Hydraulic Bumpers for the Protection of Buildings, Cranes and Operators From Impact Damage

by Paul G. Kit

Presented at the 1996 Fall Convention

Association of Iron and Steel Engineers

Subcommittee # 13: Design and Construction of Mill Buildings

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Hydraulic Bumpers have been used effectively on overhead and gantry cranes for more than forty years. Advances in hydraulic shock absorber technology have made hydraulic bumpers affordable as standard equipment for most new cranes and "retrofittable" to existing ones. New cranes are operating faster than ever before to meet the rigorous duty cycle demands of industry. To reduce capital costs, companies are building new mill buildings as light as possible. There has been insufficient attention given to crane bumpers. Several unprotected collisions and catastrophic accidents have resulted. Careful attention must now be given to ensure the proper specification and application of hydraulic bumpers, based on today's duty classes and typical operation of cranes.

Background and History:

The earliest cranes did not have bumpers as we are familiar with today. End of travel limits were typically steel to steel collisions between framed members. Because travel speeds of crane bridges and trolleys were relatively low, impact damage was seldom a problem. Additionally, operators avoided impacting the end of travel limits because of the impending "crash" and possible damage. Some cranes and buildings began to use oak timbers to offer a sacrificial cushion at end stops. This did not provide a "soft" stop; however, the impact damage occurred to the timbers and not the steel structure. Chocks that fit the curvature of the tread of the wheels were also commonly used. Again, no cushion from impact is provided, it merely served as a mechanical limit to travel. The wheel chock caused severe damage to wheels, wheel bearings and end trucks.

Just as many aspects of cranes were borrowed from the railroad industry, crane bumpers evolved similarly. Early railroad cars had varieties of coil springs, ring springs and elastomeric springs for impact management. Ultimately, many of these mechanical energy absorbers gave way to hydraulic cushions mounted in the draft gear of the rail cars. Cranes have followed a parallel progression toward standardization on hydraulic energy absorbers.

A Comparison of Current Devices:

There are several acceptable and effective methods for dissipating the impact energy from a crane impacting into its end stops or another crane. All of these devices differ substantially in physical characteristics, design, function and efficiency. The most significant comparison between energy absorber devices is Diagram Efficiency. Diagram efficiency refers to the ratio of area (energy) under the force vs. stroke curve from a particular bumper to the area (energy) under the theoretical force vs.

stroke curve of an ideal bumper. The theoretical area is defined by a rectangle at maximum force and maximum stroke available. A crane bumper dissipates the Kinetic Energy from an impact by applying a resistive force over a deflection or stroke. The kinetic energy is converted to heat energy and removed from the system. An ideal bumper is 100% efficient. A linear coil spring device with a triangular force vs. stroke curve is basically 50% efficient. Hysteretic phenomena such as friction can increase this energy absorption ratio. Note that the efficiency used for calculations is diagram efficiency and not energy absorption empirically measured. Structural flexure typically dissipates approximately 5% of the kinetic energy before the bumper begins to stroke. Typical diagram efficiencies for the most common devices used for crane bumpers are listed below for reference:

- Hydraulic Bumper 90%
- Fluid Elastomer Bumper 65%
- Ring Spring Bumper 60%
- Coil Spring Bumper 50%
- Solid Elastomer Bumper 45%
- Rubber Bumper 30%

Bumper End Force and Stroke are related to Kinetic Energy Capacity by this Efficiency.

Kinetic Energy = (End Force) x (Stroke) x (Efficiency)

Special care should be used when comparing different types of energy absorbers. Only steel coil springs and some ring springs can be analyzed in the simplified linear approach. Rubber and elastomenc bumpers have non-linear load-deflection relations as well as shape factor dependence. Hydraulic bumpers are non-linear and vary from manufacturer to manufacturer. It is the non-linearity that allows only the hydraulic bumper to change its resisting force as a function of the impacting velocity. Kinetic Energy is a function of velocity squared. Provided that the metering device of the hydraulic bumper is designed to provide bumper force as a function of velocity squared, the bumper force will always be of the correct magnitude to arrest the impacting mass in the most efficient manner. This velocity dependence is the key advantage for protecting a wide range of varying impacting weights and speeds.

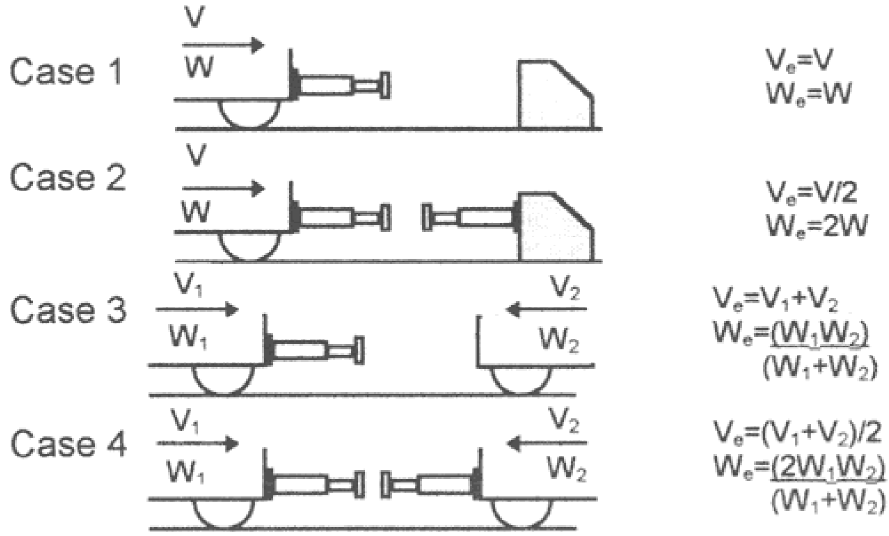
Determining the Impacting Weight:

Cranes are dynamic systems that can change their load distributions as the trolley moves or as "live" loads are lifted or set down. Since the bumper is stationary, it is common to consider the maximum impacting weight that could affect one bumper at a time. This is done similar to the method used for calculating Crane Maximum Vertical Wheel Loads. However, most crane specifications only consider "dead" loads unless the "live" load is suspended on a rigid mast or like device. The trolley is always considered to be in its maximum end approach so as to produce the maximum end reaction for bridge bumper calculation. Association of Iron and Steel Engineers (AISE) Technical Report # 6[1] and Crane Manufacturers' Association of America (CMAA) Specification # 70[2] agree that the following method is typically appropriate for an impact spread over two bumpers:

W_{eB} = Impacting Weight Per Bridge Bumper
W_{eB} = .5 x (Bridge Weight) + .9 x (Trolley Weight)

W_{eT} = Impacting Weight Per Trolley Bumper
W_{eT} = .5 x (Trolley Weight)

The following are four typical impact cases and how they affect the impact weight (We) and impact speed (Ve).



It is always wise to consult with a bumper manufacturer for sizing confirmation whenever the impact condition is other than Case 1 above.

Determining the Impact Speed:

Crane bumpers are needed to dissipate the kinetic energy of the moving crane or trolley. Kinetic energy is related to the travel speed of the body. Therefore, a bumper must be designed for a particular maximum travel speed condition. The most conservative method is to design for the maximum speed attainable. AISE Technical Report # 6 refers to 100% of full load rated travel speed. Some cranes can develop greater travel speeds in a “no load” condition, especially if DC powered. It is therefore advisable to determine the maximum rate of travel for kinetic energy calculations.

A significant percentage of cranes built in the United States each year are built to CMAA Specification # 70. CMAA only requires crane bumpers to have adequate kinetic energy capacity for the crane traveling at 40% of rated load speed. Similarly, trolleys are to be protected to 50% speed. If you were to consider how much energy must be absorbed by the bumpers, you would find a large disparity between AISE and CMAA calculation methods. Kinetic energy at 40% speed is only 16% of that at 100% speed. It is this energy capacity deficit that causes structural damage and unsafe conditions for the operator. An impact at speed greater than the 40% design speed will undoubtedly cause structural damage and poses a severe safety risk for the operator.

Duty class of the crane should influence the designer as to the amount of risk associated with fractional speed bumpers. The CMAA fractional speed recommendation is based on a probabilistic approach. It assumes that the crane will not hit the bumpers at full speed. A trained operator would be able to apply brakes, reduce travel speed or reverse plug to avoid hitting the bumpers at faster than 40% speed. Experience has shown that this approach is adequate for careful, controlled, low speed crane operation, but definitely insufficient in the case of full speed impacts. The catastrophic damage and potential injury to operators should be motivation for crane owners to demand 100% speed protection bumpers. Cost was previously the primary deterrent to this level of protection. Increased production volumes at hydraulic bumper manufacturers has made modern, heavy duty bumpers affordable.

Calculation of Kinetic Energy:

Kinetic energy is associated with a body in motion.

$KE = ½ M V^2$
Where M = Mass and v = Velocity

Consider the following example:

Bridge Weight = 225,000 lbs
Trolley Weight = 35,000 lbs
Bridge Speed = 450 ft/min
Trolley Speed = 200 ft/min

Impact weight and kinetic energy calculations follow:

W_{eB} = .5 x (Bridge Weight) + .9 x (Trolley Weight)
W_{eB} = .5 x (225,000 Ibs) + .9 x (35,000 Ibs)
W_{eB} = 144,000 lbs

$KE_B = ½ M V^2$
$KE_B = ½ (144,000 / 32.2) (450 / 60)^2$
KE_B = 125,776 ft-lbs

W_{eT} = .5 x (Trolley Weight)
W_{eT} = .5 x (35,000 lbs)
W_{eT} = 17,500 lbs

$KE_T = ½ M V^2$
$KE_T = ½ (17,500 / 32.2) (200 / 60)^2$
KE_T = 3,019 ft-lbs

The above Kinetic Energy calculations are based upon AISE 100% speed protection. If CMAA 40% speed level was used, substitute 450 ft/min x .4 = 180 ft/min in the equation. The Kinetic Energy compares as follows:

AISE	CMAA
$KE_{B100\%}$ = 125,776 ft-lbs $KE_{T100\%}$ = 3,019 ft-lbs	$KE_{B40\%}$ = 20,124 ft-lbs $KE_{T50\%}$ = 755 ft-lbs

Deceleration Requirements:

Deceleration is the rate of change in speed. It tells how quickly a body comes to rest. Both AISE Technical Report # 6 and CMAA Specification # 70 offer deceleration limits. Although these limits are similar, they are not identical. In both cases, the deceleration limit is calculated at a fractional speed. AISE calls for maximum deceleration of 16 ft/sec^2 at 50% of full load travel speed. CMAA calls for average deceleration not to exceed 3 ft/sec^2 at 20% of rated load speed for bridge bumpers and 4.7 ft/sec^2 at 1/3 rated load speed for trolleys. Maximum deceleration is calculated as follows:

$$\text{Maximum Deceleration} = \frac{(\text{Velocity})^2}{(2) \times (\text{Bumper Efficiency}) \times (\text{Bumper Stroke})}$$

As can be seen by the above equation, deceleration is only a function of Velocity and Stopping Distance (Bumper Stroke). The mass of the impacting object does not affect the deceleration.

Therefore, deceleration is unaffected when the crane impacts with full "rated load" or "no load" condition.

Reference Accelerations: [3]	
Braking of an Automobile:	26 ft/sec^2
Earth's Gravity:	32 ft/sec^2
Loss of Consciousness:	230 ft/sec^2
Explosive Ejection Seat:	492 ft/sec^2

As you can see in comparison with the above table, the deceleration requirements appear to be low and suggest comfort for the crane operator, even during a collision. However, the corresponding increase in deceleration at higher speeds is a more direct comparison. It is typical to expect as much as 64 fl/sec^2 during a full speed crane impact. This is more than twice as severe as maximum braking effort in an automobile. It would follow that a full speed impact on a bumper designed for only 40% of travel speed could result in a deceleration approaching the "blackout" threshold of the operator. Additionally, the structural design of the crane stop and girder connections do not anticipate loads of this magnitude. The hydraulic crane bumper allows the designer to determine and specify the maximum deceleration and end forces transmitted to the building. The safety of the crane operator is also addressed.

Propelling Forces:

Although most references and design guides suggest that a bumper be sized for a specific travel speed, they all consider drive motors to be off at the moment of impact. If the drive motors are still energized at impact, additional energy capacity will be required. The actual propelling force is limited by tractive effort and power rating of the drive motors. In any case, it is up to the crane designer to recognize that the crane will be used in such a way as to consider power "on" at impact.

$$\text{Propelling Force} = \frac{(\text{ Motor Rated Power HP }) \times (\text{ 33,000 })}{(\text{ Travel Speed ft/min })}$$

$$KE_{Motors} = (\text{ Propelling Force }) \times (\text{ Bumper Efficiency }) \times (\text{ Bumper Stroke })$$

Assume that the sample crane has a 100HP motor as its bridge drive. The total propelling force is 7,333 lbs over two bumpers. This amounts to 4,090 ft-lbs per bumper or 3% additional energy capacity required.

Due to the flywheel effect of rotating machinery, a significant amount of propelling force and kinetic energy can be present. If you are considering rotational inertia, determine the wk^2 term for the drive motors, gear box and brake wheel from the manufacturers' literature.

$$wk^2_{\,(Total)} = wk^2_{\,(Motor)} + wk^2_{\,(Gearbox)} + wk^2_{\,(Brakes)} \qquad (wk^2 = \text{lb-ft}^2)$$

Calculate the equivalent weight as follows:

$$W_{Equiv} = \frac{(\, wk^2_{\,(Total)}\,) \times (\text{ 39.49 }) \times (\text{ Max RPM })^2}{(\text{ Travel Speed ft/min })^2}$$

$$KE_{Rot} = \tfrac{1}{2} \times \frac{(\, W_{Equiv}\,)}{32.2} \times (\text{ Travel Speed} / 60\text{ })^2$$

If the sample crane has $wk^2_{(Motor)} = 42.3$, $wk^2_{(Gearbox)} = 31.2$, $wk^2_{(Brakes)} = 20.7$ and maximum operating RPM of 1200, then we have an additional Kinetic Energy of 23,105 ft-lbs total. Each bumper requires 11,553 ft-lbs or 8% additional capacity energy capacity.

"Another factor to consider is the ratio of the number of wheels driving the crane to the total number of wheels. This is because most cranes have their drive motors directly connected to a portion of the wheels. The rotational inertia of the drive motor and brake wheel may carry a substantial amount of kinetic energy. Furthermore, if the motors are energized at the time of impact they will feed additional propelling work through the wheels to further load the bumpers. The amount of energy and work from these two sources that can be felt by the bumper is limited by drive wheel adhesion. Very often, the rotational energy alone is enough to exceed an adhesion of say, 25%. Thus it usually doesn't matter what the rotational inertia is or if the drive motors are energized at the time of impact. To find the adhesion force, we consider the total load, the percentage of wheels driving, and the friction factor for the crane wheels on the runway rails."[4]

For the given crane example, assume 50% of the wheels are driven and adhesion between wheels and rails is .20.

$$\begin{aligned}\text{Propelling Force} &= (\text{ Total Crane Weight }) \times (\text{ Adhesion }) \times (\text{ \% of Wheels Driven }) \\ &= (\ 225{,}000 + 35{,}000\ ) \times (\ .20\ ) \times (\ .50\ ) \\ &= 26{,}000 \text{ lbs. Total}\end{aligned}$$

$$KE_{Prop} = (\text{ Propelling Force }) \times (\text{ Bumper Efficiency }) \times (\text{ Bumper Stroke })$$

At impact, with the motors energized, each bumper "feels" as much as 13,000 lbs of propelling force. This is the governing limit for tractive effort propelling force. If the selected bumper had a stroke of 15.75 inches, the additional energy capacity required would be 14,503 ft-lbs. This amounts to approximately 12% of the bumper energy capacity.

If the drive motors are energized at impact, both rotational inertia and drive force will be present. We calculated 4,090 ft-lbs + 11,553 ft-lbs = 15,643 ft-lbs per bumper. Tractive effort limits the actual transmitted propelling force and energy to 14,503 ft-lbs. Therefore, this value should be used as the maximum propelling energy from drive motors.

Outdoor cranes can be subjected to severe weather elements. Wind gusts can load a crane above the anticipated levels, often causing higher travel speeds as well as propelling forces at impact. Many cranes have electrical controls or braking systems to prevent the crane from traveling faster than the design speed. However, many cranes are still subject to this phenomena. The designer should compute the sail area or projected area in the expected worse possible direction of crane travel. Wind loading is considered to be 5 lbs. per square foot of projected area, modified by shape factors. This is equivalent to 44 miles per hour constant velocity without gust factor.[5]

$$KE_{Wind} = (\text{ Propelling Force From Wind }) \times (\text{ Bumper Efficiency }) \times (\text{ Bumper Stroke })$$

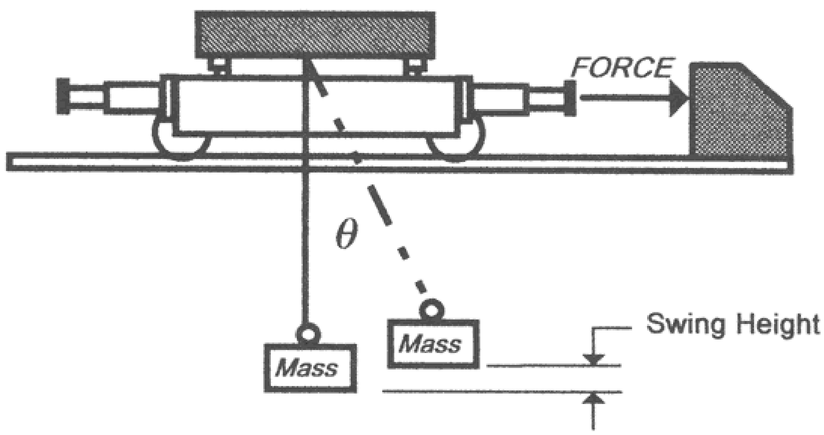
If the sample crane had span of 80 ft and projected height of 4 ft for total projected area of 320 ft^2, the propelling force would be as much as 1,600 lbs total. If the same stroke of 15.75 inch stroke is used, this is 1,785 ft-lbs or 1% of the bumper's kinetic energy capacity.

Therefore, we nave calculated an additional kinetic energy requirement of 16,288 ft-lbs. The kinetic energy at 100% speed was determined to be 125,776 ft-lbs. Propelling forces can potentially add as much as 11% to the kinetic energy requirement as shown above.

<u>Swinging, Cable Hung Loads</u>:

At impact, the crane is decelerated by the hydraulic bumpers until it comes to rest. The bumper force is transmitted through the crane and reeving system until the lifted load is also brought to rest. A simply supported, cable hung load will act much like a pendulum. Kinetic energy is converted to potential energy by lifting the load through a vertical displacement. [6]

Kinetic Energy	= Potential Energy
$½MV^2$	= (Load Free to Swing) x (Verticai Displacement)
Vertical Displacement	= ($½ V^2$) / (G)



At maximum angular displacement, (q), a right triangle is formed with hypotenuse equal to the rope length and an adjacent side equal to the rope length minus the vertical displacement.

$$\cos(q) = \frac{(\text{ Rope length - Vert Displacement })}{(\text{ Rope length })}$$

The effective propelling force is equal to the component of the lifted load acting in the horizontal direction:

$$\text{Propelling Force} = \frac{(\text{ Swinging Load }) \times \sin(q)}{\cos(q)}$$

If the load of 35 tons is considered at a rope length of 15 ft and speed is 450 ft/min,

$$\begin{aligned}
\text{Vertical Displ.} &= ½\, V^2 / G \\
&= ½\, (450 / 60)^2 / 32.2 \\
&= .87 \text{ ft} \\
\cos(q) &= \frac{(\text{ Rope length - Vert Displacement })}{(\text{ Rope length })} \\
&= \frac{(15 \text{ ft} - .87 \text{ ft})}{(15 \text{ ft})} \\
q &= 21.79°
\end{aligned}$$

$$\text{Propelling Force} = \frac{(\text{Swinging Load}) \times \sin(\theta)}{\cos(\theta)}$$

$$= \frac{(70{,}000\text{ lbs}) \times \sin(21.79°)}{\cos(21.79°)}$$

$$= 24{,}941\text{ lbs Total}$$

Again utilizing the bumper with 15.75 inches of stroke, this yields an additional energy requirement of 13,912 ft-lbs or 10% of the original bumper capacity.

It should be noted, however, that the period of the cable hung load is typically much larger than the duration of the bumper stroke. Therefore, only a portion of this propelling force as calculated above need be accounted for in bumper sizing. The residual force is transmitted through the bumper into the crane stop and building after the completion of the bumper stroke.

Between propelling force from motors, wind loading and swinging loads, we see that there can potentially be as much as 19% more kinetic energy capacity required than accounted for in AISE Technical Report # 6. Per CMAA Specification # 70, there can potentially be as much as 87% of the energy not accounted for. It is this specific discrepancy that has caused several severe accidents.

<u>Design Features of Hydraulic Bumpers</u>:

The design of a hydraulic bumper is unique to the particular manufacturer. There is no current standard for interchangeability among competitively manufactured brands. Several features are common while others are special options available upon request.

A hydraulic bumper consists of a shock absorber damping element along with a reset mechanism. Typically, the bumper has a biasing mechanism that controls the rate of reset after impact. Oil is passed through an orifice to convert mechanical energy to heat energy.

A heavy duty crane bumper will have a robust design with seals internal and shielded from the environment. Airborne dust and debris are typically removed by scraper seals.

Occupational Safety and Health Administration (OSHA) CFR 1910.179[7] identifies the need for bumpers to be designed to minimize the possibility of parts falling to the floor in the event of breakage. AISE specifically mentions the use of safety cables. It is common to see an aircraft quality wire rope looped from the bumper head to the bumper body. This is crimped for permanent attachment. Additionally, it is common practice to have an additional wire rope from the bumper to the end truck in case the mounting bolts should vibrate loose. There are only a few isolated incidents where bumpers have been damaged during operation and the plunger has fallen to the floor. This was determined to be the result of one particular proprietary design that utilized a small cap screw through the mounting base into the piston rod. Safety dictates that this connection be a substantial, fully threaded outer diameter connection. The safety cable offers an additional feature. When the bumper is manufactured, the cable is looped in place to be slightly taut. Once installed on the crane, a visual inspection of the cable will determine whether the bumper is fully extended or reset. If the cable is drooping, the bumper could be damaged and not fully reset.

Hydraulic bumpers can be mounted on the crane end trucks or on the end stops. In some operations, where two cranes come together, bumpers impact into the bumpers on an adjacent crane. The contact surface or diameter of the bumpers should not be less than 5 inches in diameter. AISE addresses this point while CMAA does not. Experience has shown that cranes can have as much as 2 inches of wheel float. This coupled with possible misalignment in the bumper mounting has led to several cases of one bumper missing the adjacent one causing severe crane damage. One manufacturer has their entire range with contact surface diameters of 5 inches and greater. Optional larger diameter and spherical contours are also offered for special conditions. As bumpers vary from manufacturer to manufacturer, It is important to keep this factor in mind when specifying and carefully check dimensional drawings for clearances and interfaces.

Each hydraulic bumper contains a reset mechanism that restores the plunger for the next impact. This reset mechanism is classified as a spring and can be identified through a static resistance curve. If a bumper is compressed at an infinitely slow rate and then held compressed, the static resistance curve is developed. The characteristics of the restoring force is available from the manufacturer of the bumper. In some installations, the crane bumpers must be compressed and held compressed to gain maximum end approach or position at a loading platform. The designer must determine the size of the drive motors and tractive effort to guarantee that the crane or trolley has sufficient torque to compress the bumpers at slow closure speeds. Reset mechanisms vary from steel coil springs with complicated biasing valves to hydro pneumatic gas springs similar to the landing gear strut on an aircraft. Some bumpers have a rubber bladder with gas inside to accommodate the displacement of the piston or plunger as the bumper is stroked. In any event, the crane designer must consult the bumper manufacturer for this data if full end approach is critical.

There has been a significant debate in the industry as to which reset mechanism is the most reliable. Experience has shown that if the restoring force is very low, the bumper can be easily jammed from debris, misalignment, corrosion, etc. A restoring force of 10-20% of the dynamic force rating is most reliable. This can be developed by use of a “liquid spring” where the hydraulic oil is of a compressible silicone type or by use of a nitrogen gas charge in polytropic compression. Bumpers with liquid elastomer are similar in characteristic to rubber and spring bumpers where energy is stored and not dissipated. The coefficient of restitution is high and the energy is returned to the impacting crane.

Since the hydraulic bumper utilizes standard seals and conventional hydraulic fluids, care must be taken to consider the ambient temperature range. Cranes often operate in environments that range from -40°F to +200°F. Depending on the manufacturer, special seals and/or oil may be required to accommodate these temperature extremes. Most bumpers can withstand transient temperature extremes as when operating near a furnace. It is prolonged, sustained heat above 165°F that can deteriorate the seals. Always identify the ambient temperature to the bumper supplier to avoid problems.

Cranes often operate in environments where there is substantial airborne dust and debris. Similarly, outdoor snow and ice conditions can present a problem for bumpers. Most manufacturers offer a nylon reinforced rubber protective bellows or boot to be fitted over the bumper plunger. As the bumper is compressed, the volume of air trapped inside the bellows changes and brass breather elements are necessary to avoid inflation and rupture of the fabric. It is my experience that protective bellows can also trap dirt and pose a threat to jamming the bumper. If the bellows becomes torn, the material can become entrained into the bumper as the bumper is stroked. The best method for keeping dirt out of a bumper is the use of a scraper seal.

Corrosion resistance is often a concern where bumpers operate in marine or caustic environments. Most manufacturers use chrome plating to provide corrosion resistance. This plating can consist of layers of nickel followed by hard chrome. The mounting brackets are typically painted along with the end truck of the crane, after installation. Most manufacturers supply their bumpers standard with a zinc chromate primer and leave the final painting to the crane builder. A multi layer epoxy paint works extremely well for prevention of rust and many types of caustic corrosion. In some cases, full stainless steel construction has been considered, but due to cost, has not yet been utilized.

The mounting methods of hydraulic bumpers are fairly common among the various manufacturers. However, some manufacturers standardize on one type of bracket and assume that the crane will be designed to suit the bumper. One manufacturer offers bumpers with boltable flanges that can be mounted from the back end of the bumper or from a flange in the mid section of the bumper. Further, the bumper is available in capsule form to allow the crane builder to use a press fit into a fabricated bracket for low profile mounting as on a bridge girder. Creative use of these available mounting styles can offer many advantages. Useful runway length and crane approach can be maximized by using bumpers with the mid mount, known as front flange mounting. The compressed length of the bumper is minimized to allow the closest approach. Often, the boltable flange is square in shape. One manufacturer makes this flange as a standard dimension and pattern. This allows bumpers to be upgraded to greater capacities without having to re-drill mounting holes. This is especially useful for an owner that bought the crane with CMAA 40% speed protection bumpers and then realizes that he wants to upgrade to AISE 100% speed bumpers.

There are no set rules as far as where bumpers must be mounted. AISE suggests that bumpers be mounted over the center line of the rail, but in cases where two bumpers are used per corner, it should be expanded to suggest symmetry about the center line of the rail. Bumpers can be interchangeably mounted on the crane end trucks or on the crane stops mounted to the runway girders. Where two cranes come together, it is often most convenient to have a pair of bumpers on one crane and steel plate striking surface on the other side. The bumpers must be sized appropriately to absorb the energy from both cranes or either crane as required.

A Comparison of Metering Types:

The dynamic characteristics of hydraulic bumpers vary from manufacturer to manufacturer and upon their physical, internal design. Currently there are only three major types of hydraulic bumpers that are manufactured. These include the metering pin bumper, the metering tube bumper and the fluidic bumper. AISE Technical Report No. 6 suggests a deceleration rate of 16 ft/sec^2 at 50% of full load rated travel speed. Additionally, the bumper must be capable of absorbing the energy of the unloaded crane at 100% travel speed with corresponding increase in deceleration. Most engineers in the crane industry agree that this maximum deceleration rate at 100% travel speed is 2G's or 64.4 ft/sec^2. Only a hydraulic bumper of the metering tube or metering pin design can accomplish both points on the curve described by 16 ft/sec^2 at 50% and 64.4 ft/sec^2 at 100% speed. A fluidic piston head hydraulic bumper has a characteristic "Force vs. Velocity" relationship that approximates the equation:

$$F = C * V^N$$

(Where N varies from .5 to .7 depending on specific design configuration)
(C is the Damping Constant)

Metering pin and metering tube hydraulic bumpers have a characteristic "Force vs. Velocity" relationship that approximates the equation:

$$F = C * V^2$$

The implications of this fundamental difference are shown in two enclosed graphs. *(See Appendix)* One shows Peak Deceleration vs. % Speed while the other shows the Typical Force vs. Stroke curves at various velocities and % Speeds. Only a bumper that follows $F=C*V^2$ will utilize all of its stroke at any partial speed. The metering pin and metering tube bumpers compensate for speed changes by adjusting end force in this way. Conversely, the fluidic bumper compensates by adjusting its stroke while maintaining nearly constant force at all speeds. It is not truly velocity sensitive and causes greater cumulative stress on the building and crane structures.

When an end force or deceleration is given for 100% speed, typically, the values at 50% speed are determined by dividing by 4. This only holds true if the bumper follows the $F=C*V^2$ relationship. The proper way to determine End Force or Deceleration at partial impacting speeds is to solve for the proportionality constant, C.

For example) if a metering pin bumper has End Force of 92,813 lbs at 100% Speed (325 FPM), then:

$$\text{End Force}_{100\%} = C*V^2$$
$$92{,}813 = C*325^2$$
$$C = .8787$$

End Force at 50% speed (162.5 FPM) is:

$$\text{End Force}_{50\%} = (.8787) * (162.5)^2$$
$$\text{End Force}_{50\%} = 23{,}203 \text{ lbs.}$$

This is not the case for a fluidic hydraulic bumper. For example, a bumper manufacturer reports that his bumper follows the relationship: $F=C*V^{.62}$. In this case, the proportionality constant, C, is $92{,}813 = C * 325^{.62}$, or C = 2,571.82. End Force at 50% speed (162.5 FPM) is F = 2,571.82 * $(162.5)^{.62}$ or End Force = 60,391 lbs. Clearly, the same holds true for deceleration rates. If End Force = 92,813 lbs corresponds to 64 ft/sec^2, then at 50% speed, a Metering Pin bumper would have deceleration of 16 ft/sec^2 while a Fluidic hydraulic bumper would yield deceleration of 41.7 ft/sec^2 , thereby not satisfying the requirements of AISE TR #6. Additionally, the fluidic bumper would only utilize 38% of its available stroke while the metering pin bumper uses all available stroke to keep the end force at its minimum.

Again, consider the sample crane:

Bridge Weight = 225,000 lbs	@	450 FPM
Trolley Weight = 35,000 lbs	@	200 FPM
(Bridge Effective Weight = 144,000 lbs)		
(Kinetic Energy = 125,776 ft-lbs)		

To properly compensate for the deceleration, it is advisable to determine the minimum stroke required. This is done as follows:

F = M * A (F = End Force at 50% Speed)
(M = Effective Weight / Gravity)
(A = Deceleration rate or 16 ft/sec^2 at 50% speed)

$$16 \text{ ft/sec}^2 = (\text{ End Force}_{50\%}) / (\text{ Effective Weight } / 32.2 \text{ ft/sec}^2)$$

$$16 = (\text{ End Force}_{50\%}) / (144{,}000 / 32.2)$$

$$(\text{ End Force}_{50\%}) = 71{,}553 \text{ lbs}$$

Now, we can solve for the proportionality constants, end forces and decelerations at any speed for any manufacturer. Remembering to use the proper $F= C*V^N$ relationships, assuming that the Fluidic hydraulic bumper from a particular manufacturer has N= .62 , we solve for C:

<u>Metering Pin</u>		<u>Fluidic</u>	
F	$= C * V^2$	F	$= C * V^{.62}$
71,553	$= C * (450 * .5)^2$	71,553	$= C * (450*.5)^{.62}$
C	= 1.413	C	=2,490.432

Now calculate the end force at 100% travel speed:

$F_{100\%} = (1.413) * (450)^2$
$F_{100\%} = 286,132$ lbs

$F_{100\%} = (2,490.432) * (450)^{.62}$
$F_{100\%} = 109,968$ lbs

From the end force at 100% speed, we can calculate how much stroke is required to absorb all of the kinetic energy:

Min. Stroke = KE / (.9) * ($F_{100\%}$)
Min Stroke = 125,776 / (.9 * 286,132)
Min. Stroke = .49 ft = 5.86 inches

Min. Stroke = KE / (.9) * ($F_{100\%}$)
Min. Stroke = 125,776 / (.9* 109,968)
Min. Stroke = 1.27 ft = 15.25 inches

Obviously, the more stroke provided by the hydraulic bumper, the lower the end force will be and thus the decelerations will also be lower. For this application, even though AISE allows as little as 3.75 inches of stroke, we have selected a metering pin bumper based on the energy capacity requirement at 100% speed. It has 15.75 inches of stroke. At 100% speed, end force is 106,477 lbs. At 50% speed, end force is 26,619 lbs. Decelerations are 24 ft/sec^2 and 6 ft/sec^2 respectively.

Assuming that the fluidic hydraulic bumper has at least 15.25 inches of stroke, it will satisfy the deceleration requirement at 50% speed. Note that a metering pin or metering tube bumper could satisfy the technical requirements in only 5.86 inches of stroke as compared to 15.25 inches of the fluidic. If the Fluidic bumper were designed to only meet the 100% kinetic energy and the 64 ft/sec^2 deceleration at 100% speed, then only 5.86 inches of stroke would be required. The deceleration at 50% speed would, however, be 42 ft/sec^2 as compared to the AISE requirement of 16 ft/sec^2. This phenomena can be more easily studied by examining the graphs of deceleration vs. % speed and typical force vs. stroke. *(See Appendix)*

The bumper specifications in AISE Technical Reports No. 6 and 13 are not in any way exclusive to a particular brand of hydraulic bumpers. Manufacturers of true hydraulic bumpers all have the capability of building a bumper with a metering pin or metering tube that follows the $F=C*V^2$ relationship. The benefit of the lower deceleration rate provided at partial speeds by the metering pin and metering tube bumpers is reduction in cumulative stress in the building and in the crane each time the bumpers are impacted.

Sample Specification:

When electric overhead traveling cranes are purchased, typically, a data sheet or complete specification is "sent out for bids". Only when this specification clearly identifies what the purchaser wants, will the crane builder be able to completely satisfy his customer.

The following specification should be used for all cranes or transfer vehicles that are CMAA Classes C and D with travel speeds greater than 250 FPM, CMAA Classes E and F (any speed) as well as all AISE classes. This specification is based upon AISE Technical Report # 6, Sept. 1991 and satisfies the requirements of CMAA Specification #70 as well as OSHA.

Technical Specification G137-B
Energy Absorber Device: Crane Bumpers

The crane shall be equipped with hydraulic bumpers on both bridge and trolley travel motions. The bumpers shall be capable of absorbing the total energy at 100% full load rated speed (or maximum free running speed if DC powered) with deceleration not to exceed 16 ft/sec^2 at 50% of the full load travel speed. End force per bumper shall not exceed 160 kips. Energy shall be calculated with trolley in full end approach so as to produce the maximum end reaction. Power is to be considered "off" at impact. Lifted loads that are free to swing are not to be included in energy calculations. If the load is rigidly or semi-rigidly supported, an appropriate ratio of 30% to 100% of the load should be included as dead load on the trolley.

Between cranes or trolleys, bumpers shall be capable of absorbing the energy from 70% of full load travel speed (or 70% of maximum free running speed if DC powered) of both cranes or trolleys traveling in opposite directions, or the energy from 100% of full toad rated speed (or maximum free running speed if DC powered) of either crane or trolley, whichever is the greatest. Trolleys are to be considered in maximum end approach and on the same side.

The design of all bumpers shall include safety cables to prevent parts from falling to the floor. Additionally, bridge bumpers or wherever two trolleys come together, shall have a contact surface not less than 5 inches in diameter to allow proper alignment between bumpers. The bumpers shall be tamper-proof and require no field adjustment.

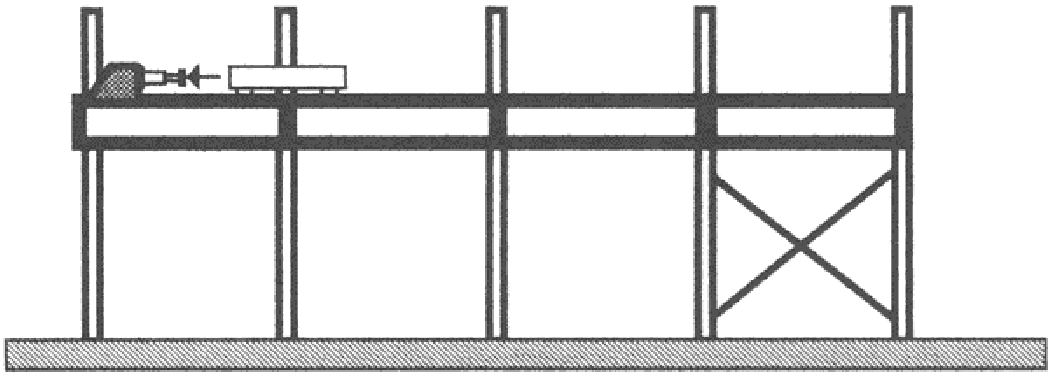
The unit shall be of heavy duty construction and shall incorporate components that are manufactured from high grade materials and to such high quality as to permit complete interchangeability of parts during assembly and servicing.

Bumper End Force and the Building:

The primary reason that hydraulic bumpers are utilized is to reduce the impact damage imparted to the building and crane stops by the crane in a collision. Most often, the building is designed prior to the design of the crane. The building designer must make certain assumptions regarding the longitudinal forces that the crane will impart. The building designer could make use of crane duty requirements and estimated speeds to input into the deceleration requirements of the AISE Technical Report # 6 and determine a maximum bumper force. However, he can not know what type of bumper will be used (i.e. metering pin, metering tube, fluidic, spring, etc.) unless it is specified. Therefore, a simplification in building design can be achieved by specifying in the structural steel drawings a maximum end force per bumper at 100% speed. The sample specification above identifies 160,000 lbs as a practical limit for end force.

The bumper end force is delivered onto the crane stop and transmitted into the building structure. This force must be transferred back to the nearest braced bay to complete the load path to the foundation. *(See Figure Below)* The greater the bumper end force, the more steel required for the longitudinal building tie to transmit loads. In most cases, a careful analysis of this load path can reduce the structural steel cost enough to more than pay the cost of the hydraulic bumpers.

It can not be emphasized enough how important it is for the building structural engineer to convey the bumper end force limits to the crane builder. This data transfer should be managed via the building and crane layout drawings. Also, please note that if a building is designed to AISE Technical Report # 138 then it is assumed that the crane will be protected by bumpers up to 100% of full load rated travel speed. This requires modification to the crane specification if it bears a CMAA rating since CMAA Specification #70 only requires fractional speed protection.



Conclusions:

Proper attention should be given to hydraulic bumpers in new and existing operations. They are engineered devices built for specific impact conditions. CMAA Specification # 70 and current OSHA recommendations do not go far enough to protect the crane operator who relies on the mechanical components of the crane for his personal safety. The mill buildings will survive the rigors of operation if and only if proper design limit states have been used. By using careful consideration of the operational characteristics, the designers can anticipate the potential for damage, accident or injury. Previously, this level of protection was not economically available. Now, this technology is available, proven reliable and cost effective. In many cases, the reduction in forces reduce costs. It is now obvious that new attention must be given to the modernization of the relevant specifications pertaining to cranes and bumpers. This paper is intended to launch that discussion.

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[1] **Association of Iron and Steel Enninneers (AISE) Technical Report No. 6**, 1991, Association of Iron and Steel Engineers, Pittsburgh, P.47-48.

[2.] **Crane Manufacturers Association of America Specification # 70**, 1994, The material Handling Institute, P.51.

[3.] **Physics, Hans C. Ohanian, 1985 W.W. Norton** & Company, New York, P.30.

[4.] **Hydraulic Energy Absorbers for Cranes,** ASME Paper 66-WA/SAF-2, Henry S. Germond, Nov. 1966.

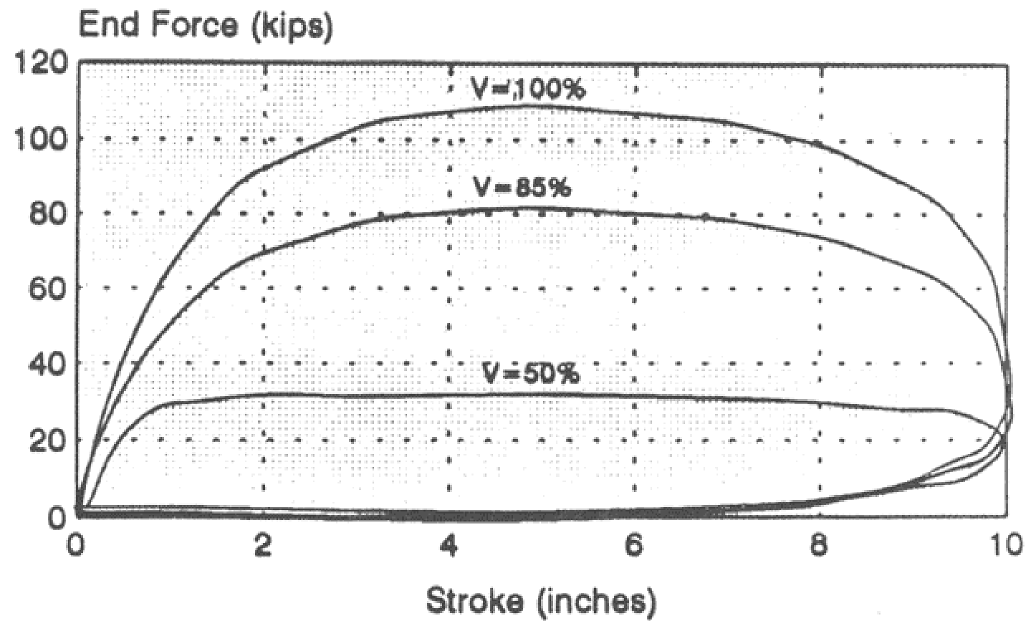
[5] **Whiting Crane Handbook**, Wm. M. Weaver, Whiting Corp., Harvey, IL., 1985.

[6] **Design and Selection Criteria for the Hydraulic Buffer,** Douglas P. Taylor, Paper presented at AISE Rolling Mill Conference, 1978.

[7] **Occupational Safety and Health Administration Code of Federal Regulations**, 1994, 1910.179, P. 520.

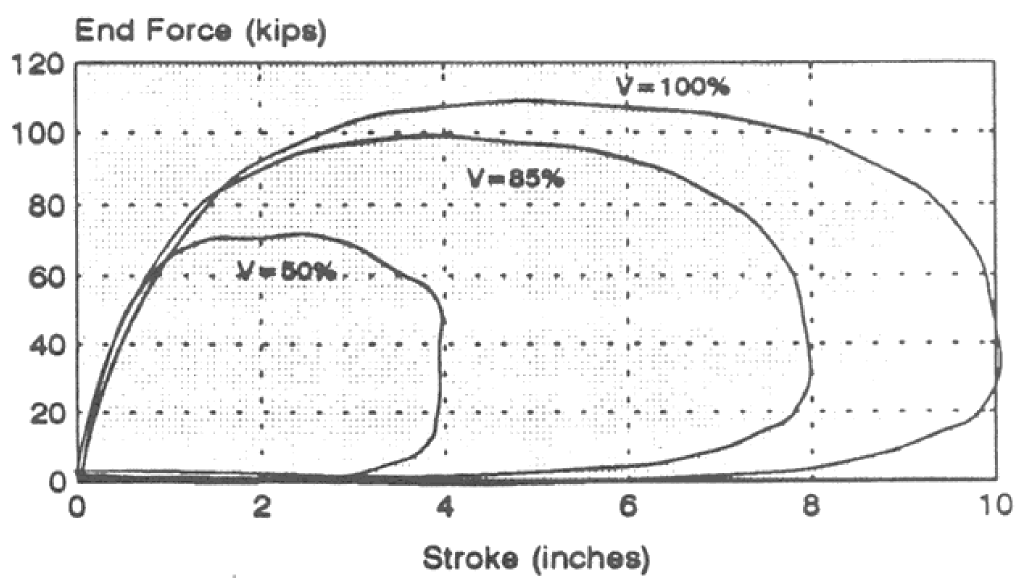
[8] **Association of Iron and Steel Engineers (AISE) Technical Report No. 13**, 1991, Association of Iron and Steel Engineers, Pittsburgh.

Typical Force vs. Stroke Curve
at Different Impact Velocities



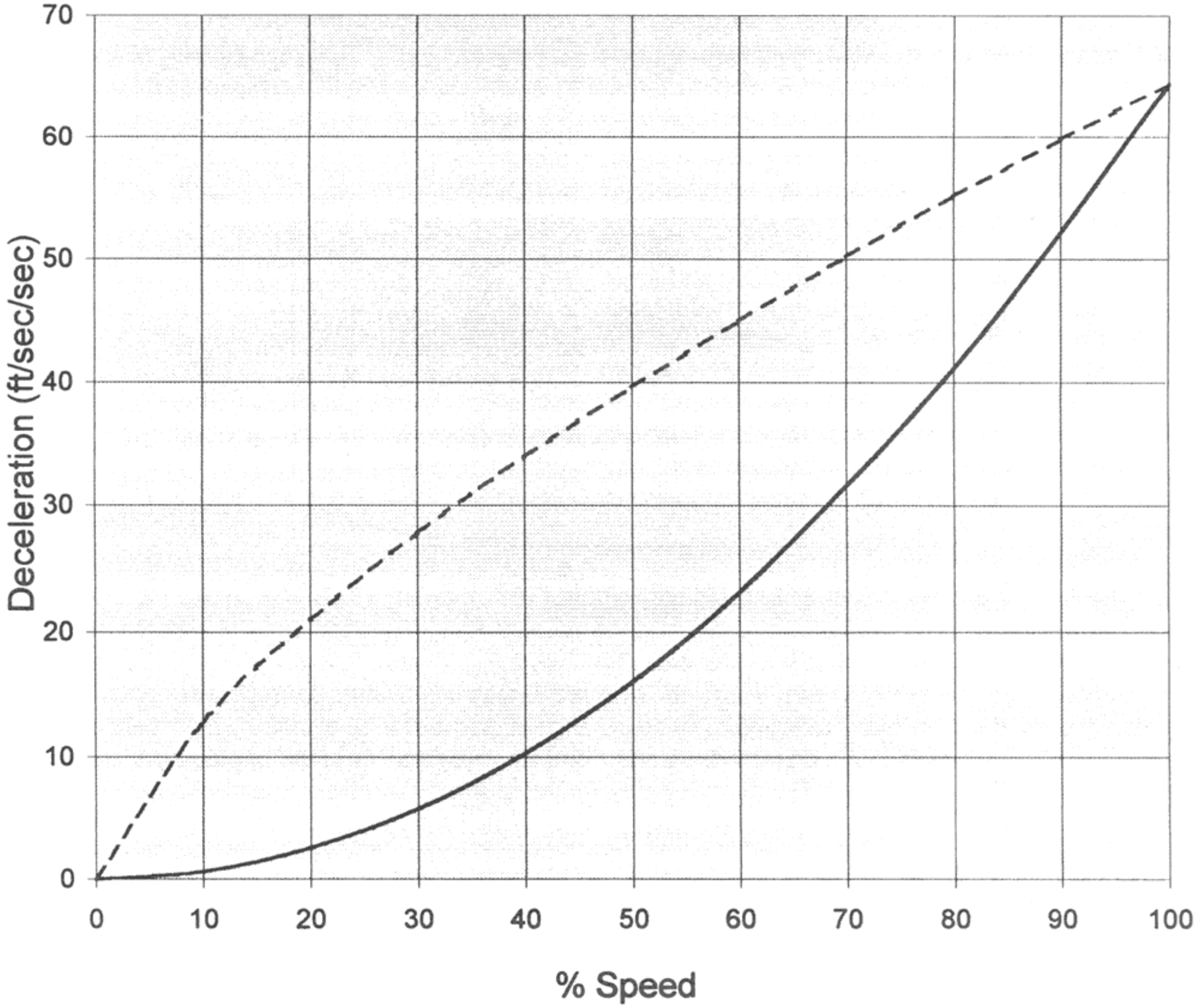
Metering Pin or Metering Tube Bumper

Typical Force vs. Stroke Curve
at Different Impact Velocities



Non-Metering Pin Type Bumper

Peak Deceleration vs. % Speed



–––––– Metering Pin – – – – Fluidic