RICE UNIVERSITY

EXPERIMENTAL AND ANALYTICAL STUDY OF
VARIABLE FLUID DAMPER AND STIFFNESS DEVICE

by

Jeffrey Dyck

A THESIS SUBMITTED
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Approved, Thesis Committee:

Satish Nagarajaiah, Chairman
Associate Professor
Civil and Environmental Engineering

Michael Terk
Assistant Professor
Civil and Environmental Engineering

Andrew J. Meade, Jr., P.E.
Associate Professor
Mechanical Engineering and Material Science

Houston, Texas

June, 2004
Abstract

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A prototype variable fluid damper and stiffness device is evaluated in this study. This damper is capable of providing continuous variation with different levels of system damping and stiffness. The damping and stiffness can be varied independent of each other or simultaneously. The following research is focussed on the evaluation of the effectiveness of such a device, and on determining the effectiveness of incorporation of such a device into a framed structure. The damper properties are determined under cyclic and ramp loadings, and the results are presented. Further experimental research is performed to evaluate the effectiveness of the afore mentioned device in a proposed Scissor-Jack energy dissipation system. This system is incorporated into a typical steel frame; whereupon the variable damping and stiffness system’s effectiveness is enhanced by a factor of four. By means of test results of the damper it is shown that the damper is capable of producing continuously and independently variable damping and stiffness. It is shown that a similar variation in frame structure damping and stiffness can be achieved by using the damper in a Scissor-Jack system. The presented device can be an effective means for response reduction of structures under earthquake and/or wind excitation.
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Chapter 1

Introduction

1.1 Overview

The use and research into the practicality and the applicability of damping devices to dissipate seismic energy in building and bridge structures has dramatically increased in present years. Implementation of such energy dissipation devices allows for the limitation of, or the elimination of structural damage produced by induced energy transferred to the structure as a result of seismic activity.

Conventional building materials, such as steel and concrete, allow for approximately 2 to 5% inherent damping. Under traditional design systems, the seismic induced energy has to be absorbed by the structure through inelastic deformations causing costly and sometimes un-repairable damage. But, with the incorporation of energy dissipation devices like the damper, the seismic energy is absorbed through the dissipation device instead of the structure itself, thus resulting in expanded life of the structure and cost savings to the owner. Further, with the incorporation of dissipation devices in mind during initial design phases of a structure, it can become possible to lower the cost of materials used for construction of the structure. This becomes not only evident in the bill of materials at initial costs of construction, as long as dissipation system costs less than traditional stiffening mechanism, but also later on in the structures life after weathering seismic activity.

The incorporation of energy dissipation devices into earthquake mitigation has evolved into three distinct approaches: passive, active, and semi-active control. Passive control systems utilize the
ability of a specially designed mechanical device(s) to absorb energy through deformation of that device(s) applied to a given structural system. Active control systems require the "active" participation of specially designed mechanical device(s), whose mechanical characteristics change as a result of feedback responses of the structure to which the device(s) is applied. Semi-active control involves variations of damping and/or stiffness properties of a mechanical device(s) applied to a given structure based upon feedback from the applied system.

The following research pertains to the complete study of a prototype variable damping and stiffness device that can be incorporated into a given semi-active control system. Further, this prototype device is incorporated into an innovative Scissor-Jack lateral bracing system, whereupon the combined system responses are enhanced significantly.

1.2 Objectives

The objective of this thesis is to systematically represent the process demonstrated in the discovery of both experimental and analytical information that could be beneficial to future incorporations of energy dissipation devices in frame structures for earthquake and/or wind response reduction. Chapter 1 as presented, gives a brief introduction to seismic dissipation devices. Chapter 2 encompasses a more complete background on seismic dissipation devices and their history regarding their incorporation into steel frame construction. Chapter 3 presents extensive background and experimental information (set-up, procedure, and results) regarding the prototype variable damping and stiffness device. Chapter 4 consists of information regarding background and theory of the Scissor-Jack energy dissipation system. Additional information is presented regarding incorporation of the damper into the test-setup and experimental results obtained. Chapter 5 presents the analytical modelling and results of the variable damping and stiffness device in the Scissor-Jack system. Finally, Chapter 6 concludes with remarks and suggestions for further research.
Chapter 2

Historical Survey

2.1 Introduction

Traditionally, methods for designing earthquake resistant structures have relied on the availability of ductile behavior of the structural members to provide energy dissipation. When such designs are pushed to inelastic responses, plastic deformations form causing permanent damage to the structure. Such system designs do not promote long term product utility in earthquake prone regions. As the cost of constructing, maintaining, and serviceability have increased, so too have the demands on a product's lifetime utility and capacity to weather multiple seismic events. As a result, designers have looked for new and innovative ways to match the demands of costs to the structures utility and performance.

Currently, engineers have turned towards using a capacity design approach for controlling both location and type of inelastic behavior experienced by the structure. Such an approach allows for more reliable and predictable failure modes, through energy dissipation, while at the same time making the structure more resistant to the uncertainties associated with seismic events. One innovative approach to controlling the dissipation of energy introduced to a given structure is through the incorporation of energy-based design which incorporates energy dissipation devices. The applied method allows for the separation of the energy dissipation from the load carrying capacity of the structure. Thus, it presents the availability of a lower cost in actual structural materials used, and becomes easier for the designer to detail and optimize the structure's energy dissipation. For such a system to maintain effectiveness, the device(s) incorporated into the structure must be properly
designed for accessibility and replaceability and/or repairability after major seismic events. To effectively evaluate the efficiency of energy dissipation devices in an applied structural system, studies are often made of the balance of the energy input and the use of various energy considerations to achieve improved performance.

The last two decades have provided a number of devices with high-energy dissipation capacity. These devices are generally linked to the main members of the lateral force resisting system (LFRS). Device performance is as enhanced semi-rigid connections or joining elements with high damping characteristics. Such devices as friction-damped devices, yielding steel inner frames, and added damping and stiffness (ADAS) elements are examples of elements often added at critical joints of steel bracing members. Performance of these devices is such that deformation occurs during severe seismic excitation before yielding or buckling in the primary system occurs. It thus prevents or nearly prevents damage to the major members of the LFRS and allows for energy dissipation by hysteretic behavior through the applied devices. The hysteretic behavior of the dissipation devices is associated with the inelastic behavior of the structure, and thus, it can be considered as damaging seismic energy [50].

2.2 History of Energy Dissipation Devices in Frames

As stated above, most of the energy inputted into a structural system is dissipated by hysteresis. In its origins, energy dissipation devices were conceived and intended to be integral members of base-isolation systems, and at later stages, adopted to bracing systems. Hysteretic devices, to date, can be classified as one of the following: yielding or friction devices. The following discussion is a brief survey of the developmental history of dissipation devices incorporated into framing systems.
2.2.1 Friction Devices

The pioneering of friction devices in seismic design was spearheaded by R. G. Tyler. He suggested a method that promoted the reduction of damage to infill panels all the while still providing an acceptable level of damping to a building. By incorporating polytetrafluoroethylene (PTFE) elements along infill panels within frame members, the infill panels are able to perform under loading, within capacity, while the joints slip within a known force range.

![PTFE sliding elements](image)

**Figure 2.1:** PTFE sliding elements [79]

In 1980, Pall developed a passive seismic control system for precast and cast-in-place concrete walls by the introduction of friction devices. Figure 2.2 illustrates typical details for the above mentioned system. The applied slipping friction joints consist of heavy-duty brake lining pads inserted between sliding steel plates jointed by high-strength bolts. For the joints to properly perform, they must be engineered to approach ideal elastoplastic behavior. Through the creation of vertical joint lines inside concrete walls plus the use of properly engineered friction joints to couple them together, the system can receive the following beneficial characteristics [50]:

Figure 2.2: Friction Joints in Concrete Walls [60]

- The walls act as a monolithic unit during service load conditions (including wind and moderate earthquakes).

- As the friction joints slip during major seismic activity, a concurrent increase in the flexibility of the system results in the elongation of the effective period and thus a reduction in seismic accelerations.

- The friction joints dissipate a large portion of the seismic input and delay the possible onset of inelasticity in the LFRS.

For economical reasons, braced frames have become the industry standard for control of lateral deflections resulting from wind and moderate earthquakes. Unfortunately, such systems do not perform satisfactorily under major earthquakes resulting from their very limited energy dissipation capacity. Such limited energy dissipation capabilities become even worse when the bracing system's effective design is for tension loadings. These tension-only mechanisms elongate during high interstory displacements, and the upon load reversal buckle in compression. Upon the next load reversal, now in the tension phase, the bracing mechanism is rendered ineffective until it becomes taut and
further stretched. Continual loading reversals only further the degradation of the load capacity of the member. [50]

With continued research by Pall, new friction device design began to focus towards the assembly in the intersection of steel bracing. Figure 2.3 shows several friction based devices proposed by Pall [59] for tension-only and tension-compression bracing systems. The conceptual purpose for such above mentioned designs is to minimize the conceived drawbacks encountered in the performance of steel bracing systems during major earthquake mitigation.

![Diagram](image)

(a) Device for tension-only cross bracing

![Diagram](image)

(b) Device effective in tension and reverse slip at zero compression

![Diagram](image)

(c) Device effective in tension and compression

**Figure 2.3:** Friction devices for bracing systems [59]

The friction device incorporated into the bracing system is designed not to slip under normal service loads and low to moderate earthquakes. If the system encounters a major earthquake, slippage of the device occurs at a predetermined loading before yielding can occur in other structural elements. For bracing designed to withstand compression, a simple slotted friction connection can be incorporated to slip in both tension and compression.
In 1986, Baktash and Marsh [9] proposed a simple friction-damped bracing system where upon the braces are connected to the structure by bolting through steel gusset plates with slotted holes. Further, brake lining pads are mounted to both sides of the plates, and a system of spring plates is used to control the slip force in the joints. Further studies by Baktash and Marsh [10] on friction devices provided guidance on the tuning of friction devices.

At the turn of the decade, Anagnostides [7] [8] proposed an alternative friction device for tension-only cross-bracing. Figure 2.5 illustrates two variations of Anagnostides's proposed device which incorporates rotational friction joints instead of friction joints that slip in a linear trajectory. The
device consists of frictional washers bolted to steel plates and a distribution washer by high-strength bolts. The device’s strength results from the material and dimensions of the above mentioned washers and the pressure applied by the bolts. The adoption of the rotational friction device as opposed to translational friction allows for the expected increase production ability per unit of time and thus a lower unit of cost; in addition there is the ability to produce more consistent hysteretic friction behavior due to the geometry of the frictional sources ability to apply a more uniform clamping pressure on the frictional material. [50]

![Friction device diagram](image)

**Figure 2.6:** Friction device for wooden panel systems [21]

Approximately the same time, Filiatrault [21] proposed incorporating friction devices, as demonstrated in Figure 2.6, into the corners of timber-sheathed wall structures. Filiatrault’s study reported approximately 60% of the hysteretic energy experienced by the testing structure during experiments was dissipated by the friction in the wall devices.

In 1992, Aiken, Nims, and Kelly [5] reported testing of the sophisticated Japanese friction "Sumitomo" device in various frame systems, as shown in Figure 2.7. Originally, designed as a shock absorber in railway rolling stock, it was later applied to steel bracing systems. Initial tests demonstrated the device's independence of loading frequency, amplitude, number of loading cycles, and temperature. Further, the device demonstrated regular and repeatable hysteretic behavior with no variation in slip load during earthquake motion.
In the same above mentioned study, Aiken's group also tested an energy-dissipating strut originally intended for nuclear power plants as a seismic restraint device for the support of piping systems. The device, as illustrated in Figure 2.8, allows for self-centering and demonstrates frictional forces proportional to experienced displacements. Further, the self-centering ability of the strut allows for the reduction of permanent off-sets resulting from a structures inelastic deformation. In addition, the study suggests that bracing systems incorporating energy-dissipating struts are more effective in reducing a structures response to relatively harmonic excitations than for impulsive excitations. [50]

By 2000, a study released by Filiatrault, Tremblay, and Kar [22] proposed a new dissipative strut. The authors referred to it as a friction-based ring spring damper. The device, Figure 2.9, comprises a friction spring composed from the assembly of cylindrical wedges or rings connected in series.

Such devices as the above mentioned struts have proven quite effective in improving seismic behavior
of braced frames, but incorporation and utilization is limited compared to energy dissipation devices of simpler designs due to costs and complexity of setup. As a result of the demand for simpler friction devices, Grigorian, Tsong-Shuoh, and Popov [25] redesigned and developed a more suitable friction device known as the slotted bolted connection (SBC). As illustrated in Figure 2.10, the device is comprised of a bolted connection where elongated holes or slots in the main connecting plate are aligned with the direction of loading. The strength of the device is a direct result of the amount of tightening given to the bolts. This research work is acknowledged in that it resulted in the development of a reliable friction device with intrinsic simplicity and low cost. [50]

As a result of ever changing codes and the ever increasing demand for retro-fitting existing structures
with systems to increase deformation capacity under earthquake loading, many conventional and innovative redesign techniques have been developed over the years. Many of the designs have proven to be worthy, but often such techniques come with such undesired "side effects" as a substantial amount of construction work, large augmentations to building weight and base shear, crucial adaptations to building layout, stern agitation to building occupants, and in some cases, the modernization of the foundation system due to augmentations in base shear.

Figure 2.11: Examples of low invasivity incorporation of hysteretic devices [50]

Martínez-Rueda [44] [45] [46] [47] [49] [50] [48] [52] has set forth alternative solutions to the above mentioned problems based upon low-invasivity retrofitting techniques for framed structures, as seen in Figure 2.11. These alternatives, based on the local or global incorporation of energy dissipation devices into the structural system, demonstrate, through many nonlinear seismic analyses, the introduction of significant hysteretic damping in regards to seismic excitation. The local approach was developed to alleviate such problems as significant disturbances to the occupants and loss of income
due to major renovations to the space availability and functionality after renovation often associated with the installation of steel bracing with or without devices to a structure. Although its ideal and cost efficient, the local approach can have drawbacks in instances where a large number of devices may have to be installed in order to provide adequate seismic protection. Further, in areas of the application, appropriate design considerations and possible alterations will have to be made to the structural system due to the introduction of modifications in the flexural and shear demands of the surrounding regions. As a result, the global approach may prove to be more efficient. This approach relies first on the incorporation of passive control of the initial period of the structure. This inactive device time period involves a brace with stable hysteretic response. Next the approach relies on high energy-dissipation capacity upon activation of the dissipation devices. With its adaptive geometry, space loss upon installation can be minimalized by the design engineer. Like the local approach, a major drawback to the global approach can be seen. To ensure beneficial device activation without excessive gain in lateral strength incompatible with strength of the structural system, the global approach must avoid large scale device strengths. [50]

2.2.2 Yielding Devices

As with early friction devices, yielding devices in their infant stages were focussed on the enhancement of infilled frame systems. In 1965, Guerrero [26] proposed the system in figure 2.32(a). This system, if properly designed, creates a passive control with regards to the lateral forces transmitted by the structure to the infill partitions. Even with the proposed systems possible benefits, primarily its effectiveness in protecting infill wall partitions and in making significant contributions in the energy dissipation capacity of the structure, the system would require expensive construction procedures that would make the cost effectiveness questionable.

By 1969 Muto [53] introduced a nonconventional seismic design with reinforced concrete infill panels.
CHAPTER 2. HISTORICAL SURVEY

Figure 2.12: Incorporation of yielding devices into infill walls

As seen in figure 2.32(b), the infilled concrete panels have vertical slits which cause the panels to act as a series of RC columns. Energy absorption occurs as a result of plastic hinges that develop at the top and bottom of each effective column during deformation of the panel during interstory deflection.

With the advent and research into steel-plate energy dissipation devices originating in the 70's, development of added damping and stiffness (ADAS) elements has quickly become one of the most popular devices incorporated into chevron bracing. Figure 2.13 shows the alliance of ADAS elements into a MRF to connect chevron braces with the floor system. Since their application into frames, it has been shown that when properly designed, ADAS elements can increase the robustness, rigidity, and energy dissipation capacity of MRFs. Further, it can be shown that in concentric-braced frames, the application of ADAS elements results in substantial increase in the energy dissipation capacity per unit-story drift.

Whittaker and colleagues [82] undertook a research program evaluating ADAS elements in upgrading moment-resisting frames. Results of their efforts confirm that the incorporation of ADAS elements improved the behavior of the bare frame. Further, testing substantiated three major advantages of the incorporation of ADAS elements in braced frames:

1. Inelastic deformations and hysteretic energy dissipation can be bounded to a limited number
Figure 2.13: Moment-resisting frame with ADAS elements [82]

of predetermined, easily replaceable elements.

2. For minor and moderate levels of earthquake excitation, yielding of the ADAS elements can produce significant reductions in interstory deformations.

3. Stable hysteretic behavior of the bracing system can be preserved throughout a severe earthquake.

Figure 2.14: TADAS elements [77]

The triangular-plate added damping and stiffness (TADAS) devices are a variation of the ADAS
elements developed by Tsai and colleagues [77] [78]. The TADAS elements, as shown in figure 2.14, incorporate a series of triangular steel plates connected chevron braces and beams of steel frames. The two systems differ in the fact that ADAS elements are X-shaped plates that are bolted together through two ends of each plate, while the TADAS elements are connected to a base plate by a welded connection. In a later publishing Tsai [76], points out that the most attractive feature of the TADAS system is the participation of gravity loading on the frame can be completely separated from the device. This is illustrated in the connection detail, figure 2.14, through the use of the slotted holes in the connection. During large deformations, the vertical displacements at the end of the triangular plate are easily tolerated and thus bending only generates plasticity within the plates. As a result, the inelastic response of the TADAS device can be easily predicted [50].

In 1990, Pocanschi, Krause, and Haendel [64] proposed an enhanced braced system that combines the cost effectiveness of tension-only cross-bracing with the hysteretic behavior of ADAS elements. As shown in figure 2.15, the system incorporates a closed cross-bracing system, effective only in tension, that is fixed at the bottom of the columns, and is capable of moving at the top corners of the frame. The frame and the cable bracing system are attached through a ADAS-like element in order to govern relative motion.

Figure 2.15: Closed cross-bracing with yield device [64]
In 1992, Aguirre and Sánchez adapted the U-shaped yielding device, studied by Kelly [34], to a new steel bracing yielding device that could handle both tension and compression cycling. Known as the "oval element", as seen in figure 2.16, this device when properly designed can be guaranteed for 100 cycles for a desired maximum displacement. [50]

![Image of oval element](image)

**Figure 2.16:** Application of oval element as yield device for bracing [4]

Originally suggested by David Smith and Robert Henry of Auckland, New Zealand in the late 1970's, the yielding device in figure 2.32(a) is described by Tyler [80] as a yielding frame. This device achieves energy consumption by the yielding of a rectangular frame constructed from round bars. During cyclic loading, distortion occurs in the shape of a parallelogram. The perversion encourages stable cyclic behavior denying the progressive slack that correlates to cross-bracing under severe seismic loading. Acceptable performance of this device is dependent upon horizontal dynamic forces oscillating about zero load. If the zero loading is not maintained, the device will have a steady movement in one direction and a consequent locking up in that direction.

Testing the above mentioned device under cyclic loadings demonstrated many hundred of cycles of loading before the development of unwanted slack was experienced. Further, the lock-up effect occurred at very large deformations which would normally be considered outside effective design range. Due to the transverse section of the yielding frame being constant, this specific yielding device is biased to high concentrations of ductility demands around the corners. This results in the inefficient contribution of material to energy dissipation.
Ciampi and Samuelli-Ferretti [14] furthered the development of the yielding frame devices with their proposals, as illustrated in figures 2.18 and 2.19. Their designs consist of a yielding inner frame fabricated from steel plates. As a result of the linear varying bending moment diagrams for the devices, uniform plastification in bending is accomplished by varying of the member sections in the yielding frame. Further, plastification, resulting primarily from bending and the devices ability to resist shear and axial forces, is maintained by a transition region of constant section in the central part of the device members.
Ciampi and his research group's [15] [16] [13] continued research developments resulted in the "E-shaped" and "C-shaped" devices, as shown in figure 2.20. As with the predecessors, the new devices are cut from a thick steel plate and are designed to work in the plane of that plate. The geometry allows for an almost uniform plastification, especially at small displacements.

As shown in figure 2.20, these devices are easily incorporated into bracing systems, but they have
proven applicable to bridges as dissipative connection devices between the decks and piers and/or abutments. For bracing systems specifically, the braces are designed to react elastically both in tension and compression while the yielding device deforms plastically.

2.3 A Look into the State of the Art Structural Control:

Semi-Active Control Systems

The current trend of control strategies based on the application of semi-active devices, appears to merge the best features of both passive and active control systems. In addition, these strategies offer the greatest likelihood for near-term acceptance of control technology as a viable means of protecting structural systems with regards to earthquake and wind mitigation. A major focus of interest is a direct result of the ability of semi-active control devices to offer the adaptability of active control devices without requiring the large power sources associated with active control devices. In fact, many semi-active control systems can operate on battery power, which is crucial advantage during seismic events when the main power source to the structure may fail.

Currently, the universally recognized definition of a semi-active control device is one which cannot inject mechanical energy into the controlled structural system (i.e., including the structure and the control device), but has properties which can be controlled to optimally reduce the responses of the system [70]. In contrast to active control devices, a semi-active control devices does not have the potential to destabilize (in the bounded input/bounded output sense) the structural system. Preliminary research indicates that, when appropriately designed and implemented, semi-active systems perform significantly better than passive devices. Further, they have the potential to achieve the majority of the performance of fully active systems, thus allowing for the possibility of effective response reduction during a wide array of dynamic loading conditions [70], [72], [?]}. Examples of such
devices that will be discussed in the remainder of this section including the following: variable-orifice fluid dampers, variable-stiffness devices, controllable friction devices, smart tuned mass dampers and tuned liquid dampers, controllable smart fluid dampers, and controllable impact dampers. [70]

2.3.1 Variable-Orifice Dampers

One method in achieving a semi-active damping device is to use a controllable, electromechanical, variable-orifice valve to alter the resistance to flow of a conventional hydraulic fluid damper. Such a device, schematically shown in figure 2.21, typically operates on approximately 50 watts of power. Feng and Shinozuka [19] where the first to conceptually apply the concept of this type of variable-damping device to control the motion of bridges experiencing seismic motion. In the following years, analytical and experimental research had been carried out by Kawashima and Unjoh [34], Sack and Patten [67], Patten [63], Symans and Constantinou [75], Nagarajuiah [55], Yang [88], and Liang [40]. In 1993, Sack and Patten [67] developed a hydraulic actuator with a controllable orifice. This device was implemented by Patten [64] in a full-scale bridge on interstate highway I-35 in Oklahoma in order demonstrate the technology’s ability to reduce the vibrations produced by vehicle traffic. Symans and Constantinou [75] and Symans and Kelly [76] analytically and experimentally studied the application of variable fluid dampers for seismic response reduction of buildings and bridges. Jabbari and Bobrow [30], Yang [86], and Iwan [29] have all studied a resettable hydraulic actuator with controllable orifice as an applicable on-off variable stiffness device.

![Figure 2.21: Schematic of the variable orifice damper](image)
2.3.2 Variable Stiffness Device

Conceived as a variable-stiffness device, Kobori [36] implemented a full-scale variable-orifice damper, using the on-off mode, in a variable stiffness system (AVS) for semiactive control of the Kajima Research Institute building (see figure 2.22. Although a variable orifice dampers can be used for producing variable stiffness, a result of the on-off mode properties consisting of a very high stiffness device resulting from hydraulic fluid compressibility (primarily due to entrapped air) when the valve is closed or no stiffness when the valve is open, these devices cannot vary stiffness continuously between different stiffness states.

![Image](a)

![Image](b)

**Figure 2.22:** Kajima Technical Research Institute with AVS system

By the end of the 1990’s, Nagarajaiah [56] had developed a semi-active continuously and independently variable stiffness device (SAIVS), scalable example is shown in figure 2.23, able to vary stiffness continuously and smoothly. Research by Nagarajaiah and Mate [57] demonstrated the effectiveness of the SAIVS device for varying the stiffness smoothly and producing a non-resonant system in scaled structural model.
2.3.3 Smart Tuned Mass Dampers

A considerable of research has been focused towards determining the advantages and effectiveness related to tuned mass dampers (TMD) and multiple tuned mass dampers (MTMD). The TMD is limited by its sensitivity to tuning frequency ratio, even when optimally designed. The MTMD has the ability overcome this limitation, but the MTMD cannot be retuned in real time. Thus, it is not adaptable, and space constraints can limit the location of a large number of TMD’s. In 1982, Hrovat [28] began conducting research on the next stage of evolution of the device(s), TMD’s with adjustable damping. By 2000, Nagarajaiah and Varadarajan [58] had continued the evolutionary process by associating the SAIVS device to the TMD to create a semi-active tuned mass damper (STMD) with a using the SAIVS device [56]. The system has the distinct advantage of being able to continuously re-tune its frequency in real time which makes it robust to changes in the building’s stiffness and damping. The variation of stiffness of the STMD is based on estimation of instantaneous frequency and a time frequency controller developed by Nagarajaiah and Varadarajan [58]. The continued research by them [82] in 2004, illustrated the device’s effectiveness in a tall benchmark building with response reductions comparable to that of an active tuned mass dampers with an order of magnitude less power consumption. STMD’s, based on variable damping, has been studied by Abe and Igusa [2]. Continued research into semi-active impact dampers have also been developed
and studied, by Caughey and Karyealis [11], and their experimental effectiveness has been shown by Masri [52].

STMD's can also be based on (1) controllable tuned sloshing dampers (CTSD), and (2) controllable tuned liquid column dampers (CTLCD). A TSD uses liquid sloshing in a tank to add damping to the structure. Similarly, in a TLCD, liquid is used as a moving mass in a column, which is driven by the vibrations of the structure, to promote damping. Because of their fixed design, these systems are not as effective for a wide variety of loading conditions, and as a result, researchers are looking to improving their effectiveness in reducing structural responses [33]. In 1994, Lou [42] proposed a semi-active CTSD device based on the passive TSD, where the length of the sloshing tank can be altered to change the properties of the device. Abe [3] and Yalla [85] have conducted research into semi-active CTLCD devices based on a TLCD with a variable orifice.

2.3.4 Variable-Friction Dampers

Various semi-active devices have been proposed which utilize forces generated by surface friction to dissipate vibratory energy in a structural system. Akbay and Aktan [6] and Kannan [32] proposed a variable-friction device that consists of a friction shaft that is rigidly connected to the structural bracing. The force at the frictional interface can be adjusted by allowing slippage in controlled amounts. Feng [20] incorporated a semi-active friction-controllable fluid bearing in parallel with a seismic isolation system. Recent developments include research into response reduction of nonlinear buildings by Yang and Agrawal [87] and experimental research into piezoelectric dampers by Garret [23].

2.3.5 Controllable-Fluid Dampers

As a result of their design, most semi-active dampers employ some electrically controlled valve(s) or mechanism(s) to achieve changes in device's characteristics. Such mechanical components have
a history of being problematic in terms of reliability and maintenance. Advancements have led to a new class of semi-active control devices that use controllable fluids in a fixed-orifice damper. As shown in figure 2.24, the advantage this type of device is its mechanical simplicity. Thus, the damper contains no moving parts other than its piston.

![Figure 2.24: Schematic of controllable-fluid damper](image)

Two fluids that are viable contenders for the development of controllable dampers are: (i) electrorheological (ER) fluids and (ii) magnetorheological (MR) fluids. To date, only MR fluids have been shown to be attractive for civil engineering applications [71]. The essential characteristic of these fluids is their ability to reversibly change from a free-flowing, linear viscous fluid to a semi-solid with a controllable yield strength in milliseconds upon exposure to a magnetic field. In the absence of an applied field, these fluids flow freely and can be modelled as Newtonian. MR fluids typically consist of micron-sized, magnetically polarizable particles dispersed in a carrier medium such as mineral or silicone oil and can operate at temperatures from -40 to 150°C with only modest variations in the yield stress. In addition, MR fluid devices can be readily controlled with a low power (e.g., less than 50 watts), low voltage (e.g., 12-24V), current-driven power supply outputting only 1-2 amps. Such power levels can be readily supplied by batteries. [70]

Through simulations and laboratory model experiments, MR dampers have been shown to significantly outperform comparable passive damping configurations, while requiring only a fraction of the input power needed by the active controller [71], [74], [72], [73], [18], [17], [56], [68], [84], [89], [24],
Moreover, the technology has been demonstrated to be scalable to devices sufficiently large for implementation in civil engineering structures.

2.3.6 Full-Scale Applications

The Kajima Technical Research Institute, shown in figure 2.22, was the first full-scale building structure to be implemented with semi-active control devices. The AVS is a hydraulic device with a bypass valve used to switch the device between the on-off positions to engage and disengage the bracing system. As a result, the structural system can be varied between the configurations of a purely moment resistant framing system to a fully braced framing system. The building's stiffness is varied based on the nature of the earthquake to produce a non-resonant system. The observed responses during several earthquakes [36] indicate the effectiveness of the AVS system in reducing the structural responses.

In the United States, the first and only to date full-scale implementation of semiactive control was conducted on the Walnut Creek Bridge, shown in figure 2.26, on interstate highway I-35 to demonstrate variable damper technology [64].

More recently, a smart damping system was implemented in the Kajima Shizuoka Building in Shizuoka, Japan. As seen in figure 2.27, semi-active hydraulic dampers are installed inside the
walls on both sides of the building to enable it to be used as a disaster relief in earthquake situations [1], [39], [38] [37], [59]. Each damper contains a flow control valve, a check valve, and an accumulator, and can develop a maximum damping force of 1000 kN (see figure 2.28). Both story shear forces and story drifts are have been shown to be greatly reduced with the control activated. In addition, shear forces are confined within their elastic-limit values in contrast to the plastic range that would be achieved with out the active control.

The use of the variable-orifice damper has prospered in Japan. Figure 2.29 shows the construction site in the Siodome area in downtown Tokyo. At present, there are 4 buildings under construction in this area that will employ switching semi-active hydraulic dampers for structural protection. One of these structures, the Kajima K-Tower, is a 172 m tall, 38-story hotel and office complex installed with 88 semi-active dampers and 2 hybrid mass dampers (see figure 2.30). In the Roppongi area of Tokyo, the Kajima R-Building, a 54-story building with 356 variable-orifice dampers and 192 passive dampers distributed throughout, is also under construction (see figure 2.31). Altogether, the Kajima Corporation is currently constructing or has recently finished 9 buildings in Japan that employ semi-active hydraulic dampers for structural protection. Upon completion of these projects, a total of nearly 800 variable-orifice dampers will be installed in building structures.
2.4 Analytical Modelling of Dampers

2.4.1 SDOF Equation of Motion

A single degree of freedom (SDOF) can be modeled in the following form.

\[ m\ddot{u} + c\dot{u} + ku = -m\ddot{u}_g \] (2.1)
where \( m, c, k, u_g, \) and \( u \) and its derivatives represent mass, linear viscous damping, elastic stiffness, ground acceleration, and displacement, velocity, and acceleration of the SDOF, respectively. By dividing equation 2.1 by the mass term, an alternative form of the equation of motion of a SDOF can be derived.

\[
\ddot{u} + 2\zeta \omega \dot{u} + \omega^2 u = -\ddot{u}_g
\]  (2.2)
2.4.2 Fluid Viscous Damper Model

A non-linear fluid viscous damper's (FVD) properties of force, $f_D$, and velocity, $\dot{u}$, can be analytically expressed in the following notation.

$$f_D = c_\alpha \text{sgn}(\dot{u}) |\dot{u}|^\alpha$$  \hspace{1cm} (2.3)

where $c_\alpha$ is the experimentally determined damping coefficient with units of force per velocity raised to the $\alpha$ power. The value of $\alpha$ is a real positive exponent with typical values in the range of 0.35-1 for seismic applications [27], and $\text{sgn}(\cdot)$ is the signum function. A linear fluid viscous damper is
represented by $\alpha = 1$ and the reduction of equation 2.3 to $f_D = c_1 \dot{u}$. When $\alpha = 0$, equation 2.3 reduces to $f_D = c_0 \text{sgn}(\dot{u})$. Therefore, it should be stated that $\alpha$ defines the non-linearity of the fluid viscous damper model \cite{41}.

During a cycle of harmonic motion represented by $u = u_0 \sin \omega t$, the energy dissipated by the FVD modeled in equation 2.3 is

\begin{equation}
E_D = \int_0^{2\pi/\omega} f_D \dot{u} dt = \int_0^{2\pi/\omega} c_\alpha |\dot{u}|^{1+\alpha} dt
\end{equation}

Upon integration of equation 2.4, the following is derived.

\begin{equation}
E_D = \pi \beta_\alpha c_\alpha \omega^\alpha u_0^{\alpha+1}
\end{equation}

for which $\beta_\alpha$ is

\begin{equation}
\beta_\alpha = \frac{2^{2+\alpha} \Gamma^2 (1 + \alpha/2)}{\pi \Gamma (2 + \alpha)}
\end{equation}
and $\Gamma(\cdot)$ is the gamma function. For a linear FVD, $\beta_\alpha = 1$ and equation 2.5 becomes

$$E_D = \pi c_1 \omega u_0^2$$  \hspace{1cm} (2.7)

For the case of pure friction dampers, when $\alpha = 0$, $\beta_\alpha = 4/\pi$ and equation 2.5 becomes

$$E_D = 4c_0 u_0$$  \hspace{1cm} (2.8)

Non-linear and linear FVDs dissipation of energy per cycle are the same when equations 2.8 and 2.7 are equal. Manipulation of this equality for $c_\alpha$ reduces to

$$c_\alpha = \frac{(\omega u_0)^{1-\alpha}}{\beta_\alpha} c_1$$  \hspace{1cm} (2.9)

By insertion of equation 2.9 into equation 2.3, an equation for damper force is derived for energy-equivalent FVDs.

$$\frac{f_D(t)}{f_{D0}(\alpha = 1)} = \frac{(\omega u_0)^{1-\alpha}}{\beta_\alpha \omega_0} \text{sgn}(\dot{u}) |\dot{u}|^\alpha$$  \hspace{1cm} (2.10)

where $f_{D0}(\alpha = 1) = c_1 u_0$ is the peak force of a linear FVD. The resulting peak value of the damper force is

$$\frac{f_{D0}(\alpha)}{f_{D0}(\alpha = 1)} = \frac{1}{\beta_\alpha} \left( \frac{\omega u_0}{\omega_0} \right)^{1-\alpha}$$  \hspace{1cm} (2.11)

For instances where the damper is undergoing harmonic motion, a peak velocity of $\dot{u}_0 = \omega u_0$ is achieved. Equation 2.11 will be reduced to

$$\frac{f_{D0}(\alpha)}{f_{D0}(\alpha = 1)} = \frac{1}{\beta_\alpha}$$  \hspace{1cm} (2.12)

For instances of non-harmonic motion, an energy equivalence must be determined due to the fact
that the system is most sensitive to damping at \( \omega = \omega_n \), where \( \omega_n \) is the natural vibration frequency of the SDOF system. By substituting \( \omega \) with \( \omega_n \), equation 2.11 gives

\[
\frac{f_{D0}(\alpha)}{f_{D0}(\alpha = 1)} = \frac{1}{\beta_\alpha} \left( \frac{\omega_n u_0}{u_0} \right)^{1-\alpha}
\]  
(2.13)

2.4.3 Equivalent Linear Viscous Damping

For energy-equivalent non-linear FVDs, the energy dissipation capacity is defined by the supplemental damping ratio \( \zeta_{sd} \) and its non-linearity by \( \alpha \), as previously mentioned. Both a linear single degree of freedom (SDF) system, characterized by mass \( m \) and stiffness \( k \), and a non-linear FVD defined by the previously mentioned equation 2.3, are defined by a \( \zeta_{sd} \) based on equivalent linear viscous damping. [12] Thus,

\[
\zeta_{sd} = \frac{E_D}{4\pi E_{sd}} = \frac{E_D}{2\pi ku_0^2}
\]  
(2.14)

\( E_{sd} \) represents the elastic energy stored at the maximum displacement \( u_0 \). By applying equation 2.5, evaluated at \( \omega = \omega_n \), with respect to equation 2.14, the supplemental damping ratio can be redefined in terms of the displacement amplitude, \( u_0 \).

\[
\zeta_{sd} = \frac{\beta_\alpha c_\alpha}{2ku_0} (\omega_n u_0)^\alpha = \frac{\beta_\alpha c_\alpha}{2m\omega_n} (\omega_n u_0)^{\alpha-1}
\]  
(2.15)

For linear FVDs (\( \alpha = 1 \)), equation 2.15 can be reduced to an equation independent of amplitude.

\[
\zeta_{sd} = \frac{c_1}{2m\omega_n}
\]  
(2.16)

For friction dampers (\( \alpha = 0 \)), equation 2.15 becomes

\[
\zeta_{sd} = \frac{2c_0}{\pi ku_0}
\]  
(2.17)
2.4.4 Equations of Motion

Now that the damper force \( f_D \) and supplemental damping ratio \( \zeta_{sd} \) have been defined, equation 2.1 can now be defined as follows:

\[
m\ddot{u} + c\dot{u} + ku + c_\alpha \text{sgn}(\dot{u}) |\dot{u}|^\alpha = -m\ddot{u}_g(t)
\]  

(2.18)

If equation 2.15 is solved for in terms of \( c_\alpha \), substituted back into equation 2.15, and then equation 2.15 is divided by \( m \), an equation of one parameter, \( \alpha \), is developed.

\[
\ddot{u} + 2\zeta_\alpha \omega_n \dot{u} + \omega_n^2 u + \frac{2\zeta_{sd} \omega_n}{\beta_\alpha} (\omega_n u_0)^{1-\alpha} \text{sgn}(\dot{u}) |\dot{u}|^\alpha = -\ddot{u}_g(t)
\]  

(2.19)

It should be noted that equation 2.19 defines the motion of SDF systems with energy-equivalent non-linear FVDs, which are depicted with the same \( \zeta_{sd} \) value but are defined by different \( \alpha \) values.

From the above mentioned equation, it can be stated that the response of the energy-equivalent SDF system with non-linear FVDs are controlled by the following parameters: [40]

- The damper non-linearity parameter of \( \alpha \), which defines the shape of the damper force hysteresis loop.

- The supplemental damping ratio, \( \zeta_{sd} \), which depicts the an independent energy dissipation capacity of the FVD with respect to \( \alpha \).

- The natural vibration period of the system defined by \( T_n = 2\pi/\omega_n \).

- The damping ratio \( \zeta \) which depicts the inherent energy dissipation capacity of the system.

2.5 Current Study

In the current study, a prototype variable fluid damper and stiffness device is tested to obtain its damping and stiffness properties. The device is then tested in a "Scissor-Jack" energy dissipation
system in a full scale steel frame. Once all information is collected on the prototype damper, it is modeled analytically. From the evaluation of both experimental and analytical results, the viability of both the prototype fluid damping and stiffness device and its subsequent application in the "Scissor-Jack" energy dissipation system is determined.
Chapter 3

Prototype Variable Fluid Damper and Stiffness Device

3.1 Introduction

Initial testing of the prototype variable fluid damper and stiffness device is conducted in a two piece testing frame developed for a shaketable setup. Tests are conducted in order to determine the energy dissipation capabilities under dynamic loadings.

![Figure 3.1: Actual Prototype Damper in Test Setup](image)

The distinct properties of adjustable stiffness and damping available by the prototype damper are controlled by separate valves, as shown in figure 3.2. The prototype damper is designed for an
allowable 4 inches of stroke upon extreme circumstances.

With a dynamic damping function of approximately 1933\(v^{0.9}\) while the damping valve is closed, where \(v\) is the velocity in inches per second, additional damping can be augmented to the system by opening the control valve. This valve allows addition flow through a 0.19 in. diameter orifice. The manufacturer highly recommends that the valve not be opened more than 4 complete revolutions from the completely closed position. This is to prevent any malfunction of the control valve.

The "high/low" liquid spring of the prototype damper is controlled by the valve actuator assembly located on the main cylinder. By rotating the small shaft sticking out at the bottom of the cylinder with a 1/16 in. Allen wrench, the spring's properties can be turned on or off. A rotation of 90° from the marked open position closes the valve, removing the low pressure accumulator from participation. This allows for the generation of high liquid spring forces upon an inward stroke of the piston. To re-activate the low pressure accumulator, another rotation of 90° is made from the position where the valve was closed. This is done to prevent unnecessary pressure build ups in the prototype damper. Further, if one attempts to open the valve in a position other than where it was closed, possible stripping of the valve can occur. In the fully extended position, the liquid spring with an open setting has low capabilities. If the damper is fully compressed under these proposed conditions, the liquid spring force will only produce approximately 19 pounds of force.

In order to take advantage of the liquid spring force, the spring valve must be closed in the fully extended position. Upon complete stroking of the damper, approximately 8,300 pounds of liquid spring force will be generated. The spring valve can be closed at any position along the stroke, and it will not change the damping of the prototype damper. But the liquid spring loads will change, resulting in lower values in generated spring force the closer the position is to being fully compressed.

The main hydraulic cylinder of the prototype damper is manufactured from 17-4PH stainless steel.
Each end of the damper is threaded with its own respective size, 1-8UNC-2A THD and 1/2-13UNC-2A THD, as illustrated in figure 3.2. For testing purposes, all connections are fabricated to meet the requirements of the testing structure.

**Figure 3.2:** Damper dimensions and the locations of the spring valve, damper valve, and accumulator

### 3.2 Test Set-up

The experimental testing setup is specifically designed to be able to test the prototype device at peak dynamic conditions. The setup, shown in figures 3.3, 3.4(a), 3.4(b), 3.4(c), and 3.4(d), allows for velocities up 21 inches per second and loadings in excess of 30 kips.

The bracket legs shown are made from welding two W8x16's on top of each other and then properly stiffened to assure proper transferal of forces to their respective base plate. As seen in the above mentioned figures, each support bracket has a face plate welded to it for contact with it's respective cross member. The left support structure, as seen in figures 3.3, 3.4(a), and 3.4(b), has a solid 1 inch face plate. Bolting to it is a w8x16 positioned horizontally to allow bolting through the bottom
flange of the member. This cross member has been adequately stiffened and 2-1/2 inch plates have been welded to the outside connected the top flanges to the bottom flanges in order to create a more box like section. This is done in order to prevent lateral torsional buckling during dynamic loading. A 1 inch thick mounting plate is then welded to the cross member to create a solid mounting point for the load cell.

Figure 3.3: Side View of Testing Set-up

The right support structure, uses two 1/2 inch plates instead of the solid 1 inch face plate used on the left support structure. This is done in order to create a moveable connection that allows for proper aligning of the right structure's mount for the damper. The cross member on the right support structure is made from two channels 6 inches in depth, adequately stiffened, and are spaced with pipe spacers to create once again a box like effect to prevent lateral torsion buckling of the section. A 1 inch thick mounting plate is welded halfway along the cross member allowing for a 1 inch threaded rode to pass through for mounting of the damper connection. Each support structure uses a 3 inch diameter solid spacer to adequately fill space left between the support structures. From the three inch spacers, a manufactured clevis protrudes providing adequate mounting of the
prototype damper. Refer to figure 3.3

Figure 3.4: Testing structure for prototype damper

3.3 Instrumentation and Data Collection

The data is collected by SIMULINK in conjunction with dSPACE from the load cell, with a maximum capacity of 56 kips dynamic and 110 kips static, and LVDT. The instrumented chamral included force and displacement measurements from the actuator and a LVDT to measure the displacement of the damper. Processed data is simulated in the form of Force vs. Displacement plots via program written in Matlab.
3.4 Types of Loading

In order to properly understand and model the properties of the damper, cyclic and ramp loadings of various frequencies and amplitudes were applied to the damper. Test signals of the forms represented in figures 3.5 and 3.6 were used via dSPACE to collect sufficient data with respect to the damper’s response under the various loadings.

![Graph of displacement over time](image)

**Figure 3.5:** Sample Full Sine Pulse Loading

Over 700 tests were conducted using cyclic waves ranging in frequency from 1/2 to 3 Hz. and amplitudes from ±1/4 to ±1 inch. Thirteen different high velocity ramp tests were conducted, refer for details to table 3.1

3.5 Test Results

Initial testing of the prototype damper began with the damper positioned in the “neutral stroking position” located at the midpoint of the stroke, i.e. at 2 inches. Systematic testing of cyclic waves
was conducted for the various damping and stiffness settings.

For an initial setting of damper valve closed and the stiffness valve open, the testing resulted in a maximum loading of 8 kips. Refer to figures 3.7, 3.8, 3.9, and 3.10.

The effects of the "hi/lo" stiffness capabilities of the prototype damper are evident. Depending on the spring accumulator valve setting and the frequency of the cyclic test, the sharpness and the extent to which the spring accumulator effects the system can be determined. This is a direct result of where the spring accumulator valve has been closed, which determines the maximum spring force produced in the system, and the amount of variable damping present in the system in figures 3.11 to 3.30.

Evaluation of the 2 inch setting, i.e. the spring accumulator valve is closed at the midpoint of the stroke, establishes an early increase in stiffness during the compression stage of the cycle. As damping levels are increased this becomes more apparent in the progressive evolution of the force vs.
CHAPTER 3. PROTOTYPE VARIABLE FLUID DAMPER AND STIFFNESS DEVICE

<table>
<thead>
<tr>
<th>Compression Velocity</th>
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<th>Damping Valve</th>
<th>Amplitude</th>
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<td>inches/sec.</td>
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<td>0.25</td>
<td>Closed</td>
<td>Closed</td>
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<tr>
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Table 3.1: Ramp Pulse Excitation Tests Conducted on Prototype Damper


The 3 inch setting, i.e. the spring valve is closed when the piston is extended 3 inches from the fully compressed position and then returned to the neutral stroking position of 2 inches, reflects the transition of the spring accumulator’s effectiveness in the tension side of the cyclic motion. This extra inch in extension of the piston allows for a greater response from the prototype damper in the form of additional spring force. The plateau effects experienced at the 3 inch setting are far less evident, and thus result in the increased stiffness in the tension side of the cyclic motion. Refer to figures A.9, A.10, A.11, A.12, A.25, A.26, A.27, A.28, A.41, A.42, A.43, A.44, A.57, A.58, A.59, A.60, A.73, A.74, A.75, and A.76 in the Appendix.

With the stiffness accumulator closed at the 4 inch mark, full extension of the stroke to develop maximum stiffness, and the damping valve still closed, the maximum force achieved by the damper is approximately 12 kips. Refer to figures 3.11, 3.12, 3.13, and 3.14. Note that closing the spring
valve at full extension and then returning the piston to the "neutral stroking position", pre-loads the system with 3 kips of force. Further, in situations of full or near full damping forces present, a "re-bounding" effect is experienced in the system on the tension direction of motion of the damper. The net effect is in an increased capability experienced on the extension phase of the loading that meets or exceeds that of the compression phase. Pre-loading occurs when the valve is closed at any position beyond the two inch neutral mark. For example, at the 3 inch mark, when the valve is closed at the point for which the damper piston is extended 3 inches from the fully compressed position, the system is preloaded to approximately 1 kip (refer to previously mentioned plots). With the stiffness accumulator valve closed at the 4 inch mark and the damping valve at 1 complete revolution, the maximum force achieved by the damper is approximately 5 kips. Refer to figures 3.15, 3.16, 3.17, and 3.18. With the stiffness accumulator valve closed at the 4 inch mark and the damping valve at 2 complete revolution, the maximum force achieved by the damper is approximately 2.75 kips. Refer to figures 3.19, A.46, 3.21, and 3.22. With the stiffness accumulator valve closed at the 4 inch mark and the damping valve at 3 complete revolution, the maximum force achieved by the damper is approximately 2 kips. Refer to figures 3.23, 3.24, 3.25, and 3.26. With the stiffness accumulator valve closed at the 4 inch mark and the damping valve at 4 complete revolution, the maximum force achieved by the damper is approximately 1.75 kips. Refer to figures 3.27, 3.28, 3.29, and 3.30.

The second stage of testing involved using ramp signals of various velocities and amplitudes. In order to protect the prototype damper, all ramp waves were centered about the neutral stroking position, as illustrated by the ramp pulse in figure 3.6.

Two tests were conducted to determine the spring accumulator's strength at a velocity of 1/4 inch per second. With the damping valve closed and the accumulator valve open, the prototype damper produced approximately 1/2 kip of force, and with the spring accumulator closed at full stroke, the prototype damper produced approximately 9 1/2 kips of force, refer to figure 3.31.
For the high velocity ramp loadings, two settings were tested. Refer to table 3.1. For both velocity settings, a peak value of approximately 12 kips is produced. Refer to figures 3.32 and 3.33

3.6 Conclusions

In summary, the testing procedures and setup were designed with the prototype device in mind. Tests were conducted at various amplitudes and frequencies for various loading waves. The data was collected with the assistance of dSPACE and plotted via a MATLAB program to analyze the results. After the completion of tests, the following conclusions were drawn.

Upon completion of numerous testings, the ability to alter the variable damping and stiffness properties of the prototype damper proved quite effective. The maximum force obtained from cyclic loading was approximately 9 kips. The prototype damper demonstrated an approximate 12 kip ability with the application of a high speed dynamic ramp loading. Significant changes in the hysteresis loops were clearly demonstrated with the application of the variable stiffness and variable damping settings.

Note that special care had to be taken with respect to the attachment of the prototype damper, because the damper having a small radius for its piston arm and the exclusion of any moment sleeve in the design to compensate for any mis-alignment is a drawback.
**Figure 3.7:** Full Sine Pulse with Damping Valve Closed and Stiffness Valve Open

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**Figure 3.8:** Full Sine Pulse with Damping Valve Closed and Stiffness Valve Open
Figure 3.9: Full Sine Pulse with Damping Valve Closed and Stiffness Valve Open

Figure 3.10: Full Sine Pulse with Damping Valve Closed and Stiffness Valve Open
Figure 3.11: Full Sine Pulse with Damping Valve Closed and Stiffness Valve Closed at 4 inches

Figure 3.12: Full Sine Pulse with Damping Valve Closed and Stiffness Valve Closed at 4 inches
Figure 3.13: Full Sine Pulse with Damping Valve Closed and Stiffness Valve Closed at 4 inches

Figure 3.14: Full Sine Pulse with Damping Valve Closed and Stiffness Valve Closed at 4 inches
**Figure 3.15:** Full Sine Pulse with Damping Valve at 1 Revolution and Stiffness Valve Closed at 4 inches

**Figure 3.16:** Full Sine Pulse with Damping Valve at 1 Rev. and Stiffness Valve Closed at 4 inches
Figure 3.17: Full Sine Pulse with Damping Valve at 1 Revolution and Stiffness Valves Closed at 4 inches

Figure 3.18: Full Sine Pulse with Damping Valve at 1 Revolution and Stiffness Valve Closed at 4 inches
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Figure 3.19: Full Sine Pulse with Damping Valve at 2 Revolutions and Stiffness Valve Closed at 4 inches

Figure 3.20: Full Sine Pulse with Damping Valve at 2 Revolutions and Stiffness Valve Closed at 4 inches
Figure 3.21: Full Sine Pulse with Damping Valve at 2 Revolutions and Stiffness Valves Closed at 4 inches

Figure 3.22: Full Sine Pulse with Damping Valve at 2 Revolutions and Stiffness Valve Closed at 4 inches
Figure 3.23: Full Sine Pulse with Damping Valve at 3 Revolutions and Stiffness Valve Closed at 4 inches

Figure 3.24: Full Sine Pulse with Damping Valve at 3 Revolutions and Stiffness Valve Closed at 4 inches
Figure 3.25: Full Sine Pulse with Damping Valve at 3 Revolutions and Stiffness Valves Closed at 4 inches

Figure 3.26: Full Sine Pulse with Damping Valve at 3 Revolutions and Stiffness Valve Closed at 4 inches
Figure 3.27: Full Sine Pulse with Damping Valve at 4 Revolutions and Stiffness Valve Closed at 4 inches

Figure 3.28: Full Sine Pulse with Damping Valve at 4 Revolutions and Stiffness Valve Closed at 4 inches
**Figure 3.29:** Full Sine Pulse with Damping Valve at 4 Revolutions and Stiffness Valve Closed at 4 inches

**Figure 3.30:** Full Sine Pulse with Damping Valve at 4 Revolutions and Stiffness Valve Closed at 4 inches
Figure 3.31: Results of Ramp Loading at 1/4 inch per second

Figure 3.32: Ramp Pulse of 15 inch/sec at ±1 inch with Damping Valve and Stiffness Valve Closed
Figure 3.33: Ramp Pulse with Compression Velocity of 21 inch/sec and Tension Velocity of 10 inch/sec at ±1 3/4 inch with Damping Valve Closed and Stiffness Valve Open
Chapter 4

Scissor-Jack Energy Dissipation System

4.1 Background

With the adoption of damping devices to protect structural systems from wind and earthquake induced vibrations, designers have attempted to optimize their results. In such countries as Japan, yielding steel devices and viscoelastic fluid or solid devices are mostly utilized. In the United States, designers have focussed primarily on fluid viscous devices. Such applications are either installed in-line with diagonal bracing or as horizontal elements atop chevron bracing or toggle bracing schemes.

Although these devices can supply adequate protection from wind and earthquake vibrations, often their requirements of entire bays in frames and frequent violation of architectural requirements make them unattractive to designers. It is with this in mind, that Constantinou and Sigaher [68] created the "Scissor-Jack-damper" system. Their design combines the displacement magnification feature with the availability of smaller sized devices. This is achieved through the compactness of the Scissor-Jack dissipation device and its near-vertical instillation [68].

4.1.1 Theory

In conventional diagonal and chevron brace configurations, the displacement of the energy dissipation devices is either less than (as in diagonal bracing) or equal to (as in chevron bracing) the drift of the story at which the devices are installed. This can be expressed in the following form.

\[ u_D = f \cdot u \]  

(4.1)
Figure 4.1: Test set-up used by Constantinou and Sigaher [68]

where \( u \) = interstory drift

\[ u_D = \text{damper relative displacement} \]

and \( f \) = magnification factor

The chevron brace configuration has a magnification factor equal to 1.0, and the diagonal brace configuration has a factor equal to \( \cos \theta \), where \( \theta \) is representative of the angle of inclination of the damper with respect to the horizontal component of the damper force exerted on the frame. In other words,

\[ F = f \cdot F_D \]

Figure 4.2 illustrates the various bracing schemes along with the applied force, \( F \), and the interstory drift, \( u \). By considering that the illustrated single-story structure has an effective weight, \( W \), and a
**CHAPTER 4. SCISSOR-JACK ENERGY DISSIPATION SYSTEM**

<table>
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<tr>
<th>Bracing Scheme</th>
<th>Equation</th>
<th>Parameters</th>
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<tr>
<td>Diagonal</td>
<td>( f = \cos \theta )</td>
<td>( \theta = 37^\circ ), ( f = 0.80 ), ( \beta = 0.63 )</td>
</tr>
<tr>
<td>Chevron</td>
<td>( f = 1.00 )</td>
<td>( f = 1.00 ), ( \beta = 0.65 )</td>
</tr>
<tr>
<td>Scissor-Jack</td>
<td>( f = \frac{\cos \varphi}{\tan \theta} )</td>
<td>( \varphi = 9^\circ, \theta = 70^\circ ), ( f = 2.16 ), ( \beta = 0.23 )</td>
</tr>
<tr>
<td>Upper Toggling</td>
<td>( f = \frac{\sin \theta_2}{\cos(\theta_1 + \theta_2)} + \sin \theta_1 )</td>
<td>( \theta_1 = 31.9^\circ, \theta_2 = 43.2^\circ ), ( f = 3.191 ), ( \beta = 0.509 )</td>
</tr>
<tr>
<td>Reverse Toggling</td>
<td>( f = \frac{\cos \theta_1}{\cos(\theta_1 + \theta_2)} + \cos \theta_2 )</td>
<td>( \theta_1 = 30^\circ, \theta_2 = 49^\circ, \alpha = 0.7 ), ( f = 2.521 ), ( \beta = 0.318 )</td>
</tr>
</tbody>
</table>

**Figure 4.2:** Illustrations of diagonal bracing schemes, magnification factors, and damping ratios of a single-story structure with linear fluid viscous devices. [68]

The fundamental period under elastic conditions, \( T \), and that it is equipped with a linear fluid viscous damper of the following properties:

\[
F_D = C_0 \cdot \dot{u}_D
\]  

(4.3)

for which \( C_0 \) is the damping coefficient and \( \dot{u}_D \) is the relative velocity between the ends of the damper along its axis. The resulting damping force, \( F \), exerted on the frame by the damper assembly is the following:
$F = C_0 \cdot f^2 \cdot \dot{u}$  \hspace{1cm} (4.4)

such that $\dot{u}$ is the interstory velocity. From this the damping ratio of a single-story frame with a linear fluid viscous device can be stated as the following:

$$\beta = \frac{C_0 \cdot f^2 \cdot g \cdot T}{4 \cdot \pi \cdot W}$$  \hspace{1cm} (4.5)

As seen in equation 4.5, the damping ratio varies proportionally to the square of the magnification factor. Further, as illustrated in figure 4.2, both the Scissor-Jack and toggle bracing schemes have the ability to achieve magnification factors greater than 1.0.

As a result of the magnification mechanisms present in the Scissor-Jack system, the utility of fluid viscous devices to stiff structural systems under seismic excitation and/or structures subjected to wind loadings all of which are characterized by small interstory drifts and velocities become viable. Further, the system may be configured or arranged in such a way to allow for open spaces and/or minimal obstruction of views, all of which are fundamental design aspects considered by architects.

### 4.1.2 Magnification Factor and Forces

The magnification factor, $f$, is fundamental in the effectiveness of the Scissor-Jack device. Figure 4.3 illustrates the movement along with the forces present in a single-story frame with a Scissor-Jack system.

Using figure 4.3 for reference, the magnification factor can be re-written as follows

$$f = \frac{u_D}{u} = \frac{\overline{X'B'} - \overline{AB}}{u}$$  \hspace{1cm} (4.6)

where $\overline{AB}$ and $\overline{X'B'}$ represent the initial and deformed lengths of the damper, respectively. This analysis neglect taking into effect any deformations in the frame as a result of only considering rigid
body motion and any reduction in height resulting from rotation in the columns. As a result of rigid body kinematics, the damper displacement can be expressed as:

\[ u_D = |A'B' - AB| = \pm 2 \cdot \ell_1 \cdot [\sin (\theta \pm \Delta \theta) - \sin \theta] \]  

(4.7)

where \( \Delta \theta \) represents the angle of rotation of the scissor-braces. To maintain lengths between points C and D

\[ 2 \cdot \ell_1 \cdot \cos (\theta \pm \Delta \theta) = 2 \cdot \ell_1 \cdot \cos \theta \mp u \cdot \cos \psi \]  

(4.8)
from equations 4.7 and 4.8, the damper displacement can be manipulated to the following form.

\[
    u_D = 2 \cdot \xi_1 \cdot \left[ \sin \left( \cos^{-1} \left( \cos \theta + \frac{\cos \psi}{2 \cdot \xi_1} \cdot u \right) \right) - \sin \theta \right]
\]  

(4.9)

Noting for reference that equations 4.7 to 4.9 have positive signs holding for drift towards the right and for damper extension (\(u_D > 0\)); the equations become valid with negative signs with respect to drift towards the left.

### 4.2 Set-up

As illustrated in the schematics of figure A.92, the test system for the Scissor-Jack Dissipation Device consists of a frame, W8x24 columns at 12 feet in height and a W8x28 as the cross member completing the frame. Sizes of the frame members were chosen such that maximum flexibility could be achieved. The same actuator and load cell from the shake table set up were mounted to a support column approximated 3 feet away in the plane of the frame, figure 4.4(a). This enables excitation of the frame to occur at the top of the frame story, 12 feet from the floor, compared to the ground used by Constantinou and Sigaher [68].

The Scissor-Jack consists of four legs, each made from TS4x4x1/4, with connection endings machined from solid 3 inch square stock. The end pieces, once fabricated are slid into the tubes and then welded to assure proper connection strength. The four legs connect to the prototype device at a joint detail represented in figures A.94 and 4.4(g). Assembly is maintained by a 1 inch diameter bolt, assuring a pinned joint (refer to figures A.94 and 4.4(h)). Placement of the Scissor-Jack device into the frame can be referenced by figures A.94, A.93, and A.92. All other details regarding individual elements of the test system can be found in the appendix under subsection A.2.
Figure 4.4: Testing Structure of Scissor-Jack Energy Dissipation Device
4.3 Test Results

Before the Scissor-Jack Dissipation Device can be applied to the structural frame, an initial test is conducted to determine the required forces to move the frame the desired 1/4 inch used in the following tests. From the force vs. displacement plot of the steel frame, shown in figure 4.5, significant friction can be observed.

![Steel Frame Subjected to Full Sine Pulse, Freq: 0.50 Hz](image)

Figure 4.5: Structural Frame Subjected to Full Sine Pulse of 1/2 Hz.

A theoretical amplification factor of 4.18 is calculated using $\psi = 42.95$ and $\theta = 9.94$ in the following equation.

$$f = \frac{\cos \psi}{\tan \theta} \quad (4.10)$$

Thus, an approximate displacement of 1 inch in the damper should correlate to 1/4.2 inch displacement in the frame. Further it can be stated that the forces in the frame should be that of approximately 4.2 times those experienced by the damper.
It is shown in the following data plots that amplification properties can be achieved in a less than perfect system. Such factors as friction in the moment brace and Scissor-Jack constructed for the prototype damper, irregularities resulting from manufacturing, construction, and individual member properties in the system, and less than perfect performance of system members result in at times a near perfect approximate amplification factor and at other times smaller but acceptable values of amplification.

### 4.3.1 Spring Valve Open

Figures 4.6 through 4.14 are given examples of various frequencies with various damping valve settings and an open spring valve.

Notice is given to the large amounts of frictional damping present in the system, refer to figures 4.6 through 4.8 with the system’s total output producing 5 to 5 1/2 kips of force. Further, it is dually noted that this set of experiments are relatively close in achieving optimum equal displacements in both the tension and compression side of the cycle. It can be further noted that in all of the 1/2 Hz. frequency testings that frictional damping participated greatly in the global system’s energy dissipation.

Upon entering the middle frequency testing range of 1 Hz., frictional damping becomes less apparent and a more elliptical energy dissipation loop is achieved producing approximately 7 kips of force, as in figures 4.9 through 4.11.

At 1 1/2 Hz. the Scissor-Jack system, with different valve positions, is able to produce in the range of 8 to 10 kips of force. It should also be noted that the greatest disparity between directional displacements occurs during the highest velocity testing. Further, reference is given to the previous
Figure 4.6: Scissor-Jack Subjected to Full Sine Pulse of 1/2 Hz. with Damping Valve at 2 Revolutions and Stiffness Valve Open

Chapter 3's results of the damping valve's contribution to the system's ability to obtain higher forces. This ability produces greater force levels that can be implied to the system given a correlation to the above mentioned chapter's test results and the current system's ability to amplify the prototype damper's properties. Under ideal circumstances, such results can be produced, but given circumstances out of the control of the researcher, forces are experienced that exceeded the ability of the actuator's support column's ability to remain rigid and thus giving data not reflective of the tests. Usually this events occur on average at 2, 1, and closed positions of the variable damping valve.
Figure 4.7: Scissor-Jack Subjected to Full Sine Pulse of 1/2 Hz. with Damping Valve at 3 Revolutions and Stiffness Valve Open

Figure 4.8: Scissor-Jack Subjected to Full Sine Pulse of 1/2 Hz. with Damping Valve at 4 Revolutions and Stiffness Valve open
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4.3.2 Spring Valve Closed At 2 Inches

As noted before, frictional forces are evident in the 1/2 Hz. frequency testings with the spring valve closed at 2 inches. Frictional forces produce a hysteretic energy dissipation loop in contrast to the normal elliptical shape normally experienced by the damper in solo testing. Compressive forces are in the range of 6 1/2 to 7 kips at maximum variable damping flow.

For both 1 and 1 1/2 Hz., compressive forces are still on the order of 6 1/2 to 7 kips. A more elliptical shape is produced by the energy loops.
Figure 4.15: Scissor-Jack Subjected to Full Sine Pulse of 1/2 Hz. with Damping Valve at 2 Revolutions and Stiffness Valve Closed at 2 inches

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Figure 4.23: Scissor-Jack Subjected to Full Sine Pulse of 1/2 Hz. with Damping Valve at 2 Revolutions and Stiffness Valve Closed at 4 inches

4.3.3 Spring Valve Closed At 4 Inches

Upon closing the spring valve at 4 inches the system becomes predictably characteristic with it upward angled parallelogram characteristic of non-linear systems. Unfortunately, desired displacements magnitudes in both the compression and tension sides of the cycle are not accomplished, but a more balanced behavior is.
**Figure 4.24:** Scissor-Jack Subjected to Full Sine Pulse of 1/2 Hz. with Damping Valve at 3 Revolutions and Stiffness Valve Closed at 4 inches

**Figure 4.25:** Scissor-Jack Subjected to Full Sine Pulse of 1/2 Hz. with Damping Valve at 4 Revolutions and Stiffness Valve Closed at 4 inches
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4.3.4 Conclusion

Test results confirm the effectiveness of the damper in the Scissor-Jack Energy Dissipation System, with its ability to use smaller dissipation devices in association with amplifying braces to perform the equivalent work of much larger and expensive devices.

Unfortunately, primarily resulting from the inability of the actuator support column to remain perfectly rigid during testing the full potential of the damper could not be achieved. Improvement of the damping device connections are necessary for enhanced performance.
Chapter 5

Analytical Model

5.1 The Maxwell Model

The Maxwell model is a simplified analytical model of a variable damping device installed in a structure by connecting it with a brace or a wall. This model is represented by a variable dashpot and a variable linear spring in series. The motion equation of the Maxwell model is as follows:

\[ \frac{\dot{F}}{k} + \frac{F}{C} = \ddot{u} \quad (5.1) \]

where \( F \) = damper force

and \( \ddot{u} \) = velocity across the Maxwell model

Re-arranging equation 5.1 into the following form,

\[ F + \frac{C}{k} \dot{F} = C \ddot{u} \quad (5.2) \]

allows for the application of the Fourier Transform to the constitutive equation. The with the application and simplification of the following property

\[ \mathfrak{F} \left\{ \frac{df}{dt} \right\} = (i\omega) \mathfrak{F} \{ f \} \quad (5.3) \]

produces the following result.

\[ \mathfrak{F} \{ F \} = [\tilde{K}_1(\omega) + i\tilde{K}_2(\omega)] \mathfrak{F} \{ u \} \quad (5.4) \]
where

\[ \tilde{K}_1(\omega) + i\tilde{K}_2(\omega) = \tilde{K}(\omega) = \frac{\sum_{n=0}^{N} b_n (i\omega)^n}{\sum_{m=0}^{M} a_m (i\omega)^m} \]  \hspace{1cm} (5.5)

\( \tilde{K}_1 \) represents the "in-phase" storage stiffness while \( \tilde{K}_2 \) represents the "out-of-phase" loss stiffness of the described system. The equivalent viscous damping constant for the system then becomes the following.

\[ \tilde{C}(\omega) = \frac{\tilde{K}_2(\omega)}{\omega} \]  \hspace{1cm} (5.6)

In theory, the system is examined under a steady-state condition of the following parameters.

\[ u = u_0 \sin(\omega t) \]  \hspace{1cm} (5.7)

\[ F = F_0 \sin(\omega t + \delta) \]  \hspace{1cm} (5.8)

where \( \delta \) represents the phase difference in the system. It can be stated that the energy dissipated per cycle will is determined by the following equation.

\[ W_D = \int F du = \pi F_0 u_0 \sin \delta \]  \hspace{1cm} (5.9)

Upon deduction equation 5.8 can be transformed into the following:

\[ F = F_0 \sin \omega t \cos \delta + F_0 \cos \omega t \sin \delta \]  \hspace{1cm} (5.10)

It can be further manipulated into the much more simplified form of the following:

\[ F = \tilde{K}_1 u_0 \sin \omega t + \tilde{K}_2 \cos \omega t \]  \hspace{1cm} (5.11)
where

\[ K_1 = \frac{F_0}{u_0} \cos \delta \]  \hspace{1cm} (5.12)

is the simplified storage stiffness and

\[ K_2 = \frac{F_0}{u_0} \sin \delta \]  \hspace{1cm} (5.13)

is the simplified loss stiffness of the system. For any arbitrary load \( F_1 = F_0 \sin \delta \), the phase difference can be solved for.

\[ \delta = \sin^{-1} \left( \frac{F_1}{F_0} \right) \]  \hspace{1cm} (5.14)

Now the phase difference is applied to equations 5.12 and 5.13 in order to solve the elastic stiffness of the system.

\[ K_0 = \frac{F_0}{u_0} = \sqrt{K_1^2 + K_2^2} \]  \hspace{1cm} (5.15)

### 5.2 Prototype Analysis

The application of Maxwell Theory is carried out in the following presented documentation through a step by step process. First, after the collection of all test data, data entries a chronologicalized so that can be used to scrutinized based upon the major characteristics of stiffness and damping. Next, coefficients are calculated by equations discussed in the previous section. Once the coefficients for damping and stiffness are calculated, plots of peak force vs. velocity and stiffness vs. displacement are used to determine if the proper assumptions regarding the prototype damper's ability to be modeled as a maxwell element. The coefficients are used to create an analytical model of the system.
CHAPTER 5. ANALYTICAL MODEL

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Frequency = 0.5 Hz, Amplitude = 1", Damping Valve at 4 Revolutions and Full Stiffness
Chapter 6

Conclusions

As demonstrated in the given study, the prototype variable fluid damper and stiffness device, has many promising attributes that are beneficial. With its variable damping valve and stiffness valve the characteristics of the system can be changed in a passive system to meet the desired needs of the designer. With its maximum cyclic force loading being approximately 9 kips and approximate 12 kip ability with the application of a high speed dynamic ramp loading, significant changes in the hysteresis loops are clearly demonstrated with the application of the variable stiffness and variable damping settings. This can prove beneficial in the aspect of one device doing the work of several complicated devices.

The incorporation of the damper in a Scissor-Jack system amplifies its effectiveness by, in our specific case, nearly four times. The system would prove to be an effective means for response reduction of structures under earthquake and wind excitations.
Bibliography


Appendix A

A.1 Prototype Variable Damping and Stiffness Device

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Figure A.78: Full Sine Pulse with Damping Valve at 4 Revolutions and Stiffness Valve Closed at 4 inches

Figure A.79: Full Sine Pulse with Damping Valve at 4 Revolutions and Stiffness Valve Closed at 4 inches
Figure A.80: Full Sine Pulse with Damping Valve at 4 Revolutions and Stiffness Valve Closed at 4 inches

Figure A.81: Ramp Pulse with Compression Velocity of 21 inch/sec and Tension Velocity of 10 inch/sec at ±1/4 inch with Damping Valve Closed and Stiffness Valve Open
Figure A.82: Ramp Pulse with Compression Velocity of 21 inch/sec and Tension Velocity of 10 inch/sec at ±1/2 inch with Damping Valve Closed and Stiffness Valve Open

Figure A.83: Ramp Pulse with Compression Velocity of 21 inch/sec and Tension Velocity of 10 inch/sec at ±3/4 inch with Damping Valve Closed and Stiffness Valve Open
Figure A.84: Ramp Pulse with Compression Velocity of 21 \textit{inch/sec} and Tension Velocity of 10 \textit{inch/sec} at \( \pm 1 \text{ inch} \) with Damping Valve Closed and Stiffness Valve Open

Figure A.85: Ramp Pulse with Compression Velocity of 21 \textit{inch/sec} and Tension Velocity of 10 \textit{inch/sec} at \( \pm 1 \ 1/4 \text{ inch} \) with Damping Valve Closed and Stiffness Valve Open
Figure A.86: Ramp Pulse with Compression Velocity of 21 inch/sec and Tension Velocity of 10 inch/sec at ±1 1/2 inch with Damping Valve Closed and Stiffness Valve Open

Figure A.87: Ramp Pulse with Compression Velocity of 21 inch/sec and Tension Velocity of 10 inch/sec at ±1 3/4 inch with Damping Valve Closed and Stiffness Valve Open
Figure A.88: Ramp Pulse of 15 inch/sec at ±1/4 inch with Damping Valve and Stiffness Valve Closed

Figure A.89: Ramp Pulse of 15 inch/sec at ±1/2 inch with Damping Valve and Stiffness Valve Closed
Figure A.90: Ramp Pulse of 15 inch/sec at ±3/4 inch with Damping Valve and Stiffness Valve Closed

Figure A.91: Ramp Pulse of 15 inch/sec at ±1 inch with Damping Valve and Stiffness Valve Closed
A.2 Scissor-Jack Dissipation Device
Figure A.92: Scissor-Jack Dissipation Device Incorporated into Steel Frame
Figure A.93: Testing Structure of Scissor-Jack Energy Dissipation Device
Figure A.94: Preliminary Structural Details for Scissor-Jack Device System
A.3 Analytical Models

A.3.1 Prototype Damper

![Graph showing peak force vs. velocity for prototype damper.](image)

**Figure A.95**: Experimental and Analytical Values of Peak Force Vs. Velocity for Prototype Damper
Figure A.96: Experimental and Analytical Values of Peak Force Vs. Velocity for Prototype Damper

Figure A.97: Experimental and Analytical Values of Peak Force Vs. Velocity for Prototype Damper
Figure A.98: Experimental and Analytical Values of Peak Force Vs. Velocity for Prototype Damper

Figure A.99: Experimental and Analytical Values of Peak Force Vs. Velocity for Prototype Damper
Figure A.100: Experimental and Analytical Values of Damping and Storage Stiffness Coefficients for Prototype Damper

Figure A.101: Experimental and Analytical Values of Damping and Storage Stiffness Coefficients for Prototype Damper
Figure A.102: Experimental and Analytical Values of Damping and Storage Stiffness Coefficients for Prototype Damper

Figure A.103: Comparison of Analytical and Experimental Force-Displacement loop: Frequency = 0.5 Hz, Amplitude = 1/2”, Damping Valve Closed and Full Stiffness
Figure A.104: Comparison of Analytical and Experimental Force-Displacement loop:
Frequency = 1.0 Hz., Amplitude = 1/2", Damping Valve Closed and Full Stiffness

Figure A.105: Comparison of Analytical and Experimental Force-Displacement loop:
Frequency = 1.5 Hz., Amplitude = 1/2", Damping Valve Closed and Full Stiffness
Figure A.106: Comparison of Analytical and Experimental Force-Displacement loop: Frequency = 2.0 Hz, Amplitude = 1/2", Damping Valve Closed and Full Stiffness

Figure A.107: Comparison of Analytical and Experimental Force-Displacement loop: Frequency = 2.5 Hz, Amplitude = 1/2", Damping Valve Closed and Full Stiffness
Figure A.108: Comparison of Analytical and Experimental Force-Displacement loop:
Frequency = 3.0 Hz., Amplitude = 1/2", Damping Valve Closed and Full Stiffness
Figure A.109: Comparison of Analytical and Experimental Force-Displacement loop of Scissor-Jack: Frequency $= 1/2 \text{ Hz}$, Damping Valve at 2 Revolutions and Stiffness Valve Open

A.3.2 Analytical Model: Scissor-Jack
Figure A.110: Comparison of Analytical and Experimental Force-Displacement loop of Scissor-Jack: Frequency $= 1/2 \text{ Hz}$, Damping Valve at 3 Revolutions and Stiffness Valve Open

Figure A.111: Comparison of Analytical and Experimental Force-Displacement loop of Scissor-Jack: Frequency $= 1/2 \text{ Hz}$, Damping Valve at 4 Revolutions and Stiffness Valve Open
Figure A.112: Comparison of Analytical and Experimental Force-Displacement loop of Scissor-Jack: Frequency = 1 Hz, Damping Valve at 2 Revolutions and Stiffness Valve Open

Figure A.113: Comparison of Analytical and Experimental Force-Displacement loop of Scissor-Jack: Frequency = 1 Hz, Damping Valve at 3 Revolutions and Stiffness Valve Open
**Figure A.114:** Comparison of Analytical and Experimental Force-Displacement loop of Scissor-Jack: Frequency $= 1$ Hz, Damping Valve at 4 Revolutions and Stiffness Valve Open

**Figure A.115:** Comparison of Analytical and Experimental Force-Displacement loop of Scissor-Jack: Frequency $= 1 1/2$ Hz, Damping Valve at 2 Revolutions and Stiffness Valve Open
Figure A.116: Comparison of Analytical and Experimental Force-Displacement loop of Scissor-Jack: Frequency = 1 1/2 Hz., Damping Valve at 3 Revolutions and Stiffness Valve Open

Figure A.117: Comparison of Analytical and Experimental Force-Displacement loop of Scissor-Jack: Frequency = 1 1/2 Hz., Damping Valve at 4 Revolutions and Stiffness Valve Open