TEST METHODOLOGY AND PROCEDURES FOR FLUID VISCOUS DAMPERS USED IN STRUCTURES TO DISSIPATE SEISMIC ENERGY

by

D.P. Taylor
Taylor Devices, Inc.
90 Taylor Drive
North Tonawanda, NY 14120

M.C. Constantinou
Department of Civil Engineering
State University of New York
Buffalo, NY 14260

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1.0 INTRODUCTION

Taylor Devices, Inc. has manufactured damping devices for use in engineered structures since 1955. Until 1990, most of our applications were for the U.S. Military, who often uses dampers to attenuate weapons effects. These military concepts, applications, and damper designs have not been publicized, due to Government security restrictions. However, with the end of the Cold War, most of these restrictions have been relaxed, and much of this damping technology is now available to the structural engineering community for use in seismic energy dissipation applications. This technology encompasses both fluid viscous dampers and various types of spring-damper elements.

Several technical reports have been published within the past two years discussing the test performance of fluid viscous dampers and related products applied within scaled building and bridge models (Reference 1). This paper addresses a frequently raised question concerning large fluid dampers:

"The structural engineering community now has access to compact fluid viscous dampers in the 100 kip to 2,000 kip output range. What testing methods are possible and appropriate for full sized devices?"

Please note that information is presented within this report concerning test levels and methodology used by the U.S. Government to shock test various structures. In some cases, hard numbers have been replaced by qualitative terminology to safeguard information restricted by the United States Department of Defense. All information herein related to weapons grade shock and shock testing is unclassified, for unlimited distribution, having been obtained solely from authorized open source publications.

2.0 DISCUSSION OF TEST METHODOLOGY FOR DAMPING DEVICES

2.1 PRIOR COMMERCIAL PRACTICE

Historically, the component testing methods used by the structural engineering community to evaluate seismic energy dissipation devices include:

1. Cyclic testing with a controlled servo-hydraulic actuator at the full scale damper design level.

2. Cyclic testing with a controlled servo-hydraulic actuator at the fractionally scaled level, prorating performance to the full scale by analysis.

2.2 PRIOR PRACTICE FOR DEFENSE APPLICATIONS

The defense community has taken a far different approach to component level testing, influenced strongly by the extreme severity of a weapons effect pulse. These approaches include:

1. Drop hammer testing at the full scale level.
2. Explosive actuator testing at the full scale level.
3. Unconfined detonation of explosives at the full scale level.

The methodology used by the military on full scale structures usually requires full scale system testing. This can be compared to commercial practice where structures that are equipped with seismic design improvements can be field inspected and evaluated only if an earthquake occurs of suitable magnitude to exercise the design. The military's approach is different, because hardware must be fully proven to meet stringent shock survivability specifications before being put into service. Because of this, it is common to test full sized, operational structures, with human occupants, under full scaled detonation of explosives. In the case of structures hardened against nuclear attack, treaty requirements against nuclear weapons testing have greatly restricted testing, so the detonation of very large charges of specially constructed conventional explosives are often used as a substitute.

Particular attention to weapons effect testing is taken by the U.S. Navy, who must "go in harm's way" much more frequently than other service branches. For this reason, Navy shock testing involves multiple steps, including:

1. <u>Component Level Testing</u> by drop hammer.

2. <u>Subsystem Testing</u> by explosives detonation on the U.S. Navy Floating Shock Platform (FSP).

3. <u>Ship Shock Trials</u>, usually run on the first ship of each new class. These involve the detonation of large, sub-surface moored depth bombs, each containing several tons of high explosives, at various "close-in" positions near the ship. The ship is manned for shock trial testing, with both Naval personnel and contractor representatives. Peak translational velocities for this test are in excess of 60 in/sec.

4. <u>Shock Training Exercises</u>, where <u>each</u> new ship, fully operational and fully manned, is required to experience at least one "high-level" shock; being exposed to the detonation of the same depth bomb used at the Ship Shock Trials.

A good engineering perspective can be obtained by comparing free-field transients representative of military systems to those typically experienced by buildings and bridges subjected to seismic inputs.

For example:

Figure 1 is a velocity-time history of the El Centro Earthquake, at the 1.0 G level, considered a relatively severe seismic transient input.

Figure 2 is an unclassified velocity-time history from the FSP, at a test level often used to shock qualify a component or system for shipboard use.

Figure 3 is an unclassified free field velocity-time history for a land based structure subjected to the detonation of a large nuclear weapon.

Observation of Figures 1-3 immediately reveals several interesting comparisons between seismic motion and weapons grade shock transients. In general, weapons inputs are far higher in velocity and occur much faster in time than a seismic event. Initial accelerations are very high, often in the range of 500-1,000 G, to cause maximum damage to rigid structures. Significant velocity is associated with these accelerations, to cause damage even to objects isolated with energy dissipation or spring type elements. Using an extremely long stroke base isolation system is not feasible on military systems, since the existence of the isolation device can often be determined by enemy reconnaissance activities prior to attack. Long travel “soft” isolators are easily defeated by the simple expedient of using two weapons instead of one, with a slight time lag between detonations. The first detonation strokes the isolation device substantially. The second detonation over-travels the isolators to their stops allowing damage to occur. The alternative approach to military shock isolation is to “rigidize” or “shock harden” a structure, by making it heavy, very stiff, and strong. This method is often impractical, due to both high cost and the very high initial acceleration of a weapons pulse, which serves to substantially excite stiff structural modes. Open source shock spectra are not available for this report, however it can be stated that the 500-1,000 G accelerations of weapons grade shock occurs over a relatively broad band frequency range. Within the defense community, a shock hardened or rigidized structure is said to be one that is intended to “fight the shock pulse.” A rigidized design can be undertaken only if the design activity has a <u>very</u> exact data base, encompassing every possible weapon that could be used against the structure.

Also of importance to the military are components of a weapons detonation that occur in addition to the free field displacements noted here. These include blast over-pressure, high velocity fragments, heat, cratering, localized structural penetration, gaseous combustion by-products, and radiation. Often, a very specific evolution of a structural design occurs to address changing parameters of the weapon of choice. Some parallels exist to this in the seismic design community, where the discovery of a new fault, or occurrence of a specific severe earthquake will be cause for upgrade or update of the design level transients and/or spectra for a specific region or site.

An excellent example of structural and weapons evolution in the military can be seen in the design of main battle tanks over the period from World War I to the present. During World War I, tanks were defeated relatively easily with conventional high explosives. Detonation would cause structural failure, or produce a crater that the tank could not cross without getting stuck. The period between World War I and World War II saw tank designers add improved armor, large engines, and better track wheel suspensions to counter the effects of high explosives. During World War II, weapons designers countered these improved tanks by developing the so-called “shaped charge” warhead. This relatively small war head literally “burned through” armor plate in a very small localized area.

If aimed at the tank’s turret, the shaped charge fully penetrated the armor and spewed molten metal plasma and fragments throughout the turret. After World War II, tank designers countered the shaped charge warhead by developing so-called “reactive armor,” an external applique to the tank that was in itself an explosive material of relatively low power. When impacted by a shaped charge, the reactive armor exploded, disrupting the ability of the shaped charge to form its concentrated burn front. During Operation Desert Storm, the public saw that the weapons designers had again struck back, with the so-called “sabot” or penetrator round. This was essentially an extremely high velocity “bullet,” usually made of hard, dense, depleted uranium. The penetrator round was unaffected by reactive armor, and penetrated tank armor purely by kinetic energy applied over a small penetration area. Once inside the tank, the metal penetrator and fragments of armor literally “bounce” around the interior of the tank, killing the crew and damaging equipment. In appearance, the penetrator looks much like a long, pointed piece of steel bar, gaining its formidable armor piercing performance only when it is hurled at great speed. It is axiomatic that the next generation of tanks will be configured to defeat high velocity penetrators, and equally true that weapons designers are working diligently to develop a new counter-weapon.

2.3 U.S. NAVY SHOCK TEST PLATFORMS AND MACHINES

The U.S. Navy has historically allowed substantial information to be published on its shock testing programs, much more so than the other military services. Many of the Navy shock test requirements are readily available to the public (Reference 2). To illustrate the lengths to which the U.S. Navy goes in its shock survivability testing, a photographic sequence of a Floating Shock Platform (FSP) test is provided as Figures 4 through 6. The FSP is similar in shape to a small barge. The test items are fixed to the Shock Platform, in the same manner in which they would be oriented and fixed on a ship. To allow these photographs to be published, the test objects are shielded from view by a security shroud. The shock transient occurs from the simultaneous detonation of two moored explosive charges, located at a point of “stand-off” from the platform so as to cause a fairly well understood time history to occur at the floating platform. The transient resembles closely that of Figure 2. One thing that is quickly evident from the photos is the noticeable horizontal distance and submerged depth of the charges, as compared to the common perception that explosives are more dangerous when they detonate in contact with the ship. Indeed, since World War II, it has been known that maximum damage will result to a ship when the weapon detonates at some distance from it. The dynamics involved are complex. In general, the explosive creates an underwater gas bubble, which undulates by repeatedly expanding and contracting in an attempt to reach equilibrium (Reference 3). Maximum ship damage occurs when the outer diameter of the undulating bubble contacts the ship’s hull, and can literally break the ship in half if the detonation is well placed. A detonation that occurs too close to the ship causes the explosive bubble to vent to the atmosphere, thus greatly reducing the available destructive energy. A contact detonation may cause localized hull penetration, but since most combat ships have numerous sealable watertight compartments, a hole in the hull is easily countered. Typically, minimum stand-off distances of around 20 feet are used during FSP testing, with the explosive charges moored 24 feet beneath the surface of the water (Reference 4). Implied by the photographs is the fact that Floating Shock Platform tests are expensive. Even more expensive are tests run on an entire ship, a project which usually is scheduled two or more years in advance of the actual tests.

To allow component level testing at reasonable costs, the U.S. Navy uses the so-called Light Weight and Medium Weight Shock Test Machines. The light weight machine can accept test objects weighing up to 550 pounds. The medium weight machine can accept objects weighing up to 7,400 pounds. Larger objects must be tested with explosives on the FSP.

The Navy Shock Test Machines consist of a rotating drop hammer, driven by gravity (Figure 7). The test object is affixed to the machine with a tuneable mounting fixture, such that the primary response frequency of the fixture and object is coincident with the expected frequency of the region of the ship on which the equipment is to be installed. The oldest of these machines, the Light Weight Shock Test Machine, dates to the 1930's in the British Navy, with a version of the British machine adapted by the U.S. Navy in 1940. The Medium Weight Shock Testing Machine is of U.S. origin, and was adapted by the U.S. Navy in 1942. To this day, no unclassified information exists as to the exact criteria used to construct and qualify these machines, only vague comments as to the problem of "the growing necessity for complex and delicate electronic devices, such as radar and sonar" (Reference 4). Due to the age of these drop test machines, it is evident that it was not possible to quantify their input transients during their design, since neither adequate instrumentation or computers existed in the 1930-1942 period. Instead, shock equivalency was established by equivalent damage analysis from field observations, and all data remains classified. After World War II, the Navy had the opportunity to update its equivalent damage criteria with nuclear weapons detonations. The most notable tests were code named Able and Baker, the first nuclear tests to occur after World War II. Over ninety ships were subjected to the tests, ranging in size from small landing craft and submarines to the aircraft carrier Saratoga and the battleships Arkansas, Nagato (Japan) and Prince Eugen (Germany). Detonation Able was an airburst, with the device dropped from a bomber. Detonation Baker was a nuclear depth bomb detonated ninety feet below the surface. The Baker detonation sunk many of the test ships, including the Saratoga, Arkansas and Nagato. Post test difficulties indicate that tests of this magnitude will never be performed again. The Baker detonation produced intense radioactive water spray and fog, of a level now known to be life threatening to test personnel. Post test damage analyses from these tests are highly classified, and will probably remain so well into the next century.

Within the past few years, substantial effort has taken place to quantify and simulate the transient responses of the various Navy shock test machines. Finite element modeling of the machines and simulations using standard techniques have been most successful (Reference 5). However, the latest revision of the U.S. Navy Shock Test Standard (Reference 2) is dated 1989, and is essentially unchanged from the previous revision of 1963 with respect to input and test criteria. This demonstrates that within the U.S. Navy, drop hammer testing has a 30+ year experience period of successful use to simulate severe shock transients, without the need for changes.

3.0 DAMPER PARAMETERS FOR SEISMIC ENERGY DISSIPATION

Fluid damping devices are well proven by the test of time, with production of dampers in the 50 kip range dating to the mid-1890's. The earliest well documented use of large fluid dampers was by the military, to attenuate recoil transients on large caliber artillery pieces. For example, the French 75 mm gun, model of 1897, utilized relatively sophisticated dampers, which featured controllable force output to compensate for the firing elevation angle of the weapon. Literally tens of thousands of large fluid dampers have been manufactured for this usage, and many of the design elements of the M1897 French damper can indeed be found on present-day artillery, nearly all of which use fluid damping devices.

For testing purposes, fluid dampers can be classified into three groups, depending on the operating design of the internal orifices used.

<u>Viscous-Shear Dampers</u> produce an output by viscous shearing of the fluid, and can operate only at relatively low damping fluid pressures. Typically, maximum pressure is less than 300 psi, making this type of device rather large and cumbersome. Output generally follows the classical equations for viscous fluid shear, where shear stress is proportional to speed. This results in the so-called “linear” or “viscous” output, where damper force is proportional to velocity. The major drawback of viscous shear dampers is a strong temperature dependency. Over a typical north central U.S. outdoor temperature range of -20^{o}F to +120^{o}F, fluid viscosity changes of ten to thirty to one are common, and this viscosity shift has a direct effect on damping forces.

<u>Inertial Fluid Dampers</u> produce an output by forcing fluid through orifice passages. The output force of this type of damper is dependent on the size and shape of the orifices. Operating pressures of 2,000-10,000 psi are common, thus minimizing the effect of fluid viscosity change, since high inertial fluid pressures dominate the output. Inertial fluid dampers have an output force which follows the Bernoulli Equation, where output force varies with the square of the damper stroking speed, and directly with the fluid density. Various mechanical construction means are used to “shape” or “tune” the output force of the damper to a specific function. These means can involve complex combinations of spaced orifice holes, tapered orifice pins, or various types of spring loaded valves. For example, the French M1897 Recoil Buffer utilized a long tapered orifice pin, ground to a non-symmetrical cross section. The pin taper fixed the damper output wave form, and by rotating the pin, the absolute magnitude of the tapered orifice could be adjusted. An external gear was attached to the pin, and as the gun was elevated, the gun elevation gear train rotated the gear on the pin, stiffening the overall output as the angle of firing elevation increased. Today, inertial drive damping devices are widely used in industry on applications ranging from small machines to large steel mill equipment.

<u>Fluidic Dampers</u> were first produced in the 1960's, and utilize the technology of passive fluidic control. Whereas viscous shear and inertial drive dampers produce an output force that varies with velocity and (velocity)^2 respectively, fluidic orifices can be specifically designed for a wide range of damping functions. Damping functions can vary velocity exponents from as low as 0.2 to as high as 1.8 depending on customer requirements (Reference 6). In general, the higher the peak translational speed of the input, the lower the

optimal damping exponent. Fluidic orifices operate in the 2,000-10,000 psi range, minimizing effects of fluid viscosity change. Output is generally unaffected by temperature, fluid type, and manufacturing tolerances.

Both viscous shear and fluidic dampers have been used as seismic energy dissipation devices; inertial drive dampers have not. It is easily observed that the velocity squared output of an inertial fluid damper is not acceptable for seismic energy dissipation, due to the high variance in potential input transients. Unless the entire velocity-time history of the real input matches very closely the transient used for design, then the risk of bottoming-out the damper or overloading the structure exists. The linear output of the viscous shear damper is often used by design teams, since available software codes usually contain viscous damping elements. Fluidic dampers can be set for a linear output also, but often a non-linear analysis will reveal that damping exponents of 0.4 to 0.7 will provide better seismic performance than that of a linear device.

4.0 TESTING CRITERIA

In general, test criteria have evolved within the defense community over the past 40 years which apply to any type of damper used within a structure. Well over 300,000 high output force dampers are known by the author to have been used previously in defense applications, most of which have been subjected to extensive testing. These well established criteria indicate that if a damping device is to be used for purposes of energy dissipation in a structure, then the following tests are necessary to verify performance:

a. Damping force vs. velocity over the entire credible velocity range of the transient
b. Deflection capability
c. Damping function consistency with respect to stroke position
d. Damping function vs. temperature
e. Cumulative energy capacity
f. Cyclic life
g. Environmental life
h. Structural yield and ultimate loads

Tests of the above items can be classified as specific (done on the production hardware), scaled (done on a scaled model of the production hardware), or generic (done on a damper of the same form and function as the production hardware).

4.1 DAMPING FORCE VS. VELOCITY OVER THE ENTIRE CREDIBLE VELOCITY RANGE OF THE TRANSIENT

This test is specific and is run on 100% of the production dampers. Usually the first few dampers produced are tested at velocities of 20%, 40%, 60%, 80%, and 100% maximum. Subsequent production is usually tested at 100% speed only, if dampers built to maximum and minimum drawing tolerances have passed the testing at the speed multiples listed.

4.2 DEFLECTION CAPABILITY

This test is specific, and is run on 100% of the production dampers. Each damper is stroked to minimum and maximum stroke position at relatively low speed, monitoring the damper for any irregularity, such as sticking, binding, or excessive friction.

4.3 DAMPING FUNCTION CONSISTENCY WITH RESPECT TO STROKE POSITION

This test is specific and is performed on the first production damper to verify the design. Usually, this test is performed 1/4, 1/2, and 3/4 stroke positions, checking each position for proper damping function at the five velocities of Section 4.1.

4.4 DAMPING FUNCTION VS. TEMPERATURE

This test may be specific, scaled, or generic, depending on the size of the damper and the available temperature test chamber. The five point damping function test of Section 4.1. is performed at the expected low, ambient, and high temperatures for the application. If this test is performed on a generic damper, then the manufacturer must demonstrate that the generic sample is similar in design, parts arrangements, operating pressures, and material to the full size production hardware.

The temperature range for test should be indicative of the worst case extremes, which vary with the geographical location of the structure. If a damper is to be qualified for service within the United States, temperature extremes to be tested should include:

32°F to 120°F - Southern United States
-20°F to 120°F - Northern United States

For world-wide service, indoors or outdoors, current military specifications recommend a range of -40°F to +160°F for all locations except for those within the Arctic and Antarctic Circles.

4.5 CUMULATIVE ENERGY CAPACITY

This criteria is verified by test or analysis on specific or scaled hardware. It is important that the damper have adequate thermal mass to produce the energy dissipation required for the maximum credible transient, without overheating damage. Generally this can be verified by analysis, using a conservative approach that assumes no heat transfer out of the damper fluid, and a near instantaneous input of the transient.

The energy output of the damper is obtained by integration of its force-displacement response, under the maximum credible event. This is set equal to the product of the fluid mass in the dampers pressure tube times the specific heat constant for the fluid times the fluid temperature differential. The terminal temperature must be such that the fluid does not boil, or cause fluid seals to melt or soften.

If cumulative energy is to be tested, the test device is cycled such that the damper absorbs the proper amount of energy within a time reflective of the real time for the full scale transient (usually less than 1 minute). Damper temperature is monitored during and immediately after the test, with post test inspection.

4.6 CYCLIC LIFE

This test may be performed on either specific or scaled devices, depending on damper size and the availability of suitable cyclic test equipment. Depending on whether the damper is expected to dissipate energy during wind and/or traffic excitations (such as on a flexible structure, like a suspension bridge), the actual number of cycles required will vary widely from application to application. For uses involving substantially seismic use, life cycles of less than 1000 full strokes will be representative of a total life cycle. If wind and traffic excitations exist, as many as 5×10^6 cycles may be necessary. In addition, examination of the materials of construction must verify that no age sensitive materials are used.

4.7 ENVIRONMENTAL LIFE

This test can be performed using a design analysis, or by accelerated testing in a salt fog chamber on specific or scaled devices. Most seismic damping devices installed within a building can be expected to operate in a damp environment, and thus require nearly as much design concern for corrosion resistance as those intended for service outdoors. To insure a life without maintenance of fifty plus years, particular care must be taken in metallic material selection. Reference 7 lists standards for metallic material selection such that unacceptable dissimilar metal couples can be avoided. Other guidelines are related to specific damper parts, such as piston rods and internal fluid reservoirs. Both of these should be fabricated from stainless steel, even if they are to be plated before use. For example, a conventional piston rod for non-seismic damper application would use hard chrome plating over carbon or alloy steel. During long term use, a rod of this type would be expected to develop corrosion pits and cracks in the steel base material, since chrome is both porous and difficult to deposit uniformly all over a part that has edges and threads. Any type of pitting or lifting of the extremely hard chrome will slit or tear the damper seals in only a single cycle, causing catastrophic fluid loss. Similarly, any non-stainless steel damper fluid reservoir containing both air and fluid will develop condensation during use, which will cause severe internal corrosion. This corrosion will spread over all internal surfaces without any apparent defects noted by an external inspection.

Other potential environmental test/design criteria for specific sites include:

Sand and dust
Fungus formation/attack
Thermal shock
Solar radiation
Ozone attack
Acid rain, smog
Ice build-up

4.8 STRUCTURAL YIELD AND ULTIMATE LOADS

Testing for load capability may be performed on specific hardware or by stress analysis, depending on damper size and contractual requirements. Most dampers utilize heat treated alloy and stainless steels in the 80,000 psi to 200,000 psi ultimate strength range. This is necessary to minimize the diameter and weight of the damper, which tends to minimize cost. Materials of this type normally have a yield stress that is 60% to 90% of the material's ultimate stress. Typical military strength requirements for dampers require a minimum safety factor of 2 or more on yield, and 2.5 or more on tensile strength. In general, materials with more than 10% elongation to failure are considered acceptable for any structural member or part that is to be subjected to transient or shock loading.

A second major concern is pressure capability of each fully assembled production damper. A proof pressure test should be run on 100% of production to verify steel, joint, and seal integrity. Normally, a pressure of 125% that of the maximum credible loading expected is applied for a period indicative of the maximum time duration of the transient. For weapon's effects dampers, the normal military requirement is simply to reach the desired proof pressure without damage, since weapon's effects occur very rapidly. For seismic use in structures, holding a 125% proof pressure for a period of one minute appears to provide a satisfactory level of proofing.

5.0 SHOCK TEST MACHINES FOR TESTING OF LARGE DAMPING DEVICES

Test machines required for the eight tests defined in 4.0 would generally be as follows, using conventional approaches from prior practice within the structural engineering community:

5.0.1 DAMPING FORCE VS. VELOCITY OVER THE ENTIRE CREDIBLE VELOCITY RANGE OF THE TRANSIENT

A hydraulic actuator with pumps, accumulators, and servo valve would be used to drive the damper with a series of sine wave motions, varying frequency and displacement to obtain various points of damping force at velocity.

5.0.2 DEFLECTION CAPABILITY

The actuator in test 5.0.1 would be used to fully compress and fully extend the damper, with measurements taken of damper length at these points.

5.0.3 DAMPING FUNCTION CONSISTENCY WITH RESPECT TO STROKE POSITION

The actuator used in test 5.0.1 would be used to cycle the damper through a series of sine waves, with the initial (zero) stroke position of the damper set at 1/4, 1/2, or 3/4 stroke position prior to initiating the sine wave sweeps.

5.0.4 DAMPING FUNCTION VS. TEMPERATURE

Test 5.0.1 would be repeated with the damper held at the required test temperature.

5.0.5 CUMULATIVE ENERGY CAPACITY

This test can be performed using the actuator from test 5.0.1, or verification can be performed by analysis.

5.0.6 CYCLIC LIFE

This test can be performed with the actuator from test 5.0.1.

5.0.7 ENVIRONMENTAL LIFE

This test requires suitable test chambers, or verification by analysis.

5.0.8 STRUCTURAL YIELD AND ULTIMATE LOADS

These tests are usually performed in a press, or so-called universal test machine.

The most problematic of the test machines for the above tests 5.0.1 through 5.0.8 is the large hydraulic actuator and its pumps and valves. If a realistic full size damper is tested, test machines in current use fall far short of the required power.

For example, a typical damper for use in a base isolated structure would have an output force in the 330 kip range, at a peak translational velocity of 60 in/sec.

Required power is:

$$\begin{aligned} \text{Power} &= \text{Force x Velocity} \\ &= 330{,}000 \text{ lb. x } 60 \text{ in/sec.} \\ &= 19{,}800{,}000 \text{ in-lb/sec.} \\ &= 1{,}650{,}000 \text{ ft-lb/sec.} \\ &= 3{,}000 \text{ horsepower} \end{aligned}$$

Since most hydraulic testing systems operate in the 60% efficiency range, a power source of 5,000 HP is required. A machine of this size is truly formidable in both size and cost.

Recently, testing has been performed to demonstrate the applicability of military type drop hammers as an effective method of testing large fluid dampers intended for seismic energy dissipation in both fixed base and base isolated structures. Used in this manner, drop testing can be substituted for conventional sine wave cyclic tests, which, in the case of large dampers in the 100 kip to 2,000 kip output range, cannot be easily tested. This is especially true when one considers that large structures can often require large numbers of high capacity devices, all of which should be tested for performance before installation.

5.1 DROP HAMMER TEST MACHINES

One of the easiest ways to generate large amounts of energy is to use gravity to accelerate a free falling weight. A typical drop hammer test machine is schematically presented in Figure 8. The energy input available is equal to the weight times its total falling distance, which includes the free fall distance plus the stroke of the test article. Power available is quite high, essentially limited only by the time necessary to decelerate the weight to a reduced speed.

For example, we can assume the following specifics for the drop hammer test of Figure 8:

Drop weight = 15,000 lb.

Damper = 330 kip output rating at 60 in/sec.

Damping function = $V^{.4}$

Assuming a “rigid” ground node, and a very stiff structure for the damper itself, if the drop weight is raised to a level of 4.66 inches above the damper, it will impact the damper at 60 in/sec, causing the damper to output 330,000 lb. force. The peak power instantaneously generated is 3,000 HP, using the same equations as used in 5.0 previously. During an actual test, the weight will need to be raised slightly higher to compensate for the slight deflection of the ground node and test fixture during the impact.

The only potential difficulty with a drop test of this type is that the power from the drop weight is available for only a short time. For example, when the 15,000 lb. weight impacts the damper, its kinetic energy is:

$$\text{KE} = \frac{1}{2}\text{MV}^2 = \frac{1}{2}\left(\frac{15{,}000}{32}\right)\left(\frac{60}{12}\right)^2 = 5{,}859 \text{ ft-lb} = 70{,}312 \text{ in-lb}$$

When the damper is impacted, its output goes from 0 to 330,000 lb. and it begins dissipating energy. With the damper described above, the entire energy of the drop weight will be dissipated within approximately .38 inches of damper deflection. This is of no particular difficulty, provided that high frequency recording equipment is available. Most drop hammer test rigs still utilize analog recording equipment rather than the more modern digital types for this reason. This is because most digital equipment has a usable “tick rate” that is relatively slow, and will lose much resolution quality compared to older analog devices during a short duration drop test. Another quality of instruments for drop hammer testing is that most recording transducers will be resistive in nature. A typical test on the damper described would use a resistive strain gaged load cell to record force and wire wound resistance potentiometer to record stroke. Other types of transducers, including accelerometers and inductive potentiomenters do not work well on short duration testing, exhibiting either ringing or phase lags which degrade the test measurements.

5.2 SIZE RANGES OF DROP HAMMERS

The effectiveness of a drop hammer is determined by its maximum throw weight, maximum shut-height, and ground node stiffness. Both commercial and Government owned drop hammers exist within the U.S. today, most of which were constructed for specific test applications with later use for generalized testing.

The drop hammer of Figure 8 is owned by Taylor Devices, and was originally built for the testing of large damping devices used on NASA’s Apollo Program of the 1960's. It has an 18,000 lb. weight capacity, a 44 ft. shut-height, and an extremely stiff ground node frequency of 270 hertz, intended to simulate the primary frequency of the Apollo Launch Pad at the Kennedy Space Center. In our military testing, even the most rigid engineered structures, such as the armored decks of warships, rarely exceed 75 hertz frequency, assuring that tests performed with this particular test rig will be conservative.

Other drop hammers known to exist in the U.S. have weights up to 350,000 lbs. (unfortunately with a limited shut-height) and heights up to 200 ft.

6.0 COMPARATIVE TEST RESULTS: DROP HAMMER VS. SINE WAVE ACTUATOR

Recently, Taylor Devices received a purchase contract for a quantity of the 330 kip dampers discussed previously. Rated output force was 300 kips plus or minus 30 kips at a speed of 60 in/sec. The damping function selected was $V^{.4}$. The dampers were to be used on a base isolated structure, using elastomer isolation bearings as the spring element in the isolation system. The $V^{.4}$ damping function was selected after extensive transient analysis had been performed to find optimum conditions of energy dissipation and building base shear loadings for the combined output of elastomer spring and damper. Important criteria that required verification by testing included:

1. Variance of damping function over the expected velocity range.
2. Change in damping with temperature.
3. Change in damping from cycle to cycle during the maximum credible earthquake.

Early in the design process it became evident that no available actuators existed to cycle the full sized dampers. A testing sequence evolved to utilize drop testing on full sized devices, with both cyclic and drop testing performed on a scaled damper to demonstrate correlation between drop testing and the traditional cyclic test method. The following is the list of tests that were performed:

1. Perform tests with a scaled damper using existing laboratory cyclic test equipment rated for output in the 100 kip range at speeds to 25 in/sec. The force scaled test damper would have output in the 50 kip range with a $V^{.4}$ damping function and a damping coefficient set for maximum force level in the 20 in/sec. range.

2. Drop test the scaled damper at various drop heights, comparing force vs. velocity plots from the drop test to those resulting from the cyclic tests. Agreement of drop test data points to within plus or minus 10% of the cyclic test data baseline would correlate the two test methods.

3. Perform extreme temperature tests on the scaled prototype, using a thermal box constructed around the damper on the cyclic test fixture.

4. Obtain cumulative energy data by cycling the scaled prototype rapidly until its total energy dissipated per unit volume of damping fluid equaled or exceeded the same value expected from the full sized device under the maximum credible earthquake.

5. Drop test the full sized device set for the specified 300 kip output at 60 in/sec. using various drop heights to verify the required damping function.

It should be noted that fluid dampers for the scaled testing above must be hydraulically scaled from the full sized device if the test is to be considered representative. This means that at their respective maximum output force rating, both the scaled and full size devices must operate at the same internal pressure, with damper parts sized such that both devices operate at the same stress levels.

The fact that the scaled damper was set to operate at a lower velocity range than the full sized device was due to limitations of the hydraulic actuator used on the cyclic test. In a fluid damper, this means only that the orifice in the device must have its total flow area adjusted by the velocity ratio of 25/60 to provide its maximum output at the reduced velocity range.

When the scaled damper was designed, a degree of uncertainty existed relative to the method used to load rate the available cyclic test machine. The test machines's actuator was factory rated at 110 kips, and the machine was equipped with a pump and control valve which should allow it to achieve full actuator output at 25 in/sec. velocity. The uncertainty was whether the equipment manufacturer had used sinusoidal wave forms during rating tests, or a more rigorous wave form approaching that of a square wave. Driving a damper with a $V^{.4}$ damping function through sinusoidal motion generates a force-displacement output that is basically a series of square waves, with the magnitude of the square wave varying with the peak velocity of the input wave raised to the 0.4 power. To avoid building a damper that could not be satisfactorily driven by the test machine, it was decided to build the scaled damper at a 50 kip rated force, set for full output at 20 in/sec. This left a suitable margin of safety to the maximum rating of the cyclic test machine. In comparison, the drop test machine is not dependant on damper wave form shapes, since its total available energy at impact is available instantaneously.

It should also be noted that the 60 in/sec. transient wave selected for this project was thought to be extremely conservative, since typical maximum credible earthquake transients are usually in the 20-30 in/sec. range for much of California. However, after the recent Northridge Earthquake, with translational velocities of 51 in/sec, it now appears that this particular high level transient is very representative!

6.1 TEST RESULTS: SINE WAVE CYCLIC TESTS, AMBIENT TEMPERATURE

Figures 9-13 show the expected quasi-square wave output, at speeds of 1 in/sec. to 17 in/sec., with sine wave frequencies of .064 Hz to 5.864 Hz Note that at the higher speeds, the 110 kip actuator is having difficulty driving the 50 kip damper, and it was necessary to alter the input command to the actuator calling for an over-speed square wave drive on the first cycle of each test. This allowed subsequent cycles to be at the specified parameters. This is particularly obvious in Figure 13, where it is evident the test machine is operating well beyond its capability, and is having difficulty driving the proper wave form. In addition, with the rather short deflections used in this particular test, some flexing of the test fixture may be occurring.

6.2 TEST RESULTS: SINE WAVE CYCLIC TESTS, EXTREME TEMPERATURES

For these tests, a thermal blanket was placed around the unit and the damper was stabilized at the required temperature prior to test. Figures 14-18 show the results, at speeds of 1 in/sec. to 17 in/sec., and frequencies of .064 Hz to 1.082 Hz with the damper temperature at 120ºF. Figures 19-23 show the results with the damper temperature at 32ºF.

6.3 TEST RESULTS: SINE WAVE CYCLIC TEST, CUMULATIVE ENERGY INPUT OF THE MAXIMUM CREDIBLE EARTHQUAKE

Figure 24 shows the results of seven complete cycles of motion at 4 in/sec. velocity and .225 Hz frequency. The cumulative energy dissipated at 3.5 cycles was equivalent in units of BTU/lb mass of fluid to that of the full scale device under the input condition of the maximum credible seismic transient for this project.

6.4 TEST RESULTS: DROP TESTING OF SCALED PROTOTYPE

Figures 25-29 show the results of drop tests at 1, 3.4, 9, 13.1, and 16.2 in/sec. Velocity was obtained by sloping the stroke vs. time curve at the point of maximum output force.

6.5 TEST RESULTS: DROP TESTING OF FULL SIZED FLUID DAMPER

Figure 30 depicts test results from a series of drop tests at speeds to 60 in/sec. and forces to the 300 kip level. All points plot within the acceptance band for the full sized device.

7.0 SUMMARY OF TEST RESULTS

The overall testing scheme used for this project proved to be highly successful. Figure 31 plots comparative cyclic test and drop test data on the scaled prototype damper. The cyclic test results were used as a functional baseline, with a curve fitted to the data and an allowable correlation band width of plus or minus 10%, represented in Figure 31 by dashed lines. All drop test points were well within the allowable band width, demonstrating the comparative results from the two test methods.

Figure 32 provides thermal test results at the 3 temperatures selected for evaluation, these being +32ºF, +70ºF, and +120ºF. Parameter drift for the damper was minimal over the entire range tested. This particular damper is of the fluidic type, and uses passive internal temperature compensation of the damping orifice. With a high velocity orifice of this type, the compensator adjusts the orifice area with temperature to allow for the change in the mass density of the fluid. However, at low flow velocities, the orifice output becomes dominated by viscous shear rather than inertial flows, and hence the compensation of the orifice is not as effective. Thus, the performance change in the damper with temperature becomes more pronounced at low speeds. This will have little change in the performance of the overall isolation system, since at reduced translational speeds, the total energy to be dissipated is rapidly dropping with the square of the velocity, which is much greater than the slight shift in damper output.

Drop testing of the full sized device was equally successful, with no difficulties or problems noted. As with any test rig using a strain-gaged load cell, parallel alignment of the 15 foot long damper to the drop rail was important to avoid noise in the force channel. Shims were required to hold parallel alignment from end to end of the damper to within .5 inch. This prevented "ringing" of the load cell at impact. The damper itself is mechanically capable of accepting mis-alignment angles in excess of 15%. Figure 33 is a photograph of the full sized damper.

8.0 CONCLUSION

The use of high capacity energy dissipation devices in buildings and bridges requires that testing be performed at full scale force and velocity levels. Drop testing has now been successfully proven to be an acceptable test method for large fluid damping devices. Test results demonstrate excellent correlation between drop test equipment, as used for many years by the defense community, and cyclic test equipment, as used for many years by the structural engineering community.

The use of drop testing is a cost effective way of testing full scale damping devices in the range of 100 kips to 2000 kips output force, utilizing methods and equipment that have been proven over many years of testing, which are readily available to the public at low cost.

The force and velocity ranges available from a drop test facility are limited only by the height of the drop rail and the size and strength of the seismic mass the rail is affixed to. This allows testing of damping devices to much higher speed ranges than were previously possible, allowing enhanced seismic transient requirements to be easily tested, without the need for costly development of new equipment.

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   UDC 620.193.7:620.197
   Confirmed March 1990

   b) MIL-STD-889B, entitled "Military Standard, Dissimilar Metals"
   Revised 1976

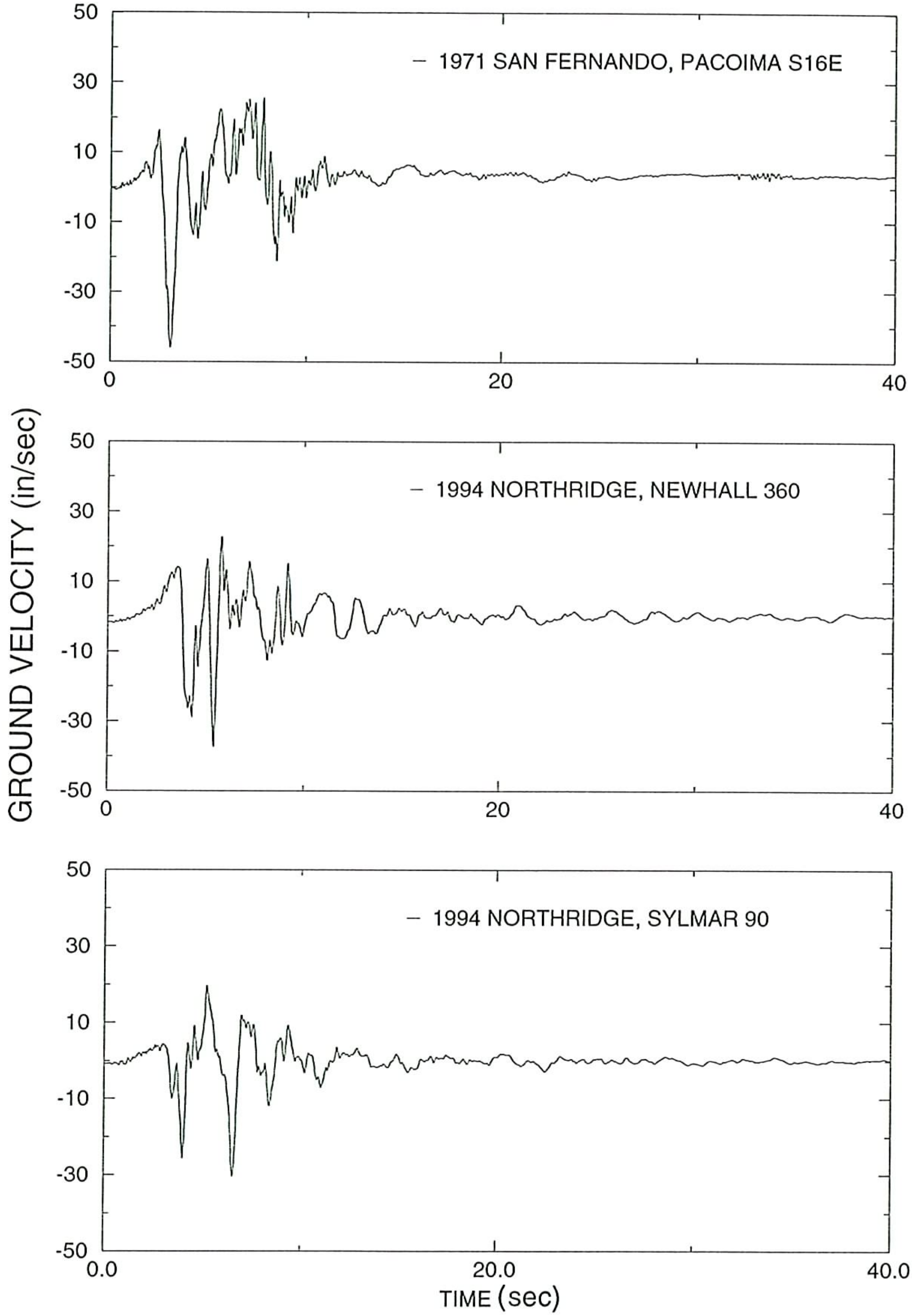

1971 SAN FERNANDO, PACOIMA S16E
1994 NORTHRIDGE, NEWHALL 360
1994 NORTHRIDGE, SYLMAR 90
GROUND VELOCITY (in/sec)
TIME (sec)
50
30
10
-10
-30
-50
0
20
40
0.0
20.0
40.0

FIGURE 1

Table 1   Characteristics of Earthquake Motions in Figure 1

Earthquake	Date	Station/ Component	Site Geology	Epicenter Distance (km)	Peak Ground Motion		
					accel (g)	vel (in/sec)	displ (in)
San Fernando, CA	2/9/71	Pacoima Dam, Comp. S16E	Rock	30	1.17	44.58	14.38
Northridge, CA	1/17/94	Newhall, LA County Fire Sta., No. 24279, Comp. 360°	Alluvium	20	0.59	37.29	11.99
Northridge, CA	1/17/94	Sylmar County Hospital, Parking Lot, No. 24514 Comp. 90°	Alluvium	16	0.60	30.23	5.99

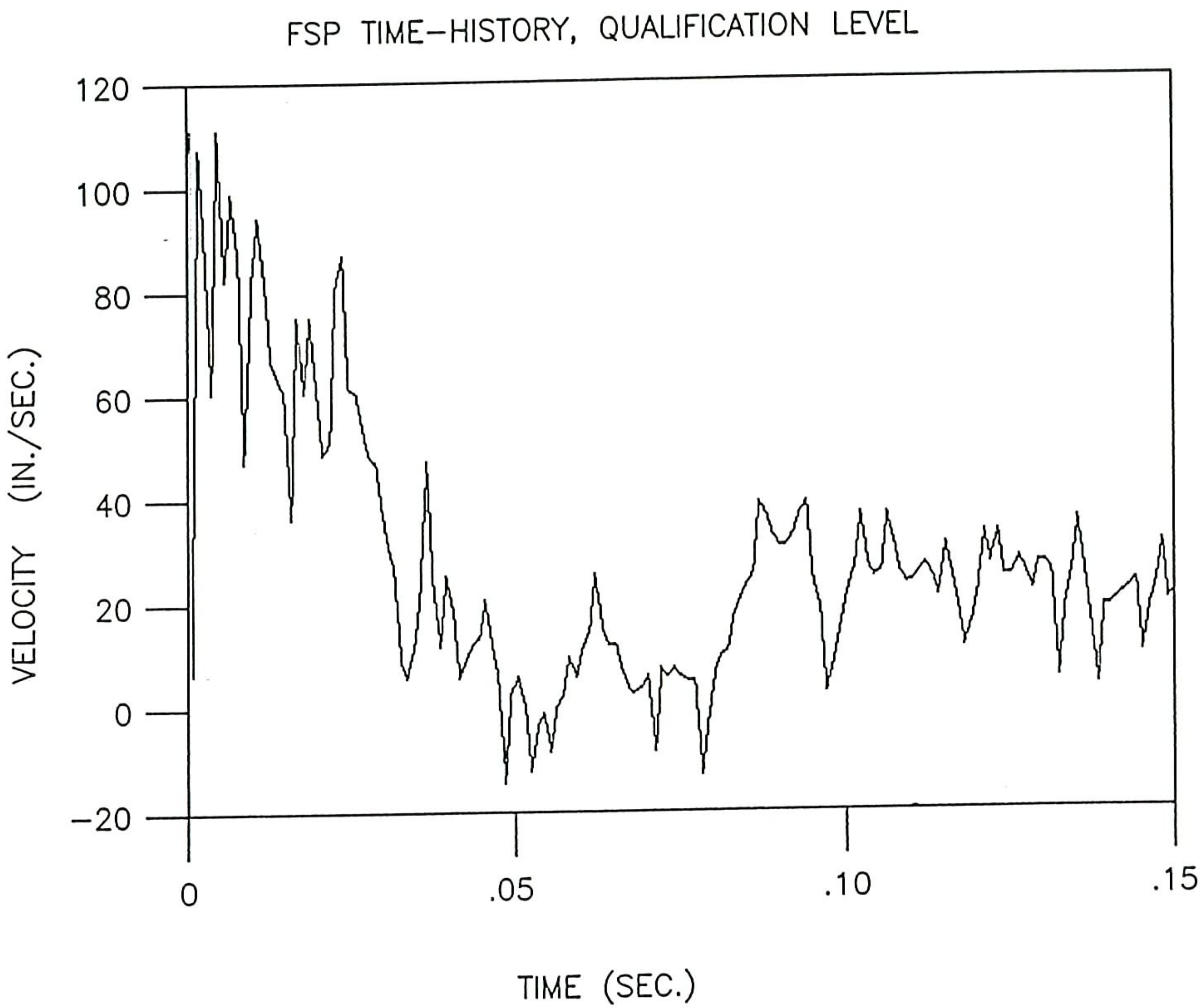

FSP TIME-HISTORY, QUALIFICATION LEVEL
120
100
80
60
40
20
0
-20
VELOCITY (IN./SEC.)
0
.05
.10
.15
TIME (SEC.)

FIGURE 2

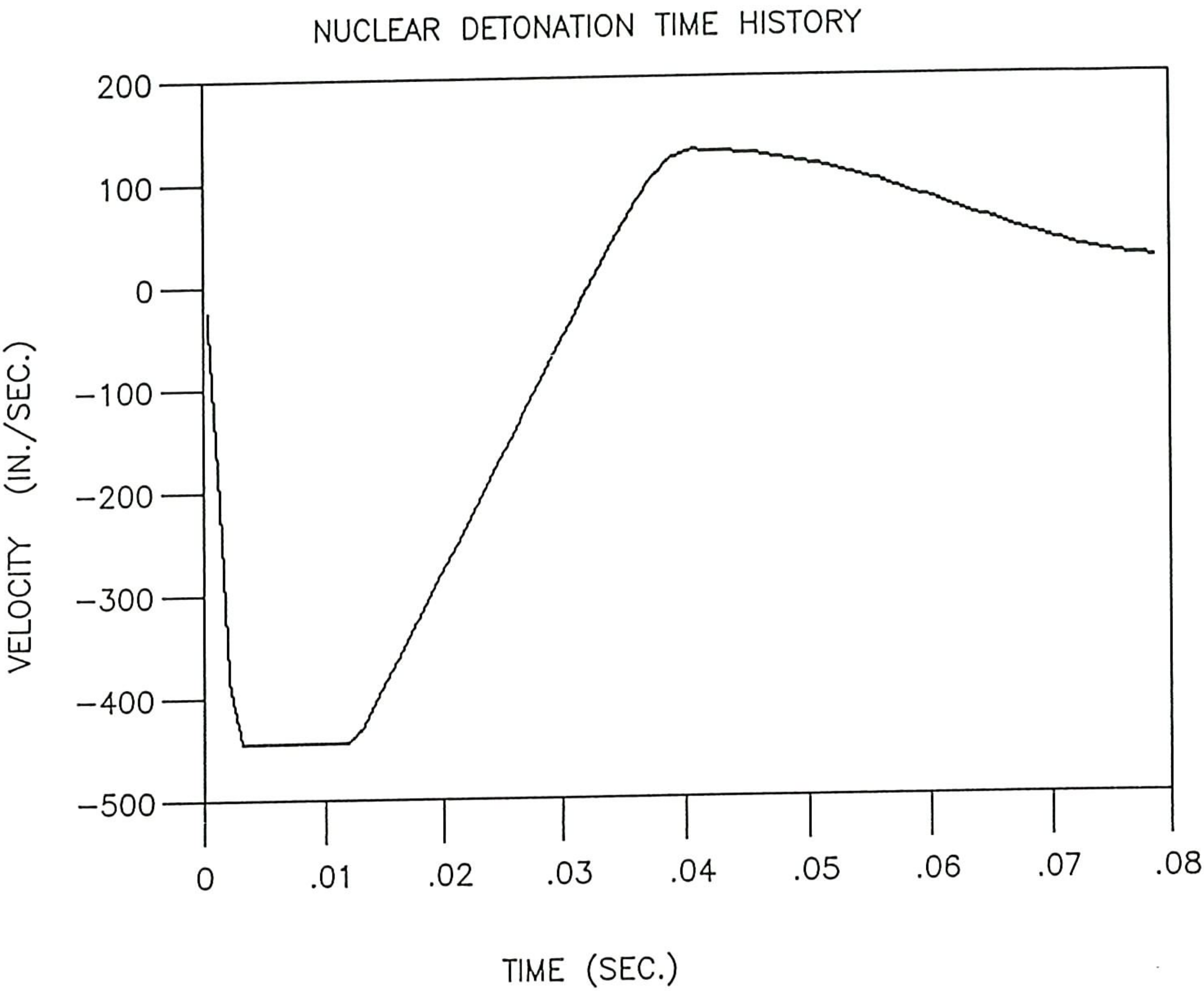

NUCLEAR DETONATION TIME HISTORY
200
100
0
−100
−200
−300
−400
−500
VELOCITY (IN./SEC.)
0
.01
.02
.03
.04
.05
.06
.07
.08
TIME (SEC.)

FIGURE 3

FIGURE 4

FIGURE 5

FIGURE 6

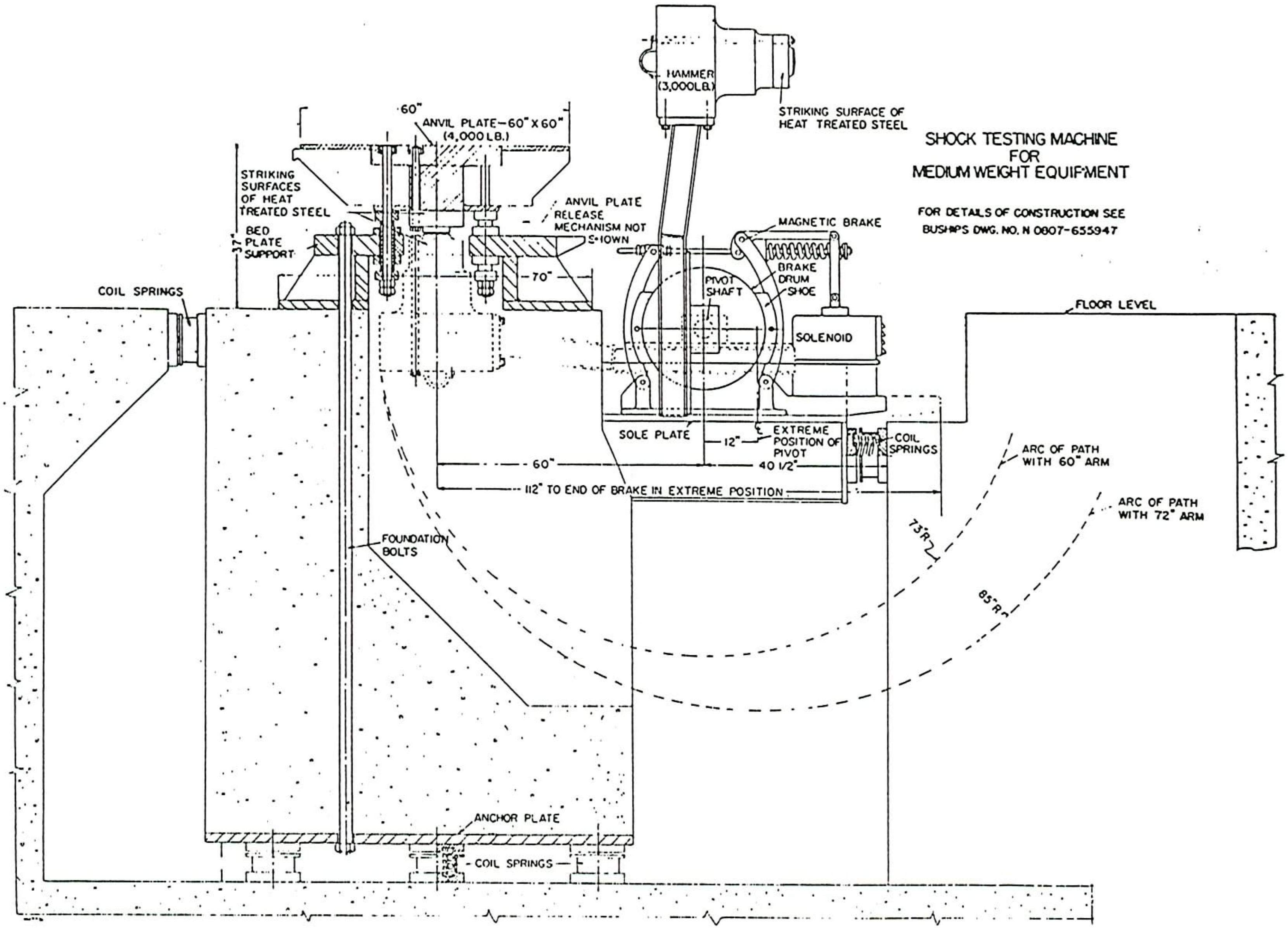
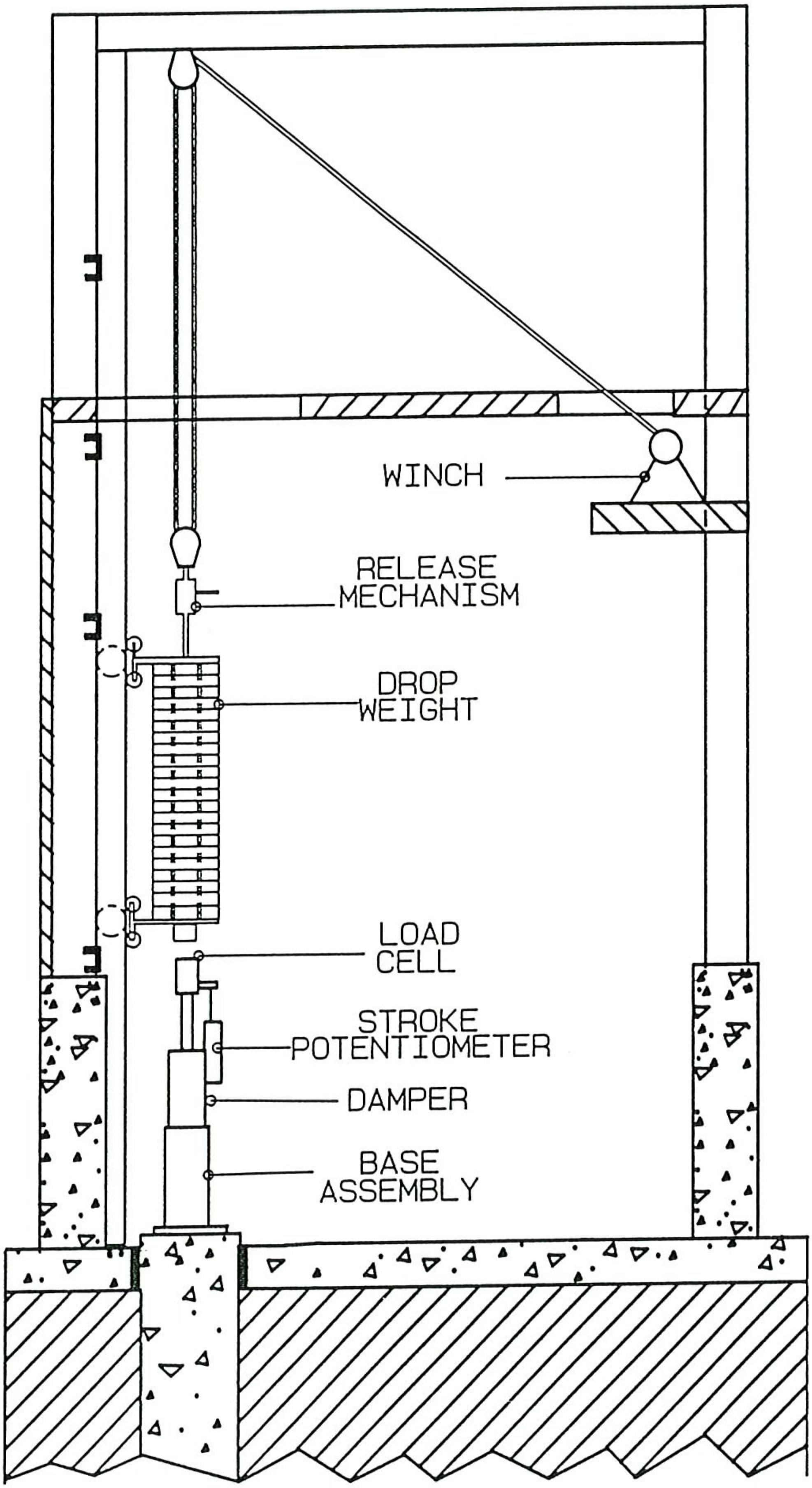

SHOCK TESTING MACHINE
FOR
MEDIUM WEIGHT EQUIPMENT
FOR DETAILS OF CONSTRUCTION SEE
BUSHIPS DWG. NO. N 0807-655947
HAMMER (3,000 LB.)
STRIKING SURFACE OF HEAT TREATED STEEL
60"
ANVIL PLATE–60" X 60" (4,000 LB.)
STRIKING SURFACES OF HEAT TREATED STEEL
ANVIL PLATE RELEASE MECHANISM NOT SHOWN
MAGNETIC BRAKE
BED PLATE SUPPORT
37"
70"
BRAKE DRUM
PIVOT SHAFT
SHOE
COIL SPRINGS
SOLENOID
FLOOR LEVEL
SOLE PLATE
12"
EXTREME POSITION OF PIVOT
COIL SPRINGS
ARC OF PATH WITH 60" ARM
60"
40 1/2"
112" TO END OF BRAKE IN EXTREME POSITION
ARC OF PATH WITH 72" ARM
73"R
FOUNDATION BOLTS
85"R
ANCHOR PLATE
COIL SPRINGS

FIGURE 7

FIGURE 8



WINCH
RELEASE
MECHANISM
DROP
WEIGHT
LOAD
CELL
STROKE
POTENTIOMETER
DAMPER
BASE
ASSEMBLY

FIGURE 9

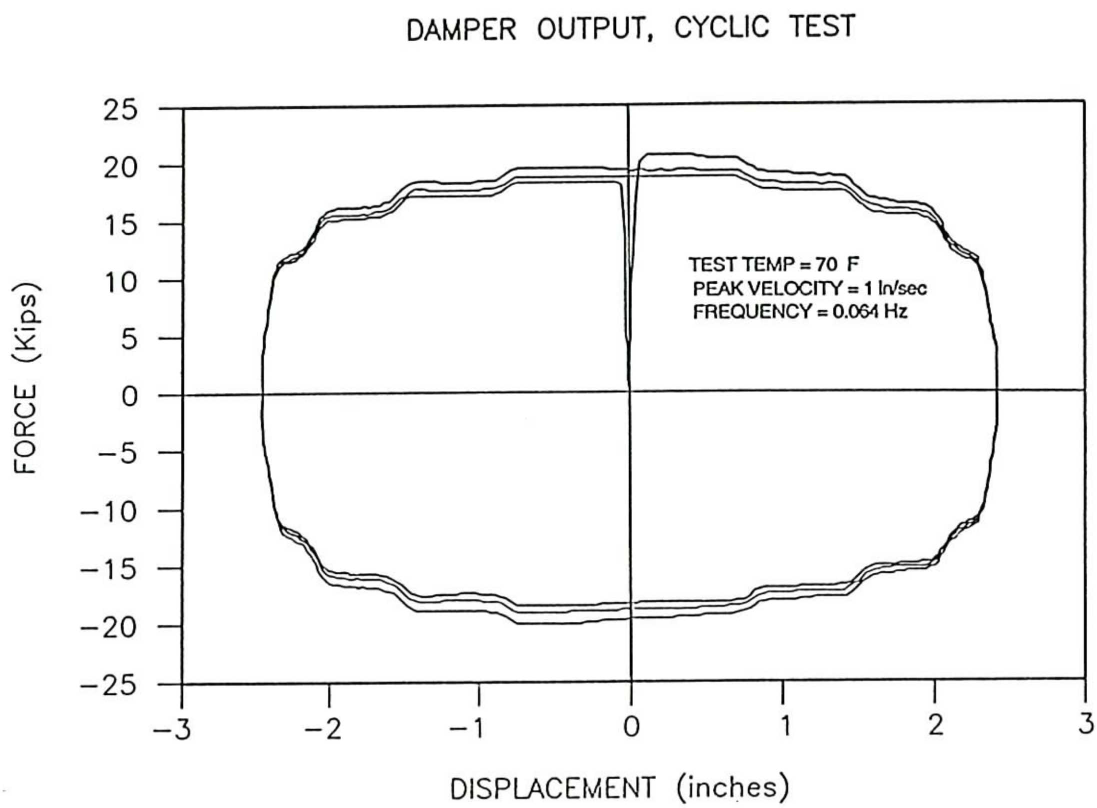


FIGURE 10

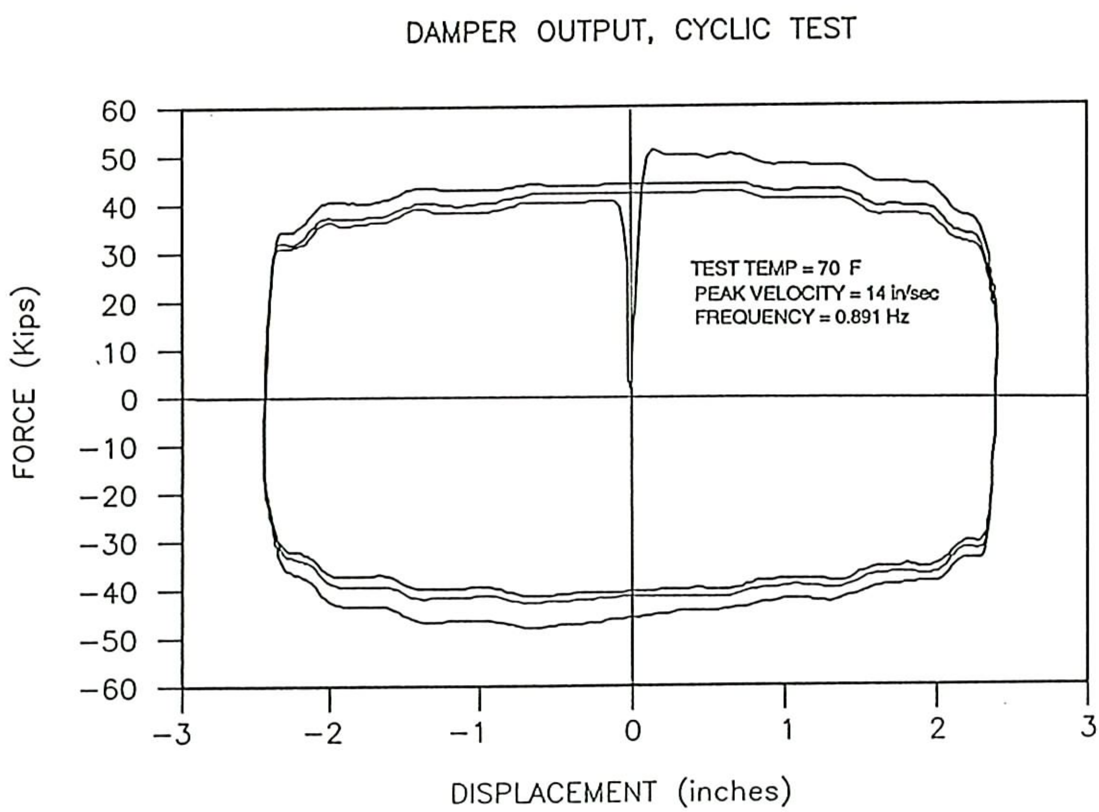


FIGURE 11

DAMPER OUTPUT, CYCLIC TEST

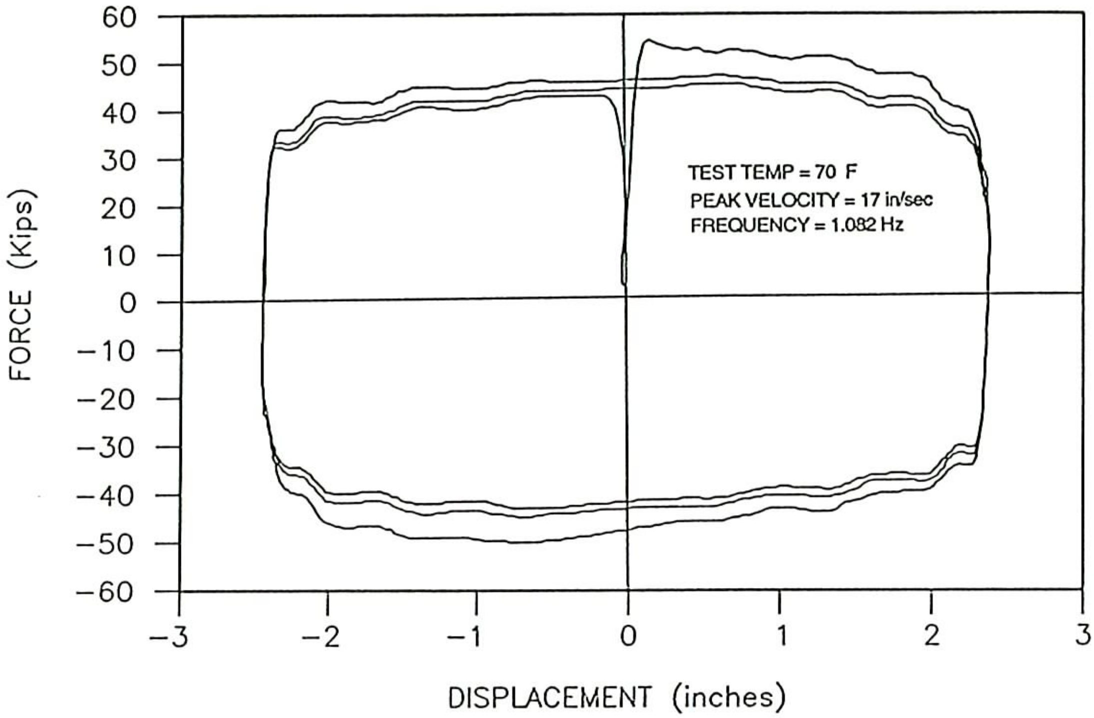


FIGURE 12

DAMPER OUTPUT, CYCLIC TEST

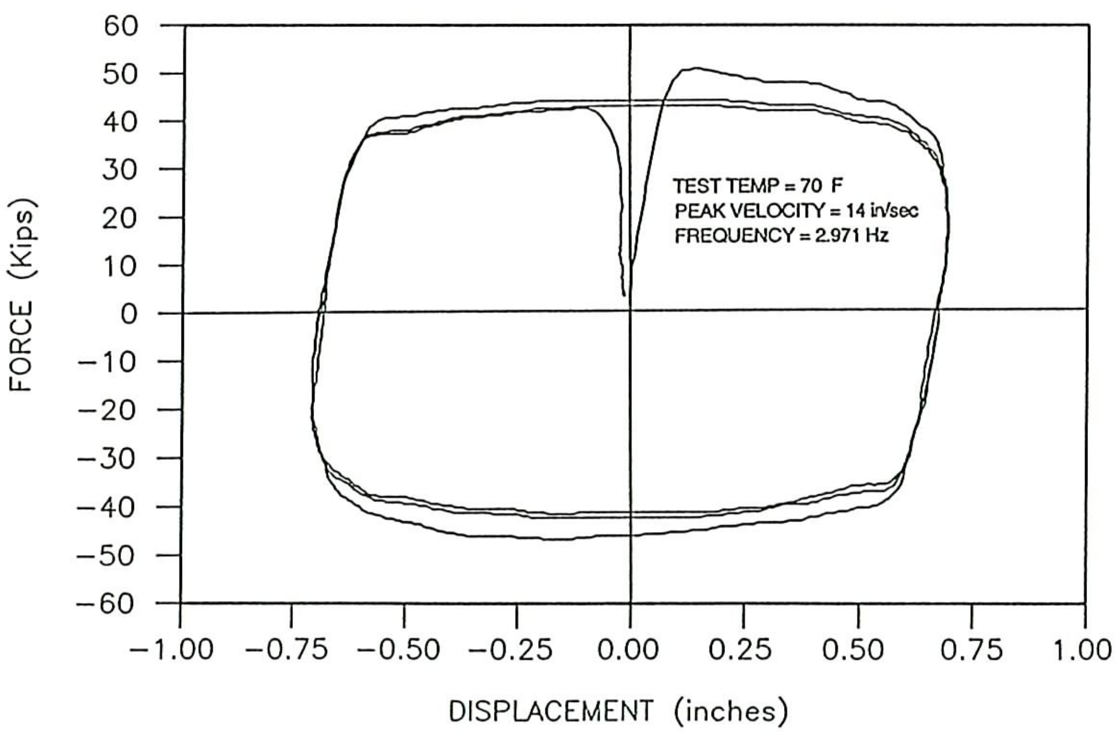


FIGURE 13

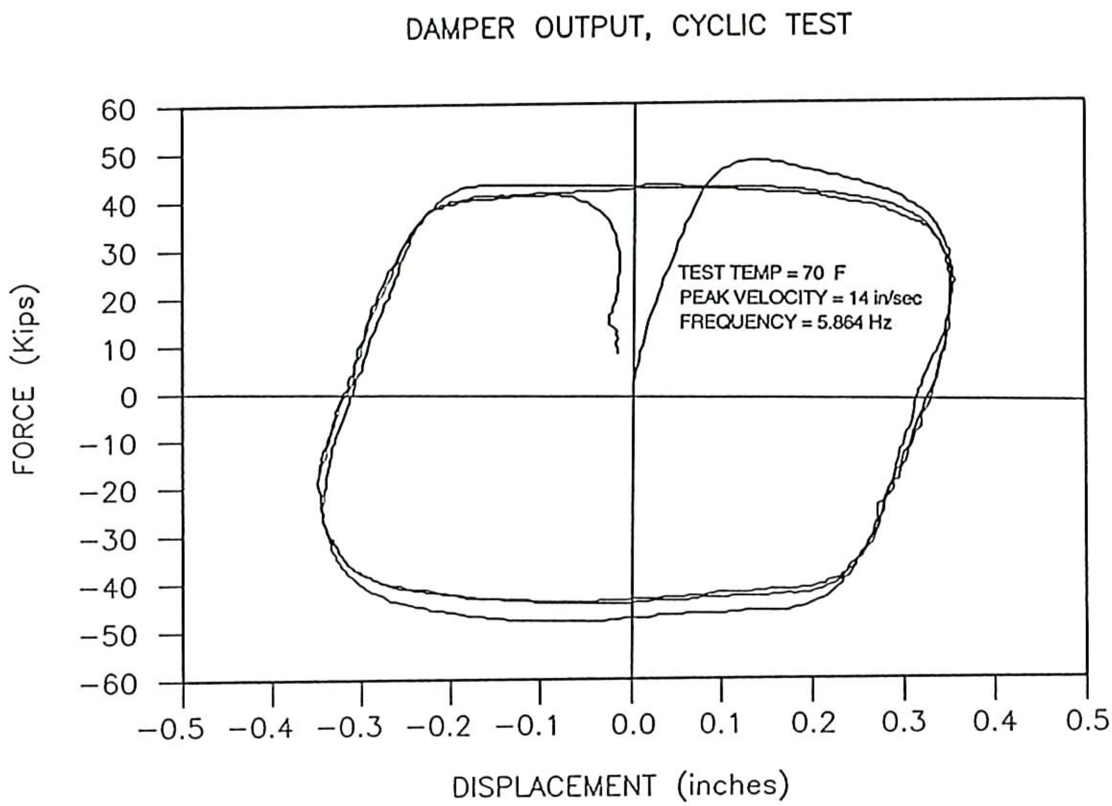


FIGURE 14

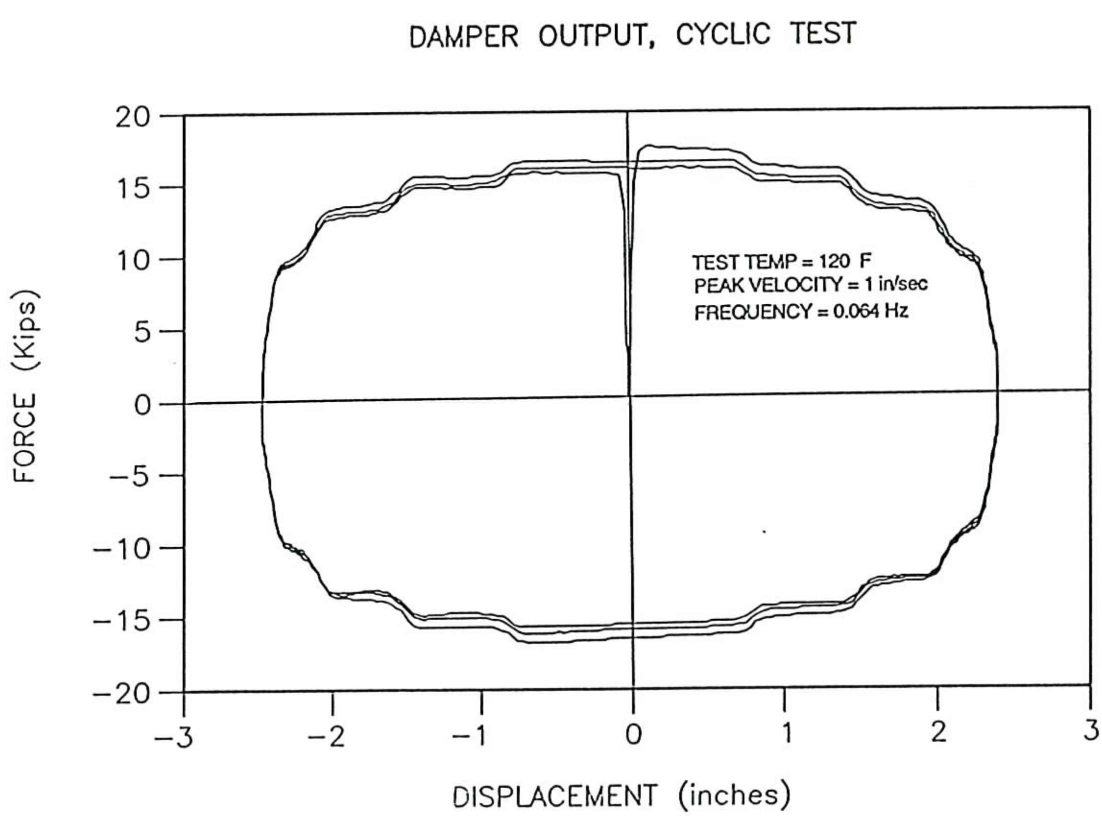


FIGURE 15

DAMPER OUTPUT, CYCLIC TEST

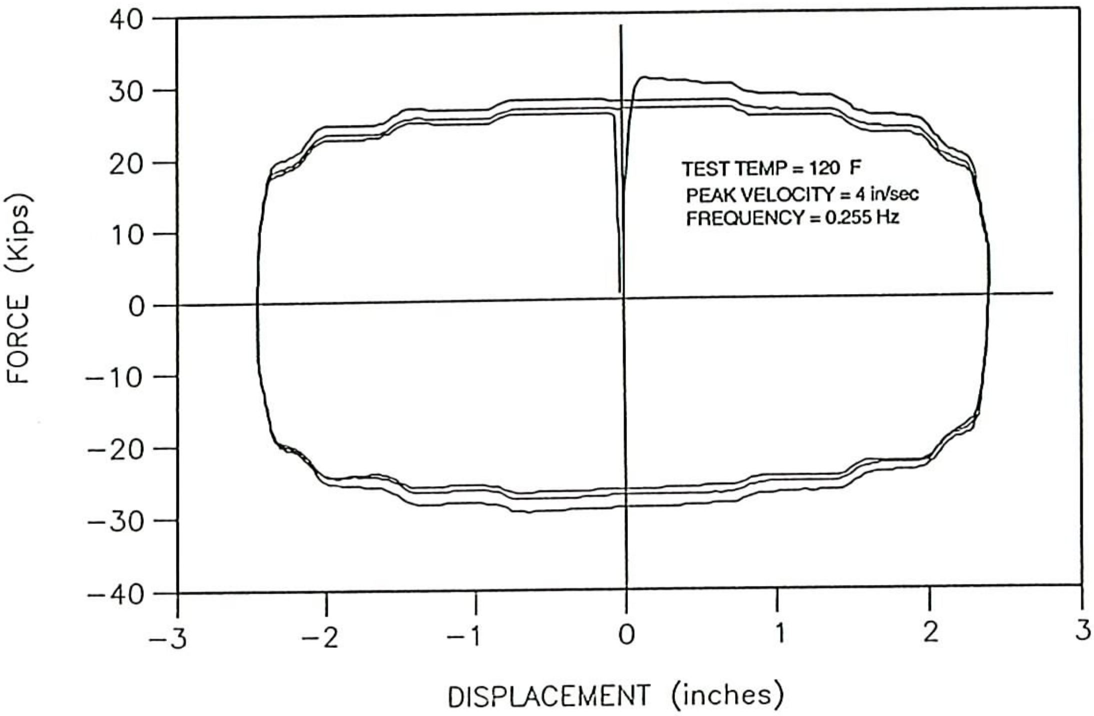


FIGURE 16

DAMPER OUTPUT, CYCLIC TEST

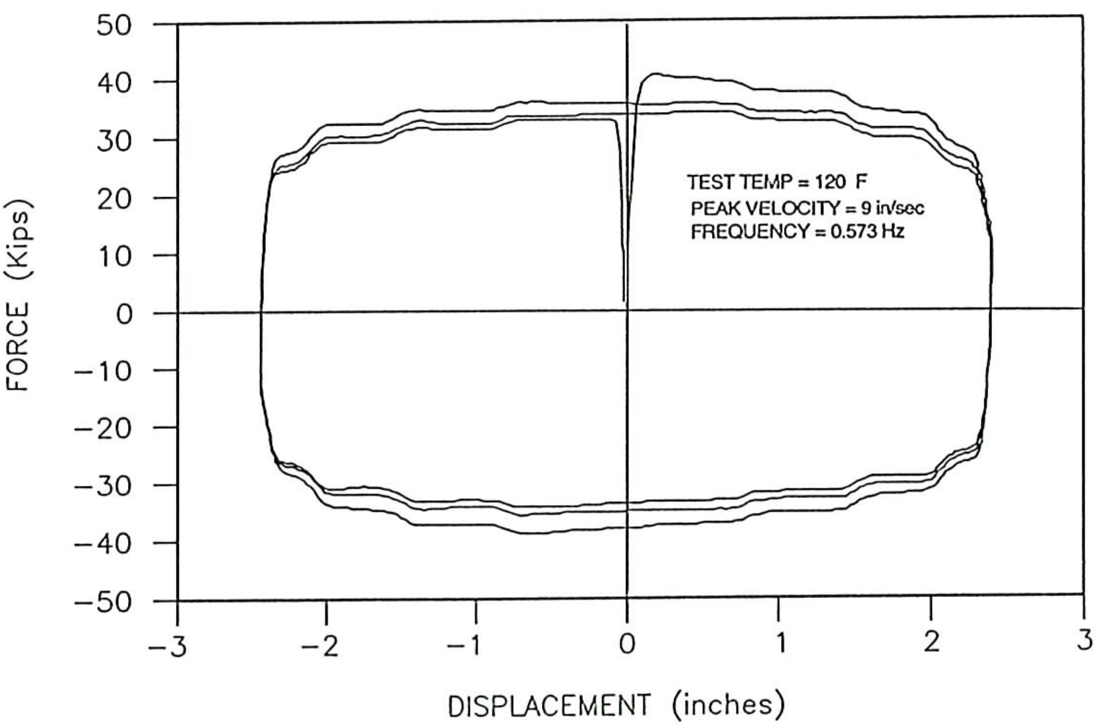


FIGURE 17

DAMPER OUTPUT, CYCLIC TEST

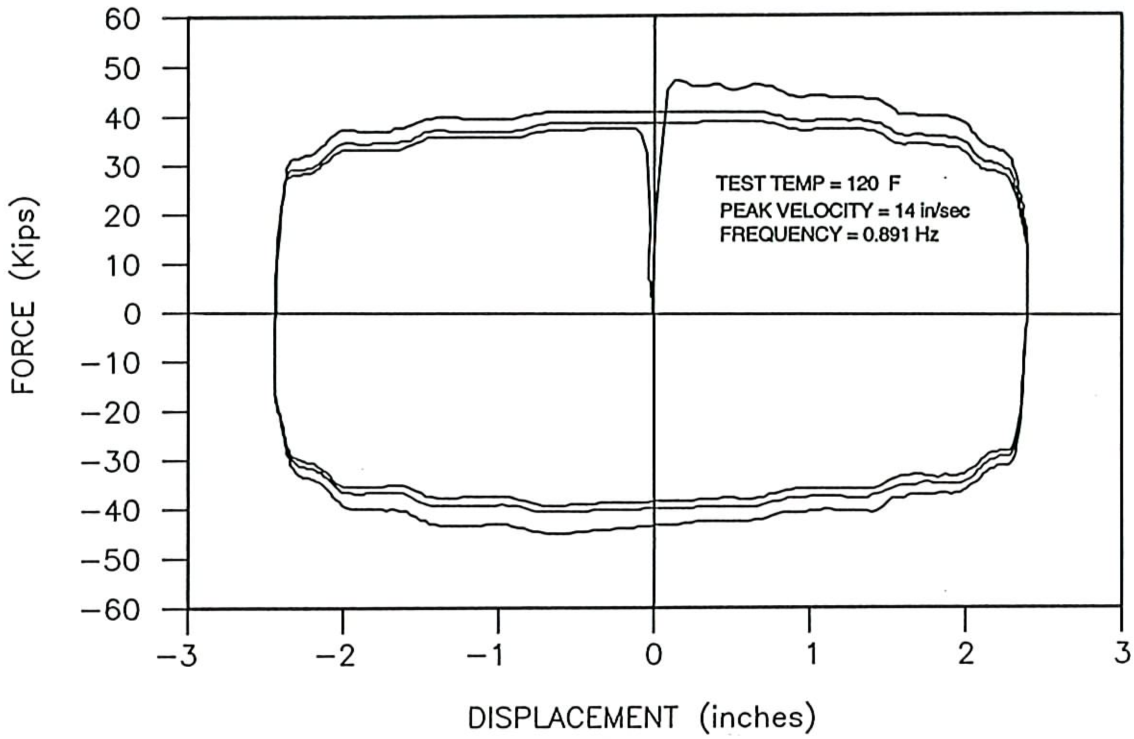


FIGURE 18

DAMPER OUTPUT, CYCLIC TEST

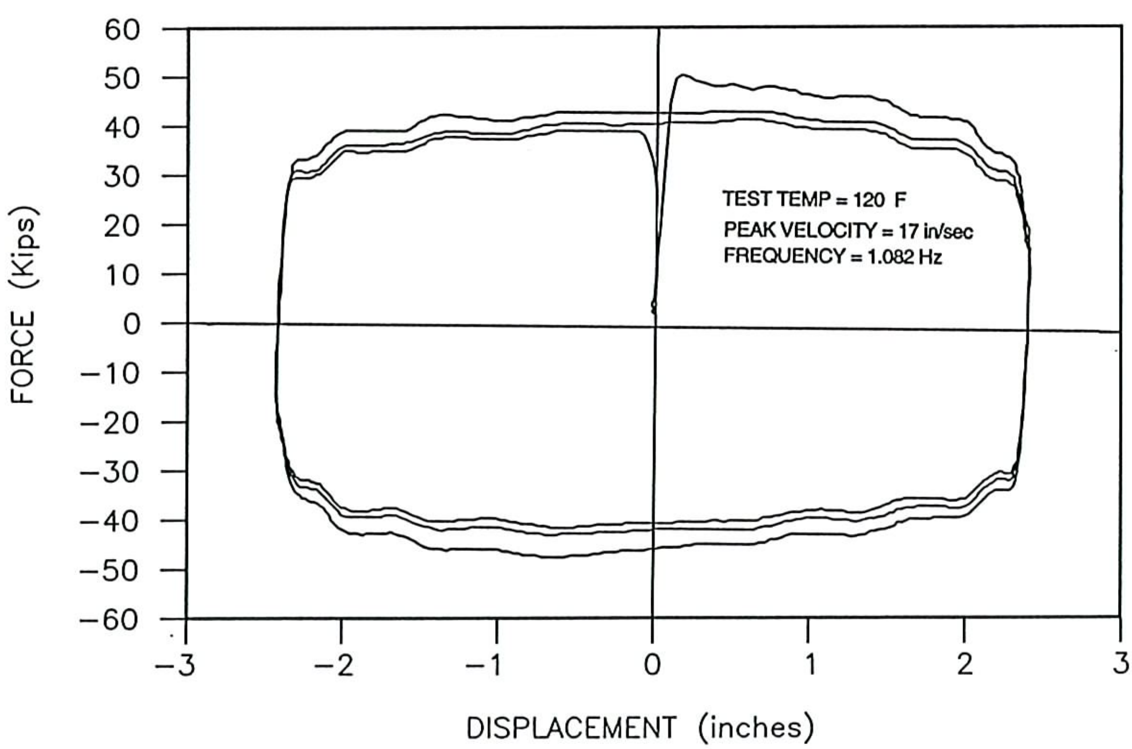


FIGURE 19

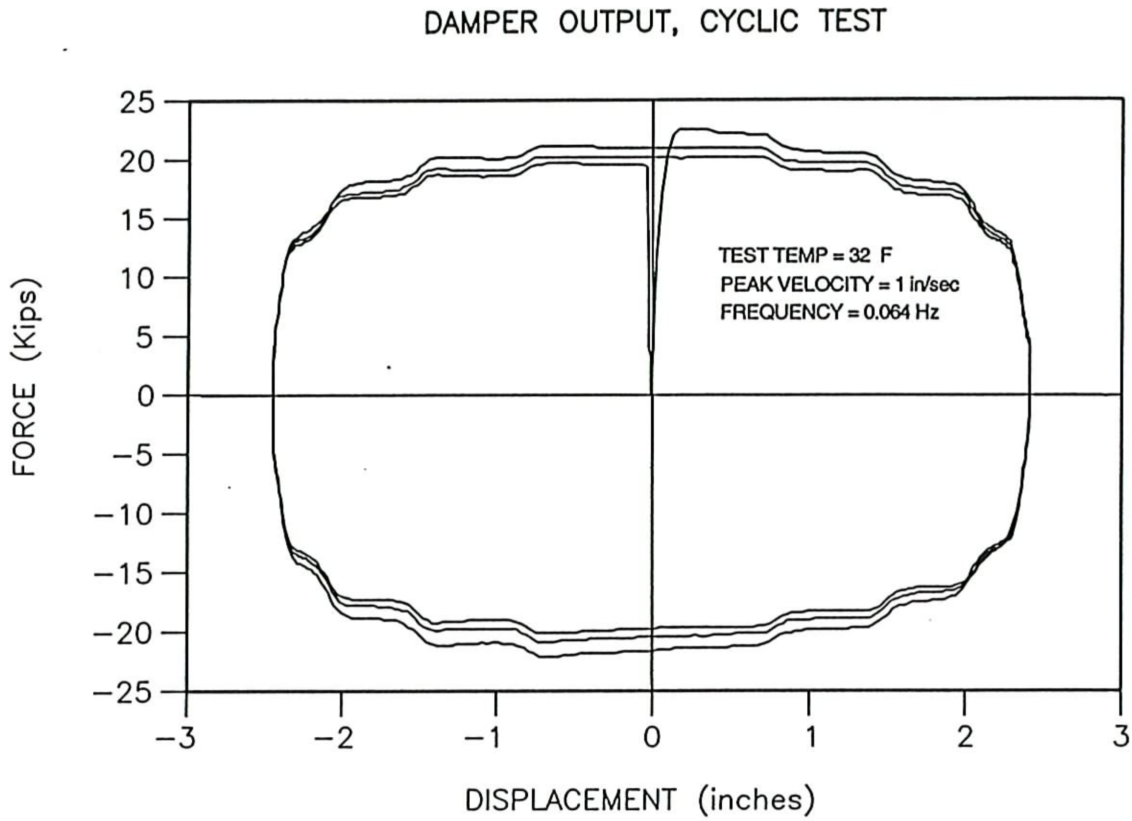


FIGURE 20

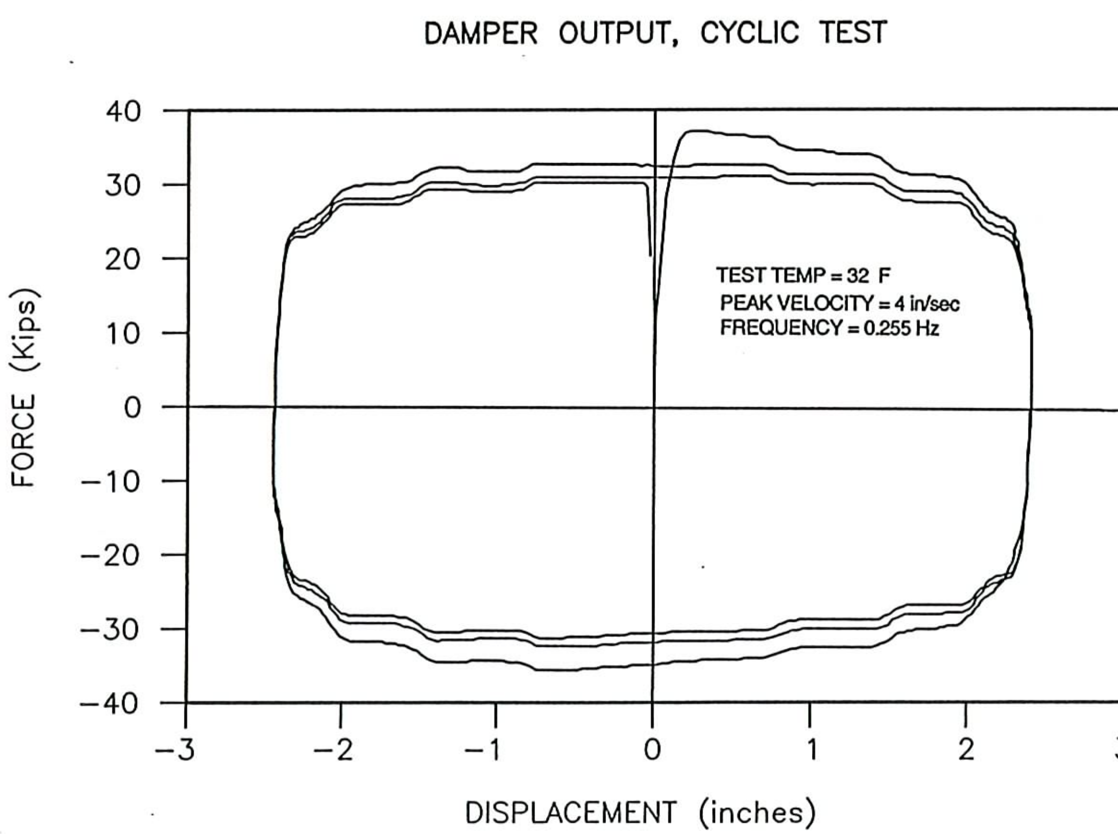


FIGURE 21

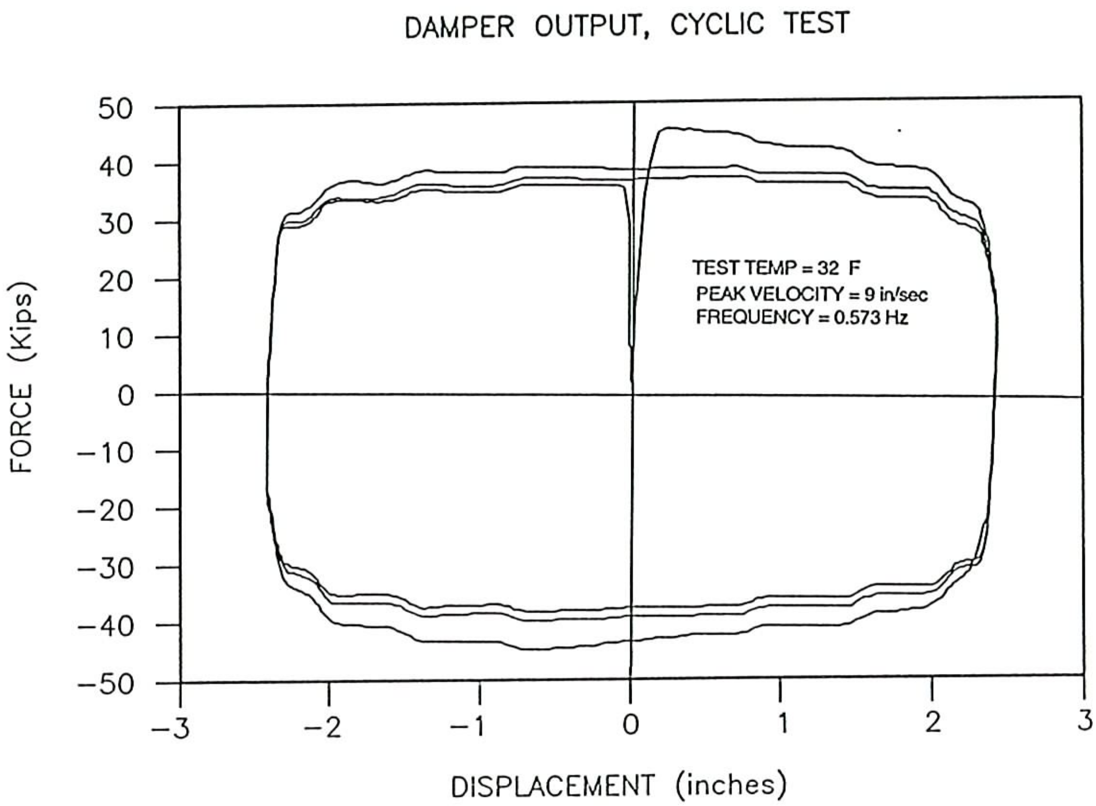


FIGURE 22

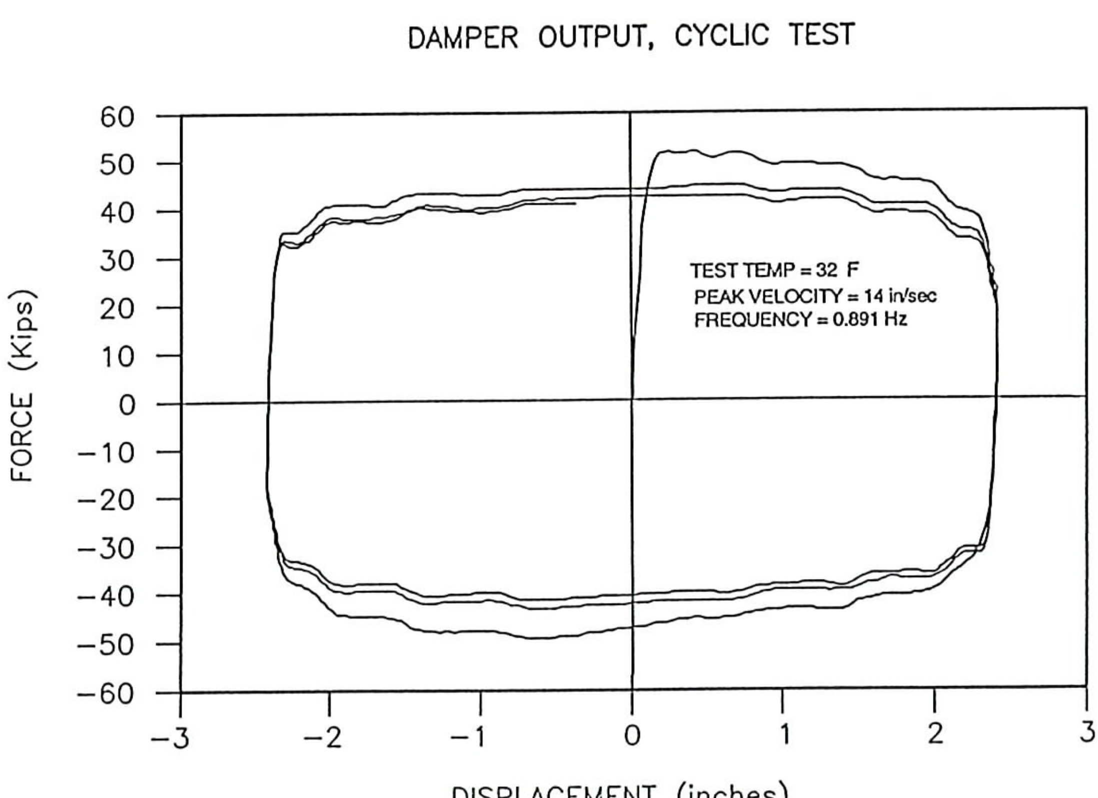


FIGURE 23

DAMPER OUTPUT, CYCLIC TEST

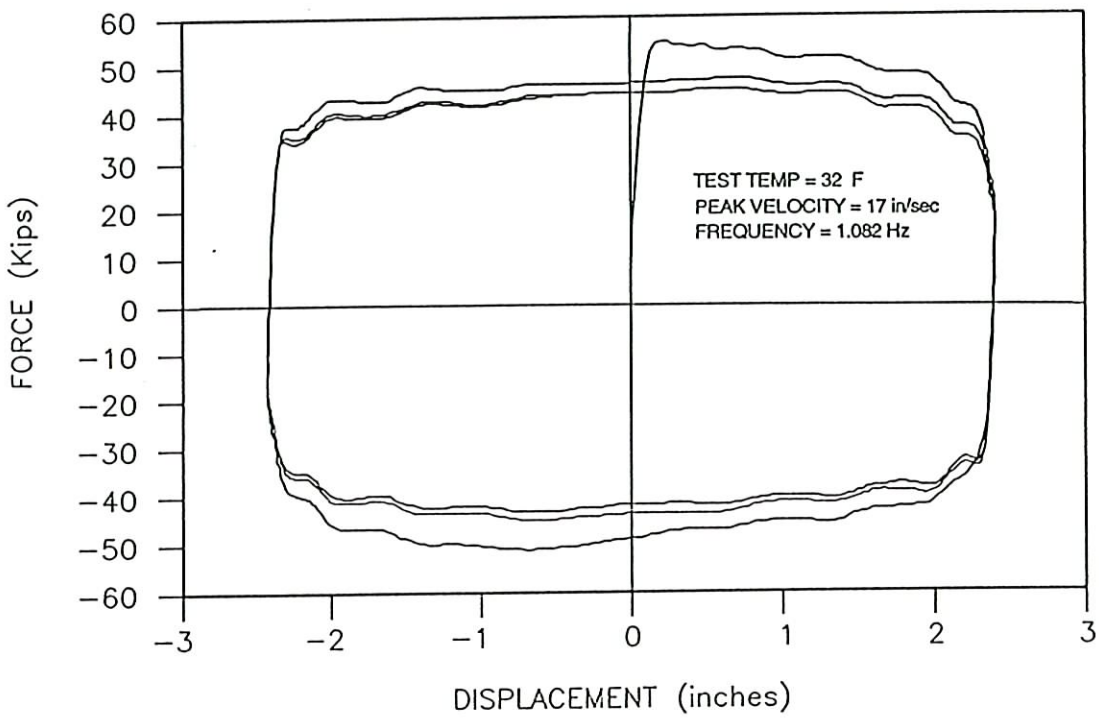


FIGURE 24

DAMPER OUTPUT, CYCLIC TEST

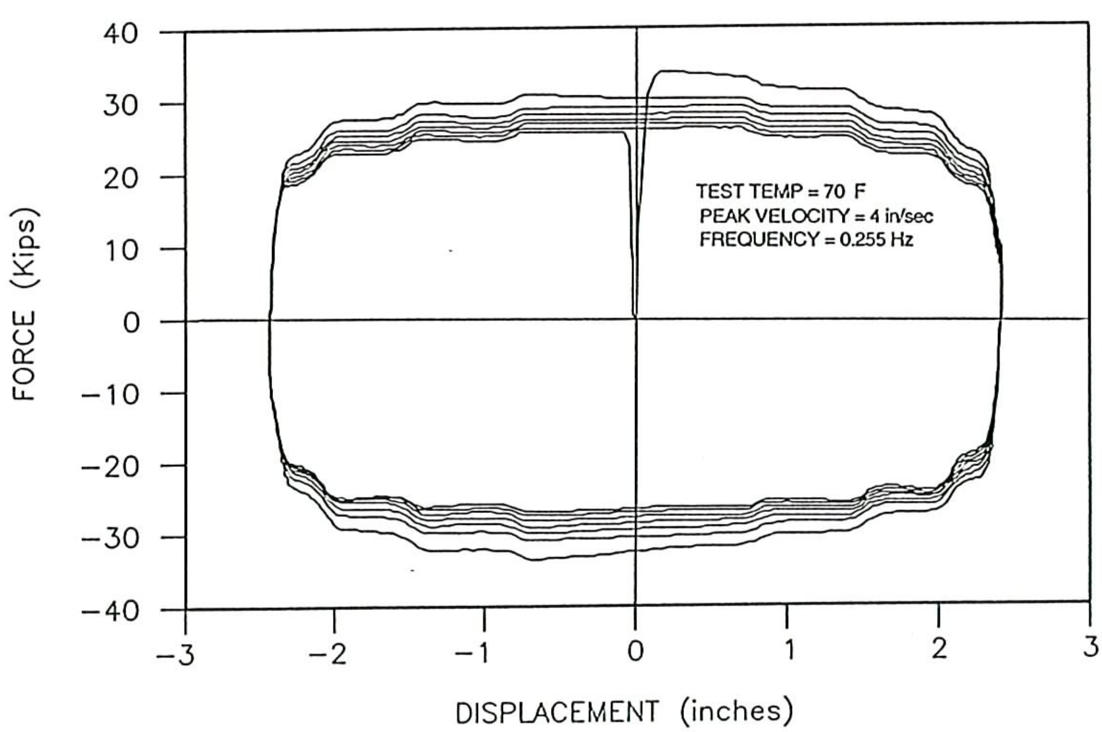
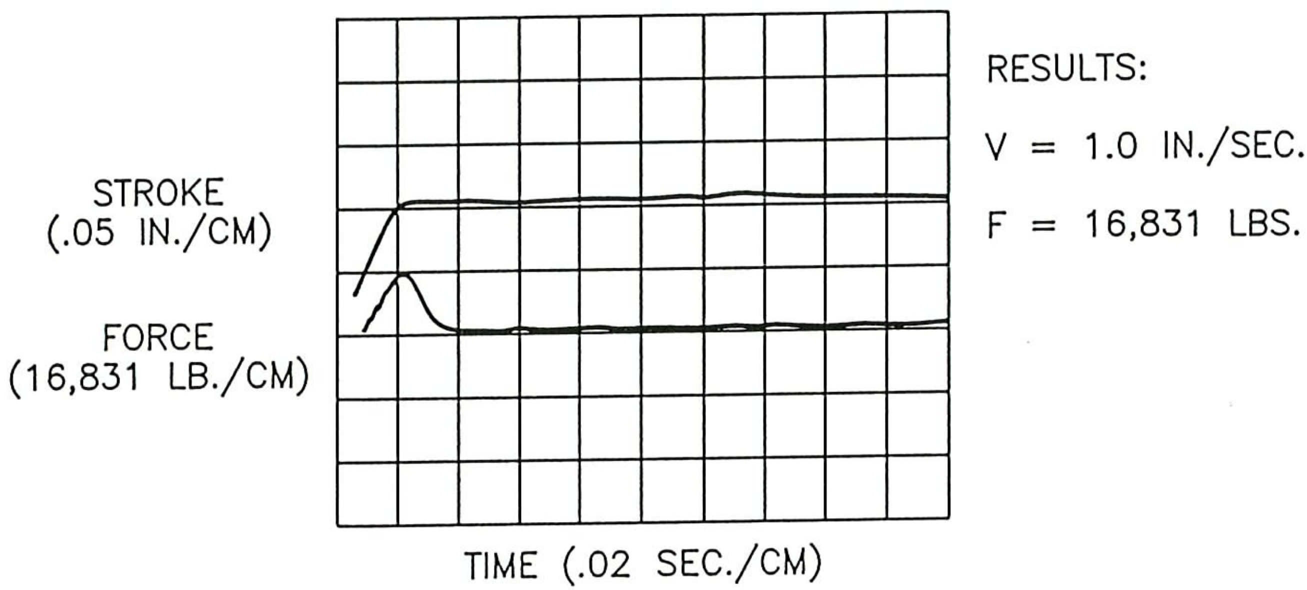


FIGURE 25



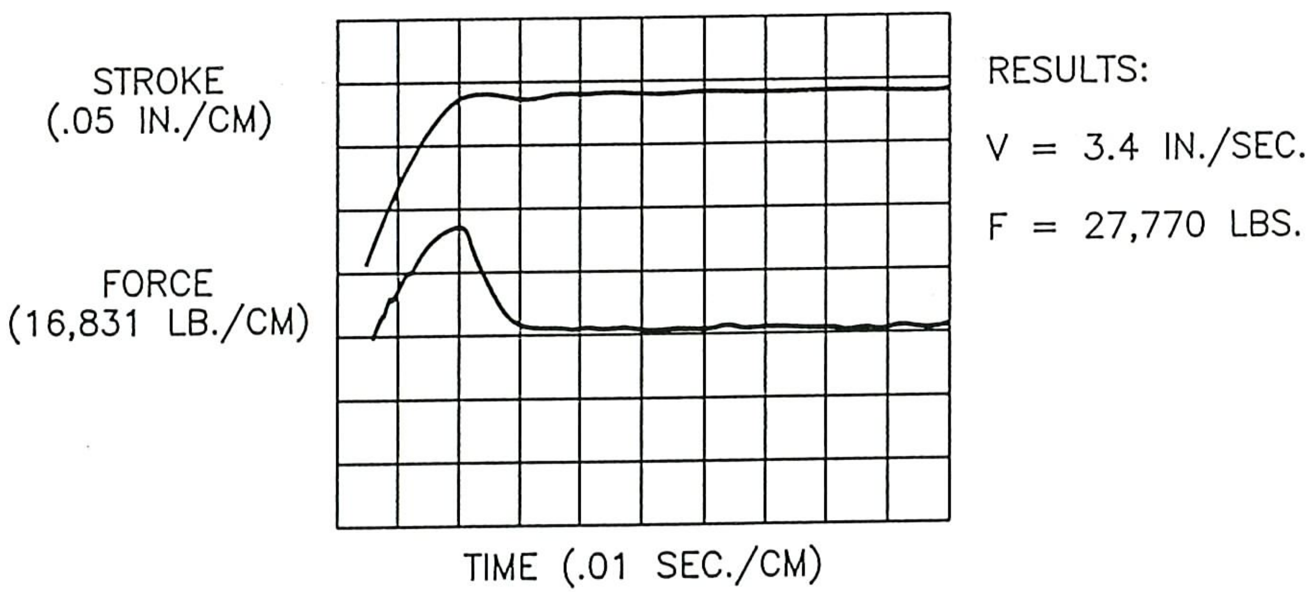
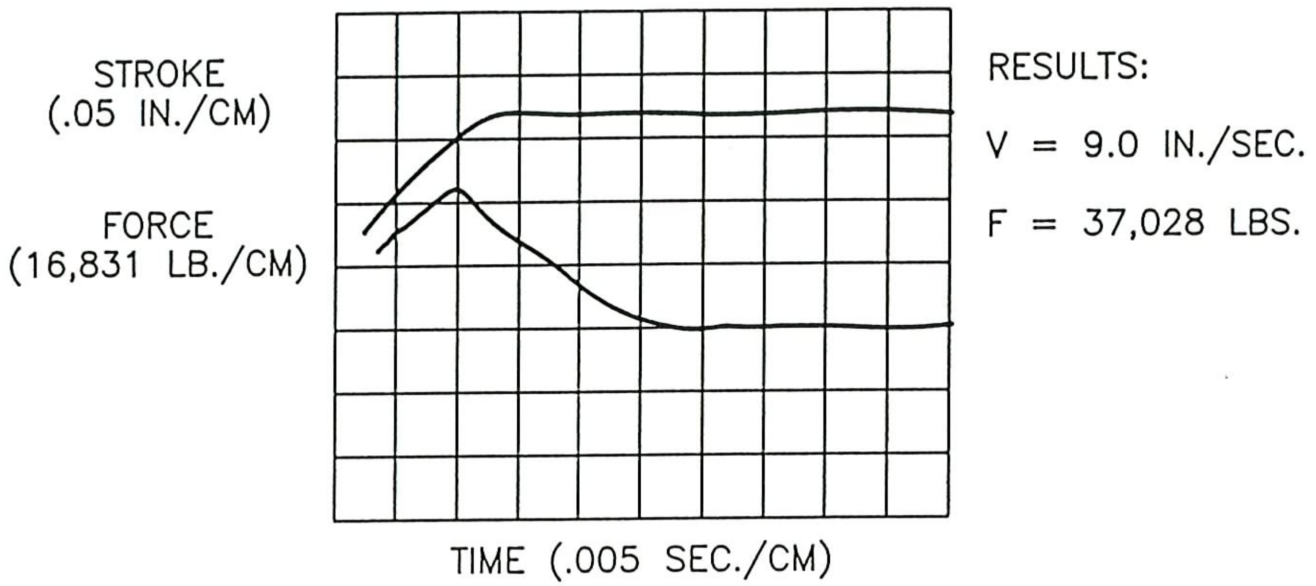


FIGURE 26

FIGURE 27



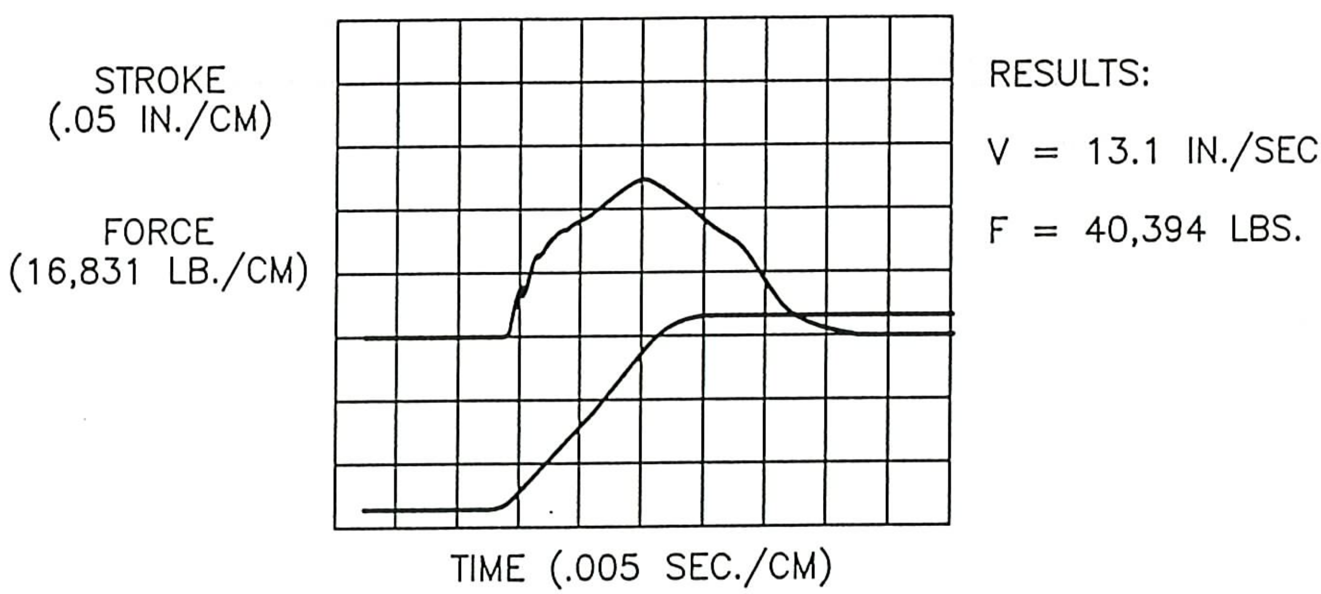


FIGURE 28

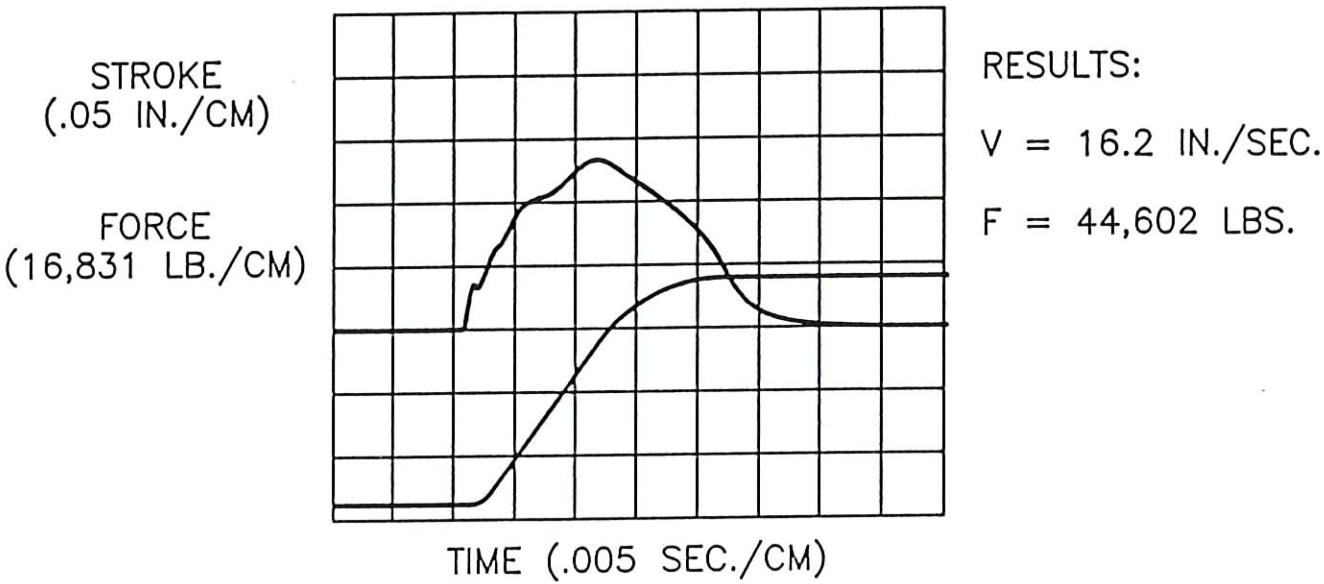

STROKE
(.05 IN./CM)
FORCE
(16,831 LB./CM)
TIME (.005 SEC./CM)
RESULTS:
V = 16.2 IN./SEC.
F = 44,602 LBS.

FIGURE 29

FORCE VS VELOCITY

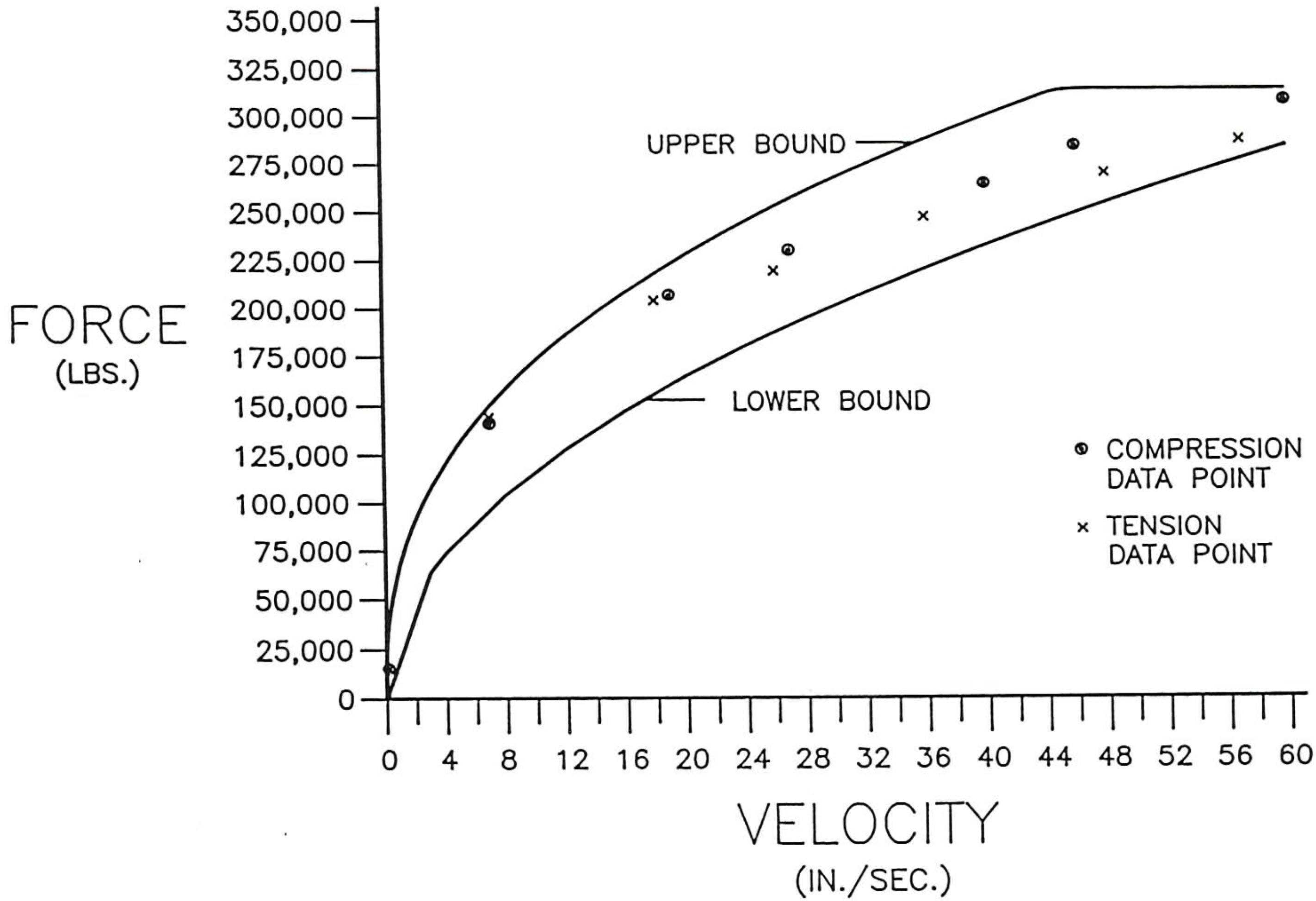


FIGURE 30

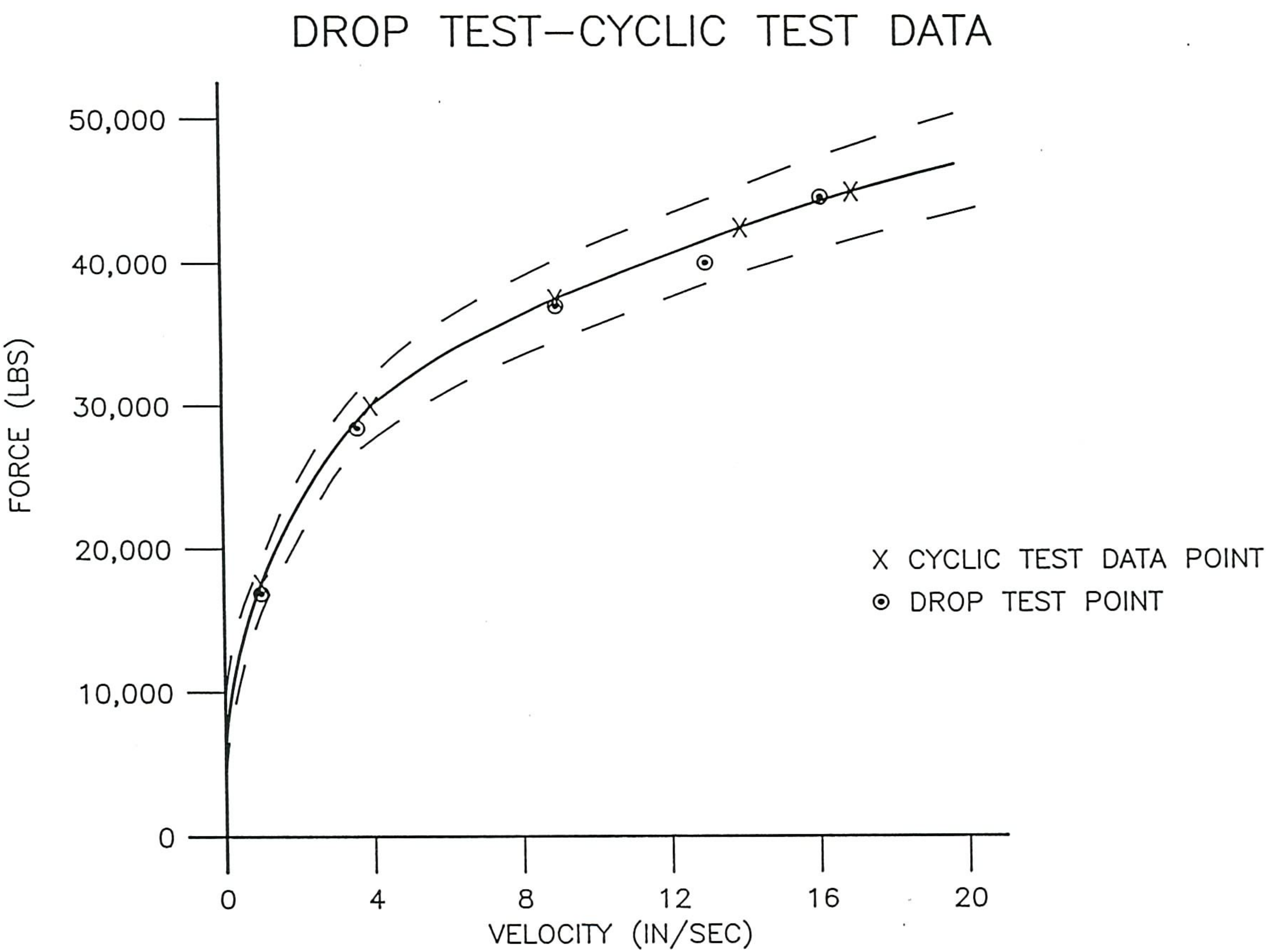

DROP TEST-CYCLIC TEST DATA
FORCE (LBS)
50,000
40,000
30,000
20,000
10,000
0
0
4
8
12
16
20
VELOCITY (IN/SEC)
X CYCLIC TEST DATA POINT
⊙ DROP TEST POINT

FIGURE 31

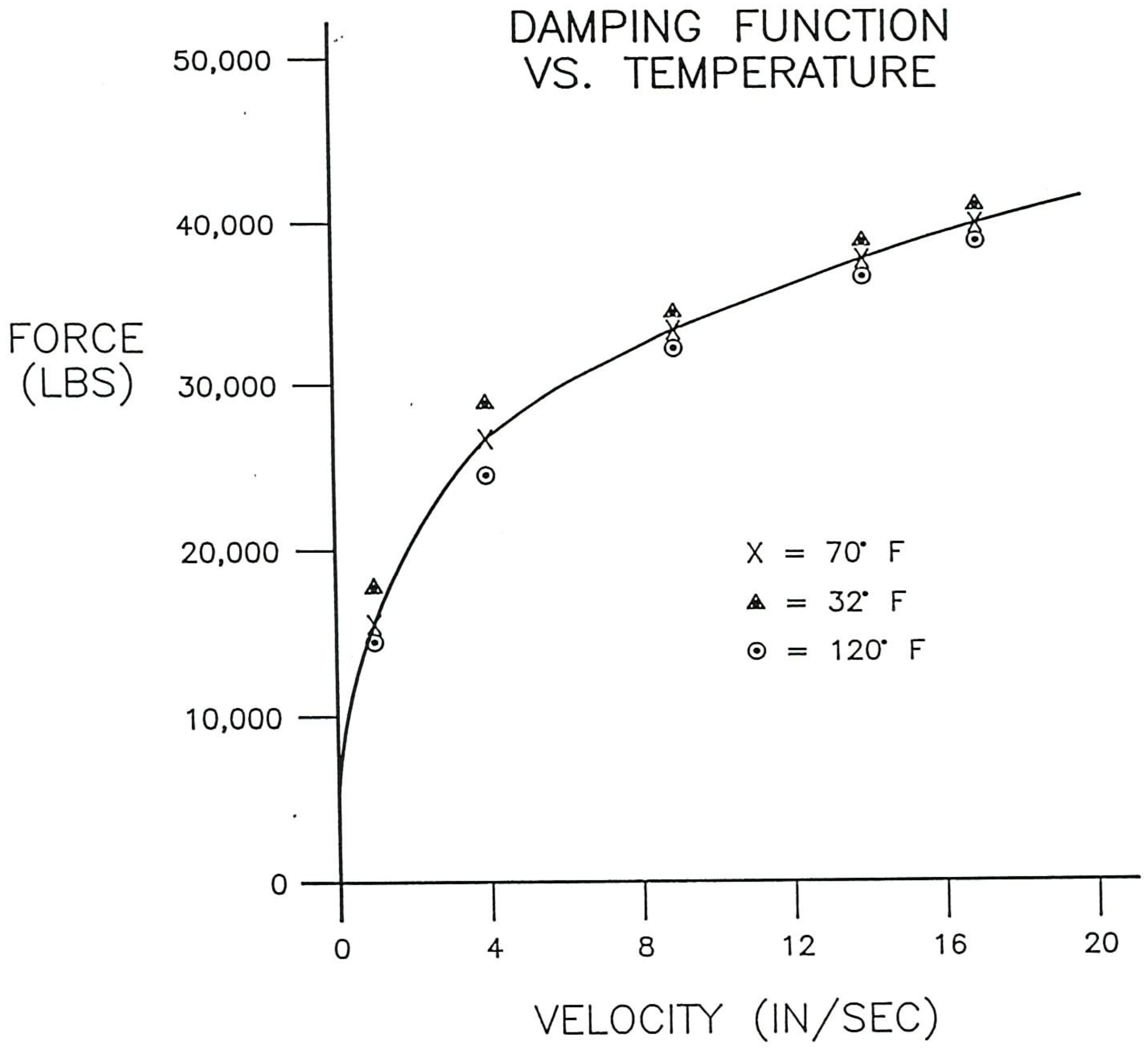

DAMPING FUNCTION
VS. TEMPERATURE
FORCE
(LBS)
50,000
40,000
30,000
20,000
10,000
0
0
4
8
12
16
20
VELOCITY (IN/SEC)
X = 70° F
▲ = 32° F
⊙ = 120° F

FIGURE 32

FIGURE 33