DESIGN AND SELECTION CRITERIA
FOR
THE HYDRAULIC CRANE BUFFER

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In a modern steel plant, maximum productivity requires maximum equipment capacity and high operational speeds. In the case of the steel mill overhead traveling crane, higher capacity and higher speed has greatly increased the potential for collision damage.

The hydraulic crane buffer has proven itself to be the most reliable and least costly method of protecting crane, operator, and plant from the hazards of crane collisions, even under 100% full speed impact conditions.

This paper examines the hydraulic buffer from a design and operational viewpoint as compared to other types of crane bumpers. Quantitative data includes all formulas and calculations necessary to select a crane buffer, along with qualitative guidelines for specific types of applications.

INTRODUCTION

Almost all electric traveling cranes built in this country utilize some sort of bumper for the purpose of eliminating or minimizing damage. Protection of this type is required for three major types of collisions:

1. Crane to crane impacts

2. Crane to building impacts

3. Trolley to bridge end stop impacts

The type of bumper system used will determine the degree of collision protection available, which may range from minimal protection up to zero damage protection under full load, full speed collisions with power on.

It is the choice of the end user of the crane as to what level of protection is desired in a crane bumper, and this choice is determined by the following:

1. The duty cycle of the crane

2. The accident history (if any) of the cranes in a given area

3. Operator habits in a given area

4. Requirements and recommendations of associations and regulatory bodies such as the AISE and OSHA

In order to have an understanding of the crane bumper, one must first define what it is. A good working definition is as follows:

"A crane bumper (or buffer) is a device installed for the purpose of storing or absorbing the energy of a moving crane, thereby protecting the crane, the building it may operate in, and personnel in the immediate area from damage caused by collision."

The words bumper and buffer are used synonymously in the above definition, simply because most present safety standards use both words in this manner. The word bumper is usually used to describe a device which **stores** the energy of impact (like a spring) and gives this energy back to the system. A buffer is generally considered to be a device which **absorbs** the energy of impact and permanently removes this energy from the colliding objects. For the balance of this paper, the word bumper and buffer will be used in this manner. Therefore, the use of coil springs or similar means as a collision protection system on a crane will be defined as a spring bumper, whereas the use of a hydraulic device for the same purpose will be defined as a hydraulic buffer.

For comparative discussion purposes, a single typical crane will be used for studying various types of crane buffers and bumpers. The following is the standard crane:

1. Bridge weight = 100,000 lb. (does not include trolley)

2. Bridge speed = 300 F.P.M.

3. Trolley weight = 20,000 lb.

4. Trolley speed = 200 F.P.M.

5. Maximum carried load = 20 tons, carried on cables

6. Bridge span = 80 ft.

7. Minimum vertical distance from the center of gravity of the lifted load to the trolley hoist mechanism = 8 ft.

For comparison purposes, we will evaluate an impact of this crane into the end of the building it operates in at 50% speed.

COMPARATIVE PERFORMANCE CURVES FOR CRANE PROTECTION SYSTEMS

In order to design a crane bumper (buffer) to meet the requirements of the various regulatory bodies, one must be concerned with 2 basic criteria:

    a.   The energy capacity of the bumper (buffer)

    b.   The deceleration rate permitted on the crane

Energy capacity basically defines how large a bumper (buffer) is, and the deceleration rate is a measurement of how quickly the crane is to be stopped at impact.

The energy a moving object possesses at any time is a function of its weight and speed. The formula which is used for determining energy is:

$$E = \tfrac{1}{2} MV^2$$

Where: E = Energy

           M = Mass of the moving body

           V = Velocity of the moving body

By converting mass to units of weight, we can rewrite this formula in an easily understood way:

$$E = .1865\ W\ V^2$$

Where: E = Energy, units of in-lb.

           W = Weight, units of lbs.

           V = Velocity, units of ft/sec.

For the sample crane, we can calculate its energy (not including lifted load) using this formula:

    W = 100,000 lb. bridge + 20,000 trolley = 120,000 lb.

    V = 300 F.P.M. bridge speed = 5 ft/sec. @ 100% speed

    V = 2½ ft/sec. @ 50% speed

Therefore, for the sample crane at 50% speed:

$$
\begin{aligned}
E &= .1865\ (120{,}000)\ (2.5^2) \\
&= .1865\ (120{,}000)\ (6.25) \\
&= 139{,}875 \text{ in-lb.}
\end{aligned}
$$

Note that because the energy varies with the square of the impact speed that the energy of a crane at 100% speed is four times that of 50% speed.

The deceleration rate imparted to a moving object is a measure of the relative magnitude of the force being used to decelerate it. For example, if a 100 lb. weight was decelerated by a 1,000 lb. force, the deceleration rate would be much higher than if the same 1,000 lb. stopping force was applied against a weight of 5,000 lb. A simple way of expressing deceleration is in units of "G," one G being the amount of acceleration imparted to an object by the gravitational attraction of the earth. Since the gravitational attraction on a body on earth is equal to its weight, an acceleration or deceleration rate is determined by:

Acceleration or deceleration in G's $= \dfrac{F}{W}$

Where $\quad$ F $\quad =\quad$ the applied force (lb.)
$\qquad\quad$ W $\quad =\quad$ the object's weight (lb.)

Hence, if we applied a 1,000 lb. stopping force to a 100 lb. weight, the deceleration would be:

Deceleration $\quad = \quad \dfrac{1000}{100} = 10 \text{ G}$

A second way of expressing deceleration is in units of ft/sec^2. To convert units of G's to units of ft/sec^2, multiply the value in G's by 32. ft/sec^2, the acceleration of gravity on earth. Hence:

10 G deceleration = 320 ft/sec^2
.2 G deceleration = 6.4 ft/sec^2

The codes which often are used to determine the size of a crane buffer (bumper) specify protection at various percentages of the crane's full load speed, with a corresponding deceleration rate. The two major codes used to specify a bumper or buffer are those of OSHA and AISE 1969. At the time of writing this paper, certain types of cranes must have bumpers or buffers complying with OSHA requirements, by law. The AISE 1969 code for bumpers and buffers is a recommended specification but nevertheless many crane users require that a crane be in compliance with AISE 1969 regulations. The reader must realize that neither of the codes referenced cover all types of crane impacts, and for special severe service usage, capacities as high as 300% that of the AISE code may be required to yield satisfactory performance. Appendix I lists the basic energy and deceleration requirements of both OSHA and AISE 1969 standards.

Figure 1 plots allowable peak deceleration rate vs. speed for both OSHA and AISE 1969. Since OSHA specifies average decelerations, these values have been plotted assuming that the allowable peak is twice the average value. For higher velocity impacts under both specifications, the deceleration is assumed to increase in direct proportion to the energy of the crane (increases with the square of the crane velocity). Figure 1 reveals that the OSHA regulations allow higher deceleration rates than AISE 1969, meaning that a crane complying with AISE 1969 will have a "softer" stop than that permitted by OSHA. However, one must be quite careful not to get "carried away" with specifying the softest stopping bumper for the sole purpose of providing operator comfort. For example, typical deceleration rates for various occurrences are:

1. Normal city driving in an automobile = .2 G (6.4 ft/sec^2)

2. Maximum emergency braking in an automobile = .8 G (25.6 ft/sec^2)

3. 5 mph crashes in 1973 and later cars = 4 G (128 ft/sec^2)

4. A "knock-out punch" = 6 G (192 ft/sec^2)

5. The minimum deceleration rate necessary to cause bone fracture in an average person = 35 G (1120. ft/sec^2) (damage to knee-thigh-hip complex with load applied at knees, ref. S.A.E. information report J885A).

6. The deceleration rate necessary to cause death in an average person = 80 G (2560 ft/sec^2).

Remembering that since collisions involving cranes at more than 50% full load speeds are considered as an emergency or accident occurrence, the regulatory restrictions on crane deceleration appear quite conservative. When compared to the normal "fender bender" car accident at 4 G deceleration, the AISE 1969 100% speed impact deceleration of 2 G appears to be quite a soft stop by comparison.

One would normally expect that the deceleration rate of a buffer (bumper) is primarily a function of its stroke and this is indeed true among similar products of a given manufacturer. Figure 2 shows force-displacement output curves of 3 different types of bumper systems with equal strokes. The 3 output curves have been superimposed over one another for comparison. The energy absorbed or stored by each bumper is equal to the area under the appropriate curve.

Curve 1 is that which normally occurs with a rubber bumper. Note the parabolic shape of the output curve, and how the load out is only slightly lower than the load in. This means that this type of bumper will impart a significant rebound velocity to the crane equal to approximately 80% of the original impact speed. Note also that to match the area under the curves of the other bumpers a very high output force is required, yielding a high deceleration rate.

Curve 2 is that which occurs with a coil spring bumper. The spring force increases in proportion to the stroke with this type of design. The load out is equal to the load in, yielding a 100% rebound rate. To obtain equal capacity, the coil spring requires less force than the rubber spring, meaning that it will decelerate the crane at a lower rate than the rubber spring.

Curve 3 is the output curve of a well-designed hydraulic buffer. Note the near-constant force output yielding the lowest deceleration, and hence the softest stop, of any of the curves shown. In addition, the output force of the hydraulic buffer drops to near zero after impact, yielding no bounce back of the crane.

The quantities of relative output force and rebound rate are technically defined as the bumper efficiency and coefficient of restitution, respectively. Efficiency is a measure of relative output force. A unit with 100% efficiency has a perfectly "square" output curve with constant force over the entire stroke of the unit. A square output curve is shown in Figure 2 with a dashed line.

$$\text{Efficiency} \; = \; \frac{\text{Square wave output force}}{\text{Actual output force}} \text{ x } 100\%$$

$$\text{Where square wave output force} \; = \; \frac{\text{Energy input (in - lb)}}{\text{Stroke (in)}}$$

Therefore, using Figure 2 values,

Square wave output force = Y

$$\text{Efficiency, curve 1} = \frac{Y}{3Y} \text{ x } 100\% = 33\%$$

$$\text{Efficiency, curve 2} = \frac{Y}{2Y} \text{ x } 100\% = 50\%$$

$$\text{Efficiency, curve 3} = \frac{Y}{1.1Y} \text{ x } 100\% = 90\%$$

The coefficient of restitution is defined as the ratio of rebound velocity to impact velocity. Therefore, if a spring bumper is impacted at 150 FPM, and the crane bounces back at 100 FPM, the coefficient of restitution is $\frac{100}{150} = .66$

In general, coil and rubber spring bumpers have a coefficient of restitution of between .75 and 1.0, whereas hydraulic buffers have a coefficient of less than .1, yielding a non-rebound impact when compared to springs.

VARIOUS TYPES OF CRANE BUMPERS

Steel Stops - The oldest form of crane bumper is the plain steel stop. When impacted, the steel stop will hopefully buckle and bend, thereby dissipating the energy of the crane. Steel stops are a rather "rigid" way of stopping a crane, since most steel structure deflects only slightly under an applied force.

In general, a plain steel stop neither acts like a bumper or a buffer, since it is usually so rigid that when it is impacted, the crane energy must be dissipated by crushing the crane, the building it runs in, or both. The output curve efficiency of a steel stop bumper is 10-15%; coefficient of restitution is .8-1.0 for most designs.

Wheel Chocks - The wheel chock is a variation of the steel stop in which the energy of the crane is dissipated by lifting the crane up and off of the crane rails as the crane wheels run up onto the tapered chock. Wheel chocks are quite satisfactory for very low velocity "bumping" and can theoretically be designed to work at greater speeds.

The operation of a wheel chock is quite simple and analogous to that of a spring bumper. As the crane impacts the wheel chock at speed, the crane wheels will "ride up" the slope of the chock and will tend to lift the crane off the rail. This effectively converts the kinetic energy of the crane to potential energy, which is entirely given back to the crane as a rebound. Therefore, a crane running into a wheel chock will be rebounded off the chock at nearly the same speed it impacted at. This, of course, pre-supposes that the crane is not going fast enough to over-run the wheel chock. The output curve efficiency of a wheel chock is 15-30%; coefficient of restitution is 1.0.

Wood Stops - Another variation of the rigid steel stop is simply to install a piece of suitable timber between the crane and the object it is going to impact. Many of these bumpers offer the feature of easily replaceable wood blocks, usually by inserting the timber into a cast iron socket. Although one could attempt to classify an impact into a piece of wood as "softer" than that obtained by running into a solid steel stop, the difference is marginal. As with steel, only a small amount of the crane's energy can be stored by the wood before crane, building, or operator damage occurs. The output curve efficiency of a wooden stop is 10-30%; coefficient of restitution averages .40-.70.

Spring Bumper Systems -

    a. Steel Spring Bumpers - Steel springs were the earliest type of bumper resilient enough to be able to control the energy of a moving crane without damage. This type of bumper is still used on many new cranes today, but if the crane is over 5 tons capacity, it is very difficult to meet OSHA or AISE requirements without excessive costs.

        Various types of steel springs have been used in crane bumpers including coil springs, Belleville springs, Volute springs, and ring springs. With the exception of the coil spring, most spring bumper designs require that the spring element be properly and frequently lubricated to prevent jamming. Because a spring bumper stores energy, a jammed bumper of this type can be an exceedingly dangerous safety hazard.

        Coil spring bumpers are relatively free from jamming if properly designed, and are indeed the most popular of this bumper style. The most common coil spring used is identical to that used in the truck suspension of railroad rolling stock. On our sample crane with its kinetic energy at 50% speed of 139,875 in-lb., a bumper using railroad type coil spring packs can readily be designed. A typical railroad spring pack has an energy capacity of 10,000 in-lb. This means that to protect both sides of the reference crane bridge to 50% speed requires 32 of these spring packs, 8 on each corner of the crane. Compared to the hydraulic buffer, designing a coil spring bumper that actually meets the mandatory code requirements (OSHA) yields a cumbersome, heavy and expensive package. Since the coil spring only stores the crane's energy, the crane will

rebound off the bumper at a speed equal to its initial impact speed. Output curve efficiency for steel springs is 45-60%; coefficient of restitution is .90-1.0.

b. Rubber Spring Bumpers - The rubber spring offers performance equal to the coil spring with the advantage of being easier to package. A second advantage is that the rubber spring does dissipate a small amount of the crane impact energy as heat, due to friction within the rubber particles under stretching and compression. This means that the rebound from a rubber spring is not quite as violent as that occurring from a coil spring, assuming that either bumper is designed to decelerate the crane equally. Typically, a rubber spring which is impacted at 100% speed will rebound the crane at about 85-90% speed. Overall package size is similar to that of the coil spring bumper, being heavy and cumbersome compared to a hydraulic buffer of equal capacity. Output curve efficiency for rubber springs is 30-40%; coefficient of restitution is .70-1.0.

c. Compressed Elastomer Spring Bumpers - This spring bumper contains a silicone rubber elastomer in a high strength steel pressure vessel and utilizes seals and a piston to compress the rubber to explosive-like pressures (up to 60,000 lbs. per square inch). The rubber material used is of the silicone family and is not completely cross-linked in its molecular structure, so that it will flow in a fashion somewhat like a fluid. Since normal rubber spring bumper material will permanently deform at stress levels of only 5,000 psi or so, the ability of the compressible elastomer bumper to compress the rubber to ultra-high pressures of 60,000 psi yields a very compact design. This type of bumper offers somewhat lower rebound speeds than other types of rubber springs because the rubber can be forced around the piston of the device at impact, thereby absorbing some of the impact energy. Rebound velocity is in the order of 80% of the initial impact speed. Compressed elastomer bumpers have seen only limited usage on overhead cranes due to the requirement that the rubber must be pre-loaded with a high static pressure for the bumper to operate properly. This static pressure can be drastically increased as the rubber expands due to the normal high temperatures often found in steel mills, causing a possible safety hazard. If a rubber spring bumper is to be used on a crane, the non-compressible rubber pad type should be considered for safety and cost reasons, since in addition to safety problems, the compressed elastomer bumper is the most expensive bumper (buffer) design built. Output curve efficiency for the compressed elastomer bumper is 30-40%; coefficient of restitution is .75-90.

THE MODERN HYDRAULIC BUFFER

The present style of hydraulic buffer became available about 1955, with widespread usage on cranes and other steel mill equipment starting in 1965. Prior to 1965, most hydraulic buffers were of a "dashpot" type design, using only a single fixed orifice to absorb energy. Because the efficiency of a fixed orifice is only about 30%, very long strokes with complex multi-spring or latch-type reset were required with these early designs. Unlike the early hydraulic dashpots, a modern hydraulic buffer contains:

1. A reset system, self-contained and internal to the buffer –
    An internal coil spring is normally used.

2. A variable metering orifice –
   The self-contained reset is used to eliminate troublesome external reset latches and pins used on early dashpot designs. Reset forces are usually kept down to about 1% of maximum output force. Therefore, a hydraulic buffer with 50,000 lb. rated output force would have a maximum reset force of 500 lb. This means that unlike any type of bumper system, most hydraulic buffers can be easily compressed by the crane at low velocity, effectively using the buffer stroke as additional crane approach. Since the reset forces are so low, the buffer will not bounce the crane back after impact.

Unfortunately, this advantage of the hydraulic buffer can also be a serious disadvantage if the crane is in the hands of a relatively unskilled operator, or the duty cycle of the crane demands that the load be made to swing by impacting the buffer with power on.

It is the purpose of the variable orifice metering system of the buffer to regulate the ability of the buffer to absorb the crane energy efficiently at all speeds and conditions of impact. To properly understand the features of the various types of metering systems available, they must be discussed at length.

TYPES OF HYDRAULIC METERING SYSTEMS

Three major types of metering systems have been used in hydraulic buffers for steel mill service. These are:

   a. Single orifice metering

   b. Variable orifice by mechanical metering

   c. Variable orifice by fluidic metering

The previous discussion of the early hydraulic dashpot type of buffer covered the primary disadvantage of single orifice metering, namely that of excessive stroke required due to its inherently low output curve efficiency. Since hydraulic buffers of this design have not been produced for the past 15 years, no further discussion will be given on this design.

The primary design criteria for a bumper (buffer) is to absorb the impact energy of the crane with a deceleration limited by various codes. As has been shown, a design with a high output curve efficiency is capable of doing this with a much shorter stroke (and usually a lower cost) than a long stroke design of low efficiency. With a hydraulic buffer, the metering of the fluid is governed by the various equations of the science of fluid mechanics. In the case of flow through an orifice, these equations state that the pressure drop across the orifice varies with the square of the speed of the fluid flowing through the orifice. This means that to obtain an efficient constant force output during an impact, the orifice must be drastically varied during the impact. For maximum efficiency, the orifice area must be greatest at the instant of impact, progressively dropping to zero at the end of stroke. The two methods used to accomplish this in the modern hydraulic buffers are by either a mechanical or fluidic type of metering system.

MECHANICAL METERING SYSTEMS

The mechanical metering systems use a metering tube or metering pin to mechanically vary the orifice area of the buffer throughout its stroke. Figure 3 shows a typical buffer with metering tube construction. A list of the internal parts of Figure 3:

1. Cylinder

2. Piston rod

3. Piston head and high pressure seal

4. Orifice hole (one of 5)

5. End cap and seal assembly (low pressure)

6. Metering tube

7. Metering tube high pressure seal

8. Piston rod displacement accumulator

9. Return spring

At impact, fluid will be forced through all of the orifice passages, yielding a fixed orifice area until the piston head passes the first orifice hole. As the impact continues, the remaining holes are passed and the process continues in like manner until the shock absorber stops the impacting weight. Piston rod displacement is compensated for by an accumulator, often of cellular rubber construction.

The metering tube buffer can be designed to yield a high output efficiency if a large number of orifice hole positions are used. The more orifice holes, the more uniform and efficient the output curve will be. The number of holes actually used relative to buffer stroke varies drastically between the various manufacturers. In 2 inch stroke units for example, some manufacturers use as few as 2 orifice positions, while others use as many as 6. To obtain maximum output efficiency, the spacing of the holes is important, since the shock must stroke past each successive orifice hole at equal intervals of time. Because of this, the holes must be parabolically spaced along the metering tube, with the holes widely spaced at the beginning of the buffer stroke and closely spaced at the end. If a linear hole placement (equal spacing between holes) is used to save costs, a loss in efficiency will result. A similar problem occurs if too few holes are used.

Depending on hole location and number of hole positions used, the efficiency of metering tube crane buffers available today ranges from 40% to 80%. Efficiencies at the low end of the range result from the attempted use of low quality industrial equipment shock absorbers (sometimes called industrial decelerators) in steel mill service. It is important to remember that when a crane impact occurs, both man and machine must be protected. The industrial shock absorber (or decelerator) is an inexpensive item designed to protect the machine only, so a low output efficiency can be tolerated.

A second style of mechanical metering uses a metering pin rather than a tube, but is a more complex design, so costly as not to be offered in any form other than an "industrial decelerator" with very low efficiency.

FLUIDIC METERING SYSTEMS*

Fluidic metering uses a specially shaped series of orifice passages to yield by hydraulic flow the effect of a mechanically varied orifice. In order to operate efficiently, a compressible working fluid must be used, such as the family of silicone base oils. The molecular structure of these oils is such that they can physically be compressed by a substantial amount when forced through the orifices of the buffer. A typical fluidic type buffer is shown in Figure 4. A description of the internal parts of Figure 4:

1. Cylinder
2. Piston rod
3. Piston head
4. Main orifice
5. Feeder orifice
6. Fluidic collector groove
7. End cap and seal assembly (low pressure)
8. Piston rod displacement accumulator
9. Return spring
10. Reset valve
11. Guide spool

When impacted, a buffer of the fluidic type will initially attempt to behave like a single orifice design, flowing all fluid across the main orifice passage. Within a few micro-seconds after impact the pressure differential across the length of the main orifice has a gradient, such that the pressure in front of the piston is greater than that existing in the fluid collector groove. This means that flow through the feeder orifices will be entrained into the main orifice flow, yielding a combined flow equal to the main orifice plus the feeder orifice. Because the working fluid is compressible, the flows can indeed be combined, yielding a large effective orifice area at impact. By proper design, the flow through the feeder orifices will be progressively decreased as the buffer absorbs energy, yielding the effect of a parabolically reduced orifice area, with a square output curve. Response is similar to that of a metering tube or metering pin metering system, although no **mechanical** variation of the orifice occurs.

The fluidic metering system consistently yields the highest efficiency of any known fluid metering system. Output efficiency will normally range between 85% and 95%, with a coefficient of restitution of less than .10 as typical.

*Patents issued, pending, applied for, Tayco Developments, Inc.

PACKAGING OF HYDRAULIC BUFFERS

Using the buffer designs shown in Figures 3 and 4, a striker cap could be added to the piston of the buffer and it would be ready for crane service. However, the exposed highly-finished piston rod would be exposed to the mill environment, which is not desirable for maximum life. For this reason, hydraulic buffers in mill service often utilize bellows over the piston rod for environmental protection . A second alternative is to turn the buffer backwards, and install it in a guide sleeve. This design, shown in Figure 5, not only protects the rod, but also is much better able to resist offset loadings than exposed rod designs.

For our sample crane, a typical hydraulic buffer suitable to absorb the crane energy at 50% speed is a Taylor Device's Fluidic Buffer, Model 3 x 3, rated at 81,000 in-lb full capacity. This buffer is 3" diameter, and has a 3" stroke, and is 13½ O.A.L. One of these buffers would be required at each corner of the crane. For protection at 100% speed, a Taylor Device's Fluid Buffer Model 5 x 4 could be used, rated at 306,000 in-lb.

**PERFORMANCE CHARACTERISTICS OF HYDRAULIC BUFFERS
UNDER POWER-ON IMPACTS**

As mentioned previously, hydraulic buffers utilize reset springs with relatively low force to enable use of the buffer stroke as crane end approach. In addition, the hydraulic buffer differs from a spring- type buffer with regard to how it operates under low velocity impacts. A spring type bumper varies its output force in proportion to how far it strokes. If an energy input is introduced which is over the capacity of the bumper, it will bottom the bumper, leaving the crane and/or building structure to absorb the remaining energy. A hydraulic buffer will vary its output force with respect to the square of the impact velocity, thereby always maintaining a constant stroke at any speed, since the impact energy of the crane also varies directly with the square of the velocity.

This means that for any two bumper systems, with equal decelerations at a given speed, the hydraulic buffer will offer the softest stop at all lower velocities.

As mentioned previously, this feature of the hydraulic buffer is also its biggest drawback, if one considers that cranes are subject to more than just power-off, no load impacts.

If a hydraulic buffer is sized for power-off impacts only, it will use nearly all of its stroke to absorb the kinetic energy of the crane at all speeds, leaving no capacity remaining for counterbalancing drive forces, or counterbalancing the equivalent driving force of a swinging load. If the hydraulic buffer is oversized for power-on impacts, it will not stroke completely under power-off conditions, but will have additional capacity to withstand power-on conditions. The lack of inclusion of driving inputs in sizing of a buffer is the major cause of premature buffer failure on cranes. It is apparent that within the steel industry, a typical duty cycle for a hydraulic buffer often includes repeated 10-20% speed impacts with power-on, for the purpose of putting the crane bridge or trolley in the end approach condition. If a crane bridge or trolley is expected to undergo this sort of duty, then the hydraulic buffers should be sized for power-on impact conditions.

Taylor Device's experience with both metering tube and fluidic buffers has shown that more than 50% of the cranes equipped with hydraulic buffers are subjected to this sort of duty cycle. This has prompted the firm to offer two distinctly different types of orificing in the fluidic type buffers:

1. Full drive down metering: Allows the fluidic design to behave exactly like a metering tube design, being easily compressed under drive inputs.

2. Non-drive down metering: A valve is installed across the fluidic orifices to enable the buffer to offer substantial resistance (20% of maximum output force) for counterbalancing drive at low speeds.

It is recommended that the use of the hydraulic buffers on the following types of mill cranes include sizing for power-on impacts, or include the use of a Fluidic Buffer without full drive down metering:

1. Bridge and trolley buffers on soaking pit and stripper cranes

2. Trolley buffers on all charging and scrap handling cranes

3. Trolley buffers on all magnet cranes

4. Bridge and trolley buffers on all pendant and radio controlled cranes

5. All crane applications where the buffer is expected to be impacted more often than "emergency" conditions

THE EFFECT OF SWINGING LOADS ON ENERGY CAPACITY

Mentioned briefly in the previous section was the "equivalent driving force of a swinging load." Most codes state that the lifted load is not to be considered as crane weight for calculating the energy to be absorbed by a buffer, but an effective drive for a swinging load can indeed be calculated. The response of the lifted load will lag the response of the crane by a short period of time, but if the duty cycle of the crane involves repeated inputs of this type, the buffer (bumper) should be sized to include swinging load effects. Calculating the effective input of a swinging load is somewhat complicated, and Appendix II shows the calculations involved.

THE SELECTION OF HYDRAULIC BUFFER BORE AND STROKE

Unlike other types of crane protection systems, hydraulic buffers are available in a seemingly endless list of various bores and strokes. For satisfactory crane service one should not consider buffers designed for only industrial service, as they are not rugged enough for mill applications. After eliminating industrial designs, one finds that even among the mill service buffers there are still a large number of sizes available.

The most common mistake made by purchasers of a hydraulic buffer is to compare designs from different manufacturers by bore and stroke, as would be done with hydraulic cylinders. Unfortunately, with buffers, this comparison does not apply since most manufacturers use varying operating stress levels, and various construction materials. Materials range from mild steel to heat- treated aircraft quality steels of types 4140, 4340, etc. When comparing buffers from different manufacturers, the purchaser should be concerned with the following:

1. Is the buffer certified by the manufacturer to the protection level desired?

2. Does the manufacturer's catalog rated energy capacity for the buffer agree with that calculated for the crane?

3. What back-up structure design loading is specified by the manufacturer for the particular buffer involved?

Because of differences in efficiency among the various hydraulic buffers, it is possible for 2 buffers to have equal back-up structure design loadings, but stroke differences of as much as 2:1. Because of efficiency, it is quite possible for the 10" stroke buffer of one manufacturer to have the same capacity and deceleration rate as the 5" stroke buffer of another.

Even among the products of a single manufacturer, a large number of sizes may be available in a given capacity. For example, in mill buffers of 250,000 in-lb. capacity, the following sizes are available from Taylor Devices.

Model	Diameter (in.)	Stroke (in.)	Output Force (lb.)	Capacity (in-lb.)
3 x 8	3	8	35,000	252,000
4 x 6	4	6	50,000	270,000
5 x 4	5	4	85,000	306,000
6 x 2	6	2	150,000	270,000

Any of the above sizes are perfectly adequate from a capacity basis, but each size has its advantages and disadvantages:

1. The 3 x 8 is the least expensive, the 6 x 2 is the most expensive.

2. The 3 x 8 has the lowest reaction force, the 6 x 2 puts out the highest force.

3. The 6 x 2 is the shortest, and hence takes less effort to package.

4. The 6 x 2 can withstand the most abuse, and is virtually impervious to scrubbing and sideload damage.

5. The 6 x 2 will give the longest cyclic life, since its short stroke yields the least number of square inches of piston rod sealing area passing through its seals each cycle.

In general, for mill service, the following rules can be used to select bore and stroke ratios:

1. For trolley buffers, the ratio of the buffer stroke to piston rod diameter should not be greater than 1.5 to 1.

2. For bridge buffers, the ratio of the buffer stroke to piston rod diameter should not exceed 3 to 1.

3. The lower the stroke/diameter ratio, the more rugged the design is. This means that of the four sizes, the 3 x 8 would be acceptable for use on small cranes only, and the other sizes are suitable for either bridge or trolley usage on any crane. For a severe service application, the Model 6 x 2 would be the best choice, followed by the Model 5 x 4.

Most manufacturers rate their hydraulic buffers as able to accept repeated angular offsets of 8-10°. However, if a long, slender buffer is put into mill service, its usual failure mode from offset impacts is due to scrubbing loads rather than angular misalignment. If the buffer is stroking and the crane should rise or fall suddenly (as when a rail joint is encountered), the buffer will be subjected to a scrubbing load. Assuming a steel on steel coefficient of friction of .25, this scrub is equivalent to an angular misalignment of 15 degrees, substantially more than would ever be expected on a crane from angular misalignment.

For this reason, one should always try to keep the stroke of the buffer as short as the code (and the budget) allows.

COST OF THE HYDRAULIC CRANE BUFFER

The cost of hydraulic buffers on a crane varies with the size of the crane, the code requirements, and the duty cycle of the crane. In general, using 2001 prices, one can roughly expect the following prices to apply for crane buffers of various capacities:

1. A 10,000 in-lb. buffer = $325. each

2. A 100,000 in-lb. buffer = $630. each

3. A 1,000,000 in-lb. buffer = $1,875. each

4. A 10,000,000 in-lb. buffer = $13,000. each

For our sample 20 ton crane, protected to 50% speed on the bridge and trolley, the total cost for 4 pc. Model 3 x 3 bridge and 4 pc. Model 2 x 2 trolley buffers is roughly $5,200.

When comparing equal capacity designs and equal decelerations, hydraulic buffers of more than 20,000 in-lb. capacity are usually the most economical design available compared with any type of spring bumper.

Maintenance on a properly designed mill buffer is limited to occasional visual inspection to make sure that the buffer is resetting completely. Rebuilding is required only if the buffer shows any obvious damage. Cycle lives of between S0,000 and 2 million impacts can be expected between rebuilds, provided that the buffer has not been overloaded.

SUMMARY

For those readers interested in formal sizing calculations for a crane buffer, Appendix III lists and explains the formulas involved. Unlike spring bumpers which often must be completely designed for each job, the wide selection of hydraulic buffers available means that a size can be quickly established for a given application. Most manufacturers are able to size buffers over the telephone, with no quotation charges involved. Optional custom mounts are available for direct bolt-in retrofit of existing bumper or steel stop designs, if standard mountings will not adapt easily. For new cranes, most crane manufacturers offer hydraulic buffers in compliance with AISE 1969 as an option.

In its relatively short history, the hydraulic buffer has proven to be of significant value to the steel industry, offering protection never before available at reasonable cost. On cranes subject to frequent bumping, reduced maintenance costs can pay for the buffer in as little as three months of operation. This means that this product can offer significant reductions in maintenance, and increased productivity, at a reasonable cost.

The author welcomes any comments or questions concerning hydraulic buffers either within or beyond the scope of this paper. Literature available on the Taylor Device's Crane Buffer product line includes:

1. Crane Buffer Sizing Graphs for OSHA Code
2. Crane Buffer Sizing Graphs for AISE 1969 Code
3. Crane Buffer Data Packet (General)
4. W-Series Self-Adjusting Crane Buffer Packet
5. O-B Series Long Stroke Buffers Information Packet

Requests for information should be sent to the following address:

Taylor Devices, Inc.
90 Taylor Drive
N. Tonawanda, New York 14120-0748
Attn: Crane Buffer Sales Dept.

APPENDIX I
CRANE BUMPER STANDARDS

1. **OSHA Impact Standards:**

   All information listed here is based on OSHA standards dated June 27, 1974. These standards show the following protection requirements:

   a. Bridge bumpers (buffers)
      Protection at 20% of maximum full load crane speed with an average deceleration rate of 3 ft/sec^2.

   b. Bridge bumpers (buffers)
      Protection at 40% of maximum full load crane speed with no specified deceleration.

   c. Trolley bumpers (buffers)
      Protection at 33-1/3% of maximum full load crane speed with an average deceleration rate of 4.7 ft/sec^2.

   d. In addition OSHA states that for calculating bumper (buffer) capacity, swinging loads are not to be considered as crane weight. Trolley placement for impact is not specified, so one must assume that the trolley is fully offset to one end of the bridge for correct sizing.

   e. OSHA does not specify whether power is to be considered as on or off for sizing purposes. One normally assumes power off.

2. **AISE 1969 Impact Standards**

   All information listed here is based on AISE standard No. 6, dated May 1, 1969. These standards show the following protection requirements:

   a. Bridge and trolley bumpers (buffers)
      Protection at 50% of maximum full load crane speed with a maximum deceleration rate of 16 ft/sec^2.

   b. Bridges and trolleys equipped with hydraulic buffers shall have protection at 100% maximum full load crane speed with a deceleration rate increased correspondingly from that specified at 50% speed.

   c. In addition, the AISE specifies that trolleys are to be fully offset to the end approach position for bridge and trolley bumper (buffer) sizing, and power is to be considered off for all sizing.

APPENDIX II
THE EFFECT OF SWINGING LOADS

At whatever speed one is concerned with, a swinging crane load will possess a certain amount of kinetic energy of its own, a function of its weight and the velocity of the crane. At impact, the load will swing until all of its kinetic energy has been converted to potential energy by lifting the load through the **vertical** displacement of the swing. Expressing this as a formula yields:

Potential energy = kinetic energy

Swinging weight x vertical swing height = $.1865 \, W \, V^2$

Since the swinging weight and "W" in the kinetic energy formula are the same, the expression reduces to:

Vertical swing height (in.) = $.1865 \, V^2$ (ft^2/sec^2)

As the load swings to its maximum angular displacement, a right triangle with hypotenuse equal to cable length, and an adjacent side equal to the hypotenuse minus the vertical swing height will be formed. This means that the angle of swing can be determined by:

$$\text{Cosine (swing angle)} = \frac{\text{cable length} - \text{vertical swing}}{\text{cable length}}$$

The effective drive of the load is equal to the component of the lifted load acting in the horizontal direction, which is:

Effective drive = swinging load x sin (swing angle) ÷ cos (swing angle)

On the trolley of our sample crane, the energy of the empty trolley at full speed is:

$$KE = .1865 \, W \, V^2 = .1865 \, (20,000) \left(\frac{200}{60} \right)^2 = 41,444 \text{ in-lb.}$$

The vertical swing height = $.1865 \, V^2 = 2.07$ in.

(Note that the formula $KE = .1865 \, WV^2$ has conversion units "built in" to the constant .1865, so the end result of this calculation is values of "inches" using values of "ft/sec" for velocity).

For our sample crane, the shortest net effective cable length of the lifted load is 8 ft. (96 in.) Therefore, if we assume that at impact the cable length is 10 ft. (120 in.), allowing the load to swing:

$$\text{Cos (swing angle)} = \frac{120 - 2.07}{120} = .982$$

swing angle = 10 degrees, 50 minutes

Therefore, the effective drive of the 20 ton swinging load is:

Effective drive = 40,000 (.188)/(.982) = 7,658 lb.

Effective drive energy = force x distance = 7658 lb. x 2 in. = 15,316 in-lb.

Using two Taylor Device's Fluidic Buffers to absorb the trolley kinetic energy in each direction would require 2 pc. Model 2 x 2 rated 27,000 in-lb. each, with a maximum output force of 15,000 lb. Note that the effect of the swinging load is an equivalent driving force of 7658 in-lb/buffer, equal to 28% of the buffer capacity.

In general, studies have shown that lifted load can usually be compensated for on most cranes with typical impacts by selecting a buffer with capacity at least 20% greater than that required for the dead weight of the crane.

APPENDIX III
A GUIDE TO HYDRAULIC CRANE BUFFER SIZING

A buffer is a device which converts mechanical energy into thermal energy. The energy transformation occurs as the shock absorber's fluid medium is forced through orifices at high velocities.

Selecting a shock absorber is not difficult if you follow the formulae presented. To insure adequate sizing, all inputs to the buffer must be known or conservatively estimated.

A) UNITS AND ABBREVIATIONS: (Use only units shown below in formulae)

W = weight (lb.)

V = linear velocity at the shock absorber (ft/sec.)

F = output shock force at impact (lb.)

F_D = motor drive force (lb.)

S = shock absorber stroke (in.)

KE = kinetic energy (in-lb.)

B) SOLVING FOR KINETIC ENERGY

1) Horizontal motion
$$KE = .1865 \ WV^2 \ \text{(in-lbs.)}$$

2) Vertical motion
$$KE = W \ (H + S) \ \text{(in-lbs.)}$$

C) SOLVING FOR KINETIC ENERGY OF OVERHEAD CRANES

1) Because of the "sling-shot" effect of cable hung loads and overspeed possibilities, effective impact weights, W_E should be used.

a) Bridge Buffer W_E/Buffer = 1.3 (½ bridge weight + trolley weight) (lbs.)

OR

W_E/Buffer = ½ bridge weight + ½ trolley weight + ½ lifted load (lbs.)

Use whichever weight is **greater** for kinetic energy calculation

        b) Trolley Buffer W_E/Buffer = 1.3 (½ trolley weight) (lbs.)

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OR

</div>

W_E/Buffer = ½ trolley weight + ½ lifted load (lbs.)

Use whichever weight is **greater** for kinetic energy calculation

    2) Solve for kinetic energy per buffer

KE/Buffer = .1865 W_E V^2 (in-lbs.)

D)   SOLVING FOR DRIVE FORCE AT THE BUFFER

    1) A.C. Motors

$$F_D = 1375 \ \frac{\text{Motor Horsepower}}{V} \quad \text{(Assumes 2.5:1 stall factor)}$$

    2) D.C. Motors

$$F_D = 1925 \ \frac{\text{Motor Horsepower}}{V} \quad \text{(Assumes 3.5:1 stall factor)}$$

Note: Both 1 and 2 neglect gearing power losses and slippage power losses.

E)   SELECTING THE BUFFER IF INPUT IS PURE KINETIC ENERGY WITH NO MOTOR DRIVE

Select a shock absorber from catalog data with adequate energy capacity for your calculated input. For cyclic rates above 120/hour, use a 30% safety factor on energy capacity. For cyclic rates above 360/hour, consult factory on your application

F) DECELERATION RATE FOR OVERHEAD CRANES

    1) AISE 1969 code limits decelerations to ½ G at 50% speed, which effectively is 2.0 G at 100% speed for a Taylor Buffer.

    2) OSHA code limits bridge decelerations to .0932 G average at 20% speed, which effectively is .373 G average at 40% speed for a Taylor Buffer.

3) OSHA code limits trolley decelerations to .146 G average at 33⅓% speed.

4) Deceleration rate for your application is:

$$\text{Number of G's} = \frac{F/\text{Buffer}}{\text{Impact weight}/\text{buffer}}$$

5) Bridge weight per buffer for deceleration calculation, use ½ bridge weight + ½ trolley weight.

   For trolley weight per buffer when calculating deceleration, use ½ trolley weight.

6) If your deceleration is too high, try a longer stroke.

G) SELECTING THE SHOCK ABSORBER IF INPUT IN KINETIC ENERGY AND MOTOR DRIVE

1) Obtain kinetic energy of your input, and the motor drive force.

2) Select a trial shock absorber diameter.

3) Solve for stroke required using the equation KE

$$S = \frac{KE}{C(F - F_D)}$$

   Where C, the efficiency coefficient, varies between .4 and .9 for various manufacturers. As in section E, use a 30% safety factor on kinetic energy for cyclic rates above 120/hour, and consult the manufacturer for sizing of units with cyclic rates above 360/hour.
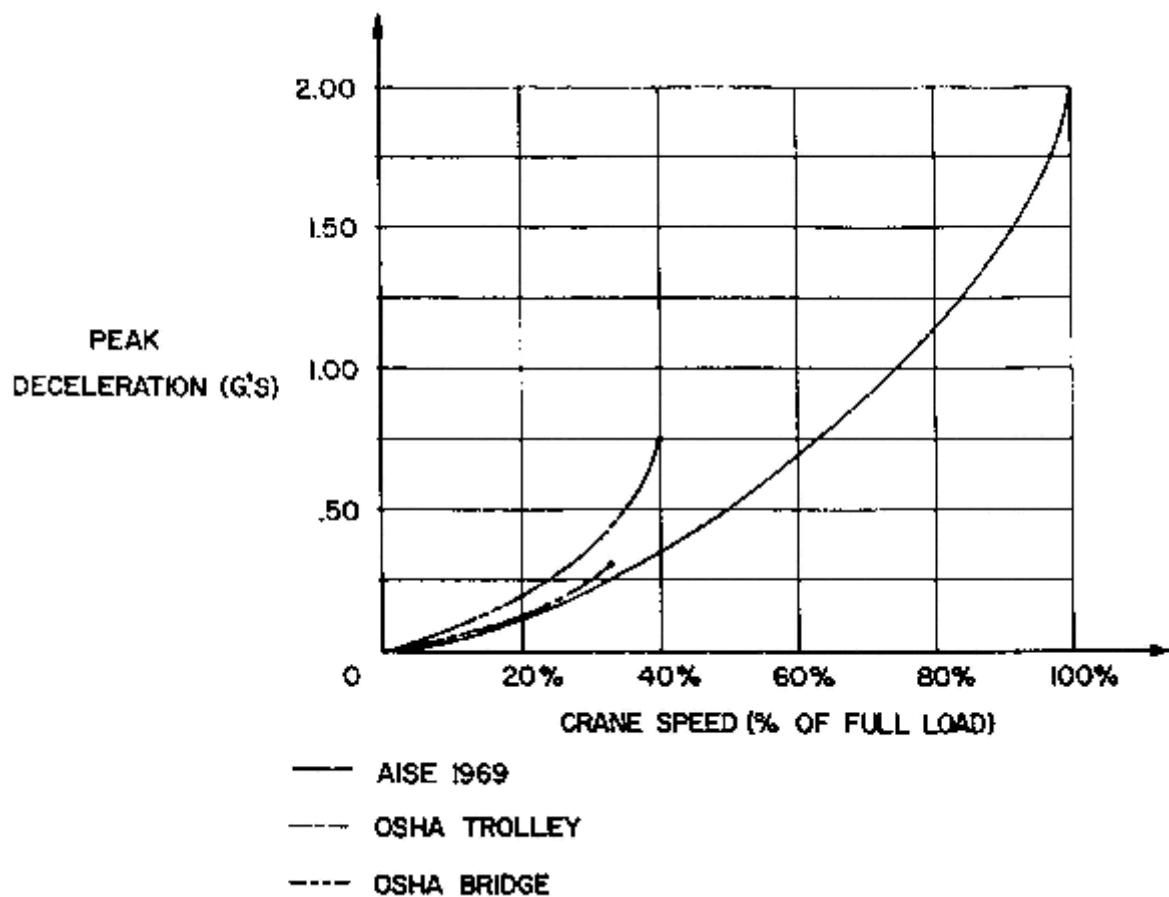
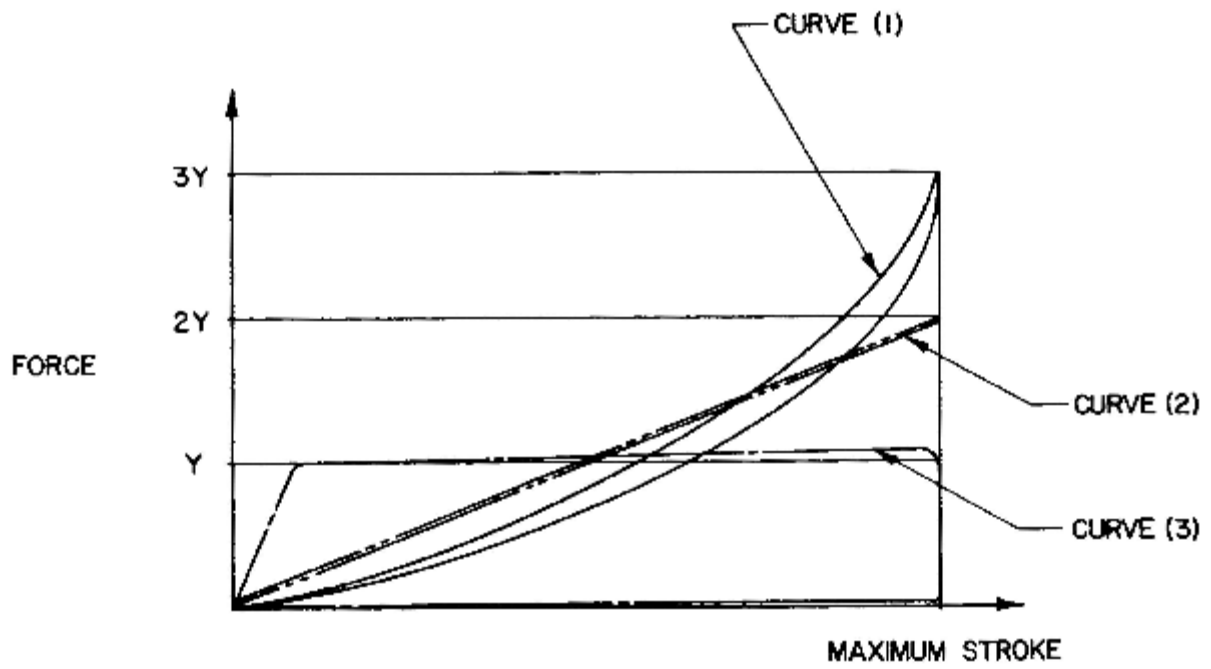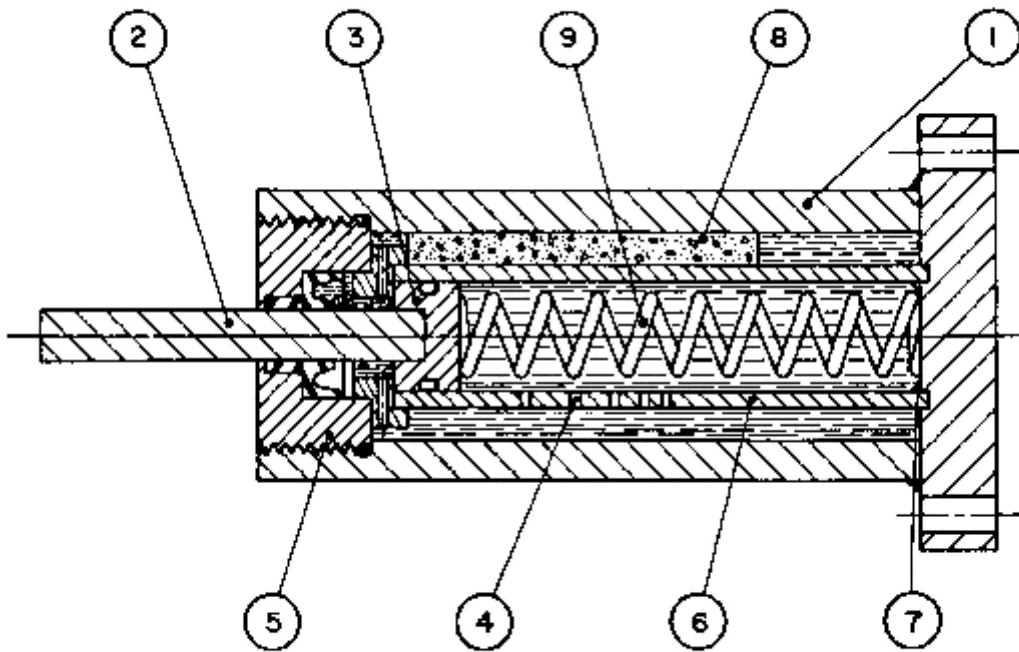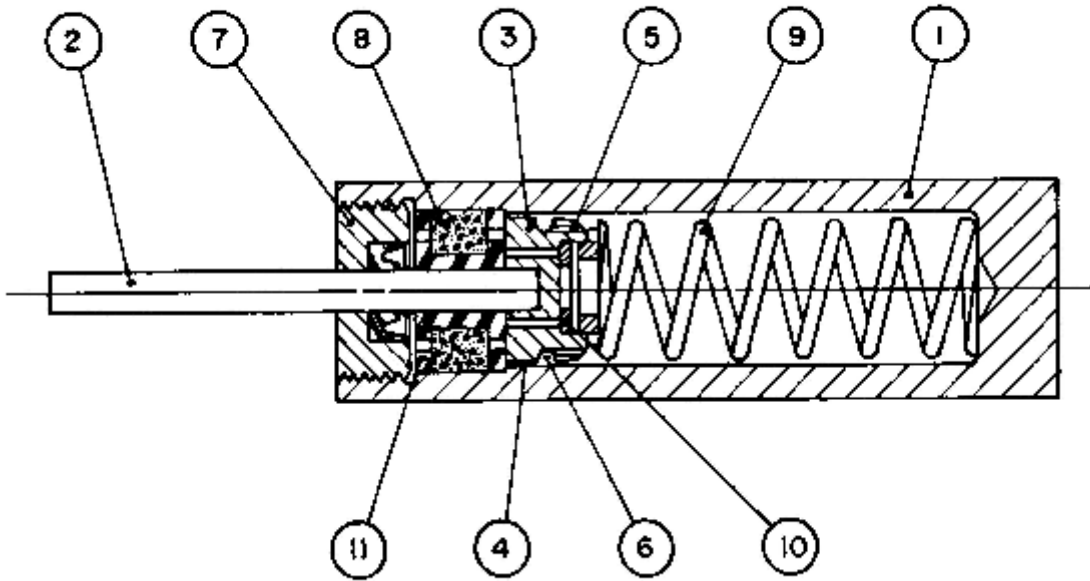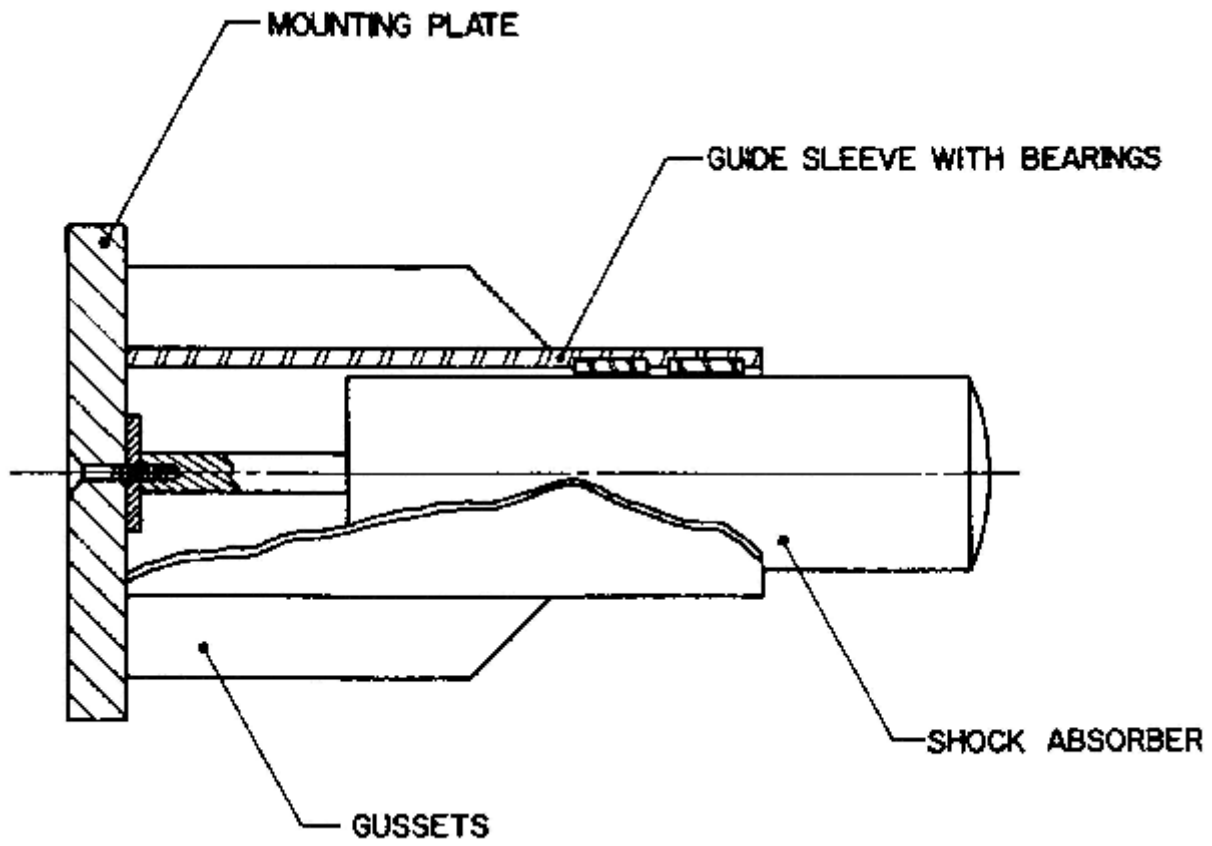4) Calculate deceleration in the same way as section F.

FIGURE 1

FIGURE 2

FIGURE 3

FIGURE 4

MOUNTING PLATE

GUIDE SLEEVE WITH BEARINGS

SHOCK ABSORBER

GUSSETS

FIGURE 5