Overview

With its world headquarters located in Orchard Park, New York, USA, **ITT ENIDINE Inc.** is a world leader in the design and manufacture of standard and custom energy absorption and vibration isolation product solutions within the Industrial, Aerospace, Defense, Marine and Rail markets. Product ranges include shock absorbers, gas springs, rate controls, air springs, wire rope isolators, heavy industry buffers and emergency stops. With facilities strategically located throughout the world and in partnership with our vast global network of distributors, Enidine Incorporated continues to strengthen its presence within marketplace.

Founded in 1966, ITT Enidine Incorporated now has close to 600 employees located throughout the globe in the United States, Germany, France, Japan, China and Korea. With a team of professionals in engineering, computer science, manufacturing, production and marketing our employees provide our customers the very best in service and application solutions.

"ITT Enidine is widely recognized as the preferred source for energy absorption and vibration isolation products."

From Original Equipment Manufacturers (OEM) to aftermarket applications, ITT Enidine offers a unique combination of product selection, engineering excellence and technical support to meet even the toughest energy absorption application requirements.

Global Manufacturing and Sales Facilities offer our customers:

- ***Highly Trained Distribution Network***
- ***State-of-the Art Engineering Capabilities***
- ***Custom Solution Development***
- ***Customer Service Specialists***
- ***Multiple Open Communication Channels***

If you are unsure whether one of our standard products meets your requirements, feel free to speak with one of our technical representatives **toll-free at 1-800-852-8508**, or contact us via **e-mail at techsales@enidine.com**.

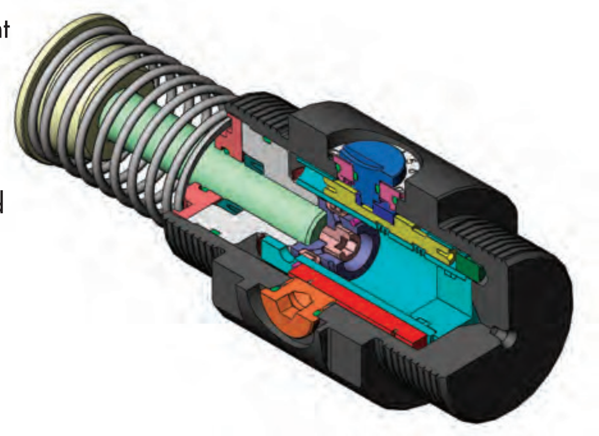
Products/Engineering/Technical Support

ITT Enidine continually strives to provide the widest selection of shock absorbers and rate control products in the global marketplace. Through constant evaluation and testing, we bring our customers the most cost effective products with more features, greater performance and improved ease of use.

New Products and Services

ITT Enidine engineers continue to monitor and influence trends in the motion control industry, allowing us to remain at the forefront of new energy absorption product development such as our new ECO Series shock absorbers and our new HDN Series shock Absorbers.

Our experienced engineering team has designed custom solutions for a wide variety of challenging applications, including automated warehousing systems and shock absorbers for hostile industrial environments such as glass manufacturing, among others. These custom application solutions have proven to be critical to our customers' success. Let ITT Enidine engineers do the same for you.



Custom designs are not an exception at ITT Enidine, they are an integral part of our business. Should your requirements fit outside of our standard product range, Enidine engineers can assist in developing special finishes, components, hybrid technologies and new designs to ensure a "best-fit" product solution customized to your exact specifications.

A talented engineering staff works to design and maintain the most efficient energy absorption product lines available today, using the latest engineering tools:

- ***Solid Modeling***
- ***3-D CAD Drawings***
- ***3-D Soluable Support Technology***
- ***Finite Element Analysis***
- ***Complete Product Verification Testing Facility***

New product designs get to market fast because they can be fully developed in virtual environments before a prototype is ever built. This saves time and lets us optimize the best solution using real performance criteria.

Global Service and Support

ITT Enidine offers its customers a global network of customer service staff technical sales personnel that are available to assist you with all of your application needs.

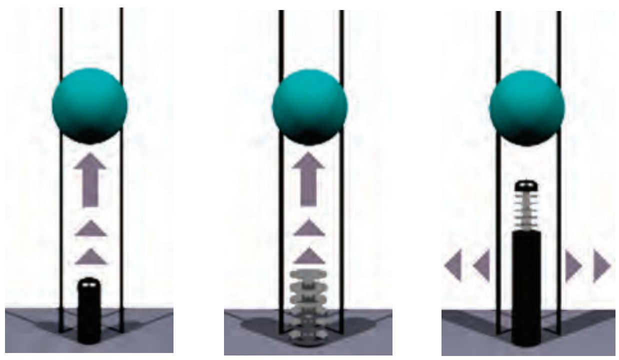
- Operating with lean manufacturing and cellular production, ITT Enidine produces higher quality custom and standard products with greater efficiency and within shorter lead times.
- An authorized Global Distribution Network is trained regularly by ITT Enidine staff on new products and services ensuring they are better able to serve you.
- New Enisize sizing portal provides our customer with the necessary sizing and design tools. www.enisize.com
- Global operations in United States, Germany, France, China, Japan and Korea.
- A comprehensive, website full of application information, technical data, sizing examples and information to assist in selecting the product that's right for you.

Our website also features a searchable worldwide distributor lookup to help facilitate fast, localized service. Contact us today for assistance with all of your application needs.

Our global customer service and technical sales departments are available to assist you find the solution that's right for your application needs. Call us at 1.800.852.8508 or e-mail us at industrialsales@enidine.com and let us get started today.

Overview

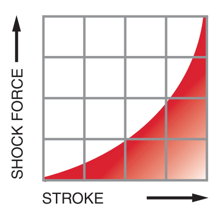
As companies strive to increase productivity by operating machinery at higher speeds, often the results are increased noise, damage to machinery/products, and excessive vibration. At the same time, safety and machine reliability are decreased. A variety of products are commonly used to solve these problems. However, they vary greatly in effectiveness and operation. Typical products used include rubber bumpers, springs, cylinder cushions and shock absorbers. The following illustrations compare how the most common products perform:



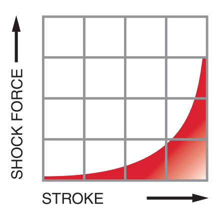
Rubber Bumper **Metal Spring** **ITT Enidine Shock Absorber**

All moving objects possess kinetic energy. The amount of energy is dependent upon weight and velocity. A mechanical device that produces forces diametrically opposed to the direction of motion must be used to bring a moving object to rest.

Rubber bumpers and springs, although very inexpensive, have an undesirable recoil effect. Most of the energy absorbed by these at impact is actually stored. This stored energy is returned to the load, producing rebound and the potential for damage to the load or machinery. Rubber bumpers and springs initially provide low resisting force which increases with the stroke.



Cylinder cushions are limited in their range of operation. Most often they are not capable of absorbing energy generated by the system. By design, cushions have a relatively short stroke and operate at low pressures resulting in very low energy absorption. The remaining energy is transferred to the system, causing shock loading and vibration.



Shock absorbers provide controlled, predictable deceleration. These products work by converting kinetic energy to thermal energy. More specifically, motion applied to the piston of a hydraulic shock absorber pressurizes the fluid and forces it to flow through restricting orifices, causing the fluid to heat rapidly. The thermal energy is then transferred to the cylinder body and harmlessly dissipated to the atmosphere.

The advantages of using shock absorbers include:

1. Longer Machine Life – The use of shock absorbers significantly reduces shock and vibration to machinery. This eliminates machinery damage, reduces downtime and maintenance costs, while increasing machine life.

2. Higher Operating Speeds – Machines can be operated at higher speeds because shock absorbers control or gently stop moving objects. Therefore, production rates can be increased.

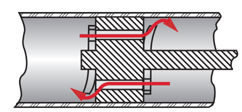
3. Improved Production Quality – Harmful side effects of motion, such as noise, vibration and damaging impacts, are moderated or eliminated so the quality of production is improved. Therefore, tolerances and fits are easier to maintain.

4. Safer Machinery Operation – Shock absorbers protect machinery and equipment operators by offering predictable, reliable and controlled deceleration. They can also be designed to meet specified safety standards, when required.

5. Competitive Advantage – Machines become more valuable because of increased productivity, longer life, lower maintenance costs and safer operation.

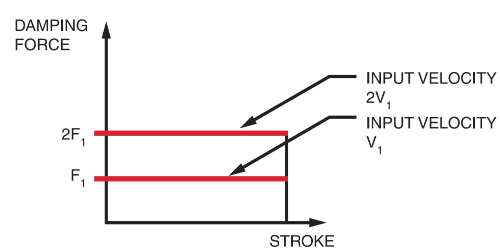
Automotive vs. Industrial Shock Absorbers

It is important to understand the differences that exist between the standard automotive-style shock absorber and the industrial shock absorber.

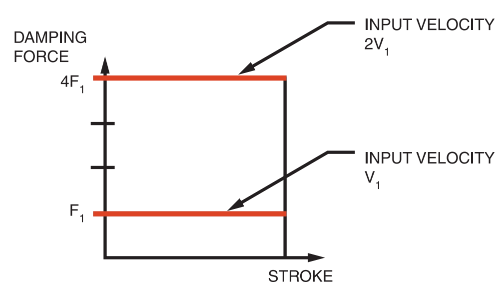


DEFLECTIVE BEAM ORIFICE

The automotive style employs the deflective beam and washer method of orificing. Industrial shock absorbers utilize single orifice, multi-orifice and metering pin configurations. The automotive type maintains a damping force which varies in direct proportion to the velocity of the piston, while the damping force in the industrial type varies in proportion to the square of the piston velocity. In addition, the damping force of the automotive type is independent of the stroke position while the damping force associated with the industrial type can be designed either dependent or independent of stroke position.



AUTOMOTIVE TYPE SHOCK ABSORBER



INDUSTRIAL TYPE SHOCK ABSORBER

Equally as important, automotive-style shock absorbers are designed to absorb only a specific amount of input energy. This means that, for any given geometric size of automotive shock absorber, it will have a limited amount of absorption capability compared to the industrial type.

This is explained by observing the structural design of the automotive type and the lower strength of materials commonly used. These materials can withstand the lower pressures commonly found in this type. The industrial shock absorber uses higher strength materials, enabling it to function at higher damping forces.

Adjustment Techniques

A properly adjusted shock absorber safely dissipates energy, reducing damaging shock loads and noise levels. For optimum adjustment setting see useable adjustment setting graphs. Watching and "listening" to a shock absorber as it functions aids in proper adjustment.

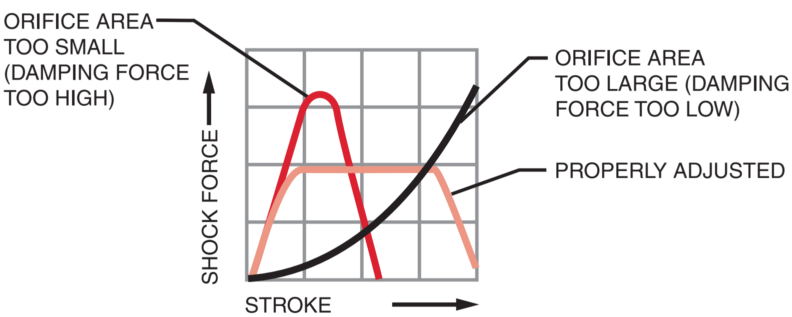
To correctly adjust a shock absorber, set the adjustment knob at zero (0) prior to system engagement. Cycle the mechanism and observe deceleration of the system.

If damping appears too soft (unit strokes with no visual deceleration and bangs at end of stroke), move indicator to next largest number. Adjustments must be made in gradual increments to avoid internal damage to the unit (e.g., adjust from 0 to 1, not 0 to 4).

Increase adjustment setting until smooth deceleration or control is achieved and negligible noise is heard when the system starts either to decelerate or comes to rest.

When abrupt deceleration occurs at the beginning of the stroke (banging at impact), the adjustment setting must be moved to a lower number to allow smooth deceleration.

If the shock absorber adjustment knob is set at the high end of the adjustment scale and abrupt deceleration occurs at the end of the stroke, a larger unit may be required.



Shock Absorber Performance When Weight or Impact Velocity Vary

When conditions change from the original calculated data or actual input, a shock absorber's performance can be greatly affected, causing failure or degradation of performance. Variations in input conditions after a shock absorber has been installed can cause internal damage, or at the very least, can result in unwanted damping performance. Variations in weight or impact velocity can be seen by examining the following energy curves:

Varying Impact Weight: Increasing the impact weight (impact velocity remains unchanged), without reorificing or readjustment will result in increased damping force at the end of the stroke. Figure 1 depicts this undesirable bottoming peak force. This force is then transferred to the mounting structure and impacting load.

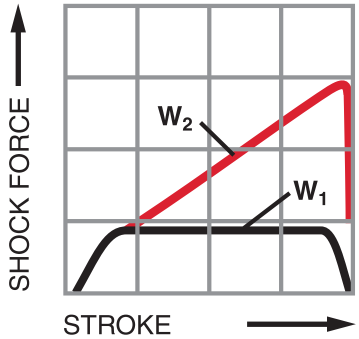


Figure 1

Varying Impact Velocity: Increasing impact velocity (weight remains the same) results in a radical change in the resultant shock force. Shock absorbers are velocity conscious products; therefore, the critical relationship to impact velocity must be carefully monitored. Figure 2 depicts the substantial change in shock force that occurs when the velocity is increased. Variations from original design data or errors in original data may cause damage to mounting structures and systems, or result in shock absorber failure if the shock force limits are exceeded.

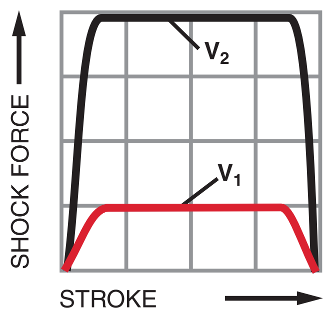


Figure 2

SHOCK ABSORBER SIZING

Follow the next six steps to manually size Enidine shock absorbers:

STEP 1: Identify the following parameters. These must be known for all energy absorption calculations. Variations or additional information may be required in some cases.

A. Weight of the load to be stopped (lbs.)(Kg).
B. Velocity of the load upon impact with the shock absorber (in./sec.)(m/s).
C. External (propelling) forces acting on the load (lbs.)(N), if any.
D. Cyclic frequency at which the shock absorber will operate.
E. Orientation of the application's motion (i.e. horizontal, vertical up, vertical down, inclined, rotary horizontal, rotary vertical up, rotary vertical down).

NOTE: For rotary applications, it is necessary to determine both the radius of gyration (K) and the mass moment of inertia (I). Both of these terms locate the mass of a rotating object with respect to the pivot point. It is also necessary to determine the angular velocity (ω) and the torque (T).

STEP 2: Calculate the kinetic energy of the moving object.

$$E_K = \frac{W}{772} \times V^2 \text{ (linear) or } E_K = \frac{I}{2}\omega^2 \text{ (rotary) or } E_K = \frac{1}{2}MV^2 \text{ (metric)}$$

(Note: 772 = 2 x acceleration due to gravity)

Utilizing the Product Locators for Shock Absorbers located at the beginning of each product family section, select a model, either adjustable or non-adjustable, with a greater energy per cycle capacity than the value just calculated.

STEP 3: Calculate the work energy input from any external (propelling) forces acting on the load, using the stroke of the model selected in Step 2.

$$E_W = F_D \times S \text{ (linear) or } E_W = \frac{T}{R_S} \times S \text{ (rotary)}$$

Caution: The propelling force must not exceed the maximum propelling force listed for the model chosen. If the propelling force is too high, select a larger model and recalculate the work energy.

STEP 4: Calculate the total energy per cycle $E_T = E_K + E_W$

The model selected must have at least this much energy capacity. If not, select a model with greater energy capacity and return to Step 3.

STEP 5: Calculate the energy that must be absorbed per hour. Even though the shock absorber can absorb the energy in a single impact, it may not be able to dissipate the heat generated if the cycle rate is too high.

$$E_TC = E_T \times C$$

The model selected must have an energy per hour capacity greater than this calculated figure. If it is not greater, there are two options:

1. Choose another model that has more energy per hour capacity (because of larger diameter or stroke). Keep in mind that if the stroke changes, you must return to Step 3.
2. Use an Air/Oil Tank. The increased surface area of the tank and piping will increase the energy per hour capacity by 20 percent.

STEP 6: If you have selected an HP, PM, SPM, TK, or PRO Series model, refer to the sizing graph(s) in the appropriate series section to determine the required damping constant. If the point cannot be found in the sizing graph, you must select a larger model or choose a different series. Note that if the stroke changes, you must return to Step 3.

If you have selected an adjustable model (OEM, HP or HDA series), refer to the Useable Adjustment Setting Range graph for the chosen model. The impact velocity must fall within the limits shown on the graph.

RATE CONTROL SIZING

Follow the next five steps to manually size ITT Enidine rate controls:

STEP 1: Identify the following parameters. These must be known for all rate control calculations. Variations or additional information may be required in some cases.

A. Weight of the load to be controlled (lbs.)(Kg)
B. Desired velocity of the load (in/sec.)(m/s)
C. External (propelling) force acting on the load (lbs.)(N), if any.
D. Cyclic frequency at which the rate control will operate.
E. Orientation of the application's motion (i.e. horizontal, vertical up, vertical down, inclined, rotary horizontal, rotary vertical up, rotary vertical down.)
F. Damping direction (i.e., tension [T], compression [C] or both [T and C].
G. Required stroke (in.)(mm)

STEP 2: Calculate the propelling force at the rate control in each direction damping is required. (See sizing examples on page 6-12).

CAUTION: The propelling force in each direction must not exceed the maximum propelling force listed for the chosen model. If the propelling force is too high, select a larger model.

STEP 3: Calculate the total energy per cycle

$$E_T = E_W \text{ (tension)} + E_W \text{ (compression)}$$
$$E_W = F_D \times S$$

STEP 4: Calculate the total energy per hour

$$E_TC = E_T \times C$$

The model selected must have an energy per hour capacity greater than this calculated figure. If not, choose a model with a higher energy per hour capacity.

Compare the damping direction, stroke, propelling force, and total energy per hour to the values listed in the Rate Controls Engineering Data Charts (pages 97-106).

STEP 5: If you have selected a rate control, refer to the sizing graphs in the Rate Controls section to determine the required damping constant.

If you have selected an adjustable model (ADA), refer to the Useable Adjustment Setting Range graph for the chosen model. The desired velocity must fall within the limits shown on the graph.

www.enidine.com Email: industrialsales@enidine.com Tel.: 1-800-852-8508 Fax: 1-716-662-0406

SYMBOLS

- a = Acceleration (in./sec.^{2})(m/s^2)
- A = Width (in.)(m)
- B = Thickness (in.)(m)
- C = Number of cycles per hour
- d = Cylinder bore diameter (in.)(mm)
- D = Distance (in.)(m)
- E_K = Kinetic energy (in-lbs.)(Nm)
- E_T = Total energy per cycle (in-lbs./c)(Nm/c), $E_K + E_W$
- E_TC = Total energy to be absorbed per hour (in-lbs./hr)(Nm/hr)
- E_W = Work or drive energy (in-lbs.)(Nm)
- F_D = Propelling force (lbs.)(N)
- F_P = Shock force (lbs.)(N)
- H = Height (in.)(m)
- Hp = Motor rating (hp)(kw)
- I = Mass moment of inertia (in-lbs./sec^2)(Kgm^2)
- K = Radius of gyration (in.)(m)
- L = Length (in.)(m)
- P = Operating pressure (psi)(bar)
- R_S = Mounting distance from pivot point (in.)(m)
- S = Stroke of shock absorber (in.)(m)
- t = Time (sec.)
- T = Torque (in-lbs.)(Nm)
- V = Impact velocity (in./sec.)(m/s)
- W = Weight (lbs.)(Kg)
- α = Angle of incline (degrees)
- θ = Start point from true vertical 0° (degrees)
- μ = Coefficient of friction
- Ø = Angle of rotation (degrees)
- ω = Angular velocity (radians/sec)

USEFUL FORMULAS

1. To Determine Shock Force

$$F_P = \frac{E_T}{S \times \mathbf{.85}}$$

For PRO and PM Series only, use

$$F_P = \frac{E_T}{S \times \mathbf{.50}}$$

2. To Determine Impact Velocity

A. If there is no acceleration (V is constant) (e.g., load being pushed by hydraulic cylinder or motor driven.) $V = \frac{D}{t}$

B. If there is acceleration. (e.g., load being pushed by air cylinder) $V = \frac{\mathbf{2} \times D}{t}$

3. To Determine Propelling Force Generated by Electric Motor

$F_D = \frac{\mathbf{19{,}800} \times Hp}{V}$ $F_D = \frac{\mathbf{3\ 000} \times Hp}{V}$ (metric)

4. To Determine Propelling Force of Pneumatic or Hydraulic Cylinders

$F_D = \mathbf{.7854} \times d^2 \times P$ $F_D = \mathbf{0{,}07854} \times d^2 \times P$ (metric)

5. Free Fall Applications

A. Find Velocity for a Free Falling Weight:
$V = \sqrt{\mathbf{772} \times H}$ $V = \sqrt{19{,}6 \times H}$ (metric)

B. Kinetic Energy of Free Falling Weight:
$E_K = W \times H$

6. Deceleration and G Load

A. To Determine Approximate G Load with a Given Stroke

$G = \frac{F_P - F_D}{W}$ $G = \frac{F_P - F_D}{kg \times 9{,}81}$ (metric)

B. To Determine the Approximate Stroke with a Given G Load (Conventional Damping Only)

$$S = \frac{E_K}{GW\ .85 - .15\ F_D}$$

*For PRO/PM and TK Models:

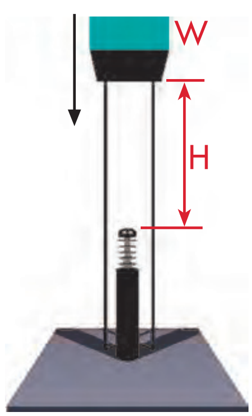
$$S = \frac{E_K}{GW\ .5 - .5\ F_D}$$

NOTE: Constants are printed in **bold**.

The following examples are shown using Imperial formulas and units of measure.

Shock Absorbers

EXAMPLE 1:
Vertical Free Falling Weight



STEP 1: Application Data
(W) Weight = 3,400 lbs.
(H) Height = 20 in.
(C) Cycles/Hr = 2

STEP 2: Calculate kinetic energy
$E_K = W \times H$
$E_K = 3{,}400 \times 20 = 68{,}000$ in-lbs.

Assume Model OEM 4.0M x 6 is adequate (Page 31).

STEP 3: Calculate work energy
$E_W = W \times S$
$E_W = 3{,}400 \times 6$
$E_W = 20{,}400$ in-lbs.

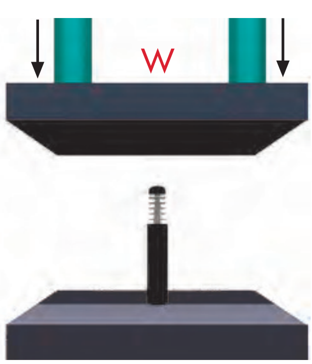
STEP 4: Calculate total energy per cycle
$E_T = E_K + E_W$
$E_T = 68{,}000 + 20{,}400$
$E_T = 88{,}400$ in-lbs./c

STEP 5: Calculate total energy per hour
$E_TC = E_T \times C$
$E_TC = 88{,}400 \times 2$
$E_TC = 176{,}800$ in-lbs./hr

STEP 6: Calculate impact velocity and confirm selection
$V = \sqrt{\mathbf{772} \times H}$
$V = \sqrt{\mathbf{772} \times 20}$
$V = 124$ in./sec.

Model OEM 4.0M x 6 is adequate.

EXAMPLE 2:
Vertical Moving Load with Propelling Force Downward



STEP 1: Application Data
(W) Weight = 3,400 lbs.
(V) Velocity = 80 in./sec.
(d) Cylinder bore dia. = 4 in.
(P) Pressure = 70 psi
(C) Cycles/Hr = 200

STEP 2: Calculate kinetic energy
$E_K = \frac{W}{\mathbf{772}} \times V^2 = \frac{3{,}400}{\mathbf{772}} \times 80^2$
$E_K = 28{,}187$ in-lbs.

Assume Model OEM 4.0M x 4 is adequate (Page 31).

STEP 3: Calculate work energy
$F_D = [\mathbf{.7854} \times d^2 \times P] + W$
$F_D = [\mathbf{.7854} \times 4^2 \times 70] + 3{,}400$
$F_D = 4{,}280$ lbs.
$E_W = F_D \times S$
$E_W = 4{,}280 \times 4$
$E_W = 17{,}120$ in-lbs.

STEP 4: Calculate total energy per cycle
$E_T = E_K + E_W$
$E_T = 28{,}187 + 17{,}120$
$E_T = 45{,}307$ in-lbs./c

STEP 5: Calculate total energy per hour
$E_TC = E_T \times C$
$E_TC = 45{,}307 \times 200$
$E_TC = 9{,}061{,}400$ in-lbs./hr

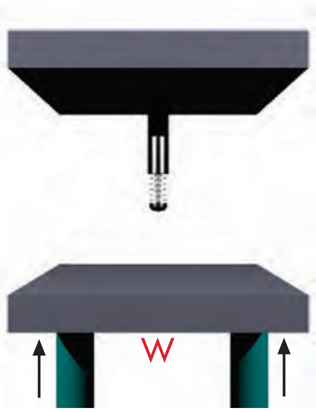
Model OEM 4.0M x 4 is adequate.

Shock Absorber Sizing Examples

Typical Shock Absorber Applications

EXAMPLE 3:

Vertical Moving Load with Propelling Force Upward



STEP 1: Application Data

(W) Weight = 3,400 lbs.
(V) Velocity = 80 in./sec.
(d) 2 Cylinders bore dia. = 6 in.
(P) Operating pressure = 70 psi
(C) Cycles/Hr = 200

STEP 2: Calculate kinetic energy

$E_K = \frac{W}{\mathbf{772}} \times V^2 = \frac{3{,}400}{\mathbf{772}} \times 80^2$

$E_K = 28{,}187$ in-lbs.

Assume Model OEM 3.0M x 5 is adequate (Page 31).

STEP 3: Calculate work energy

$F_D = 2 \times [\mathbf{.7854} \times d^2 \times P] - W$

$F_D = 2 \times [\mathbf{.7854} \times 6^2 \times 70] - 3{,}400$

$F_D = 558$ lbs.

$E_W = F_D \text{ X } S$

$E_W = 558 \times 5$

$E_W = 2{,}790$ in-lbs.

STEP 4: Calculate total energy per cycle

$E_T = E_K + E_W$

$E_T = 28{,}187 + 2{,}790$

$E_T = 30{,}977$ in-lbs./c

STEP 5: Calculate total energy per hour

$E_TC = E_T \times C$

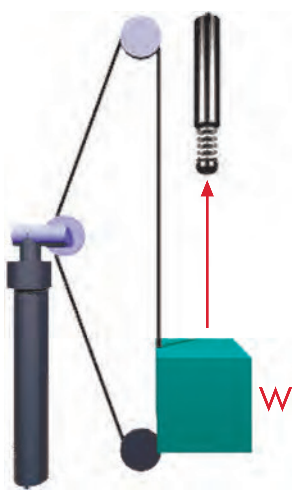
$E_TC = 30{,}977 \times 200$

$E_TC = 6{,}195{,}400$ in-lbs./hr

Model OEM 3.0M x 5 is adequate.

EXAMPLE 4:

Vertical Moving Load with Propelling Force from Motor



(e.g., Load Moving Force Up)

STEP 1: Application Data

(W) Weight = 200 lbs.
(V) Velocity = 60 in./sec.
(Hp) Motor horsepower = 1.5 Hp
(C) Cycles/Hr = 100

STEP 2: Calculate kinetic energy

$E_K = \frac{W}{\mathbf{772}} \times V^2 = \frac{200}{\mathbf{772}} \times 60^2$

$E_K = 933$ in-lbs.

CASE A: UP

STEP 3: Calculate work energy

$F_D = \frac{\mathbf{19{,}800} \times Hp}{V} - W$

$F_D = \frac{\mathbf{19{,}800} \times 1.5}{60} - 200$

$F_D = 295$ lbs.

Assume Model OEM 1.25 x 2 is adequate (Page 24).

$E_W = F_D \text{ X } S$

$E_W = 295 \times 2$

$E_W = 590$ in-lbs.

STEP 4: Calculate total energy per cycle

$E_T = E_K + E_W$

$E_T = 933 + 590$

$E_T = 1{,}523$ in-lbs./c

STEP 5: Calculate total energy per hour

$E_TC = E_T \times C$

$E_TC = 1{,}523 \times 100$

$E_TC = 152{,}300$ in-lbs./hr

Model OEM 1.25 x 2 is adequate.

CASE B: DOWN

STEP 3: Calculate work energy

$F_D = \frac{\mathbf{19{,}800} \times Hp}{V} + W$

$F_D = \frac{\mathbf{19{,}800} \times 1.5}{60} + 200$

$F_D = 695$ lbs.

Assume Model OEMXT 2.0M x 2 is adequate (Page 29).

$E_W = F_D \times S$

$E_W = 695 \times 2$

$E_W = 1{,}390$ in-lbs.

STEP 4: Calculate total energy per cycle

$E_T = E_K + E_W$

$E_T = 933 + 1{,}390$

$E_T = 2{,}323$ in-lbs./c

STEP 5: Calculate total energy per hour

$E_TC = E_T \times C$

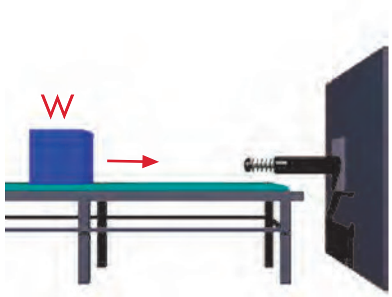
$E_TC = 2{,}323 \times 100$

$E_TC = 232{,}300$ in-lbs./hr

Model OEMXT 2.0M x 2 is adequate.

EXAMPLE 5:

Horizontal Moving Load



STEP 1: Application Data

(W) Weight = 1,950 lbs.
(V) Velocity = 60 in./sec.
(C) Cycles/Hr = 200

STEP 2: Calculate kinetic energy

$E_K = \frac{W}{\mathbf{772}} \times V^2$

$E_K = \frac{1950}{\mathbf{772}} \times 60^2$

$E_K = 9{,}093$ in-lbs.

Assume Model OEMXT 2.0M x 2 is adequate (Page 29).

STEP 3: Calculate work energy: N/A

STEP 4: Calculate total energy per cycle

$E_T = E_K = 9{,}093$ in-lbs./c

STEP 5: Calculate total energy per hour

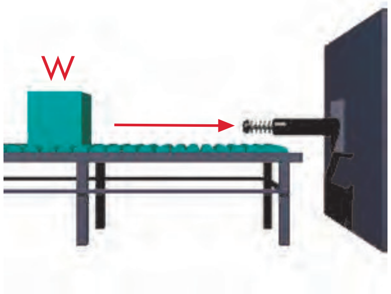
$E_TC = E_T \times C$

$E_TC = 9{,}093 \times 200$

$E_TC = 1{,}818{,}600$ in-lbs./hr

Model OEMXT 2.0M x 2 is adequate.

EXAMPLE 6:
Horizontal Moving Load with Propelling Force



STEP 1: Application Data
(W) Weight = 1,950 lbs.
(V) Velocity = 60 in./sec.
(d) Cylinder bore dia. = 3 in.
(P) Operating pressure = 70 psi
(C) Cycles/Hr = 200

STEP 2: Calculate kinetic energy

$E_K = \frac{W}{\mathbf{772}} \times V^2$

$E_K = \frac{1{,}950}{\mathbf{772}} \times 60^2$

$E_K = 9{,}093$ in-lbs.

Assume Model OEMXT 2.0M x 2 is adequate (Page 29).

STEP 3: Calculate work energy

$F_D = \mathbf{.7854} \times d^2 \times P$

$F_D = \mathbf{.7854} \times 3^2 \times 70$

$F_D = 495$ lbs.

$E_W = F_D \times S$

$E_W = 495 \times 2$

$E_W = 990$ in-lbs.

STEP 4: Calculate total energy per cycle

$E_T = E_K + E_W$

$E_T = 9{,}093 + 990$

$E_T = 10{,}083$ in-lbs./c

STEP 5: Calculate total energy per hour

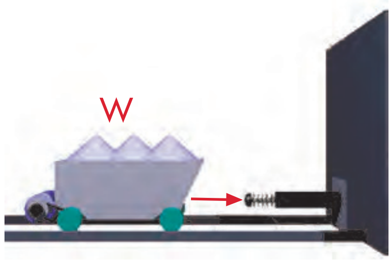
$E_TC = E_T \times C$

$E_TC = 10{,}083 \times 200$

$E_TC = 2{,}016{,}600$ in-lbs./hr

Model OEMXT 2.0M x 2 is adequate.

EXAMPLE 7:
Horizontal Moving Load, Motor Driven



STEP 1: Application Data
(W) Weight = 2,200 lbs.
(V) Velocity = 60 in./sec.
(Hp) Motor horsepower = 1.5 Hp
(C) Cycles/Hr = 120

STEP 2: Calculate kinetic energy

$E_K = \frac{W}{\mathbf{772}} \times V^2$

$E_K = \frac{2{,}200}{\mathbf{772}} \times 60^2$

$E_K = 10{,}259$ in-lbs

Assume Model OEMXT 2.0M x 2 is adequate (Page 29).

STEP 3: Calculate work energy

$F_D = \frac{\mathbf{19{,}800} \times Hp}{V}$

$F_D = \frac{\mathbf{19{,}800} \times 1.5}{60}$

$F_D = 495$ lbs.

$E_W = F_D \times S$

$E_W = 495 \times 2$

$E_W = 990$ in-lbs.

STEP 4: Calculate total energy per cycle

$E_T = E_K + E_W$

$E_T = 10{,}259 + 990$

$E_T = 11{,}249$ in-lbs./c

STEP 5: Calculate total energy per hour

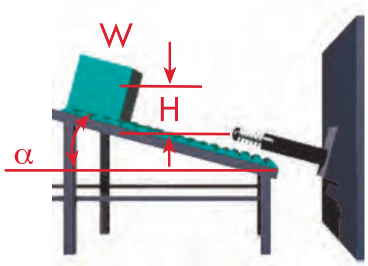
$E_TC = E_T \times C$

$E_TC = 11{,}249 \times 120$

$E_TC = 1{,}349{,}880$ in-lbs./hr

Model OEMXT 2.0M x 2 is adequate.

EXAMPLE 8:
Free Moving Load Down an Inclined Plane



STEP 1: Application Data
(W) Weight = 550 lbs.
(H) Height = 8 in.
(α) Angle of incline = 30°
(C) Cycles/Hr = 250

STEP 2: Calculate kinetic energy

$E_K = W \times H$

$E_K = 550 \times 8$

$E_K = 4{,}400$ in-lbs.

Assume Model OEMXT 1.5M x 3 is adequate (Page 27).

STEP 3: Calculate work energy

$F_D = W \times \sin \alpha$

$F_D = 550 \times .5$

$F_D = 275$ lbs.

$E_W = F_D \times S$

$E_W = 275 \times 3$

$E_W = 825$ in-lbs.

STEP 4: Calculate total energy per cycle

$E_T = E_K + E_W$

$E_T = 4{,}400 + 825$

$E_T = 5{,}225$ in-lbs./c

STEP 5: Calculate total energy per hour

$E_TC = E_T \times C$

$E_TC = 5{,}225 \times 250$

$E_TC = 1{,}306{,}250$ in-lbs./hr

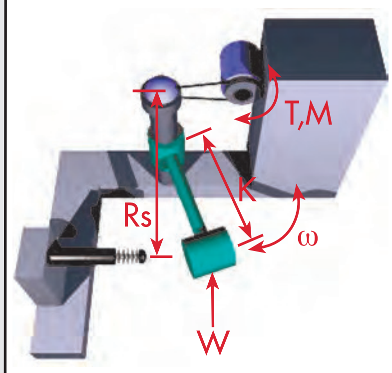
STEP 6: Calculate impact velocity and confirm selection

$V = \sqrt{\mathbf{772} \times H}$

$V = \sqrt{\mathbf{772} \times 8} = 79$ in./sec.

Model OEMXT 1.5M x 3 is adequate.

EXAMPLE 9:
Horizontal Rotating Mass



STEP 1: Application Data
(W) Weight = 200 lbs.
(ω) Angular velocity = 1.5 rad./sec.
(T) Torque = 1,065 in-lbs.
(K) Radius of gyration = 15 in.
(R_S) Mounting radius = 20 in.
(C) Cycles/Hr = 120

STEP 2: Calculate kinetic energy

$I = \frac{W}{\mathbf{386}} \times K^2$

$I = \frac{200}{\mathbf{386}} \times 15^2$

$I = 117$ in-lbs./sec.^{2}

$E_K = \frac{I \times \omega^2}{\mathbf{2}}$

$E_K = \frac{117 \times 1.5^2}{\mathbf{2}}$

$E_K = 132$ in-lbs.

Assume Model STH .5M is adequate (Page 40).

STEP 3 Calculate work energy

$F_D = \frac{T}{R_S}$

$F_D = \frac{1{,}065}{20}$

$F_D = 53$ lbs.

$E_W = F_D \text{ X } S$

$E_W = 53 \text{ X } .5$

$E_W = 27$ in-lbs.

STEP 4: Calculate total energy per cycle

$E_T = E_K + E_W$

$E_T = 132 + 27$

$E_T = 159$ in-lbs./c

STEP 5: Calculate total energy per hour

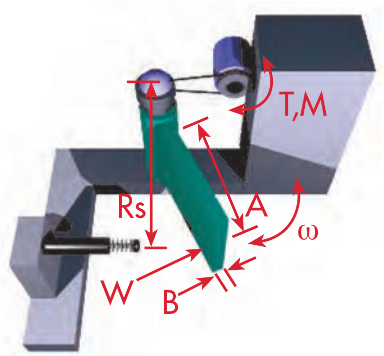
$E_TC = E_T \text{ X } C$

$E_TC = 159 \text{ X } 120$

$E_TC = 19{,}080$ in-lbs./hr

Model STH .5M is adequate.

EXAMPLE 10:
Horizontal Rotating Door



STEP 1: Application Data
(W) Weight = 50 lbs.
(ω) Angular velocity = 2.5 rad./sec.
(T) Torque = 100 in-lbs.
(R_S) Mounting radius = 20 in.
(A) Width = 40 in.
(B) Thickness = .5 in.
(C) Cycles/Hr = 250

STEP 2: Calculate kinetic energy

$K = .289 \times \sqrt{\mathbf{4} \times A^2 + B^2}$

$K = .289 \times \sqrt{\mathbf{4} \times 40^2 + .5^2}$

$K = 23.12$

$I = \frac{W}{\mathbf{386}} \times K^2$

$I = \frac{50}{\mathbf{386}} \times 23.12^2$

$I = 69$ in-lbs./sec.^{2}

$E_K = \frac{I \times \omega^2}{\mathbf{2}}$

$E_K = \frac{69 \times 2.5^2}{\mathbf{2}}$

$E_K = 216$ in-lbs.

Assume Model OEM .5 is adequate (Page 21).

STEP 3: Calculate work energy

$F_D = \frac{T}{R_S}$

$F_D = \frac{100}{20}$

$F_D = 5$ lbs.

$E_W = F_D \times S = 5 \times .5 = 2.5$ in-lbs.

STEP 4: Calculate total energy per cycle

$E_T = E_K + E_W = 216 + 2.5 = 218.5$ in-lbs./c

STEP 5: Calculate total energy per hour

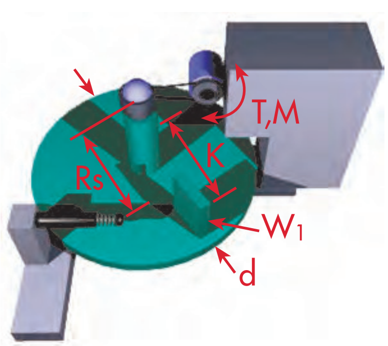
$E_TC = E_T \times C = 218.5 \times 250 = 54{,}625$ in-lbs./hr

STEP 6: Calculate impact velocity and confirm selection

$V = R_S \times \omega = 20 \times 2.5 = 50$ in./sec.

Model OEM .5 is adequate.

EXAMPLE 11:
Horizontal Moving Load, Rotary Table Motor Driven with Additional Load Installed



STEP 1: Application Data
(W) Weight = 440 lbs.
(W_1) Installed load = 110 lbs.
Rotational speed = 10 RPM
(T) Torque = 2,200 in-lbs.
Rotary table dia. = 20 in.
(K_{Load}) Radius of gyration = 8 in.
(R_S) Mounting radius = 8.86 in.
(C) Cycles/Hr = 1
(ω) Direction

Step 2: Calculate kinetic energy
To convert RPM to rad./sec., multiply by **.1047**

$\omega = \text{RPM} \times \mathbf{.1047} = 10 \times \mathbf{.1047} = \mathbf{1.047}$ rad./sec.

$I = \frac{W}{\mathbf{386}} \times K^2$

In this case, the mass moment of inertia of the table and the mass moment of inertia of the load on the table must be calculated.

K_{Table} = Table Radius x **.707**

K_{Table} = 10 x **.707** = 7.07 in.

$I_{Table} = \frac{W}{\mathbf{386}} \times K^2_{Table}$

$I_{Table} = \frac{440}{\mathbf{386}} \times 7.07^2 = 57$ in-lbs./sec.^{2}

$I_{Load} = \frac{W_1}{\mathbf{386}} \times K^2_{Load}$

$I_{Load} = \frac{110}{\mathbf{386}} \times 8^2 = 18$ in-lbs./sec.^{2}

$E_K = \frac{(I_{Table} + I_{Load}) \times \omega^2}{\mathbf{2}}$

$E_K = \frac{(57 + 18) \times 1.047^2}{\mathbf{2}} = 41$ in-lbs.

Assume Model ECO 50 is adequate (Page 46).

STEP 3: Calculate work energy

$F_D = \frac{T}{R_S} = \frac{2{,}200}{8.86} = 248$ lbs.

$E_W = F_D \times S = 248 \times .875 = 217$ in-lbs.

STEP 4: Calculate total energy per cycle

$E_T = E_K + E_W = 41 + 217 = 258$ in-lbs./c

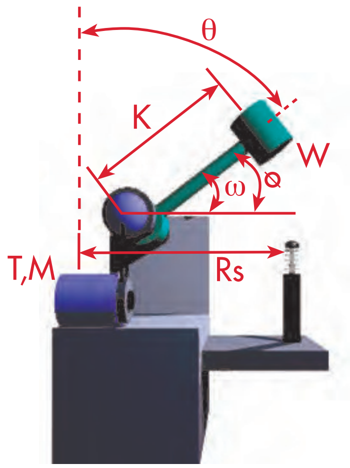
STEP 5: Calculate total energy per hour: not applicable, C = 1

STEP 6: Calculate impact velocity and confirm selection

$V = R_S \times \omega = 8.86 \times 1.047 = 9$ in./sec.

From ECO Sizing Graph.
Model ECO 50 is adequate.

EXAMPLE 12:
Vertical Motor Driven Rotating Arm with Attached Load
CASE A–Load Aided by Gravity



STEP 1: Application Data
(W) Weight = 110 lbs.
(ω) Angular velocity = 2 rad./sec.
(T) Torque = 3,100 in-lbs.
(θ) Starting point of load from true vertical = 20°
(Ø) Angle of rotation at impact = 30°
(K_{Load}) Radius of gyration = 24 in.
(R_S) Mounting radius = 16 in.
(C) Cycles/Hr = 1

STEP 2: Calculate kinetic energy

$I = \frac{W}{386} \times K^2 = \frac{110}{386} \times 24^2$

$I = 164$ in-lbs-sec^2

$E_K = \frac{I \times \omega^2}{2}$

$E_K = \frac{164 \times 2^2}{2}$

$E_K = 328$ in-lbs.

Assume Model OEM 1.0 is adequate (Page 21).

CASE A
STEP 3: Calculate work energy

$F_D = \frac{[T + (W \times K \times \text{Sin}(\theta + Ø))]}{R_S}$

$F_D = \frac{[3{,}100 + (110 \times 24 \times .77)]}{16}$

$F_D = 320.8$ lbs.

$E_W = F_D \times S = 320.8 \times 1 = 320.8$ in-lbs.

STEP 4: Calculate total energy per cycle

$E_T = E_K + E_W = 328 + 320.8$

$E_T = 648.8$ in-lbs./c

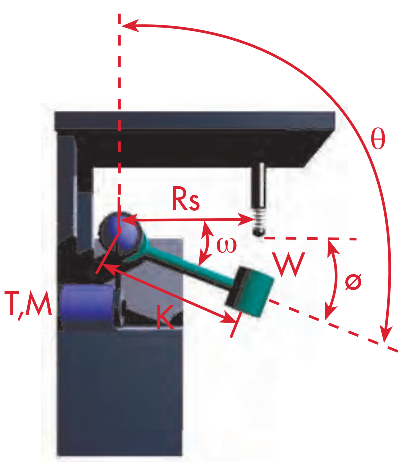
STEP 5: Calculate total energy per hour: not applicable, C = 1

STEP 6: Calculate impact velocity and confirm selection

$V = R_S \times \omega = 16 \times 2 = 32$ in./sec.

Model LROEM 1.0 is adequate. Needed for higher calculated propelling force.

EXAMPLE 13:
Vertical Motor Driven Rotating Arm with Attached Load
CASE B–Load Opposing Gravity



STEP 1: Application Data
(W) Weight = 110 lbs.
(ω) Angular velocity = 2 rad./sec.
(T) Torque = 3,100 in-lbs.
(θ) Starting point of load from true vertical = 30°
(Ø) Angle of rotation at impact = 150°
(K_{Load}) Radius of gyration = 24 in.
(R_S) Mounting radius = 16 in.
(C) Cycles/Hr = 1

STEP 2: Calculate kinetic energy

$I = \frac{W}{386} \times K^2 = \frac{110}{386} \times 24^2$

$I = 164$ in-lbs-sec^2

$E_K = \frac{I \times \omega^2}{2}$

$E_K = \frac{164 \times 2^2}{2}$

$E_K = 328$ in-lbs.

Assume Model OEM 1.0 is adequate (Page 21).

CASE B
STEP 3: Calculate work energy

$F_D = \frac{[T - (W \times K \times \text{Sin}(\theta - Ø))]}{R_S}$

$F_D = \frac{[3{,}100 - (110 \times 24 \times .77)]}{16}$

$E_W = F_D \times S = 67 \times 1 = 67$ in-lbs.

STEP 4: Calculate total energy per cycle

$E_T = E_K + E_W = 328 + 67$

$E_T = 394.7$ in-lbs./c

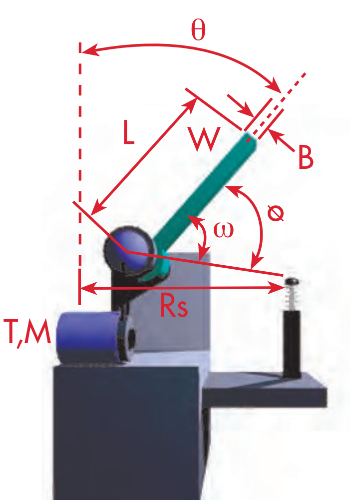
STEP 5: Calculate total energy per hour: not applicable, C = 1

STEP 6: Calculate impact velocityand confirm selection.

$V = R_S \times \omega = 16 \times 2 = 32$ in./sec.

Model OEM 1.0 is adequate**.**

EXAMPLE 14:
Vertical Rotating Beam



STEP 1: Application Data
(W) Weight = 540 lbs.
(ω) Angular velocity = 3.5 rad./sec.
(T) Torque = 250 in-lbs.
(θ) Starting point of load from true vertical = 20°
(Ø) Angle of rotation at impact = 50°
(R_S) Mounting radius = 20 in.
(B) Thickness = 2.5 in.
(L) Length = 24 in.
(C) Cycles/Hr = 1

STEP 2: Calculate kinetic energy

$K = .289 \times \sqrt{4 \times L^2 + B^2}$

$K = .289 \times \sqrt{4 \times 24^2 + 2.5^2} = 13.89$

$I = \frac{W}{386} \times K^2 = \frac{540}{386} \times 13.89$

$I = 270$ in-lbs./sec.^{2}

$E_K = \frac{I \times \omega^2}{2} = \frac{270 \times 3.5^2}{2} = 1{,}653$ in-lbs.

Assume Model OEM 1.5M x 2 is adequate (Page 27).

STEP 3: Calculate work energy

$F_D = \frac{T + (W \times K \times \text{Sin}(\theta + Ø))}{R_S}$

$F_D = \frac{250 + (540 \times 13.89 \times \text{Sin}(20° + 50°))}{20}$

$F_D = 365$ lbs.

$E_W = F_D \times S = 365 \times 2 = 730$ in-lbs.

STEP 4: Calculate total energy per cycle

$E_T = E_K + E_W = 1{,}653 + 730 = 2{,}383$ in-lbs./c

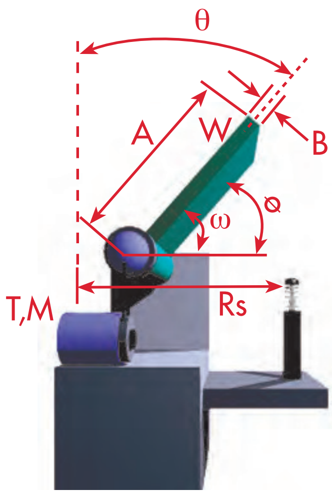
STEP 5: Calculate total energy per hour: not applicable, C = 1

STEP 6: Calculate impact velocity and confirm selection

$V = R_S \times \omega = 20 \times 3.5 = 70$ in./sec.

Model OEM 1.5M x 2 is adequate.

EXAMPLE 15:
Vertical Rotating Lid



STEP 1: Application Data

(W) Weight = 2,000 lbs.
(ω) Angular velocity = 2 rad./sec.
(Hp) Motor horsepower = .25 Hp
(θ) Starting point of load from true vertical = 30°
(Ø) Angle of rotation at impact = 60°
(R_S) Mounting radius = 30 in.
(A) Width = 60 in.
(B) Thickness = 1 in.
(C) Cycle/Hr = 1

STEP 2: Calculate kinetic energy

$K = .289 \times \sqrt{4} \times A^2 + B^2$

$K = .289 \times \sqrt{4} \times 60^2 + 1^2 = 34.68$ in.

$I = \frac{W}{386} \times K^2 = \frac{2{,}000}{386} \times 34.68^2$ in.

$I = 6{,}232$ in-lbs./sec.^{2}

$E_K = \frac{I \times \omega^2}{2} = \frac{6{,}232 \times 2^2}{2}$

$E_K = 12{,}464$ in-lbs.

Assume Model OEM 3.0M x 2 is adequate (Page 31).

STEP 3: Calculate work energy

$T = \frac{19{,}800 \times Hp}{\omega}$

$T = \frac{19{,}800 \times .25}{2} = 2{,}475$ in-lbs.

$F_D = \frac{T + (W \times K \times \sin(\theta + Ø))}{R_S}$

$F_D = \frac{2{,}475 + (2{,}000 \times 34.68 \times \sin(30° + 60°))}{30}$

$F_D = 2{,}395$ lbs.

$E_W = F_D \times S = 2{,}395 \times 2 = 4{,}790$ in-lbs.

STEP 4: Calculate total energy per cycle

$E_T = E_K + E_W = 12{,}464 + 4{,}790$

$= 17{,}254$ in-lbs./c

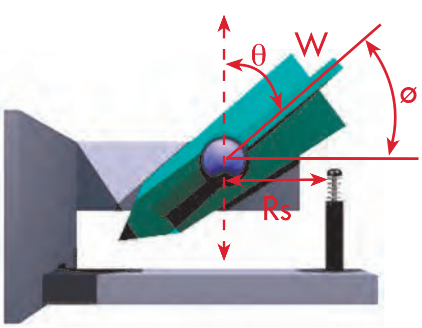
STEP 5: Calculate total energy per hour: not applicable, C = 1

STEP 6: Calculate impact velocity and confirm selection

$V = R_S \times \omega = 30 \times 2 = 60$ in./sec.

Model OEM 3.0M x 2 is adequate.

EXAMPLE 16:
Vertical Rotation with Known Intertia Aided by Gravity



STEP 1: Application Data

(W) Weight = 220.5 lbs
(I) Known Intertia = 885 in-lbs/sec.^{2}
(C/G) Center-of-Gravity = 12 in.
(θ) Starting point from true vertical = 60°
(Ø) Angle of rotation at impact = 30°
(R_S) Mounting radius = 10 in.
(C) Cycles/Hr = 1

STEP 2: Calculate kinetic energy

$H = C/G \times [\cos(\theta) - \cos(Ø+\theta)]$

$H = 12 \times [\cos(60°) - \cos(30°+60°)]$

$E_K = W \times H$

$E_K = 220.5 \times 6$

$E_K = 1{,}323$ in-lbs.

STEP 3: Calculate work energy

$F_D = (W \times C/G \times \sin(\theta + Ø))/R_S$

$F_D = (220.5 \times 12 \times \sin(60° + 30°))/10$

$F_D = 264.6$ lbs.

$E_W = F_D \times S = 264.6 \times 1 = 264.6$ in-lbs.

STEP 4: Calculate total energy per cycle

$E_T = E_K + E_W = 1{,}323 + 264.6$

$E_T = 1{,}587.6$ in-lbs/cyc.

STEP 5: Calculate total energy per hour: not applicable, C = 1

$E_TC = E_T \times C$

$E_TC = 1{,}587.6 \times 1$

$E_TC = 1{,}587.6$ in-lbs/hr.

STEP 6: Calculate impact velocity and confirm selection

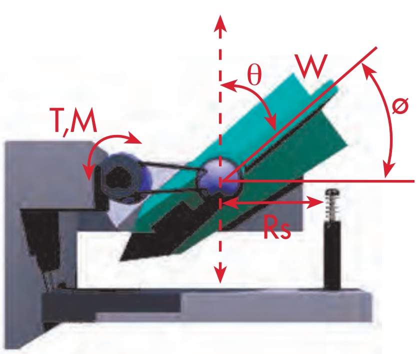
$\omega = ((2 \times E_K)/I)\ 0.5$

$\omega = ((2 \times 1{,}323)/885)\ 0.5 = 1.7$

$V = R_S \times \omega = 10 \times 1.7 = 17$ in./sec.

Model OEM 1.15 x 1 is adequate (Page 24).

EXAMPLE 17:
Vertical Rotation with Known Intertia Aided by Gravity (w/Torque)



STEP 1: Application Data

(W) Weight = 220.5 lbs
(ω) Angular Velocity = 2 rad/sec.
(T) Torque = 2,750 in-lbs.
(I) Known Intertia = 885 in-lbs/sec.^{2}
(C/G) Center-of-Gravity = 12 in.
(θ) Starting point from true vertical = 60°
(Ø) Angle of rotation at impact = 30°
(R_S) Mounting radius = 10 in.
(C) Cycles/Hr = 100

STEP 2: Calculate kinetic energy

$E_K = (I \times \omega^2)/2$

$E_K = (885 \times 2^2)/2$

$E_K = 1{,}770$ in-lbs.

STEP 3: Calculate work energy

$F_D = [T - (W \times C/G \times \sin(\theta + Ø)]/R_S$

$F_D = [2{,}750 - (220.5 \times 12 \times \sin(60°+30°)]/10$

$F_D = 539.6$ lbs.

$E_W = F_D \times S = 539.6 \times 1 = 539.6$ in-lbs.

STEP 4: Calculate total energy per cycle

$E_T = E_K + E_W = 1{,}770 + 539.6$

$E_T = 2{,}309.6$ in-lbs/cyc.

STEP 5: Calculate total energy per hour: not applicable, C = 1

$E_TC = E_T \times C$

$E_TC = 2{,}309.6 \times 1$

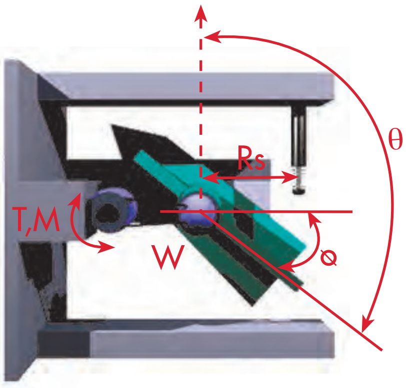
$E_TC = 230{,}960$ in-lbs/hr.

STEP 6: Calculate impact velocity and confirm selection

$V = R_S \times \omega = 10 \times 2 = 20$ in./sec.

Model OEM 1.15 x 1 is adequate (Page 24).

EXAMPLE 18:
Vertical Rotation with Known Intertia Aided by Gravity (w/Torque)



STEP 1: Application Data
(W) Weight = 220.5 lbs
(ω) Angular Velocity = 2 rad/sec.
(T) Torque = 2,750 in-lbs.
(I) Known Inertia = 885 in-lbs/sec.^{2}
(C/G) Center-of-Gravity = 12 in.
(θ) Starting point from true vertical = 120°
(Ø) Angle of rotation at impact = 30°
(R_S) Mounting radius = 10 in.
(C) Cycles/Hr = 100

STEP 2: Calculate kinetic energy
$E_K = (I \times \omega^2)/2$
$E_K = (885 \times 2^2)/2$
$E_K = 1{,}770$ in-lbs.

STEP 3: Calculate work energy
$F_D = [T - (W \times C/G \times \sin(\theta - Ø))]/R_S$
$F_D = [2{,}750 - (220.5 \times 12 \times \sin(120^\circ - 30^\circ))]/10$
$F_D = 10.4$ lbs.
$E_W = F_D \times S = 10.4 \times 1 = 10.4$ in-lbs.

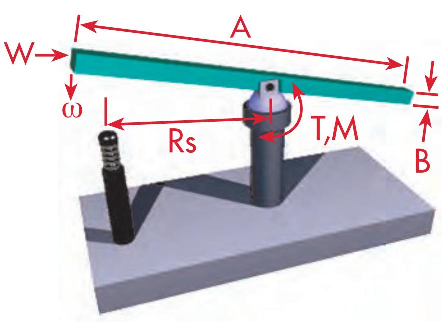
STEP 4: Calculate total energy per cycle
$E_T = E_K + E_W = 1{,}770 + 10.4$
$E_T = 1{,}780.4$ in-lbs/cyc.

STEP 5: Calculate total energy per hour: not applicable, C = 1
$E_TC = E_T \times C$
$E_TC = 1{,}780.4 \times 100$
$E_TC = 178{,}040$ in-lbs/hr.

STEP 6: Calculate impact velocity and confirm selection
$V = R_S \times \omega = 10 \times 2 = 20$ in./sec.

Model OEMXT 1.5M x 1 is adequate (Page 27).

EXAMPLE 19:
Vertical Roation Pinned at Center (w/Torque)



STEP 1: Application Data
(W) Weight = 220.5 lbs.
(ω) Angular velocity = 2 rad./sec.
(T) Torque = 2,750 in-lbs.
(A) Length = 40 in.
(R_S) Mounting radius = 10 in.
(B) Thickness = 2 in.
(C) Cycles/Hr = 100

STEP 2: Calculate kinetic energy
$K = .289 \times (A^2 + B^2)^{0.5}$
$K = .289 \times (40^2 + 2^2)^{0.5} = 11.6$ in.
$I = (W/386) \times K^2$
$I = (220.5/386) \times 11.6^2 = 76.9$ in-lb/sec^2
$E_K = (I \times \omega^2)/2$
$E_K = (76.9 \times 2^2)/2$
$E_K = 153.8$ in-lbs.

Assume Model OEM 1.0 is adequate (Page 21).

STEP 3: Calculate work energy
$F_D = T/R_S$
$F_D = 2{,}750/10$
$F_D = 275$ lbs.
$E_W = F_D \times S = 275 \times 1 = 275$ in-lbs.

STEP 4: Calculate total energy per cycle
$E_T = E_K + E_W = 153.8 + 275$
$E_T = 428.8$ in-lbs./cycle

STEP 5: Calculate total energy per hour
$E_TC = E_T \times C$
$E_TC = 428.8 \times 100$
$E_TC = 42{,}880$ in-lbs./hr.

STEP 6: Calculate impact velocity and confirm selection
$V = R_S \times \omega = 10 \times 2 = 20$ in./sec.

Model OEM 1.0 is adequate.

Shock Absorber Sizing Examples
Typical Shock Absorber and Crane Applications

Overview

Calculaions assume worst case scenario of 90% trolley weight over one rail.

Crane A		Per Buffer
Propelling Force Crane	lbs.	
Propelling Force Trolley	lbs.	
Weight of Crane (W_a)	lbs.	
Weight of Trolley (W_{ta})	lbs.	
Crane Velocity (V_a)	in./sec.	
Trolley Velocity (V_{ta})	in./sec.	

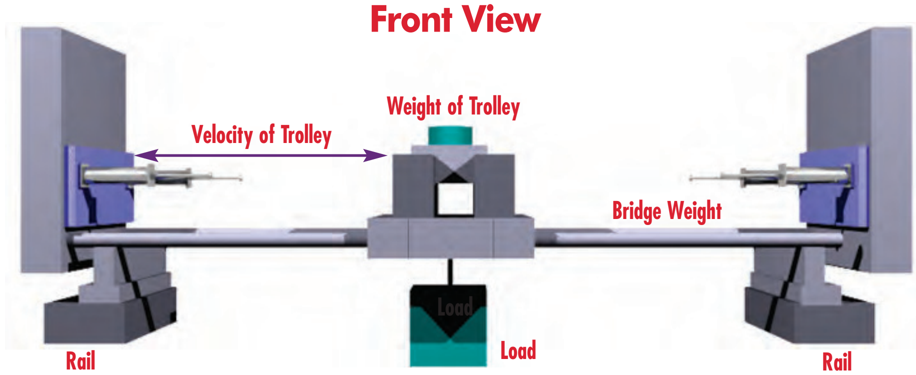
Crane B		Per Buffer
Propelling Force Crane	lbs.	
Propelling Force Trolley	lbs.	
Weight of Crane (W_b)	lbs.	
Weight of Trolley (W_{tb})	lbs.	
Crane Velocity (V_b)	in./sec.	
Trolley Velocity (V_{tb})	in./sec.	

Crane C		Per Buffer
Propelling Force Crane	lbs.	
Propelling Force Trolley	lbs.	
Weight of Crane (W_c)	lbs.	
Weight of Trolley (W_{tc})	lbs.	
Crane Velocity (V_c)	in./sec.	
Trolley Velocity (V_{tc})	in./sec.	

Please note:
Unless instructed otherwise, ITT Enidine will always calculate with:

- 100% velocity v, and
- 100% propelling force F_D

Front View



Plan Views

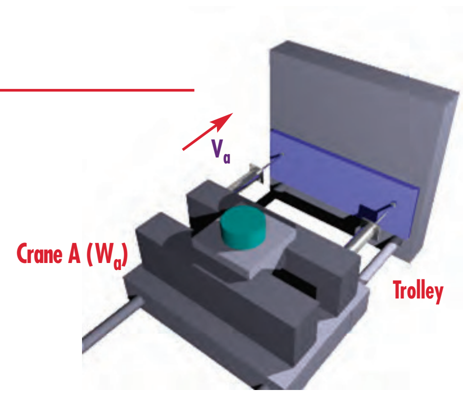
Application 1
Crane A against Solid Stop

Velocity:

$V_r = V_a$

Impact weight per buffer:

$$W_d = \frac{Wa + (1.8)\ Wta}{\text{Total Number of Shocks}}$$



Application 2
Crane A against Crane B

Velocity:

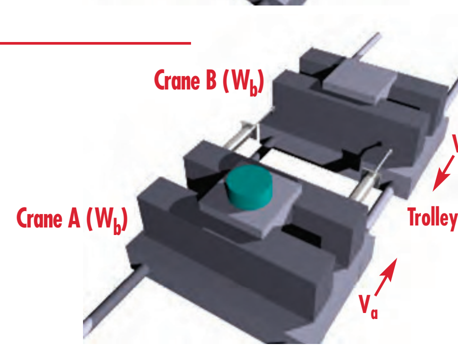
$V_r = V_a + V_b$

Impact weight per buffer:

$W_1 = Wa + (1.8)\ Wta$

$W_2 = Wb + (1.8)\ Wtb$

$$W_d = \frac{W_1\ W_2}{(W_1 + W_2)(\text{Total Number of Shocks})}$$



Application 3
Crane B against Crane C

Velocity:

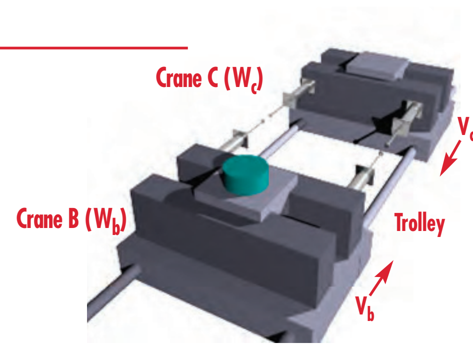
$$V_r = \frac{V_b + V_c}{2}$$

Impact weight per buffer:

$W_1 = Wb + (1.8)\ Wtb$

$W_2 = Wc + (1.8)\ Wtc$

$$W_d = \frac{2\ W_1\ W_2}{(W_1 + W_2)(\text{Number of Shocks Per Rail})}$$



Application 4
Crane C against Solid Stop with Buffer

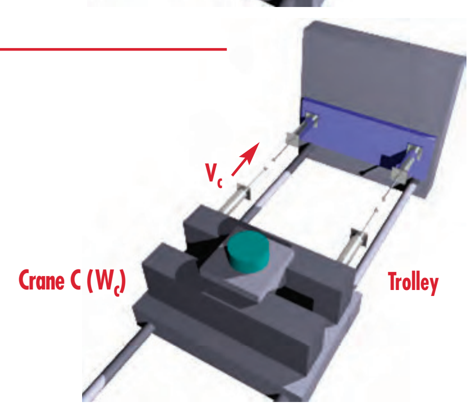
Velocity:

$$V_r = \frac{V_c}{2}$$

Impact weight per buffer:

$W_1 = W_c + 1.8\ (W_{tc})$

$$W_d = \frac{2\ W_1}{\text{Number of Shocks Per Rail}}$$



Overview

Calculation Example for Harbor Cranes as Application 1

Please note that this example is not based on any particular standard. The slung load can swing freely, and is therefore not taken into account in the calculation.

Given Values

Bridge Weight:	**837,750 lbs.**
Weight of Trolley:	**99,200 lbs.**
Crane Velocity:	**60 in./sec.**
Required Stroke:	**24 in.**
Trolley Velocity:	**160 in./sec.**
Required Stroke:	**40 in.**

Determination of the Maximum Impact Weight W_d per Buffer

$$W_d = \frac{Wa + (1.8)\ Wta}{\text{Total Number of Shocks}}$$

$$W_d = \frac{837{,}750 + (1.8)(99{,}200)}{2}$$

$$\mathbf{W_d = 508{,}155\ lbs.}$$

Determine Size of Shock Absorber for Crane

$$E_K = \frac{W_d}{772} \bullet V_r^2$$

$$E_K = \frac{508{,}155\ \text{lbs.}}{772} \bullet (60\ \text{in./sec.})^2$$

$$\mathbf{E_K = 2{,}369{,}635\ in\text{-}lbs.}$$

$V_r = V_a$ (Application 1)

E_K = Kinetic Energy

η = Efficiency

Selecting for required 24-inch stroke:

HD 5.0 x 24, maximum shock force ca. 116,159 lbs = $F_s = \frac{E_K}{s \bullet \eta}$

Determine Size of Shock Absorber for Trolley

W_t = Trolley Weight per Shock Absorber

$$W_t = \frac{99{,}200\ \text{lbs.}}{2}$$

$$\mathbf{W_t = 49{,}600\ lbs.}$$

$$E_K = \frac{W_t}{772} \bullet V_t^2$$

$$E_K = \frac{49{,}600\ \text{lbs.}}{772} \bullet (160\ \text{in./sec.})^2$$

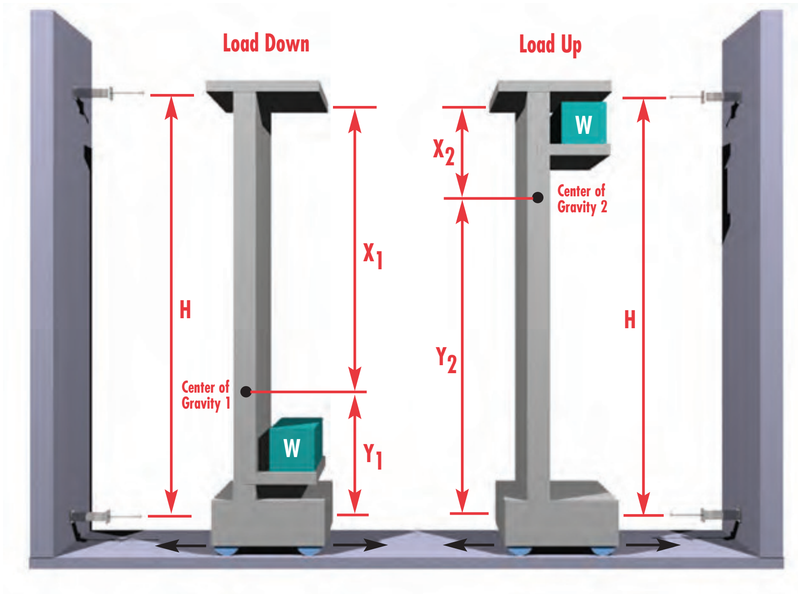
$$\mathbf{E_K = 1{,}644{,}767\ in\text{-}lbs.}$$

Selecting for required 40-inch stroke:

HDN 4.0 x 40, maximum shock force ca. 48,376 lbs. = $F_s = \frac{E_K}{s \bullet \eta}$

Overview

Application 1	Value
Buffer Distance H	ft.
Distance X_1	ft.
Distance Y_1	ft.
Distance X_2	ft.
Distance Y_2	ft.
Total Weight	lbs.
$W_{max\ d}$	lbs.
$W_{min\ d}$	lbs.
$W_{max\ u}$	lbs.
$W_{min\ u}$	lbs.



Calculation Example Stacker Cranes

Please note that this example shows how to calculate the maximum impact weight on the upper and lower shock absorbers for a stacker crane.

Given Values

Distance Between Buffers:	H = 60 ft.
Distance to C of G1 - Upper:	X_1 = 45 ft.
Distance to C of G1 - Lower:	Y_1 = 15 ft.
Distance to C of G2 - Upper:	X_2 = 21 ft.
Distance to C of G1 - Lower:	Y_2 = 39 ft.
Total Weight:	W = 40,000 lbs.

Calculation for Lower Shock Absorbers

$$W_{max\ d} = \frac{X_1}{H} \bullet W \qquad W_{max\ d} = \frac{X_2}{H} \bullet W$$

$$W_{max\ d} = \frac{15\ m}{20\ m} \bullet 20\ t \qquad W_{max\ d} = \frac{21\ ft.}{60\ ft.} \bullet 40{,}000\ lbs.$$

$$W_{max\ d} = 15\ t \qquad W_{max\ d} = 14{,}000\ lbs.$$

Calculation for Upper Shock Absorbers

$$W_{max\ d} = \frac{Y_1}{H} \bullet W \qquad W_{max\ d} = \frac{Y_2}{H} \bullet W$$

$$W_{max\ d} = \frac{5\ m}{20\ m} \bullet 20\ t \qquad W_{max\ d} = \frac{39\ ft.}{60\ ft.} \bullet 40{,}000\ lbs.$$

$$W_{max\ d} = 5\ t \qquad W_{max\ d} = 26{,}000\ lbs.$$

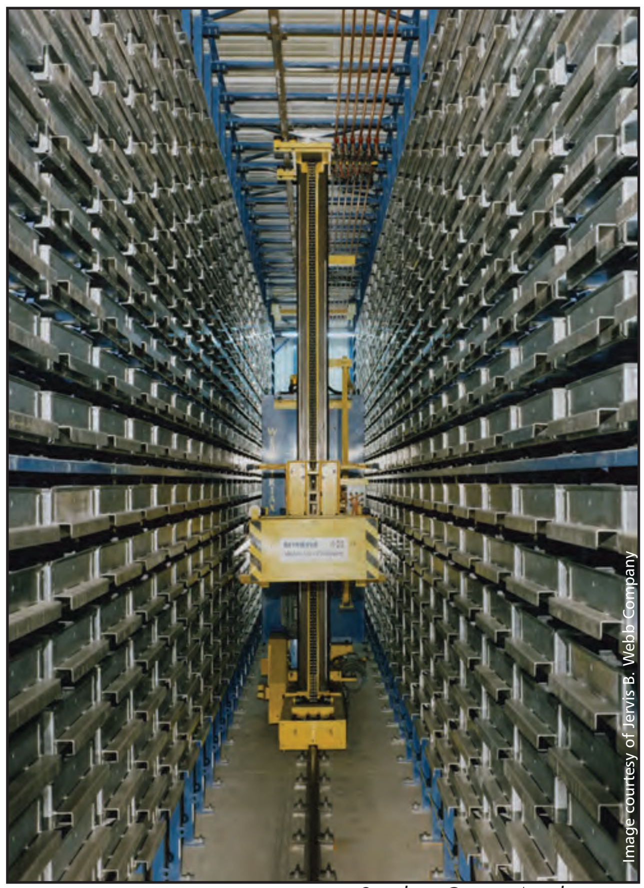
Shock Absorber Selection

Using the value for W_{max} obtained above, the kinetic energy can be calculated, and a shock absorber selected.

Typical Applications

Overhead Crane Applications

Cargo Crane Applications



Stacker Crane Applications

Shock Absorber and Rate Controls Quick Selection Guide
Typical Selections

Technical Data

Use this **ITT Enidine Product Quick Selection Guide** to quickly locate potential shock absorber models most suited for your requirements. Models are organized in order of smallest to largest energy capacity per cycle within their respective product families.

ITT Enidine Adjustable Shock Absorbers

Catalog No. (Model)	(S) Stroke (in.)	(E_T) Max. in.-lbs./cycle	(E_TC) Max. in.-lbs./hour	Damping Type	Page No.
	1 in. = 25,4mm	1 in.-lb. = .11 Nm			
OEM 0.1M (B)	0.28	62	120,000	D	21
ECO OEM .15M (B)	0.38	62	185,000	D	21
ECO OEM .25 (B)	0.38	62	195,000	D	21
ECO LROEM .25 (B)	0.38	62	195,000	D	21
ECO OEM .35 (B)	0.50	170	331,000	D	21
ECO LROEM .35 (B)	0.50	170	331,000	D	21
ECO OEM .5 (B)	0.50	275	311,000	D	21
ECO LROEM .5 (B)	0.50	275	311,000	D	21
ECO OEM 1.0 (B)	1.00	715	681,000	C	21
ECO LROEM 1.0 (B)	1.00	715	681,000	C	21
ECO OEM 1.15 X 1	1.00	1,900	737,000	C	24
ECO LROEM 1.15 X 1	1.00	1,900	737,000	C	24
ECO OEM 1.15 X 2	2.00	3,750	963,000	C	24
ECO LROEM 1.15 X 2	2.00	3,750	963,000	C	24
ECO OEM 1.25 x 1	1.00	1,900	886,000	C	24
ECO LROEM 1.25 x 1	1.00	1,900	886,000	C	24
ECO OEM 1.25 x 2	2.00	3,750	1,084,000	C	24
ECO LROEM 1.25 x 2	2.00	3,750	1,084,000	C	24
LROEMXT ¾ x 1	1.00	3,750	1,120,000	C	27
OEMXT ¾ x 1	1.00	3,750	1,120,000	C	27
LROEMXT 1.5M x 1	1.00	3,750	1,120,000	C	27
OEMXT 1.5M x 1	1.00	3,750	1,120,000	C	27
LROEMXT ¾ x 2	2.00	7,500	1,475,000	C	27
OEMXT ¾ x 2	2.00	7,500	1,475,000	C	27
LROEMXT 1.5M x 2	2.00	7,500	1,475,000	C	27
OEMXT 1.5M x 2	2.00	7,500	1,475,000	C	27
OEMXT ¾ x 3	3.00	11,500	1,775,000	C	27
OEMXT 1.5M x 3	3.00	11,500	1,775,000	C	27
LROEMXT 1⅛ x 1	1.00	6,000	2,000,000	C	27
LROEMXT 1⅛ x 2	2.00	20,000	2,400,000	C	29
OEMXT 1⅛ x 2	2.00	20,000	2,400,000	C	29
LROEMXT 2.0M x 2	2.00	20,000	2,400,000	C	29
OEMXT 2.0M x 2	2.00	20,000	2,400,000	C	29
OEM 3.0M x 2	2.00	20,000	3,290,000	C	31
OEMXT 1⅛ x 4	4.00	40,000	3,200,000	C	29
OEMXT 2.0M x 4	4.00	40,000	3,200,000	C	29
OEM 4.0M x 2	2.00	34,000	13,300,000	C	31
OEM 3.0M x 3.5	3.50	35,000	5,770,000	C	31
OEMXT 1⅛ x 6	6.00	60,000	3,730,000	C	29
OEMXT 2.0M x 6	6.00	60,000	3,730,000	C	29
OEM 3.0M x 5	5.00	50,000	8,260,000	C	31
OEM 3.0M x 6.5	6.50	65,000	10,750,000	C	31
OEM 4.0M x 4	4.00	68,000	16,000,000	C	31
OEM 4.0M x 6	6.00	102,000	18,600,000	C	31
OEM 4.0M x 8	8.00	136,000	21,300,000	C	31
OEM 4.0M x 10	10.00	170,000	24,000,000	C	31

Key for Damping Type:
D – Dashpot
C – Conventional
P – Progressive
SC – Self-compensating

ITT Enidine Non-Adjustable Shock Absorbers

Catalog No. (Model)	(S) Stroke (in.)	(E_T) Max. in.-lbs./cycle	(E_TC) Max. in.-lbs./hour	Damping Type	Page No.
	1 in. = 25,4mm	1 in.-lb. = .11 Nm			
TK 6	0.25	9	31,863	D	39
TK 8	0.25	50	42,480	D	39
TK 21	0.25	20	36,000	D	40
ECO 8	0.25	35	55,000	SC	47
TK 10M	0.25	50	115,000	D	40
ECO 10	0.28	62	120,700	SC	47
ECO 15	0.41	106	275,000	SC	47
STH .25M	0.25	100	39,000	D	41
ECO 25	0.50	265	389,000	SC	47
ECOS 50	0.50	285	440,000	SC	47
ECO 50	0.88	550	523,000	SC	47
STH .5M	0.50	585	390,000	D	41
ECO 100	1.00	930	681,500	SC	47
PRO 110	1.56	1,860	743,500	P	50
ECO 120	1.00	1,640	743,000	SC	50
ECO 125	1.00	1,640	920,500	SC	50
PMXT 1525	1.00	3,250	1,120,000	SC	59
STH .75M	0.75	2,150	780,000	D	41
ECO 220	2.00	3,100	911,600	SC	50
ECO 225	2.00	3,100	1,124,000	SC	50
PMXT 1550	2.00	6,500	1,475,000	SC	59
STH 1.0M	1.00	4,400	1,300,000	D	41
PMXT 1575	3.00	10,000	1,775,000	SC	59
STH 1.0M x 2	2.00	8,800	2,100,000	D	41
PMXT 2050	2.00	16,500	2,400,000	SC	59
STH 1.5M x 1	1.00	10,200	2,200,000	D	41
PMXT 2100	4.00	33,000	3,200,000	SC	59
STH 1.5M x 2	2.00	20,400	3,200,000	D	41
PMXT 2150	6.00	50,000	3,730,000	SC	59

Key for Damping Type:
D – Dashpot
C – Conventional
P – Progressive
SC – Self-compensating

The design of Jarret Series Industrial Shock Absorber utilizes the unique compression and shear characteristics of specially formulated silicone elastomers.

These characteristics allow the energy absorption and return spring functions to be combined into a single unit **without the need for an additional gas or mechanical spring stroke return mechanism**.

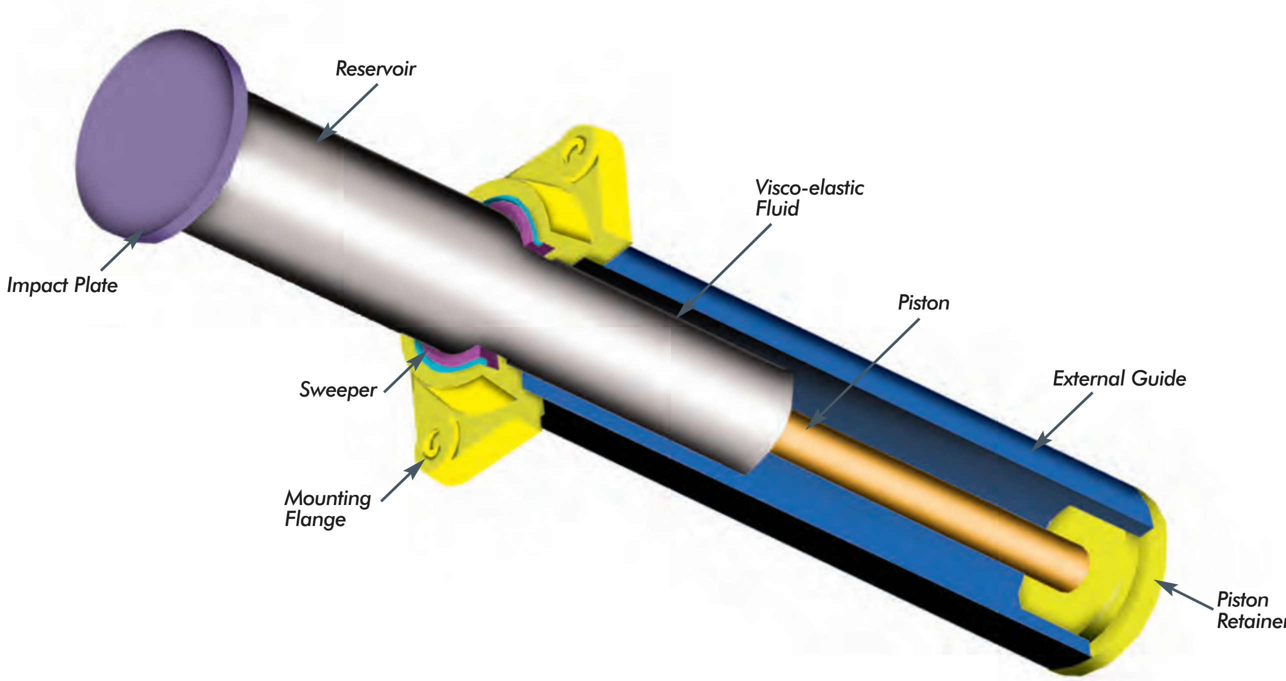
Applications

Shock protection for all types of industries including:
Defense, Automotive, Railroad, Materials Handling, Marine, Pulp/Paper, Metal Production and Processing.

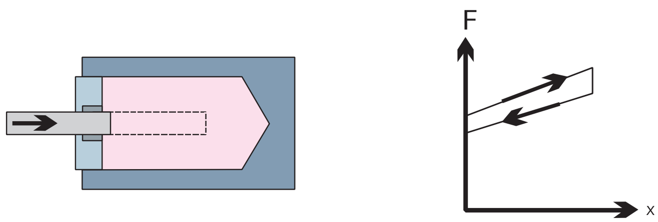
Advantages:

- Simple design
- High reliability
- High damping coefficient
- Low sensitivity to temperature variances

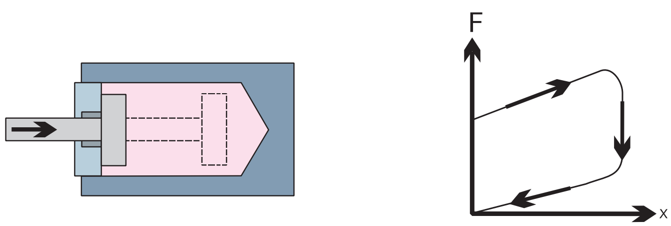
Visco-elastic Technology



Visco-elastic technology makes use of the fundamental properties of specially formulated Jarret visco-elastic medium.



Compressibility:
Preloaded spring function
$\mathbf{F = F_0 + Kx}$



Viscosity:
Shock absorber function
$\mathbf{F = F_0 + Kx + CV^{\alpha}}$ **with** α **between 0,1 and 0,4**

The two functions can be used separately or in combination, in the same product:

Preloaded Spring:
Spring Function Only
- Hysteresis of between 5% and 10%
- Reduced weight and space requirement
- Force/stroke characteristic is independent of actuation speed

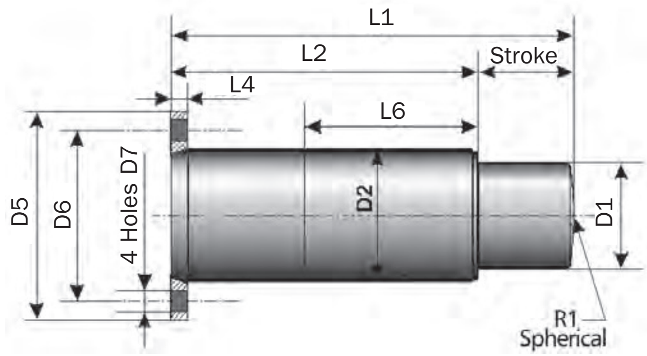
Shock Absorber Without Spring Return:
Shock Absorbing Function Only
- Dampening devices
- Blocking devices

Preloaded Spring Shock Absorbers:
Combine Spring and Shock Absorber Functions
- Dissipate between 30% and 100% of energy
- Force/stroke characteristics remain relatively unchanged between 15°F and 160°F (-10°C and + 70°C)

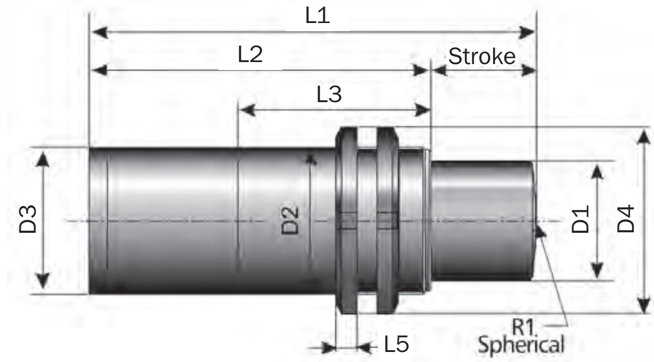
* Spring and shock absorber products are capable of functioning between 15°F and 160°F (-10°C and + 70°C). However, standard products are not intended for use over the full rated temperature range. Consult factory for special product considerations required to accommodate operation over a wide temperature range.

BC1ZN → BC1GN Series

Technical Data



Rear Flange Mounting - Fa



Threaded Body Mounting - Fc

Catalog No./ Model	Max Energy Capacity in-lbs. (kJ)	Stroke in. (mm)	Return Force Extension lbs. (kN)	Return Force Compression lbs. (kN)	Rdy_0 lbs. (kN)	Rdymax Max Shock Force lbs. (kN)
BC1ZN	885 (0,1)	0.47 (12)	211 (0,94)	1,213 (5,4)	1,349 (6)	2,473 (11)
BC1BN	3,806 (0,43)	0.87 (22)	562 (2,5)	3,147 (14,0)	3,147 (14)	6,070 (27)
BC1DN	13,276 (1,5)	1.4 (35)	1,169 (5,2)	6474 (28,8)	6,295 (28)	13,489 (60)
BC1EN	30,093 (3,4)	1.8 (45)	1,753 (7,8)	9,666 (43,0)	10,116 (45)	22,481 (100)
BC1FN	61,955 (7)	2.4 (60)	3,057 (13,6)	17,220 (76,6)	20,233 (90)	33,721 (150)
BC1GN	123,910 (14)	3.1 (80)	4,271 (19,0)	29,225 (130,0)	29,225 (130)	51,706 (230)

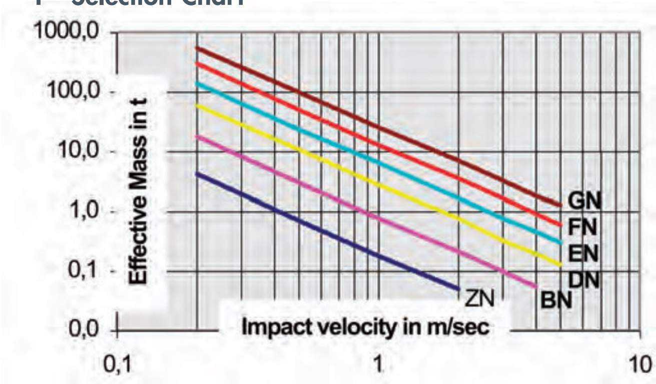
Catalog No./ Model	L1 in. (mm)	L2 in. (mm)	L3 in. (mm)	L4 in. (mm)	L5 in. (mm)	L6 in. (mm)	R1 in. (mm)	D1 in. (mm)	D2 in. (mm)	D3 in. (mm)	D4 in. (mm)	D5 in. (mm)	D6 in. (mm)	D7 in. (mm)	Weight lbs. (kg.)
BC1ZN	2.95 (75)	2.1 (53)	2.1 (52)	0.39 (10)	0.28 (7)	1.7 (43)	– –	0.75 (19)	M25 x 1,5	0.79 (20)	1.5 (38)	2.2 (57)	1.6 (41)	0.28 (7)	0.7 (0,3)
BC1BN	4.7 (120)	3.9 (98)	3.8 (96)	0.47 (12)	0.31 (8)	3.4 (86)	– –	1.0 (25)	M35 x 1,5	1.3 (32)	2.0 (52)	3.1 (80)	2.4 (60)	0.35 (9)	1.5 (0,7)
BC1BN-M	4.7 (120)	3.9 (98)	3.8 (96)	0.47 (12)	0.35 (9)	– -	– –	1.0 (25)	M40 x 1,5	1.3 (32)	2.3 (58)	– –	– –	– –	1.8 (0,8)
BC1DN-70	6.9 (175)	5.5 (140)	5.4 (138)	0.47 (12)	0.43 (11)	5.0 (128)	– –	1.5 (38)	M50 x 1,5	1.8 (45)	2.8 (70)	3.5 (90)	2.8 (70)	0.35 (9)	4.2 (1,9)
BC1DN-85	6.9 (175)	5.5 (140)	5.4 (138)	0.47 (12)	0.43 (11)	5.0 (128)	– –	1.5 (38)	M50 x 1,5	1.8 (45)	2.8 (70)	4.2 (106)	3.3 (85)	0.43 (11)	4.4 (2)
BC1DN-M	6.9 (175))	5.5 (140)	5.4 (138)	0.47 (12)	0.43 (11)	– –	– –	1.5 (38)	M60 x 2	1.8 (45)	2.8 (70)	– –	– –	– –	4.4 (2)
BC1EN	8.4 (213)	6.6 (168)	6.2 (158)	0.39 (10)	0.51 (13)	6.2 (158)	5.1 (130)	2.4 (60)	M75 x 2	2.8 (72)	3.9 (98)	4.8 (122)	4.0 (100)	0.43 (11)	11 (5)
BC1FN	10.6 (270)	8.3 (210)	5.1 (130)	0.47 (12)	0.63 (16)	5.1 (130)	5.9 (150)	2.9 (74,5)	M90 x 2	3.5 (90)	4.7 (120)	5.9 (150)	4.7 (120)	0.51 (13)	23.1 (10,5)
BC1GN	13.3 (337)	10.1 (257)	5.7 (145)	0.55 (14)	0.75 (19)	5.7 (145)	13.8 (350)	3.5 (90)	M110 x 2	4.3 (110)	5.7 (145)	6.9 (175)	5.6 (143)	0.70 (18)	37.5 (17)

Notes: Spring and shock absorber products are capable of functioning between 15°F and 160°F (-10°C and + 70°C). However, standard products are not intended for use over the full rated temperature range.
Consult factory for special product considerations required to accommodate operation over a wide temperature range.

BC1ZN → BC1GN Series

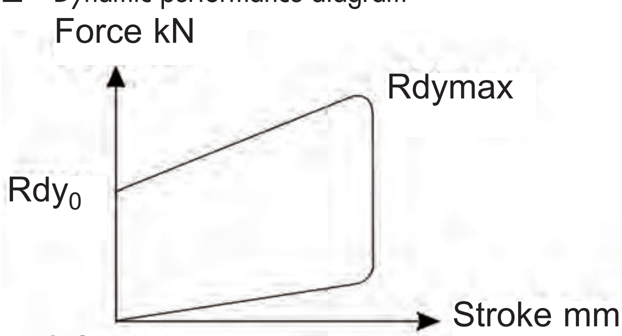
Sizing Example

1 - Selection Chart



Based On

- ☐ Impact velocity (V) : 2 m/s
- ☐ Operating temperature : 20° to + 40°C
- ☐ Surface protection : Electrolytic zinc
- ☐ Dynamic performance diagram



Symbols:

En = Energy Capacity (kJ)
C = Maximum Stroke (mm)
Rdy = Dynamic Reaction Force (kN)

2 - Energy Calculation

$$E = \frac{1}{2} M_e V_e^2$$

3 - Allowable Impact Velocity

$$IF < 20 \times \frac{En}{E} \text{ Impacts/hour}$$

4 - Effective (Actual) Stroke Calculation

$$Ce = C\left(\sqrt{\frac{E}{En\,(0{,}03\,V + 0{,}24)} + 1{,}36} - 1{,}17\right)$$

5 - Calculation of Effective Reaction Force Rdy_e

$$Rdy_e = \left[\left(\frac{Rdymax - Rdy_0}{C}\right) \times Ce + Rdy_0\right](0{,}1V + 0{,}8)$$

6 - Application Example

Given data: Effective mass = 15 t
Effective velocity = 0,8 m/s
Impact frequency: 25 impacts/hour

1. Energy dissipated per impact: $E = \frac{1}{2}(15)(0{,}8) = 4{,}8$ kJ

2. BC1FN Selected

3. Allowable impact frequency IF < 20x7/4.8 = 29
25 < 29

4. Effective (Actual) Stroke:

$$Ce = 60\left(\sqrt{\frac{4{,}8}{7\,(0{,}03 \times 0{,}8 + 0{,}24)} + 1{,}36} - 1{,}17\right)$$

Ce = 49 mm

5. Effective Reaction Force:

$$Rdy_e = \left[\frac{(150 - 90)}{60} \times 49 + 90)\right](0{,}1 \times 0{,}8 + 0{,}8)$$

Rdy_e = 122 kN

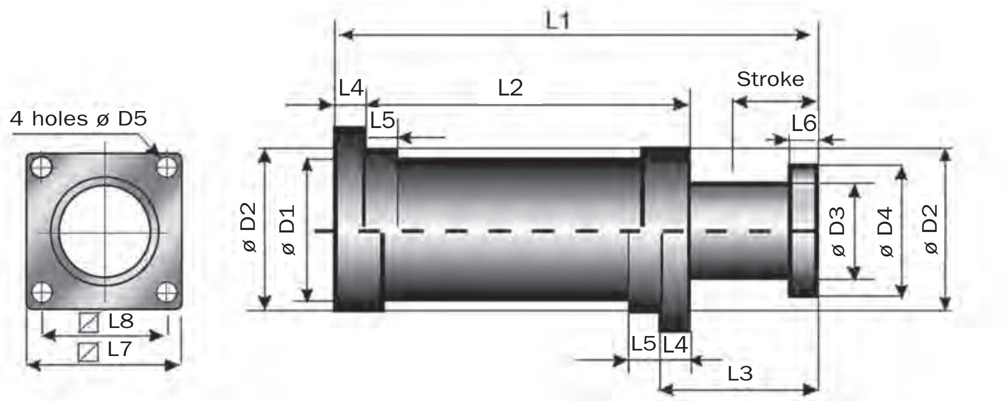
6. Compare standards to results:

	BC1FN		APPLICATION
E (kJ) =	7	>	4,8
C (mm) =	60	>	49
Rdymax (kN)	150	>	122

All performance characteristics can be modified.
Please advise us of your specific requirements.

BC5A → BC5E Series

Technical Data



Rear Flange Mount - Fa **Front Flange Mount - Fc**

Catalog No./ Model	Max Energy Capacity in-lbs. (kJ)	Stroke in. (mm)	Return Force Extension lbs. (kN)	Return Force Compression lbs. (kN)	Rdy_0 lbs. (kN)	Rdymax Max Shock Force lbs. (kN)
BC5A-105	221,268 (25)	4.1 (105)	4,159 (18,5)	31,630 (140,7)	37,543 (167)	69,691 (310)
BC5B-130	442,537 (50)	4.7 (130)	13,039 (58,0)	58,416 (259,9)	69,691 (310)	121,397 (540)
BC5C-140	663,806 (75)	5.5 (140)	11,015 (49,0)	73,827 (328,4)	89,924 (400)	157,366 (700)
BC5D-160	885,075 (100)	6.3 (160)	13,376 (59,5)	85,427 (380,0)	105,660 (470)	184,343 (820)
BC5E-180	1,327,612 (150)	7.1 (180)	26,269 (117,0)	122,656 (546)	143,878 (640)	247,290 (1 100)

Catalog No./ Model	L1 in. (mm)	L2 in. (mm)	L3 in. (mm)	L4 in. (mm)	L5 in. (mm)	L6 in. (mm)	L7 in. (mm)	L8 in. (mm)	D1 in. (mm)	D2 in. (mm)	D3 in. (mm)	D4 in. (mm)	D5 in. (mm)	Weight lbs. (kg)
BC5A-105	16.3 (415)	10.8 (275)	5.5 (140)	0.79 (20)	1.2 (30)	0.59 (15)	5.3 (135)	4.1 (105)	4.6 (116)	4.6 (116)	3.4 (87)	4.7 (120)	0.55 (14)	55 (25)
BC5B-130	19.7 (500)	12.8 (325)	6.9 (175)	1.0 (25)	1.3 (33)	1.2 (30)	6.1 (155)	4.9 (125)	5.6 (142)	5.6 (142)	4.5 (115)	5.4 (138)	0.55 (14)	88 (40)
BC5C-140	20.5 (520)	12.4 (315)	8.1 (205)	1.2 (30)	1.4 (36)	1.4 (35)	6.9 (175)	5.5 (140)	6.3 (160)	6.3 (160)	5.2 (132)	6.2 (158)	0.70 (18)	99 (45)
BC5D-160	23 (585)	13.8 (350)	9.3 (235)	1.4 (35)	1.6 (40)	1.6 (40)	8.5 (215)	6.7 (170)	7.1 (180)	7.1 (180)	6.0 (153)	7.3 (185)	0.87 (22)	161 (73)
BC5E-180	26.4 (670)	15.9 (405)	10.4 (265)	1.6 (40)	1.8 (45)	1.8 (45)	9.8 (250)	7.7 (195)	8.5 (215)	8.5 (215)	7.2 (182)	8.7 (220)	1.0 (26)	258 (117)

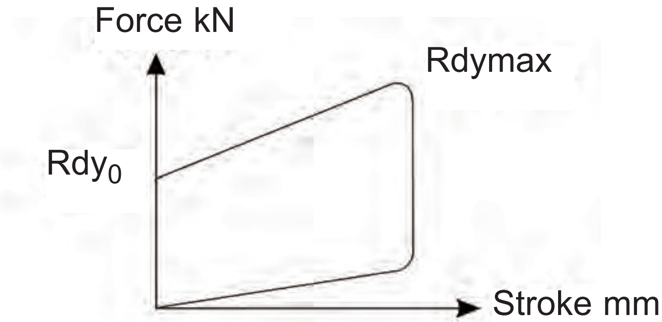
Impact Speed: BC5 Series shock absorbers are designed for impact velocities of up to 4 m/sec. Higher impact velocities require custom modification.
Spring and shock absorber products are capable of functioning between 15°F and 160°F (-10°C and + 70°C). However, standard products are not intended for use over the full rated temperature range.
Consult factory for special product considerations required to accommodate operation over a wide temperature range.

BC5A → BC5E Series

Sizing Example

Based On

- ☐ Impact velocity (V) : 2 m/s
- ☐ Operating temperature : 20° to + 40°C
- ☐ Surface protection : Electrolytic zinc
- ☐ Dynamic performance diagram



Symbols:

En = Energy Capacity (kJ)
C = Maximum Stroke (mm)
Rdy = Dynamic Reaction Force (kN)

1 - Energy Calculation

$$E = \frac{1}{2} M_e V_e^2$$

2 - Allowable Impact Frequency (IF)

$$IF < 15 \times \frac{En}{E} \text{ Impacts/hour}$$

3 - Effective Stroke Calculation

$$Ce = C\left(\sqrt{\frac{E}{En\ (0{,}03\ V + 0{,}24)} + 1{,}36} - 1{,}17\right)$$

4 - Calculation of Effective Reaction Rdy_e

$$Rdy_e = \left[\left(\frac{Rdymax - Rdy_0}{C}\right) \times Ce + Rdy_0\right](0{,}1V + 0{,}8)$$

5 - Application Example

Data: Two shock absorbers in series, Effective mass m =300 t, Impact speed v = 1,2 m/s (which is an impact of 0,6 m/s on each shock absorber), Impact frequency = 15 impacts/hour, Maximum allowable structural load 1000 kN

1: $E = \frac{1}{2}\left(\frac{1}{2} mV^2\right)$

$E = \frac{1}{2}\left(\frac{1}{2}\ 300 \times 1{,}2^2\right) = 108$ kJ

2. Selection BC5E-180

3. Maximum allowable impact frequency is $15 \times \frac{150}{108}$ 21 impacts/hour. Therefore 15 impacts/hour is acceptable.

$$15 < 15 \times \frac{150}{108}$$

$$15 < 21$$

4. Effective (actual) stroke is 167 mm

$$Ce = 180 \times \left(\sqrt{\frac{108}{150\ (0{,}03 \times 0.6 + 0{,}24)} + 1{,}36} - 1{,}17\right) = 156 \text{ mm}$$

5. $Rdye = \left[(1\ 100 - 640) \times \frac{156}{180} + 640\right](0{,}1 \times 0{,}6 + 0{,}8)$

Rdye = 893 kN < 1000 kN

6. Compare standards to results:

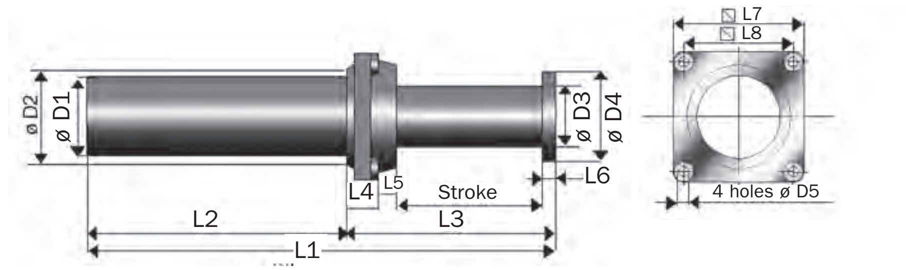
	BC5E-180		APPLICATION
E (kJ) =	150	>	108
IF =	21	>	15
C (mm) =	180	>	156
Rdymax (kN)	1100	>	893

Note: maximum allowed structural load is 1 000 kN > 893 kN

All performance characteristics can be modified.
Please advise us of your specific requirements.

XLR6-150 → XLR-800 Series

Technical Data



XLR Series - Front Flange Mount- Fc

Catalog No./ Model	Max Energy Capacity in-lbs. (kJ)	Stroke in. (mm)	Return Force Extension lbs. (kN)	Return Force Compression lbs. (kN)	Rdy_0 lbs. (kN)	Rdymax Max Shock Force lbs. (kN)
XLR6-150	53,104 (6)	5.9 (150)	652 (2,9)	4,609 (20,5)	5,620 (25)	11,240 (50)
XLR12-150	106,209 (12)	5.9 (150)	1,866 (8,3)	8,655 (38,5)	14,837 (66)	22,481 (100)
XLR12-200	106,209 (12)	7.9 (200)	1,259 (5,6)	6,744 (30,0)	9,442 (42)	17,535 (78)
XLR25-200	221,269 (25)	7.9 (200)	3,012 (13,4)	16,726 (74,4)	21,537 (95)	33,721 (150)
XLR25-270	221,269 (25)	10.6 (270)	2,495 (11,1)	11,555 (51,4)	14,837 (66)	25,179 (112)
XLR50-275	442,537 (50)	10.8 (275)	4,429 (19,7)	29,225 (130,0)	26,527 (118)	51,706 (230)
XLR50-400	442,537 (50)	15.7 (400)	2,900 (12,9)	18,839 (83,8)	16,861 (75)	33,721 (150)
XLR100-400	885,075 (100)	15.7 (400)	5,620 (25,0)	36,531 (162,5)	39,342 (175)	71,939 (320)
XLR100-600	885,075 (100)	23.6 (600)	2,608 (11,6)	29,765 (132,4)	19,109 (85)	51,706 (230)
XLR150-800	1,327,612 (150)	31.5 (800)	5,216 (23,2)	34,216 (152,2)	17,984 (80)	56,202 (250)

Impact Speed: Types XLR and BCLR Series shock absorbers are designed for impact velocities of up to 2 m/sec.
Higher impact velocities require custom modification.

Catalog No./ Model	L1 in. (mm)	L2 in. (mm)	L3 in. (mm)	L4 in. (mm)	L5 in. (mm)	L6 in. (mm)	L7 in. (mm)	L8 in. (mm)	D1 in. (mm)	D2 in. (mm)	D3 in. (mm)	D4 in. (mm)	D5 in. (mm)	Weight lbs. (kg.)
XLR6-150	16.1 (410)	9.1 (231)	7.0 (179)	0.75 (19)	0 (0)	0.39 (10)	3.5 (90)	2.8 (70)	2.0 (50)	3.5 (90)	1.5 (38)	2.0 (50)	0.35 (9)	9.3 (4,2)
XLR12-150	18.9 (480)	11.2 (285)	7.7 (195)	0.71 (18)	0.60 (15)	0.47 (12)	4.3 (110)	3.3 (85)	3.0 (75)	3.5 (90)	2.2 (57)	3.1 (80)	0.43 (11)	24.3 (11)
XLR12-200	20.9 (530)	11.2 (285)	9.6 (245)	0.71 (18)	0.60 (15)	0.47 (12)	4.3 (110)	3.3 (85)	3.0 (75)	3.5 (90)	2.2 (57)	3.1 (80)	0.43 (11)	24.3 (11)
XLR25-200	24.4 (620)	14.6 (370)	9.8 (250)	0.79 (20)	0.71 (18)	0.47 (12)	5.3 (135)	4.1 (105)	3.5 (90)	4.3 (110)	2.8 (72)	4.0 (100)	0.6 (14)	44.1 (20)
XLR25-270	27.2 (690)	14.6 (370)	12.6 (320)	0.79 (20)	0.71 (180	0.47 (12)	5.3 (135)	4.1 (105)	3.5 (90)	4.3 (110)	2.8 (72)	4.0 (100)	0.6 (14)	55.1 (25)
XLR50-275	33.7 (855)	20.5 (520)	13.2 (335)	1.0 (25)	0.79 (20)	0.60 (15)	6.9 (175)	5.5 (140)	4.3 (110)	5.9 (150)	3.4 (87)	4.7 (120)	0.71 (18)	88.2 (40)
XLR50-400	38.6 (980)	20.5 (520)	18.1 (460)	1.0 (25)	0.79 (20)	0.60 (15)	6.9 (175)	5.5 (140)	4.3 (110)	5.9 (150)	3.4 (87)	4.7 (120)	0.71 (18)	88.2 (40)
XLR100-400	53.9 (1370)	35.8 (910)	18.1 (460)	1.0 (25)	0.79 (20)	0.60 (15)	6.9 (175)	5.5 (140)	4.3 (110)	5.9 (150)	3.4 (87)	4.7 (120)	0.71 (18)	143.3 (65)
XLR100-600	61.8 (1570)	35.8 (910)	26.0 (660)	1.0 (25)	0.79 (20)	0.60 (15)	6.9 (175)	5.5 (140)	4.3 (110)	5.9 (150)	3.4 (87)	4.7 (120)	0.71 (18)	143.3 (65)
XLR150-800	103.9 (2640)	70.1 (1780)	33.9 (860)	1.0 (25)	0.79 (20)	0.60 (15)	6.9 (175)	5.5 (140)	4.3 (110)	5.9 (150)	3.4 (87)	4.7 (120)	0.71 (18)	253.5 (115)

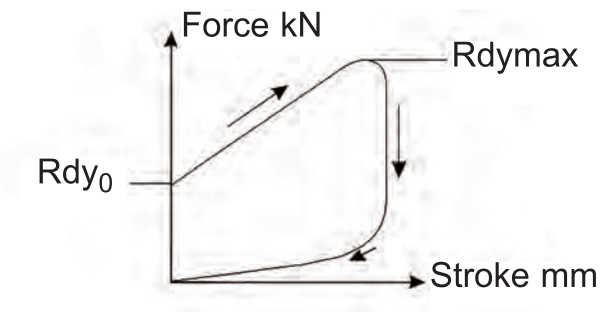
Rear Flange Mounting - Fa on Request.
Spring and shock absorber products are capable of functioning between 15°F and 160°F (-10°C and + 70°C). However, standard products are not intended for use over the full rated temperature range.
Consult factory for special product considerations required to accommodate operation over a wide temperature range.

Jarret Shock Absorbers
XLR Series

XLR6-150 → XLR-800 Series

Based On

- ☐ Impact velocity (V) : 2 m/s
- ☐ Operating temperature : 20° to + 40°C
- ☐ Surface protection : Electrolytic zinc & Painting
- ☐ Dynamic performance diagram



Symbols:

En = Energy Capacity (kJ)
C = Maximum Stroke (mm)
Rdy = Dynamic Reaction Force (kN)

1 - Energy Calculation

$$E = \frac{1}{2} M_e V_e^2$$

2 - Allowable Impact Frequency (IF)

$$IF < 8 \times \frac{En}{E} \text{ Impacts/hour}$$

3 - Required Stroke Calculation

$$Ce = C\left(\sqrt{\frac{E}{En\,(0{,}027\,V + 0{,}22)}} + 1{,}83 - 1{,}35\right)$$

4 - Calculation of Effective Reaction Rdy_e

$$Rdy_e = \left[\left(\frac{Rdymax - Rdy_0}{C}\right) \times Ce + Rdy_0\right](0{,}1V + 0{,}8)$$

5 - Application Example **Data**:

Effective mass = 30 t
Effective impact speed = 2,2
Maximum allowable structural force = 350 kN
Impact frequency = 10/hr

1: Energy dissipated/impact is 72,6 kJ

$$E = \frac{1}{2} \times 15 \times (2{,}2)^2$$

$$E = 72{,}6 \text{ kJ}$$

2: XLR100-400 selected

3: Maximum allowable impact frequency
IF < 8 x 100 / 72,6 = 11
(10<11 impacts/hour is acceptable)

4: Effective (actual) stroke:

$$Ce = 400 \times \left(\sqrt{\frac{72{,}6}{100\,(0{,}027 \times 2{,}7 + 0{,}22)}} + 1{,}83 - 1{,}35\right)$$

$$Ce = 290{,}3 \text{ mm}$$

5: $$Rdye = \left[\left(\frac{320 - 175}{400}\right) 290{,}3 + 175\right](0{,}1 \times 2{,}2 + 0{,}8)$$

$$Rdye = 285{,}8 \text{ kN}$$

(which is less than maximum allowable reaction force of 350 kN)

6. Compare standards to results:

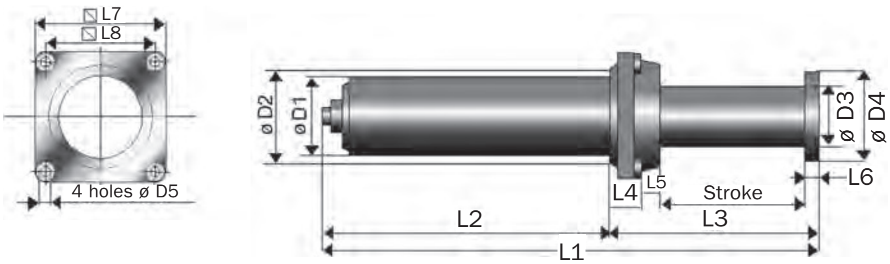
	XLR100-400		APPLICATION
E (kJ) =	100	>	72,6
IF =	11	>	10
C (mm) =	400	>	301,8
Rdymax (kN)	320	>	290,1

Note: maximum allowed structural load is 350 kN > 290,1 kN

All performance characteristics can be modified.
Please advise us of your specific requirements.

BCLR-100 → BCLR-1000 Series

Technical Data



BCLR Series - Front Flange Mount- Fc

Catalog No./ Model	Max Energy Capacity in-lbs. (kJ)	Stroke in. (mm)	Return Force Extension lbs. (kN)	Return Force Compression lbs. (kN)	Rdy_0 lbs. (kN)	Rdymax Max Shock Force lbs. (kN)
BCLR-100	885,075 (100)	15.7 (400)	6,744 (30,0)	36,403 (161,9)	42,714 (190)	69,691 (310)
BCLR-150	1,327,612 (150)	19.7 (500)	9,330 (41,5)	47,300 (201,4)	44,962 (200)	85,427 (380)
BCLR-220S	1,947,614 (220)	15.7 (400)	10,116 (45,0)	60,698 (270,0)	85,427 (380)	153,994 (685)
BCLR-250	2,212,686 (250)	25.6 (650)	10,116 (45,0)	56,877 (253,0)	60,698 (270)	110,156 (490)
BCLR-400	3,540,298 (400)	33.5 (850)	11,144 (49,6)	69,214 (307,9)	74,187 (330)	134,885 (600)
BCLR-600	5,310,477 (600)	41.3 (1050)	10,678 (47,5)	79,020 (351,5)	83,179 (370)	166,359 (740)
BCLR-800	7,080,597 (800)	47.2 (1200)	14,433 (64,2)	99,141 (441,0)	96,668 (430)	193,336 (860)
BCLR-1000	8,850,746 (1000)	51.2 (1300)	19,109 (85,0)	120,048 (534,0)	112,405 (500)	224,809 (1000)

Impact Speed: Types XLR and BCLR Series shock absorbers are designed for impact velocities of up to 2 m/sec.
Higher impact velocities require custom modification.

Catalog No./ Model	L1 in. (mm)	L2 in. (mm)	L3 in. (mm)	L4 in. (mm)	L5 in. (mm)	L6 in. (mm)	L7 in. (mm)	L8 in. (mm)	D1 in. (mm)	D2 in. (mm)	D3 in. (mm)	D4 in. (mm)	D5 in. (mm)	Weight lbs. (kg.)
BCLR-100	44.1 (1120)	26.0 (660)	18.1 (460)	1.0 (25)	0.79 (20)	0.60 (15)	6.9 (175)	5.5 (140)	5.1 (130)	5.9 (150)	4.3 (110)	5.5 (140)	0.71 (18)	139.0 (63)
BCLR-150	53.1 (1350)	30.5 (775)	22.6 (575)	1.2 (30)	1.0 (25)	0.79 (20)	8.5 (215)	6.7 (170)	5.5 (140)	7.3 (185)	4.7 (120)	5.9 (150)	0.87 (22)	198.4 (90)
BCLR-220S	49.5 (1258)	30.8 (783)	18.7 (475)	1.2 (30)	1.0 (25)	0.79 (20)	8.5 (215)	6.7 (170)	6.3 (160)	N/A	5.3 (134)	6.3 (160)	0.87 (22)	243 (110)
BCLR-250	68.9 (1750)	40.4 (1025)	28.5 (725)	1.2 (30)	1.0 (25)	0.79 (20)	8.5 (215)	6.7 (170)	6.1 (155)	7.3 (185)	6.9 (135)	6.7 (170)	0.87 (22)	297.6 (135)
BCLR-400	86.0 (2185)	49.2 (1250)	36.8 (935)	1.4 (35)	1.0 (25)	1.0 (25)	10.4 (265)	8.3 (210)	6.9 (175)	9.3 (235)	5.9 (150)	7.5 (190)	1.1 (27)	480.6 (218)
BCLR-600	100.6 (2555)	55.9 (1420)	44.7 (1135)	1.4 (35)	1.0 (25)	1.0 (25)	10.4 (265)	8.3 (210)	7.9 (200)	9.3 (235)	6.9 (175)	8.5 (215)	1.1 (27)	650.4 (295)
BCLR-800	115.6 (2935)	64.2 (1630)	51.4 (1305)	1.6 (40)	1.4 (35)	1.2 (30)	11.8 (300)	9.4 (240)	8.7 (220)	10.6 (270)	7.5 (190)	9.3 (235)	1.2 (30)	926 (420)
BCLR-1000	127.0 (3225)	71.7 (1820)	55.3 (1405)	1.6 (40)	1.4 (35)	1.2 (30)	11.8 (300)	9.4 (240)	9.1 (230)	10.6 (270)	8.1 (205)	9.8 (248)	1.2 (30)	1036.2 (470)

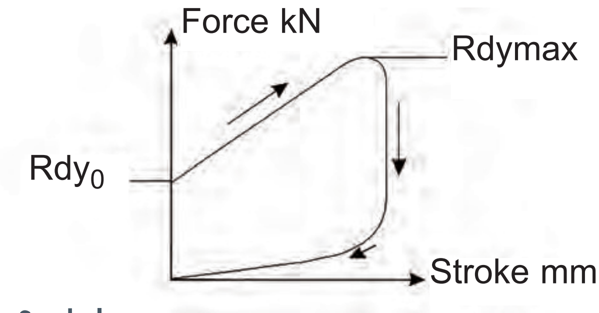
Rear Flange Mounting - Fa on Request.
Spring and shock absorber products are capable of functioning between 15°F and 160°F (-10°C and + 70°C). However, standard products are not intended for use over the full rated temperature range.
Consult factory for special product considerations required to accommodate operation over a wide temperature range.

Sizing Example

BCLR-100 → BCLR-1000 Series

Based On

- ☐ Impact velocity (V) : 2 m/s
- ☐ Operating temperature : 20° to + 40°C
- ☐ Surface protection : Electrolytic zinc & Painting
- ☐ Dynamic performance diagram



Symbols:

En = Energy Capacity (kJ)
C = Maximum Stroke (mm)
Rdy = Dynamic Reaction Force (kN)

1 - Energy Calculation

$$E = \frac{1}{2} M_e V_e^2$$

2 - Allowable Impact Frequency (IF)

$$IF < 8 \times \frac{En}{E} \text{ Impacts/hour}$$

3 - Required Stroke Calculation

$$Ce = C\left(\sqrt{\frac{E}{En\,(0{,}027\,V + 0{,}22)} + 1{,}83} - 1{,}35\right)$$

4 - Calculation of Effective Reaction Rdy_e

$$Rdy_e = \left[\left(\frac{Rdymax - Rdy_0}{C}\right) \times Ce + Rdy_0\right](0{,}1V + 0{,}8)$$

5 - Application Example:

Effective mass = 75 t
Effective impact speed = 2,7
Maximum allowable structural force: 650 kN
Impact frequency = 10/hr

1: Energy dissipated/impact is 274 kJ

2: BCLR-400 selected

3: Maximum allowable impact frequency
IF < 8 x 400 / 274 = 12 (10 impacts/hour is acceptable)
10 < 12

4: Effective (actual) stroke:

$$Ce = 850 \times \left(\sqrt{\frac{274}{400\,(0{,}027 \times 2{,}7 + 0{,}22)} + 1{,}83} - 1{,}35\right)$$

Ce = 587mm

5: Rdye = 520 (0,1 x 2,7 + 0,8) = 556 kN

(which is less than maximum allowable reaction force of 650 kN)

6. Compare standards to results:

	BCLR-400		APPLICATION
E (kJ) =	400	>	274
IF =	12	>	10
C (mm) =	850	>	587
Rdymax (kN)	600	>	556

Note: maximum allowed structural load is 650 kN > 556 kN

All performance characteristics can be modified. Please advise us of your specific requirements.

Typical Applications

Construction Elevator Emergency Stops

Mining Applications

Refinery Material Handling Applications

Typical Applications

High Speed Elevator Applications

Material Transport Crane Applications

Amusement Ride Emergency Stops

FAX NO.: ____________________

DATE: ____________________

ATTN: ____________________

COMPANY: ____________________

The ITT Enidine Application Worksheet makes shock absorber sizing and selection easier.

Fax, phone, or mail worksheet data to Enidine headquarters or your nearest ITT Enidine subsidiary/affiliate or distributor. (See catalog back cover for ITT Enidine locations, or visit www.enidine.com for a list of ITT Enidine distributors.)

Upon ITT Enidine's receipt of this worksheet, you will receive a detailed analysis of your application and product recommendations. (For custom design projects, Enidine representatives will consult with you for specification requirements.)

GENERAL INFORMATION

CONTACT: ____________________

DEPT/TITLE: ____________________

COMPANY: ____________________

ADDRESS: ____________________

____

TEL: ____________ FAX: ____________

EMAIL: ____________________

PRODUCTS MANUFACTURED: ____________________

____

____

____

____

APPLICATION DESCRIPTION

____

____

____

____

____

____

____

____

____

Motion Direction (Check One):

☐ Horizontal ☐ Vertical ☐ Up ☐ Down ☐ Incline Angle ______ Height ______

☐ Rotary Horizontal ☐ Rotary Vertical ☐ Up ☐ Down

Weight (Min./Max.): ____________________ (lbs.)(Kg)

Cycle Rate ____________________ (cycles/hour)

Additional Propelling Force (If Known) ____________ (lbs.)(N)

☐ Air Cyl: Bore ____ (in.)(mm) Max. Pressure ____ (psi)(bar) Rod Dia. ____ (in.)(mm)

☐ Hydraulic Cyl: Bore ____ (in.)(mm) Max. Pressure ____ (psi)(bar) Rod Dia. ____ (in.)(mm)

☐ Motor ________ (hp)(kW) Torque ________ (in-lbs.)(Nm)

Ambient Temp. ____________________ °F (°C)

Environmental Considerations: ____________________

____

SHOCK ABSORBER APPLICATION

(All Data Taken at Shock Absorbe

Number of Shock Absorbers to Stop Load

Impact Velocity (min./max.) ____________ (in./sec.)(m/sec.)

Shock Absorber Stroke Requirements: ____________ (in.)(mm)

G Load Requirements ________ (G)(m/sec^2)

APPLICATION SKETCHES / NOTES

Enidine is a diversified leading manufacturer of highly engineered critical components and customized technology solutions for growing industrial end-markets in energy infrastructure, electronics, aerospace and transportation.

As part of our strategy to make the customer central to everything we do, our core technologies, engineering strength and global scale offers greater value for customers in terms of quality, cost and delivery.

.

Common Applications:

- Automotive
- Auto, Storage and Retrieval
- Bridges and Structures
- Conveyor Systems
- Steel Mills
- Plastic Bottle Manufacturing
- Packaging Machinery
- Overhead Cranes
- Robotics
- Electronics Cabinets
- Sub-Sea Equipment
- Medical Equipment

Enidine provides energy absorption and vibration isolation solutions to meet the challenging demands of global industrial markets.

Enidine
7 Centre Drive
Orchard Park, New York 14127
Phone: 716-662-1900
Fax: 716-662-1909
Email: industrialsales@enidine.com
www.enidine.com

Enidine GmbH
Werkstrasse 5
D-64732, Bad Koenig, Germany
Phone: +49 6063 9314 0
Fax: +49 6063 9314 44
Email: info@enidine.eu
www.enidine.eu

ITT Control Technologies (APAC)
570 Xida Rd., Meicun, New Dist.,
Wuxi, China 214112
Phone: 86 510 8855 6197
Fax: 86 510 8855 6193
Email: enidineCN@itt.com
Email: controltechnologiesAPAC@itt.com
www.enidine.cn

Enidine Co. Ltd.
1-7-20 Maginu, Miyamae-ku, Kawasaki-shi
Kanagawa-kan, 216-0035
Japan
Phone: 81-44-870-8112
Fax: 81-44-870-8133
Email: support.enidinejp@itt.com
www.enidine.co.jp

ENI872R3 09/23