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Design and Operation of Equipment for Impact Test of a Hydraulic Cushion

Harshadbhai R. Patel
Brigham Young University - Provo

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DESIGN AND OPERATION OF EQUIPMENT FOR IMPACT TEST OF A HYDRAULIC CUSHION

A Thesis

Presented to the

Department of Mechanical Engineering Science

Brigham Young University

In Partial Fulfillment

of the Requirements for the Degree

Master of Science

by

Harshadbhai R. Patel

August 1968
This thesis, by Harshadbhai R. Patel, is accepted in its present form by the Department of Mechanical Engineering Science of Brigham Young University as satisfying the thesis requirement for the degree of Master of Science.

20 JUNE 1963
Date
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The Rich Soft Cushion Bumper Company, Sacramento, California, is appreciated for the purchase of equipment and materials used in these tests.
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CHAPTER I

INTRODUCTION

In recent years, experiments have been carried out to evaluate the performance of water-filled cushion cells used to attenuate energy of automobile collisions. The water-filled cushion cell is a vinyl plastic cylinder of 6 inches nominal outside diameter, 1/4 inch wall thickness, 40 inches length, closed at the bottom by a cast-in-place vinyl plug and partially closed by a bolted-in vinyl diaphragm at the upper end. These cells are designed to be installed in the path of a crashing automobile to absorb and dissipate the kinetic energy of impact. Properly designed cushions could be used as one means of saving life and property.

The main purpose of this thesis was to design and construct equipment which can be used for testing the hydraulic behavior of the cushion cell, including the capability of geometrically different orifices. The validity of the operation of this equipment was shown by impacting the cell with the hammer and measuring the deceleration of the hammer after impact and the pressure inside the cell. The design criteria for this device included a weight of 200 lbm maximum and velocity of 35 mph. A hammer, hammer arm, releasing mechanisms for hammer and hammer arm, a seismic barrier and other supporting equipment were designed to satisfy the desired requirements. Necessary instrumentation
was employed for measuring the above-mentioned variables at different speeds as functions of time.

The swinging hammer was released to strike a cushion cell resting against the seismic barrier on which was mounted a catching device for the free hammer arm. Provisions were made for varying the weight of the hammer and the height at which it was dropped. This, in turn, varied the impact force on the cell.

This thesis describes the design of various members of the equipment and the instrumentation necessary for measuring the deceleration and the pressure as functions of time. The apparatus was tested for 80 lbm weight dropped from different heights. The peak acceleration and pressure were plotted against the velocity at the time of impact. It was observed that the peak pressure inside the cushion cell and the peak acceleration of the hammer were increased with the increased speed. However, the rate of increase of pressure was not linear with speed.

This study did not fully analyze the effect of the orifice configuration on individual cell basis. It is suggested that tests of geometrically different orifice configurations be conducted in the future.
CHAPTER II

DESIGN CRITERIA

The impact force exerted by the pendulum hammer is a function of mass and velocity. A higher impact force on the desired object can be produced by heavier mass striking it with larger velocity. In a stationary unit, one way of generating a large impact is to drop a heavy mass through a very large height.

The design criteria for this device included a weight of 200 lbm maximum and velocity of 35 mph. It was so designed that possible combinations involving lower speed and/or mass could be made.

The range for the weight of hammer was restricted to between 80 lbm and 200 lbm; a 140 lbm hammer can be obtained with a slight deviation in the center of gravity of the mass. The actual design was such that weight can be added or removed from the hammer assembly. The weight was restricted to 200 lbm maximum by the design strength of the hammer arm and the supports. The hammer was designed to cover not more than three cells when it struck them. This in turn limited the shape and the size of the hammer, and hence the weight of it.

The maximum height through which the weight could be dropped was 4.4 feet. This height gives approximately 35 mph at impact. Imposition of this maximum was due to the limitations of space and cost. Arrangements were made to drop the hammer from lower heights.
CHAPTER III

DESIGN OF THE HAMMER ASSEMBLY

This chapter discusses the design of the hammer, the hammer arm, the releasing device for hammer and the positioning mechanism (see Fig. 1). The positioning mechanism was used to hold the hammer assembly at the desired height and to release it at the required moment.

Figure 1. The Hammer Arm with Positioning Pulley
(a) Hammer arm, (b) Forks (c) Axle
(d) Positioning pulley (e) Releasing device for hammer arm

3.1 The Hammer and Releasing Device

The hammer was designed to serve the purpose of a test weight dropped through different heights. The weight and the shape of the hammer were the two main factors which influenced its design. It was
designed to allow variation of its weight without changing its shape. It was shaped to cover either a single water-filled cushion cell or a small cluster of cells.

A 2' x 2' x 1/2" steel plate was sandwiched between two 2' x 2' x 3/8" steel plates to obtain the desired weight of 200 lbm. The main middle plate weighed 80 lbm and could serve as a hammer when two side-plates were removed from the assembly. The plates were engaged by bolting them together with countersunk bolts. When only the front plate was engaged with the middle plate, the hammer weighed about 140 lbm.

A steel tongue (Fig. 2) having hard grooved surfaces on its two opposite sides was attached on the middle of the top edge of the main plate. Two hard steel balls in the hammer releasing device (shown in Fig. 2) were fitted into the hard grooved surfaces of the steel piece to hold the hammer. This was designed so that the hammer could be released for free flight shortly before impact, by pulling up the actuator. The action of the actuator will be described in Sec. 3.4.
Two steel knife-edges (Fig. 3) were screwed to the 1/2 inch thick plate on both the side edges.

Two forks on the hammer arm fitted into these knife-edges. These forks supported the load of the hammer during the lifting process, and fixed the hammer with respect to the hammer arm during the fall of the weight and the arm.

An accelerometer was mounted on one side of the 1/2 inch thick plate to measure the deceleration of the hammer after impact. Safety chains tied onto the hammer arm were used to carry the cable from the accelerometer and to catch the hammer after impact.
3.2 The Hammer Arm

The hammer was to be dropped at a maximum height of 42 feet from the ground. This was accomplished with a 20 foot pendulum. The use of a single steel pipe was suggested as a possible design for the pendulum arm. It was found that the deflection of the free end of the pipe was not under safety limits for economical cross-sections.

The present hammer arm (Fig. 1) was designed to meet all the criteria. Two 1 1/2 inch diameter steel pipes formed the basic member of the hammer arm frame. The frame structure limited the deflection of the hammer arm to a minimum. All the members of the frame structure were found to be safe against the working load.
The top most member of the frame structure, was welded at one end onto the rim of the positioning pulley and the other end was attached onto the junction of 1 1/2 inch pipes. The two 1 1/2 inch pipes were attached to the axle housing. A detail drawing of axle housing is given in Appendix C. One important advantage gained by designing the frame structure was the ease with which the releasing mechanism fixed at the outer end of the hammer arm could be operated. The actuator rod connected with the bell crank which was operated by contact with an adjustable rod point fixed on the top of the catching mechanism (Fig. 5).

The cable from the accelerometer was carried along safety chains which were tied onto the hammer arm. This cable was then run along one of the pipes of the hammer arm to the axle, taken down along the support pipe, and connected with power supply and recorder electronics.

3.3 The Positioning Mechanism

The function of the positioning mechanism was to position the hammer assembly at desired height and release it when desired. A 6-foot diameter positioning pulley was mounted on the axle of the hammer arm. One quadrant of the positioning pulley was cut out to reduce the overall weight of the hammer assembly. The weight was further minimized by supporting a 4 inch wide rim with 1/2 inch steel spokes spaced 45° apart. Holes were drilled in the rim at 15° intervals to fit a holding pin. A special rim was fastened onto the circumference of the pulley to
Figure 4. Link Mechanism for the Holding Pin

H   Holding Pin
P   Positioning Pulley
R   Handle
carry steel cable. One end of the cable was tied by means of three
cable clamps at the welded junction of the hammer arm and the rim of
the pulley. A loop was made at the other end so that a hook could be
inserted to link it with a cable operated by a winch mounted on one of the
supporting members of the catching device.

The hammer assembly was kept at a desired height by inserting
the holding pin in the drilled holes on the rim of the positioning pulley.
When it was desired to drop the hammer, the link mechanism shown in
Figure 4 retracted the holding pin and let the hammer assembly fall.
The holding pin could be operated by lifting or lowering the handle at the
bottom. The movement of the handle was such that when it was lifted,
the holding pin could be inserted to hold the hammer assembly. A hook
located at a convenient height of the support structure held the handle in
the safe position.

The holding pipe was checked for bending and shear failure and
found to be safe.

3.4 The Catching Device

The swinging hammer arm was brought to rest by means of
catching mechanism after the hammer was released. The catching
device served the dual purpose of actuating the link for the release
mechanism and also of stopping the swinging hammer arm. A group of
cushion cells was fixed on the vertical plate of the catching mechanism
for the energy absorption of the swinging arm.
The catching mechanism was a simple frame structure which supported two plates. A movable clamp (Fig. 5) was bolted to the top of the horizontal plate.

![Diagram of catching mechanism](image)

Figure 5. Movable Clamp with Projecting Rod
(a) Movable clamp (b) Projecting rod
(c) Bell crank (d) Actuator (e) Cushions for free hammer arm

The projecting rod was positioned in the clamp to strike the bell crank on the swinging hammer assembly and operate the link. This, in turn, pulled the actuator rod and the hammer was released.
CHAPTER IV

DESIGN OF SUPPORT TOWER

The support tower was designed to support the 20-foot hammer arm in any position (see Fig. 6). The main criteria for design were the deflection of the top end of the tower and the strength of the supporting structure. It was found that the support structure was suitable. The deflection of the end was under safety limits and the cross-sections of every member were strong enough to resist the impact load with a safety factor of about three. This chapter discusses the present form of support.

The main supports were 3 inch diameter, 23 feet long steel pipes. The deflection of this support was kept to a safe value by means of a frame structure. The distance between the bottom ends of the main support pipe and the 1-inch structural steel pipe was 4 feet. Supporting plates for the axle on which the hammer arm and the positioning pulley were mounted were fastened at the junction of the main support pipes, and 1 inch structural pipes at the top.

The base of the support tower was so designed that the supports assembled with the hammer arm and the positioning mechanism could be conveniently located at their positions. The base consisted of plates
Figure 6. The Support Tower
(a) Main pipe support (b) structural pipe (c) ladder (d) Seismic block (e) Seismic base (f) hammer arm (g) hammer arm release (h) Winch

(Fig. 7) welded at the bottom ends of the main support and structural pipes. These ends of the supports could easily be seated on the plates.
which were bolted with the seismic base. To properly locate the hammer assembly and to prevent the deflection of the support tower on the sides, two cross-bars were welded onto the supports. The positioning mechanism which was used to hold the hammer arm at desired height and to release it at desired moment was mounted at convenient height on the main support pipes.

One side of the support tower (Fig. 5) incorporated a ladder which allowed access to the tower. On the other side was mounted a linking mechanism for holding and releasing the hammer arm. Loops were positioned around the pulley to properly guide the steel cable into the pulley.
CHAPTER V

DESIGN OF THE SEISMIC BARRIER

All of the kinetic energy of the mass after impact was to be absorbed by the cushion cell resting against the seismic block. The seismic block, therefore, was built to be large and strong enough to resist the impact force. The seismic base was designed to take the loads of support tower and the seismic block. The design of both seismic block and seismic base is described in this chapter.

5.1 The Seismic Block

It was calculated that it should weigh about three tons to withstand the expected impact. The block used weighed about 4 1/2 tons. The seismic block was designed to support the cushion cells against heavy impact. It also supported the catching mechanism for hammer arm at a convenient location. To prevent tensile load in concrete due to pressure waves, a 4 1/2' x 4' x 3 1/2' box was constructed by welding 1/4'' steel plates. Slots were cut in the bottom of the block (Fig. 8) to make provision for a fork lift which could lift the seismic block and also to pass other apparatus.

Two pipes were run from the front side to the back side and welded to pass threaded screws through them. This allowed the fastening
Figure 8. The Seismic Block
(a) Slots for fork lift
(b) Bolt-down channels
(c) Provision for other apparatus
of apparatus to the block. The block was reinforced by welding steel bars at different intervals into the form before pouring. Four channels were welded at the bottom corners of the block to stiffen the assembly.

Two bolt-down channels (channels made by welding a 3/4 inch wide piece of steel in between two 2 inch standard channels to accommodate bolts in the gap) were welded on the top of the block running from the front edge to the back edge. The catching mechanism for the hammer bolted to these channels. Angle-brace members were provided to fasten the block to the base. These members were designed to be joined by bolts with the bolt-down channels fixed on the top of the seismic base.

5.2 The Seismic Base

The seismic base was designed to withstand the load of concrete block and served as a foundation for the supports. A detail drawing of the seismic base is given in Appendix B. Four J-bolts were fixed within pieces of pipe and cast into the concrete base at each support position. The base was reinforced with bars and wire mesh. Four bolt-down channels to which the brackets on the seismic block could be attached were placed at required intervals. The channels were cast into the concrete and held in place by welded reinforcements.
CHAPTER VI

INSTRUMENTATION

It was desired to measure the deceleration of the hammer and the pressure inside the cushion cell as functions of time during impact at different impacting speeds of hammer. The set of instruments used to measure these variables is shown in Figure 9. The selections of measuring systems for these quantities are described in this chapter.

![Figure 9. Set of Instruments](image)

6.1 Selection of Deceleration Measuring System

The impact acceleration is directly proportional to the height through which the weight is dropped and inversely proportional to the
stopping distance. A unidirectional accelerometer measuring 0 to 100g was mounted on the hammer (Fig. 10).

Figure 10. Accelerometer Mounting on the Hammer

Proper connections for 28 V D.C. supply, and 5 V D.C. output were made. The cable carrying both input and output wires was carried along one of the pipes of the hammer arm up to the top and along one of the main support pipes from the top back to ground level.

The specifications of the accelerometer, recorder and the constant d.c. power supply source are given in Appendix A.

6.2 Selection of the Pressure Measuring System

A piezoelectric pressure transducer was mounted on the water-filled cushion cell as shown in Figure 11. The response from the pressure
transducer was conditioned in a charge amplifier. The charge amplifier signal was fed into the recorder.

The specifications for these instruments appear in Appendix A.

![Figure 11. Pressure Transducer Mounting on a Cushion Cell (a) Piezoelectric pressure transducer (b) Cushion cell (c) Orifice]

6.3 Velocity Measuring System

Arrangements were made to measure the impacting velocities (see Fig. 12). Two switches spaced one foot apart were mounted on a stand. The stand was positioned near the impact point so that one of the forks of the swinging hammer arm actuated the switches in turn giving a signal which was recorded on a digital counter. Constant d.c. power was supplied to the circuit. The velocity was then calculated by noting the time taken by a swinging hammer in passing a distance of one foot.
Figure 12. Velocity Measuring Device
Fig. 13 Schematic Diagram for Instrumentation
Precautions were taken before impacting the cell. The system of instruments was calibrated and the test procedure was established. Tests were performed by impacting a single water-filled cushion cell fitted with an orifice with 80 lbm weight dropped through different elevations.

One test was performed without the orifice diaphragm in the cell. Readings were compared and graphs were plotted for the peak acceleration and peak pressure versus the impacting velocity.

7.1 Test Procedure

Having made the proper connections of the apparatus, as shown in Figure 13 the entire system was calibrated. The calibration procedure was as follows:

The charge amplifier, the digital counter and the recorder were balanced after allowing the entire system to reach equilibrium (about 30 minutes). The accelerometer was tapped gently to assure that it was working properly.

In preparation for test, the hammer assembly was positioned by winding the steel cable on the winch. The hammer assembly was held in
position at the desired height by inserting the holding pin in one of the holes in the positioning pulley.

The tension on the steel cable was released by unwinding the cable on the winch. The hook joining the two cables was removed. This allowed the cable attached to the hammer arm to move freely on the positioning pulley.

The cushion cell was filled with the water before the hammer arm was released for impact. The hammer assembly was then released by retracting the holding pin from the hole for impacting the cushion cell.
CHAPTER VIII

DISCUSSION OF RESULTS

Tests were performed for the impact on the single cell with the 80 lbm hammer dropped through five different heights. The responses of the water pressure and the acceleration of the impacting mass were collected as functions of time, at different impacting velocities. The peak values for both variables were measured and plotted against the impacting velocities. Figure 14 and 15 show curves of pressure and acceleration versus impacting velocity.

The impact acceleration was proportional to the height through which the weight was dropped. Mathematically, acceleration is the first derivative of velocity with respect to time and hence if the velocity increases with respect to time, the acceleration increases. It was found that as the impacting velocity increased, the acceleration was increased. The same was the case with the water pressure inside the cell. It increased as the velocity increased. The impact force is the function of acceleration. As acceleration increases, the force increases and hence the pressure.

It was noticed that the impacting velocities calculated by noting the time taken by the hammer arm in travelling a distance of one foot were
Fig. 14 Pressure versus Impacting Velocity

SCALE
x-axis 1 in = $20\frac{2}{3}$ mph
y-axis 1 in = 20 psi
Fig. 15  Acceleration versus Impacting velocity

SCALE
x-axis  1 in. = 20/3 mph
y-axis  1 in. = 20 g
Fig. 16. Variation of Actual Velocity from Theoretical Velocity
less than the expected velocities calculated on the basis of conservation of energy. This was due to the friction in the ball bearings and the air resistance to the hammer arm. It was also observed at higher velocity that the theoretical velocity calculated on the basis of conservation of energy was less than the actual velocity. This is an unusual situation. The only reason that could be given is an operational defect of speed trap. It was calculated that a slight variation in the microsecond reading at larger velocity affected the overall speed in miles per hour more significantly than at lower velocity. The uncertainty bands are shown in Fig. 16 for 0.5 and 1.0 microsecond time trap error.

It was noted that the peak pressure in the cell without orifice was of the same order of magnitude as the peak pressure in the cell with orifice. In both cases the cell was struck with the same mass dropped from the same height. The acceleration histories for both tests was quite similar.
CHAPTER IX

CONCLUSIONS AND RECOMMENDATIONS

9.1 Conclusion

The main aim of this work, as described in the introductory chapter of this thesis, was accomplished. Equipment was designed and was constructed for the impact test of an hydraulic cushion. The validity of the operation of the equipment was checked by performing impact tests on a single water-filled cushion cell. As expected, the release device for hammer worked well and released the hammer for free fall shortly before impact.

The dependence of pressure and acceleration upon the impacting velocity was roughly as expected.

9.2 Recommendations

The following recommendations were made in relation to the present equipment.

(1) The maximum acceleration achieved by the impact was about 100 g for the maximum height of 24 1/2 feet. The hammer could be dropped at the maximum height of 44 feet. The maximum range for the accelerometer mounted on the impacting mass was about 100 g. It should be replaced for high impacting velocities.
(2) A second piezoelectric pressure transducer should be mounted in the cell to measure the pressure at the bottom.

(3) Arrangements for joining plates to the main plate were made so it is recommended that more tests should be carried out with larger impacting masses.

(4) It will be interesting to carry out tests on a group of cells. A cell or group of cells with geometrically different orifices should be tested for their mass-matching capability.

(5) The speed trap is apparently not capable of measuring speed accurately. It is recommended that a more satisfactory speed trap be developed.
LIST OF REFERENCES


APPENDIX
APPENDIX A

SPECIFICATIONS OF THE INSTRUMENTS USED

This part of the Appendix contains the specifications of the transducers used in this test and the instruments used in recording the data.

(1) Accelerometer

Range 0 to 100g (in only one direction)

Power Supply Requirements

Voltage 28 V d-c.

Current 20 ma

Output Voltage 0 to 5 V

Resolution Continuous

Operating Temperature Range 0° F to 160° F

Natural Frequency 20 cps to 580 cps (depending on range)

Type A2-1106

Make Wiancko Engineering, Pasadena, California

(2) Piezoelectric Pressure Transducer

Range 0 to 2000 psi (+ Range)

Make Kistler
(3) Charge Amplifier

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(4) Electronic Counter (and Time Interval Unit)

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<td>Sensitivity</td>
<td>1 mv/div to 10v/div</td>
</tr>
<tr>
<td>Make</td>
<td>Clevite Corporation,</td>
</tr>
<tr>
<td></td>
<td>Cleveland, Ohio</td>
</tr>
</tbody>
</table>
(7) Constant D.C. Supply

Ranges for Current 0 to 30 V
Voltage 0 to 1 Amp
APPENDIX B

The section includes the data sheets obtained during actual tests.

[Graphs showing pressure and acceleration data at different velocities.]
APPENDIX C

The drawings of various members of the equipment are included in the pocket located inside the back cover of the master copy of the thesis on file with The Mechanical Engineering Department, Brigham Young University.
DESIGN AND OPERATION OF EQUIPMENT FOR
IMPACT TEST OF A HYDRAULIC CUSHION

An Abstract of a Thesis
Presented to the
Department of Mechanical Engineering Science
Brigham Young University

In Partial Fulfillment
of the Requirements for the Degree
Master of Science

by
Harshadbhai R. Patel
August 1968
ABSTRACT

The work reported in this thesis included design and operation of the equipment for impact test of an hydraulic cushion.

Instrumentation was provided to measure the pressure and the acceleration as functions of time. The main goal of this thesis, the design and the safe operation, was achieved. The dependance of pressure and the acceleration upon the impacting velocity was roughly as expected.

APPROVED: