence, the velocity of system (by using Equation 3-5) was $\dot{x} = 0.0628\text{m/s}$ system, and thus the damping coefficient of the system was found to be $c = 1.512\text{kN. s/m}$.

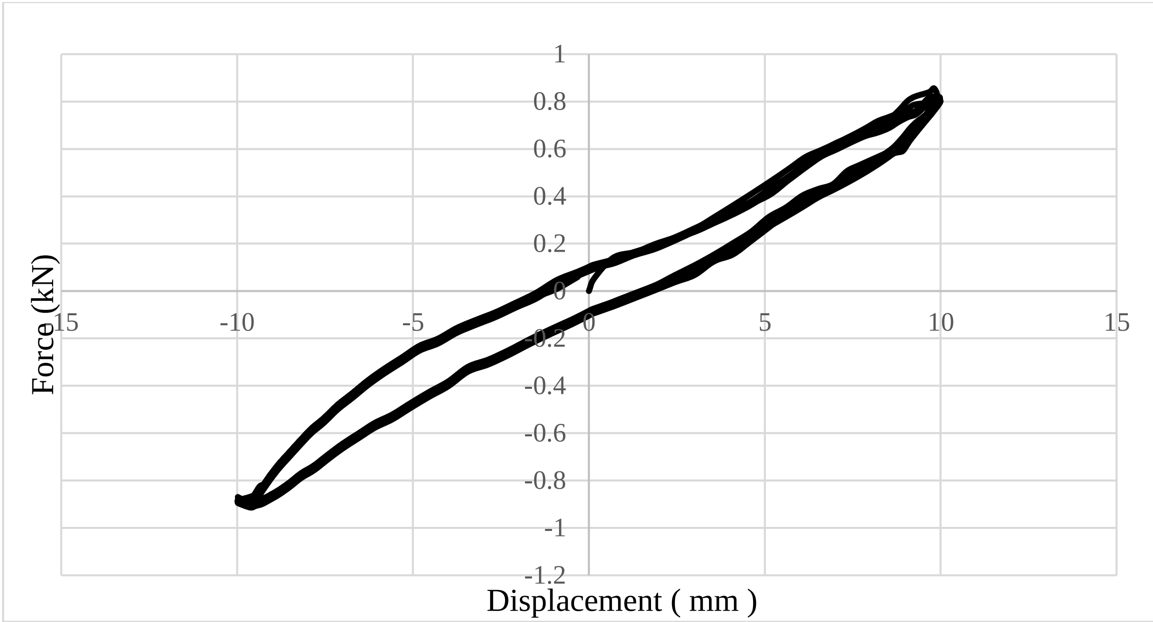


Figure 3-19: Displacement- damping force graph of FFFluid under dynamic load

The experimental procedure was also conducted by using polyethylene foam at two other densities; $30\mathrm{kg/m^3}$ (PE-30) and $120\mathrm{kg/m^3}$ (PE-120). The results obtained from different foam densities are displayed in Table 3-1. The figures of these results are shown in Appendix A.

Table 3-1: Parameter of stiffness and damping coefficients for different foams.

Density ($\mathrm{kg/m^3}$)	Stiffness ($\mathrm{kN/m}$)		Damping coefficient ($\mathrm{Ns/m}$)
	Constant value	Varied value	
PE - 30	$k = 23.4$	$k = 400x + 4.7$	$c = 959$
PE -70	$k = 30$	$k = 600x + 0.4$	$c = 1512$
PE - 125	$k = 34.2$	$k = 300x + 18.5$	$c = 2189$

3.6.4 Evaluation the Performance

To assess the performance of vibration isolators, the force transmissibility is usually used for such systems. It is the ratio between input and output load, which should be less than one to reduce vibrations.

For a system with one degree of freedom, the transmissibility is expressed as equation 2-13. Figure 3-20 shows the evaluation of the transmissibility as a function of the frequency ratio that is a ratio between natural frequency and applied frequency. According to the results, transmissibility is less than one at a frequency ratio more than 1.4, and the amount of transmissibility is improved at higher frequencies.

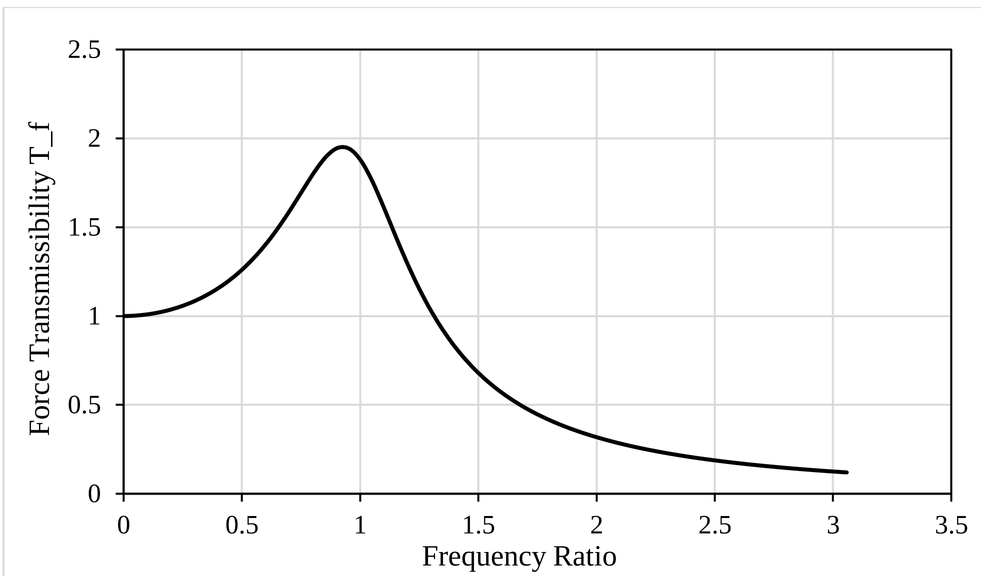


Figure 3-20: Transmissibility of FFFluid isolator.

To evaluate the FFFluid test rig as a suspension system for L7e vehicles, various criteria to measure ride comfort and handling should be considered. A quarter passive suspension system of the vehicle is demonstrated in Figure 3-21.

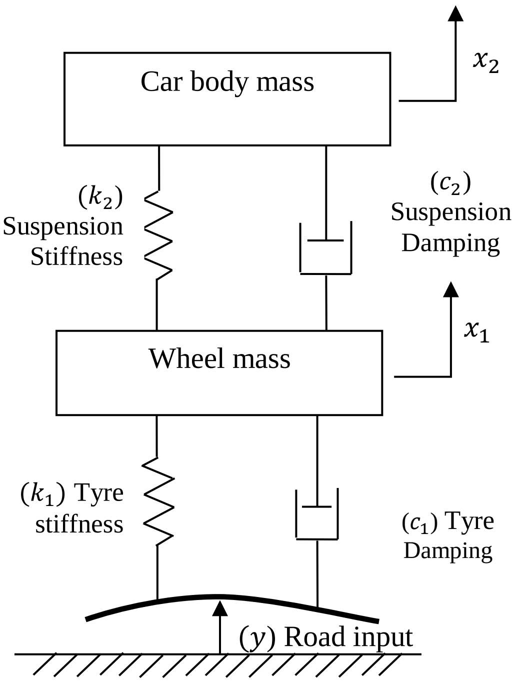


Figure 3-21: The quarter car model representation.

Where m_1 and m_2 are the masses of wheel and body. The displacements of wheel and car body are x_1 and x_2 respectively. The spring coefficients of tyre and body car are k_1 and k_2 respectively. The damper coefficients of tyre and body vehicle are c_1 and c_2 respectively. The road disturbance is y. The following are the equations of motion for passive suspension system of the quarter car model:

$$m_2\ddot{x}_2 = c_2(\dot{x}_1 - \dot{x}_2) + k_2(x_1 - x_2) \tag{3-9}$$

$$m_1\ddot{x}_1 = c_2(\dot{x}_2 - \dot{x}_1) + k_2(x_2 - x_1) + c_1(\dot{y} - \dot{x}_1) + k_1(y - x_1) \tag{3-10}$$

The evaluation was conducted using Simulink tool in Matlab. The block diagram of step input road for a quarter passive car model is illustrated in Appendix B. A comparison with an existing suspension system was also achieved [125] The FFFluid isolator has a stiffness of 30 kN/m and a damping coefficient of 1512 Ns/m. While, the existing system has the next values: tyre stiffness (k_1) 110 kN/m; suspension stiffness (k_2) 44.7 kN/m; suspension damping (c) 1031 Ns/m; wheel mass 15kg; and car body mass 120kg [125].

The input signal of the suspension system was a step input with an amplitude of 0.1m. The vehicle responses are displayed in Figures 3-22 and 3-23. The proposed FFFluid rig produces reasonable performance values comparing with the current suspension systems. The reduction of the driver's vertical displacement peak is evident, along with the settling time and the peak deflection of the suspension system. The settling time of vertical displacement is about 1 second by using the FFFluid rig, while it is almost 2 seconds by using the spring-damper unit. Also, the amplitude of the acceleration decays 90% after the first loop by using the FFFluid rig, while it decays about %80 only after the first loop by using the spring-damper unit.

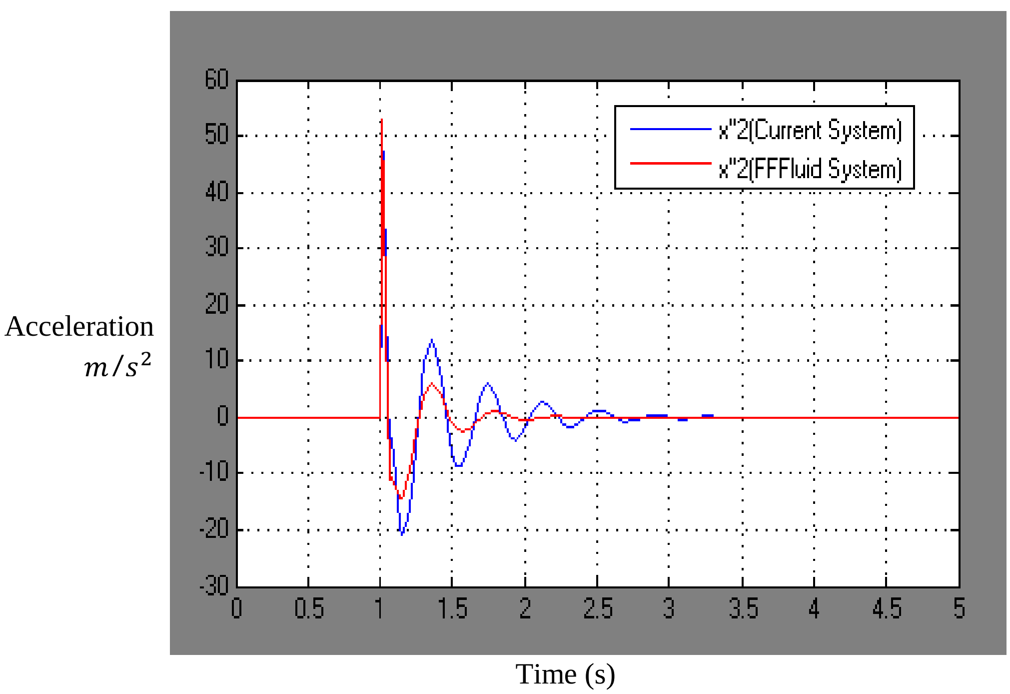


Figure 3-22: Ride comfort

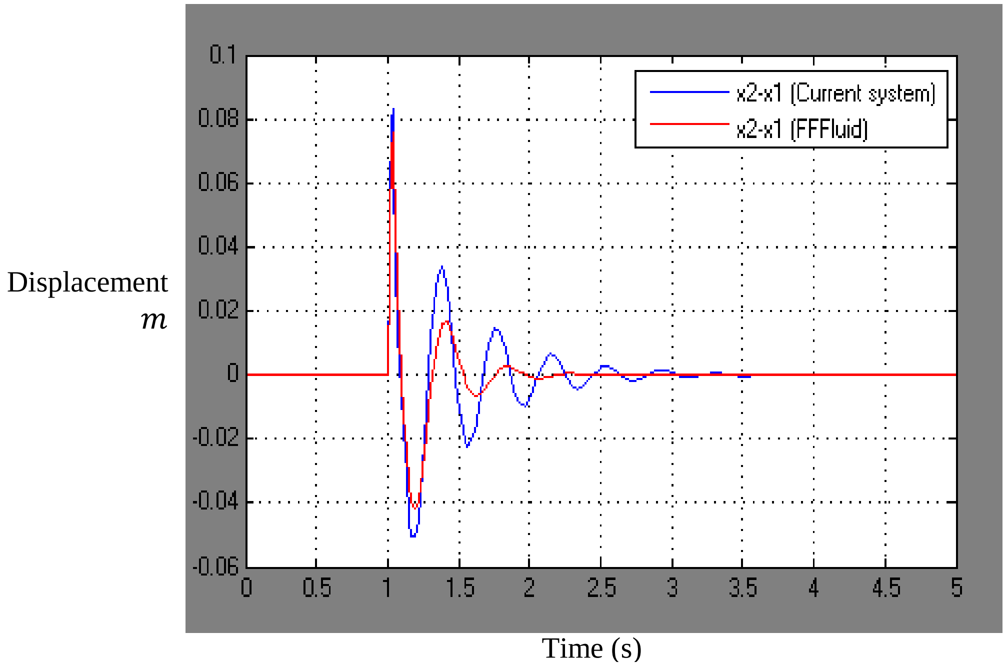


Figure 3-23: Suspension system deflection

3.7 Advantages of Foam Filled Fluid

The main advantage of FFFluid mixture over existing ones is the ability to adjust the stiffness and damping coefficients of the system via adaptive system. The previous thought of such mixture was concentrated on designing passive FFFluid devices. However, the FFFluid concept combines different isolation mechanisms, and controlling the parameters of such composite material lead to control the value of stiffness and damping coefficient of the system.

Generally, it is not easy to adjust both the stiffness and damping of the structures or systems without adding extra actuator such as active control system. However, altering the FFFluid combination of isolating mechanisms modifies the stiffness and damping coefficients. These combinations could be altered by changing the properties of the constituent component. Hence, changing any physical parameters of FFFluid such as increasing the amount of matrix fluid into the package, or modifying the total volume of the FFFluid mixture lead to varied stiffness and damping coefficients of the system.

Adaptive FFFluid system could be achieved via designing different packages. Two packages were proposed as shown in Figure 3-24. In Figure 3-24a, the volume of the FFFluid mixture can be modified by applying an external force. In Figure 3-24b, the volume of FFFluid can be controlled by closing/opening the orifice with an MR valve or servomechanism valve.

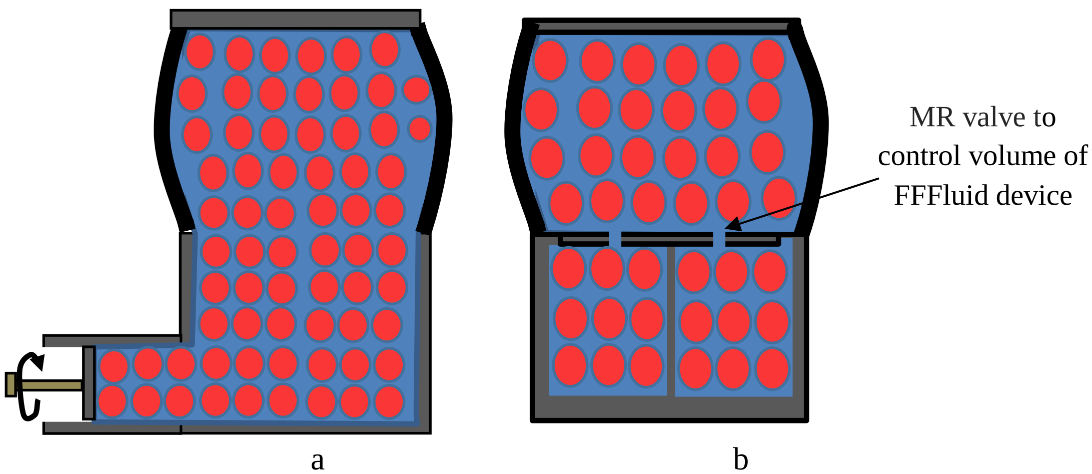


Figure 3-24: Proposed active system of FFFluid.

FFFluid technology possesses other desirable characteristics that can have advantages over exsting systems. The following section presents the benefits of this isolator over the existing one:

3.7.1 Advantages of FFFluid over metal spring and hydraulic damper

- ***Low weight:***

As with the majority of elastomer isolators, FFFluid is characterised by low weight in comparison with an equivalent metal spring-damper unit. The packaging of FFFluid unit may be made of relatively heavy materials, but the main components are the flexible foam elements. These elements have low densities due to their gas voids [66]. So replacing an amount of fluid with foam materials leads to a reduction in the weight of the devices.

Experimentally, the weight of the FFFluid test-rig, which has a piston and cylinder as a package, was found to be around 4.5kg, while the weight of an equivalent spring-damper system is around 6kg [125]. Moreover, the FFFluid rig is an experimental rig, with further potential for saving weight since the current model is over-dimensioned. Thus, the total weight of the car could be reduced by around 6kg by using four FFFluid units together, consequently reducing fuel consumption. Many studies have found that a reduction in vehicle weight (mass) has a very significant effect on fuel consumption in city driving [19], with every 10% decrease in vehicle weight leading to a 8% reduction in fuel consumption.

- ***Ease of design*:**

There is usually a certain degree of complexity in designing coil spring and hydraulic dampers, and this complexity is increased when using other types of springs. However, manufacturing of FFFluid systems is easier than these traditional isolators, as it does not require precision spring and damper. Also, it has less moving parts.

- ***Cost*:**

Besides the ease of designing an FFFluid rig, FFFluid is composed of low cost and easily sourced materials. Moreover, such systems can be manufactured from scrap material, and thus the cost of FFFluid systems is usually low compared with other applications. The design of a low-cost FFFluid buffer has been proposed by Courtney [14]. The current FFFluid rig requires about 750g of polyethylene, 450ml of silicon oil, and a piston and cylinder arrangement. These components cost less than £10 in total. In comparison, the cost of an equivalent spring alone is more than £20 [126].

Also, it is important for a designer to consider the cost of recycling an isolator system at the end of its working life. FFFluid rig is easy to disassemble, and the package and foam particles can be reused in other devices. Alternatively, they can be recycled cold, whereas hot recycling is required for metal springs, which is a more expensive process.

3.7.2 Advantages of Foam Filled Fluid over Elastomer

The FFFluid has other advantages over alternative visco-elastic materials such as dry foam, rubber, and fluid-filled foam (FFFoam) material as described below:

- **Stress Distribution**

Dry foam parts usually fail to distribute a load uniformly under compression load, resulting in local areas of stress concentration in the foam, then shortening the life span of this material. By contrast, FFFluid device are designed to provide proper load distribution during loading. The fluid in an FFFluid mixture is used to transfer stresses equally among and around all particles of flexible foam. Therefore, the stress concentrationis prevented in any local area, and the overall system life span is improved.

Experimentally, two identical samples of expanded polystyrene bead with a mean diameter of 7mm were subjected to quasi-static displacement (65cm). This expermantas were carried out by using Losenhausen Test System. The first sample was a block of dry foam, this sample had dimensions of 120mm in width, 120mm in depth, and 150mm in length. The result of the load application is shown in Figure 3-25. Figure 3-25a shows the foam deformed plasticity in the area directly beneath the applied force. The remaining area was not deformed, as is shown in Figure 3-25b.

The second sample was particles of foam mixed with silicon oil (1000cSt) in close package (FFFluid sample). The FFFluid package has a cylindrical shape, with dimensions 120mm (diameter) and 150mm (length). The results for the FFFluid show that the particles of foam are not deformed plastically, as can be seen in Figure 3-26. A random sample of the polystyrene particles were unchanged in their diameters after the loading was removed. Figure 3-26a shows the sample of polystyrene that was taken from the top of the cylinder, while Figure 3-26a shows the sample of polystyrene that was taken from the bottom of the cylinder.

a

b

Figure 3-25: a sample of dry foam: a) top area, b) bottom area.

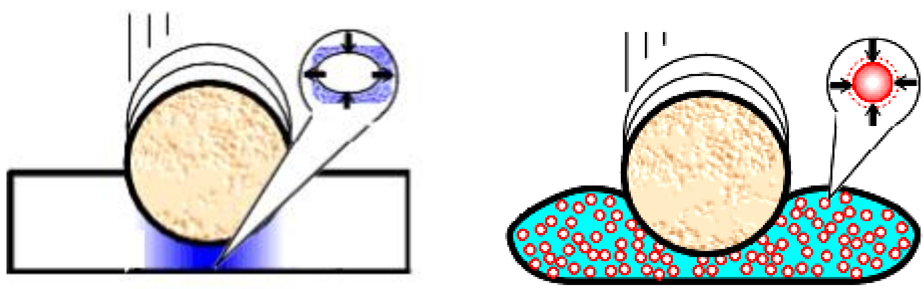
a

b

Figure 3-26: sample of the FFFluid after loading: a) top area, b) bottom area.

- **Axes loading of the elastomer material**

When a dry (traditional) foam sample is subjected to quasi-static compression loads, the cells inside the foam are compressed uniaxially. By contrast, in the case of the foam particles suspended in the FFFluid, the fluid transfers the load equally throughout the surface of all particles. Hence, the foam capsules are compressed multi-axially, as shown in Figure 3-27. In multi-axial loading, the stress caused along one axis is partly cancelled by the others, such that plastic collapse during multi-axial loading occurs with at least three times higher loading than plastic collapse during uni-axial loading [66].

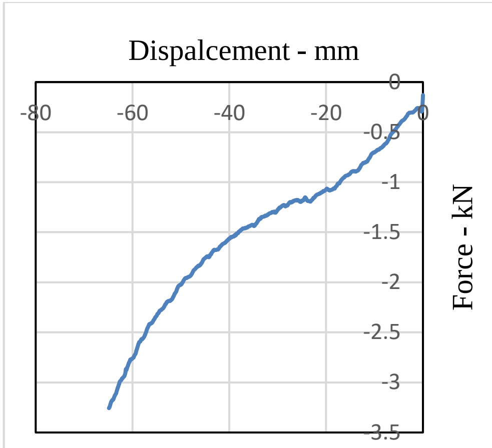


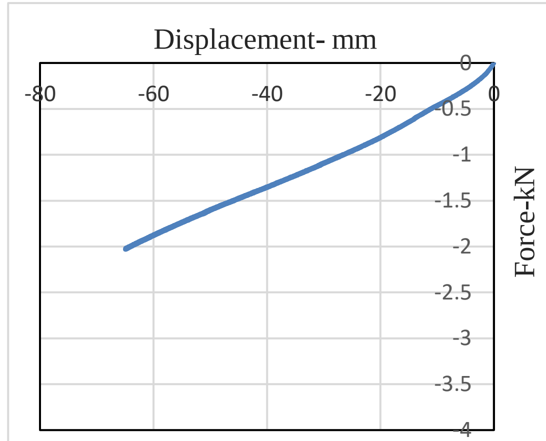
a b

Figure 3-27: Foam under compression load: (a) dry foam, (b) FFFluid.

From the previous experiments, the reaction load resulting from applying static displacement (65mm) to the dry foam sample reached 2kN, as shown in Figure 3-28a. This sample was plastically deformed by a given static displacement equal to 65cm as shown in Figure 3-25a.

For the sample of FFFluid, the reaction load resulting from applying static displacement (65mm) reached approximately 3.5kN, which is more than 60% greater than the dry foam's load as shown in Figure 3-28b. The flexible foam in this sample was deformed elastically as shown in Figure 3-26.





(a) (b)

Figure 3-28: The force-displacement graph of the samples: (a) The FFFluid sample, (b) The dry Foam

- **Temperature**

High temperatures are known to affect the performance of elastomeric polymer materials. For example, the glass temperature of polystyrene (PS) is about 100 °C. At this temperature the Young's Modulus of PS changes from 1000MPa to about 1MPa [66]. The limited operating temperature range is one of the usual disadvantages of elastomeric polymers in practical applications [45, 67]. A thermal advantage of using FFFluid instead of flexible foam is that a matrix liquid with good thermal conductivity can be included to help dissipate heat. Also, the package could be designed specifically to dissipate the heat outside the package, for example by adding fins to increase the external surface area.

- **Package**

The packaging is one of the main components of an FFFluid device. It is used to maintain the blend of the FFFluid. The packaging itself could be an elastic material such as elastic tubes, or stiffer materials such as in piston and cylinder arrangements. Although elastic packaging provides an extra elastomer material for the purpose of dissipating kinetic energy, there are advantages to using stiff packaging, which are:

a) Stiff packaging provides a higher buckling resistance than a dry elastomer. Therefore, devices that have high ratios of lengths to width and depth could be designed using an FFFluid mixture, while it may not possible to produce such devices using dry foam.

b) Designs incorporating FFFluid within a stiff package are capable of coping the side bulges in the elastomeric polymer. The bulge may not be considered a significant issue in the use of elastomer pads in a dry workplace, but if the workplace is oily or sandy, the bulge between the elastomer and the metal can cause malfunction of the isolator [45].

- **Sustainability device (*Environmental Considerations)***

There has been a significant increase in elastomer scrap produced every year, including flexible polymer foam. This largely comprises discarded packaging and industrial waste, and can constitute a significant environmental threat. It has been suggested that FFFluid devices could be manufactured using such industrial scrap, the result being that the device may be produced in a low cost and eco-friendly way. An additional appealing characteristic of FFFluid devices is that they can be recycled ‘cold’; i.e. the used mixture of fluid and foam can be re-used by placing it directly into another FFFluid device.

- **Smart Properties**

A smart structure is one that has the ability to adapt to environmental conditions according to the design requirements. As a rule, the adjustments are developed and performed in order to increase the efficiency or safety of the structure.

A smart structure could be designed using an FFFluid mixture without any external control system. For example, a vehicle bumper system has been designed using the technology [15]. This structure can have a high stiffness in an impact between opposing vehicles, and lower stiffness in the case of an impact between a vehicle and pedestrian. With a small impacting body, the elastic fluid is shifted sideways of the impact zone (Figure 3-29a). The impacting body would "see" the FFFluid package as a wide, soft cushion. The matrix fluid transmits pressure changes, allowing the foam material to the sides of the impact zone to also participate in absorbing impact energy. A larger object involves a wider contact area during an impact. In this case, if the object is larger than the bumper, there would be no movement of the foam material sideways from the impact zone (Figure 3-29b). The rate at which the FFFluid material is crushed would be higher, while the FFFluid material would have a higher stiffness.

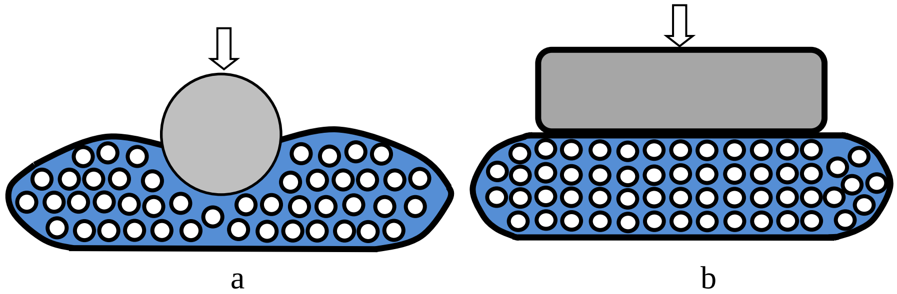


Figure 3-29: Concept of FFFluid shock isolator: (a) Stiffness (b) Stiffer

Although there are several advantages to using an FFFluid isolator, there are also some drawbacks, which are the weight and leakage problems. The unit cost and weight of an FFFluid pad are considered to be advantages of this system in comparison with metal springs-damper unit; however, these are still relatively high compared with many available elastomer materials. According to Courtney, an FFFluid pad is three times the weight of a dry elastomer; however, this percentage depends on the size and the package used, which varies between applications for this technology [115]. Leakage problems are another issue which can occur if an elastic package is punctured by a sharp instrument or if bad sealing is used.

3-8 Summary of Chapter 3

Foam Filled Fluid (FFFluid) is a promising composite material for adaptive vibration suspension systems in lightweight vehicles. FFFluid is a technology, which utilises the viscosity of the fluid and the visco-elastic properties of foam in vibration isolation systems. Therefore, this chapter had presented a study of the FFFluid rig, including the components of the system and the mechanism of FFFluid in isolation vibration.

Moreover, the manufacture and testing of FFFluid rig were described in this chapter. The rig was examined experimentally under a quasi-static load and a dynamic load. The transmissibility of such a system was determined based on the results of the experiments, while other criteria were evaluated using the Matlab Simulink tool. Such a rig offers a reasonable performance as well as having the advantages of reducing the cost and weight of the overall system.To further highlight these benefits, additional experiments were carried out.

The performance of FFFluid was found to be affected by several parameters, such as the relative density of foam and the volume of the mixture. Therefore, the parameters of the FFFluid vibration isolator deserve further investigation.

In conclusion, this chapter represented an important first step to help demonstrate the possibility of designing a vibration isolator based on Foam Filled Fluid (FFFluid). The contents of this chapter were published as two peer-reviews papers, as shown in Appendix H and I.

CHAPTER 4

CHARACTERISATION PARAMETERS OF FOAM FILLED FLUID (FFFluid)

4.1 Introduction

In Chapter 3, a test-rig of FFFluid was designed and the behaviour of such rig was examined. The experimental results of the FFFluid performance concluded that the performance of such a system depends on several parameters, these include the relative density of foam used and the volume of the mixture. This chapter aims to present characterisation process parameters that have an influence on the behaviour of the FFFluid isolation system.

This chapter is organised into three parts. The first part introduces the developed methodology of this chapter. The second part presents a theoretical study of Foam Filled Fluid (FFFluid) parameters. This study covers the characteristics of the foam, the characteristics of the fluid, and the characteristics of the package. The third part of the chapter introduces the experimental investigation of FFFluid parameters. The background of the Design of Experiments (DoE) is reviewed, including the importance and impact of DoE and the common methods of DoE. The experimental methodology for examining the FFFluid system is then explained, including parameter selection, the Taguchi method for conducting experiments and the statistical analysis of the results.

4.2 The Methodology of the Chapter

Different methods could be used to characterise parameters of the FFFluid system. A review of previous studies may not sufficiently explore all of the system's parameters due to the inadequacy and incompleteness of published articles [14, 15]. On the other hand, investigation all of the FFFluid parameters experimentally would be costly in terms of both money and time due to the complexity of interaction between the system parameters [14].

Hence, to characterise the parameters of this material, a unique approach in this area which combining theoretical exploration with experimental activity is proposed. The systematic study of the FFFluid, which was carried out in this research, consisted of the following steps as illustrated in Figure 4-1. These steps begin with an identification of the system parameters. This is followed by formulation of the hypothesis which explains the influence of one parameter on the total system' coefficients. Then, the hypotheses are collected to validate the system responses. The theoretical study is followed by an experimental investigation of the most critical parameters of the system to underpin this study. The experimental programme is also used to determine contributions of the critical parameters on the coefficients of the system.

4.3 The Theoretical Study of Foam Filled Fluid (FFFluid) Parameters

A FFFluid system may be considered as a device that transforms inputs to outputs. There are different input types and several factors (parameters) that directly influence the output of the system. Before starting a theoretical study, it is important to define the elements of the input, the process parameters, and the output.

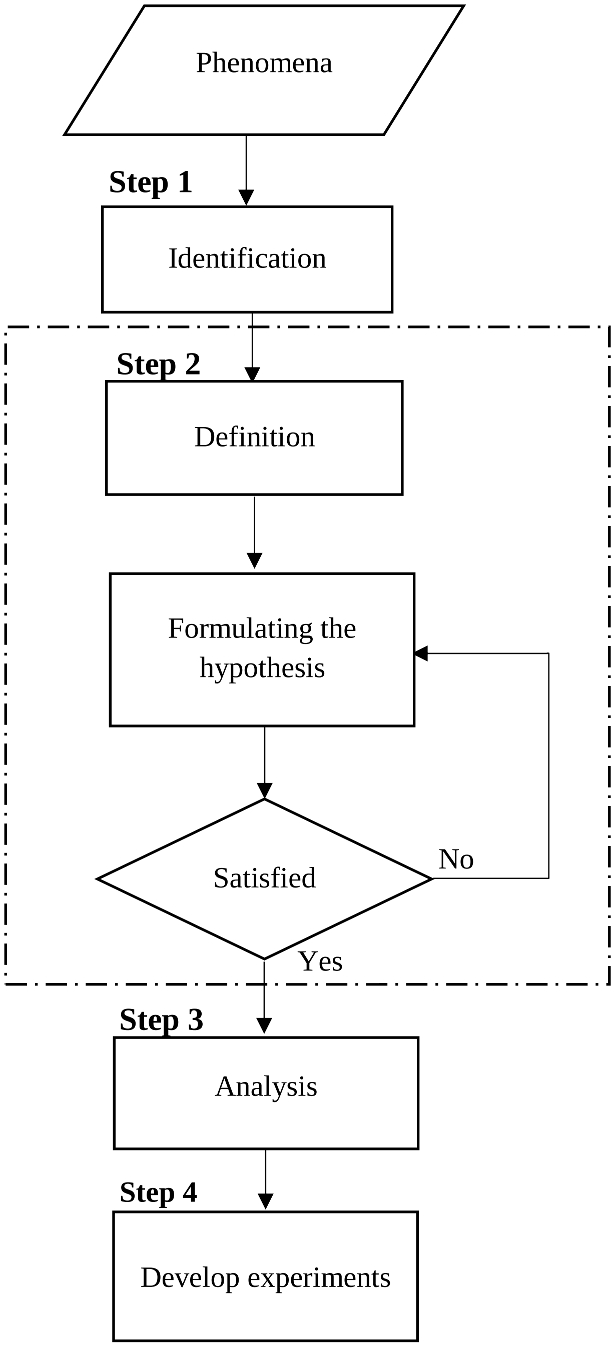


Figure 4-1: Steps of systematic study of FFFluid device.

Input: This relates to the applied loads on the system, which could be expressed in terms of forces, deformations or accelerations. These loads can be divided into two types: static load or dynamic load.

- **Static load:** it is a force that is relatively constant for an extended time, it could be the weight of the isolated object, input displacement (deflection) or a step load with low rate velocity. The amplitude of input force or deflection affect the contributions of the components of the FFFluid mixture. For example, the relationship between stress and strain for the flexible foam is linear at low loads, becoming nonlinear when a critical elastic stress is reached (see Figure 3-2). Hence, the response of an FFFluid mixture depends upon the amplitude of the input displacement or force.
- **Dynamic load:** it involves accelerations of moving loads, it can be harmonic motion (sine or cosine wave), random vibrations or impact load (step load with high rate velocity). Not only the amplitude of the input load have an effect on the response of a FFFluid mixture, but the accelerations of a moving load (frequency) have an influence the performance of the FFFluid mixture. According to Gibson and Meyers [66, 127] the properties of the flexible foam are dependent on the frequency of input loads.

Output: There are two primary coefficients that effect on the performance of FFFluid device. These coefficients are the stiffness coefficient and the damping coefficient of the system. Such coefficients are the main coefficients that are determined by examining FFFluid rig.

Parameters: The behaviour of the FFFluid mixture depends upon several process parameters such as: the viscosity of the fluid, the size of the foam, the relative density of the foam and total volume of the FFFluid mixture. These parameters are grouped into three main categories: foam material variables; matrix fluid variables; and packaging material variables. Each category contains several parameters. The next section presented these parameters in more detail.

4.3.1 Parameters of Flexible Foam

The main component of the FFFluid mixture is the flexible foam particles. The purpose of the particles is to provide a contribution to reducing the negative effect of unwanted vibration in the dynamic system. The characterisation of the foam depends on two separate sets of parameters: the first set contains the intrinsic properties of the materials that the foam is made from; the second set contains the properties of the geometric structure of foam [128, 129]. The most important parameters that affect foam properties are:

Type of Cells in the Foam

The classification of flexible foam cells can be divided into two kinds: closed-cell and open-cell [66, 130]. Both types absorb energy if they were subjected to compression loads due to their viscoelastic properties. However, their cells respond differently; specifically, the linear elastic region and the maximum elastic collapse stresses behaviours are different.

Under a static load or displacement, open cell foam deforms linearly by bending of the cells' walls. The cell walls bend until a critical load is reached at which the cell walls buckle. In the linear region, the Young's modulus of the open-cell structure (E^*) with cubic cells consisting of interconnected cell edges can be estimated with the following model [66].

$$\frac{E^*}{E_s} = \left(\frac{\rho^*}{\rho_s}\right)^2. \tag{4-1}$$

While the maximum elastic collapse is calculated by using the next expression:

$$\sigma_{el}^* = 0.05 E_s \left(\frac{\rho^*}{\rho_s}\right)^2. \tag{4-2}$$

where σ_{el}^* is the elastic stress, E_s is the Young's modulus of the material the foam is made from, ρ^* is the density of the cellular solid structure, and ρ_s is the density of the solid material.

For closed cell foam, the Young's modulus of such cells derives from the sum of the contribution of the foam cell bending and the compression of the air in the cells as shown in Figure 4-2 [130]. The Young's modulus for such structure (E^*) in linear region is calculated by using the next expression

$$\frac{E^*}{E_s} = \left(\frac{\rho^*}{\rho_s}\right)^2 + \frac{p_0(1-2v^*)}{E_s(1-\rho^*/\rho_s)} \tag{4-3}$$

It was assumed that the thickness of cell edges and faces are equal. Also, the elastic collapse stress of such cells can be expressed as [66]:

$$\sigma_{el}^* = 0.05E_s\left(\frac{\rho^*}{\rho_s}\right)^2 + p_0 - p_{at}\ . \tag{4-4}$$

where: v^* is Poisson's ratio, p_0 is the initial pressure of fluid inside the cells, and p_{at} is atmosphere pressure.

Therefore, applying the same load to the two identical types of foams (one is an open-cell foam and another is a closed-cell foam), the maximum elastic stress of the closed-cell foam is greater than maximum elastic stress of the open-cell one by the value of ($p_0 - p_{at}$). Hence, the closed-cell foam is able to carry higher static load than the open-cell foam. While the stiffness of the closed-cell foam is greater than the open cell foam due to the contribution of trapped gas, which is equal to $\frac{p_0(1-2v^*)}{E_s(1-\rho^*/\rho_s)}$.

For dynamic loads, when an open-cell foam is compressed, the air is squeezed out. As the air has a viscosity, work is done forcing air through the interconnected porosity of the foam. Therefore, there is some damping force due to squeezing the air out. In the closed-cell foams, the effect of airflow can be neglected as air stays inside the cells.

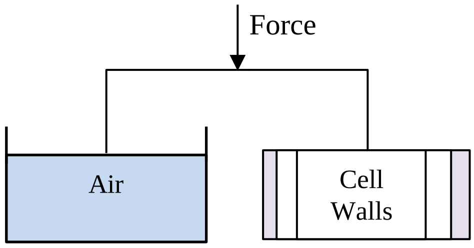


Figure 4-2: Closed-cell foam [130].

It was shown that closed-cell foams would increase the stiffness coefficient of the FFFluid system due to the effect of the enclosed gas. Conversely, the gas in open-cell foams is expected to be squeezed out of cells during dynamic loading which causes to increase the damping contributions. Thus, it can be concluded that the performance of the FFFluid device is influenced by the type of foam cells.

Mechanical Properties of Foam

The maximum elastic stress (σ_{el}^{*}) and Young' modulus of flexible foam are given by equations 4-1 to 4-4 [4, 5]. According to these equations, the main mechanical properties that affect a foam's response are (i) the Young's modulus of the material that the foam is made of, (ii) the initial pressure of gas in case of using closed cell, and (iii) the relative density of the foam, which is expressed as $\rho = \frac{\rho^*}{\rho_s}$. Therefore, changing the material of the flexible foam or increasing the relative density of the foam can increase in the elastic region of the system. Increasing the relative density of flexible foam or the pressure of trapped gas causes to increase the resistance of cell's walls to deformation; hence, the elastic region of such cells is increasing [66].

As displayed in equations 4-2 and 4-4, the relationship between elastic stress and Young's modulus is linear. Doubling the Young's modulus causes to double the maximum elastic stress of the system. While the relationship between the elastic stress and the relative density is a quadratic. This means that doubling the relative density of foam increases the maximum elastic stress of the system four times.

Also, applying a dynamic load to these particles, hysteresis loop (damping) appears in the stress-strain curve. This means that not all of the work is recovered, a fraction is dissipated as heat. This fraction is depending on the type of foam material [66, 129]. Therefore, changing the material of the flexible foam can result in a different hysteresis loop which changes the damping coefficient of the system [18, 66].

Consequently, these parameters (Young's modulus, initial gas pressure, and the relative density of the foam) have an influence on the resulting stiffness and damping coefficients of the system.

The Geometric Structure of the Particles

Any structure of flexible foam particles could be used in FFFluid mixtures. A variety of shapes has been used previously, such as hollow spheres, open tubes, uniform hollow tubes, cubes, and irregular shapes. These particles usually have small sizes [18, 121]. However, the stiffness coefficient of the flexible material depends on its size and shape [45, 131, 132]. It is determined by the following expression [45, 133]:

$$k = \frac{EA}{T_r} \tag{4-5}$$

where: k is the stiffness, E is Young's Modulus, A is the cross-sectional area, and T_r is the thickness. When applying a load to an flexible material, the resulting stress is not equally distributed across all layers of the material. The compression predominantly deforms the surface layers, with little or no deformation of the centre layers. However, by increasing the load, the deformation progressively propagates through the collapsed layers tending towards a more uniform strain distribution at high bulk compression strains. Therefore, if the size of the foam particle is increased, the proportion of compressible material decreases. Hence, increasing the size of foam particle leads to an increase in the stiffness coefficients of the system.

In addition, different volumes and surface areas of the flexible foams can result from change the shape or the size of the particles. The total volume and effective area of the mixture varies based on these parameters. For example, using a spherical shape (diameter 20mm), the surface area and volume are 1256mm^2 and 4187mm^3 respectively. While for a cubic particle (side length 20mm) the surface area is 2400mm^2, and the volume is 8000mm^3. Moreover, changing the diameter of the spherical particle from 20mm to 40mm causes the volume to increase from 4187mm^3 to 33500mm^3. These changes in volume and surface area can modify the maximum elastic region for the flexible particle of foam. Therefore, changing the shape of the particle amends the performance of the FFFluid system.

Number of Foam Particles

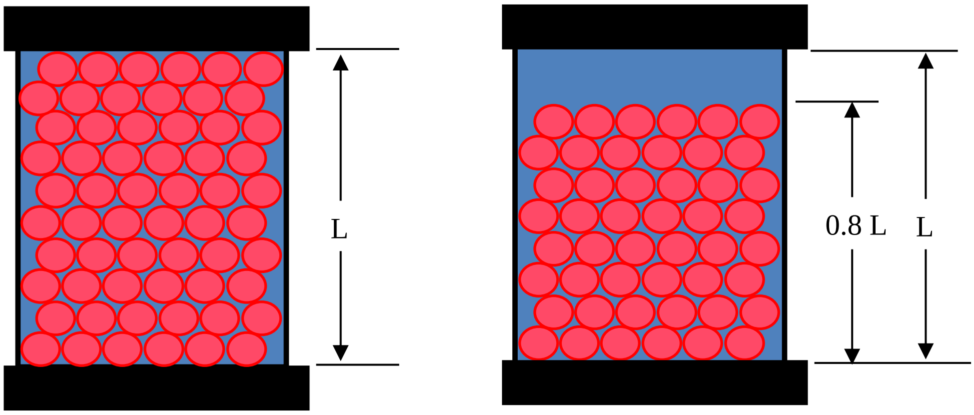
The quantity of foam particles used in this research is measured as a ratio between the height of the package containing foam to the full height of mixture. When the sample of FFFluid is prepared by filling the whole package with foam, this ratio becomes 100%. When 0.8 of the package height is filled with foam and the rest by fluid, the ratio is 80%, as shown in Figure 4-3.

Although the majority of previously studied FFFluid samples filled the full package with foam, some experiments used different percentages of foam to the fluid. Increasing the number of foam particles raises in the amount and the size of the flexible material, which change the response of the system [131, 134]. Hence, increasing the number of particles increases the damping coefficients of the system. However, increasing the number of foam particles lead to decrease the amount of fluid in the container, which leads to decrease the stiffness coefficient of the system. This is because of the incompressibility of the fluid.

An initial investigation to study the effect of the number particle was performed. Two syringes (total volume 60ml) were filled with foam and water; the first syringe was filled with one polystyrene ball (d=18mm) and water. The another syringe was filled with five balls (same size, d=18mm) and water as shown in Figure 4-4. A load equal to 6kN was applied to both containers. The first container was compressed less than 2% of its original volume with the load applied, while the other one was compressed more than 5% of its volume. Therefore, these differences in the compressions level cause a change in the stiffness coefficient of the system.

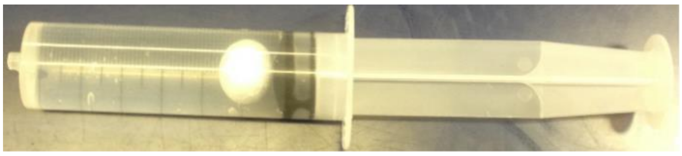
Due to the constant of input load and the piston area of the container for both cases, the pressures around foams particles are equal. However, in the first case, one particle of foam was compressed, which deformed to a certain level. Therefore, there was a small change in the total volume of the container. However, five particles (in the same size) were compressed in the second case. Therefore, the change in the total volume for this case was higher than a change in the total volume for the first case. Hence, the stiffness coefficient of FFFluid increases by decreasing the number of foam particles.

For a dynamic load, the damping mechanism that results from the effect of flexible foam particles increases with raising the number of these particles because of increasing the total volume of flexible material. While the viscous damping of the fluid decreases in such the case due to decline the amount of fluid. For examples, the first package in the previous experiment required 56ml of fluid while the second package used about 40ml of fluid. Therefore, the damping coefficients of the FFFluid system modifies by changing the number of foams particles.



a b

Figure 4-3: Preparing FFFluid mixture: (a) 100% FFFluid, (b) 80% FFFluid



a

b

Figure 4-4: The influence of foam number: (a) one particle, (b) five particles.

4.3.2 Parameters of Fluid

The purpose of the fluid in FFFluid system is to provide an additional contribution to eliminating vibration through shearing of the fluid between and around the foam particles as they are compressed/expand. The fluid also distributes the pressure around all of the particles of flexible foam – depending on the rate of compression/expansion. The main fluid parameter that affect the overall performance of FFFluid devices is the viscosity. Hence, this section presents the impact of this parameter.

Viscosity

Viscosity is the resistance of a liquid to flow; it is the ratio of the shearing stress to the velocity gradient in the fluid [135, 136]. Increased fluid viscosity raises the damping coefficient of a FFFluid system [14], but decreased viscosity fluid transfers pressure more effectively to equalise pressure around the foam particles.

Under the static load, an insignificant changing may occur in the stiffness of the system by changing the viscosity of fluid. However, the viscosity of fluid is a significant factor in the total damping coefficient of the FFFluid system when applying a dynamic load. A previous investigation of fluid viscosity in FFFluid shock absorbers was carried out by Davies and Courtney [14, 110]. Courtney compared the performance of different FFFluid pads by comparing the accelerations extracted from impulse responses. The pad provided a lower acceleration was better in absorbing shocks than the pad provided higher acceleration. The relationship between the gradients of the acceleration and fluid viscosity is shown in Figure 4-5 [12]. This figure shows that the energy absorbing performance is increased by raising the viscosity of the fluid. The figure also shows that the relationship between energy absorption and fluid viscosity is not linear. The greatest improvement in the performance occurs when moving from low viscity (1800cst) to medium viscosity (10400cst) [110]. This trend may be due to the previously highlighted conflict.

Fluid viscosity is a critical factor in the contribution of viscous damping of the system, and changing the fluid viscosity causes the damping coefficient of FFFluid devices to change.

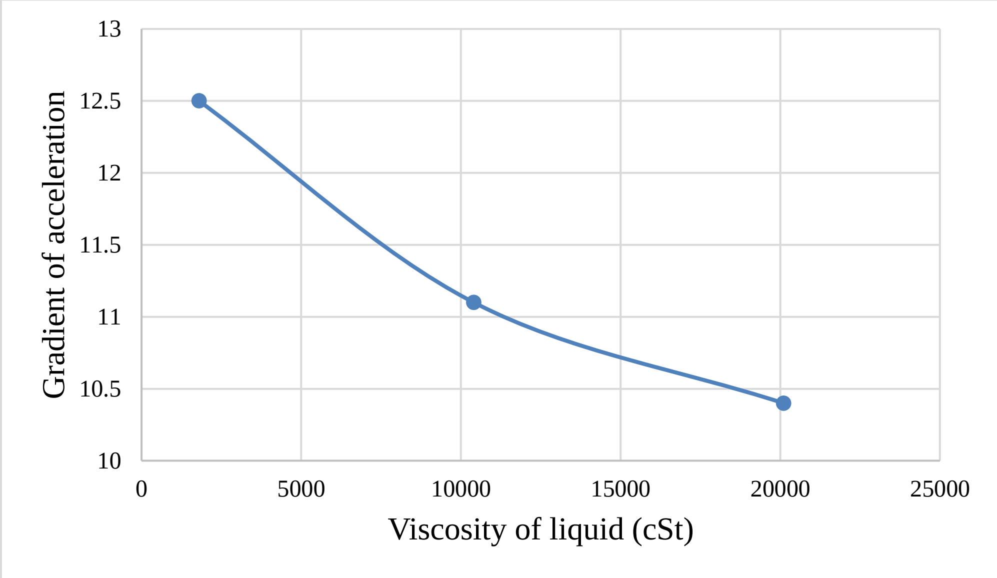


Figure 4-5: Effect of viscosity on the performance of an FFFluid shock absorber [110].

4.3.3 Parameters of Packaging

The function of the packaging is to retain the mixture in an enclosed container. Several designs and geometries of packages could be used in FFFluid devices. These designs and geometries affect the performance of the system, with the most influential parameters of the packaging being:

The Design of Package

Different types of the package could be used, such as flexible bag and piston/cylinder arrangement. The stretch packaging increases the elasticity of the system. This packaging is a critical part of the FFFluid device. If the packaging stretches significantly during the load, then some energy is dissipated. However, overall, the damping coefficient may reduce. This is because the foam particles suffer less compression, and the matrix fluid provides less viscous damping.

For piston/cylinder package, the advantage of using such stiffer packaging is to increase the stiffness of the system. Therefore, it can carry higher a static load, friction force may also appear by using this package. Hence, the total performance of an FFFluid system changes based on the type of package used.

The Volume of Packaging

By increasing the volume of the FFFluid package, there increases in the amount of fluid and the numbers of foam particles. Hence, the contributions of the flexible materials and compressed gas are increased.

For a given loading, within a larger volume of the FFFluid mixture, the stress is distributed on a large number of foam particles. Therefore, the strain rate of each individual foam particles is reduced in comparison with loading a smaller volume of the FFFluid mixture. Experimental work on polymer foams [66] has shown that foam is strain rate dependent and that stress increases at higher strain rates. Therefore, for a larger volume of the FFFluid the work done in compressing the flexible material is reduced.

Increasing the length or the cross section area of the package increase the volume of flexible particles. Hence, the maximum elastic stress for FFFluid mixture is increased. Therefore, the result of increasing the volume of FFFluid modifies the stiffness and damping coefficient of the system.

The theoretical study has identified and explored the parameters of FFFluid, but it is clear that these parameters are not equal in their effect on the total performance of the system. Some parameters have stronger contributions than others. To determine the contributions of these parameters, an experimental study is conducted. This study is presented in the next section.

4.4 Design of Experiments (Experimental Programmes)

Experimentation plays a critical role in the scientific study and characterisation of systems. It can unveil many important issues and provide unexpected opportunities for further detailed study. Hence, this section aims to perform an experimental investigation into the critical parameters of FFFluid isolators. Before presenting the experimental programme, introductions are given to experimental design and design tools.

4.4.1 Introduction of the Experiment Designs

In today's highly competitive environment, with markets and technologies changing rapidly, studies cannot afford to experiment by trial and error. A strategy for experimentation is required. Design of Experiments (DoE) refers to the design of any information-gathering exercise where variation is present, whether under the full control of the experimenter or not. Experimentation is a powerful technique for understanding a process, studying the impact of potential variables or factors affecting the process and thereby providing spontaneous insight for continuous quality improvement possibilities. Such methods have wide potential applications in engineering design and development stages; they enable researchers and scientists to study the effects of several variables affecting the response or output of a certain process [137-139]. Some potential benefits of using sound experimental design are:

- ✓ decreased time spent in developing, designing and manufacturing a product;
- ✓ reduced overall product cost via reduced scrap and rework time;
- ✓ knowledge of the relationship between an independent variable (i.e. process parameters) and the output (i.e. response), over a broad range of operating conditions;

- ✓ development of a mathematical model that can capture the behaviour of a compound system;
- ✓ improved performance of the system via a deeper understanding of the system processes.

4.4.2 Method of Experiment Design

Different approaches can be used for the Design of Experiments (DoE). The conventional approach involves varying one factor while maintaining the others at constant levels. This is known as One-Factor-At-Time (OFAT). Other techniques investigate all possible conditions of the experiment and are known as factorial design methods. This section presents the design techniques in more detail.

One-Factor-At-Time (OFAT) experiments vary only one factor or variable at a time, while keeping all others fixed. The varied factor is changed until its optimum setting is found, and it is then fixed at this level. Next, another factor is changed until its best setting is found, and it held constant at this setting. The process is repeated for all other factors. OFAT is used widely in many industrial applications [140, 141]. The main advantage of this method is that it usually requires less time and money for multivariable levels and factors [142]. However, testing on the basis of an OFAT approach always carries risks, and this method has several disadvantages. Firstly, OFAT can miss optimal settings of factors, and secondly, OFAT cannot estimate interactions between factors. Therefore, further investigation is often required to show the interaction of complex systems. Finally, decisions based on the OFAT approach are less precise because they are drawn after collecting the data of each trial run by comparing the observed outcome with the previous result. Typically, only two of the observations in OFAT experiments are used to estimate the effect of each factor. This form of technique can be regarded as trial and error, requiring experience and intuition for it to succeed.

Because of the limitations of OFAT experimental design, other approaches were developed. Full factorial experiment designs are one alternative to the OFAT methodology. A full factorial experiment is an experiment whose design consists of two or more factors, each with a discrete number of levels, and whose experimental units take on all possible combinations of these levels across all such factors. Such an experiment allows the investigator to study the effect of each factor on the response variable, as well as the impact of interactions between factors on the response variable. They have many advantages over OFAT including the fact that the interaction between factors can be calculated systematically. Ultimately, the true optimal behaviour of systems can be determined.

Despite the precision and accuracy of this technique, the main disadvantage of full factorial design is that it requires more resource (experiments, time, material) for the amount of information obtained. For a simple case of experimental design such as one involving two factors at two different levels, there are four possible combinations; these experiments are easy to run and inexpensive to execute. However, factors in a single investigation can reach ten factors or more. Even if each variable only has two possible levels, this means the designer needs to run more than 1000 possible variations. The number of experiments is calculated by using the expression:

$$N = L^M \tag{4-6}$$

where: N is the number of experiments, L is the number of levels in each factor, and M is the number of factors. Full experimental sets can be very expensive and time-consuming to carry out due to the potentially vast number of combinations.

To minimise the number of combinations, the fractional factorial experiment is used. The fractional factorial experiment investigates only a subset of all the possible combinations, but still obtains similar results. It consists of a carefully chosen fraction of the experimental runs of a full factorial design. The subset is selected on the basis of the most important features of the problem studied, while using a fraction of the effort of a full factorial design in terms of experimental runs and resources. Fractional factorial experiments are arguably the most superior DoE technique, characterised by the cost-effectiveness and simplicity. The Taguchi method is one type of fractional factorial experiment which used in this research. Hence, it is described in the next section.

4.4.3 Taguchi Method

The Taguchi method is a well-known technique that provides a systematic and efficient methodology for optimising experimental design. It is a fractional factorial experiment that optimises the process of engineering experimentation. Taguchi design uses an orthogonal array that can estimate the effect of each factor based on each factor's median and variation. The orthogonal array design is capable of investigating the effect of each factor independently from the others in an economical way that reduces experimental time and cost.

Taguchi's parameter design is discussed extensively by different researchers [143-145]. Their major point is that Taguchi methods do not have a statistical base. However, Taguchi methods have been used successfully for product design and process optimisation in applications worldwide [138, 139, 146-150]. The advantages of Taguchi method; it is very straightforward, easy to understand and easy to follow. Also, it does not require a strong background in statistics and mathematics [139, 148], while other methods require more work and more statistical knowledge [145]. Moreover, optimal conditions can be predicted and confirmed, and the interaction between factors also can be examined by using Taguchi method [139, 147]. For these reasons, the approach of this research tends to focus on an experimental approach using Taguchi method. The main steps of the Taguchi method are:

1-Obtain knowledge of the system: The theory knowledge of the system is obtained via previous research, textbooks, initial experiments and others.

2-Idenetfity the parameters of the system and determine the level of these parameters.

3-Design and conduct the experiments: Select the most suitable orthogonal array, assigning the factors to the appropriate column and finally describing the combination of the individual experiments (this is called the trial condition). The experiments are run according to these arrays.

4-Analyse the results: The result of the Taguchi method is a given part of the full result. Therefore, various statistical tools such as analysis of means (ANOM) and analysis of variance (ANOVA) are used to estimate the contribution of individual factors.

The ANOM tool is used to compute the average performance of the factors. The average output of a factor at any level is calculated by summing the results of all trials that include that factor, and dividing by the number of such tests. For example, to compute the average performance of factor $\overline{A}_1$ (the first factor) at level 1 for orthogonal array $L16$ experiment, the following equation is used:

$$\overline{A}_1 = (Y_1 + Y_2 + Y_3 + Y_4)/4 \tag{4-7}$$

The equation used to calculate the average performance of factor $\overline{D}_4$ (the fourth factor) at level 4 for the same orthogonal array would hence be:

$$\overline{D}_4 = (Y_4 + Y_6 + Y_9 + Y_{15})/4 \tag{4-8}$$

where: Y_N is the output of the Nth trial, $\overline{A}_1$ is the average of factor A at level L.

The second tool is Analysis of Variance (ANOVA): ANOVA is used to measure the contribution of each parameter. In this analysis, many quantities are computed. These quantities and their interrelationships are presented.

- Correction factor (C.F) is the square of the total sum of experimental output divided by the number of trials

$$C.F = \frac{T^2}{N} \tag{4-9}$$

where: T is the grand total of all results and it equal to $T = Y_1 + Y_2 + \cdots \dots \dots + Y_N$.
While N is total number of experiments

- Total Variation (S_T) is the subtraction the correction factor from the sum of squares of all trials.

$$S_T = (Y_1 + Y_2 + \cdots \dots \dots + Y_N)^2 - C.F \tag{4-10}$$

- The variance of each factor is subtraction the correction factor from the sum of the square of this factor trials. The variance of factor A is S_A which calculated as:

$$S_A = (A_1^2 + A_2^2 \dots + A_N^2)/N - C.F \tag{4-11}$$

- The degree of freedom (DoF): The number of the degree of freedom for a factor or column equals one less that the number of levels. Thus for a 4 level factor assigned to DoF=3. There is also total degree of freedom f_T and degree of freedom of error f_e, these quantities are calculated by:

$$f_A = Number\ of\ levels\ of\ A - 1 \tag{4-12}$$

$$f_T = Number\ of\ results - 1 \tag{4-13}$$

$$f_e = f_T - DoF\ for\ all\ factors \tag{4-14}$$

where: f_A is the degree of freedom of factor A.

- Variance is calculated by dividing the total variance of each factor the DoF of this factor, while variance ratio is computed by divided variance of this factor upon the total variance.

There are many software, such as: R, SPSS and Excel, used to calculate these quantities which are computed and organised in the standard tabular format as shown in Table 4-1.

Table 4-1: Standard tabular format of ANOVA

ANOVA Table - Average values					
Factors	DoF	Variance factor	Variance	Variance Ratio (F)	Percent contribution
A	f_A	S_A	S_A/f_A	$(S_A/f_A)/\sum$ Variance	Ratio of F
B	f_B	S_B	S_B/f_B		
… …					
Error	f_e				
Total	f_T	S_T	$\sum$ Variance		%100

4.5 Experimental Programme of FFFluid Isolator

The main objective of the experimental programme is to understand the relationships between process parameters and output performance. More specifically, the aim is to quantify the contribution of individual process parameters (independent variables) upon the performance (response variables). The experimental programme followed the following four steps:

1. Select parameters of FFFluid;
2. Identify the levels of FFFluid parameters;
3. Perform and execute designed experiment;
4. Statistically analyse the results.

The next sections describe these steps.

4.5.1 Selection of Parameters

The performance of FFFluid devices is affected by several parameters. A study of FFFluid parameters had been presented in Section 4-2. The most important factors, which were examined experimentally in this research, are: (i) the size of the foam particles, (ii) the ratio of foam to fluid, (iii) the viscosity of the fluid, (iv) the mixture volume, and (v) input displacement. There are also several other important parameters: (i) type of cells in the foam, (ii) shape of the foam, (iii) mechanical properties of the foam, (iv) design of the packaging, and (vi) the input frequencies. The latter parameters were not examined experimentally in this research, but instead were set constant for the following reasons.

Type of foam cell: The flexible foam capsules can be grouped into two classes: Open-cell and closed-cell. When applying a load to the FFFluid device by using open cell foam, the gases are squeezed out under loads. Then, the viscous fluid could occupy their space. Therefore, these foams do not return to their original shape when the loads are removed, which leads to change foam permanently. In the case of using closed-cell foam, the gases are sealed into cells, and they can not squeezed out of the cells during the elastic deformation. Therefore, closed-cell foam is the only type of cell that is used in the FFFluid isolation system.

The shape of the foam: One of the important issues when examining FFFluid mixtures is the consistent preparation and packaging of the mixture. It is important to fill the flexible foam in the container properly; To ensure different responses result from changing the system's parameters and it does not result from non-packaged correctly. However, consistent packaging may not be easy to achieve for the majority of particle shapes. Spherical shaped foam is the only shape that could be packaged in a consistent manner, as shown in Figure 4-6a. For other different foam shapes, different amounts of foam and fluid could be used to fill the same package volume as shown in Figure 4-6b, c.

Also, it is important to ensure uniform compression on the surface of the flexible foam material. Stresses applied to spherical beads within the fluid are always uniform and perpendicular to the bead surfaces. Conversely, stresses applied to other shapes are concentrated at corners and sharp edges. Therefore, changes in stress distribution around edge cells for irregular shapes result in changes to the stress-strain relationship [66]. Since consistent packaging and perpendicular loads cannot be guaranteed by using other shapes, the spherical foam is the only shape employed in this research.

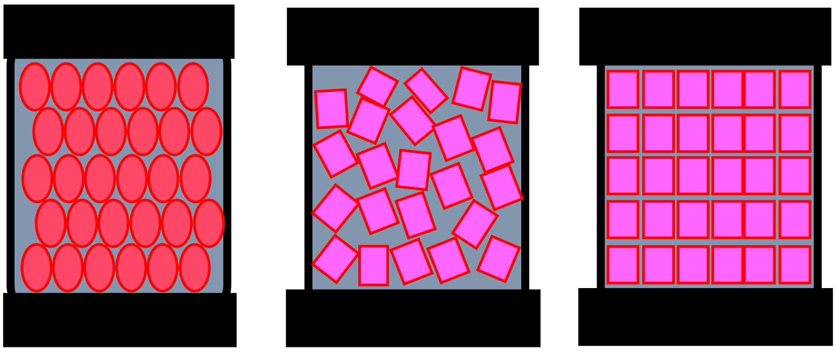


Figure 4-6: Effect of foam shape on particle packing in the FFFluid system.

Mechanical properties of the foam: Changing the Young's modulus or relative density of flexible foam or the pressure of trapped gas lead the response of the foams to be modified as described in Equations 4-1 to 4-4. Although there are various types of foam, polystyrene (PS), with a relative density of about 0.09 to 0.1, was the only foam used in this research for the following reasons:

- Flexible polystyrene foam is the only foam that is readily available in a spherical shape. The spherical shape is one of the crucial factors in the study FFFluid performance.
- The cost of Polystyrene foams with a relative density of about 0.09 to 0.1 is about £10 per 10 cubic metres. Different relative densities of polystyrene foams are not available on the market, but could be manufactured by specialist companies for higher costs. Such foams could cost more than £1000 per cubic metre [151], which is prohibitively expensive for designing vibration isolators.
- The Young's Modulus of polystyrene material is about 3-3.5 GPa [127]. It exhibits good stiffness and damping properties, and it is the most common foam structure used in anti-vibration devices [45].

The design of packaging: Different types of FFFluid packages have been used previously [18, 110]. However, an elastic package may stretch significantly, the flexible foam suffers less compression and the fluid provides less damping. To focus solely on the characterisation of FFFluid mixtures, a non-deformable package must be used. Therefore, only the piston/cylinder package was considered in this research.

The input frequency: The behaviour of polymeric foams depends on input frequencies or strain-rates. According to Gibson et al. [63] the input frequencies are divided into three main groups where the behaviour differs considerably. These groups are low, medium and high strain rate.

The range of low frequency is below 0.01Hz, which is too small to be applicable to automotive products. The range of medium frequency is from 0.01Hz up to 40Hz, which is relevant to automotive products. The high-frequency range extends from 40Hz up to 40 kHz and is used to examine blasting products.

Therefore, different values from the medium-frequency range should be examined. However, according to the same the reference, the loss coefficient of the polystyrene is constant in the frequency range of (0.01-100 Hz). More importantly, this research is focusing on modifying the coefficients of the FFFluid mixture by changing the constituent components.

4.5.2 Level of FFFluid Parameters

Having determined the critical parameters, it is important to select their appropriate levels. In general, these parameters can take a wide range of values. For example, the diameters of the foam particles can vary from 1mm to more than 200mm. To minimise these ranges, appropriate levels could be determined by a series of trial experiments. To simplify the experiments in this study, a test rig was used to simulate the vibration isolator of a medium sized machine. In this case, the suspension system of a lightweight vehicle

(L7e) is used. This application was chosen because it is possible to modify the stiffness and damping coefficient of these systems according to the laden weight of the vehicle or the road condition. This section presented the levels chosen for these parameters.

The dimensions of the package: The dimensions of the package were the first parameter whose level must be determined. Based on these dimensions, the levels of other parameters can be chosen. As mentioned previously, the piston and cylinder arrangement is used as a package for the test rig. The piston and cylinder must have a circular plan to ensure forces are distributed equally around all the foam surfaces.

The space available for the suspension system of a lightweight vehicle varies based on the type of car. However, the space available for the suspension system in a typical lightweight vehicle [30] enabled the package to be designed with a length of 300mm, a radius of 60mm, and movement of up to ±50mm. Figure 4-7 shows the assembly of the test rig.

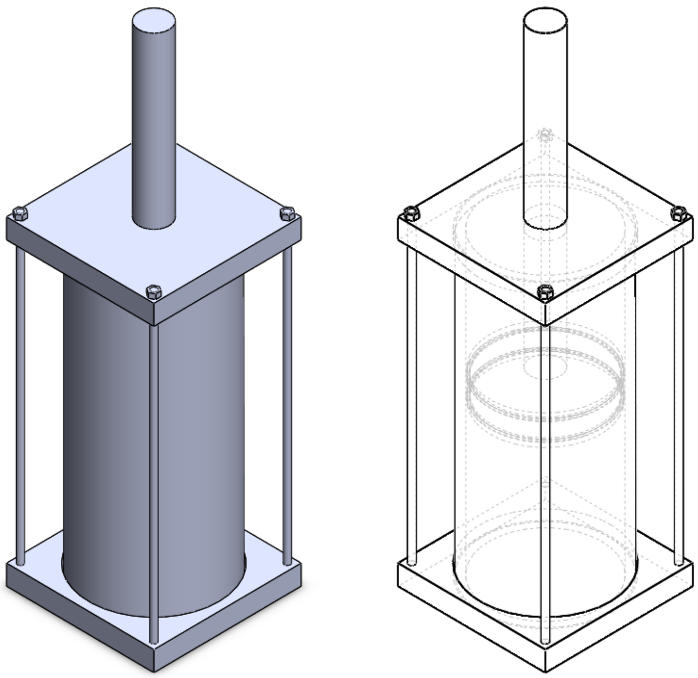


Figure 4-7: Assembly

Levels of FFFluid volume: Based on the dimension of the package, the length of the cylinder was 300mm, while the thickness of the piston was 30mm. The device was designed to vibrate with a stroke of ±50mm; therefore, the maximum height of mixture in the cylinder was 220mm, giving a volume about 2.2Liter. The minimum volume could be as small volume as possible. However, the test rig was examined under a static displacement equal to 45mm. So, the minimum volume of the test-rig was chosen to be at a depth of 90mm. Because less than this volume, the mixture will be compressed more than 50% of its volume during the static load, and the flexible foam may be deformed plastically in such case. The third and fourth volumes of the mixture were chosen to produce four equidistant volumes from the minimum to the maximum. These intermediate volumes were 1.3Liter (130mm height), and 1.8Liter (18mm height).

The level of foam to fluid ratio: Different ratios of foam to fluid could be used when preparing an FFFluid sample. The maximum ratio of foam to fluid is 100%. The minimum ratio is a zero, this means the cylinder is filled with fluid only, and there are no flexible foam particles in the test rig. It is difficult to compressing such a mixture, because the fluid is usually incompressible. Therefore, reducing a considerable amount of flexible foam can affect the performance of the system.

Experimentally, it was found that filling 85% of the package with flexible foam (7mm in diameter) is required around 30% fluid, and this amount of fluid increases by using a bigger size of flexible foam. Therefore, to avoid reducing a considerable amount of the foam, the ratio percentage (85%) was the smallest percentage used in the current study. Other levels of the foam to fluid ratio were chosen based on spacing them equally between the maximum and minimum ratios. Therefore, the four ratios used were 85, 90, 95, and 100%.

Levels of flexible foams sizes: Diameters of foams can vary from 1mm to more than 200mm. The minimum diameter of foam that was commercially available in bulk was 1.0 to 2.0mm; this is therefore the smallest size examined here. The maximum diameter of particles used in the research was 18mm. This diameter was chosen based on a preliminary experimental investigation. It was found experimentally that filling the test rig by foams with a diameter of 18mm required 600ml of the fluid. This amount of fluid was more than 25% of the total container's volume. This percentage is increased when decreasing the ratio of foam to fluid. Therefore, to avoid reducing a considerable amount of the foam, The maximum diameter of particles used in the research was 18mm.

An intermediate-sized foam with a diameter of 6-8mm was also used. In addition, a sample known as a nested sample was tested, which is a mixture of the three sizes of foam. This type of sample was chosen because Davies et al. [15, 18] examined such sample in FFFluid shock absorbers, and outstanding performance was achieved. Therefore, four levels of flexible foam size were chosen (1-2mm, 6-8mm, 18mm and the nested sample).

The level of fluid viscosity: Increasing the viscosity of the fluid leads to an increase in the damping coefficient of the system. The relationship between fluid viscosities is nonlinear, there is a greater improvement in the performance when moving from viscosity 1800cSt to viscosity 10400cSt copmaring with improvement when moving from viscosity 10400cSt to higher one at 20100cSt [14]. Therefore, three viscosities of silicon oil were chosen: 100, 1000, and 12500cSt. The same type of the fluid was selected to ensure the viscosity is the only factors that is changed.

The level of the input displacement: The test rig is designed to simulate suspension systems in lightweight vehicles, and the maximum displacement of suspension in a commercial car is approximately 60mm (±30mm) [152]. Therefore, 30mm displacement is the maximum input to the system. Two levels were also chosen in equal spaced 10 and 20mm. While the last level was chosen to be equal to 5mm, to examine the performance of the system at low displacements.

Table 4-2 shows the selected parameters and their levels.

Table 4-2: Parameters and their levels.

Parameter	**Input**			
Diameter of PS particles (mm)	1-2	6-8	18	nested
Foam height/Package length	100%	95%	90%	85%
Viscosity of fluid (cSt)	100	1000	12500	
Volume of FFFluid (litre)	0.9	1.3	1.8	2.2
Displacement (mm)	5	10	20	30

4.5.3 Experiments

The sample composition and preparation procedure have been described in the previous chapter (section 3.6.2). The test facilities are within the Cardiff University Structural Performance (CUSP) Laboratory. Experiments were performed on a Losenhausen Test System. The integrated data acquisition system comprises an instrumented type of load cell that digitally records the load–displacements. The experimental tests were conducted using the Taguchi method. In the experiment, the number of factors is 5, four of the factors have four levels, while one factor (viscosity) has three levels. Therefore the orthogonal array is L16, as shown in Table 4-3. The experiments were repeated for three different frequencies levels: 0.75, 1.00 and 1.25Hz.

Table 4-3: Orthogonal Array -L16

Experiment	Foam Diameter	Foam/Package length	Fluid Viscosity	FFFluid Volume	Input Displacement
1	1	1	1	1	1
2	1	2	2	2	2
3	1	3	3	3	3
4	1	4	3	4	4
5	2	1	2	3	4
6	2	2	1	4	3
7	2	3	3	1	2
8	2	4	3	2	1
9	3	1	3	4	2
10	3	2	3	3	1
11	3	3	1	2	4
12	3	4	2	1	3
13	4	1	3	2	3
14	4	2	3	1	4
15	4	3	2	4	1
16	4	4	1	3	2

4.5.4 Results and Discussion (Performance)

Two experimental procedures were employed: static load and dynamic load. The experimental procedures were described below.

The first part of the experiments was the static load. For this test, a quasi-static displacement (45 mm displacement) was applied to the test rig in total time equal to 45 seconds. The input displacement and output force were recorded in this test; therefore, the displacement-force graph could be obtained from this test. Figure 4-8 shows displacement versus force graph of FFFluid under static load for trial 1. The stiffness coefficient of the test rig could be calculated by using the Trend line function in Microsoft Excel. The stiffness coefficient for this test is equal to $k = 14.1kN/m$

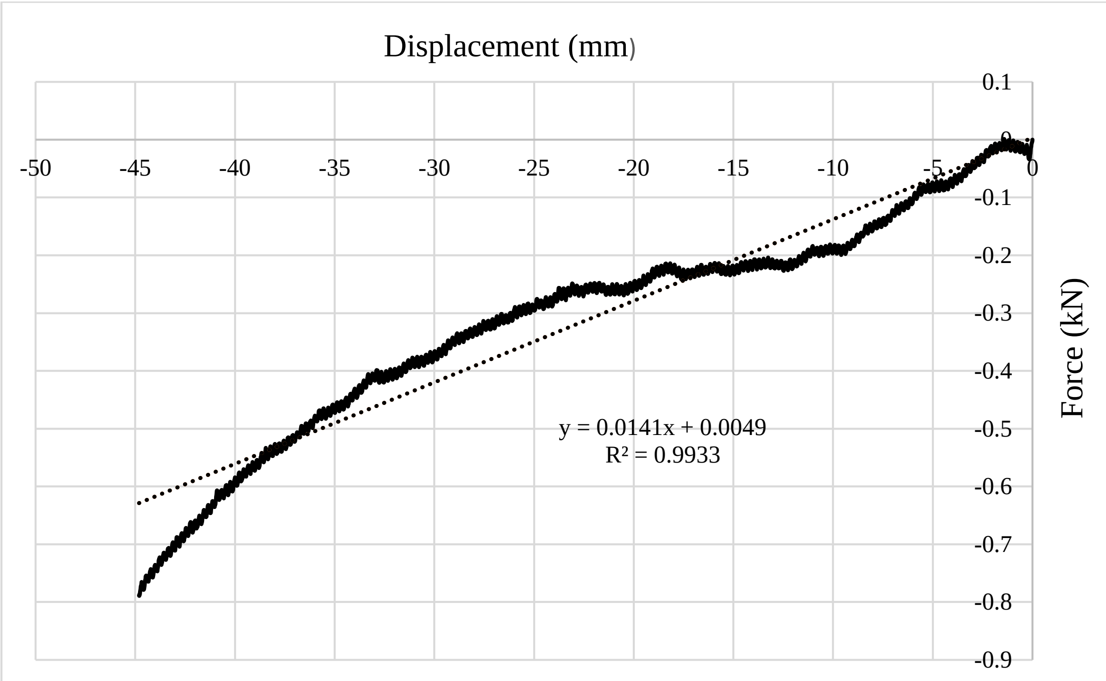


Figure 4-8: Displacement- Force graph of FFFluid under static load for trial 1.

The second part of the experiment was a dynamic test. A dynamic displacement was applied to the FFFluid rig at three different frequencies 0.75, 1, 1.25. This series of tests aimed to calculate the damping coefficient of the FFFluid rig. From this test, the input displacement against damping force could also be plotted as shown in Figure 4-9. The damping force was determined by subtracting the friction force and stiffness force from the dynamic force. From Figure 4-9, the differences between the upper and lower loops at the middle of the loop was $2c\dot{x} = 0.048kN$. By analysing the harmonic motion such as it was explained in section 3.6, the velocity at the middle position was $\dot{x} = 0.0314\text{m/s}$ system. Thus, the damping coefficient of the system was $c = 0.764\text{kN. s/m}$.

The rest results of these experiments were displayed in Appendix C. The stiffness and damping coefficients are displayed in Tables 4-4 and 4-5 respectively.

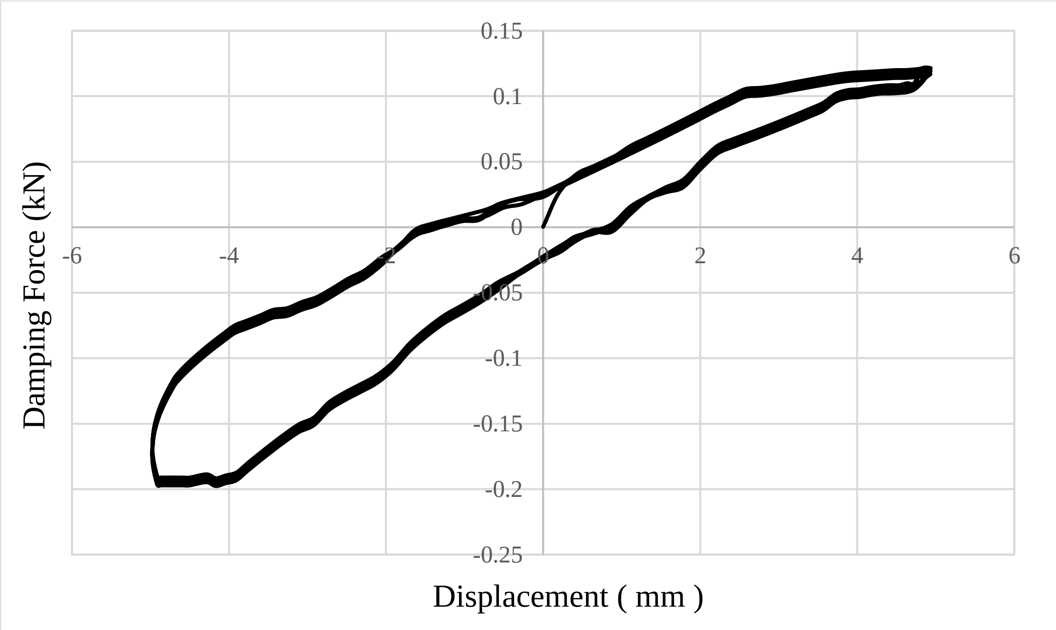


Figure 4-9: Displacement- damping force graph of FFFluid under dynamic load and frequency 1.0Hz for trial 1.

Table 4-4: Stiffness coefficient of the system

Trial	Stiffness coefficient kN/m
1	14.1
2	16.8
3	24.7
4	28
5	22.4
6	24.6
7	18.2
8	22.8
9	31.2
10	27.4
11	23.6
12	24
13	18.7
14	21.0
15	26.8
16	26.0

Table 4-5: Damping coefficient of the system at different frequencies

Trial	Damping Coefficients $kN.s/m$		
	0.75Hz	1Hz	1.25Hz
1	0.743	0.764	0.764
2	1.062	1.035	1.083
3	1.433	1.433	1.433
4	1.274	1.274	1.274
5	1.345	1.327	1.338
6	1.592	1.513	1.529
7	1.699	1.672	1.656
8	1.805	1.752	1.783
9	1.008	1.035	1.051
10	1.274	1.274	1.274
11	0.92	0.929	0.934
12	0.743	0.717	0.732
13	1.221	1.234	1.242
14	0.92	0.929	0.934
15	1.486	1.433	1.401
16	1.168	1.154	1.146

4.6 Discussion of the Influence of Factors

The results were obtained by testing all the 16 trials for the three different frequencies (0.75, 1.00, and 1.25Hz). Each prototype represented an individual experiment in the orthogonal array as shown in Tables 4-4 and 4-5. To compute the average parameters of each factor, Equations 4-7 and 4-8 were used. The average stiffness coefficient and damping coefficient at the different frequencies are summarised in Table 4-6 and Table 4-7.

Table 4-6: Average of stiffness coefficient

Parameters	Level	Stiffness coefficient kN/m
Foam Diameter	A1	20.9
	A2	22.0
	A3	26.5
	A4	23.125
Foam/Package length	B1	21.6
	B2	22.45
	B3	23.325
	B4	25.2
Fluid Viscosity	C1	22.15
	C2	22.375
	C3	24
FFFluid Volume	D1	19.325
	D2	20.6
	D3	25.125
	D4	27.65
Input displacement	E1	22.775
	E2	23.05
	E3	23
	E4	23.75

Table 4-7: Average Damping coefficients at various frequencies

Parameter	Level	Damping coefficients at various frequencies $kN.s/m$		
		0.75Hz	1Hz	1.25Hz
Foam Diameter	A1	1.128	1.127	1.139
	A2	1.61	1.566	1.576
	A3	0.986	0.989	0.998
	A4	1.199	1.188	1.181
Foam/Package length	B1	1.08	1.09	1.099
	B2	1.212	1.188	1.205
	B3	1.385	1.367	1.356
	B4	1.247	1.224	1.234
Fluid Viscosity	C1	1.106	1.09	1.093
	C2	1.159	1.128	1.139
	C3	1.292	1.287	1.30
FFFluid Volume	D1	1.026	1.02	1.022
	D2	1.252	1.237	1.261
	D3	1.305	1.297	1.298
	D4	1.34	1.314	1.314
Input displacement	E1	1.327	1.306	1.306
	E2	1.234	1.224	1.234
	E3	1.247	1.224	1.234
	E4	1.115	1.115	1.120

4.6.1 Influence of Bead Size

The effects of changes in the flexible foam size upon the efficiency of isolator were investigated. Figure 4-10 shows the stiffness coefficients for each of the tests, while Figure 4-11 illustrates the damping coefficients for each of the tests in comparison with the changes in frequency. These figures show that an increase in the foam size leads to an increase in the stiffness coefficients of the FFFluid system, while the highest damping coefficient could be achieved by using foam of size 6-8mm. Furthermore, the damping coefficient of the system is shown to be constant across the range of applied frequencies.

When using large sizes of foam beads in the FFFluid container, the amount of incompressible fluid in the container increases in comparison with the quantity of fluid required by using the small sizes of the foam. Therefore, the stiffness coefficient of the FFFluid system is raised. The opposite occurs when using smaller sizes of foam; the total amount of incompressible fluid reduces, causing a reduction in the stiffness coefficient of the system. For example, filling the container with foam of size 18mm requires 410ml of fluid and the stiffness coefficient about 26.5kN/m, while using foam size 1.5mm requires 300ml of fluid, giving a stiffness coefficient about 21kN/m.

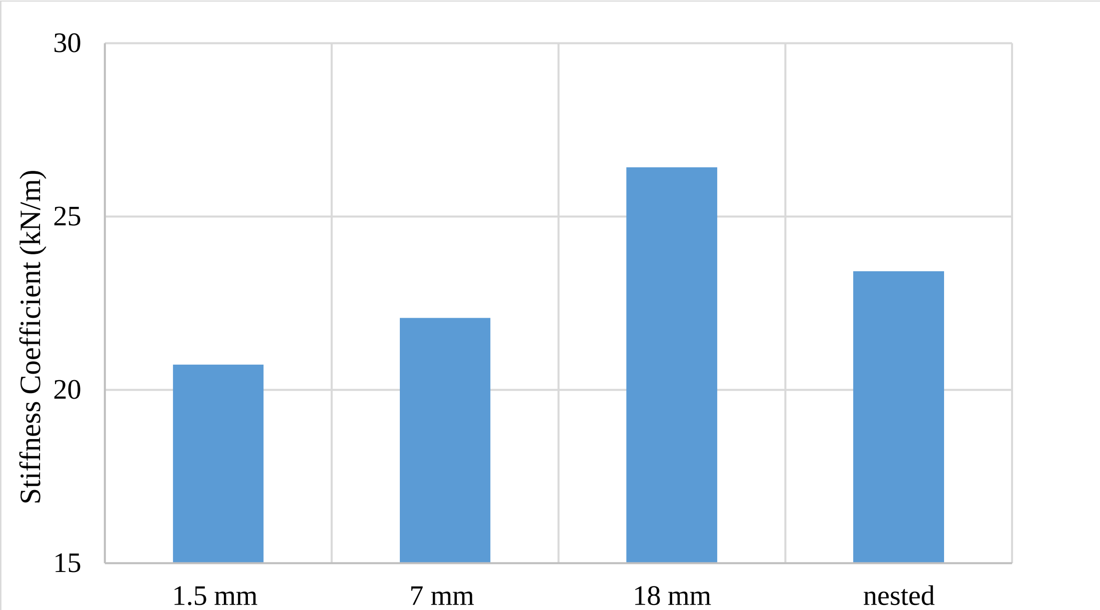


Figure 4-10: Stiffness coefficient of the FFFluid system for the different size of foam.

When examining the damping coefficient (Figure 4-11), it can be seen that the intermediate level (i.e. 6-8mm) provides the highest value of damping coefficient. Before interpreting such results, it is necessary to consider the damping mechanism of the system. The damping mechanism by an FFFluid composite material occurs in two ways. One is the work done in compressing the flexible material, and the other is the work done by the matrix fluid as it shears.

By using the biggest foam size (18mm) in a container that has a radius of 56.5mm, the amount of flexible material is reduced. This leads to a decrease in the damping value of the flexible material. In the case of using foam that has average diameter 1.5mm in the same container, there is less space available for fluid. Therefore, it is not as easy to distribute the pressure around all the foam particles, and the amount of damping due to fluid shear is hence reduced. This leads to a reduction in the damping coefficient of FFFluid devices. The damping coefficient of the system is increased by using the nest sample, but the best sample in terms of damping coefficient is the one containing (6-8) mm foam particles. The foam size (6-8) mm provides the highest damping coefficient of the system, but this size may not provide the biggest size for a different sized container.

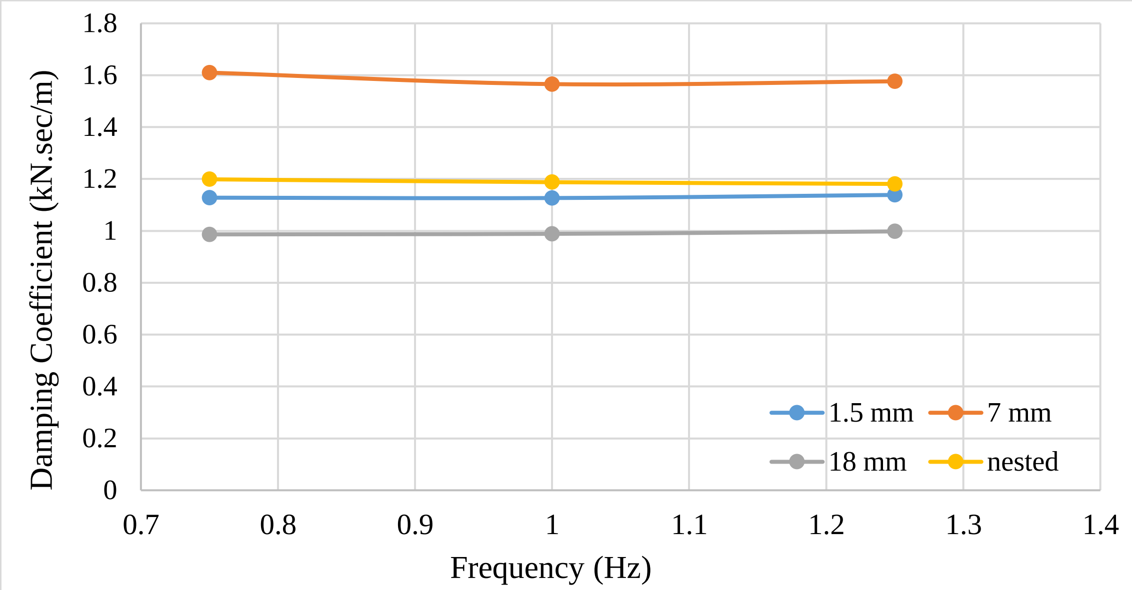


Figure 4-11: Damping coefficient against frequency for different size of foam in FFFluid pad sample.

4.6.2 Influence of Foam to Fluid ratio

Figures 4-12 and 4-13 show the effect on stiffness and damping coefficients of FFFluid of using different values of the foam to fluid ratio. These results indicate that decreasing the ratio of foam to fluid leads to an increase in the stiffness coefficients of the FFFluid system, while filling 90% of the package with foam capsules provides the highest value of damping coefficient.

To interpret this result, the reduction in the amount of foam can be thought to cause an increase in the volume of incompressible fluid, which leads to an increase in the stiffness coefficient of the system. Therefore, decreasing the ratio of foam to fluid leads to a rise in the stiffness of the FFFluid system.

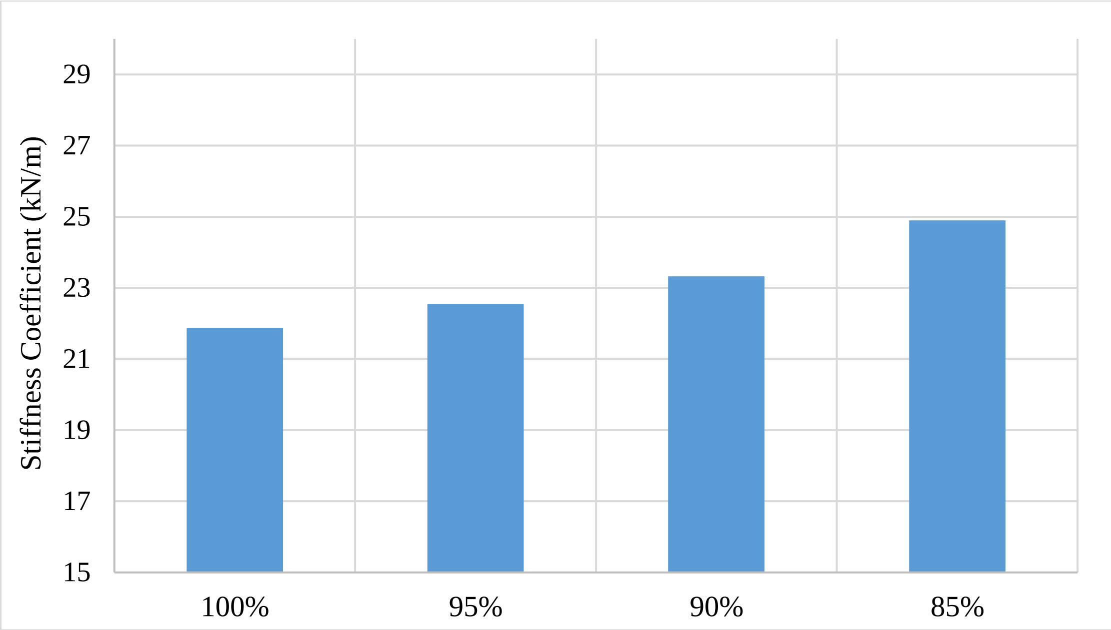


Figure 4-12: Stiffness coefficient for different ratio of foam in FFFluid pad sample.

For damping coefficients, filling 100% of the package with elastomeric foams leads to a smaller amount of fluid, especially for small size foam, which may result in more contact between the foam particles and less ability to distribute the load equally around all of the foam. Therefore, the amount of damping required to reduce unwanted vibration due to fluid shear will be reduced. While filling 85% of the package volume with foam reduces the amount of elastomer material, the damping value of the system is also reduced. Filling the container with the ratio of foam to the total length of a package of 90% provides the best value of damping coefficient for the system.

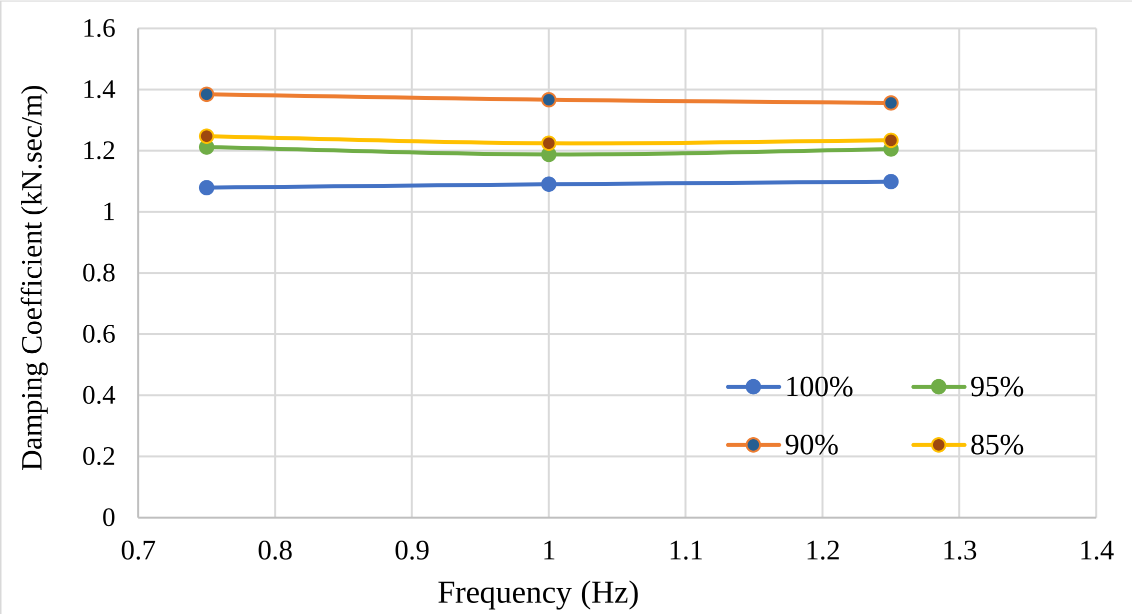


Figure 4-13: Damping coefficient against frequency for a different ratio of foam in FFFluid pad sample.

4.6.3 Influence of Matrix Fluid Viscosity

Figures 4-14 and 4-15 illustrate the stiffness and damping coefficients of the system for three viscosities of fluid in the FFFluid pad sample. As shown in these figures, fluid viscosity has limited effect on the stiffness coefficient of the system, while increasing the fluid viscosity of the system leads to a higher damping coefficient of the system.

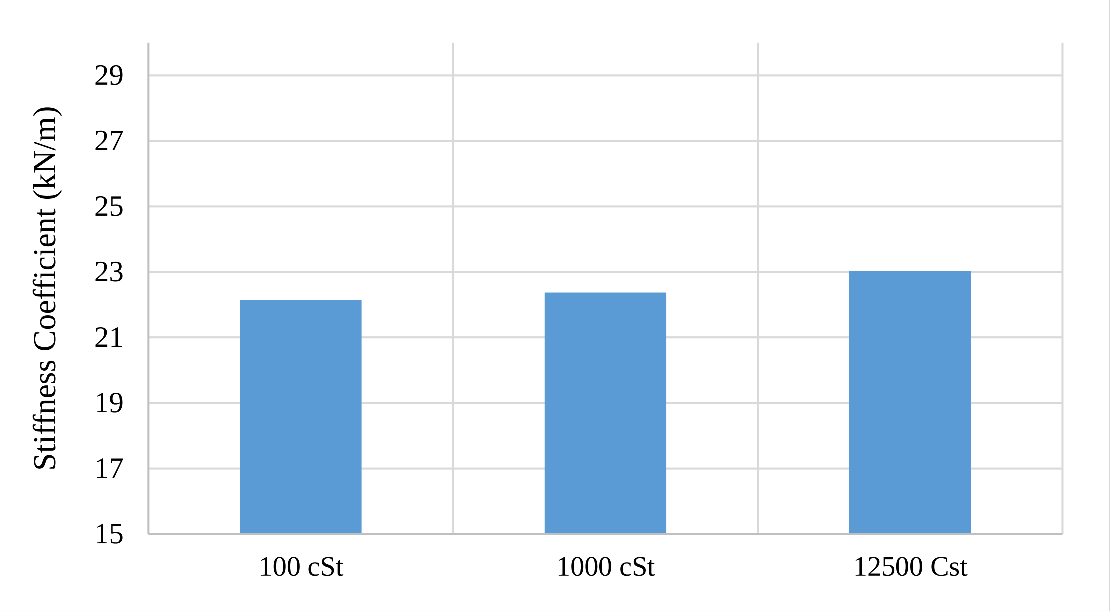


Figure 4-14: Stiffness coefficient for the different viscosity of the fluid in FFFluid pad sample.

Although low viscosity fluid has an advantage in distributing pressure around the foam, the main purpose of the fluid is to provide isolation of vibrations. If the viscosity of the matrix fluid is increased, the shear stress is far greater. This result demonstrates that higher viscosity fluids work better for improving the damping contributions of FFFluid vibration isolators.

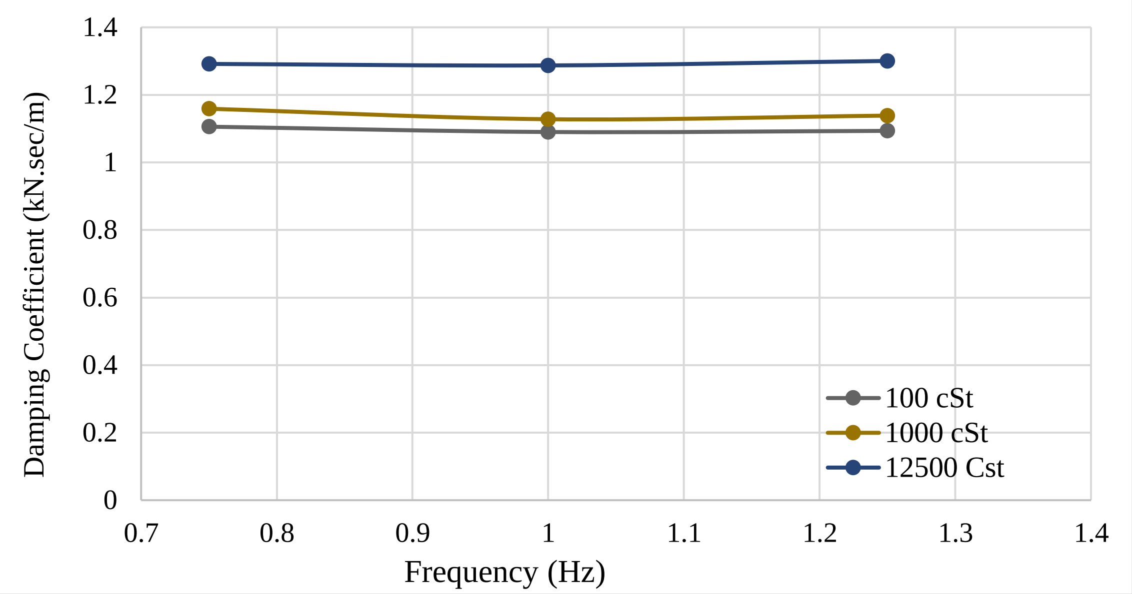


Figure 4-15: Damping coefficient against frequency for the different viscosity of fluid in FFFluid pad sample.

4.6.4 Influence of the volume of the device

This series of tests investigated the effect of changes in FFFluid volume upon the parameters of the system by increasing the depth of FFFluid mixture in the cylinder. The stiffness and damping coefficients for each of the tests are compared with the different values of FFFluid volume, as shown in Figures 4-16 and 4-17. From these figures, it can be observed that the stiffness and damping coefficients are increased by increasing the volume of the FFFluid mixture.

To explain these observations, an investigation of the isolation process is required. The mechanism of damping in this system works by compressing the flexible foam material and shearing the matrix fluid, while the stiffness of the system is raised by increasing the amount of incompressible fluid. By looking at these contributions, it can be seen that the strain-rate of the individual foam beads is reduced by increasing the volume of FFFluid [153]. Therefore, for a larger volume of FFFluid the work done in compressing the elastomeric material is reduced. This lead to increase the stiffness coefficient of the system. Also, the volume of elastomer material increases for a large volume of FFFluid which leading to increase the damping coefficient of the system.

To conclude, an increase in the volume of the FFFluid mixture leads to an increase of foam and fluid. Therefore, both stiffness and damping coefficient of the FFFluid device are increased.

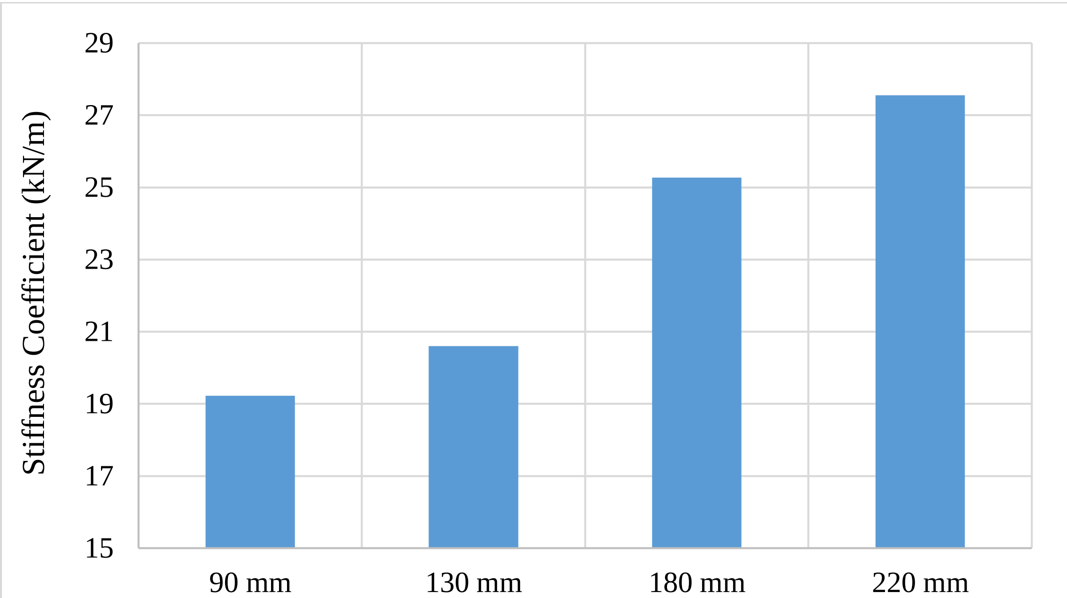


Figure 4-16: Stiffness coefficient for different volume of foam in FFFluid pad sample.

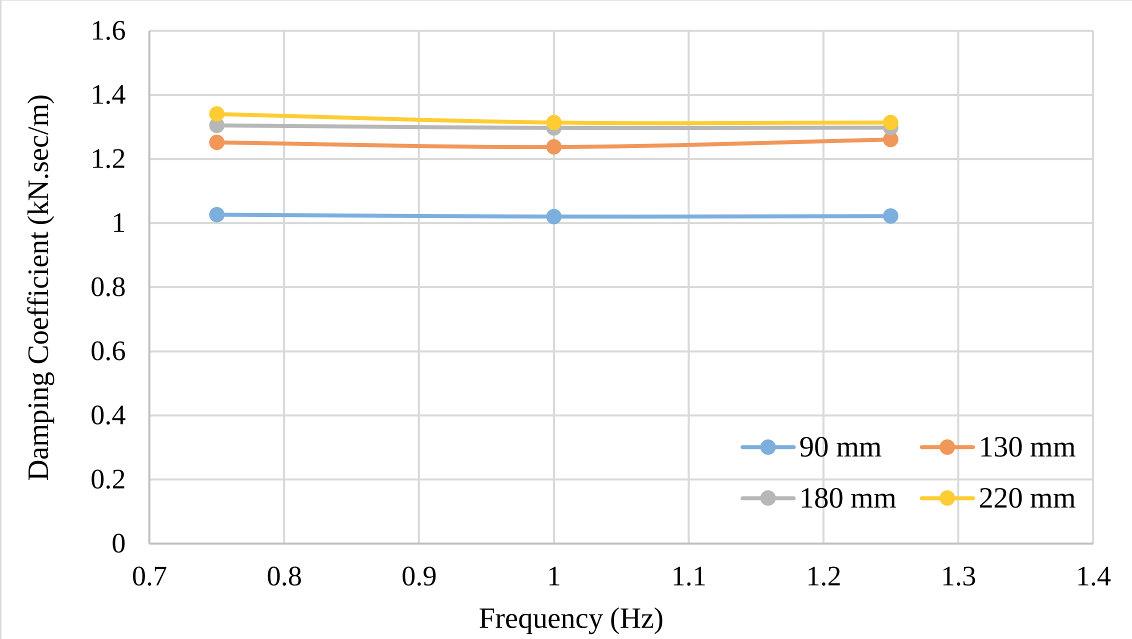


Figure 4-17: Damping coefficient against frequency for different volume of FFFluid pad sample.

4.6.5 Influence of applied displacements

This series of tests investigated the effect of changes in input displacement on the coefficients of the system. Although the displacement is the input for the devices, and it is not controlled by the design of the FFFluid. However, this parameter is investigated here in order to observe its effect on the total performance, and also to explore the best range of displacement for the optimal performance of this device.

Figure 4-18 shows the stiffness coefficient of the system by applying different input displacements, while the damping coefficient for each of the test results, compared against the change in frequency for different displacements, is shown 4-19. From these figures, the maximum displacement is seen to be 30mm, which is in the linear region. It can also be observed that there is no big change in the stiffness and damping coefficients, as all inputs were in the linear region.

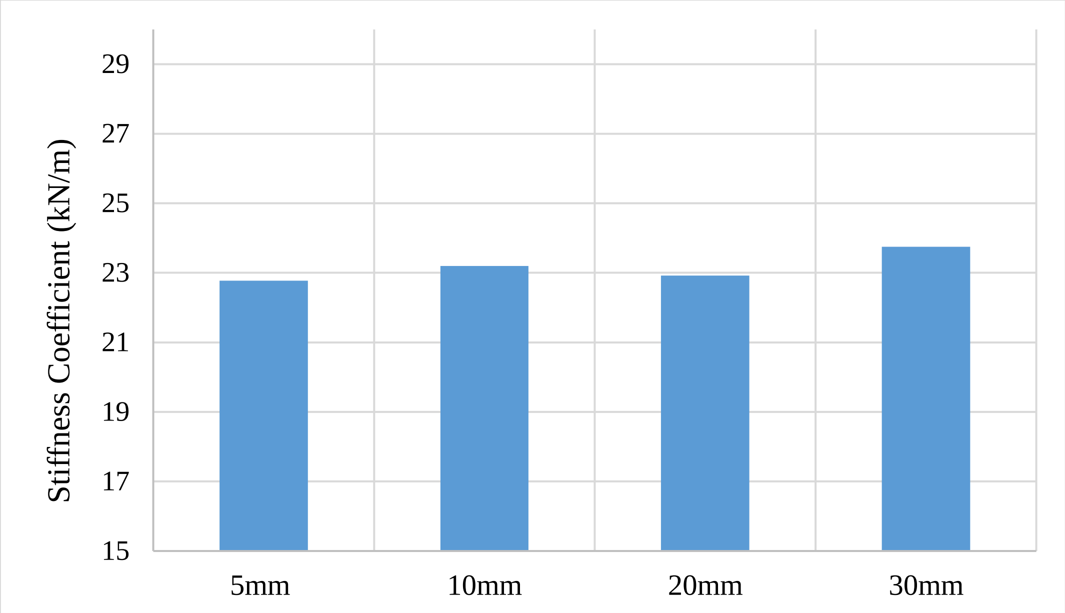


Figure 4-18: Stiffness coefficient of the system for different input displacement

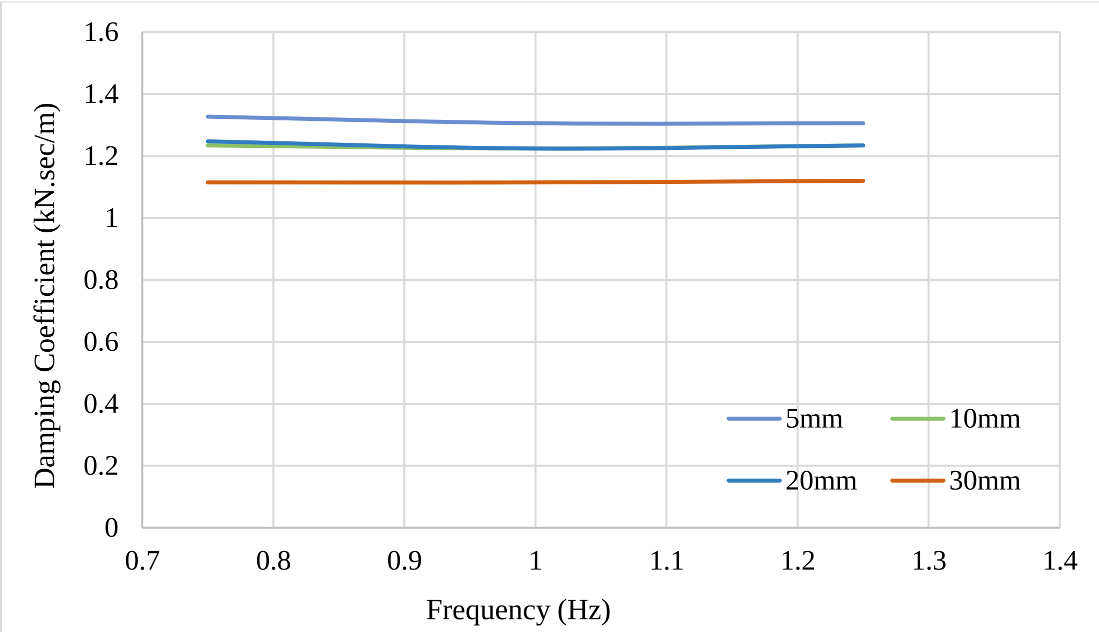


Figure 4-19: Damping coefficient against frequency for different input displacement to the system

4.7 The contribution of each factor

Analysis of variance (ANOVA) is used to measure the contribution of each parameter. Such analysis was carried out for both the stiffness and damping coefficients of the system to measure the contributions for each of the parameters to each coefficient. ANOVA analysis was explained in Section 4-4-3. As mentioned in that section, many quantities are needed to compute ANOVA analysis. For the stiffness coefficient parameters, the following describes these values:

The grand total of all results, T = 370.3;

Total number of experiments, N = 16;

Correlation factor, C.F = 8570.13;

Total Variation, S_t = 305.099.

The result of the ANOVA table is shown in Table 4-8. From this table, it can be seen that the most important parameter affecting the stiffness coefficient of the system is the volume of the mixture; it contributes about 60%. The size of foam and the ratio of foam/fluid also have some effect on the stiffness coefficient of the system, but the influence of input displacement is insignificant with regard to the stiffness coefficients of the system.

Table 4-8: Analysis of variance (ANOVA) for Stiffness coefficient

ANOVA Table - Stiffness coefficient					
Factors	DOF	Sum of Squares	Variance	Variance Ratio (F)	Percent contribution %
Foam size	3	71.781	23.927	2.3E-01	23.53
Ratio foam/fluid	3	28.502	9.50	9.3E-02	9.34
Viscosity	3	18.937	6.31	6.2E-02	6.21
Mixture Volume	3	183.747	61.25	6.0E-01	60.23
Displacement	3	2.132	0.71	6.9E-03	0.70

For damping coefficient, the calculation of these values:

The grand total of all results $T = 19.47$.

Total number of experiments $N = 16$.

Correlation factor $C.F$ =23.704.

Total Variation S_t =1.384.

Analysis of variance (ANOVA) for the damping coefficient is shown in Table 4-9. The most considerable contribution is the size of the foam, at 50%, which is the highest contribution to overall performance. The viscosity, volume of the mixture and ratio of foam/fluid also make a considerable contribution at 15%, 15% and 11% respectively. Again the effect of input displacement is insignificant to the damping coefficient of the system.

Table 4-9: Analysis of variance (ANOVA) Damping coefficient

ANOVA Table - Damping coefficient					
Factors	DOF	Sum of Squares	Variance	Variance Ratio (F)	Percent contribution %
Foam size	3	0.732	0.244	5.2E-01	52.85
Ratio foam/fluid	3	0.158	0.05	1.1E-01	11.40
Viscosity	3	0.202	0.07	1.4E-01	14.57
Mixture Volume	3	0.22	0.07	1.5E-01	15.84
Displacement	3	0.074	0.02	5.3E-02	5.33

4.8 Summary of Chapter 4

Previous investigations into FFFluid test rig have concluded that Foam Filled Fluid exhibits the appropriate characteristics for use in vibration isolators. FFFluid is a composite material that has the potential to provide a technically non-restrictive countermeasure while enhancing safety. However, there are considerable challenges that still need to be overcome. Characterisation of the innovative material proposed presents real difficulties because of the large variations that are possible in the physical and mechanical properties of the constituent components. The characterisation of the mechanical properties of FFFluid vibration isolators was therefore reported in this chapter.

The characterisation of such composite material was achieved via a systemic study which is a unique in this area. The systematic study of the FFFluid began by identifying of the system parameters, then formulating of the hypothesis and linked these hypotheses in system behaviour. The theoretical study was followed by the experimental programme to validate such study.

An experimental programme was carried out, focusing on selected key factors at a range of levels. It was shown that increasing volume of FFFluid provided better stiffness and damping coefficients for the system, while filling 90% of a container with foam of diameters 6-8mm gave better damping compared to other examined ratios. Furthermore, reducing the volume of the flexible particles by reducing the ratio of foam to fluid, or by using larger elements of flexible particles leads to an increase in the stiffness coefficient of the system. The contribution of each of the critical parameters was also determined, and it was observed that the input displacement up to 30mm has limited effect on the stiffness or damping coefficients of the system. The size of flexible beads, the ratio of foam to fluid, viscosity and volume of composite material had the significant contributions in stiffness and damping coefficients of the system. The condensed version of this chapter was written as a published paper, which is included in Appendix J.

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CHAPTER 5

DEVELOPING A MATHEMATICAL MODEL OF FFFLUID

5.1 Introduction

The advantages of Foam Filled Fluid (FFFluid) have been studied in Chapter 3, while Chapter 4 has dealt with the characterisation of the FFFluid isolator parameters. Packaging such composite material correctly can produce a promising suspension system for lightweight vehicles. However, the accurate packaging of FFFluid technology is achieved through a series of trials so that the proper performance of the system can be determined. This requires a substantial investment of time and money. Developing a mathematical model for the FFFluid device concept would eliminate much of the work involved in specifying parameters for a given task. It would also mean that the performance of a FFFluid system could be predicted at the prototype stage of manufacturing. Therefore, the purpose of the research reported in this chapter is to establish a model of Foam Filled Fluid mixtures based on experimental results. This model includes both stiffness and damping coefficients of the isolator. Also, the validation of the models is conducted.

This chapter is organised as follows. First, the methodology of the chapter is developed. Second, an introduction of a regression model is presented, this includes: how can the regression model be determined and validated. Third, the experimental programme is developed, which consists of (i) developing the design of experiments, (ii) executing the experiments, and (iii) analysing the results. The final part of the chapter develops a regression model of the system based on the experimental results; this part includes (i) obtaining equations for stiffness and damping coefficient of the system, (ii) checking the accuracy and optimising, and (iii) validating of the equations experimentally.

5.2 The Methodology of the Chapter

A mathematical model is a description of a system using mathematical concepts and language. Physical theories are almost invariably expressed using mathematical models, and researchers continually attempt to improve the accuracy of mathematical models. According to the literature, mathematical models can take many forms, and they can be classified as either deterministic or probabilistic (statistical) [154].

A deterministic model is one in which every set of variable states is uniquely determined by parameters in the model and/or by sets of previous states of these variables. In deterministic models, the output of the model is fully determined by the parameter values and the initial conditions. Such model usually applies where the material properties and applied load are well known. Conversely, a statistical model is a probability distribution constructed to enable inferences to be drawn or decisions made from the data. A statistical model is used when the properties of the system and system's input have random or unknown properties. Based on statistical design and analysis, a statistical model can be established that changes based on independent variables related to dependent responses. However, statistical models allow for the possibility of error in describing a relationship, as randomness may be present in a statistical model. Therefore, validation is usually required. For example, a statistical model to estimate the time dependence of metal loss in underground pipelines was created for in situ soil parameters [155]. The coefficient of determination was used to check its accuracy. The coefficient of determination (R^2) score 0.96, which meaning the model data almost fit the real data. Moreover, a variety of statistical models has been developed that relate to the prediction, understanding or evaluation of energy consumption in buildings [156, 157]. Different statistical tools were used to check the accuracy for such models.

Deterministic models of FFFluid based on a theoretical investigation are complicated to achieved, this is due to the complex behaviour resulting from interaction between its components, the different contributions of damping mechanisms, and non-quantified properties of the system. To overcome such difficulties, the developed methodology of this research that is presented in this chapter is a statistical analysis of the experimental data for the FFFluid rig. This is a unique method for developing models in this area; this approach starts by executing the design of experiments in order to obtain the experimental data. The design of experiments in this research is carried out via Taguchi method. Then, a regression model of the system based on these experimental results is obtained. The accuracy of such model then is checked by different statistical tools. This is followed by optimization the model if it is required (the accuracy is not accepted). Finally, the regression model is validated experimentally. Figure 5–1 shows a flowchart that summarises the development of such model.

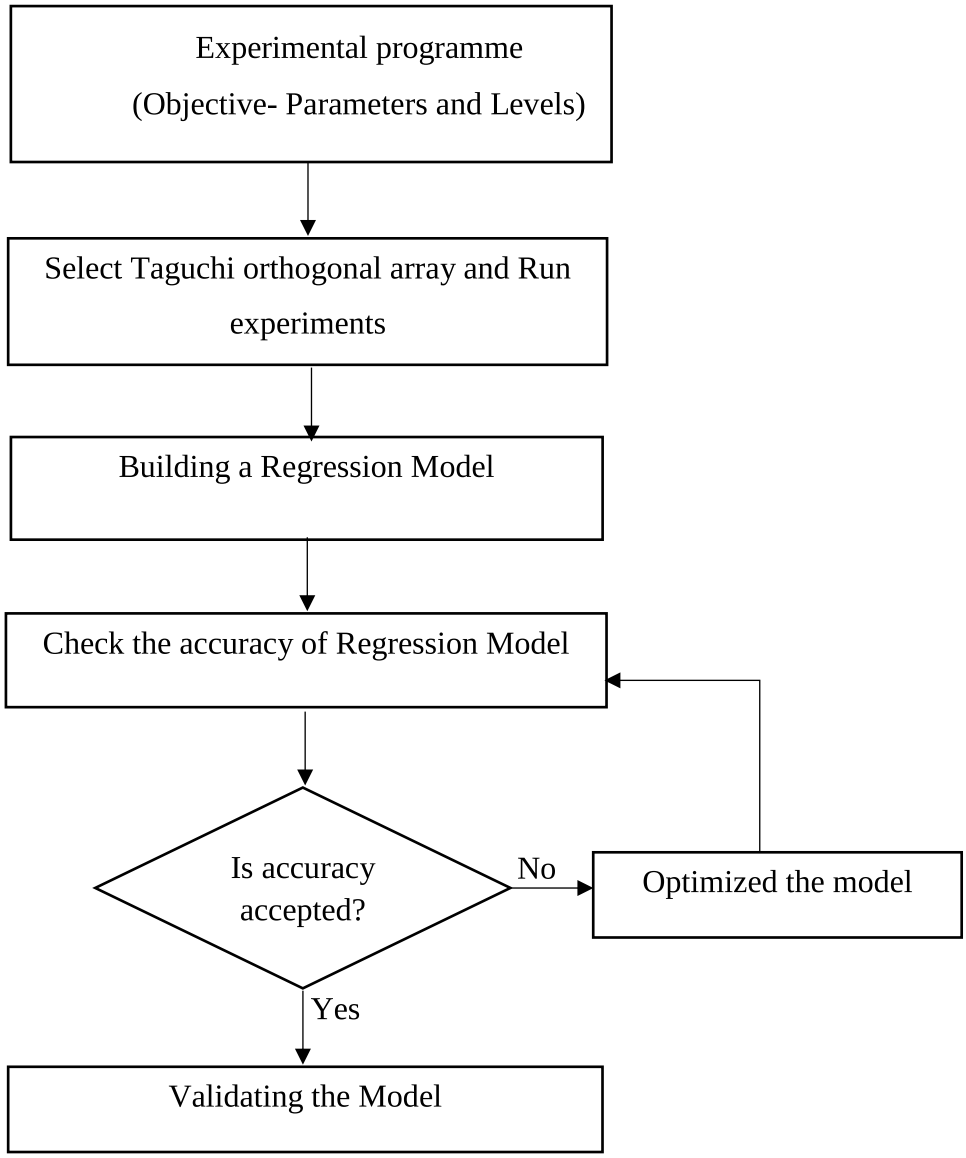

Figure 5-1: Flow chart for development of a regression model.

5.3 Regression Model

Several methods are used for the analysis of multivariable data, such as multi discriminate analysis, cluster analysis, principal component analysis and regression analysis [158-161]. If one of variable (or more) is nonparametric for example (high- medium- low), the multi discriminate analysis is used, while Cluster analysis is the technique of grouping rows together that share similar values across a number of variables, it aims to develop a meaningful of subgroups. The purpose of the principal component analysis is to derive a small number of independent linear combinations (principal components) of a set of measured variables that capture as much of the variability in the original variables as possible [158, 159]. Of these methods, a regression model is the appropriate method of analysis single metric variable presumed to be linked to two or more independent variables. It is the simplest and most widely used modelling method [154, 162, 163]. Regression represents an efficient use of data, and it can produce good estimates of the unknown parameters in a model from relatively small data sets. It is a well-developed theory for computing confidence, prediction and calibration intervals to answer scientific and engineering questions.

The major cost of the model regression technique is the number of iterations needed to optimise the procedures necessary for computing parameters. However, various types of statistical computer software have been developed to address this issue, and handling the process effortlessly such as Microsoft Excel and SPSS software. Therefore, the regression method was e chosen for this research.

Regression analysis is a statistical tool for investigating the relationships between variables. It is valuable for quantifying the impact of various simultaneous influences upon a single dependent variable. The investigator usually seeks to ascertain the causal effect of variables upon the response, for example, the influence of an FFFluid volume increase upon the stiffness coefficient of the system, or the impact of changes in the foam to fluid ratio upon the damping of the system.

Regression techniques have become powerful tools for design of experimental techniques in engineering. Numerous researchers have used regression analysis to develop a mathematical model of a system. One example is given in [164], where regression models are developed to predict the surface roughness under dry, flooded and lubricated conditions for three different tools. These models show good agreement with experimental results [164]. A set of empirical relations for a welding application is developed in [165] to estimate weld bead dimensions and mechanical properties as a function of welding conditions. Mathematical models of reinforcement and penetration profiles in high deposition welds produced by multiple-wire processes have also been achieved [166].

Regression analysis is used to quantify the effects of independent variables (inputs) upon dependent variables (responses). It is employed in a wide range of scientific disciplines including medicine, biology, economics, engineering and social sciences [167-169]. However, as it is a statistical model, validation procedures are required; hence, the following sections answer two questions:

- How can the regression model be determined?
- How can the model be validated?

5.3.1 Determining the Regression Model

The first step in exploring model regression techniques involves assembling data on the underlying variables of interest, and employs regression to estimate the quantitative effect of the causal variables upon the variable that they influence. For a simple problem with one independent variable and one dependent variable, the result can easily be described graphically, as shown in Figure 5-2. The data are drawn in a scatter diagram, then the curve or line that best fits the scatter points of X and Y is determined. However, the response is usually affected by multiple parameters [137, 154, 162], therefore, the hypothesised relationship (first order model) between variables may be written as:

$$Y = \alpha_0 + \alpha_1(X_1) + \alpha_2(X_2) + \cdots + \alpha_n(X_n) + \varepsilon \qquad (5\text{-}1)$$

where: α_0, α_1,. . .,α_n are linear coefficients and ε is the residual. Often a first order model may not fit with the response surface of many engineering problems. Therefore, a higher-order polynomial model may provide better correlation between input and output variables, and a better fit with the response surfaces. Second-order polynomial models are widely used, and can be described by the following equation:

$$Y = \alpha_0 + \alpha_1(X_1) + \alpha_{11}(X_1)^2 + \alpha_2(X_2) + \alpha_{12}(X_2)^2 + \alpha_3(X_3) + \alpha_{13}(X_3)^2 + \cdots + \alpha_n(X_n) + \alpha_{1n}(X_n)^2 + \varepsilon \qquad (5\text{-}2)$$

where α_{11}, α_{12}…. α_{1n} are quadratic coefficients; these values are optimised to produce the best fitting equation. Higher numbers of independent variables (greater than 2) are not easy to describe graphically, and it requires many axes (number of parameters +1). To find the best fit equation, the least-squares method is used. The least-squares method minimises the sum of squares of the lengths of the vertical line segments drawn from the observed data points on the scatter diagram to the fitting equation [154, 162, 169]. It is determined by using the following equation:

$$\sum_{\mathrm{i=1}}^{\mathrm{n}}(Y_i - \hat{Y}_i)^2 = \sum_{\mathrm{i=1}}^{\mathrm{n}}(Y_i - (\alpha_0 + \alpha_1(X_1) + \alpha_{11}(X_1)^2 \ldots\ldots..+\alpha_{1n}(X_n)^2 + \varepsilon)^2 \qquad (5\text{-}3)$$

where y_i is the response at x_i , and $\hat{Y}_i$ is the estimated response as x_i based on the fitted regression line. The equation that provides the smallest value of the sum of the square of this deviation is defined as the best fitting equation for these data.

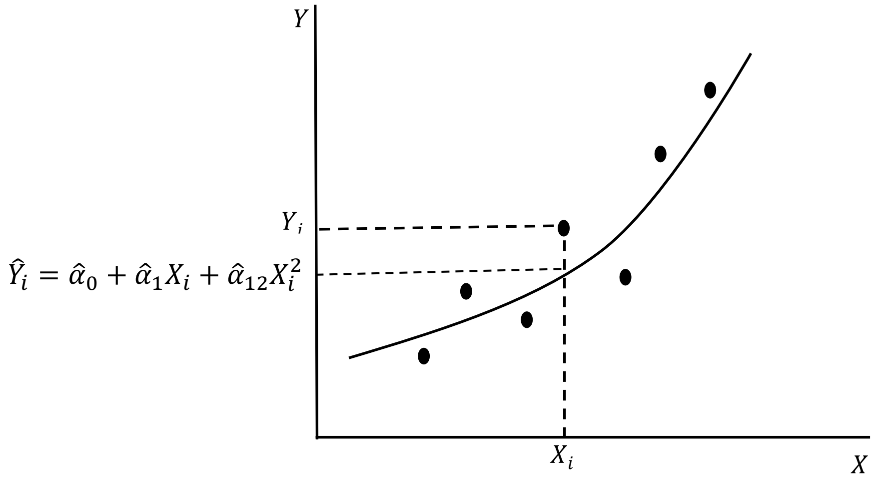

Figure 5-2: Fitting curve in the regression model.

5.3.2 Testing the Regression Model

Once the regression model is obtained to estimate the system response, it is important to evaluate whether this model predicts the output correctly or not, and if so, to what accuracy. Also, does the one particular variable add to the estimation of response achieved by other variable already presented in the regression model?

The regression model can be assessed using three measures, which are:

- The standard error of the estimate.
- The coefficient of determination.
- Testing the hypotheses of the regression model.

These methods have been explained in various textbooks [158, 162, 163, 170]. An explanation of these tools is introduced below.

Sum of squared errors of prediction (SS_E): To measure the accuracy of the model, the sum of the squared error SS_E is used as simple and easy method. It is a measure of the discrepancy between the data and the estimated model. SS_E is calculated by:

$$SS_E = \sum_{i=1}^{n}(Y_i - \hat{Y}_i)^2 \tag{5-4}$$

A small SS_E indicates a close fit of the model to the data, and an SS_E of zero indicates that the predicted equation fits the correct answer perfectly. As the SS_E gets larger, the accuracy of the prediction from the equation reduces. However, SS_E may not give the best indication of an accurate result, because a high value of SS_E can result from an increase in the number of samples rather than an inaccurate model. Instead, the standard error of estimates (S_ε) may be used. The standard error (S_ε) is the standard deviation of the errors of the prediction. It is defined using the following expression:

$$S_\varepsilon = \sqrt{\frac{SSE}{N-Z-1}} \tag{5-5}$$

where Z is the number of independent variables in a regression model, and N is the total number of samples.

The Coefficient of Determination: The coefficient of determination represents the degree to which the variation in the data is matched by the variation in the regression model. Two computational formulas provide useful interpretations; the multi-correlation coefficient, and its square. These quantities are calculated as:

$$R = \frac{\sum_{i=1}^{n}(Y_i-\bar{\mathrm{Y}})(\hat{Y}_i-\bar{\hat{\mathrm{Y}}})}{\sqrt{\sum_{i=1}^{n}(Y_i-\bar{\mathrm{Y}})^2\sum_{i=1}^{n}(\hat{Y}_i-\bar{\hat{\mathrm{Y}}})^2}} \tag{5-6}$$

and

$$R^2 = \frac{\sum_{i=1}^{n}(Y_i-\bar{\mathrm{Y}})^2-\sum_{i=1}^{n}(\hat{Y}_i-\bar{\hat{\mathrm{Y}}})^2}{\sum_{i=1}^{n}(Y_i-\bar{\mathrm{Y}})^2} \tag{5-7}$$

If the value of R^2 is close to 1 then most of the variability in the response is explained by the model (R^2 = 1 indicates a perfect fit). If the value of R^2 is close to 0, the model is not suitable for predicting the response. The percentage of variation in the measured response that is matched by the regression is calculated as $100R^2$ or $100R$.

Testing the Hypotheses of the Regression Model (F-test)

The previous tools (coefficient of determination and standard error) measure the accuracy of the model. Such tools have limitations because they are not able to answer the following questions:

- Is the entire set of independent variables contributing significantly to the output?
- Can the output of the model be predicted by chance?
- Does one of the parameters significantly contribute to the prediction?

Such questions could be answered by testing the hypotheses of the regression models. When building the regression model, it is important to check the accuracy of the model for a given significance level (β). The significance of the model is verified by conducting hypothesis testing. An overall summary of the results of regression analysis can be provided by an analysis of variance (ANOVA) table associated with regression analysis as shown in Table 5-1. Various textbooks, research papers, and computer programs have different ways of presenting the ANOVA tables associated with regression analysis, but in general they contain the same information. This table is the output result of a test recorded using MS Excel.

Table 5-1: Analysis of variance (ANOVA) method for testing the significance of regression.

	dof	*SS*	*MS*	*F*	*Significance F*
Regression	Z	SS_Y	$MS_Y = \frac{SS_Y}{DOF_Y}$	$F_Y = \frac{MS_Y}{MS_E}$	
Residual	$N - Z - 1$	SS_E	$MS_E = \frac{SS_E}{DOF_E}$		
Total	$N - 1$	SS_T			

	Coefficients	*Standard Error*	*t Stat*	*P-value*
Intercept(α_0)	α_0	$S_{\varepsilon i}$	$= \alpha_0 * \varepsilon_n$	
X_1	α_1	$S_{\varepsilon - x1}$		
X_{12}				
X_2				
X_{1n}	α_{1n}	$S_{\varepsilon - x1n}$		

In Table 5-1, SS_Y refers to the regression sum of squares, and is calculated by the following expression:

$$SS_Y = \sum_1^n (Y_i - \bar{Y})^2 \tag{5-8}$$

SS_E is the residual sum of squares and is equal to the sum of squared errors of the prediction (Equation 5-4). SS_T refers to the total sum of squares, and is calculated as:

$$SS_T = SS_Y + SS_E \tag{5-9}$$

MS_Y and MS_E refer to the regression mean squares and the residual mean squares respectively. F_Y is the value of the F-test that is calculated by the following expression:

$$F_Y = \frac{MS_Y}{MS_E} \tag{5-10}$$

Significance F refers to the F-statistic required for testing the hypotheses of the regression model. This value is calculated using the corresponding F-statistic value and the F-distribution table.

For the second (lower table): The term 'Standard error' gives the standard errors (i.e. the estimated standard deviation) of the least square estimates, α. Moreover, it is calculated by using the following expression:

$$S_{\varepsilon i} = \frac{S_{\varepsilon}}{\sum_{i=1}^{n}(Y_i - \hat{Y}_i)^2} \tag{5-11}$$

't_Stat' refers to the T-test, which is determined by the equation:

$$t\_Stat \ = \alpha_n * \varepsilon_n \tag{5-12}$$

The $P - value$ is determined from the F-tables. The term P-value refers to the smallest significance level at which the data leads to rejection of the null hypothesis. To check the model adequacy, the followed step is to state an appropriate hypothesis for testing. The hypothesis for such tests may be generally stated as:

H_0: The regression model is adequate (Null hypothesis)

H_1: The regression model is not adequate (Alternate hypothesis)

For a given significance level (β), the null hypothesis is rejected if the computed P-value is greater than β, and the value of β is considered as computed Significance F. If this condition is not satisfied, then the null hypothesis is accepted.

The Significance F and P-values, are used to check whether the results are reliable (statistically significant). This indicates the probability that the regression output could have been obtained by chance. A small number of Significance F confirms the validity of the regression output. In the example of Significance F is 0.03; this means there is only a 3% chance that the regression output was merely a chance occurrence. However, in the other situation where Significance F and P-values are larger than the confidence level β, the equation is unfit and the result was probably obtained by chance. In this case the equation would need to be optimised. To acquire more reliable results, the parameters that have higher P-values should be deleted, the regression model should be rerun, and another ANOVA table should be produced, until Significance F drops below a predetermined confidence level β.

5.4 Experimental Programme for Developing the Regression Models

The parameters of FFFluid have been studied and the most critical parameters examined in the previous chapter. The critical parameters examined were: size of foam, ratio of foam to fluid, the viscosity of the fluid, the volume of the FFFluid mixture, and input displacement. It was concluded that the size of foam, the viscosity of the fluid, ratio of foam to fluid and the volume of the FFFluid mixture make the greatest contribution while the input displacement gives the lowest contribution out of these parameters. Therefore, displacement was constant in all set of these experiments.

The objective of this set of experimental programme is to develop a regression model that can capture the relationships between process parameters and isolator behaviour with accuracy 85% or more. The model is also intended to predict the behaviour of the system in low input displacement (until 30mm). The parameters of this set were: the size of foam, ratio of foam to fluid, the viscosity of the fluid, and the volume of the FFFluid mixture. The levels of these parameters were chosen based on the previous set of experiments. Only the nested sample does not study because such sample does not provide the outstanding performance; also, such sample does not have a numeric value to use in the regression model. In addition, the levels of foam/fluid ratio in this set are 0.8, 0.9 and100, to examine this parameter in equally spaced. Therefore, the levels of parameters are shown in Table 5-2.

Table 5-2: Parameters and their levels.

Parameters	Level		
Diameter of particles (PS) m	0.0015	0.007	0.018
Ratio Foam height/Package length	0.8	0.9	1
Viscosity of fluid (m^2/s)	0.0001	0.001	0.0125
FFFluid Volume (height of mixture (m))	0.10	0.15	0.20

5.5 Experiments (Taguchi orthogonal array)

After determining the parameters and their levels, the next step is to perform and execute the designed experiment. The experimental tests in this study are conducted using the Taguchi method. To select an appropriate orthogonal array for the experiments, the total degrees of freedom must be determined. In this study, four parameters, each with three levels, means eight degrees of freedom. The appropriate orthogonal array in this case was the standard L9, with four columns and nine rows. The L9 orthogonal array for this study is shown in Table 5-3. The test facilities used are located within the Cardiff University Structural Performance (CUSP) Laboratory.

Table 5-3: Experimental plan using L9 orthogonal array

Trial	Diameter	Ratio Foam to Package	Viscosity of fluid	Volume of FFFluid
1	1	1	1	1
2	1	2	2	2
3	1	3	3	3
4	2	1	2	3
5	2	2	3	1
6	2	3	1	2
7	3	1	3	2
8	3	2	1	3
9	3	3	2	1

As previous experiments, the FFFluid test rig was tested under two types of experiment: static load and dynamic load. The static load test aimed to find the stiffness coefficient of the system. The dynamic load test aimed to calculate the damping coefficient of a FFFluid isolator. A dynamic displacement was applied to the FFFluid isolator; in particular, selected displacement (10mm) were applied at frequency value 1 Hz.

5.6 Results (performance)

The results of these experiments are displayed in Appendix D. From the static test figures, stiffness coefficients were determined for each experiment by using curve fitting. The damping coefficients of the system were determined from the dynamic results as described in chapter 3. For each experiment, these values are displayed in Table 5-4.

Table 5-4: System stiffness and damping coefficient values.

Trial	Stiffness coefficient kN/m	Damping coefficient $(kN.s/m)$
1	18.5	0.955
2	22	1.234
3	24.75	1.385
4	31	1.417
5	19	1.337
6	20.3	1.353
7	29	1.218
8	34	1.281
9	23	1.075

5.7 Mathematical Model (Regression Model)

Regression models are a combination of statistical and mathematical techniques useful for modelling and optimisation of engineering problems in which the relationships between several input parameters and dependent response variables are obtained. By executing the experiments and collecting the data, it is possible to develop analytical or empirical regression models of the response characteristics.

The performance of the isolator is mainly calculated from two coefficients: the stiffness coefficient and the damping coefficient. Therefore, two models were developed: the stiffness model that predicts the stiffness coefficient of the system; and the damping model that predicts the damping coefficient of the system.

5.7.1 Stiffness Coefficient Model

Based on the data of stiffness coefficient given in Table 5-4, and by minimising the sum squares of the error, the second order equations for the stiffness are determined as shown in Equations 5-13. This equation was developed by using the 'Solve' tool in the Microsoft Excel software.

$$k = 218.29(X_1) + 10300.85(X_1)^2 + 80.33(X_2) - 54.31(X_2)^2 + 1279.85(X_3) - 103113.76\,(X_3)^2 - 55.54(X_4) + 510.12\,(X_4)^2 - 11.019 \qquad (5\text{-}13)$$

Where*:* X_1 is the diameter of foam (m), X_2 is a ratio of foam/fluid, X_3viscosity (m^2/s), X_4 is the height of mixture (m).

The next step in developing a model was to check the adequacy of the model. This step focussed on the accuracy and robustness of the established model. To verify the capability of the developed regression model, the validation tools used were: coefficient of determination, R^2, standard error, and testing the hypotheses of the regression model. These tools were implemented using the regression tool in the Microsoft Excel programme. The output results are shown in Table 5-5.

Table 5-5: Summary Output of Regression Model

Regression Statistics	
Multiple R	0.99
R Square	0.99
Adjusted R Square	0.94
Standard Error	1.35
Observations	9.0

ANOVA

	dof	*SS*	*MS*	*F*	*Significance F*
Regression	7	243.24	34.75	18.95	0.175
Residual	1	1.83	1.83		
Total	8	245.08			

	Coefficients	*Standard Error*	*t Stat*	*P-value*
Intercept	-12.933	77.527	-0.167	0.895
X1	218.641	333.532	0.656	0.631
X12	10284.66	16118.53	0.638	0.638
X2	86.083	172.456	0.499	0.705
X22	-57.500	95.76	-0.60	0.656
X3	-40.810	79.964	-0.510	0.700
X4	-55.500	115.443	-0.481	0.715
X42	510.000	383.039	1.331	0.410

From this table, the next results can be obtained: "R Square" in Table 5-5 is the Coefficient of Determination (R^2), which states how well the regression equation approximates the real data. In this case this number equals 0.993. The closer R^2 is to 1, the better the developed regression fits the data. This result therefore indicates that the model is adequate.

The second tool is the Standard Error (SE); this value is equal 1.35. A small SE indicates a close fit of the model to the data. Therefore, this result also indicates the predicted equation fits well with the correct answer.

Another critical tool is the F-test. In Table 5-5, the value of Significance F is = 0.175. Also, the P-value for some parameters is higher than 0.7. This means that there is around 70% chance that the regression output was merely a chance occurrence.

To achieve a confidence level of 85%, it is important that the values of Significant F and P-value for each parameter be below 0.15. Therefore, the produced model must be optimised. To improve the model results, the parameter with the highest P-value was deleted, and the regression model was run again. This was repeated until the Significance F and P-value for each parameter dropped below 0.15.

From the Table (5-5) it can be seen that the P-value of X4 is the highest in comparison with the other P-values. Therefore, this parameter was deleted, and the regression equation was constructed again. This is shown in Appendix F, where it can be seen that the Significant F was still higher than 0.15. The steps were therefore repeated, each time removing the parameter with the highest P-value. After three iterations, Equation 5-14 was derived. This represents the optimised equation, and the regression table for this result is given in Table 5-6.

$$k = 427.27\ (X_1) - 9.725(X_2)^2 + 326.697\ (X_4)^2 + 20.89 \tag{5-14}$$

Table 5-6: Summary Output of optimised Regression Model

Regression Statistics	
Multiple R	0.992
R Square	0.984
Adjusted R Square	0.974
Standard Error	0.887
Observations	9

ANOVA

	dof	*SS*	*MS*	*F*	*Significance F*
Regression	3	241.140	80.380	102.026	0.6E-04
Residual	5	3.9391	0.7878		
Total	8	245.08			

	Coefficients	*Standard Error*	*t Stat*	*P-value*
Intercept	20.889	1.808	11.551	8E-05
X1	427.272	43.131	9.906	1.8E-04
X22	-9.725	2.012	-4.83	0.0047
X42	326.69	24.046	13.58	3.8E-05

From this table it can be seen that:

- R^2 = 0.99
- Standard Error (SE) = 0.88
- Value of Significance F is = 0.6E-04
- and highest P-value for the coefficient is =0.005

Therefore, the accuracy of this model is greater than 85 %. To confirm this model, several randomly selected validation experiments were carried out within the range of explored experimental parameters. The results of these experiments are shown in Appendix E. Table 5-7 presents the experimental order, the actual values, the predicted values and their deviations (percentage errors) for the stiffness coefficients.

Table 5-7: Comparison between model and experimental result of stiffness coefficients

Trials	Size (m)	Ratio	Viscosity m^2/s	Mixture height (m)	k-Model	k-Exp.	Error
1	0.0015	1	0.0125	0.15	19.156	18	6%
2	0.0015	0.8	0.001	0.2	28.374	27.4	4%
3	0.007	0.9	0.001	0.15	23.353	24.4	-4%
4	0.007	0.8	0.0001	0.2	30.724	32.5	-5%
5	0.018	0.9	0.0125	0.1	23.969	21.8	10%
6	0.018	1	0.0001	0.1	22.122	22.1	0%

5-7-2 Damping Coefficient Model

By using the same steps of stiffness model, the damping coefficient model is determined. The initial regression model by using the Excel software is:

$$\mathrm{C} = 57.32(\mathrm{X}_1) - 2939.6(\mathrm{X}_1)^2 + 9.03(\mathrm{X}_2) - 4.81(\mathrm{X}_2)^2 + 53.84(\mathrm{X}_3) - 3526.79(\mathrm{X}_3)^2 + 5.75(\mathrm{X}_4) - 11.21(\mathrm{X}_4)^2 - 3.74 \tag{5-15}$$

Where: X_1 is diameter of foam (m), X_2 is a ratio of foam/fluid, X_3 viscosity (m^2/s), X_4 is height of mixture (m). The following step of developing the modelling is checking the adequate of the model. To validate such these equation, the excel software was used. The output result are shown in Table 5-8

Table 5-8: Summary Output of developing damping Regression Model.

Regression Statistics	
Multiple R	0.994
R Square	0.988
Adjusted R Square	0.906
Standard Error	0.046
Observations	9

ANOVA

	dof	*SS*	*MS*	*F*	*Significance F*
Regression	7	0.182	0.026	12.057	0.218
Residual	1	0.002	0.002		
Total	8	0.184			

	Coefficients	*Standard Error*	*t Stat*	*P-value*
Intercept	-3.889	2.660	-1.462	0.382
X1	57.312	11.444	5.008	0.125
X12	-2939.06	553.064	-5.314	0.118
X2	9.448	5.917	1.597	0.356
X22	-5.042	3.286	-1.535	0.368
X3	8.063	2.744	2.939	0.209
X4	5.573	3.961	1.407	0.393
X42	-10.616	13.143	-0.808	0.567

From Table 5-8. R^2 is 0.98 and Standard Error (SE)=0.046, this result indicates the predict equation is fit the correct answer. (the model is predicting the data with high confidence)

However, the value of Significance F is 0.218. This is meaning, the developed equation needs to be optimized. To get confidence level 85%, the parameters that have higher P-value is deleted. Then, developing the regression model again until Significance F drops below 0.15. After two iterations shown in Appendix G. The equation 5-16 is presented the optimized equation, and the regression table for this result is Table 5-9.

$$C = 57.31(X_1) - 2939.06(X_1)^2 + 0.201(X_2)^2 + 8.06(X_3) + 2.388(X_4) + 0.553,$$

(5-16)

Table 5-9: Summary Output of optimised damping Regression Model

Regression Statistics	
Multiple R	0.975
R Square	0.950
Adjusted R Square	0.868
Standard Error	0.054
Observations	9

ANOVA

	dof	*SS*	*MS*	*F*	*Significance F*
Regression	5	0.175	0.035	11.595	0.035
Residual	3	0.009	0.003		
Total	8	0.184			

	Coefficients	*Standard Error*	*t Stat*	*P-value*
Intercept	0.553	0.131	4.211	0.044
X1	57.312	13.544	4.232	0.064
X12	-2939.06	654.52	-4.490	0.021
X22	0.201	0.125	1.613	0.15
X3	8.063	3.247	2.483	0.081
X4	2.389	0.449	5.320	0.13

From this table: R^2=0.95

Standard Error (SE) =0.055

Value of Significance F is = 0.035

The highest P-value for coefficient is =0.15.

Therefore, the accuracy of this model is more than 85%. To validate this model, several randomly picked validation experiments were carried out within the range of explored experimental parameters. The results of these experiments are shown in Appendix E. Table 5-10 present the experiments order, the actual values, the predicted values and their deviations (percentage errors) for damping coefficients.

Table 5-10: Comparison between model and experimental result of damping coefficient

Trials	Size (m)	Ratio	Viscosity m^2/s	Mixture height (m)	C-Model	C-Exp.	Error
1	0.0015	1	0.0125	0.15	1.292	1.210	7%
2	0.0015	0.8	0.001	0.2	1.247	1.082	15%
3	0.007	0.9	0.001	0.15	1.339	1.464	-9%
4	0.007	0.8	0.0001	0.2	1.417	1.592	-11%
5	0.018	0.9	0.0125	0.1	1.135	1.337	-15%
6	0.018	1	0.0001	0.1	1.073	0.955	12%

5.8 Summary of Chapter 5

It is necessary to find an optimal process condition capable of producing the desired isolating quality and also to automate the parameters. The performance of FFFluid isolators depends mainly on several key parameters such as the size of foam particles and the volume of the package. These parameters should be selected in a judicious manner in order to reach the desired results dictated by the area of application of the FFFluid device. A proper model should be constructed and tested before implementing in real applications.

The mathematical models that predict the stiffness and damping coefficients of the system were defined in the selected case. Such models were determined based on statistical analysis of experimental results which is a unique developed methodology in this area. These models were determined by using regression method based on experimental data. Therefore, different tools were used to check the accuracy of these models. Experimental results also validated the mathematical models.

From such result, it could be concluded that the mathematical model of other FFFluid cases could be predicted by using the same method 'regression analysis of experimental data'. Therefore, the performance of the FFFluid device could be determined and the control system could be established for further investigations.

CHAPTER 6

CONCLUSIONS AND RECOMMENDATIONS FOR FUTURE WORK

6.1 Introduction

The primary motivation for this work was to address future requirements for anti-vibration devices, especially in the context of suspension systems for lightweight vehicles. Indeed, suspension systems that exhibit variable stiffness and damping capabilities are of interest in this case. The design of an adaptive suspension systems or vibration isolators with variable stiffness and damping coefficients can also be of benefit for other applications such as engine mounts. The possibility of achieving this by implementing the concept of Foam Filled Fluid (FFFluid) has been suggested in this thesis. Thus, the main aim of this research was to conduct a systematic study to understand, optimise and contribute to the prediction of the performance of FFFluid–based vibration isolators.

The conclusions reached in this thesis are brought together in this Chapter to summarise the achievements of this work. The overall scientific contributions are also presented and discussed. Finally, this chapter suggests research areas that should be considered for further studies.

6.2 Conclusions

The main conclusions from the work completed in this research can be summarised as follows:

The key findings from the work completed in this research can be summarised as follows:

- Theoretical exploration of the phenomena of interest identified several mechanisms of elasticity and damping contributions that could be achieved within a FFFluid isolator. These mechanisms are (i) the work done by compressing the elastomeric material; (ii) the work done by the matrix fluid as it shears; (iii) the stress-strain characteristics of the packaging material; and (iv) the elastic behaviour of the FFFluid due to the compressed elastomeric particles and cell gas.

- The FFFluid composite system exhibits interesting potential to enhance the performance of anti-vibration devices. The main benefit is the capability of adjusting stiffness and damping coefficients by changing one component or more of this mixture.

- The benefits of FFFluid-based systems over spring and damper devices were also studied theoretically. The main advantages of a FFFluid isolator in this case are that it weighs less, it is cheaper, and it is easier to design. It is anticipated that the total weight of a car could be reduced by around 6kg when using the FFFluid unit as a suspension system instead of a coil spring and hydraulic damper unit. Consequently, this could contribute to reducing the fuel consumption by 1%.

- A comparison between FFFluid and dry foam was also conducted experimentally. It was shown that the examined FFFluid sample was able to carry a static load 60% higher compared to a dry foam sample occupying the same volume.

- The performance of a FFFluid-based isolator could be affected by many parameters. The theoretical study conducted in this research identified that these parameters could include: the size of the foam particles; the ratio of foam to fluid; the viscosity of the fluid; the volume of composite material the type of cells in the foam; the shape of the foam; the mechanical properties of the foam; the design of the packaging; and the input load amplitude and frequency.

- From the Taguchi parameters design method, which was conducted on a selected set of input factors, the most influential ones on the stiffness coefficient could be identified. These were: the volume of composite material, the ratio of foam to fluid and the foam size. Also, the most influential parameters on the damping coefficient were found to be: the foam size, the viscosity of the fluid, the volume of the mixture and the ratio of foam to fluid.

- It was shown that increasing volume of FFFluid provides higher stiffness and damping coefficients.The higher stiffness coefficient could be achieved by increasing the amount of fluid and using the bigger size for the foam particles. It was also observed that an input displacement up to 30mm has a negligible effect on the stiffness or damping coefficients of the system.

- An empirical mathematical model was also established. Such a model could be used to predict both stiffness and damping coefficients of a FFFluid-based isolator.

- The developed mathematical model was also validated by conducting further experimental tests. The accuracy of this model was found to be 85%.

6.3 Contributions of the Research

The work undertaken by the author and described in this thesis includes several aspects which are novel and not previously implemented by other researchers or practitioners. More specifically, the contributions of the thesis can be summarised as follows:

Contribution 1**:** The author believes that the application of the FFFluid concept as a vibration isolator solution is novel. No previous studies have been found in the literature that describes the application of the FFFluid in this context. Both stiffness and damping coefficients of FFFluid-based isolator systems could be adjusted via changing one or more parameters without the complexity associated with an active system. The associated benefit could be utilised in many applications, especially in suspension systems for lightweight vehicles. Therefore, this research studied this complex material in vibration isolator applications.

Contribution 2*:* Some advantages of the FFFluid mixture have been reported previously. This study extended this knowledge by showing the benefit of using the FFFluid composite material to reduce stress concentration in the elastomeric material. The sample of FFFluid carried a static load 60% higher compared to a dry foam sample occupying the same volume. Therefore, the lifespan of elastomer material could be increased by mixing it with fluid in enclosed package. This is an effect that has not been reported previously.

Contribution 3: This research found that the performance of a FFFluid device depends on a number of parameters. Previous studies had investigated the influence of some parameters under shock loads. However, in the context of vibrating loads, a knowledge gap still exists regarding the effects and the importance of such parameters on the performance of FFFluid-based isolators. Therefore, the author believes that the reported characterisation and study of the parameters for FFFluid vibration isolation systems is a contribution to knowledge. A unique approach was developed to characterising this mixture. This was achieved as follows. The phenomena of interest were identified first. This was followed by the formulation of hypotheses with respect to the theoretical influence of different parameters. Then, the hypotheses were collected to describe the system behaviour. An experimental programme was conducted to validate this study. Thus, this investigation provided a better understanding of the behaviour of the FFFluid mixture in the context of vibration isolation.

Contribution 4: The author believes that the development of a mathematical model, which can capture the behaviour of a FFFluid isolator, has not previously been achieved. The model reported in this work can predict the performance of such FFFluid devices in selected case. Such model was achieved by using regression analysis of experimental data. Therefore, the mathematical model of other FFFluid designs could be predicted by using the same method. In practice, this means that the control system of a FFFluid-based design could be established for development adaptive system. Thus, this can contribute to reducing the time and cost associated with the design of FFFluid devices, which is another distinguishing characteristic of this work. Indeed, the performance of FFFluid systems has commonly been evaluated by designing prototypes, which are then assessed experimentally.

6.4 Recommendations for Future Work

In light of the conclusions of this work, the following topics are recommended for further study:

- This thesis has made a contribution towards the development of the FFFluid concept for application in vibration isolation. The next step should be concerned with developing an effective control system for such a device. The design of an actual vibration system, which incorporates real masses and MR valve is also required.

- The behaviour of flexible foams also depends on the input frequency and amplitude. These parameters were not studied in this research. Thus, more investigations are recommended to examine the effect of these parameters on the response of the system.

- This study was limited to evaluate a set of parameters, which were selected as most critical. More experimental work should be conducted to investigate the influence of different shapes of particles, various types of foams and different designs of packages.

- The fluid may have an effect on particles over a long-term period, or trapped gas may diffuse into the surrounding liquid. Therefore, the long-term behaviour of different formulations of FFFluid also needs to be examined.

- Many engineering systems require varying the stiffness and damping coefficients to meet requirements such as engine mounts. The FFFluid mixture is also a promising solution in this case. Hence, it is recommended to investigate the FFFluid concept for this particular application.

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APPENDIX- A: Results of initial experiments

Density (kg/m^3)	Stiffness (kN/m)	Damping force at middle position 2CV (kN)	Frequency (Hz)	Max. Displacement (X)mm	Damping coefficient ($N.\frac{sec}{m}$)
PE - 30	23.4	0.12	1	10	959

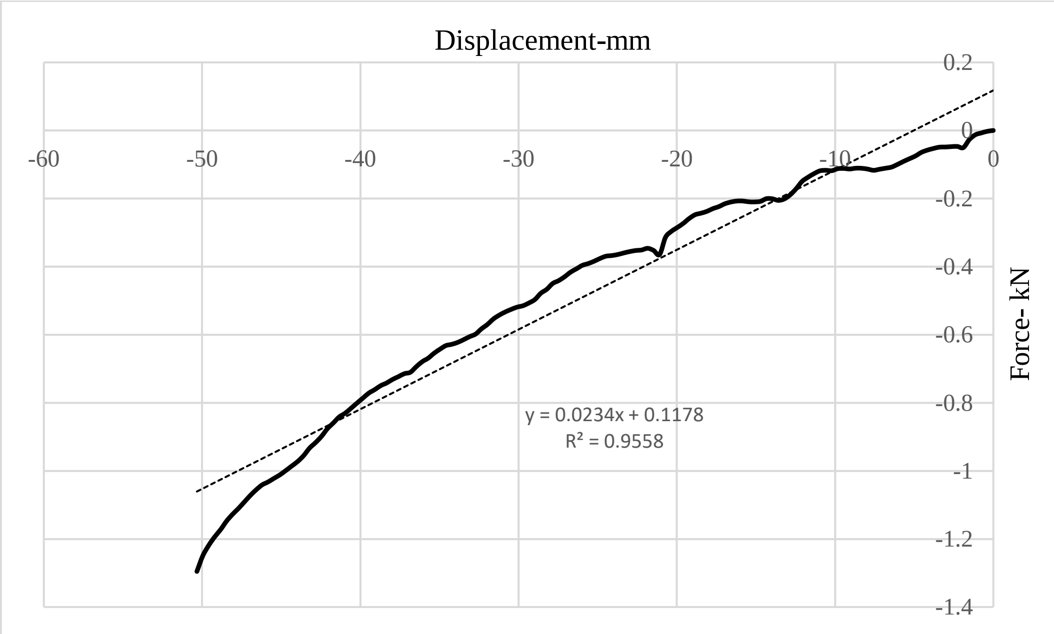

Figure A-1: Displacement- Force graph of FFFluid under static load by using PE-30.

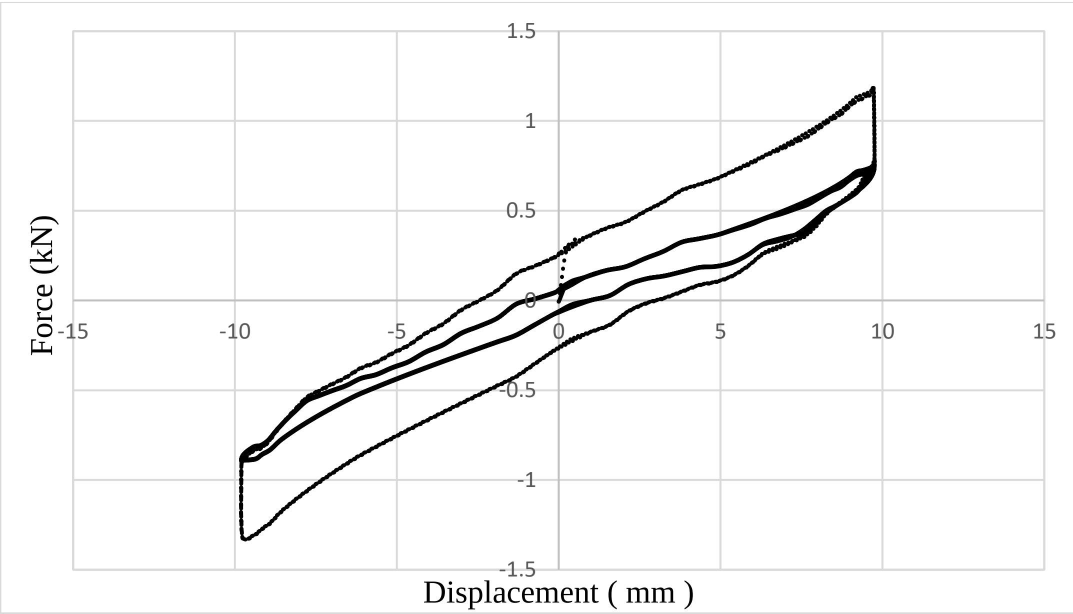

Figure A-2: Displacement- viscous force graph of FFFluid under dynamic load by using PE-30

Density (kg/m^3)	Stiffness (kN/m)	Damping force at middle position 2CV (kN)	Frequency (Hz)	Max. Displacement (X)mm	Damping coefficient $(N.\frac{sec}{m})$
PE - 125	34.2	0.275	1	10	2189

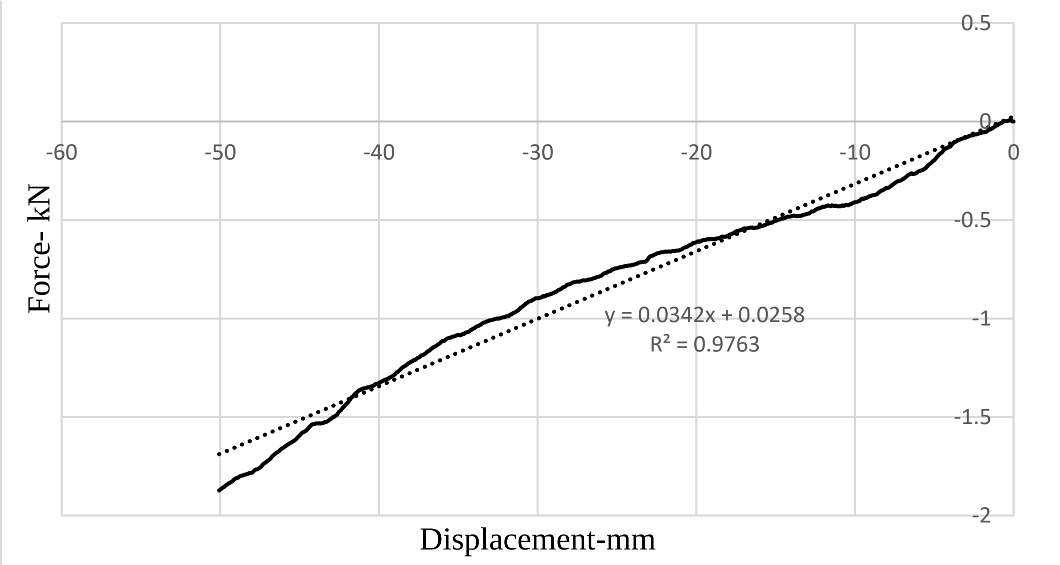

Figure A-3: Displacement- Force graph of FFFluid under static load by using PE-120.

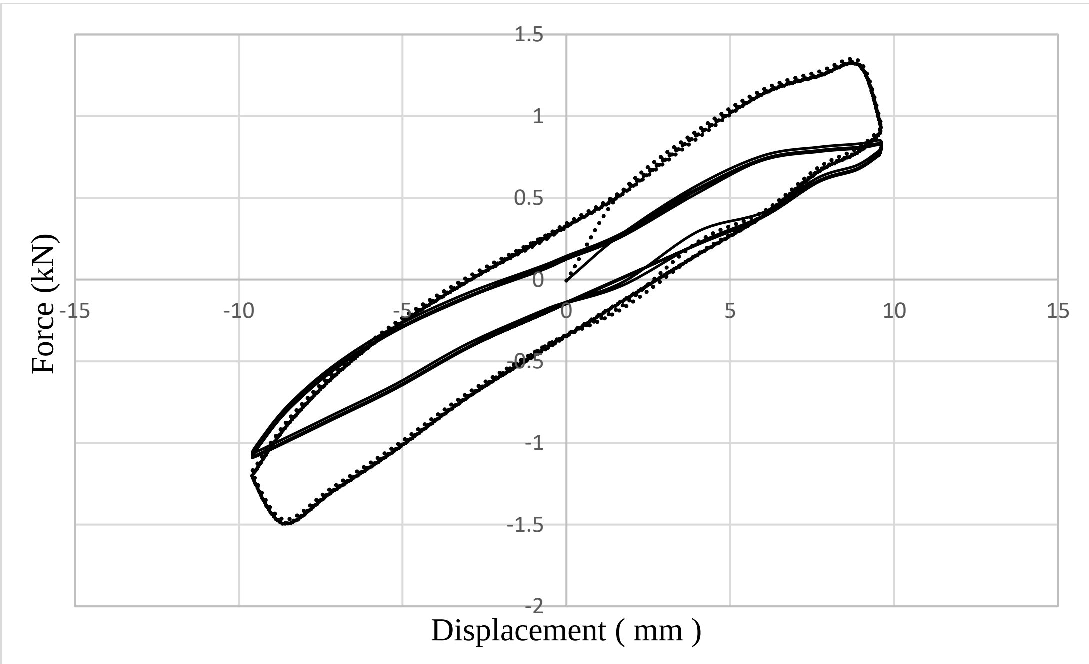

Figure A-4: Displacement- viscous force graph of FFFluid under dynamic load by using PE-120

APPENDIX- B : Matlab Simulink

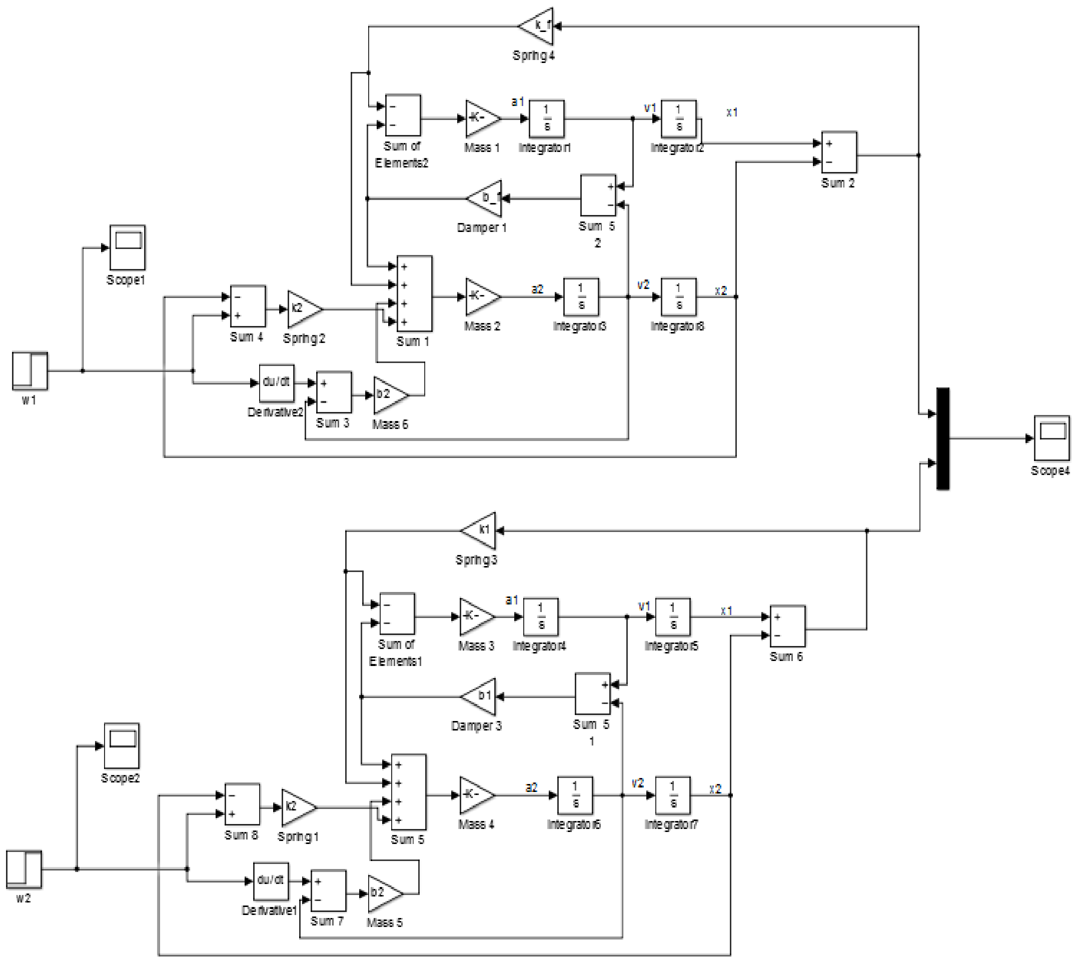

Figure B-1: Block diagram of step input road for a quarter car model

APPENDIX- C: Results of set 1

Trial 1	Stiffness (kN/m)	Damping force at middle position 2CV (kN)	Frequency (Hz)	Damping coefficient $(\mathrm{N.}\frac{\mathrm{sec}}{\mathrm{m}})$
Static test	14.1	-	-	-
Dynamic test	14.1	0.035	0.75	743
Dynamic test	14.1	0.048	1	765

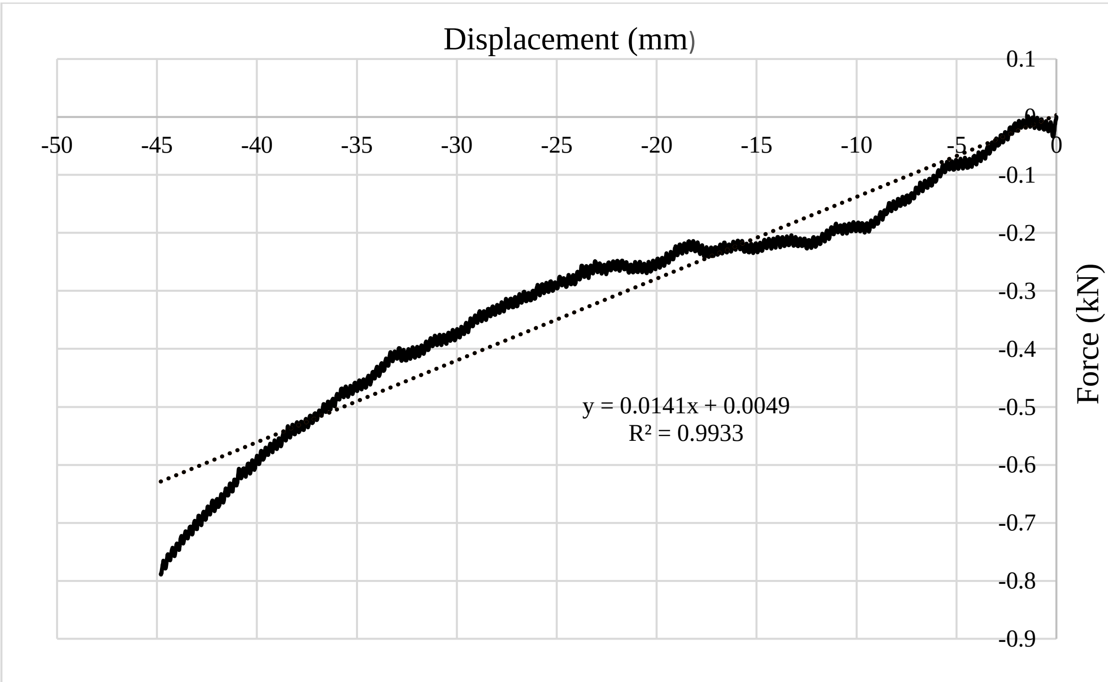

Figure C-1: Displacement- Force graph of FFFluid under static load for trial 1.

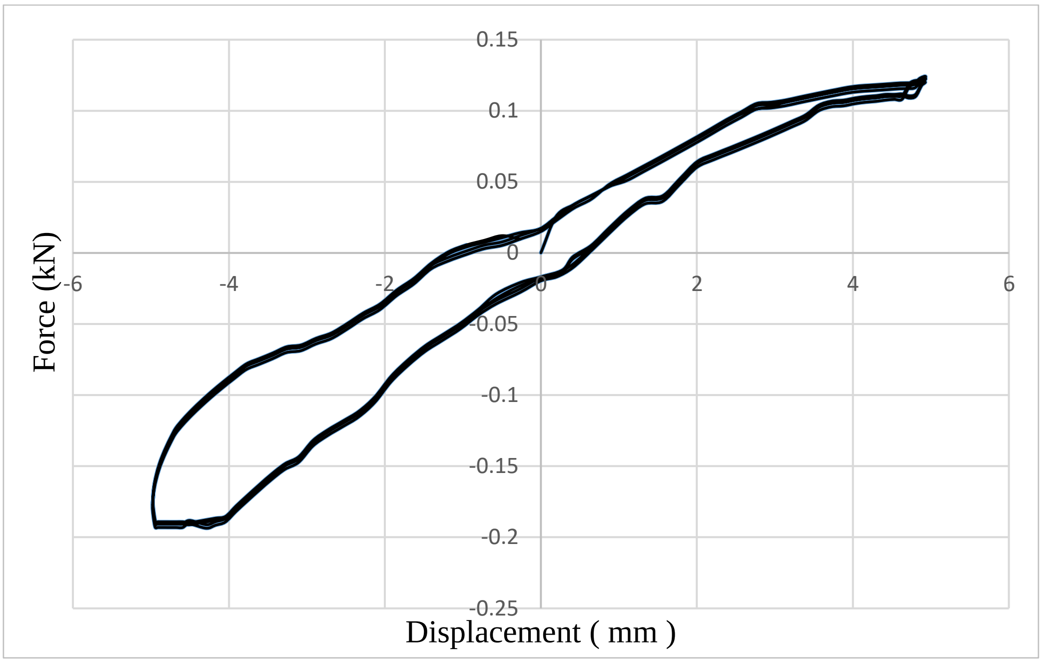

Figure 0-2: Displacement- viscous force graph of FFFluid under dynamic load and frequency 0.75Hz for trial 1.

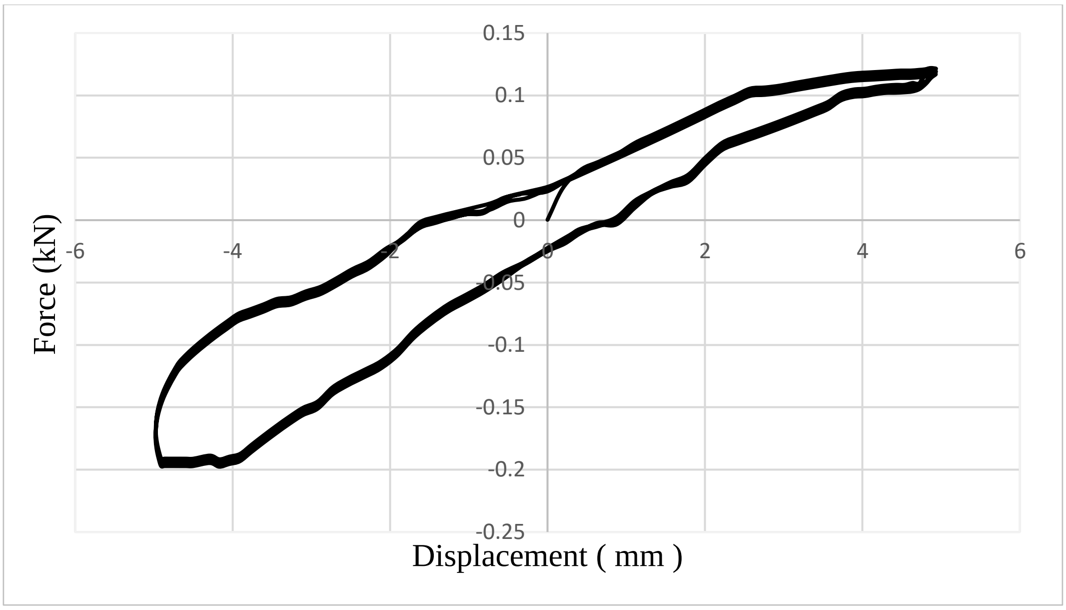

Figure C-3: Displacement- viscous force graph of FFFluid under dynamic load and frequency 1.0Hz for trial 1.

Trial 2	Stiffness (kN/m)	Damping force at middle position 2CV (kN)	Frequency (Hz)	Damping coefficient $(N.\frac{sec}{m})$
Static test	16.8	-	-	-
Dynamic	16.8	0.1	0.75	1062
Dynamic	16.8	0.13	1	1035

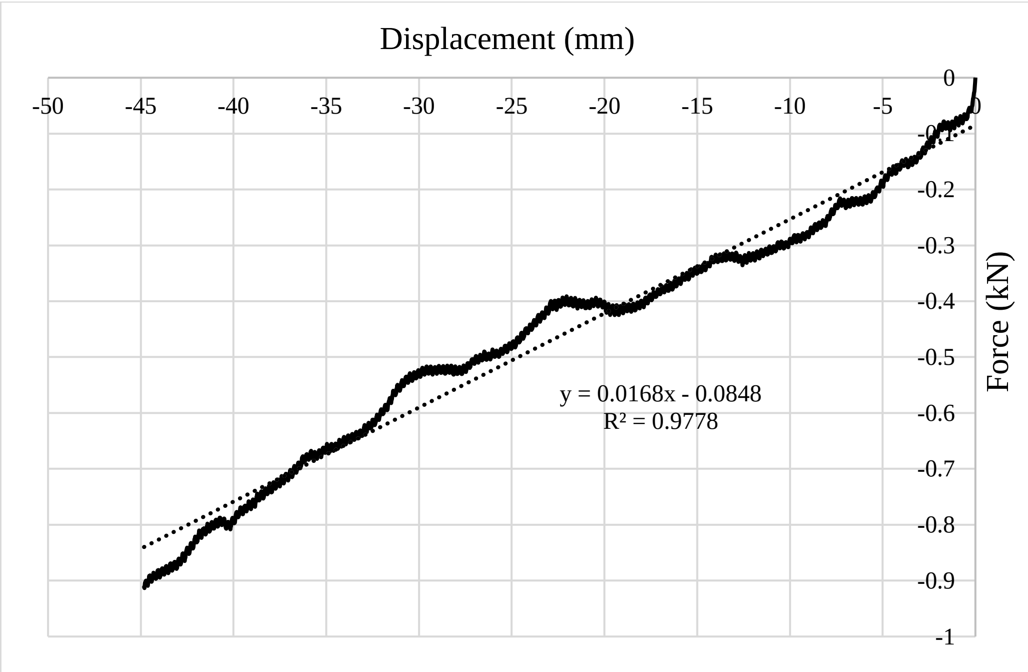

Figure C-4: Displacement- Force graph of FFFluid under static load for trial 2.

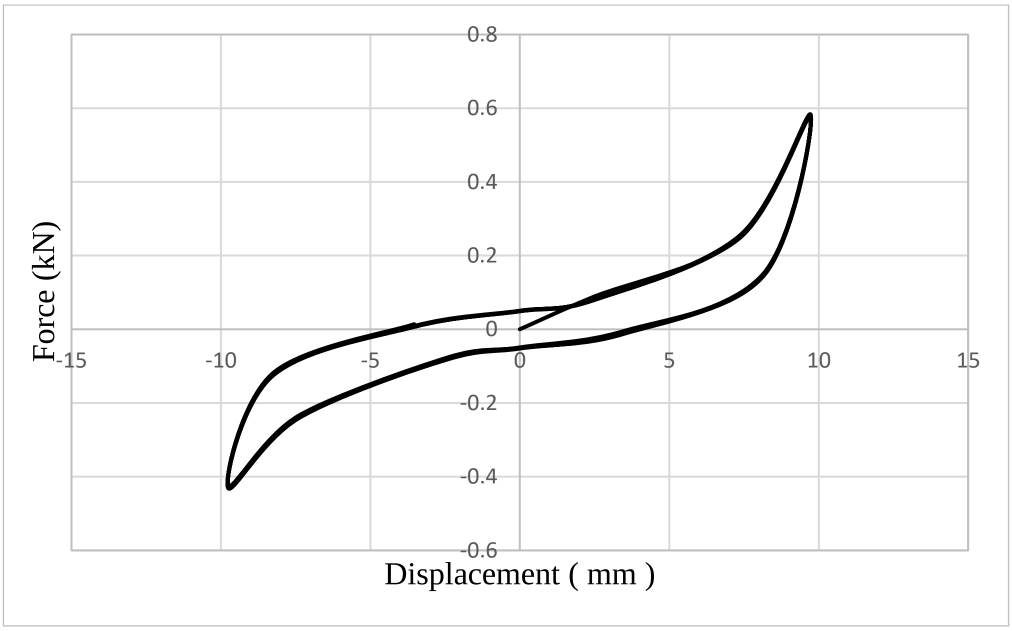

Figure C-5: Displacement- viscous force graph of FFFluid under dynamic load and frequency 0.75Hz for trial 2.

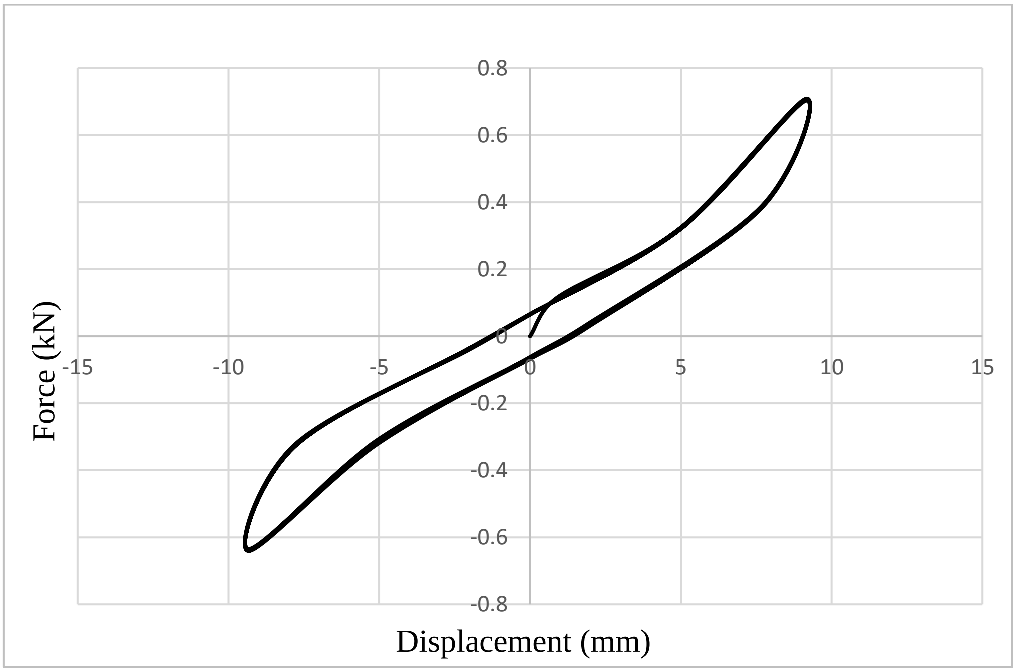

Figure C-6: Displacement- viscous force graph of FFFluid under dynamic load and frequency 1.0Hz for trial 2.

Trial 3	Stiffness (kN/m)	Damping force at middle position 2CV (kN)	Frequency (Hz)	Damping coefficient (N.$\frac{sec}{m}$)
Static test	24.7	-	-	-
Dynamic	24.7	0.27	0.75	1433
Dynamic	24.7	0.36	1	1433

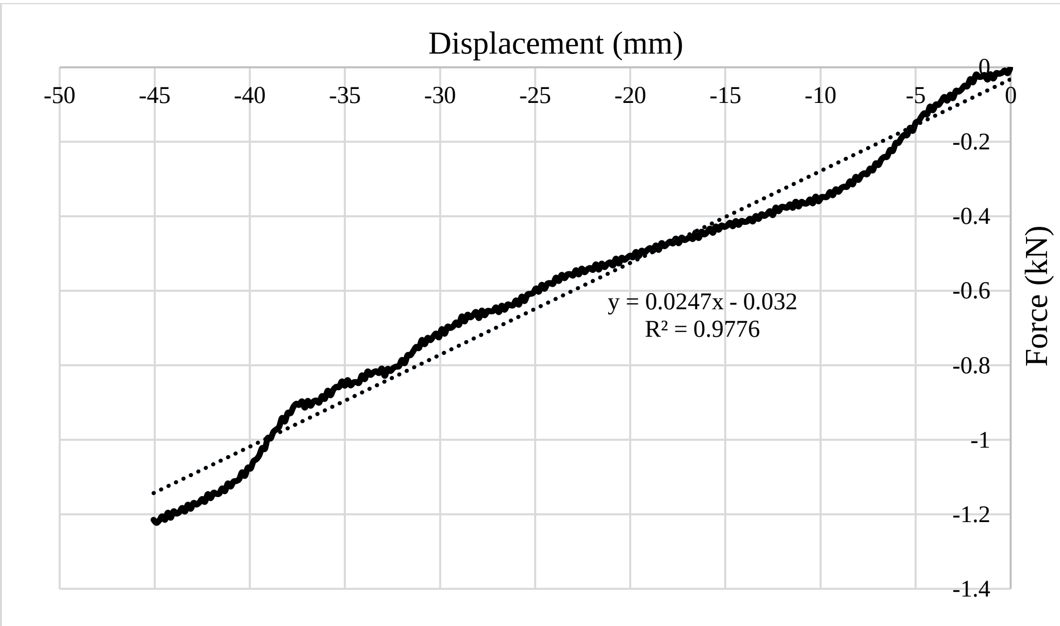

Figure C-7: Displacement- Force graph of FFFluid under static load for trial 3.

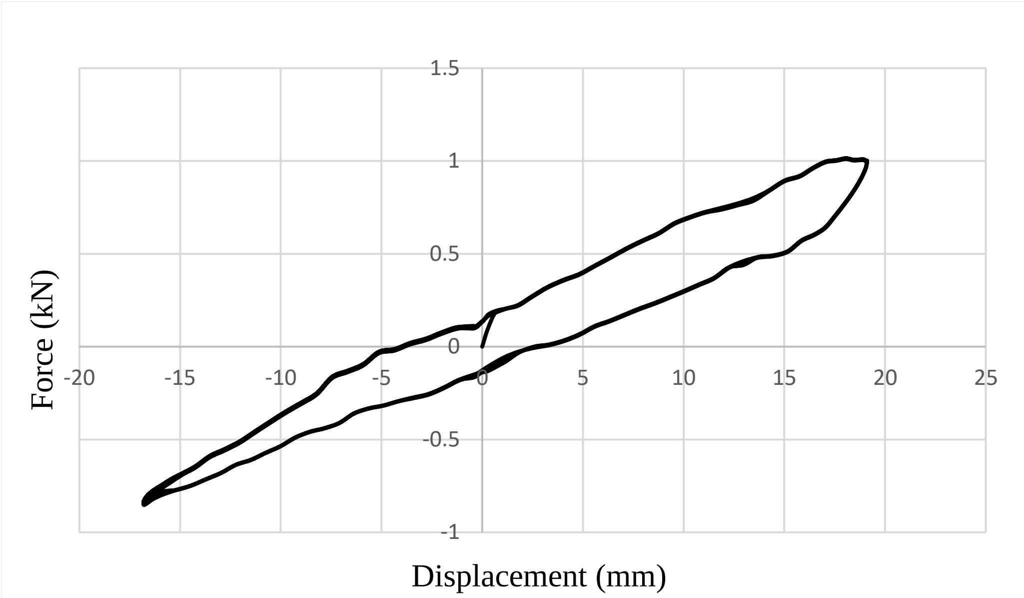

Figure C-8: Displacement- viscous force graph of FFFluid under dynamic load and frequency 0.75Hz for trial 3.

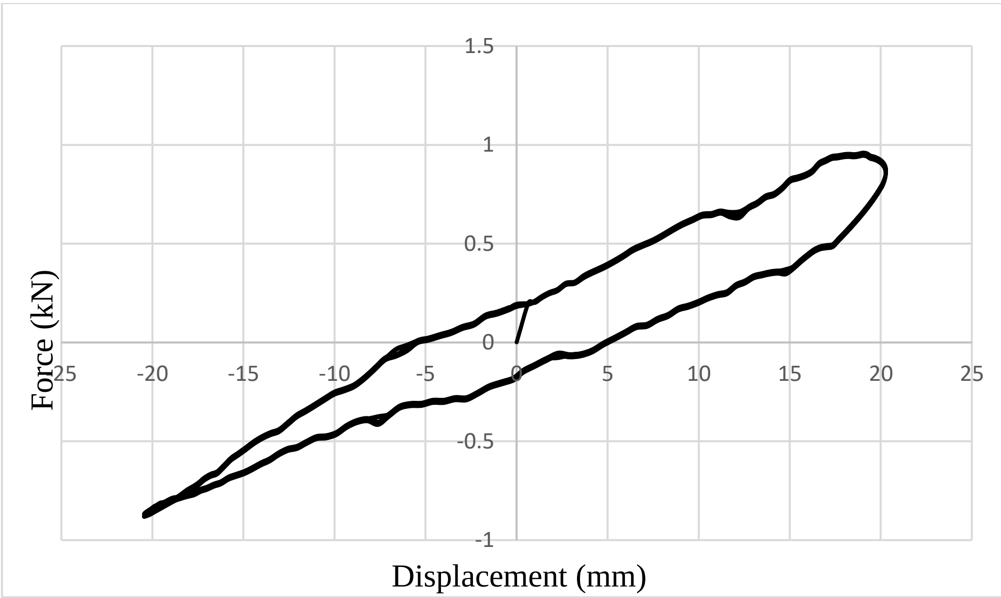

Figure C-9: Displacement- viscous force graph of FFFluid under dynamic load and frequency 1.0Hz for trial 3.

Trial 4	Stiffness (kN/m)	Damping force at middle position 2CV (kN)	Frequency (Hz)	Damping coefficient $(N.\frac{sec}{m})$
Static test	28	-	-	-
Dynamic	28	0.36	0.75	1273
Dynamic	28	0.48	1	1273

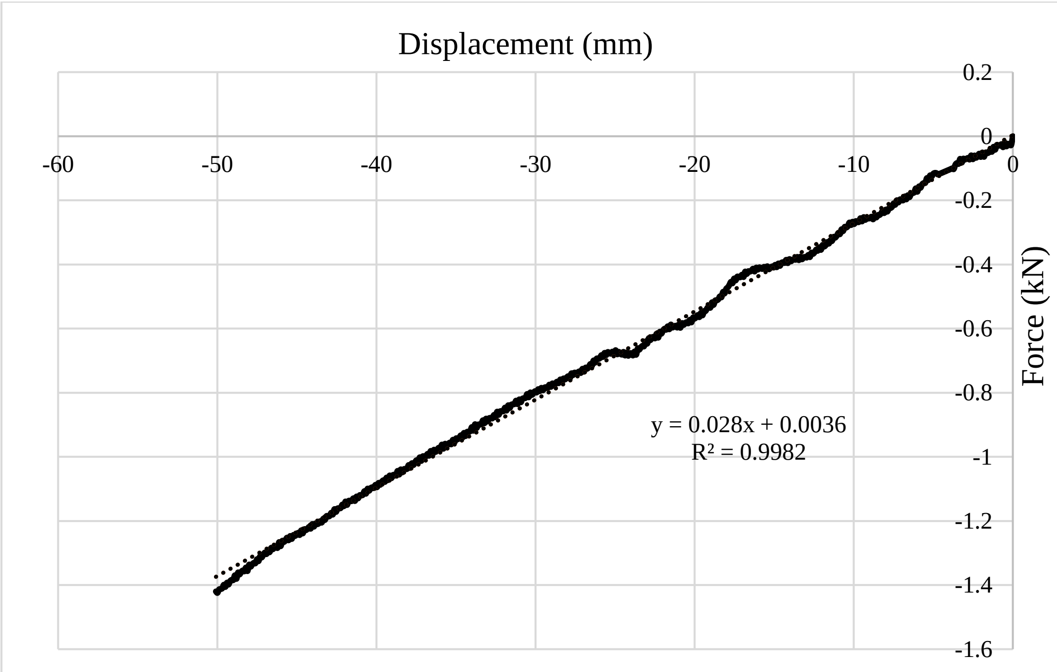

Figure C-10: Displacement- Force graph of FFFluid under static load for trial 4.

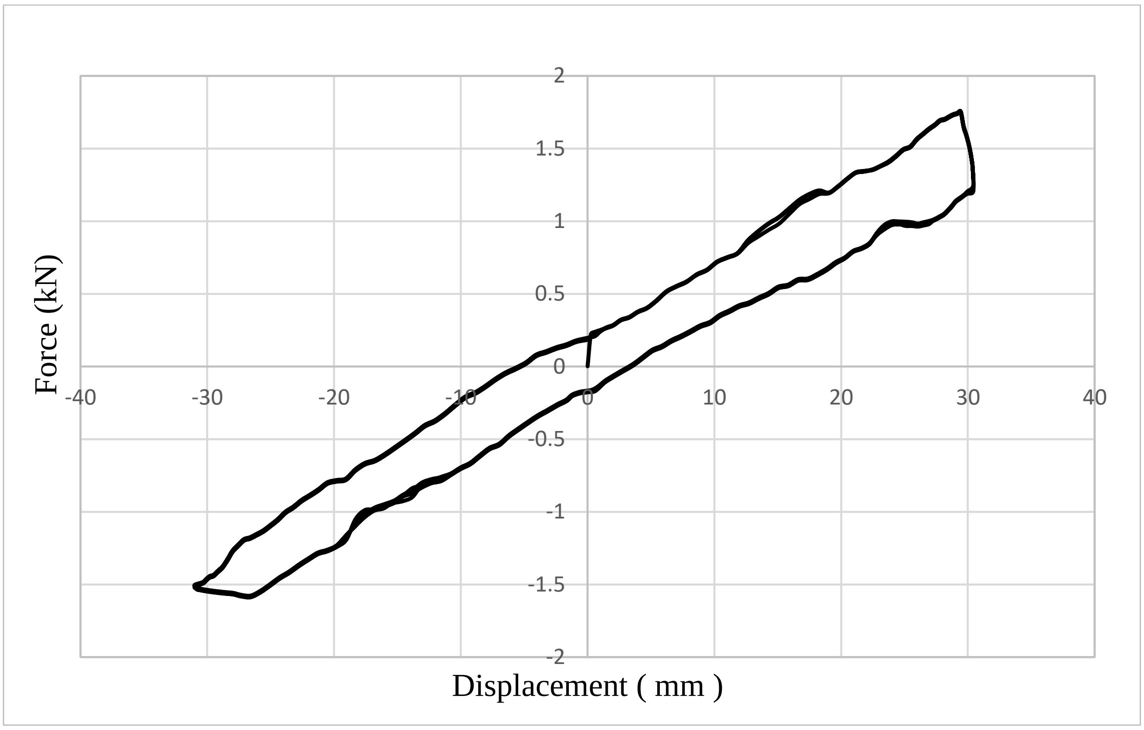

Figure C-11: Displacement- Damping force graph of FFFluid under dynamic load and frequency 0.75Hz for trial 4.

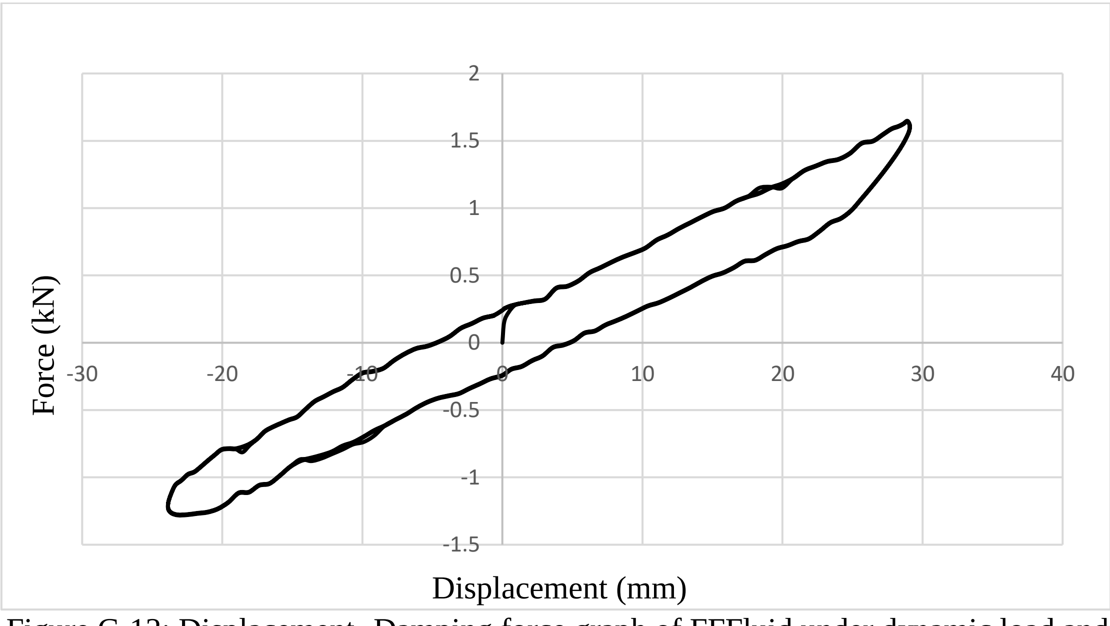

Figure C-12: Displacement- Damping force graph of FFFluid under dynamic load and frequency 1.0Hz for trial 4.

Trial 5	Stiffness (kN/m)	Damping force at middle position 2CV (kN)	Frequency (Hz)	Damping coefficient (N. $\frac{sec}{m}$)
Static test	22.4	-	-	-
Dynamic	22.4	0.38	0.75	1344
Dynamic	22.4	0.5	1	1327

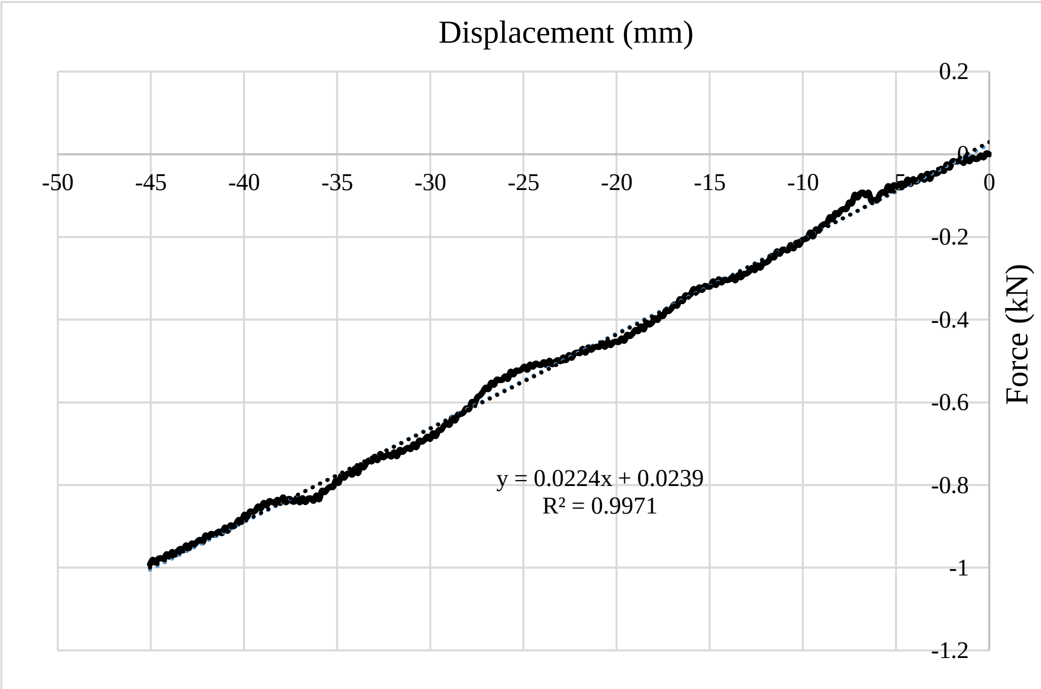

Figure C-13: Displacement- Force graph of FFFluid under static load for trial 5.

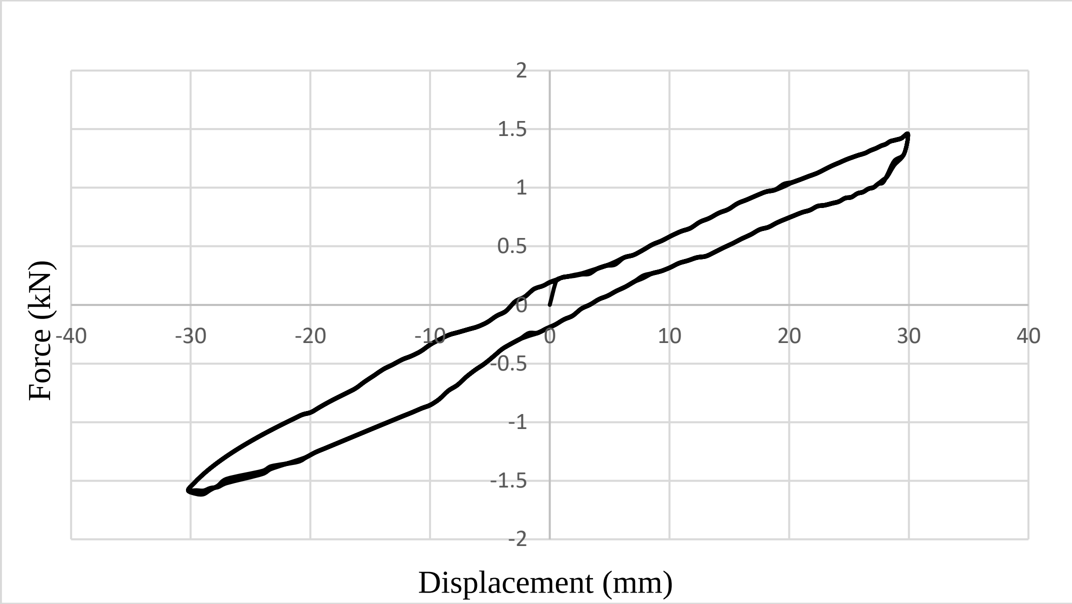

Figure C-14: Displacement- Damping force graph of FFFluid under dynamic load and frequency 0.75Hz for trial 5

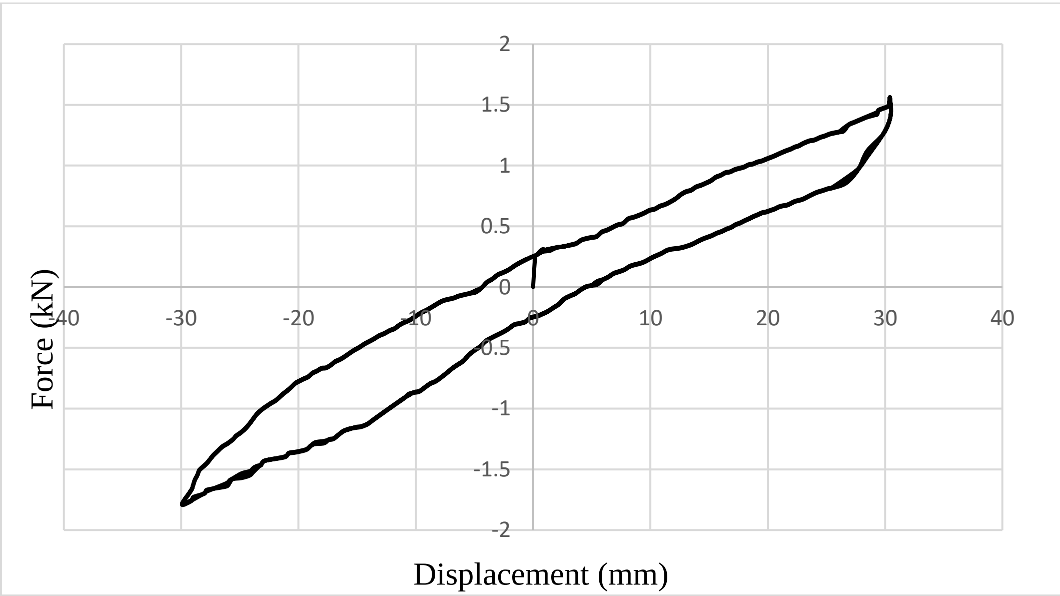

Figure C-15: Displacement- Damping force graph of FFFluid under dynamic load and frequency 1.0Hz for trial 5.

Trial 6	Stiffness (kN/m)	Damping force at middle position 2CV (kN)	Frequency (Hz)	Damping coefficient $(\mathrm{N.}\frac{\sec}{\mathrm{m}})$
Static test	24.6	-	-	-
Dynamic	24.6	0.29	0.75	1540
Dynamic	24.6	0.38	1	1513

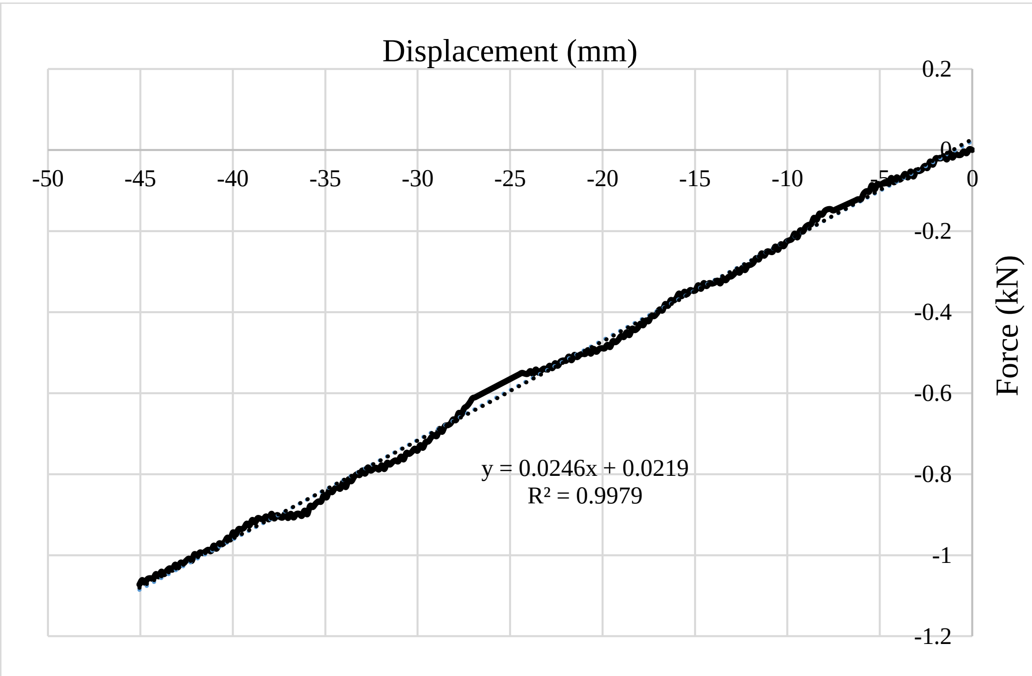

Figure C-16: Displacement- Force graph of FFFluid under static load for trial 6.

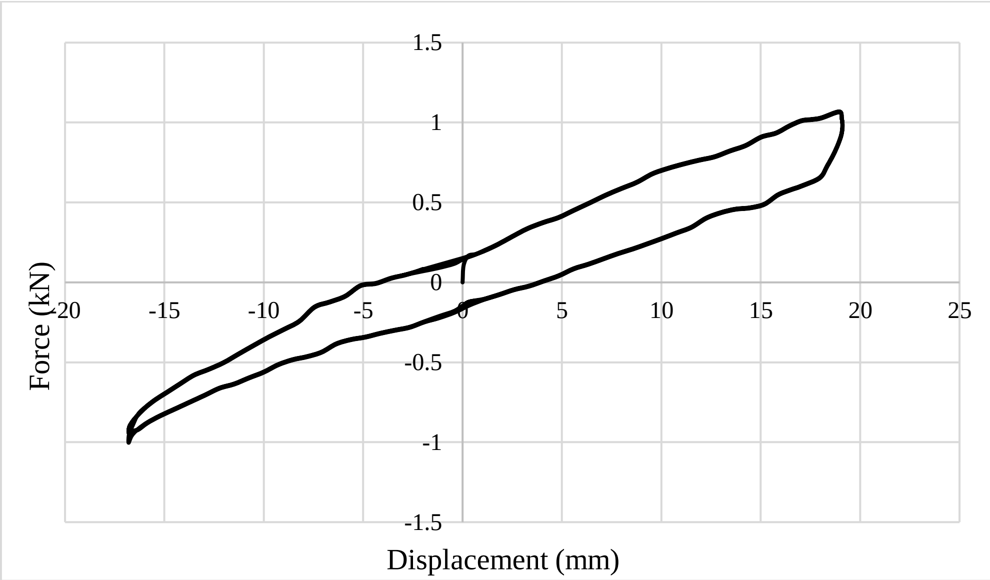

Figure C-17: Displacement- Damping force graph of FFFluid under dynamic load and frequency 0.75Hz for trial 6.

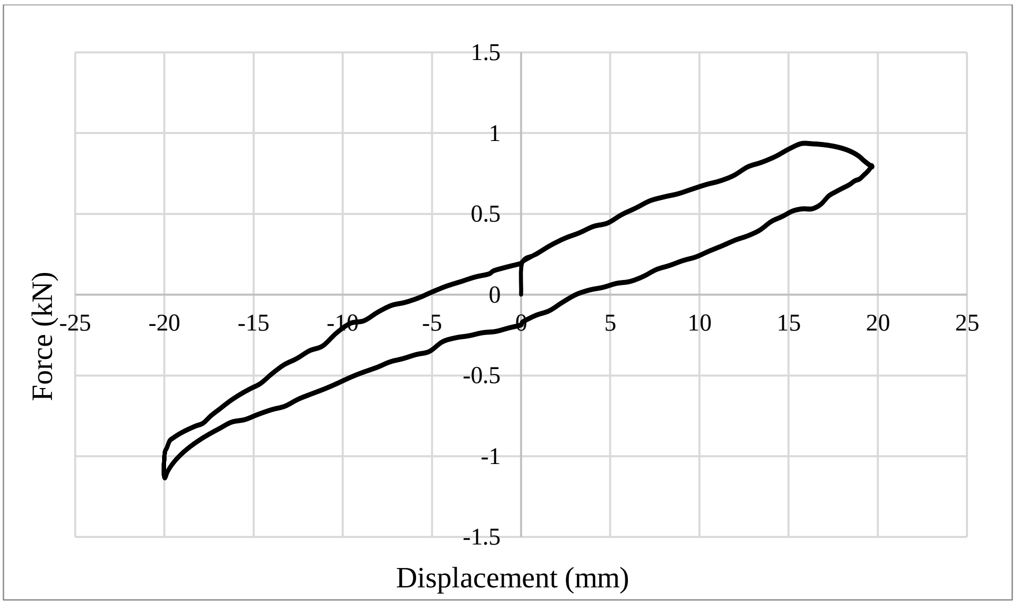

Figure C-18: Displacement- Damping force graph of FFFluid under dynamic load and frequency 1.0Hz for trial 6.

Trial 7	Stiffness (kN/m)	Damping force at middle position 2CV (kN)	Frequency Hz	Damping coefficient $(N.\frac{sec}{m})$
Static test	18.2	-	-	-
Dynamic	18.2	0.16	0.75	1698
Dynamic	18.2	0.21	1	1672

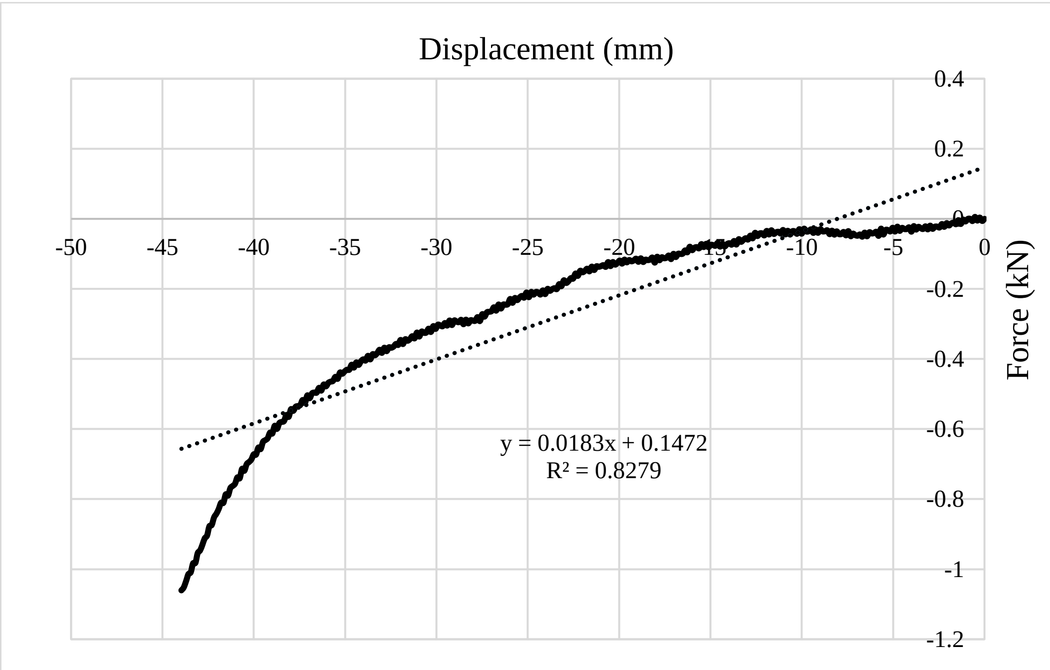

Figure C-19: Displacement- Force graph of FFFluid under static load for trial 7.

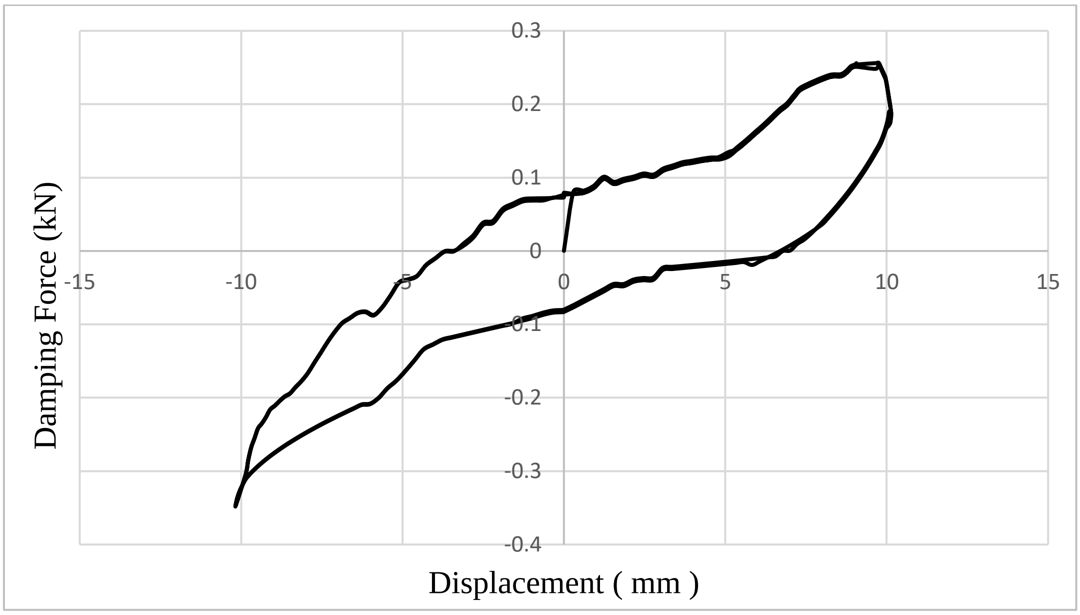

Figure C-20: Displacement- Damping force graph of FFFluid under dynamic load and frequency 0.75Hz for trial 7.

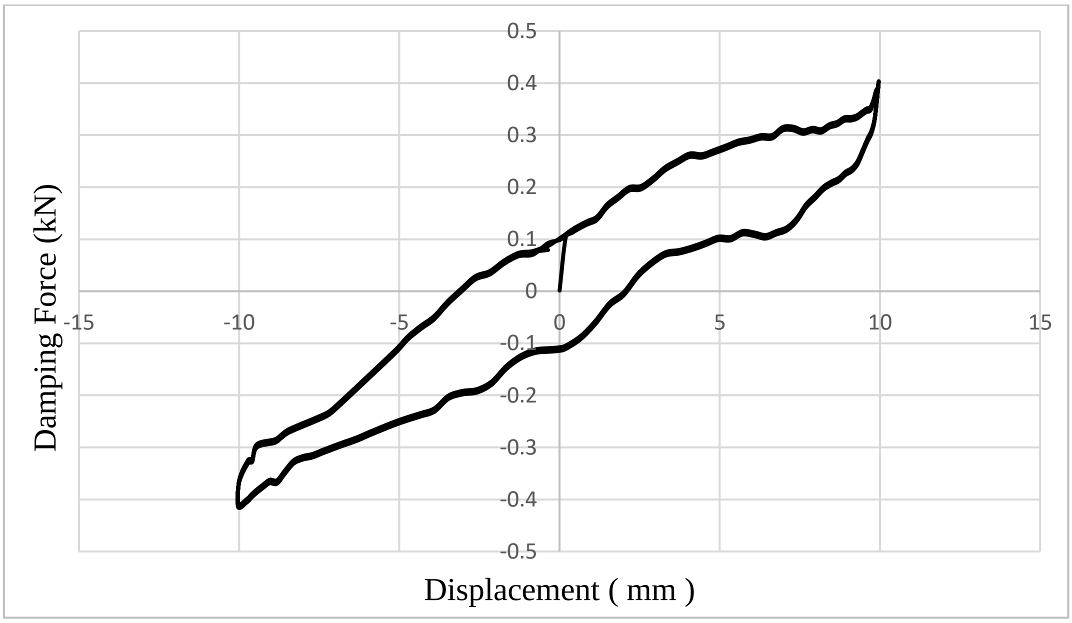

Figure C-21: Displacement - Damping force graph of FFFluid under dynamic load and frequency 1.0Hz for trial 7.

Trial 8	Stiffness (kN/m)	Damping force at middle position 2CV (kN)	Frequency Hz	Damping coefficient (N.$\frac{sec}{m}$)
Static test	22.8	-	-	-
Dynamic	22.8	0.083	0.75	1762
Dynamic	22.8	0.11	1	1752

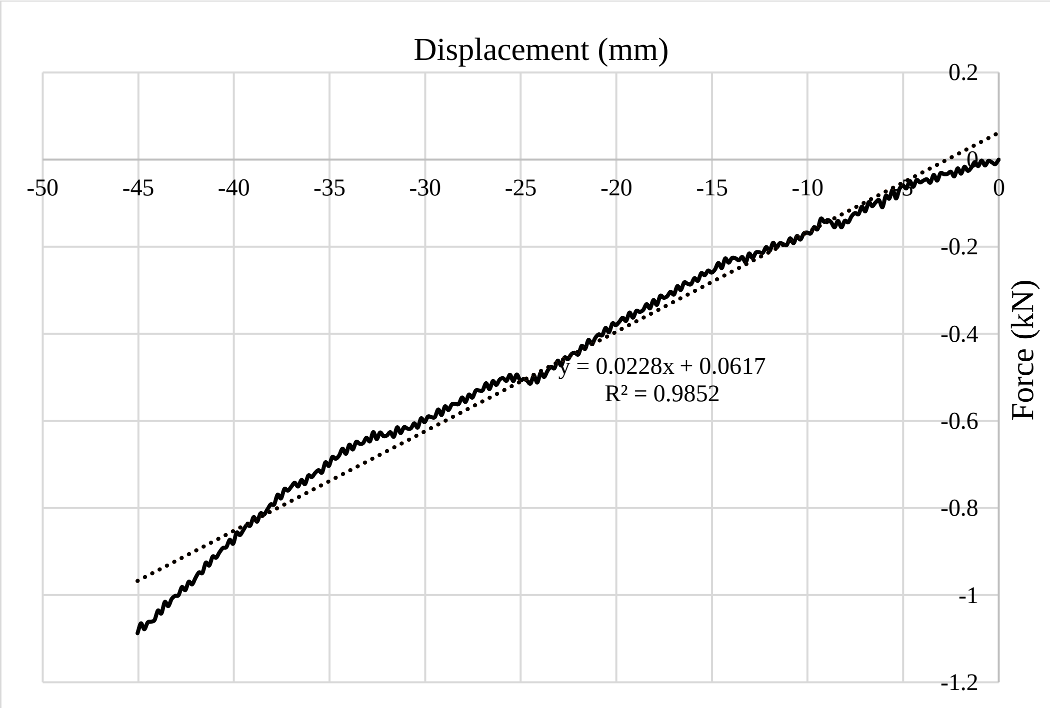

Figure C-22: Displacement- Force graph of FFFluid under static load for trial 8.

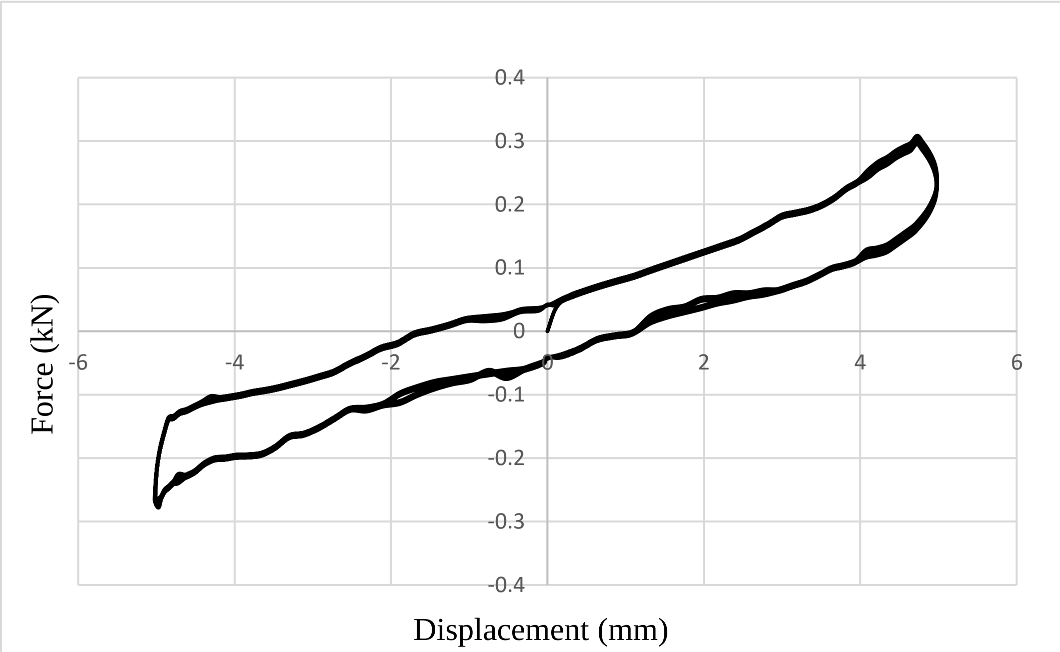

Figure C-23: Displacement- Damping force graph of FFFluid under dynamic load and frequency 0.75Hz for trial 8.

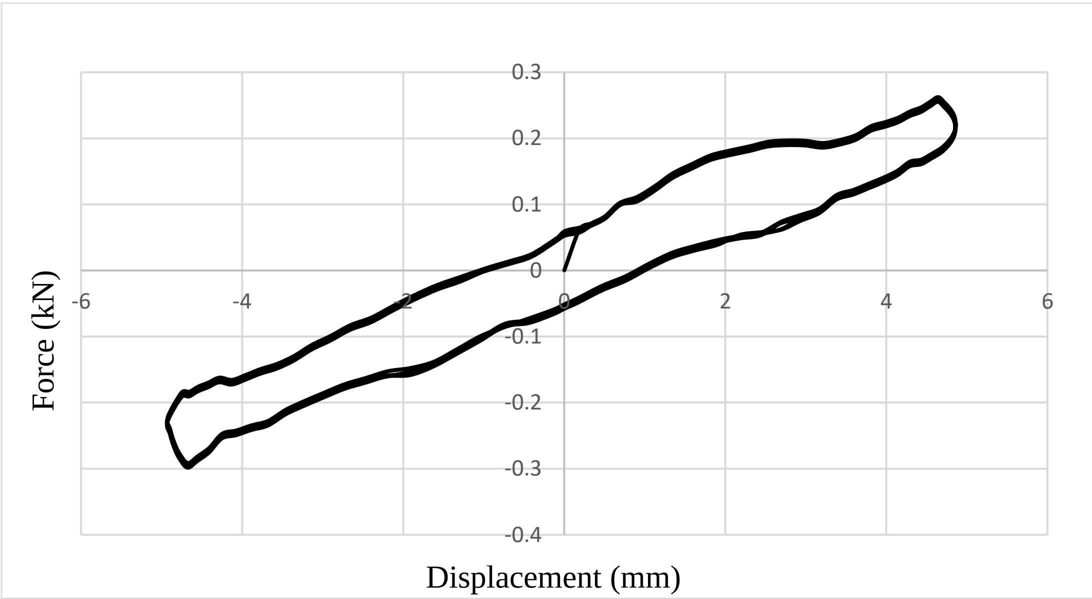

Figure C-24: Displacement- Damping force graph of FFFluid under dynamic load and frequency 1.0Hz for trial 8.

Trial 9.	Stiffness (kN/m)	Damping force at middle position 2CV (kN)	Frequency Hz	Damping coefficient ($N.\frac{sec}{m}$)
Static test	31.2	-	-	-
Dynamic	31.2	0.095	0.75	1010
Dynamic	31.2	0.13	1	1035

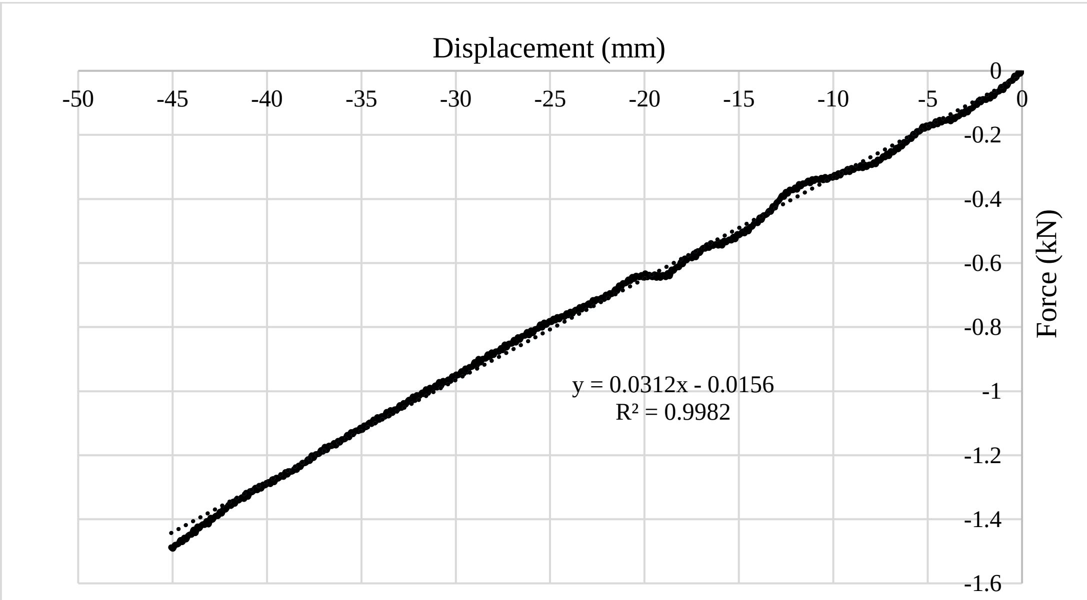

Figure C-25: Displacement- Force graph of FFFluid under static load for trial 9.

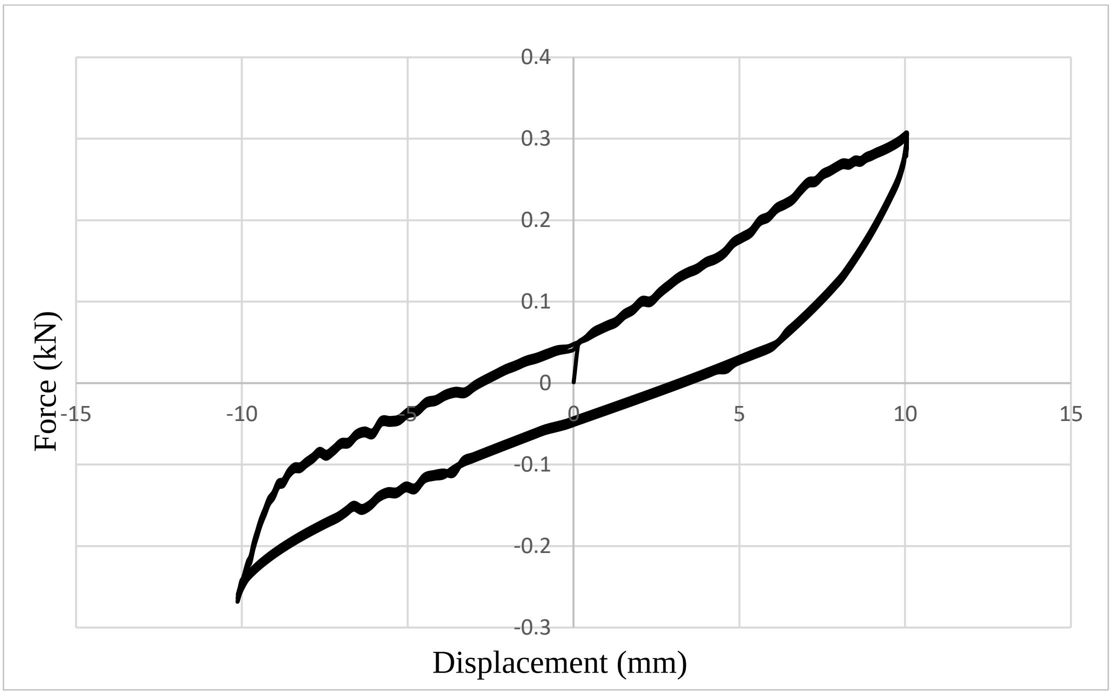

Figure C-26: Displacement- viscous force graph of FFFluid under dynamic load and frequency 0.75Hz for trial 9.

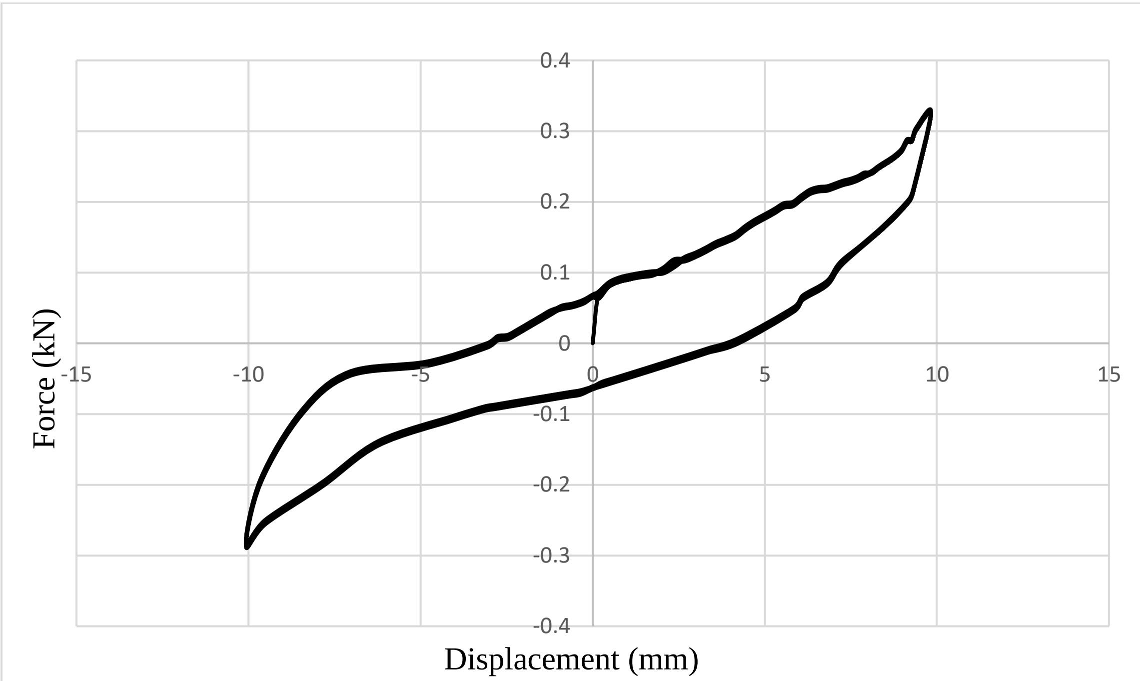

Figure C-27: Displacement- Damping force graph of FFFluid under dynamic load and frequency 1.0Hz for trial 9.

Trial 10.	Stiffness (kN/m)	Damping force at middle position 2CV (kN)	Frequency Hz	Damping coefficient $(N.\frac{sec}{m})$
Static test	27.4	-	-	-
Dynamic	27.4	0.06	0.75	1273
Dynamic	27.4	0.08	1	1273

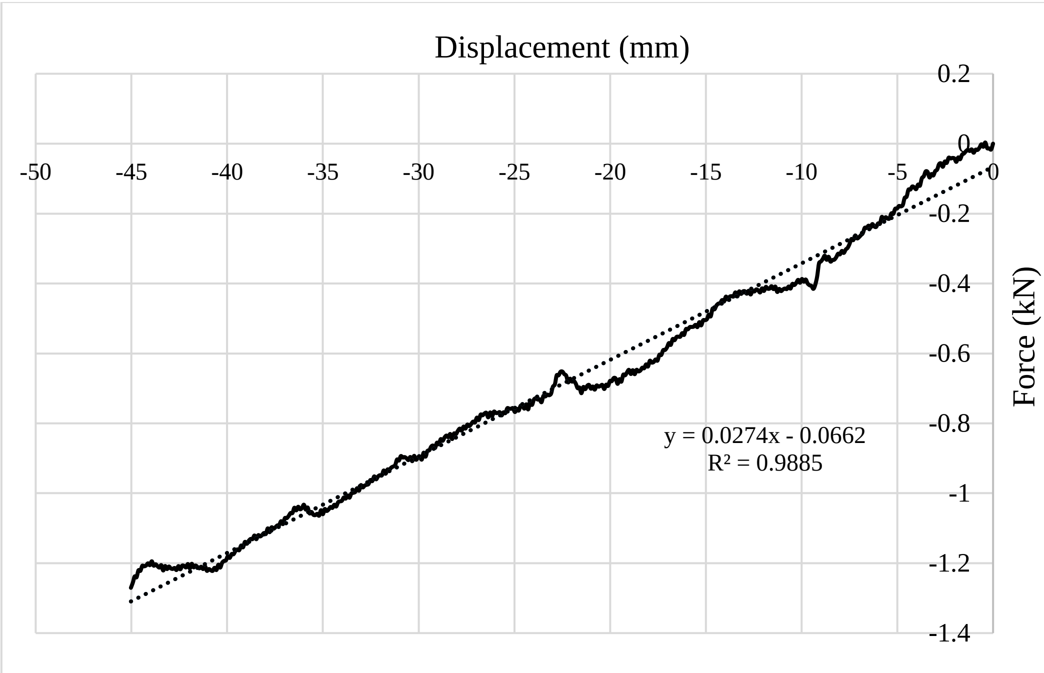

Figure C-28: Displacement- Force graph of FFFluid under static load for trial 10.

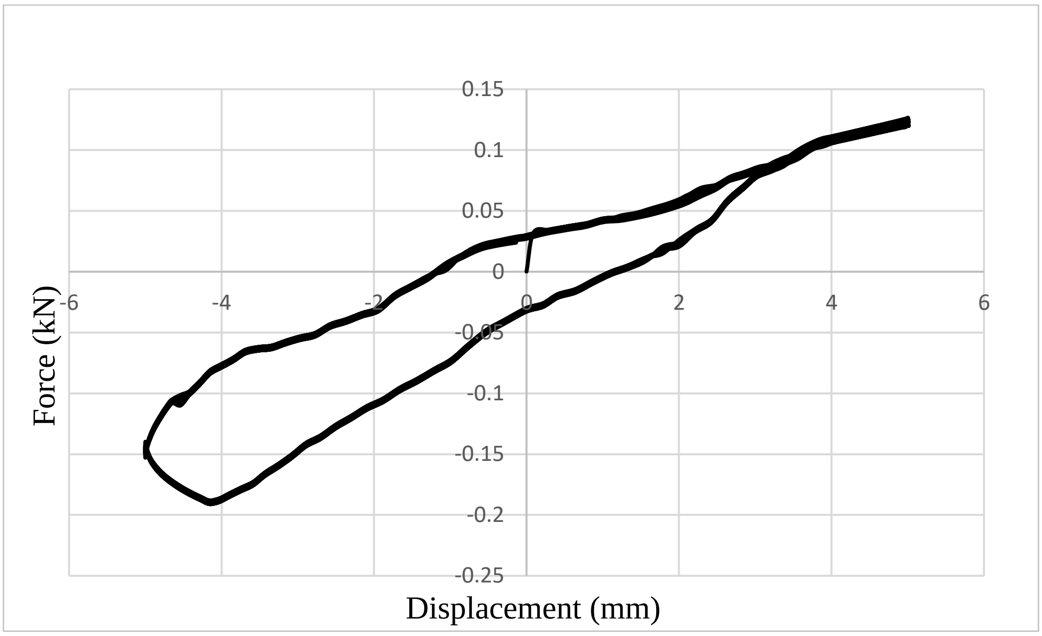

Figure C-29: Displacement- Damping force graph of FFFluid under dynamic load and frequency 0.75Hz for trial 10.

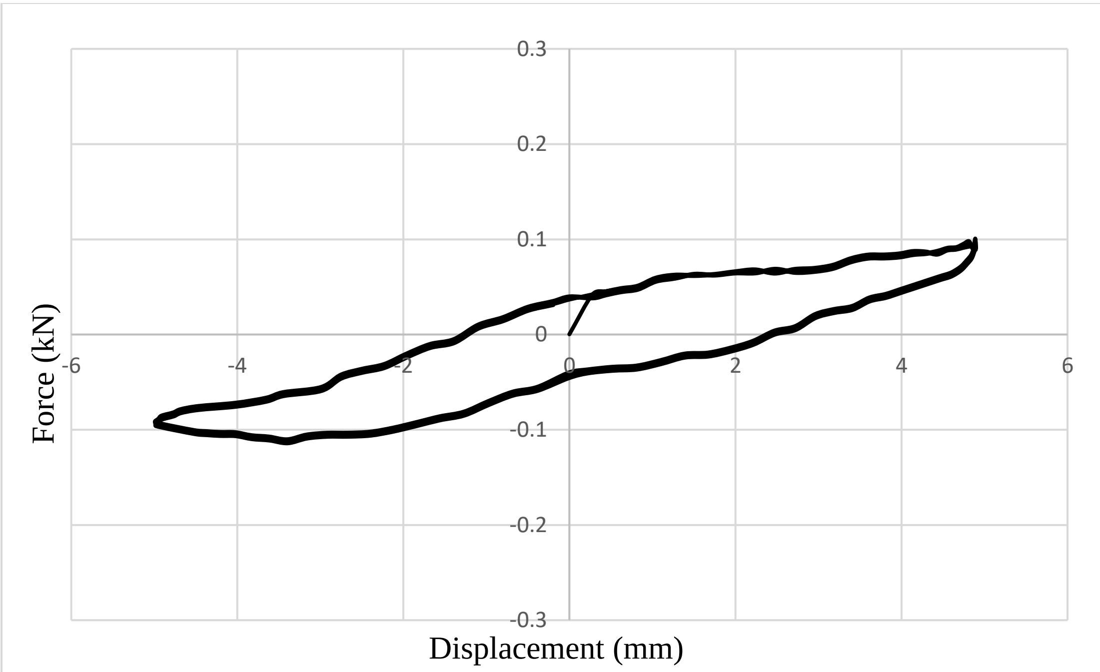

Figure C-30: Displacement- Damping force graph of FFFluid under dynamic load and frequency 1.0Hz for trial 10.

Trial 11	Stiffness (kN/m)	Damping force at middle position 2CV (kN)	Frequency Hz	Damping coefficient ($N.\frac{sec}{m}$)
Static test	23.6	-	-	-
Dynamic	23.6	0.26	0.75	920
Dynamic	23.6	0.35	1	929

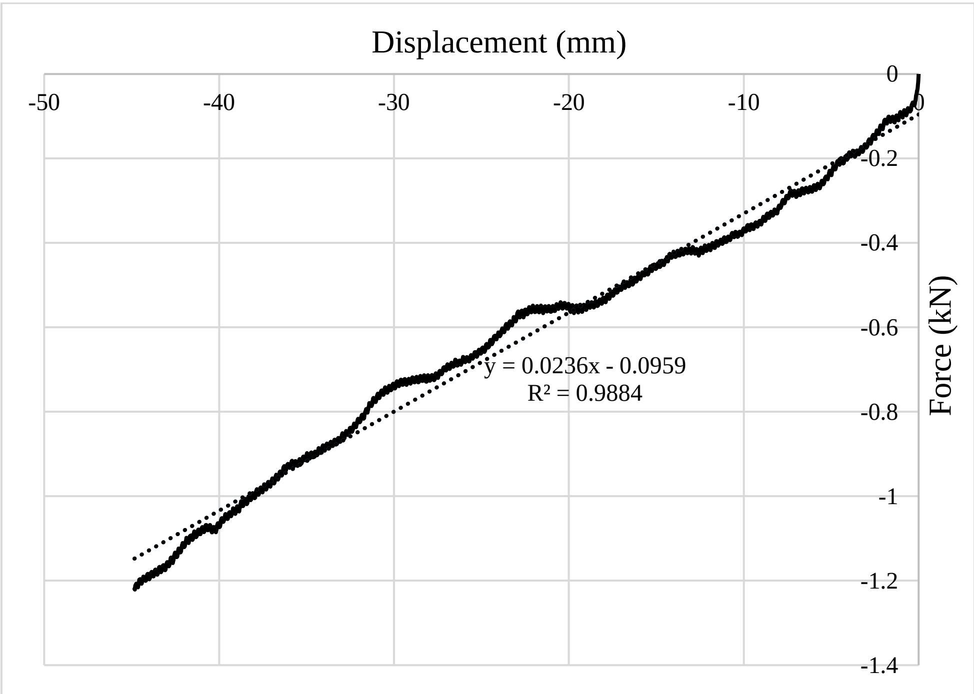

Figure C-31: Displacement- Force graph of FFFluid under static load for trial 11.

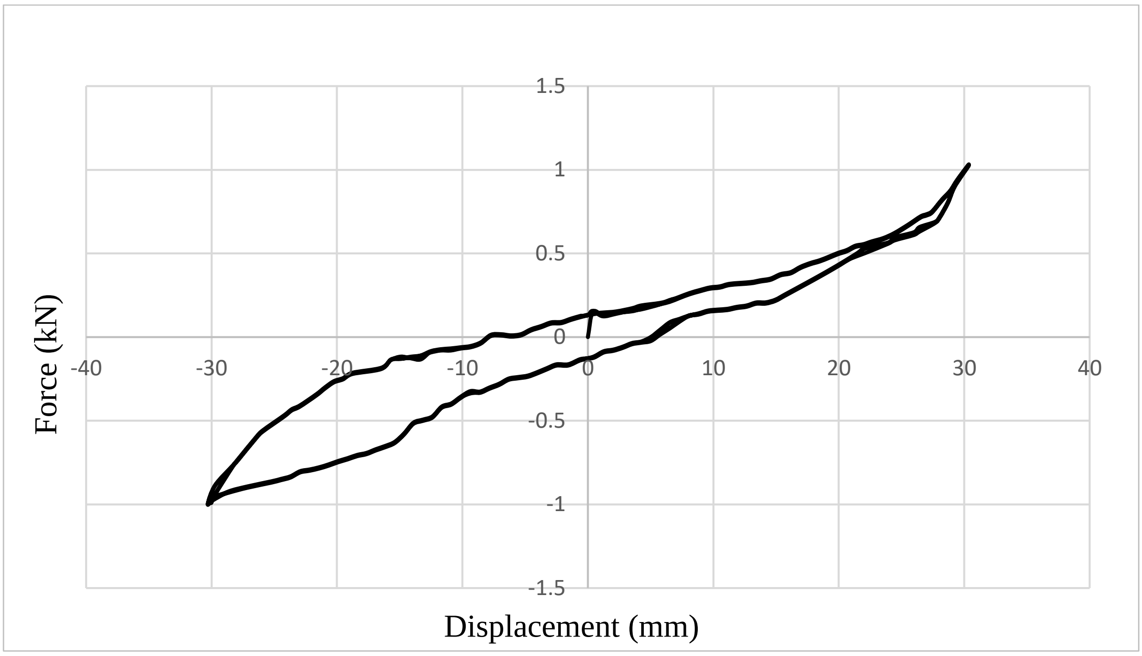

Figure C-32: Displacement- Damping force graph of FFFluid under dynamic load and frequency 0.75Hz for trial 11.

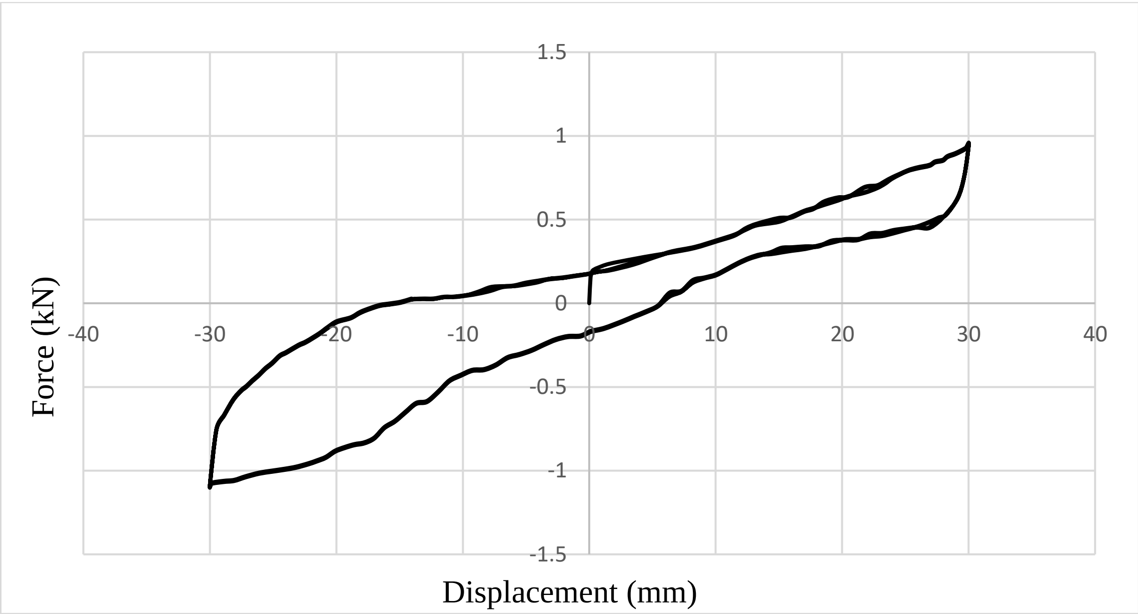

Figure C-33: Displacement- Damping force graph of FFFluid under dynamic load and frequency 1.0Hz for trial 11.

Trial 12	Stiffness (kN/m)	Damping force at middle position 2CV (kN)	Frequency Hz	Damping coefficient $(\mathrm{N.}\frac{\mathrm{sec}}{\mathrm{m}})$
Static test	24	-	-	-
Dynamic	24	0.135	0.75	714
Dynamic	24	0.18	1	716

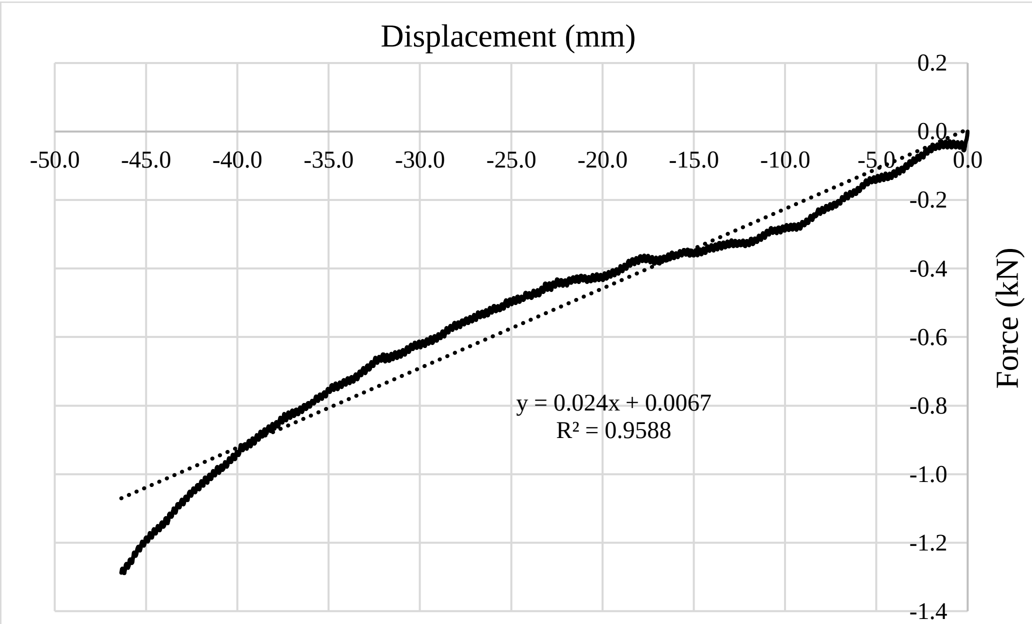

Figure C-34: Displacement- Force graph of FFFluid under static load for trial 12.

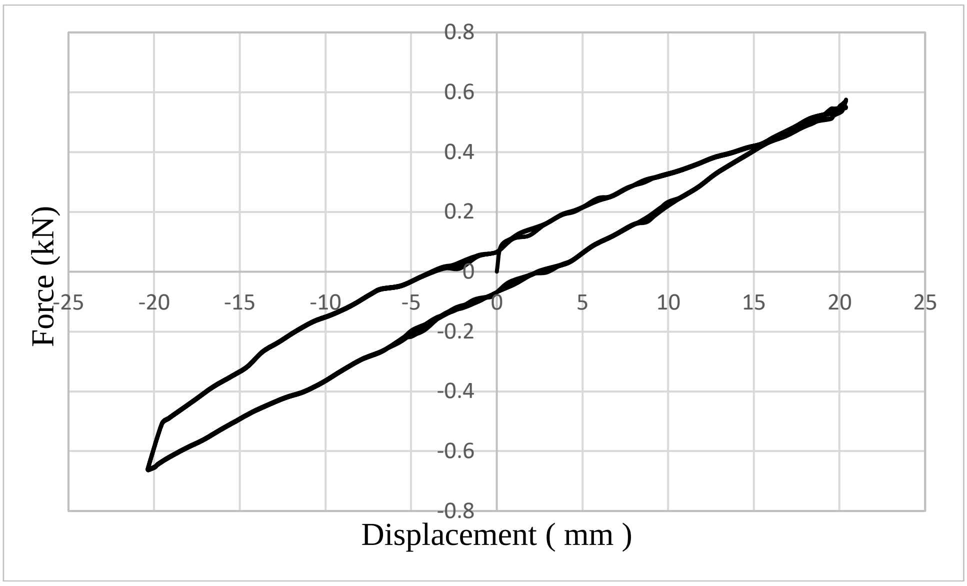

Figure C-35: Displacement - Damping force graph of FFFluid under dynamic load and frequency 0.75Hz for trial 12.

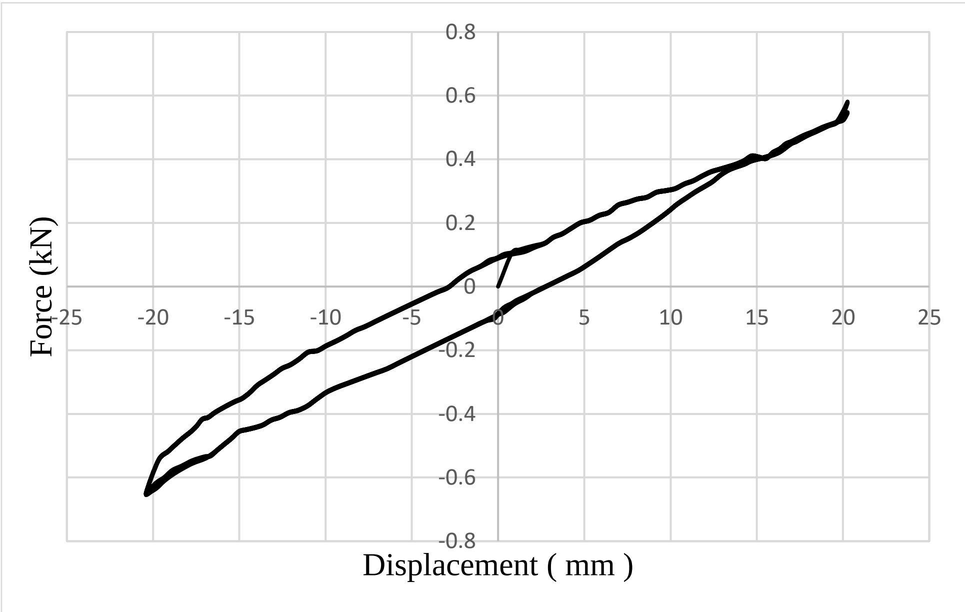

Figure C-36: Displacement- Damping force graph of FFFluid under dynamic load and frequency 1.0Hz for trial 12.

Trial 13	Stiffness (kN/m)	Damping force at middle position 2CV (kN)	Frequency Hz	Damping coefficient (N.$\frac{sec}{m}$)
Static test	18.7	-	-	-
Dynamic	18.7	0.23	0.75	1220
Dynamic	18.7	0.31	1	1234

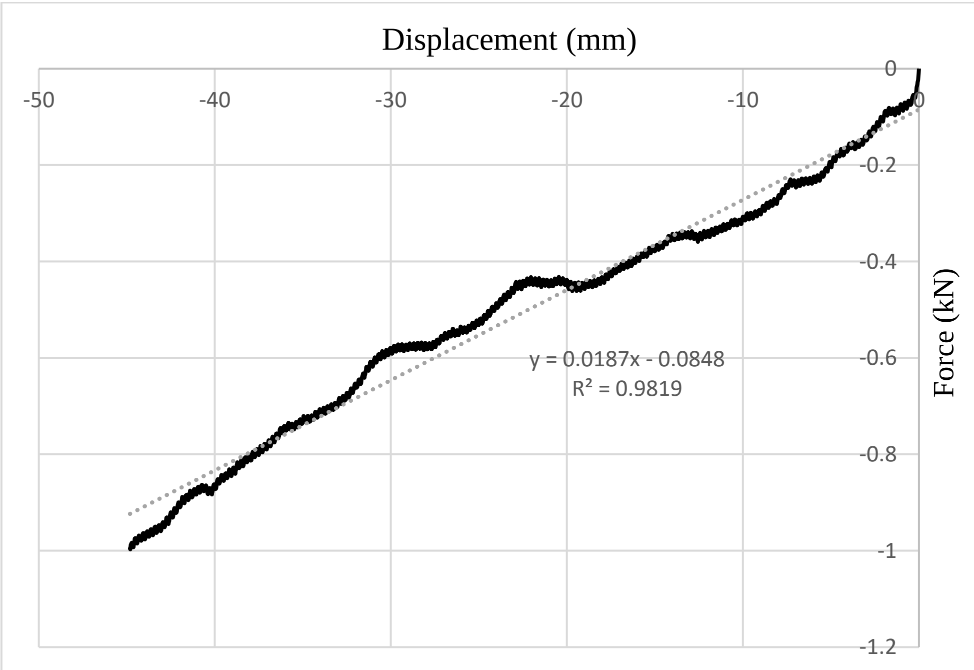

Figure C-37: Displacement- Force graph of FFFluid under static load for trial 13.

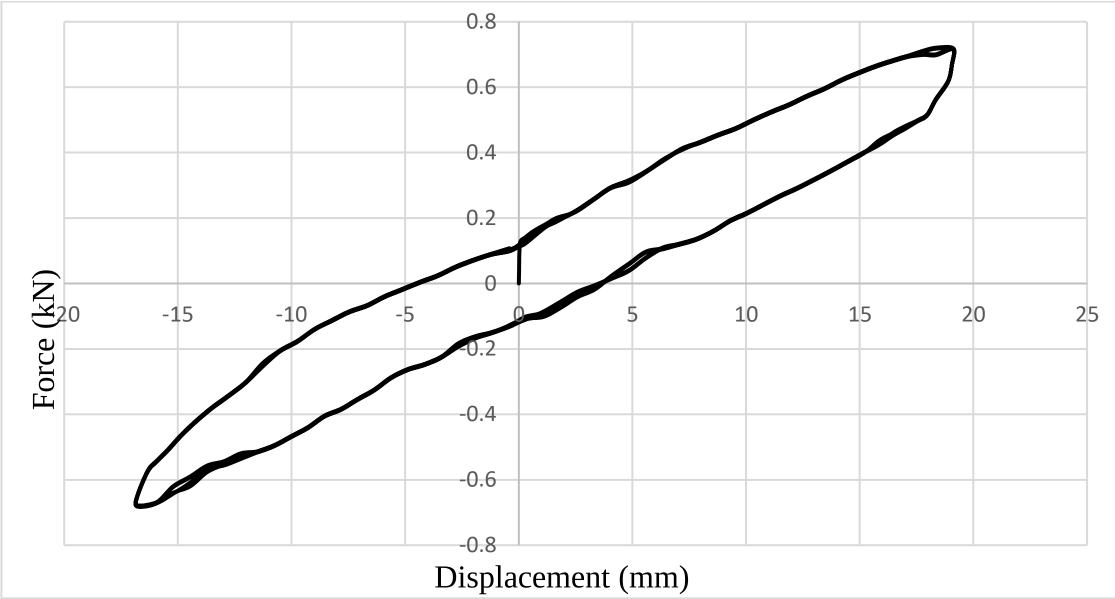

Figure C-38: Displacement- Damping force graph of FFFluid under dynamic load and frequency 0.75Hz for trial 13.

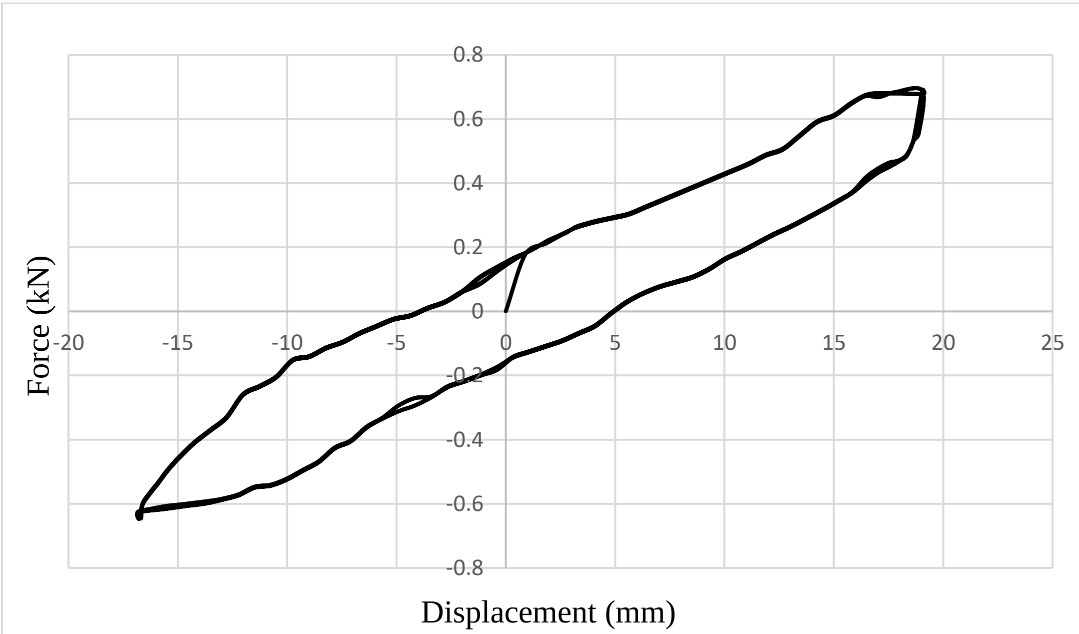

Figure C-39: Displacement- Damping force graph of FFFluid under dynamic load and frequency 1.0Hz for trial 13.

Trial 14	Stiffness (kN/m)	Damping force at middle position 2CV (kN)	Frequency Hz	Damping coefficient $(N.\frac{sec}{m})$
Static test	21.0	-	-	-
Dynamic	21.0	0.26	0.75	920
Dynamic	21.0	0.35	1	928

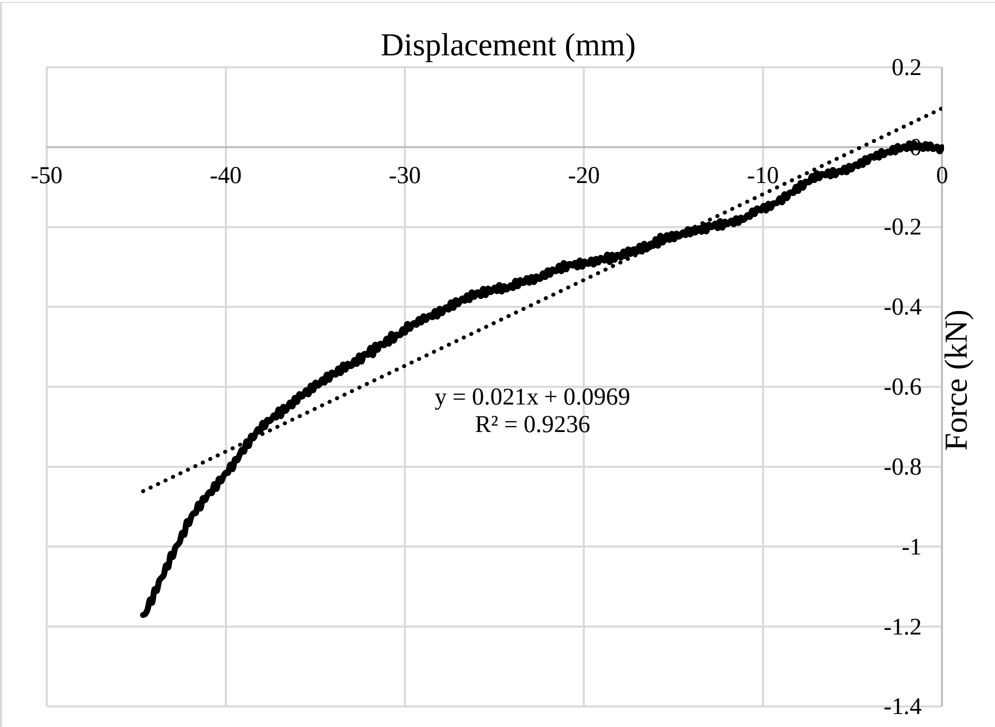

Figure C-40: Displacement- Force graph of FFFluid under static load for trial 14.

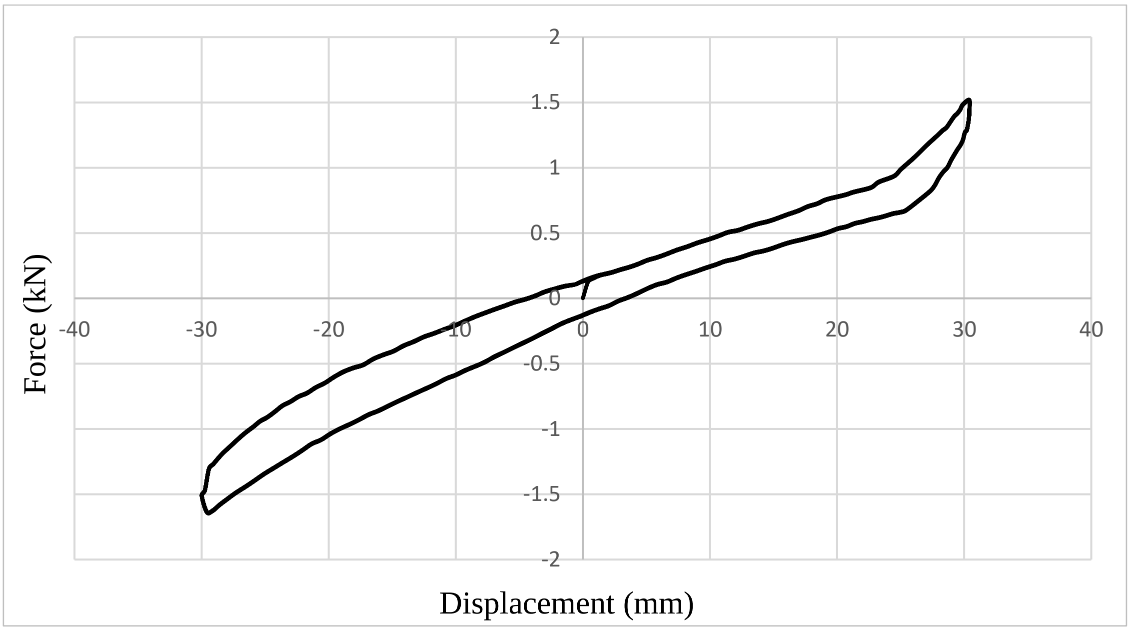

Figure C-41: Displacement- Damping force graph of FFFluid under dynamic load and frequency 0.75Hz for trial 14.

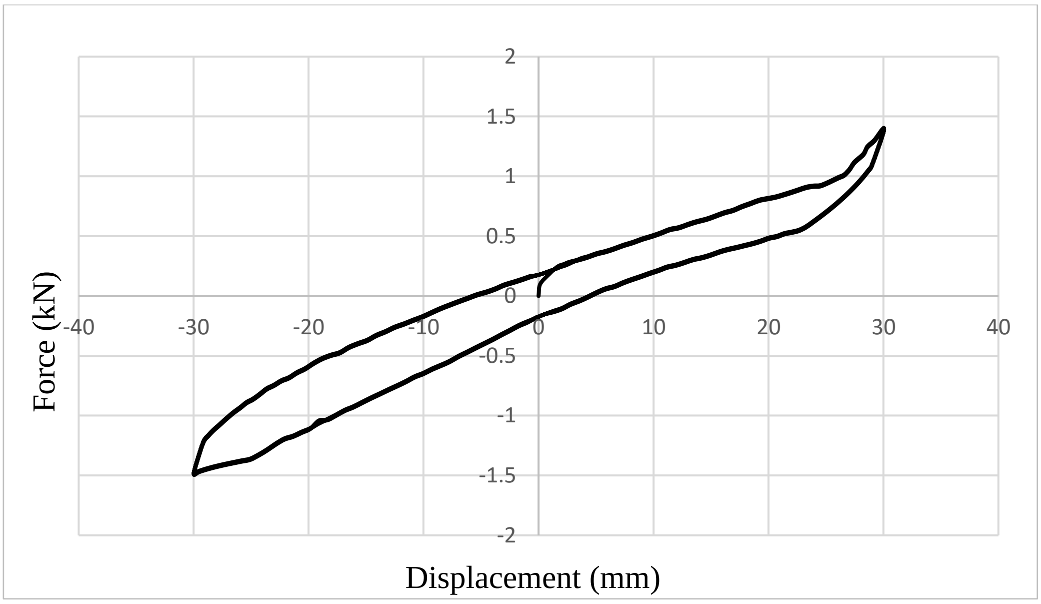

Figure C-42: Displacement- Damping force graph of FFFluid under dynamic load and frequency 1.0Hz for trial 14.

Trial 15	Stiffness (kN/m)	Damping force at middle position 2CV (kN)	Frequency Hz	Damping coefficient $(\mathrm{N}.\frac{\mathrm{sec}}{\mathrm{m}})$
Static test	26.8	-	-	-
Dynamic	26.8	0.07	0.75	1480
Dynamic	26.8	0.09	1	1440

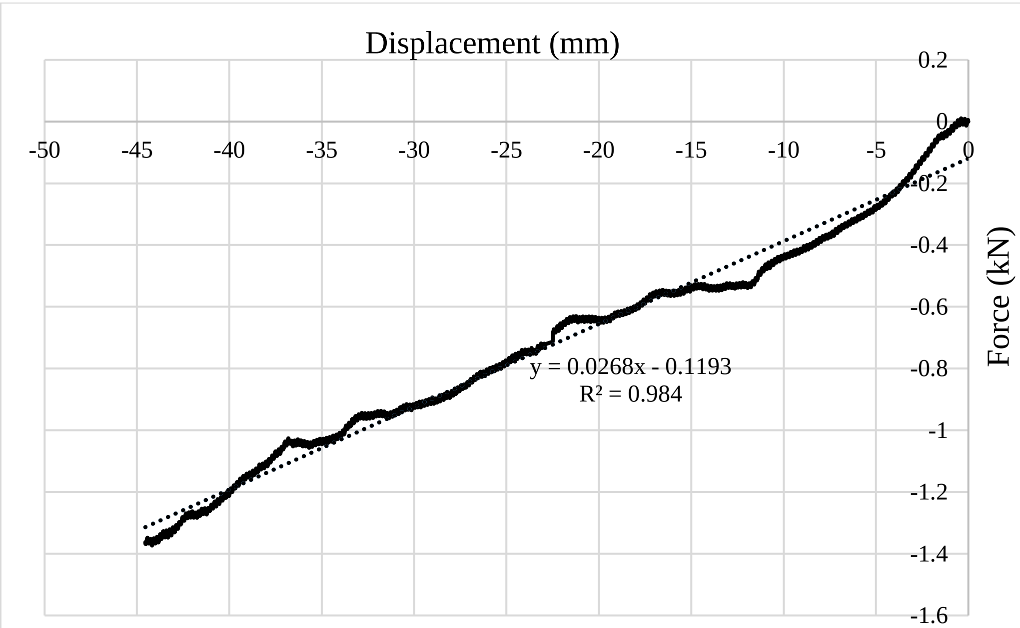

Figure C-43: Displacement- Force graph of FFFluid under static load for trial 15.

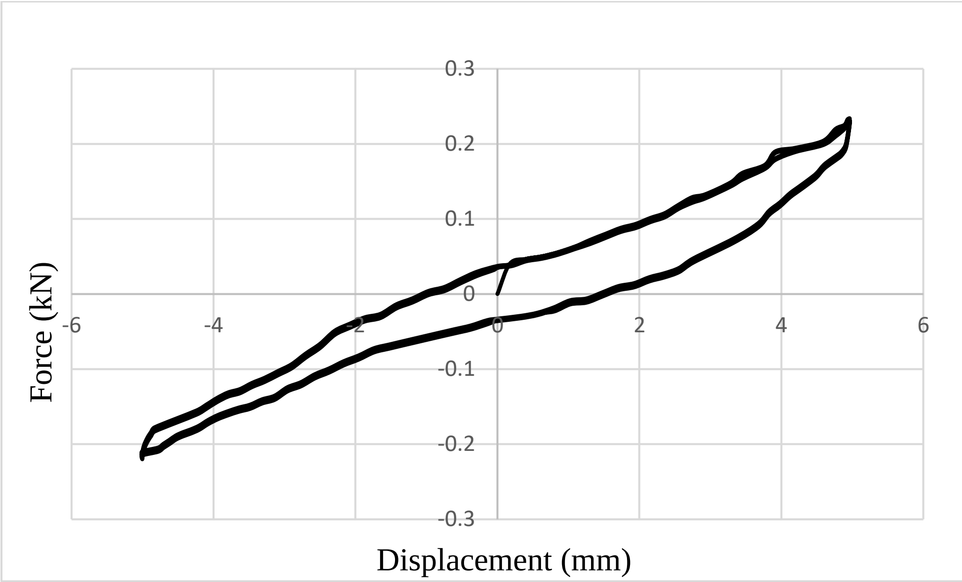

Figure C-44: Displacement- Damping force graph of FFFluid under dynamic load and frequency 0.75Hz for trial 15.

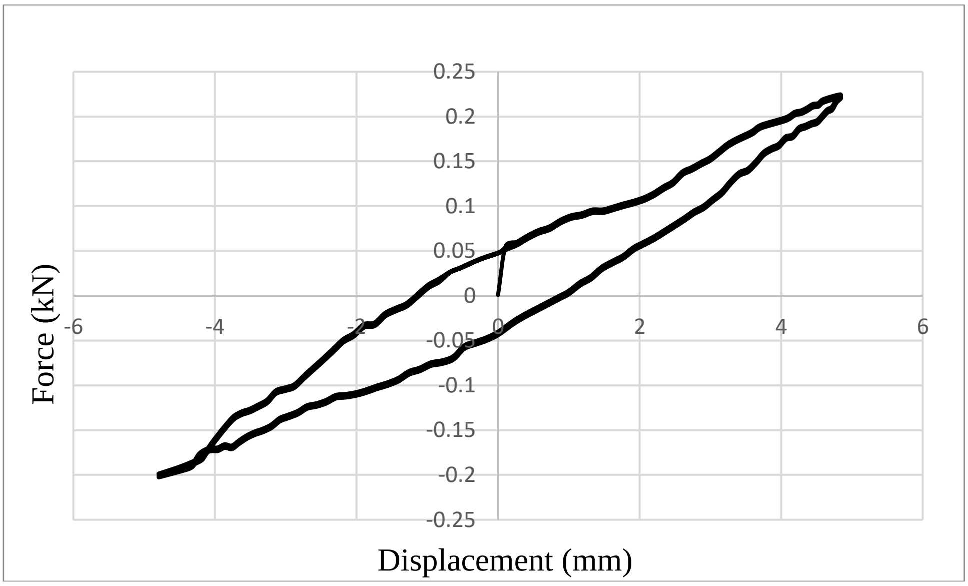

Figure C-45: Displacement- Damping force graph of FFFluid under dynamic load and frequency 1.0Hz for trial 15.

Trial 16	Stiffness (kN/m)	Damping force at middle position 2CV (kN)	Frequency Hz	Damping coefficient $(N.\frac{sec}{m})$
Static test	26.0	-	-	-
Dynamic	26.0	0.11	0.75	1157
Dynamic	26.0	0.144	1	1167

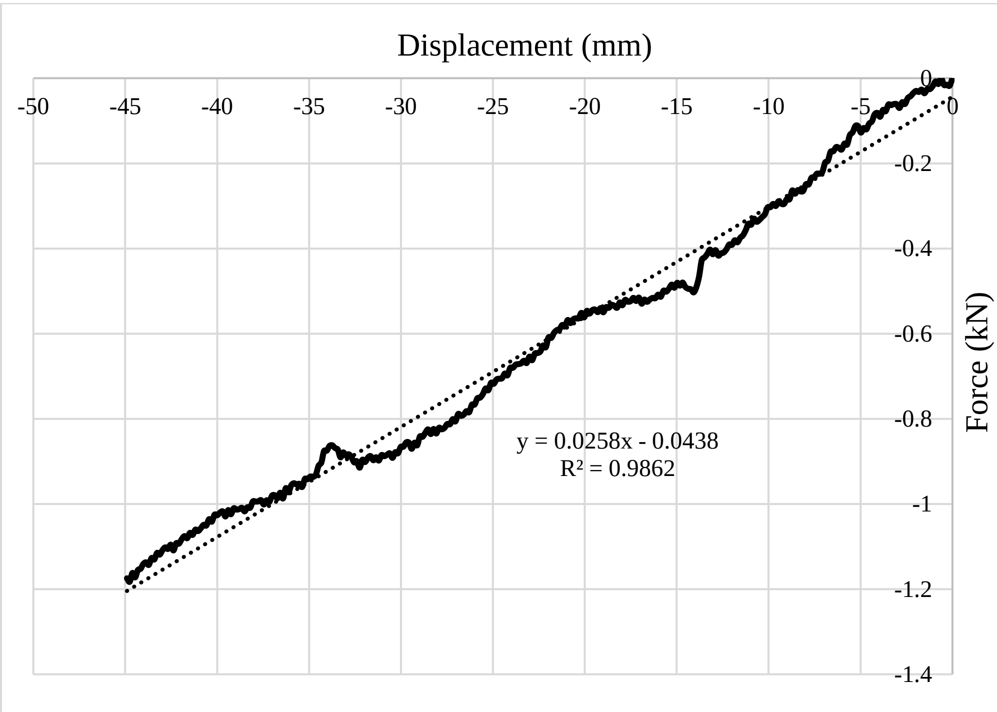

Figure C-46: Displacement- Force graph of FFFluid under static load for trial 16.

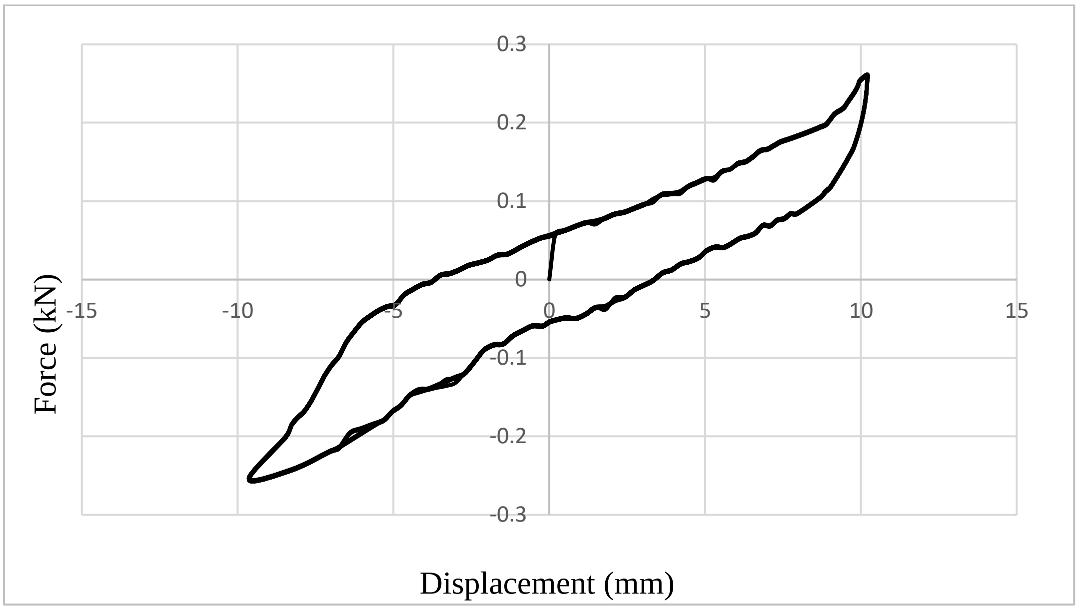

Figure C-47: Displacement- Damping force graph of FFFluid under dynamic load and frequency 0.75Hz for trial 16.

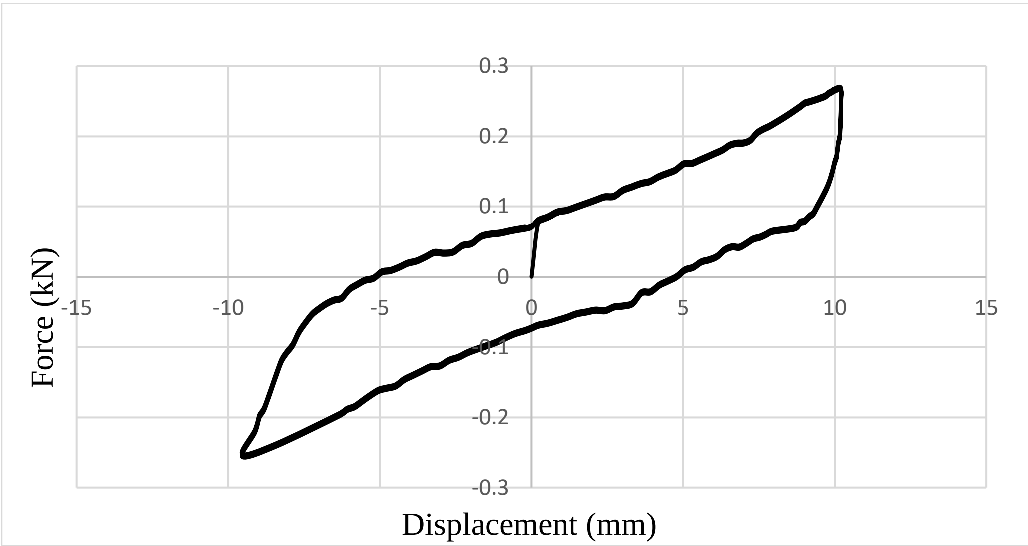

Figure C-48: Displacement- Damping force graph of FFFluid under dynamic load and frequency 1.0Hz for trial 16.

APPENDIX- D: Result of set 2

Trial 1	Stiffness (kN/m)	Damping force at middle position 2CV (kN)	Frequency Hz	Damping coefficient (N.$\frac{\text{sec}}{\text{m}}$)
Static test	18.5	-	-	-
Dynamic test	18.5	0.12	1	955

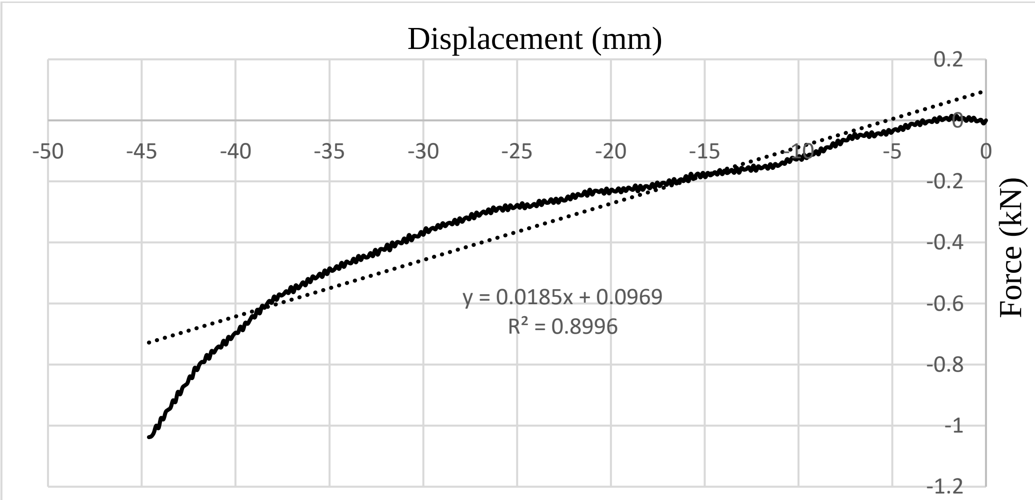

Figure D-1: Displacement- Force graph of FFFluid under static load for trial 1.

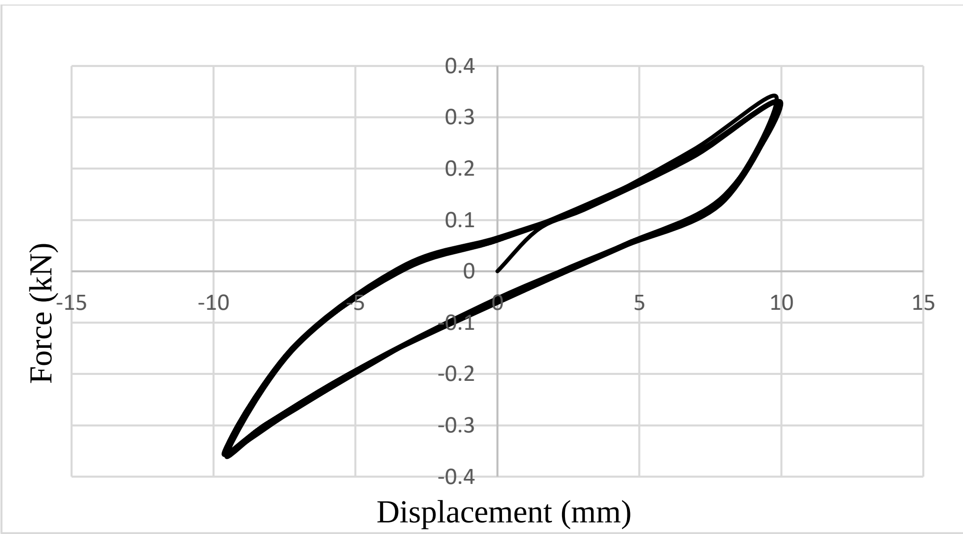

Figure 0-2: Displacement- Damping force graph of FFFluid for trial 1.

Trial 2	Stiffness (kN/m)	Damping force at middle position 2CV (kN)	Frequency Hz	Damping coefficient $(\mathrm{N}.\frac{\mathrm{sec}}{\mathrm{m}})$
Static test	22	-	-	-
Dynamic	22	0.155	1	1234

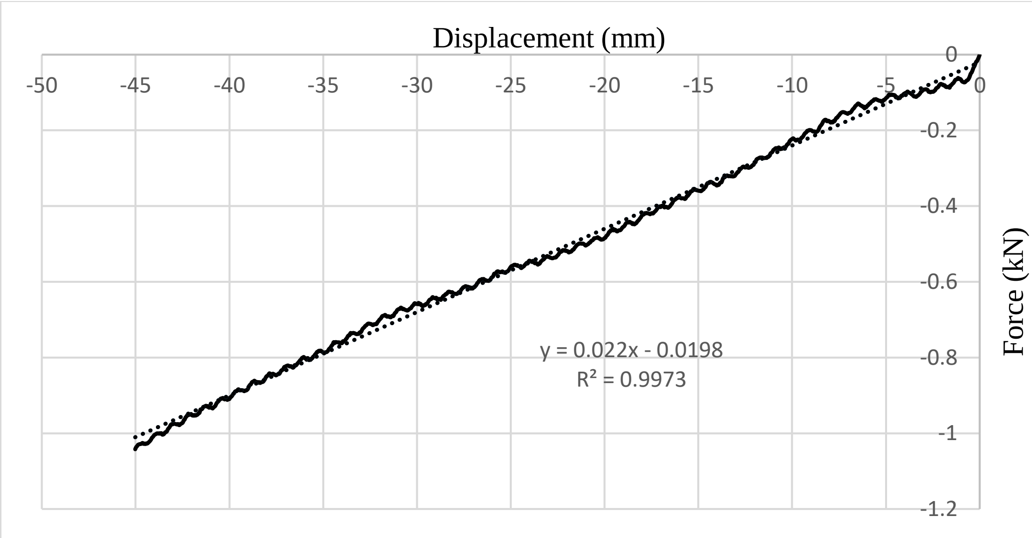

Figure D-3: Displacement- Force graph of FFFluid under static load for trial 2.

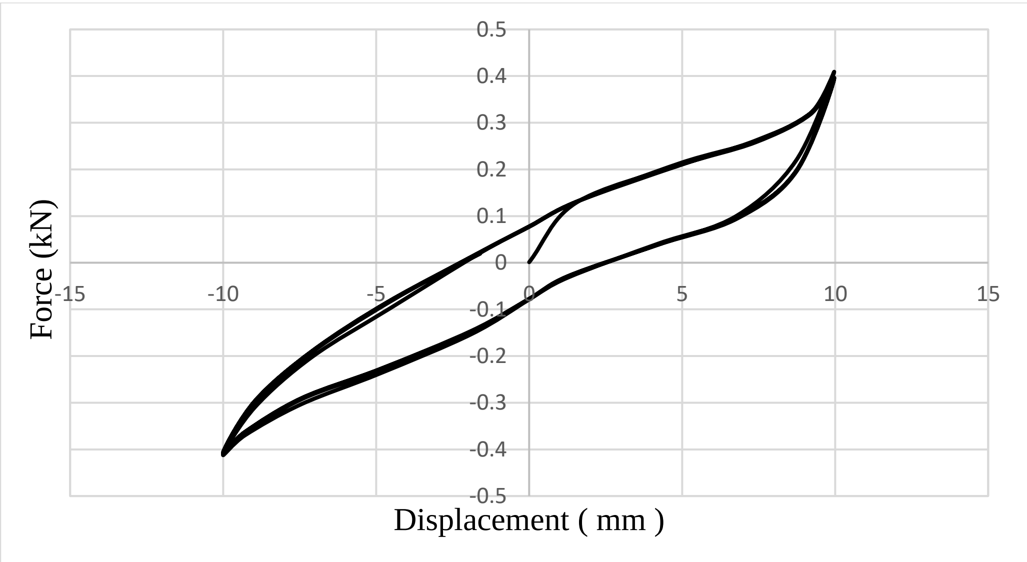

Figure D-4: Displacement - Damping force graph of FFFluid for trial 2.

Trial 3	Stiffness (kN/m)	Damping force at middle position 2CV (kN)	Frequency Hz	Damping coefficient (N. $\frac{sec}{m}$)
Static test	24.7	-	-	-
Dynamic	24.7	0.174	1	1385

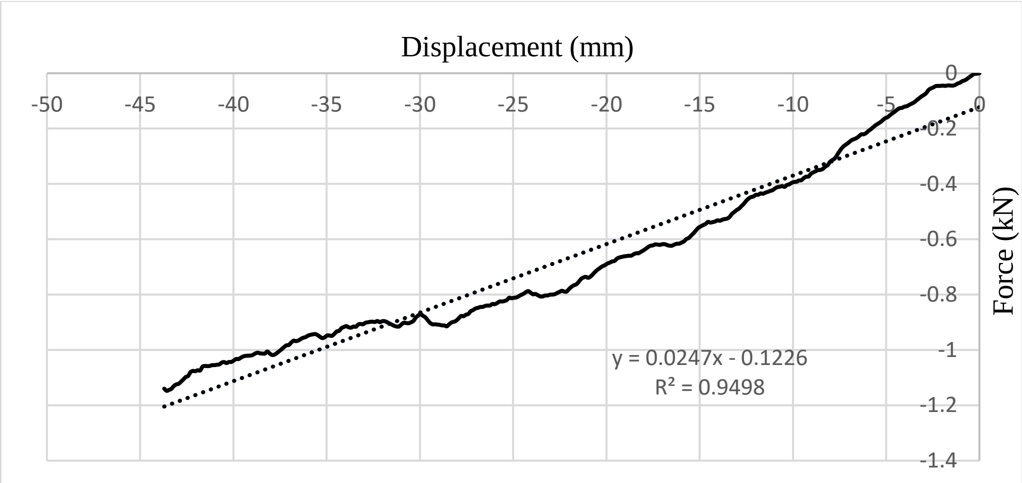

Figure D-5: Displacement- Force graph of FFFluid under static load for trial 3.

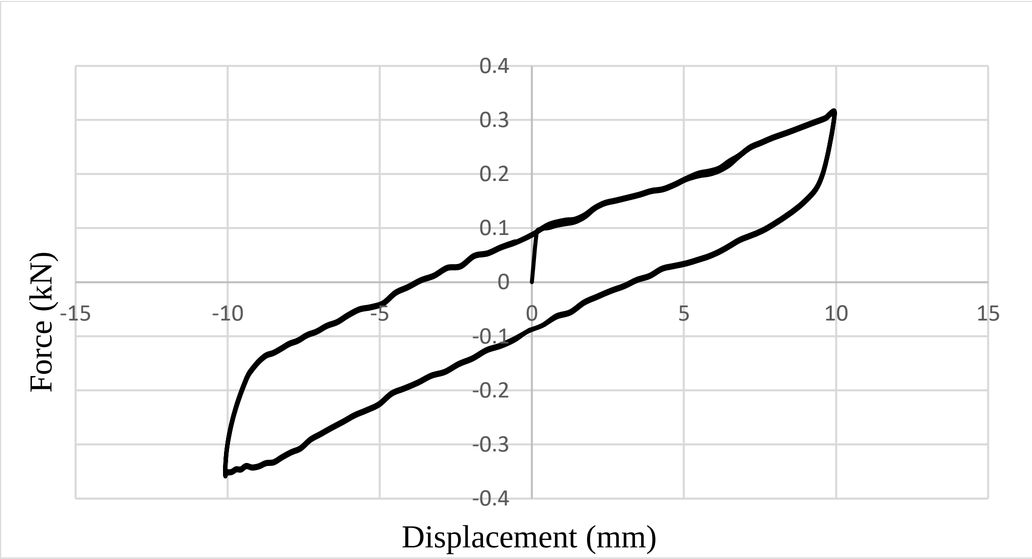

Figure D-6: Displacement - Damping force graph of FFFluid for trial 3.

Trial 4	Stiffness (kN/m)	Damping force at middle position 2CV (kN)	Frequency Hz	Damping coefficient $(N.\frac{sec}{m})$
Static test	31	-	-	-
Dynamic	31	0.178	1	1417

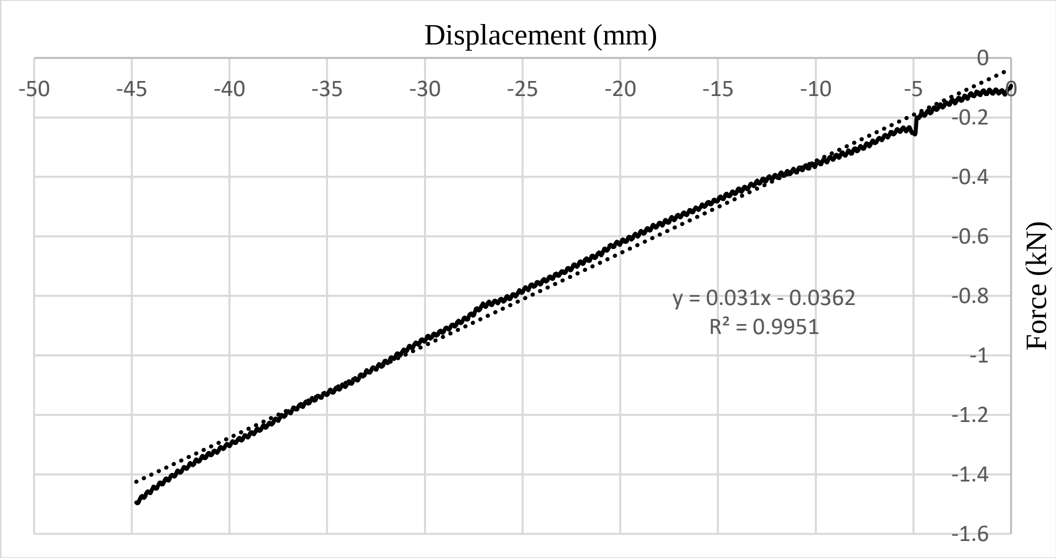

Figure D-7: Displacement- Force graph of FFFluid under static load for trial 4.

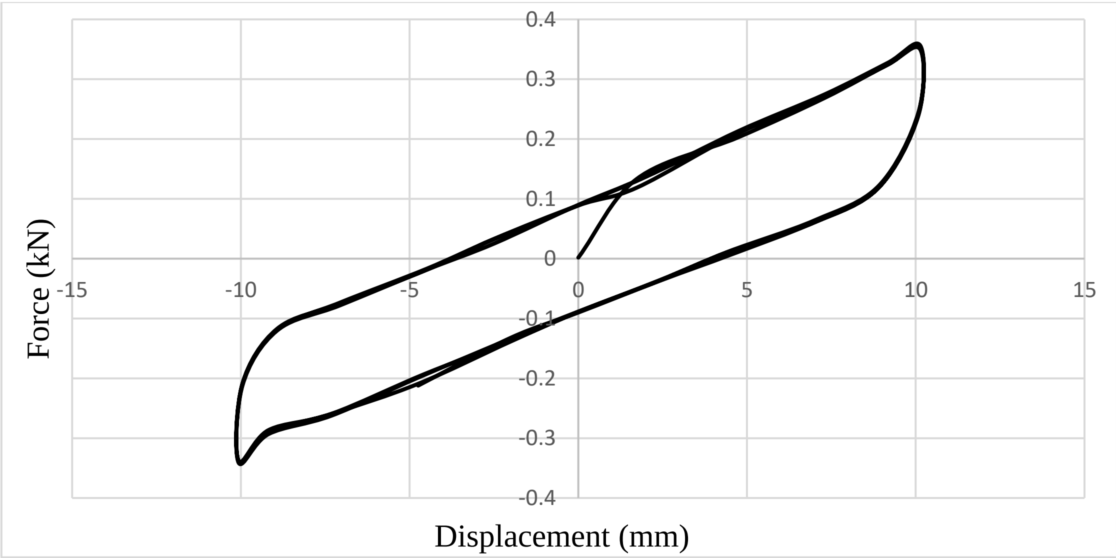

Figure D-8: Displacement - Damping force graph of FFFluid for trial 4.

Trial 5	Stiffness (kN/m)	Damping force at middle position 2CV (kN)	Frequency Hz	Damping coefficient (N.$\frac{sec}{m}$)
Static test	19	-	-	-
Dynamic	19	0.168	1	1338

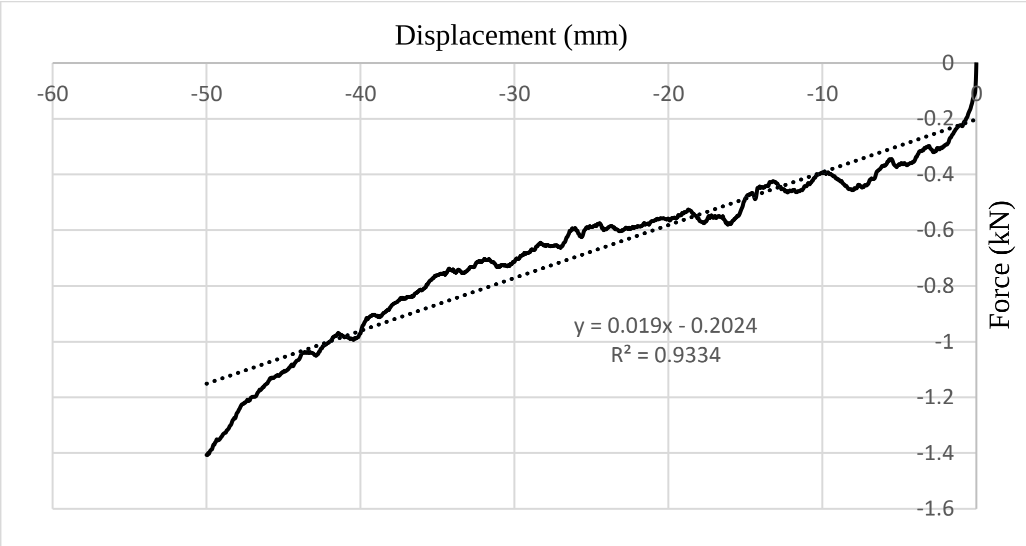

Figure D-9: Displacement- Force graph of FFFluid under static load for trial 5.

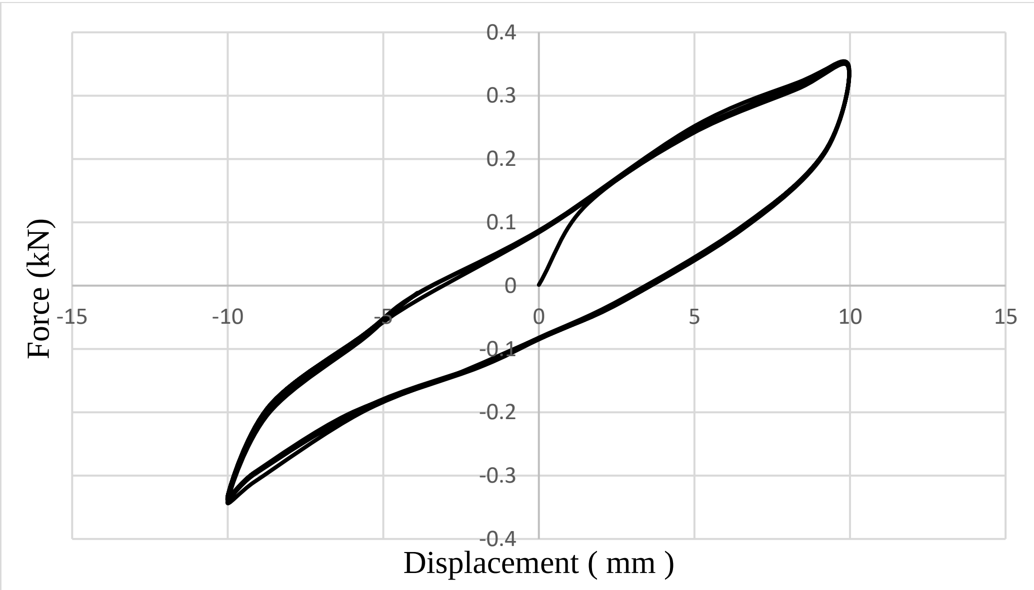

Figure D-10: Displacement - Damping force graph of FFFluid for trial 5.

Trial 6	Stiffness (kN/m)	Damping force at middle position 2CV (kN)	Frequency Hz	Damping coefficient $(N.\frac{sec}{m})$
Static test	20.3	-	-	-
Dynamic	20.3	0.169	1	1353

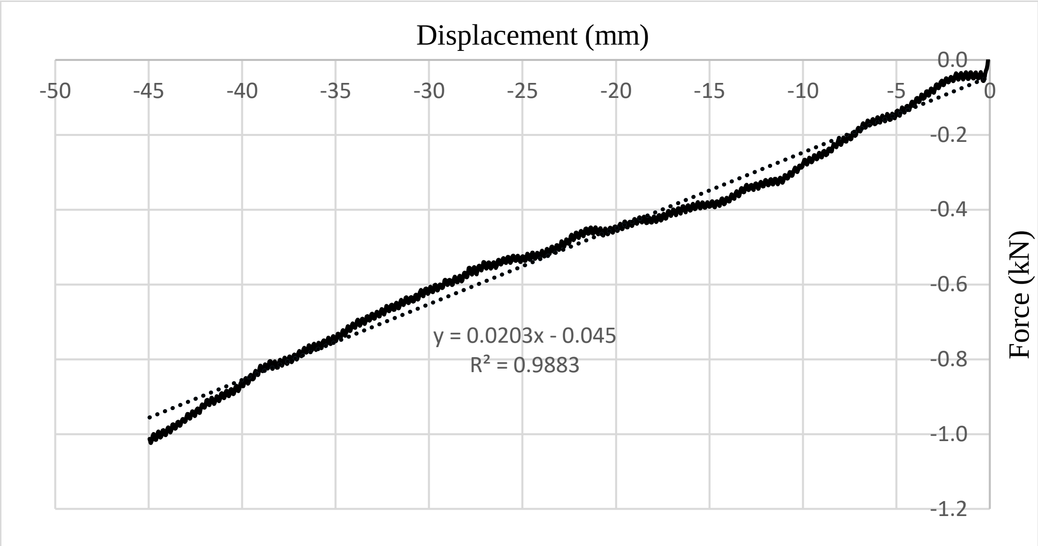

Figure D-11: Displacement- Force graph of FFFluid under static load for trial 6.

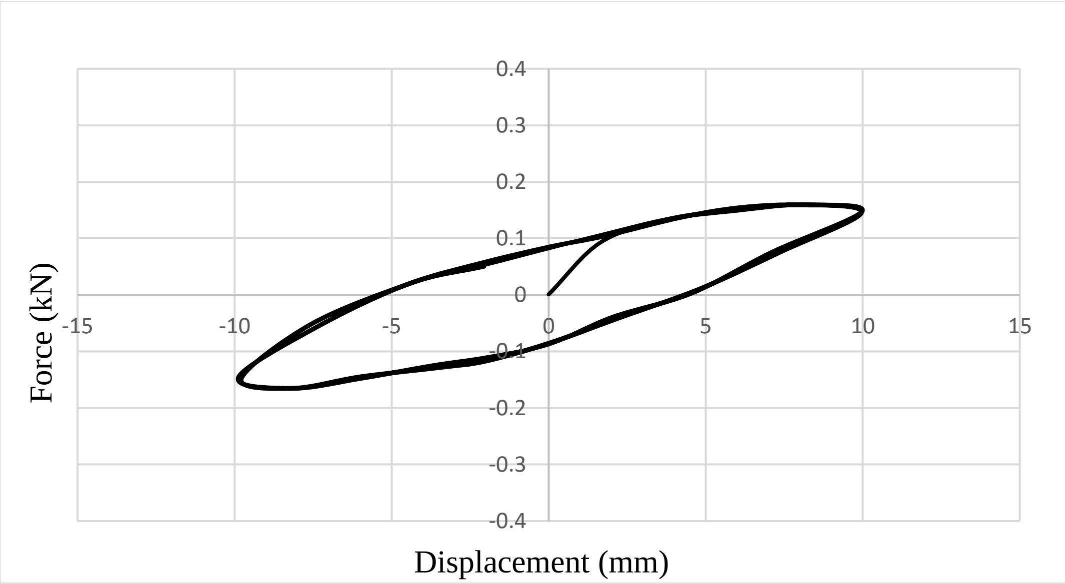

Figure D-12: Displacement- Damping force graph of FFFluid for trial 6.

Trial 7	Stiffness (kN/m)	Damping force at middle position 2CV (kN)	Frequency Hz	Damping coefficient $(N.\frac{sec}{m})$
Static test	29	-	-	-
Dynamic	29	0.154	1	1218

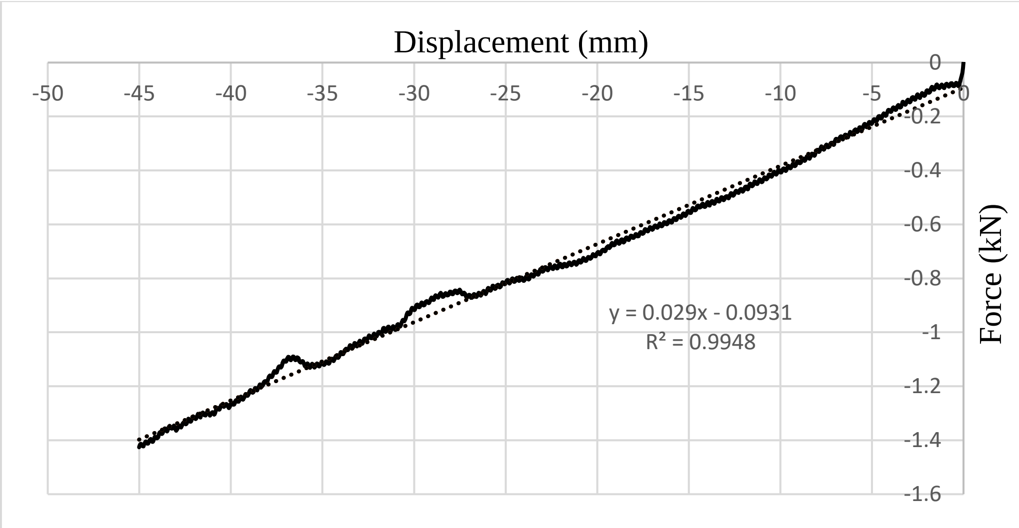

Figure D-13: Displacement- Force graph of FFFluid under static load for trial 7.

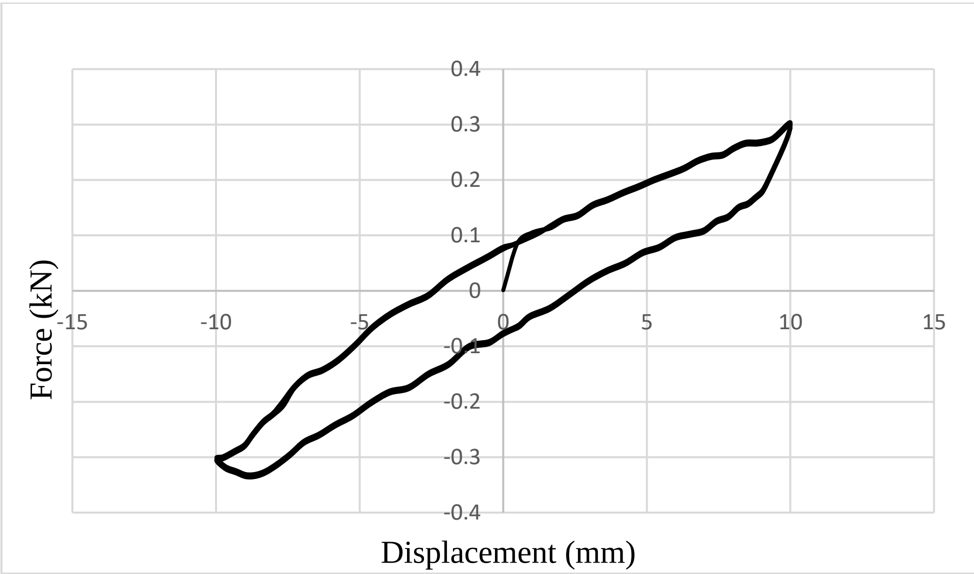

Figure D-14: Displacement- Damping force graph of FFFluid for trial 7.

Trial 8	Stiffness (kN/m)	Damping force at middle position 2CV (kN)	Frequency Hz	Damping coefficient $(N.\frac{sec}{m})$
Static test	34	-	-	-
Dynamic	34	0.161	1	1282

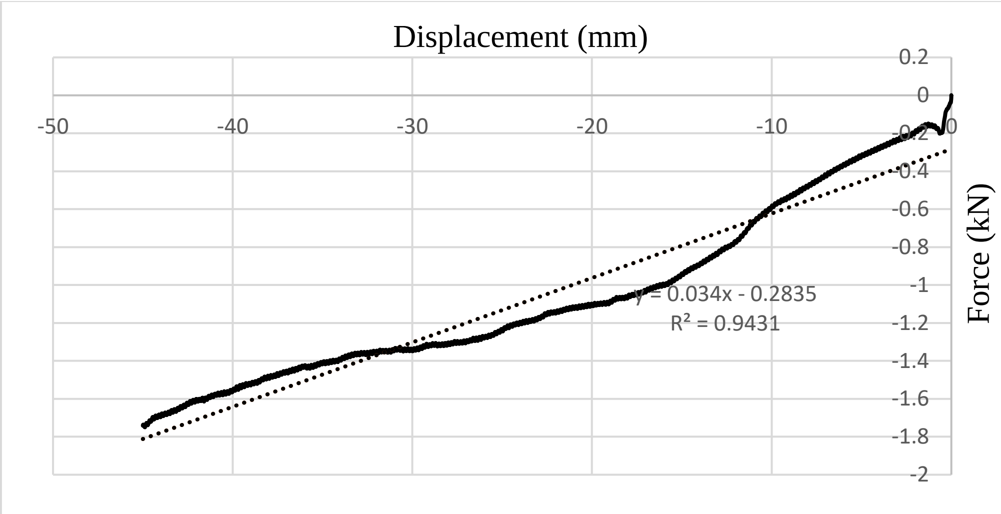

Figure D-15: Displacement- Force graph of FFFluid under static load for trial 8.

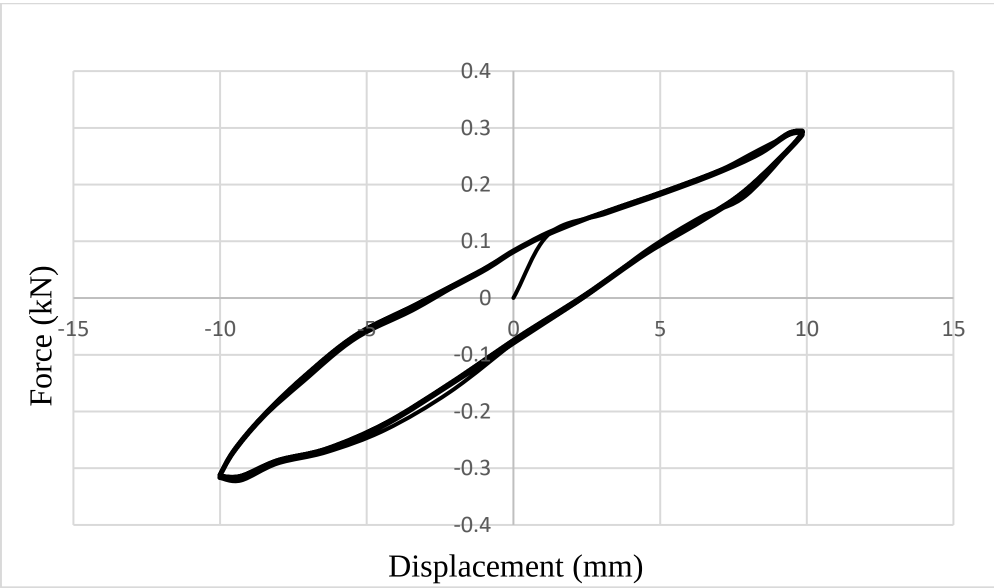

Figure D-16: Displacement- Damping force graph of FFFluid for trial 8.

Trial 9.	Stiffness (kN/m)	Damping force at middle position 2CV (kN)	Frequency Hz	Damping coefficient $(N.\frac{sec}{m})$
Static test	23	-	-	-
Dynamic	23	0.135	1	1075

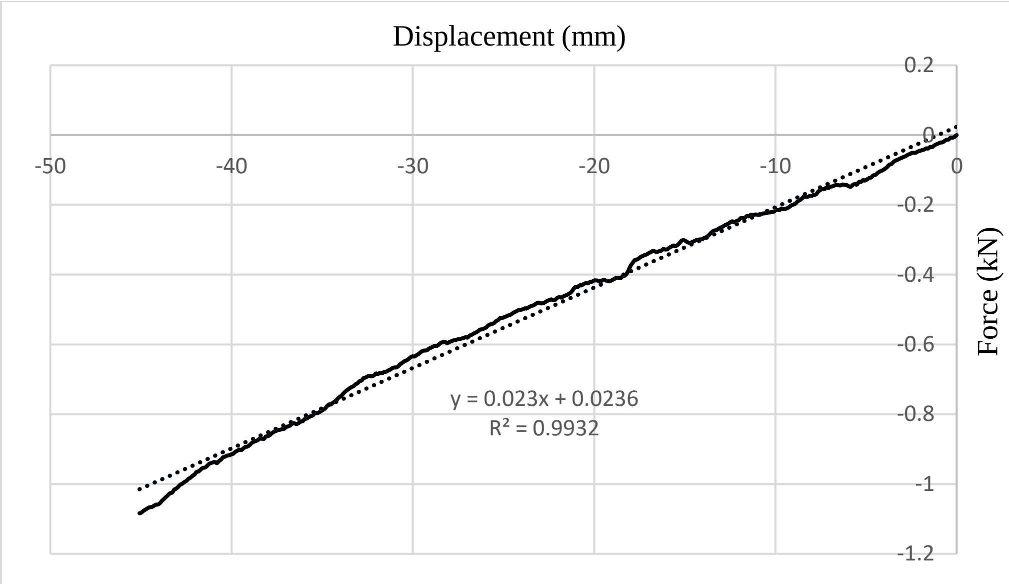

Figure D-17: Displacement- Force graph of FFFluid under static load for trial 9.

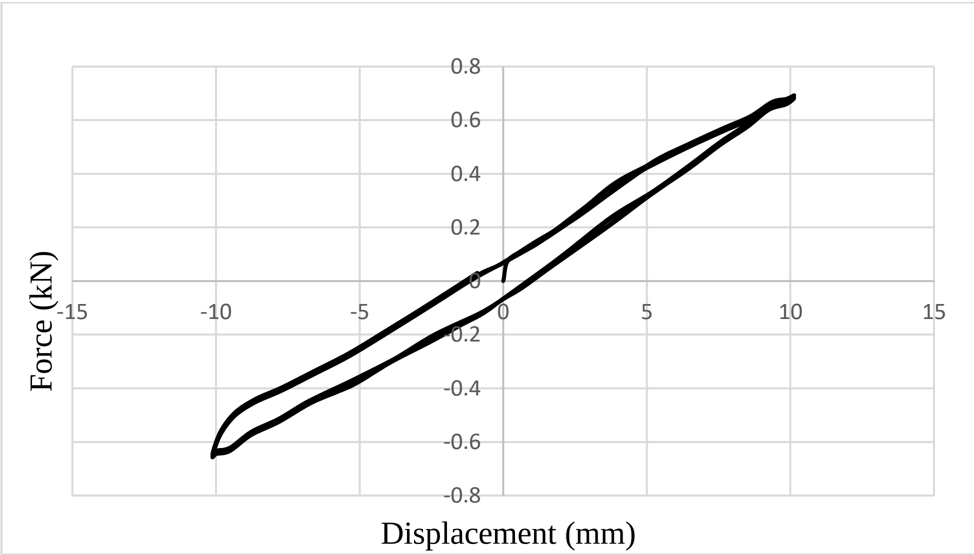

Figure D-18: Displacement- Damping force graph of FFFluid for trial 9.

APPENDIX- E: Results of validation experiments

Trial 1	Stiffness (kN/m)	Damping force at middle position 2CV (kN)	Frequency Hz	Damping coefficient $(N.\frac{sec}{m})$
Static test	18	-	-	-
Dynamic test	18	0.19	1.25	1210

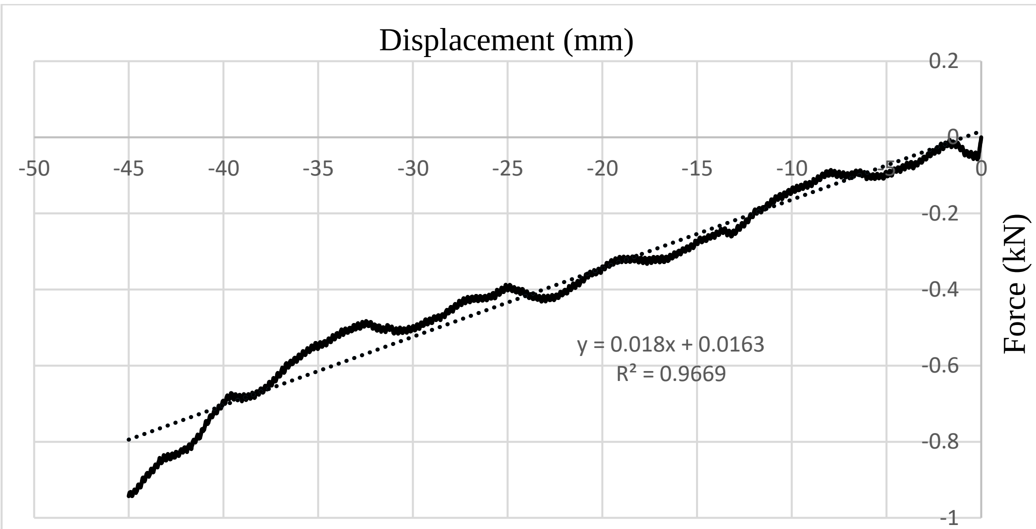

Figure E-1: Displacement- Force graph of FFFluid under static load for trial 1.

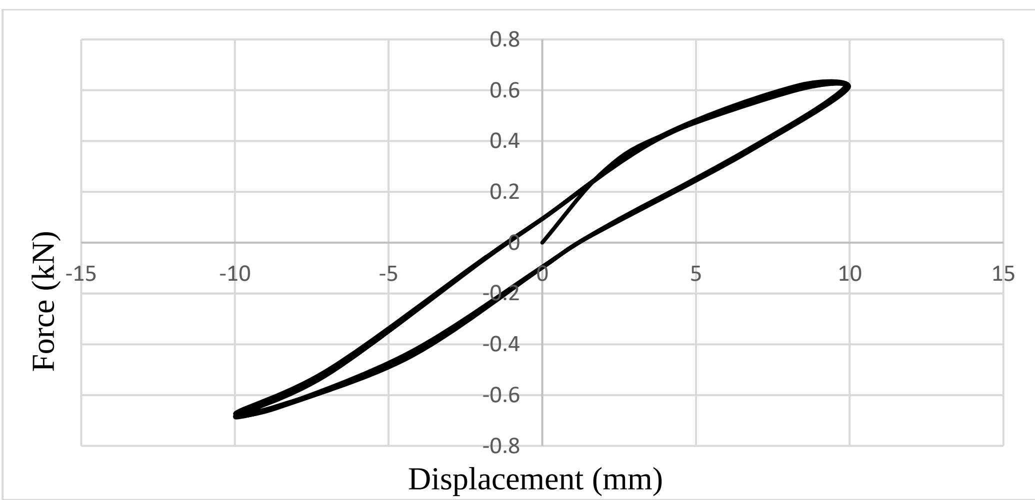

Figure E-2: Displacement- Damping force graph of FFFluid for trial 1.

Trial 2	Stiffness (kN/m)	Damping force at middle position 2CV (kN)	Frequency Hz	Damping coefficient $(N.\frac{sec}{m})$
Static test	27.4	-	-	-
Dynamic	27.4	0.17	1.25	1082

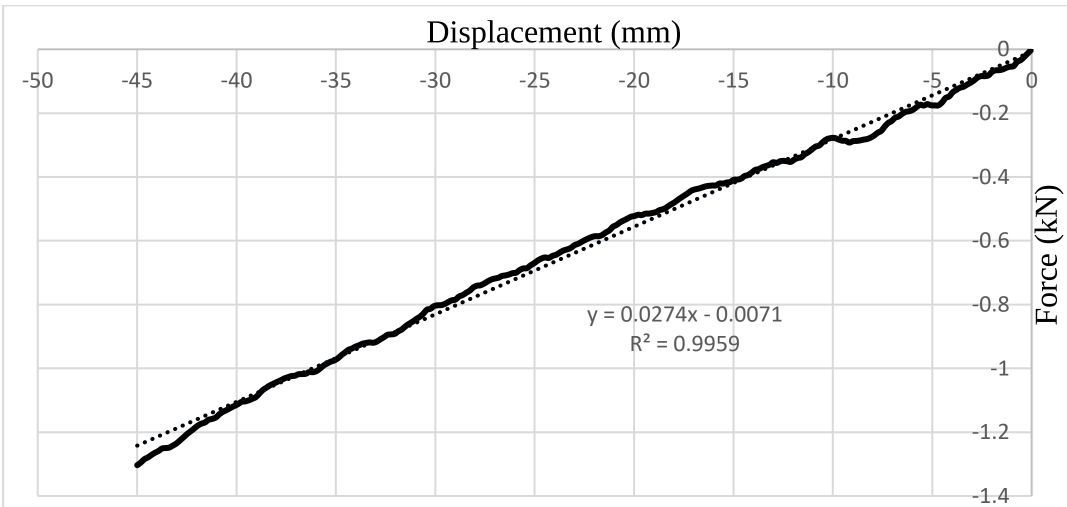

Figure E-3: Displacement- Force graph of FFFluid under static load for trial 2.

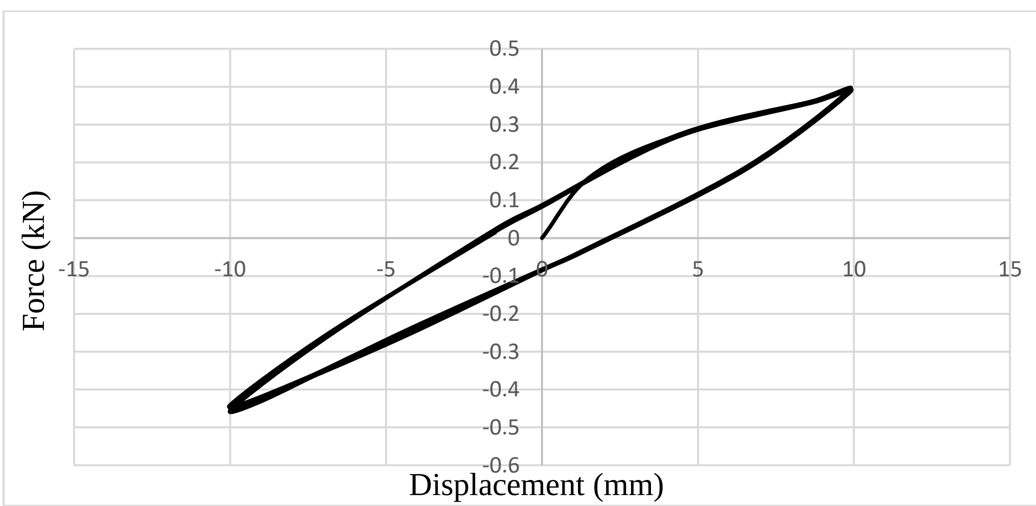

Figure E-4: Displacement- Damping force graph of FFFluid for trial 2

Trial 3	Stiffness (kN/m)	Damping force at middle position 2CV (kN)	Frequency Hz	Damping coefficient $(\mathrm{N.}\frac{\mathrm{sec}}{\mathrm{m}})$
Static test	24.4	-	-	-
Dynamic	24.4	0.23	1.25	1465

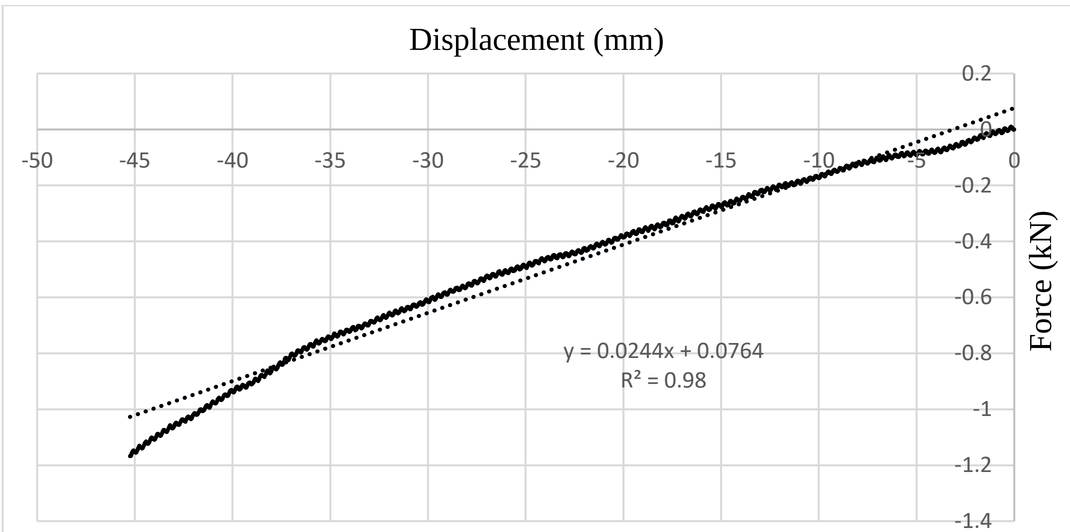

Figure E-5: Displacement- Force graph of FFFluid under static load for trial 3.

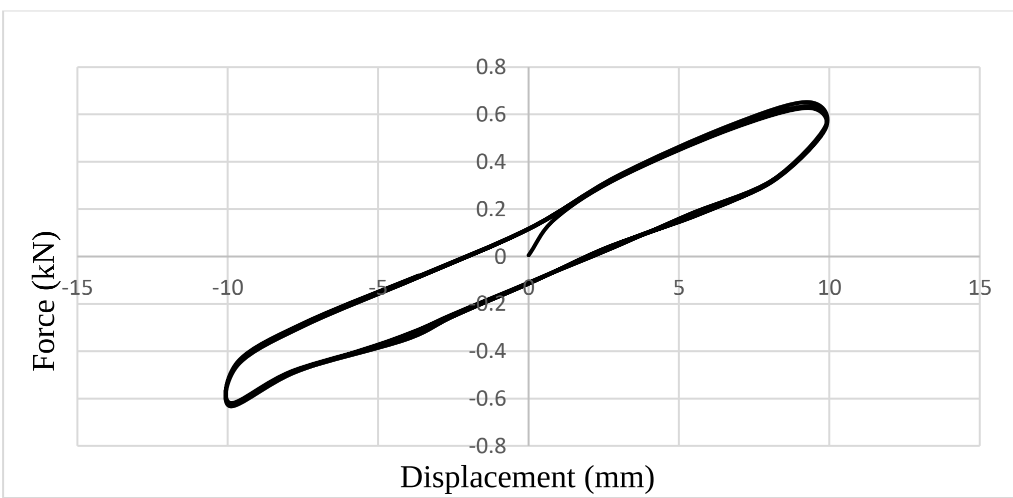

Figure E-6: Displacement- Damping force graph of FFFluid for trial 3.

Trial 4	Stiffness $(\mathrm{kN/m})$	Damping force at middle position 2CV (kN)	Frequency Hz	Damping coefficient $(\mathrm{N.}\frac{\mathrm{sec}}{\mathrm{m}})$
Static test	32.5	-	-	-
Dynamic	32.5	0.25	1.25	1592

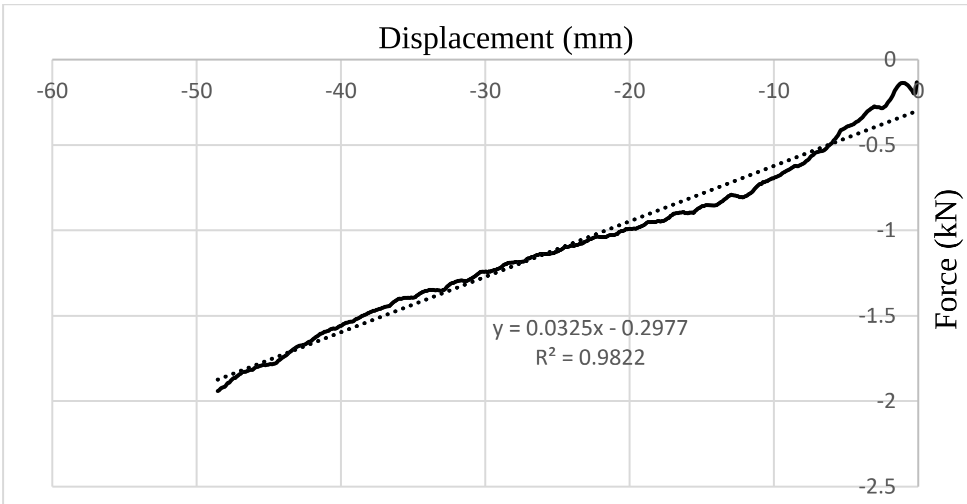

Figure E-7: Displacement- Force graph of FFFluid under static load for trial 4.

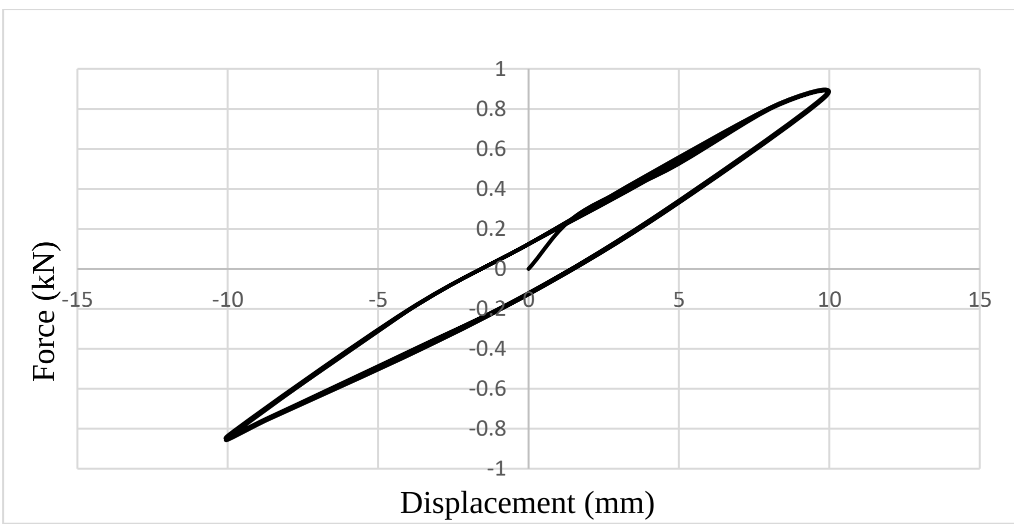

Figure E-8: Displacement- Damping force graph of FFFluid for trial 4.

Trial 5	Stiffness (kN/m)	Damping force at middle position 2CV (kN)	Frequency Hz	Damping coefficient $(N.\frac{sec}{m})$
Static test	21.8	-	-	-
Dynamic	21.8	0.21	1.25	1338

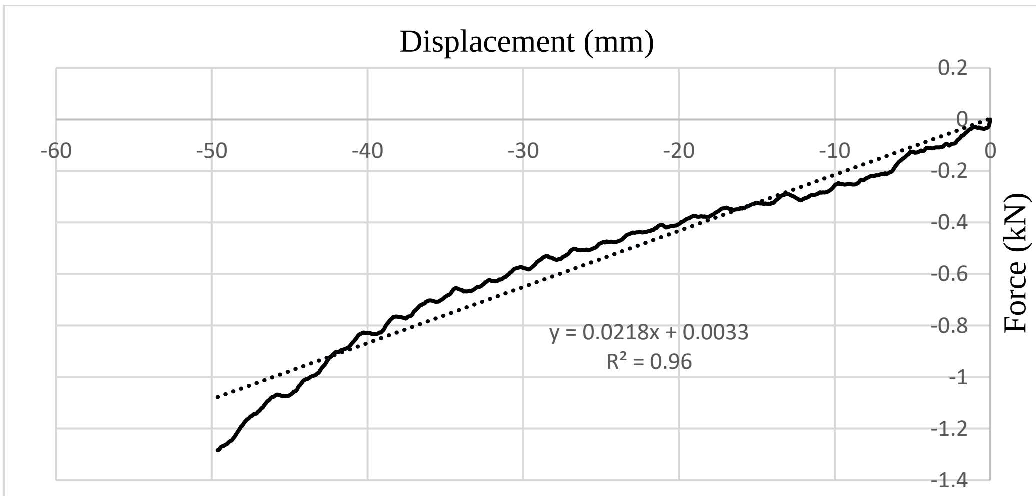

Figure E-9: Displacement- Force graph of FFFluid under static load for trial 5.

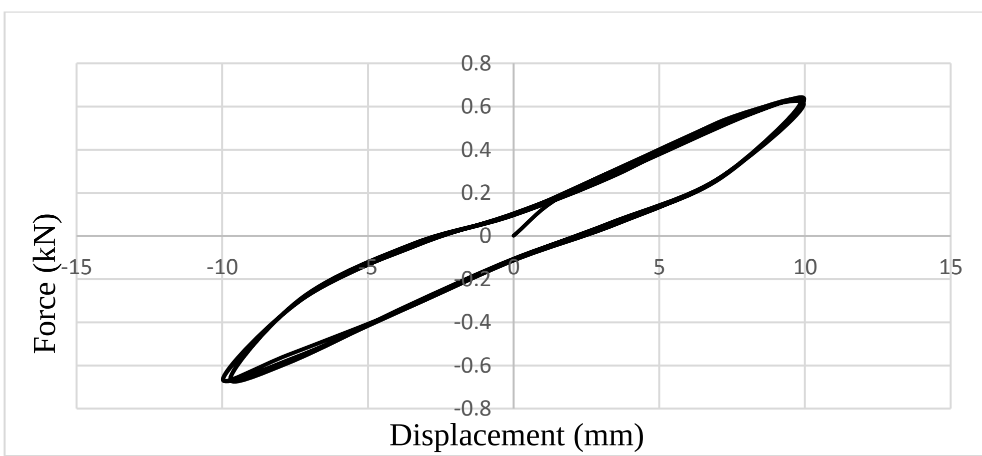

Figure E-10: Displacement- Damping force graph of FFFluid for trial 5.

Trial 6	Stiffness (kN/m)	Damping force at middle position 2CV (kN)	Frequency Hz	Damping coefficient $(N.\frac{sec}{m})$
Static test	22.1	-	-	-
Dynamic	22.1	0.15	1.25	955

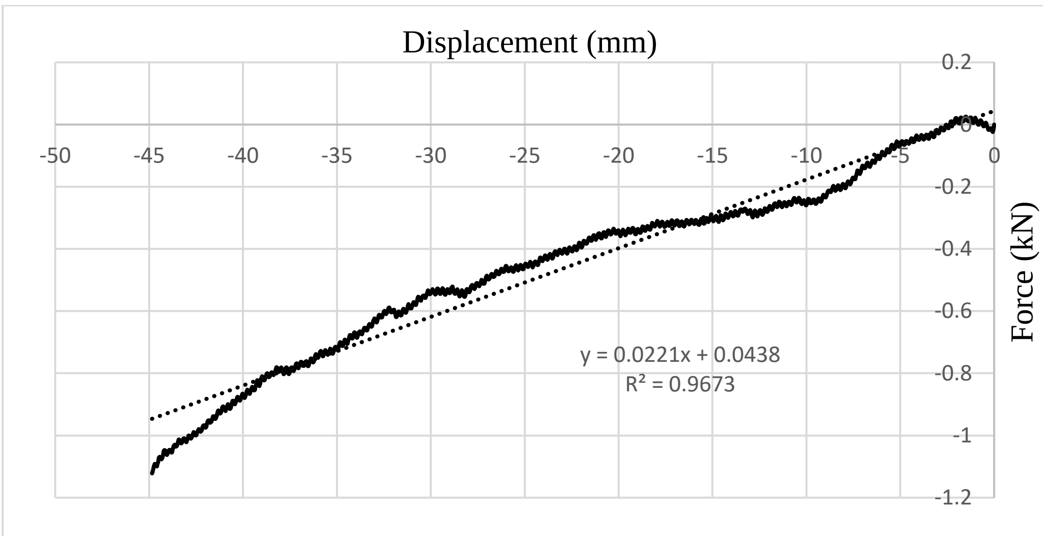

Figure E-11: Displacement- Force graph of FFFluid under static load for trial 6.

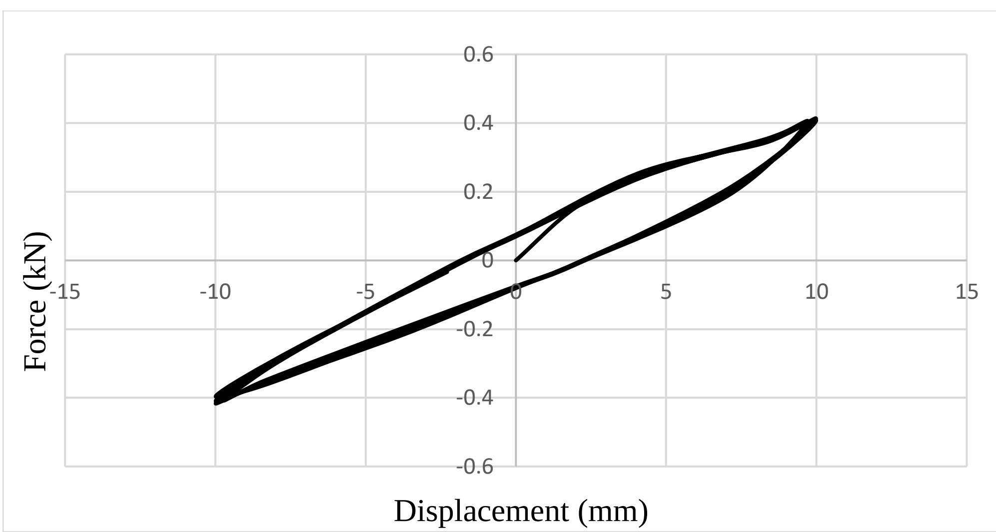

Figure E-12: Displacement- Damping force graph of FFFluid for trial 6

APPENDIX- F: Summary Output of Regression Model for stiffness Coefficient

Table 0-1: Summary Output of Regression Model after deleting X4

Regression Statistics	
Multiple R	0.995
R Square	0.991
Adjusted R Square	0.963
Standard Error	1.063
Observations	9

ANOVA

	dof	*SS*	*MS*	*F*	*Significance F*
Regression	6	242.82	40.471	35.848	0.027
Residual	2	2.258	1.129		
Total	8	245.08			

	Coefficients	*Standard Error*	*t Stat*	*P-value*
Intercept	-16.828	60.493	-0.278	0.807
X1	218.641	261.682	0.836	0.491
X12	10284.665	12646.258	0.813	0.501
X2	86.083	135.306	0.637	0.590
X22	-57.50	75.130	-0.766	0.524
X3	-40.801	62.738	-0.650	0.582
X42	326.697	28.785	11.350	0.008

Table F-2: Summary Output of Regression Model after deleting X2

Regression Statistics	
Multiple R	0.994
R Square	0.989
Adjusted R Square	0.970
Standard Error	0.951
Observations	9

ANOVA

	dof	*SS*	*MS*	*F*	*Significance F*
Regression	5	242.366	48.473	53.565	0.005
Residual	3	2.714	0.905		
Total	8	245.08			

	Coefficients	*Standard Error*	*t Stat*	*P-value*
Intercept	21.63	2.049	10.559	0.002
X1	218.641	234.289	0.933	0.419
X12	10284.66	11322.38	0.908	0.430
X22	-9.726	2.157	-4.510	0.020
X3	-40.81	56.17	-0.727	0.520
X42	326.698	25.77	12.677	0.001

Table F-3: Summary Output of Regression Model after deleting X3

Regression Statistics	
Multiple R	0.993
R Square	0.987
Adjusted R Square	0.974
Standard Error	0.893
Observations	9

ANOVA

	dof	*SS*	*MS*	*F*	*Significance F*
Regression	4	241.888	60.472	75.768	0.0005
Residual	4	3.193	0.798		
Total	8	245.08			

	Coefficients	*Standard Error*	*t Stat*	*P-value*
Intercept	21.445	1.909	11.234	0.0004
X1	218.64	220.027	0.994	0.376
X12	10284.66	10633.17	0.968	0.388
X22	-9.725	2.025	-4.802	0.008
X42	326.698	24.203	13.499	0.0002

APPENDIX- G: Summary Output of Regression Model for damping Coefficient

Table G-1: Summary Output of Regression Model after deleting X42

Regression Statistics	
Multiple R	0.99
R Square	0.980
Adjusted R Square	0.922
Standard Error	0.042
Observations	9

ANOVA

	dof	*SS*	*MS*	*F*	*Significance F*
Regression	6	0.180	0.030	16.89	0.057
Residual	2	0.004	0.002		
Total	8	0.185			

	Coefficients	*Standard Error*	*t Stat*	*P-value*
Intercept	-3.668	2.405	-1.525	0.266
X1	57.311	10.402	5.51	0.032
X12	-2939.061	502.71	-5.85	0.028
X2	9.448	5.378	1.757	0.221
X22	-5.043	2.987	-1.688	0.233
X3	8.063	2.494	3.233	0.084
X4	2.389	0.345	6.927	0.020

APPENDIX-H: Investigation of the FFFluid Technology for Anti-Vibration Devices.

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INVESTIGATION OF THE FOAM FILLED FLUID TECHNOLOGY FOR ANTI-VIBRATION DEVICES

[Haithem Elderrat, Huw Davies, Emmanuel Brousseau]

***Abstract*— In this study, Foam Filled Fluid (FFFluid) technology was investigated for the design of a novel vibration isolator, which is referred to as an FFFluid isolator. This technology relies on the utilisation of both the strain of foam capsules and fluid motion for reducing unwanted vibrations. Such an FFFluid isolator basically consists of compressible elastic particles of foam, mixed with an incompressible fluid while this mixture is contained in a controlled volume. When the FFFluid isolator is affected by vibrations, energy is absorbed due to the elastic strain of the foam. As the foam strain also enables movement of the fluid, this contributes to further energy absorption due swirling and the viscous effect of the fluid. The packaging could also contribute to attenuate vibration through the generated friction between the piston and the cylinder used to contain the FFFluid. Former studies showed that promising performances in reducing unwanted forces can be achieved with shock absorbing devices using the FFFluid technology. Such studies also highlighted the importance of defining key parameters of FFFluid devices properly. The present study was focused on characterising the FFFluid technology for vibration isolation. The performance of the system was determined based on experimental data in order to assess the stiffness and damping coefficients of the developed device.**

***Keywords*—FFFluid, Vibration Isolator**

I. Introduction

Vibrations can be utilized in many applications. However, for most of the time the presence of vibrations causes several issues in mechanical systems, such as decreasing the quality of manufactured products, producing noise, generating fatigue in mechanical components and others [1]. For these reasons, the theory of vibrations within an engineering context has been studied for over a century, and it is still a major engineering concern in the design of mechanical systems [2, 3]. There are several ways for controlling unwanted disturbances [2], one of these methods consists of inserting devices between sources of vibration and systems. These devices, called vibration isolators, help to attenuate problems arising from vibrations. Various types of anti-vibration devices have been designed previously [4, 5]. These isolators were developed using different concepts such as; elastomer isolators [6], controllable isolators (isolators with control systems) [7], pneumatic isolation systems [8] and others [9, 10]. Although there are a variety of methods for designing practical vibration isolators, the design of alternative novel isolators is still an area of active research. This is due to the increasing demand for anti-vibration devices, and to limitations with respect to cost and size, for example of current isolators in some applications especially for light automobile applications. For instance; air leakage is admitted in most types of pneumatic systems, and this issue causes to reduce performance of isolators quickly [8]. High cost and low reliability are the main problems for almost any controllable isolation system [11]. High temperature and local stress are disadvantages of viscous-elastic materials such as natural rubber, elastomer and foam material [12].

Foam Filled Fluid (FFFluid) is a composite material used for eliminating unwanted forces. It is made of a blend of polymeric foam capsules and a fluid contained in a closed package. Due to the viscoelastic effect of polymeric foams and the viscous effect of fluid, Foam Filled Fluid has the ability to reduce the effect of vibrations. Some applications have been designed using this technology such as vehicle bumper systems [13] and shock absorbers [14]. Designing a vibration isolator by using a similar method was also investigated [15]. However, the contribution of attenuating energy due to the viscous effect of fluid was not accounted. The polymeric foam will be used due to its low cost and availability. This paper has investigated the possibility of characterising FFFluid isolators. The objectives of this research were: (i) to present advantages of FFFluid technology over other existing technologies (ii) to carry out initial experimental works to evaluate the performance of an FFFluid isolator.

The paper is organised as follows: section II presents the FFFluid technology; this includes its components and the working mechanism of FFFluid devices. This is followed by presenting the advantages of FFFluid isolators over current vibration isolators in section III. In section IV, the methodology of characterising FFFluid isolator is described, then the experimental procedures and results analysis are explained in sections V, and VI respectively, and finally conclusions are drawn in section VII.

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APPENDIX- I: Improving the Exploitation of Fluid in Elastomeric Polymeric

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Improving the Exploitation of Fluid in Elastomeric Polymeric Isolator

Haithem Elderrat, Huw Davies, Emmanuel Brousseau

Abstract**—**Elastomeric polymer foam has been used widely in the automotive industry, especially for isolating unwanted vibrations. Such material is able to absorb unwanted vibration due to its combination of elastic and viscous properties. However, the 'creep effect', poor stress distribution and susceptibility to high temperatures are the main disadvantages of such a system.

In this study, improvements in the performance of elastomeric foam as a vibration isolator were investigated using the concept of Foam Filled Fluid (FFFluid). In FFFluid devices, the foam takes the form of capsule shapes, and is mixed with viscous fluid, while the mixture is contained in a closed vessel. When the FFFluid isolator is affected by vibrations, energy is absorbed, due to the elastic strain of the foam. As the foam is compressed, there is also movement of the fluid, which contributes to further energy absorption as the fluid shears. Also, and dependent on the design adopted, the packaging could also attenuate vibration through energy absorption via friction and/or elastic strain.

The present study focuses on the advantages of the FFFluid concept over the dry polymeric foam in the role of vibration isolation. This comparative study between the performance of dry foam and the FFFluid was made according to experimental procedures. The paper concludes by evaluating the performance of the FFFluid isolator in the suspension system of a light vehicle. One outcome of this research is that the FFFluid may preferable over elastomer isolators in certain applications, as it enables a reduction in the effects of high temperatures and of 'creep effects', thereby increasing the reliability and load distribution. The stiffness coefficient of the system has increased about 60% by using an FFFluid sample. The technology represented by the FFFluid is therefore considered by this research suitable for application in the suspension system of a light vehicle.

Keywords**—**Anti-vibration devices, dry foam, FFFluid.

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I. Introduction

UNWANTED vibrations can raise hazardous conditions in any type of dynamic system, ranging from large multi-storey buildings to small measurement component systems [1]-[3]. Consequently, vibration control has been a point of concern for engineers aiming to improve system performance. The essential features of an isolator are resilient load-supporting characteristics and the opportunity for energy dissipation. The means of providing resilient load-carrying is performed by springs; while the means of energy-dissipation is provided by dampers. In some types of isolators, the functions of the load-supporting elements and the energy-dissipating elements may be performed by a single element, e.g., a layer or natural rubber or other viscoelastic material.

Viscoelastic materials, often termed as elastomeric polymers, are extensively used as a means to mitigate resonant vibration responses. A very common approach is to use pads or sheets made of elastomeric polymer between the base and the movable component. This method is characterized by its low weight, low cost and its ability to be formed into different shaped absorbers. It has been used in many applications, such as automobile car seats, body armor, construction equipment, etc. [4]. However, in certain applications its poor load distribution, creep and high temperature effect is usually the main disadvantages of such material [5], [6]. One solution to minimize the effect of these disadvantages is to implement elastomeric isolators in bonded configurations, where metal inserts are bonded to the elastomer on all load-carrying surfaces. Although the performance of the part is improved, this approach has a higher cost, due to both the additional material and also the processing required because of the special chemical preparation required to achieve a bond with strength in excess of that of the elastomer itself [7]. The solution proposed as part of this research is to mix small particles of elastomeric polymer with a viscous carrier fluid, after which the mixture is contained in an enclosed package. This mixture is a relatively new method that is designed to improve the utilization of the elastomer material, and reduce the early damage due to the concentration of stress in a local area [8]. Both solid dense and foam of elastomeric polymers could be exploited in the design of such a device. However, this research will be focused on elastomeric foamed materials. Specifically, the mixture investigated here is called Foam Filled Fluid (FFFluid). There have already been some applications using this technology, such as the FFFluid shock absorber [9], and vibration isolator [10].

The objectives of this research were to present the advantages of the FFFluid technology over dry elastomeric polymer foam, experimental investigation was carried out to illustrate some of these benefits. Then, the performance of an FFFluid device in the suspension system of a light vehicle was evaluated.

The paper is organized as follows: Section II presents the FFFluid technology; this includes the working mechanisms of FFFluid devices. In Section III, the advantages of FFFluid over dry foam are presented. The performance of the FFFluid suspension system is then evaluated in Section IV, and finally, conclusions are drawn in Section V.

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APPENDIX- J: The Characterisation of a Foam Filled Fluid Vibration Isolator

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CHARACTERISATION OF A NOVEL FOAM FILLED FLUID VIBRATION ISOLATOR

[Haithem Elderrat- Huw Davies- Emmanuel Brousseau]

Abstract— **Foam-filled fluid (FFFluid) is an innovative material technology that can be used to design anti-vibration devices. This material technology is able to dissipate energy via a number of mechanisms due to the combination of elastic and buckling properties of foams, viscosity properties of fluids and the method of packing the material. The relationship between the energy isolation mechanisms can be altered by modifying the properties of the constituent components which provides significant opportunity for tailoring the response of the isolator to meet the different demands placed upon it.**

While previous studies have characterized the FFFluid shock absorbers, this paper covers the characterization of the mechanical properties of FFFluid vibration isolators. The work presented here starts by studying the parameters that have an effect on the performance of an FFFluid isolator theoretically. Then, the influence of a selected number of factors on the transmissibility of an FFFluid isolator is systematically studied following the Taguchi Parameters Design method. The investigated factors include size of particles, ratio of foam/fluid, viscosity of the fluid, size of package, displacement and frequency. The main conclusions of this research are (1) the biggest volume of FFFluid and highest viscosity of fluid is providing better results in isolating the vibration sources, and (2) filling 90% of a container with foams that have diameters 5 mm are providing the best force transmissibility characteristic compared to other examined ratios or used sizes.

I. Introduction

The presence vibrations can give rise to hazardous conditions in any type of dynamic system, ranging from large multi-bladed helicopters to small electronic component systems. Consequently vibration control has been a point of concern for engineers to improve system performance. There are several ways for controlling unwanted disturbances [1]. Polymeric foam materials are one of the materials which have been used to design vibration isolation system. It is characterized by its low weight and low cost and it is also able to be shaped into small sizes [2-4]. Polymeric foam is able to dissipate a large quantity of energy due to the combination of its elastic, plastic and buckling modes of the constituent cells that occur during the loading and unloading, it also has the ability to dissipate energy due to the viscosity of the fluid inside the structure of foam. Although utilization of fluid in polymeric foam is leading to improving their performance and minimizing their drawbacks, this mechanism either is not exploited in dry foam or partly exploited in fluid filled foam (FFFoam) [5].

Foam Filled Fluid (FFFluid) is another innovation that improves the exploitation of fluid in polymeric foam. It is made of a blend of polymeric foam capsules that are suspended in the fluid and contained in a closed package. A number of applications have been investigated using this technology such as vehicle bumper system [6], shock absorbers [7] and vibration isolator [8]. There are several advantages of FFFluid mixtures over dry foam for eliminating vibrations. The multi-axis loading of the foam prevents localised stress concentration keeping the loading of the foam in the elastic region and also enabling lower density foams to be utilised. These advantages help raise the ability of the FFFluid for absorbing the vibrations. Moreover the FFFluid composite material has smart properties. It could be designed as a smart structure by using FFFluid mixture without any external control system. A vehicle system bumper has been designed by using FFFluid [6]. This structure is able to have a stiff design impact between vehicle to vehicle, and stiffness in the case of an impact between a vehicle and pedestrian. When FFFluid is subjected to a compression load, the energy is absorbed due to the elastic and buckling properties of foams, viscosity properties of fluids and the method of packing the material. This combination can be altered by changing the properties of the constituent components. In turn, this provides significant opportunity for tailoring the response of the FFFluid to meet different demands placed upon it. Previous studies focused on characterising shock absorbing performances of the FFFluid technology [7, 9], while this paper investigates the contribution of a selected set of design parameters on the transmissibility of an FFFluid anti-vibration system using the Taguchi Parameter Design method.

The paper is organised as follows: section II presents a theoretical study on the parameters that have an effect on the total performance of the isolator, then the experimental procedures to examine the most critical parameters are described in section III, the experimental program consists of the following steps: determine the objectives and identification of the variables; then followed by performing and executing the designed experiment. In section V, the results are analysed statistically, and final conclusions are drawn in sections VI.

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