Shock Absorbers and Rate Controls

LDA
Solutions for Life

ENIDINE

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, ITT Enidine continues to strengthen its presence within marketplace.

Founded in 1966, ITT Enidine now has close to 400 employees located throughout the globe in the United States, Germany, France, Japan and China. 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 Inc. 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 at **+49 6063 9314 0**, or contact us via e-mail at **info@enidine.eu**.

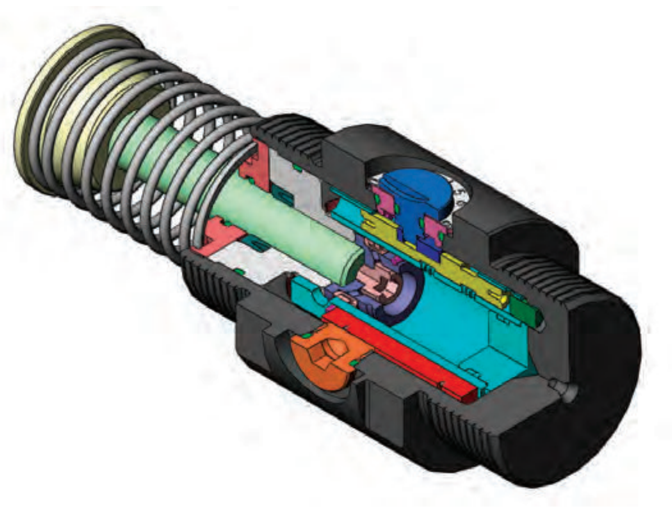
Products/Engineering/Technical Support

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 and vibration isolation product development.

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, ITT 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:

- ***3-D CAD Solid Modeling***
- ***3-D Soluble Prototype Printing Capabilities***
- ***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.

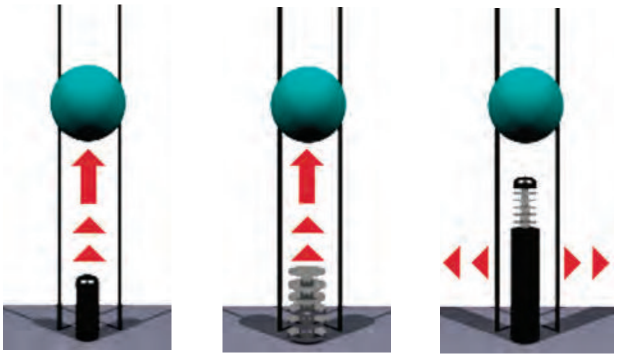
- Operating with lean manufacturing and cellular production, ITT Enidine produces higher quality custom and standard products with greater efficiency and within shorter lead times.
- An authorized Global Distribution Network is trained regularly by the ITT Enidine staff on new products and services ensuring they are better able to serve you.
- **New Enisize Sizing Portal provides our customers with the necessary sizing and design tools. www.enisize.com**
- Global operations in United States, Germany, France, China and Japan.
- 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 +49 6063 9314 0 or e-mail us at info@enidine.eu and let us get started today.

Overview

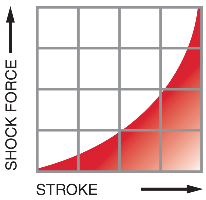
As companies strive to increase productivity by operating machinery at higher speeds, often the results are increased noise, damage to machinery/products, and excessive vibration. At the same time, safety and machine reliability are decreased. A variety of products are commonly used to solve these problems. 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:



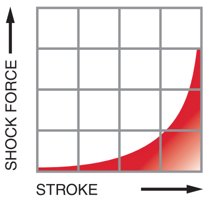
Rubber Bumper | **Metal Spring** | **ITT Enidine Shock Absorber**

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.

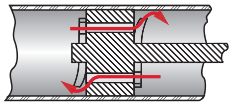
3. Improved Production Quality – Harmful side effects of motion, such as noise, vibration and damaging impacts, are moderated or eliminated so the quality of production is improved. Therefore, tolerances and fits are easier to maintain.

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.

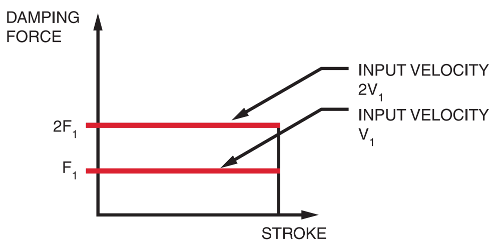
Automotive vs. Industrial Shock Absorbers

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

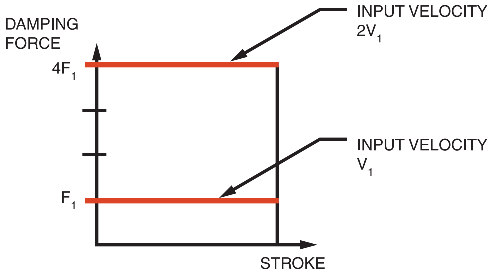


DEFLECTIVE BEAM ORIFICE

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.

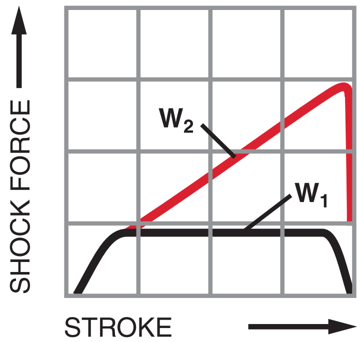
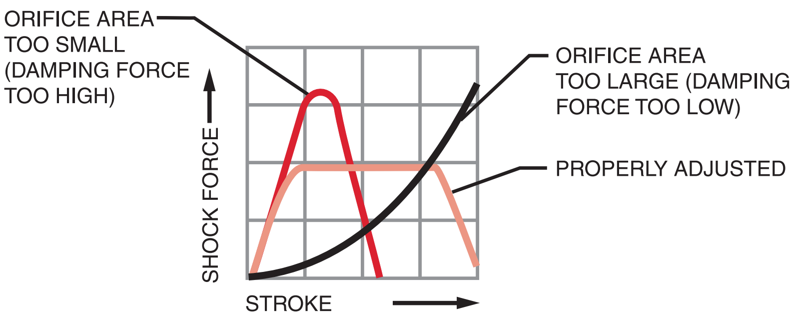


Figure 1

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.



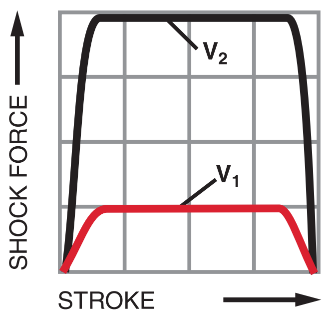


Figure 2

SHOCK ABSORBER SIZING

Follow the next six steps to manually size ITT 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 (Kg).
B. Velocity of the load upon impact with the shock absorber (m/s).
C. External (propelling) forces acting on the load (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.

$$E_K = \frac{I}{2}\omega^2 \quad \text{(rotary) or } E_K = \frac{1}{2}MV^2 \text{ (linear)}$$

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 TK or ECO 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 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 Kg
B. Desired velocity of the load m/s
C. External (propelling) force acting on the load 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 mm

NOTE: For rotary applications, please submit the application worksheet on page 175 to ITT Enidine for sizing.

STEP 2: Calculate the propelling force at the rate control in each direction damping is required. (See sizing examples on page 6-15).

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 99-104).

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.

SYMBOLS

a = Acceleration (m/s^2)
A = Width (m)
B = Thickness (m)
C = Number of cycles per hour
d = Cylinder bore diameter (mm)
D = Distance (m)
E_K = Kinetic energy (Nm)
E_T = Total energy per cycle (Nm/c), $E_K + E_W$
E_TC = Total energy to be absorbed per hour (Nm/hr)
E_W = Work or drive energy (Nm)
F_D = Propelling force (N)
F_P = Shock force (N)
H = Height (m)
Hp = Motor rating (kw)
I = Mass moment of inertia (kgm^2)
K = Radius of gyration (m)
L = Length (m)
P = Operating pressure (bar)
R_S = Mounting distance from pivot point (m)
S = Stroke of shock absorber (m)
t = Time (s)
T = Torque (Nm)
V = Impact velocity (m/s)
M = Mass (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 (rad/s)

USEFUL FORMULAS

1. To Determine Reaction Force

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

For Non-Adjustable ECO 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

$$F_D = \frac{\mathbf{3000} \times kw}{V}$$

4. To Determine Propelling Force of Pneumatic or Hydraulic Cylinders

$F_D = 0{,}\mathbf{0785} \times d^2 \times P$

5. Free Fall Applications

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

B. Kinetic Energy of Free Falling Weight:
$E_K = \mathbf{9{,}8} \times M \times H$

6. Deceleration

A. To Determine the Approximate Stroke

$$a = \frac{F_P - F_D}{M}$$

B. To Determine the Approximate Stroke (Conventional Damping Only)

$$S = \frac{E_K}{a \times M\ 0{,}85 - 0{,}15\ F_D}$$

*For ECO and TK Models:

$$S = \frac{E_K}{a \times M\ 0{,}5 - 0{,}5\ F_D}$$

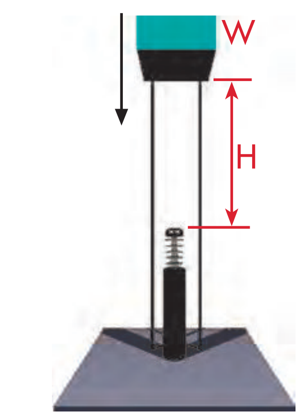
NOTE: Constants are printed in **bold**.

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

Shock Absorbers

EXAMPLE 1:

Vertical Free Falling Weight



STEP 1: Application Data

(M) Mass = 1 550 kg
(H) Height = 0,5 m
(C) Cycles/Hr = 2

STEP 2: Calculate kinetic energy

E_K = **9,8** x M x H
E_K = **9,8** x 1 550 x 0,5
E_K = 7 595 Nm

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

STEP 3: Calculate work energy

E_W = **9,8** M x S
E_W = **9,8** x 1 550 x 0,15
E_W = 2 278,5 Nm

STEP 4: Calculate total energy per cycle

E_T = E_K + E_W
E_T = 7 595 + 2 278,5
E_T = 9 873,5 Nm/c

STEP 5: Calculate total energy per hour

E_TC = E_T x C
E_TC = 9 873,5 x 2
E_TC = 19 747 Nm/hr

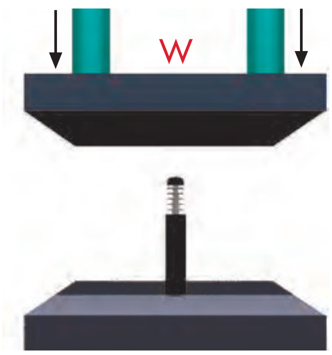
STEP 6: Calculate impact velocity and confirm selection

V = $\sqrt{\mathbf{19{,}6} \times H}$
V = $\sqrt{\mathbf{19{,}6} \times \mathbf{0{,}5}}$
V = 3,1 m/s

Model OEM 4.0M x 6 is adequate for this application.

EXAMPLE 2:

Vertical Moving Load with Propelling Force Downward



STEP 1: Application Data

(M) Mass = 1 550 kg
(V) Velocity = 2,0 m/s
(d) Cylinder bore dia. = 100mm
(P) Pressure = 5 bar
(C) Cycles/Hr = 200

STEP 2: Calculate kinetic energy

$$E_K = \frac{M}{2} \times V^2 = \frac{1\,550}{2} \times 2^2$$

E_K = 3 100 Nm

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

STEP 3: Calculate work energy

F_D = [**0,0785** x d^2 x P] + [**9,8** x M]
F_D = [**0,0785** x 100^2 x 5] + [**9,8** x 1 550]
F_D = 19 117 N
E_W = F_D x S
E_W = 19 117 x 0,1
E_W = 1 911,7 Nm

STEP 4: Calculate total energy per cycle

E_T = E_K + E_W
E_T = 3 100 + 1 911,7
E_T = 5 011,7 Nm/c

STEP 5: Calculate total energy per hour

E_TC = E_T x C
E_TC = 5 011,7 x 200
E_TC = 1 002 340 Nm/hr

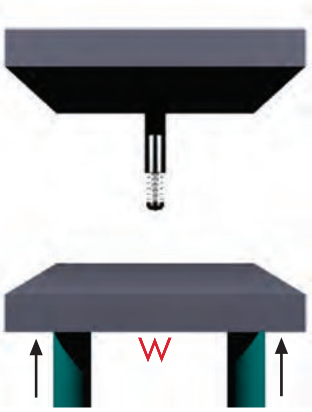
Model OEM 4.0M x 4 is adequate.

Shock Absorber Sizing Examples

Typical Shock Absorber Applications

EXAMPLE 3:

Vertical Moving Load with Propelling Force Upward



STEP 1: Application Data

(M) Mass = 1 550 kg
(V) Velocity = 2 m/s
(d) 2 Cylinders bore dia. = 150mm
(P) Operating pressure = 5 bar
(C) Cycles/Hr = 200

STEP 2: Calculate kinetic energy

$E_K = \frac{M}{2} \times V^2 = \frac{1\,550}{2} \times 2^2$

$E_K = 3\,100$ Nm

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

STEP 3: Calculate work energy

$F_D = 2 \times [\mathbf{0{,}0785} \times d^2 \times P] - [\mathbf{9{,}8} \times M]$

$F_D = 2 \times [\mathbf{0{,}0785} \times 150^2 \times 5] - [\mathbf{9{,}8} \times 1\,550]$

$F_D = 2\,472{,}5$ N

$E_W = F_D \times S$

$E_W = 2\,472{,}5 \times 0{,}125$

$E_W = 309$ Nm

STEP 4: Calculate total energy per cycle

$E_T = E_K + E_W$

$E_T = 3\,100 + 309$

$E_T = 3\,409$ Nm/c

STEP 5: Calculate total energy per hour

$E_TC = E_T \times C$

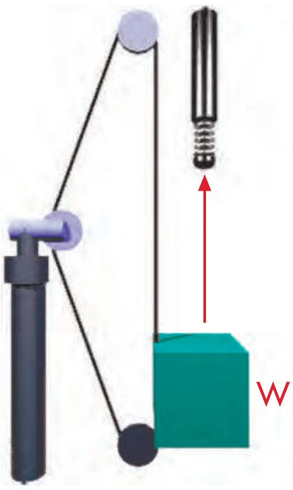
$E_TC = 3\,409 \times 200$

$E_TC = 681\,800$ Nm/hr

Model OEM 3.0M x 5 is adequate.

EXAMPLE 4:

Vertical Moving Load with Propelling Force from Motor



STEP 1: Application Data

(M) Mass = 90 kg
(V) Velocity = 1,5 m/s
(kW) Motor rating = 1 kW
(C) Cycles/Hr = 100

STEP 2: Calculate kinetic energy

$E_K = \frac{M}{2} \times V^2 = \frac{90}{2} \times 1{,}5^2$

$E_K = 101$ Nm

CASE A: UP

STEP 3: Calculate work energy

$F_D = \frac{\mathbf{3\,000} \times kW}{V} - \mathbf{9{,}8} \times M$

$F_D = \frac{\mathbf{3\,000} \times 1}{1{,}5} - 882$

$F_D = 1\,118$ N

Assume Model OEM 1.25 x 2 is adequate (Page 26).

$E_W = F_D \times S$

$E_W = 1\,118 \times 0{,}5$

$E_W = 56$ Nm

STEP 4: Calculate total energy per cycle

$E_T = E_K + E_W$

$E_T = 101 + 56$

$E_T = 157$ Nm/c

STEP 5: Calculate total energy per hour

$E_TC = E_T \times C$

$E_TC = 157 \times 100$

$E_TC = 15\,700$ Nm/hr

Model OEM 1.25M x 2 is adequate.

CASE B: DOWN

STEP 3: Calculate work energy

$F_D = \frac{\mathbf{3\,000} \times kW}{V} + \mathbf{9{,}8} \times M$

$F_D = \frac{\mathbf{3\,000} \times 1}{1{,}5} + 882$

$F_D = 2\,882$ N

Assume Model OEMXT 2.0M x 2 is adequate (Page 30).

$E_W = F_D \times S$

$E_W = 2\,882 \times 0{,}05$

$E_W = 144$ Nm

STEP 4: Calculate total energy per cycle

$E_T = E_K + E_W$

$E_T = 101 + 144$

$E_T = 245$ Nm/c

STEP 5: Calculate total energy per hour

$E_TC = E_T \times C$

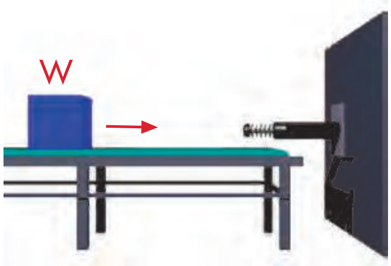
$E_TC = 245 \times 100$

$E_TC = 24\,500$ Nm/hr

Model OEMXT 2.0M x 2 is adequate.

EXAMPLE 5:

Horizontal Moving Load



STEP 1: Application Data

(M) Mass = 900 kg
(V) Velocity = 1,5 m/s
(C) Cycles/Hr = 200

STEP 2: Calculate kinetic energy

$E_K = \frac{M}{2} \times V^2$

$E_K = \frac{900}{2} \times 1{,}5^2$

$E_K = 1\,012{,}5$ Nm

Assume Model OEMXT 2.0M x 2 is adequate (Page 30).

STEP 3: Calculate work energy: N/A

STEP 4: Calculate total energy per cycle

$E_T = E_K = 1\,012{,}5$ Nm/c

STEP 5: Calculate total energy per hour

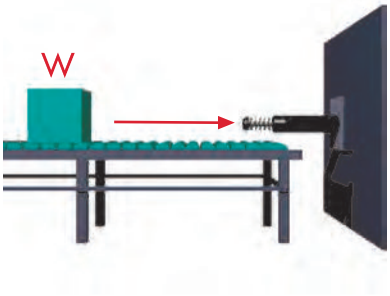
$E_TC = E_T \times C$

$E_TC = 1\,012{,}5 \times 200$

$E_TC = 202\,500$ Nm/hr

Model OEMXT 2.0M x 2 is adequate.

EXAMPLE 6:
Horizontal Moving Load with Propelling Force



STEP 1: Application Data
(M) Mass = 900 kg
(V) Velocity = 1,5 m/s
(d) Cylinder bore dia. = 75mm
(P) Operating pressure = 5 bar
(C) Cycles/Hr = 200

STEP 2: Calculate kinetic energy

$E_K = \frac{M}{2} \times V^2$

$E_K = \frac{900}{2} \times 1{,}5^2$

$E_K = 1\,012{,}5$ Nm

Assume Model OEMXT 2.0M x 2 is adequate (Page 30).

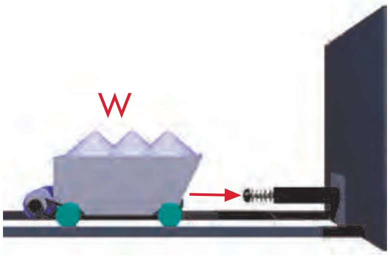
STEP 3: Calculate work energy
$F_D = \mathbf{0,0785} \times d^2 \times P$
$F_D = \mathbf{0,0785} \times 75^2 \times 5$
$F_D = 2\,208{,}9$ N
$E_W = F_D \times S$
$E_W = 2\,208{,}9 \times 0{,}05$
$E_W = 110$ Nm/c

STEP 4: Calculate total energy per cycle
$E_T = E_K + E_W$
$E_T = 1\,012{,}5 + 110$
$E_T = 1\,122{,}5$ Nm/c

STEP 5: Calculate total energy per hour
$E_TC = E_T \times C$
$E_TC = 1\,122{,}5 \times 200$
$E_TC = 224\,500$ Nm/hr

Model OEMXT 2.0M x 2 is adequate.

EXAMPLE 7:
Horizontal Moving Load, Motor Driven



STEP 1: Application Data
(M) Mass = 1 000 kg
(V) Velocity = 1,5 m/s
(kW) Motor rating = 1 kW
(C) Cycles/Hr = 120

STEP 2: Calculate kinetic energy

$E_K = \frac{M}{2} \times V^2$

$E_K = \frac{1\,000}{2} \times 1{,}5^2$

$E_K = 1\,125$ Nm

Assume Model OEMXT 2.0M x 2 is adequate (Page 30).

STEP 3: Calculate work energy

$F_D = \frac{\mathbf{3\,000} \times kW}{V}$

$F_D = \frac{\mathbf{3\,000} \times 1}{1{,}5}$

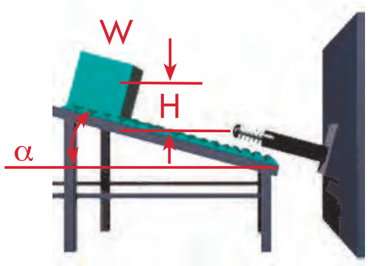
$F_D = 2\,000$ N
$E_W = F_D \times S$
$E_W = 2\,000 \times 0{,}05$
$E_W = 100$ Nm

STEP 4: Calculate total energy per cycle
$E_T = E_K + E_W$
$E_T = 1\,125 + 100$
$E_T = 1\,225$ Nm/c

STEP 5: Calculate total energy per hour
$E_TC = E_T \times C$
$E_TC = 1\,225 \times 120$
$E_TC = 147\,000$ Nm/hr

Model OEMXT 2.0M x 2 is adequate.

EXAMPLE 8:
Free Moving Load Down an Inclined Plane



STEP 1: Application Data
(M) Mass = 250 kg
(H) Height = 0,2 m
(α) Angle of incline = 30°
(C) Cycles/Hr = 250

STEP 2: Calculate kinetic energy
$E_K = \mathbf{9,8} \times M \times H$
$E_K = \mathbf{9,8} \times 250 \times 0{,}2$
$E_K = 490$ Nm

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

STEP 3: Calculate work energy
$F_D = \mathbf{9,8} \times M \times \sin\alpha$
$F_D = \mathbf{9,8} \times 250 \times 0{,}5$
$F_D = 1\,225$ N
$E_W = F_D \times S$
$E_W = 1\,225 \times 0{,}075$
$E_W = 91{,}9$ Nm

STEP 4: Calculate total energy per cycle
$E_T = E_K + E_W$
$E_T = 490 + 91{,}9$
$E_T = 581{,}9$ Nm/c

STEP 5: Calculate total energy per hour
$E_TC = E_T \times C$
$E_TC = 581{,}9 \times 250$
$E_TC = 145\,475$ Nm/hr

STEP 6: Calculate impact velocity and confirm selection
$V = \sqrt{\mathbf{19,6} \times H}$
$V = \sqrt{\mathbf{19,6} \times 0{,}2} = 2{,}0$ m/s

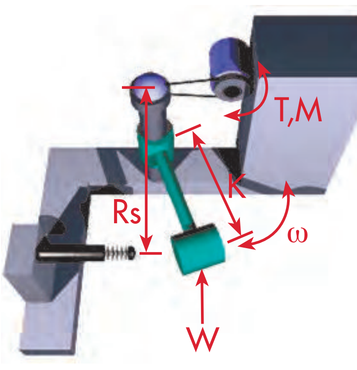
Model OEMXT 1.5M x 3 is adequate.

Shock Absorber Sizing Examples

Typical Shock Absorber Applications

EXAMPLE 9:

Horizontal Rotating Mass



STEP 1: Application Data

(M) Mass = 90 kg
(ω) Angular velocity = 1,5 rad/s
(T) Torque = 120 Nm
(K) Radius of gyration = 0,4 m
(R_S) Mounting radius = 0,5 m
(C) Cycles/Hr = 120

STEP 2: Calculate kinetic energy

$I = M \times K^2$
$I = 90 \times 0{,}4^2$
$I = 14{,}4\ kgm^2$
$E_K = \frac{I \times \omega^2}{2}$
$E_K = \frac{14{,}4 \times 1{,}5^2}{2}$
$E_K = 16{,}2\ Nm$

Assume Model STH 0.5M is adequate (Page 41).

STEP 3: Calculate work energy

$F_D = \frac{T}{R_S}$
$F_D = \frac{120}{0{,}5}$
$F_D = 240\ N$
$E_W = F_D \times S$
$E_W = 240 \times 0{,}013$
$E_W = 3\ Nm$

STEP 4: Calculate total energy per cycle

$E_T = E_K + E_W$
$E_T = 16{,}2 + 3$
$E_T = 19{,}2\ Nm/c$

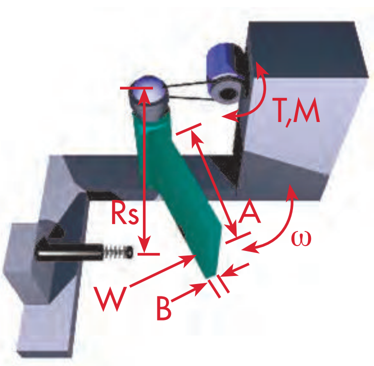
STEP 5: Calculate total energy per hour

$E_TC = E_T \times C$
$E_TC = 19{,}2 \times 120$
$E_TC = 2\,304\ Nm/hr$

Model STH 0.5M is adequate.

EXAMPLE 10:

Horizontal Rotating Door



STEP 1: Application Data

(M) Mass = 25 kg
(ω) Angular velocity = 2,5 rad/s
(T) Torque = 10 Nm
(R_S) Mounting radius = 0,5 m
(A) Width = 1,0 m
(B) Thickness = 0,1 m
(C) Cycles/Hr = 250

STEP 2: Calculate kinetic energy

$K = 0{,}289 \times \sqrt{\mathbf{4} \times A^2 + B^2}$
$K = 0{,}289 \times \sqrt{\mathbf{4} \times 1{,}0^2 + 0{,}1^2}$
$K = 0{,}58\ m$
$I = M \times K^2$
$I = 25 \times 0{,}58^2$
$I = 8{,}4\ kgm^2$
$E_K = \frac{I \times \omega^2}{2}$
$E_K = \frac{8{,}4 \times 2{,}5^2}{2}$
$E_K = 26{,}3\ Nm$

Assume Model OEM .5M is adequate (Page 19).

STEP 3: Calculate work energy

$F_D = \frac{T}{R_S}$
$F_D = \frac{10}{0{,}5}$
$F_D = 20\ N$
$E_W = F_D \times S$
$E_W = 20 \times 0{,}025$
$E_W = 0{,}5\ Nm$

STEP 4: Calculate total energy per cycle

$E_T = E_K + E_W$
$E_T = 26{,}3 + 0{,}5$
$E_T = 26{,}8\ Nm/c$

STEP 5: Calculate total energy per hour

$E_TC = E_T \times C$
$E_TC = 26{,}8 \times 250$
$E_TC = 6\,700\ Nm/hr$

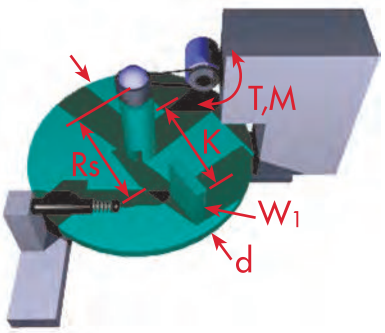
STEP 6: Calculate impact velocity and confirm selection

$V = R_S \times \omega$
$V = 0{,}5 \times 2{,}5$
$V = 1{,}25\ m/s$

Model OEM 0.5M is adequate.

EXAMPLE 11:

Horizontal Moving Load, Rotary Table Motor Driven with Additional Load Installed



STEP 1: Application Data

(M) Mass = 200 kg
(M_1) Installed load = 50 kg
Rotational speed = 10 RPM
(T) Torque = 250 Nm
Rotary table dia. = 0,5 m
(K_{Load}) Radius of gyration = 0,2 m
(R_S) Mounting radius = 0,225 m
(C) Cycles/Hr = 1

Step 2: Calculate kinetic energy

To convert RPM to rad/s, multiply by **0,1047**
$\omega = RPM \times \mathbf{0{,}1047}$
$\omega = 10 \times \mathbf{0{,}1047}$
$\omega = \mathbf{1{,}047}\ rad/s$
$I = M \times K$

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.

$K_{Table} = \text{Table Radius} \times \mathbf{0{,}707}$
$K_{Table} = 0{,}25 \times \mathbf{0{,}707} = 0{,}176\ m$
$I_{Table} = M \times K^2_{Table}$
$I_{Table} = 200 \times 0{,}176^2$
$I_{Table} = 6{,}2\ kgm^2$
$I_{Load} = M_1 \times K^2_{Load}$
$I_{Load} = 50 \times (0{,}20)^2 = 2\ kgm^2$
$E_K = \frac{(I_{Table} + I_{Load}) \times \omega^2}{2}$
$E_K = \frac{(6{,}2 + 2) \times 1{,}047^2}{2}$
$E_K = 4{,}5\ Nm$

Assume model ECO 50M-4 is adequate (Page 47).

STEP 3: Calculate work energy

$F_D = \frac{T}{R_S} = \frac{250}{0{,}225} = 1\,111{,}1\ N$
$E_W = F_D \times S = 1\,111{,}1 \times 0{,}022$
$E_W = 24{,}4\ Nm$

STEP 4: Calculate total energy per cycle

$E_T = E_K + E_W$
$E_T = 4{,}5 + 24{,}4$
$E_T = 28{,}9\ Nm/c$

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

STEP 6: Calculate impact velocity and confirm selection

$V = R_S \times \omega$
$V = 0{,}225 \times 1{,}047$
$V = 0{,}24\ m/s$

From ECO Sizing Graph.
Model ECO 50M-4 is adequate.

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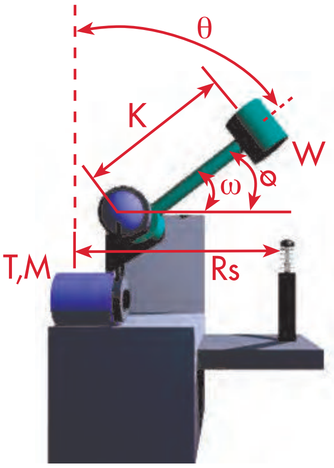
EXAMPLE 12:
Vertical Motor Driven Rotating Arm with Attached Load

CASE A–Load Aided by Gravity



STEP 1: Application Data

(M) Mass = 50 kg
(ω) Angular velocity = 2 rad/s
(T) Torque = 350 Nm
(Ø) Angle of rotation = 30°
(K_{Load}) Radius of gyration = 0,6 m
(R_S) Mounting radius = 0,4 m
(C) Cycles/Hr = 1

STEP 2: Calculate kinetic energy

$I = M \times K^2 = 50 \times 0{,}6^2$
$I = 18\ kgm^2$

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

$E_K = \dfrac{18 \times 2^2}{2}$

$E_K = 36\ Nm$

Assume Model OEM 1.0 is adequate (Page 21).

CASE A

STEP 3: Calculate work energy

$F_D = \dfrac{T + (\mathbf{9{,}8} \times M \times K \times \sin\theta)}{R_S}$

$F_D = \dfrac{350 + (\mathbf{9{,}8} \times 50 \times 0{,}6 \times 0{,}5)}{0{,}4}$

$F_D = 1\,242{,}5\ N$
$E_W = F_D \times S$
$E_W = 1\,242{,}5 \times 0{,}025$
$E_W = 31{,}1\ N$

STEP 4: Calculate total energy per cycle

$E_T = E_K + E_W$
$E_T = 36 + 31{,}1$
$E_T = 67{,}1\ Nm/c$

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

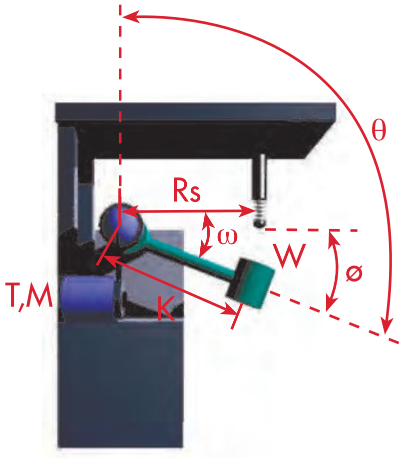
STEP 6: Calculate impact velocity and confirm selection.

$V = R_S \times \omega$
$V = 0{,}4 \times 2$
$V = 0{,}8\ m/s$

Model LROEM 1.0M 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

(M) Mass = 50 kg
(ω) Angular velocity = 2 rad/s
(T) Torque = 350 Nm
(Ø) Angle of rotation = 30°
(K_{Load}) Radius of gyration = 0,6 m
(R_S) Mounting radius = 0,4 m
(C) Cycles/Hr = 1

STEP 2: Calculate kinetic energy

$I = M \times K^2 = 50 \times 0{,}6^2$
$I = 18\ kgm^2$

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

$E_K = \dfrac{18 \times 2^2}{2}$

$E_K = 36\ Nm$

Assume Model OEM 1.0M is adequate (Page 21).

CASE B

STEP 3: Calculate work energy

$F_D = \dfrac{T - (\mathbf{9{,}8} \times M \times K \times \sin\theta)}{R_S}$

$F_D = \dfrac{350 - (\mathbf{9{,}8} \times 50 \times 0{,}6 \times 0{,}5)}{0{,}4}$

$F_D = 507{,}5\ N$
$E_W = F_D \times S$
$E_W = 507{,}5 \times 0{,}025$
$E_W = 12{,}7\ Nm$

STEP 4: Calculate total energy per cycle

$E_T = E_K + E_W$
$E_T = 36 + 12{,}7$
$E_T = 48{,}7\ Nm/c$

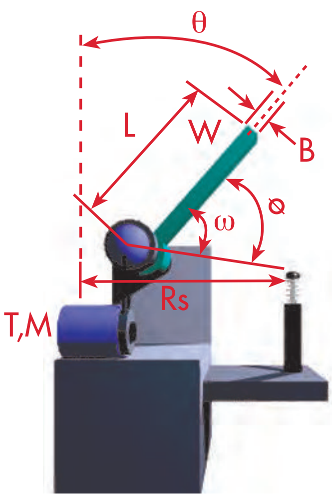
STEP 5: Calculate total energy per hour: not applicable, C=1

STEP 6: Calculate impact velocity and confirm selection

$V = R_S \times \omega$
$V = 0{,}4 \times 2$
$V = 0{,}8\ m/s$

Model OEM 1.0M is adequate.

EXAMPLE 14:
Vertical Rotating Beam



STEP 1: Application Data

(M) Mass= 245 kg
(ω) Angular velocity = 3,5 rad/s
(T) Torque = 30 Nm
(θ) Starting point from true vertical = 20°
(Ø) Angle of rotation = 50°
(R_S) Mounting radius = 0,5 m
(B) Thickness = 0,06 m
(L) Length = 0,6 m
(C) Cycles/Hr = 1

STEP 2: Calculate kinetic energy

$K = \mathbf{0{,}289} \times \sqrt{\mathbf{4} \times L^2 + B^2}$
$K = \mathbf{0{,}289} \times \sqrt{\mathbf{4} \times 0{,}6^2 + 0{,}06^2}$
$K = 0{,}35\ m$
$I = M \times K^2 = 245 \times 0{,}35^2$
$I = 30\ kgm^2$

$E_K = \dfrac{I \times \omega^2}{2} = \dfrac{30 \times 3{,}5^2}{2} = 184\ Nm$

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

STEP 3:

$F_D = \dfrac{T + [\mathbf{9{,}8} \times M \times K \times \sin(\theta + Ø)]}{R_S}$

$F_D = \dfrac{30 + [\mathbf{9{,}8} \times 245 \times 0{,}35 \times \sin(20° + 50°)]}{0{,}5}$

$F_D = 1\,640\ N$

$E_W = F_D \times S$
$E_W = 1\,640 \times 0{,}05$
$E_W = 82\ Nm$

STEP 4: Calculate total energy per cycle

$E_T = E_K + E_W$
$E_T = 184 + 82$
$E_T = 266\ Nm/c$

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

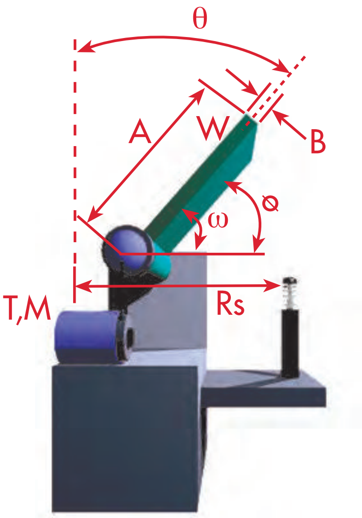
STEP 6: Calculate impact velocity and confirm selection

$V = R_S \times \omega$
$V = 0{,}5 \times 3{,}5$
$V = 1{,}75\ m/s$

Model OEMXT 1.5M x 2 is adequate.

Shock Absorber Sizing Examples
Typical Shock Absorber Applications

EXAMPLE 15:
Vertical Rotating Lid



STEP 1: Application Data
(M) Mass = 910 kg
(ω) Angular velocity = 2 rad/s
(kW) Motor rating = 0,20 kW
(θ) Starting point from true vertical = 30°
(Ø) Angle of rotation = 60°
(R_S) Mounting radius = 0,8 m
(A) Width = 1,5 m
(B) Thickness = 0,03 m
(C) Cycle/Hr = 1

STEP 2: Calculate kinetic energy
$K = \mathbf{0,289} \times \sqrt{\mathbf{4} \times A^2 + B^2}$
$K = \mathbf{0,289} \times \sqrt{\mathbf{4} \times 1,50^2 + 0,03^2}$
$K = 0,87$ m

$I = M \times K^2 = 910 \times 0,87^2$
$I = 688,8$ kgm²
$E_K = \dfrac{I \times \omega^2}{2} = \dfrac{688,8 \times 2^2}{2}$
$E_K = 1\,377,6$ Nm
Assume Model OEM 3.0M x 2 is adequate (Page 21).

STEP 3: Calculate work energy
$T = \dfrac{\mathbf{3\,000} \times kW}{\omega}$
$T = \dfrac{\mathbf{3\,000} \times 0,20}{2} = 300$ Nm
$F_D = \dfrac{T + (\mathbf{9,8} \times M \times K \times \text{Sin}\,(\theta + Ø))}{R_S}$
$F_D = \dfrac{300 + (\mathbf{9,8} \times 910 \times 0,87 \times \text{Sin}\,(60° + 30°))}{0,8}$
$F_D = 10\,073$ N
$E_W = F_D \times S$
$E_W = 10\,073 \text{ N} \times 0,05$
$E_W = 503,7$ Nm

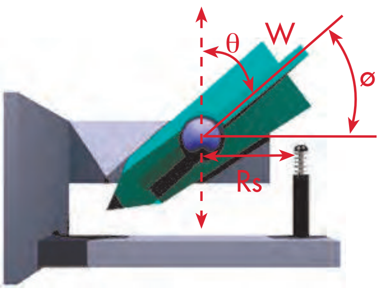
STEP 4: Calculate total energy per cycle
$E_T = E_K + E_W$
$E_T = 1\,377,6 + 503,7$
$E_T = 1\,881,3$ Nm/c

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

STEP 6: Calculate impact velocity and confirm selection
$V = R_S \times \omega$
$V = 0,8 \times 2$
$V = 1,6$ m/s

Model OEM 3.0M x 2 is adequate.

EXAMPLE 16:
Vertical Rotation with Known Intertia Aided by Gravity



STEP 1: Application Data
(M) Mass = 100 kg
(I) Known Intertia = 100 kgm²
(C/G) Center-of-Gravity = 305 mm
(θ) Starting point from true vertical = 60°
(Ø) Angle of rotation at impact = 30°
(R_S) Mounting radius = 254 mm
(C) Cycles/Hr = 1

STEP 2: Calculate kinetic energy
$H = C/G \times [\text{Cos}(\theta) - \text{Cos}(Ø+\theta)]$
$H = 0,305 \times [\text{Cos}(60°) - \text{Cos}(30°+60°)]$
$E_K = \mathbf{9,8} \times M \times H$
$E_K = \mathbf{9,8} \times 100 \times 0,5$
$E_K = 149,5$ Nm

STEP 3: Calculate work energy
$F_D = (\mathbf{9,8} \times M \times C/G \times \text{Sin}\,(\theta + Ø))/R_S$
$F_D = (\mathbf{9,8} \times 100 \times 0,305 \times \text{Sin}\,(60° + 30°))/0,254$
$F_D = 1176,8$ N
$E_W = F_D \times S = 1176,8 \times 0,025 = 29,4$ Nm

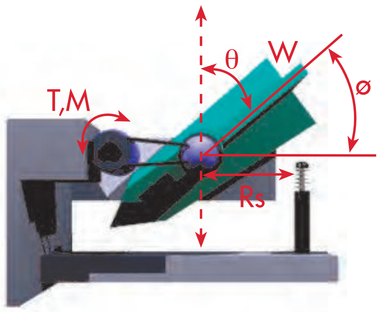
STEP 4: Calculate total energy per cycle
$E_T = E_K + E_W = 149,5 + 29,4$
$E_T = 178,9$ Nm/c

STEP 5: Calculate total energy per hour: not applicable, C=1
$E_TC = E_T \times C$
$E_TC = 178,9 \times 1$
$E_TC = 178,9$ Nm/hr

STEP 6: Calculate impact velocity and confirm selection
$\omega = \sqrt{(2 \times E_K)/I}$
$\omega = \sqrt{(2 \times 149,5)/100)} = 1,7$ rad/s
$V = R_S \times \omega = 0,254 \times 1,7 = 0,44$ m/s

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

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



STEP 1: Application Data
(M) Mass = 100 kg
(ω) Angular Velocity = 2 rad/s
(T) Torque = 310 Nm
(I) Known Intertia = 100 kgm²
(C/G) Center-of-Gravity = 305 mm
(θ) Starting point from true vertical = 60°
(Ø) Angle of rotation at impact = 30°
(R_S) Mounting radius = 254 mm
(C) Cycles/Hr = 100

STEP 2: Calculate kinetic energy
$E_K = (I \times \omega^2)/2$
$E_K = (100 \times 2^2)/2$
$E_K = 200$ Nm

STEP 3: Calculate work energy
$F_D = [T + (\mathbf{9,8} \times M \times C/G \times \text{Sin}\,(\theta + Ø))]/R_S$
$F_D = [310 + (\mathbf{9,8} \times 100 \times 0,305 \times \text{Sin}\,(60°+30°)]/0,254$
$F_D = 2\,397,2$ N
$E_W = F_D \times S = 2\,397 \times 0,025 = 59,9$ Nm

STEP 4: Calculate total energy per cycle
$E_T = E_K + E_W = 200 + 59,9$
$E_T = 259,9$ Nm/c

STEP 5: Calculate total energy per hour: not applicable, C=1
$E_TC = E_T \times C$
$E_TC = 259,9 \times 100$
$E_TC = 25\,990$ Nm/hr

STEP 6: Calculate impact velocity and confirm selection
$V = R_S \times \omega = 0,254 \times 2 = 0,51$ m/s

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

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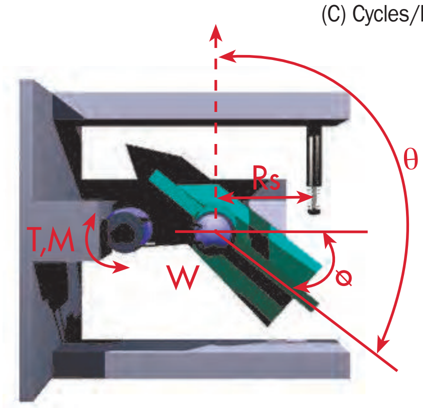
EXAMPLE 18:
Vertical Rotation with Known Intertia Opposing Gravity (w/Torque)



STEP 1: Application Data
(M) Mass = 100 kg
(ω) Angular Velocity = 2 rad/s
(T) Torque = 310 Nm
(I) Known Intertia = 100 kgm²
(C/G) Center-of-Gravity = 305 mm
(θ) Starting point from true vertical = 120°
(Ø) Angle of rotation at impact = 30°
(R_S) Mounting radius = 254 mm
(C) Cycles/Hr = 100

STEP 2: Calculate kinetic energy
$E_K = (I \times \omega^2)/2$
$E_K = (100x\ 2^2)/2$
$E_K = 200$ Nm

STEP 3: Calculate work energy
$F_D = [T - (\mathbf{9{,}8} \times M \times C/G \times \sin(\theta - Ø))]/R_S$
$F_D = [310 - (\mathbf{9{,}8} \times 100 \times 0{,}305 \times \sin(120° - 30°))]/0{,}254$
$F_D = 43{,}7$ N
$E_W = F_D \times S = 43{,}7 \times 0{,}025 = 1{,}1$ Nm

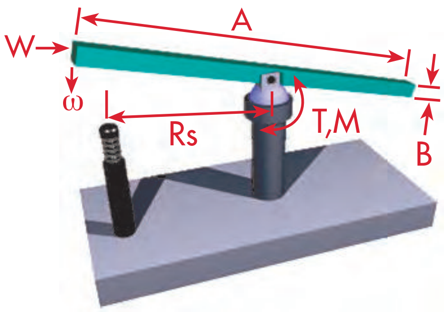
STEP 4: Calculate total energy per cycle
$E_T = E_K + E_W = 200 + 1{,}1$
$E_T = 201{,}1$ Nm/c

STEP 5: Calculate total energy per hour: not applicable, C=1
$E_TC = E_T \times C$
$E_TC = 201{,}1 \times 100$
$E_TC = 20\,110$ Nm/hr

STEP 6: Calculate impact velocity and confirm selection
$V = R_S \times \omega = 0{,}254 \times 2 = 0{,}51$ m/s

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

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



STEP 1: Application Data
(M) Mass = 100 kg
(ω) Angular velocity = 2 rad./s
(T) Torque = 310 Nm
(A) Length = 1,016 mm
(R_S) Mounting radius = 254 mm
(B) Thickness = 50,8 mm
(C) Cycles/Hr = 100

STEP 2: Calculate kinetic energy
$K = \mathbf{0{,}289} \times \sqrt{A^2 + B^2}$
$K = \mathbf{0{,}289} \times \sqrt{1{,}016^2 + 0{,}0508^2}$
$= 0{,}29$ m
$I = M \times K^2$
$I = 100 \times 0{,}29^2 = 8{,}6$ kgm²
$E_K = (I \times \omega^2)/2$
$E_K = (8{,}6 \times 2^2)/2$
$E_K = 17{,}2$ Nm

Assume Model OEM 1.0 is adequate (Page 21).

STEP 3: Calculate work energy
$F_D = T/R_S$
$F_D = 310/0{,}254$
$F_D = 1\,220{,}5$ N
$E_W = F_D \times S = 1\,220{,}5 \times 0{,}025$
$= 30{,}5$ Nm

STEP 4: Calculate total energy per cycle
$E_T = E_K + E_W = 17{,}2 + 30{,}5$
$E_T = 47{,}7$ Nm/c

STEP 5: Calculate total energy per hour
$E_TC = E_T \times C$
$E_TC = 47{,}7 \times 100$
$E_TC = 4\,770$ Nm/hr

STEP 6: Calculate impact velocity and confirm selection
$V = R_S \times \omega = 0{,}254 \times 2 = 0{,}51$ m/s

Model OEM 1.0M is adequate.

Shock Absorber Sizing Examples

Typical Shock Absorber and Crane Applications

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

Crane A		Per Buffer
Propelling Force Crane	kN	
Propelling Force Trolley	kN	
Weight of Crane (W_a)	t	
Weight of Trolley (W_{ta})	t	
Crane Velocity (V_a)	m/s	
Trolley Velocity (V_{ta})	m/s	

Crane B		Per Buffer
Propelling Force Crane	kN	
Propelling Force Trolley	kN	
Weight of Crane (W_a)	t	
Weight of Trolley (W_{ta})	t	
Crane Velocity (V_a)	m/s	
Trolley Velocity (V_{ta})	m/s	

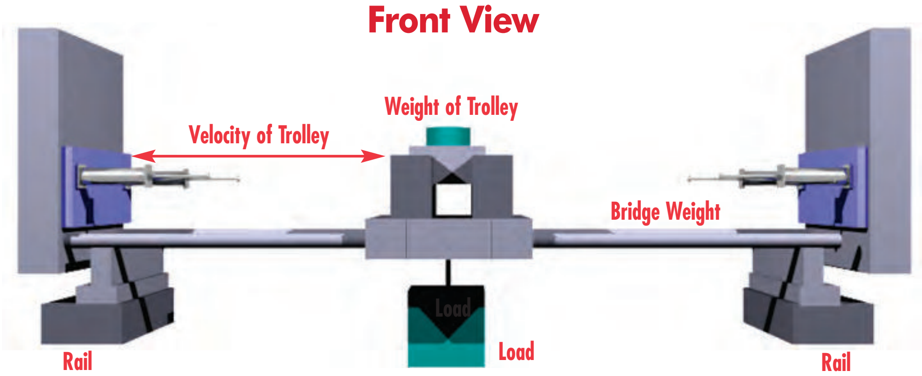
Crane C		Per Buffer
Propelling Force Crane	kN	
Propelling Force Trolley	kN	
Weight of Crane (W_a)	t	
Weight of Trolley (W_{ta})	t	
Crane Velocity (V_a)	m/s	
Trolley Velocity (V_{ta})	m/s	

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

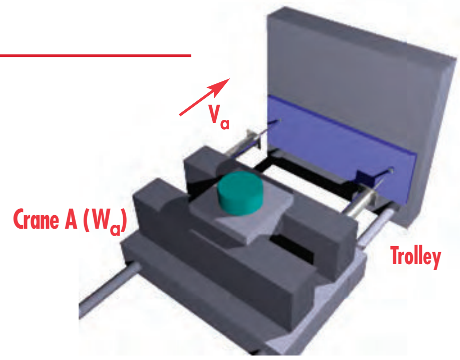
Crane A against Solid Stop

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:

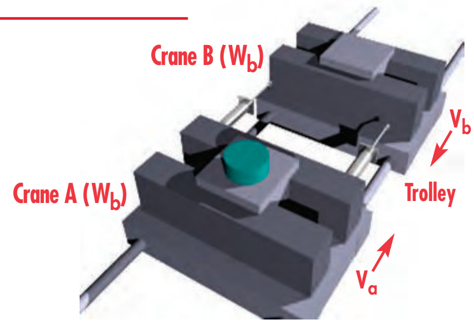
$V_r = V_a + V_b$

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:

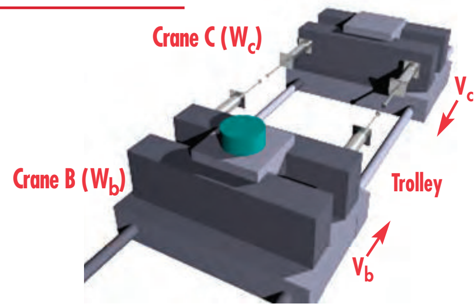
$$V_r = \frac{V_b + V_c}{2}$$

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

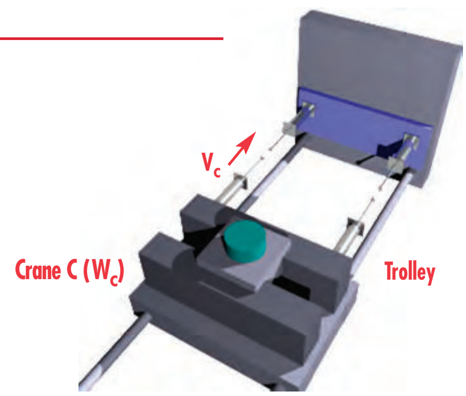
Velocity:

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

Impact weight per buffer:

$W_1 = W_c + 1{,}8\ (W_{tc})$

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



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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.

Calculation Example for Harbor Cranes as Application 1

Given Values

Total Weight of Bridge:	**380 t**
Weight of Trolley:	**45 t**
Crane Velocity:	**1,5 m/s**
Required Stroke:	**600 mm**
Trolley Velocity:	**4,0 m/s**
Required Stroke:	**1 000 m**

Determination of the Maximum Impact Mass W_d per Buffer

$$W_d = \frac{Wa + 1{,}8\ W{+}a}{\text{Total Number of Shocks}}$$

$$W_d = \frac{380\ t + (1{,}8)45\ t}{2}$$

$$\mathbf{W_d = 230.5\ t}$$

Determine Size of Shock Absorber for Crane

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

$$E_K = \frac{230.5}{2} \bullet (1{,}5\ m/s)^2$$

$$\mathbf{E_K = 259\ kN}$$

$V_r = V_A$ (Application 1)

E_K = Kinetic Energy

η = Efficiency

Selecting for required 600mm stroke:
HD 5.0 x 24, maximum shock force ca. 460 kN = $F_s = \frac{E_K}{s \bullet \eta}$

Determine Size of Shock Absorber for Trolley

M_D = Trolley Mass per Shock Absorber

$$M_D = \frac{45\ t}{2}$$

$$\mathbf{M_D = 22{,}5\ t}$$

$V_r = V_A$ Application 1

$$E_K = \frac{M_D}{2} \bullet V_r^2$$

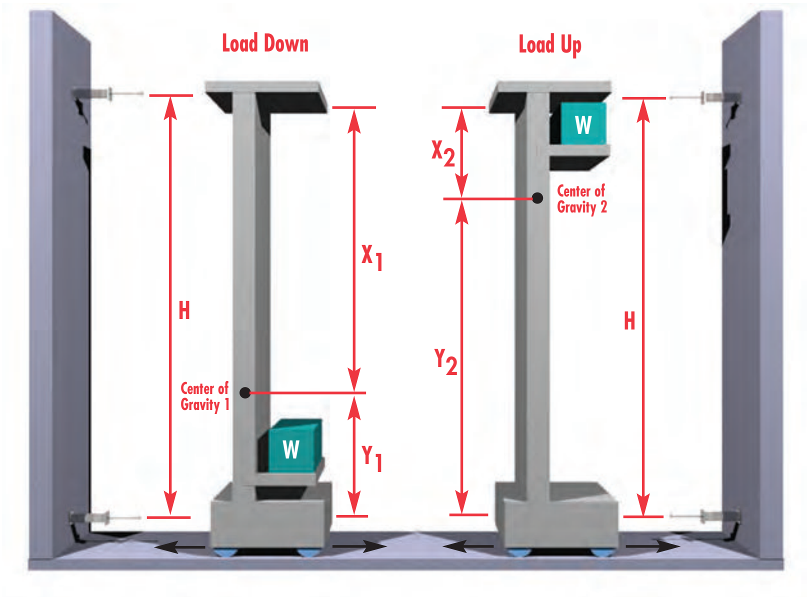
$$E_K = \frac{22{,}5\ t}{2} \bullet (4\ m/s)^2$$

$$\mathbf{E_K = 180\ kNm}$$

Selecting for required 1 000 mm stroke:
HDN 4.0 x 40, maximum shock force ca. 212 kN = $F_s = \frac{E_K}{s \bullet \eta}$

Overview

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



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 = 20 m
Distance to C of G1 - Upper:	X_1 = 15 m
Distance to C of G1 - Lower:	Y_1 = 5 m
Distance to C of G2 - Upper:	X_2 = 7 m
Distance to C of G1 - Lower:	Y_2 = 13 m
Total Weight:	W = 20 t

Calculation for Lower Shock Absorbers

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

$$W_{max\ d} = \frac{15\ m}{20\ m} \bullet 20\ t \qquad W_{max\ d} = \frac{7\ m}{20\ m} \bullet 20\ t$$

$$W_{max\ d} = 15\ t \qquad W_{max\ d} = 7\ t$$

Calculation for Upper Shock Absorbers

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

$$W_{max\ d} = \frac{5\ m}{20\ m} \bullet 20\ t \qquad W_{max\ d} = \frac{13\ m}{20\ m} \bullet 20\ t$$

$$W_{max\ d} = 5\ t \qquad W_{max\ d} = 13\ t$$

Shock Absorber Selection

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

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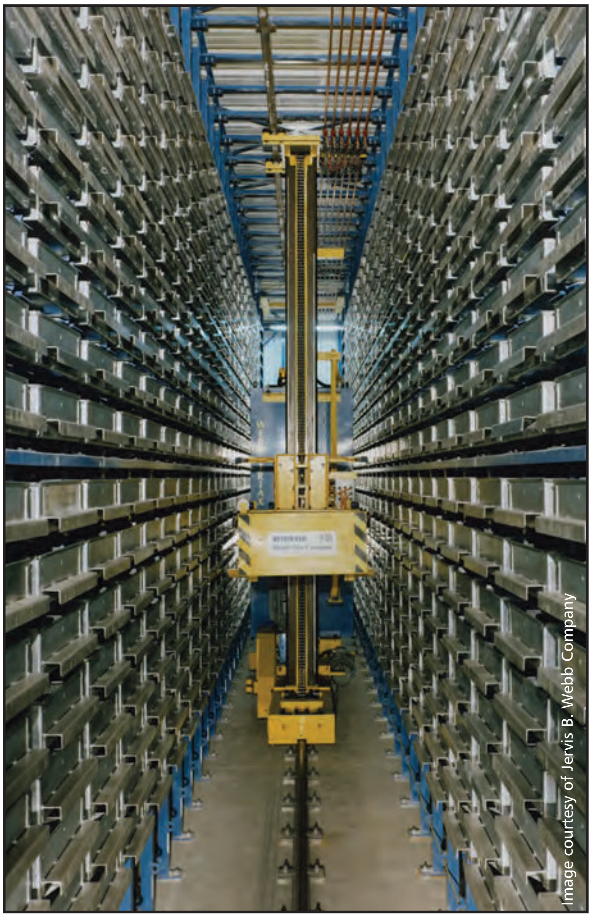
Overhead Crane Applications

Cargo Crane Applications



Stacker Carne Applications

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