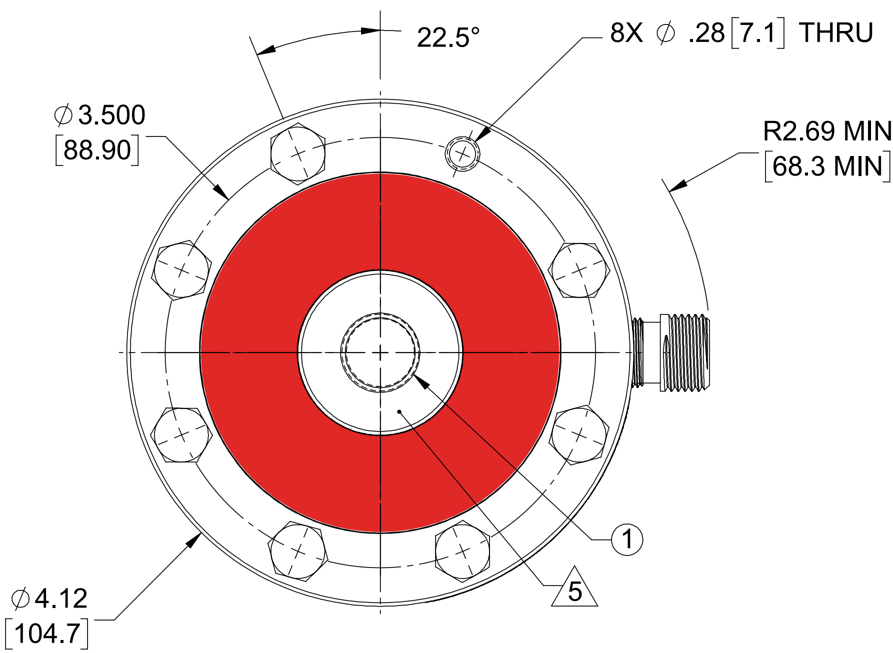
Interface

FORCE MEASUREMENT SOLUTIONS®

LOAD CELLS 301 GUIDE

LOAD CELL CHARACTERISTICS & APPLICATIONS



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Welcome to the Interface Load Cell 301 Guide, an indispensable technical resource written by industry force measurement experts. This advanced guide is designed for test engineers and measurement device users seeking comprehensive insights into load cell performance and optimization.

In this practical guide, we explore critical topics with technical explanations, visualizations, and scientific details essential for understanding and maximizing the functionality of load cells in diverse applications.

Learn how the inherent stiffness of load cells affects their performance under different loading conditions. Next, we investigate load cell natural frequency, analyzing both lightly loaded and heavily loaded scenarios to comprehend how load variations influence frequency response.

Contact resonance is another crucial aspect covered extensively in this guide, shedding light on the phenomenon and its implications for accurate measurements. Additionally, we discuss the application of calibration loads, emphasizing the importance of conditioning the cell and addressing impacts and hysteresis during calibration procedures.

Test protocols and calibrations are thoroughly examined, providing sensible guidelines for ensuring precision and reliability in measurement processes. We also delve into the application of in-use loads, focusing on on-axis loading techniques and strategies for controlling off-axis loads to enhance measurement accuracy.

Furthermore, we explore methods for reducing extraneous loading effects by optimizing design, offering valuable insights into mitigating external influences on load cell performance. Overload capacity with extraneous loading and dealing with impact loads are also discussed in detail to equip engineers with the knowledge needed to safeguard load cells against adverse conditions.

The Interface Load Cell 301 Guide provides invaluable information to optimize performance, enhance accuracy, and ensure the reliability of measurement systems in various applications.

Your Interface Team

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LOAD CELL CHARACTERISTICS & APPLICATIONS

LOAD CELL STIFFNESS

Customers frequently want to use a load cell as an element in the physical structure of a machine or assembly. Therefore, they would like to know how the cell would react to the forces developed during the assembly and operation of the machine.

For the other parts of such a machine that are made from stock materials, the designer can look up their physical characteristics (such as thermal expansion, hardness, and stiffness) in handbooks and determine the interactions of his parts based on his design. However, since a load cell is built on a flexure, which is a complex machined part whose details are unknown to the customer, its reaction to forces will be difficult for the customer to determine.

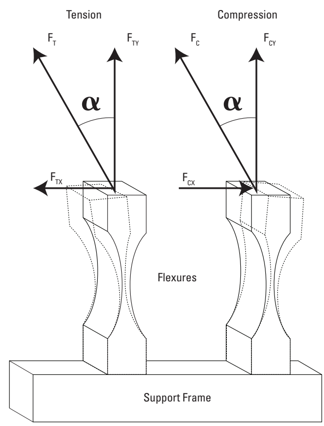

Figure 1. Comparison of Tension and Compression Reactions to Off-Axis Load

It is a useful exercise to consider how a simple flexure responds to loads applied in different directions. *Figure 1*, shows examples of a simple flexure made by grinding a cylindrical groove into both sides of a piece of steel stock. Variations of this idea are used extensively in machines and test stands to isolate load cells from side loads. In this example, the simple flexure represents a member in a machine design, not an actual load cell.

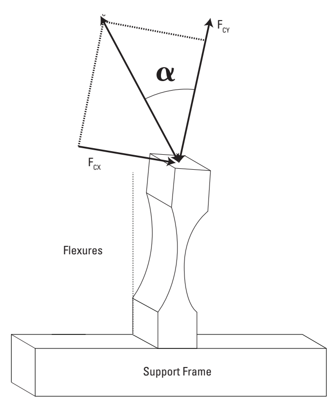

Figure 2. Off-Axis Compressive Loading with Possible Destructive Results to the Flexure

The thin section of the simple flexure acts as a virtual frictionless bearing having a small rotational spring constant. Therefore, the spring constant of the material may have to be measured and factored into the response characteristics of the machine.

If we apply a tensile force (F_T) or a compressive force (F_C) to the flexure at an angle off of its centerline, the flexure will be distorted sideways by the vector component (F_{TX}) or (F_{CX}) as shown by the dotted outline. Although the results look quite similar for both cases, they are drastically different.

In the tensile case in *Figure 1*, the flexure tends to bend into alignment with the off-axis force and the flexure assumes an equilibrium position safely, even under considerable tension.

In the compressive case, the flexure's reaction, as shown in *Figure 2*, can be highly destructive, even though the applied force is exactly the same magnitude and is applied along the same line of action as the tensile force, because the flexure bends away from the line of action of the applied force. This tends to increase the side force (F_{CX}) with the result that the flexure bends even more. If the side force exceeds the ability of the flexure to resist the turning motion, the flexure will continue to bend and will ultimately fail. Thus, the failure mode in compression is bending collapse, and will occur at a much lower force than can be safely applied in tension.

The lesson to be learned from this example is that extreme caution must be applied when designing compressive load cell applications using columnar structures. Slight misalignments can be magnified by the motion of the column under compressive loading, and the result can range from measurement errors to complete failure of the structure.

The previous example demonstrates one of the major advantages of the Interface® LowProfile® cell design. Since the cell is so short in relation to its diameter, it does not behave like a column cell under compressive loading. It is much more tolerant of misaligned loading than a column cell is.

The stiffness of any load cell along its primary axis, the normal measurement axis, can be calculated easily given the rated capacity of the cell and its deflection at rated load. Load cell deflection data can be found in the Interface® catalog and website.

NOTE:

Keep in mind that these values are typical, but are not controlled specifications for the load cells. In general, the deflections are characteristics of the flexure design, the flexure material, the gage *factors and the final calibration of the cell. These parameters are each individually controlled, but the cumulative effect may have some variability.*

Using the SSM-100 flexure in *Figure 3,* as an example, the stiffness in the the primary axis (Z) can be calculated as follows:

$$S_Z \quad \frac{C_R}{D_{RO}}$$

Where:

S_Z = *Stiffness on Primary Axis*

C_R = *Rated Capacity*

D_{RO} = *Deflection at Rated Output*

For the SSM-100:

$$S_Z = \frac{100}{0.004} = 25,000 \ lb/in$$

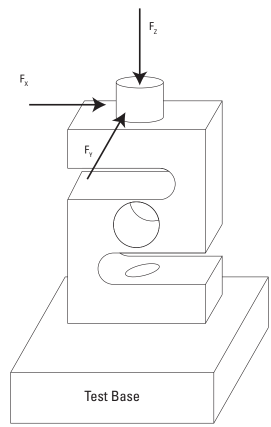

Figure 3. Test Forces Applied to S-Beam Cell.

This type of calculation is true for any linear load cell on its primary axis. In contrast, the stiffnesses of the (X) and (Y) axes are much more complicated to determine theoretically, and they are not usually of interest for users of Mini Cells, for the simple reason that the response of the cells on those two axes is not controlled as it is for the LowProfile® series. For Mini Cells, it is always advisable to avoid the application of side loads as much as possible, because the coupling of off-axis loads into the primary axis output can introduce errors into the measurements.

For example, application of the side load (F_X) causes the gages at A to see tension and the gages at (B) to see compression. If the flexures at (A) and (B) were identical and the gage factors of the gages at (A) and (B) were matched, we would expect the output of the cell to cancel the effect of the side load. However, since the SSM series is a low-cost utility cell which is typically used in applications having low side loads, the extra cost to the customer of balancing out the side load sensitivity is usually not justifiable.

The correct solution where side loads or moment loads may occur is to uncouple the load cell from those extraneous forces by the use of a rod end bearing at one or both of the ends of the load cell.

For example, *Figure 4*, shows a typical load cell installation for weight of a barrel of fuel sitting on a weigh pan, in order to weigh the fuel used in engine tests.

A clevis is mounted firmly to the support beam by its stud. The rod end bearing is free to rotate around the axis of its support pin, and can also move about ±10 degrees in rotation both in and out of the page and around the primary axis of the load cell. These freedoms of motion ensure that the tension load stays on the same centerline as the load cell's primary axis, even if the load is not properly centered on the weigh pan.

Note that the nameplate on the load cell reads upside down because the dead end of the cell must be mounted to the support end of the system.

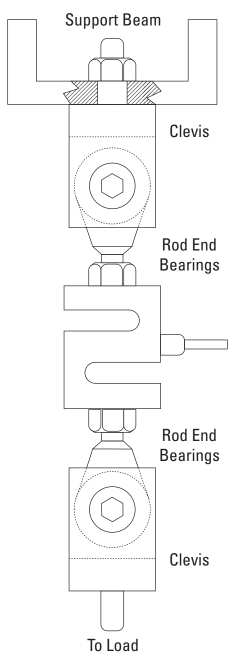

Figure 4. Using Rod End Bearings to Protect Load Cell from Extraneous Loads.

LOAD CELL NATURAL FREQUENCY: LIGHTLY LOADED CASE

Frequently a load cell will be used in a situation in which a light load, such as a weigh pan or small test fixture, will be attached to the live end of the cell. The user would like to know how quickly the cell will respond to a change in loading. By connecting the output of a load cell to an oscilloscope and running a simple test, we can learn some facts about the dynamic response of the cell. If we firmly mount the cell on a massive block and then tap the cell's active end very lightly with a tiny hammer, we will see a damped sine wave train (a series of sine waves which progressively decrease to zero).

NOTE:

Use extreme caution when applying impact to a load cell. The force levels can damage the cell, even for very short intervals.

The frequency (number of cycles occurring in one second) of the vibration can be determined by measuring the time (T) of one complete cycle, from one positive-going zero crossing to the next. One cycle is indicated on the oscilloscope picture in *Figure 5*, by the bold trace line. Knowing the period (time for one cycle), we can calculate the natural frequency of free oscillation of the load cell (f_O) from the formula:

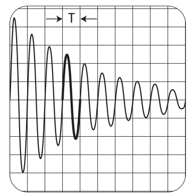

Figure 5. Oscilloscope Trace of Damped Oscillation.

$$f_O = \frac{1}{T}$$

Where:
f_O = *Natural Frequency*
T = *Time for One Cycle*

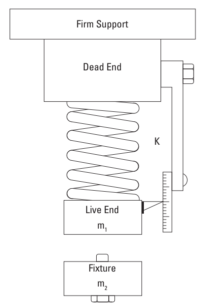

Figure 6. Load Cell Equivalent Elements

The natural frequency of a load cell is of interest because we can use its value to estimate the dynamic response of the load cell in a lightly loaded system.

NOTE:

Natural frequencies are typical values, but are not a controlled specification. They are given in the Interface® catalog only as an assistance to the user.

The equivalent spring-mass system of a load cell is shown in *Figure 6*. The mass (M_1) corresponds to the mass of the live end of the cell, from the attachment point to the thin sections of the flexure. The spring, having spring constant (K), represents the spring rate of the thin measurement section of the flexure. The mass (M_2), represents the added mass of any fixtures which are attached to the live end of the load cell.

Figure 7 relates these theoretical masses to the actual masses in a real load cell system. Note that the spring constant (K) occurs on the dividing line at the thin section of the flexure.

Natural frequency is a basic parameter, the result of the design of the load cell, so the user must understand that the addition of any mass on the active end of the load cell will have the effect of lowering the total system's natural frequency. For example, we can imagine pulling down slightly on the mass M_1 in *Figure 6* and then letting go. The mass will oscillate up and down at a frequency that is determined by the spring constant (K) and the mass of M_1. In fact, the oscillations will damp out as time progresses in much the same way as in *Figure 5*.

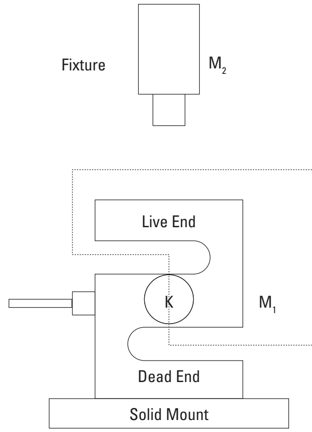

Figure 7. Partitioning of Elements in a Load Cell System

If we now bolt the mass (M_2) on (M_1), the increased mass loading will lower the natural frequency of the spring-mass system. Fortunately, if we know the masses of (M_1) and (M_2) and the natural frequency of the original spring-mass combination, we can calculate the amount that the natural frequency will be lowered by the addition of (M_2), in accordance with the formula:

$$f_S = f_O \sqrt{\frac{M_1}{M_1 + M_2}}$$

Where:

f_S = *System Natural Frequency*
f_O = *Load Cell Natural Frequency*
M_1 = *Load Cell Live End Mass*
M_2 = *Added Fixture Mass*

To an electrical or electronic engineer, the static calibration is a (DC) parameter, whereas the dynamic response is an (AC) parameter. This is represented in *Figure 7*, where the DC calibration is shown on the factory calibration certificate, and users would like to know what the response of the cell will be at some driving frequency they will be using in their tests.

Note the equal spacing of the "Frequency" and "Output" grid lines on the graph in *Figure 7*. Both of these are logarithmic functions; that is, they represent a factor of 10 from one grid line to the next. For example, "0 db" means “no change”; “+20 db” means “10 times as much as 0 db”; “–20 db” means “1⁄10 as much as 0 db”; and “–40 db” means “1⁄100 as much as 0 db.”

By using logarithmic scaling, we can show a larger range of values, and the more common characteristics turn out to be straight lines on the graph. For example, the dashed line shows the general slope of the response curve above the natural frequency. If we continued the graph down and off to the right, the response would become asymptotic (closer and closer) to the dashed straight line.

NOTE:

The curve in Figure 63 is provided only to portray the typical response of a lightly loaded load cell under optimum conditions. In most installations, the resonances in the attaching fixtures, test frame, driving mechanism and UUT (unit under test) will predominate over the load cell’s response.

LOAD CELL NATURAL FREQUENCY: HEAVILY LOADED CASE

In cases where the load cell is mechanically tightly coupled into a system where the masses of the components are significantly heavier than the load cell’s own mass, the load cell tends more to act like a simple spring that connects the driving element to the driven element in the system.

The problem for the system designer becomes one of analyzing the masses in the system and their interaction with the very stiff spring constant of the load cell. There is no direct correlation between the load cell’s unloaded natural frequency and the heavily loaded resonances which will be seen in the user’s system.

CONTACT RESONANCE

Almost everyone has bounced a basketball and noticed that the period (time between cycles) is shorter when the ball is bounced closer to the floor. Anyone who has played a pinball machine has seen the ball rattling back and forth between two of the metal posts; the closer the posts get to the diameter of the ball, the faster the ball will rattle. Both of these resonance effects are driven by the same elements: a mass, a free gap, and a springy contact which reverses the direction of travel.

The frequency of oscillation is proportional to the stiffness of the restoring force, and inversely proportional both to the size of the gap and to the mass. This same resonance effect can be found in many machines, and the buildup of oscillations can damage the machine during normal operation.

Figure 8. Contact Resonance: Living Demonstration.

For example, in *Figure 9*, a dynamometer is used to measure the horsepower of a gasoline engine. The engine under test drives a water brake whose output shaft is connected to a radius arm. The arm is free to rotate, but is constrained by the load cell. Knowing the RPM of the engine, the force on the load cell, and the length of the radius arm, we can calculate the horsepower of the engine.

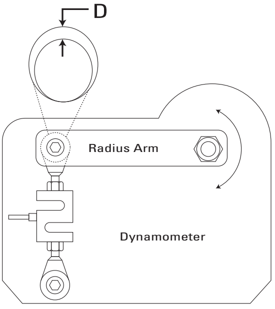

Figure 9. Destructive Contact Resonance Caused by Ball Clearance.

If we look at the detail of the clearance between the ball of the rod end bearing and the sleeve of the rod end bearing in *Figure 9*, we will find a clearance dimension, *(D)*, because of the difference in size of the ball and its constraining sleeve. The sum of the two ball clearances, plus any other looseness in the system, will be the total "gap" which can cause a contact resonance with the mass of the radius arm and the spring rate of the load cell.

As the engine speed is increased, we may find a certain RPM at which the rate of firing of the engine's cylinders matches the contact resonance frequency of the dynamometer. If we hold that RPM, magnification (multiplication of the forces) will occur, a contact oscillation will build up, and impact forces of ten or more times the average force could easily be imposed on the load cell.

This effect will be more pronounced when testing a one-cylinder lawn mower engine than when testing an eight cylinder auto engine, because the firing impulses are smoothed out as they overlap in the auto engine. In general, raising the resonant frequency will improve the dynamic response of the dynamometer.

The effect of contact resonance can be minimized by:

- Using high quality rod end bearings, which have very low play between the ball and socket.
- Tightening the rod end bearing bolt to ensure that the ball is tightly clamped in place.
- Making the dynamometer frame as stiff as possible.
- Using a higher capacity load cell to increase the load cell stiffness.

APPLICATION OF CALIBRATION LOADS: CONDITIONING THE CELL

Any transducer that depends upon the deflection of a metal for its operation, such as a load cell, torque transducer, or pressure transducer, retains a history of its previous loadings. This effect occurs because the minute motions of the crystalline structure of the metal, small as they are, actually have a frictional component that shows up as hysteresis (non-repeating of measurements that are taken from different directions).

Prior to the calibration run, the history can be swept out of the load cell by the application of three loadings, from zero to a load which exceeds the highest load in the calibration run. Usually, at least one load of 130% to 140% of the Rated Capacity is applied, to allow the proper setting and jamming of the test fixtures into the load cell.

If the load cell is conditioned and the loadings are properly done, a curve having the characteristics of (*A-B-C-D-E-F-G-H-I-J-A*), as in *Figure 10*, will be obtained. The points will all fall onto a smooth curve, and the curve will be closed on the return to zero.

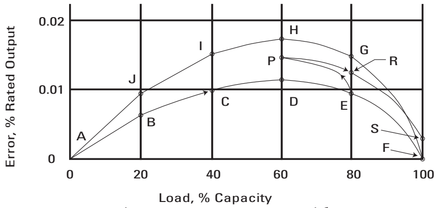

Figure 10. Output Errors with Different Loading Sequences

Furthermore, if the test is repeated and the loadings are properly done, the corresponding points between the first and second runs will fall very close to each other, demonstrating the repeatability of the measurements.

APPLICATION OF CALIBRATION LOADS: IMPACTS AND HYSTERESIS

Whenever a calibration run yields results that don't have a smooth curve, don't repeat well, or don't return to zero, the test setup or loading procedure should be the first place to check.

For example, *Figure 10* shows the result of the application of loads where the operator was not careful when the 60% load was applied. If the weight was dropped slightly onto the loading rack and applied an impact of 80% load and then returned to the 60% point, the load cell would be operating on a minor hysteresis loop that would end up at point (*P*) instead of at point (*D*). Continuing the test, the 80% point would end up at (*R*), and the 100% point would end up at (*S*). The descending points would all fall above the correct points, and the return to zero would not be closed.

The same type of error can occur on a hydraulic test frame if the operator overshoots the correct setting and then leaks back the pressure to the correct point. The only recourse for impacting or overshooting is to recondition the cell and retest.

TEST PROTOCOLS AND CALIBRATIONS

Load cells are routinely conditioned in one mode (either tension or compression), and then calibrated in that mode. If a calibration in the opposite mode is also required, the cell is first conditioned in that mode prior to the second calibration. Thus, the calibration data reflects the operation of the cell only when it is conditioned in the mode in question.

For this reason, it is important to determine the test protocol (the sequence of load applications) which the customer is planning to use, before a rational discussion of the possible sources of error can occur. In many cases, a special factory acceptance must be devised to ensure that the user's requirements will be met.

For very stringent applications, users are generally able to correct their test data for the nonlinearity of the load cell, thus removing a substantial amount of the total error. If they are unable to do so, nonlinearity will be part of their error budget.

Nonrepeatability is essentially a function of the resolution and stability of the user's signal conditioning electronics. Load cells typically have nonrepeatability that is better than the load frames, fixtures, and electronics that are used to measure it.

The remaining source of error, hysteresis, is highly dependent on the loading sequence in the user's test protocol. In many cases, it is possible to optimize the test protocol so as to minimize the introduction of unwanted hysteresis into the measurements.

However, there are cases in which users are constrained, either by an external customer requirement or by an internal product specification, to operate a load cell in an undefined way that will result in unknown hysteresis effects. In such instances, the user will have to accept the worst case hysteresis as an operating specification.

Also, some cells must be operated in both modes (tension and compression) during their normal use cycle without being able to recondition the cell before changing modes. This results in a condition called toggle (non-return to zero after looping through both modes). In normal factory output, the magnitude of toggle is a broad range where the worst case is approximately equal to or slightly larger than hysteresis, depending on the load cell's flexure material and capacity.

Fortunately, there are several solutions to the toggle problem:

- Use a higher capacity load cell so that it can operate over a smaller range of its capacity. Toggle is lower when the extension into the opposite mode is a smaller percentage of rated capacity.
- Use a cell made from a lower toggle material. Contact the factory for recommendations.
- Specify a selection criterion for normal factory production. Most cells have a range of toggle that may yield enough units from the normal

distribution. Depending on the factory build rate, the cost for this selection is usually quite reasonable.

- Specify a tighter specification and have the factory quote a special run.

APPLICATION OF IN-USE LOADS: ON-AXIS LOADING

All on-axis loadings generate some level, no matter how small, of off-axis extraneous components. The amount of this extraneous loading is a function of the tolerancing of the parts in the design of the machine or load frame, the precision with which the components are manufactured, the care with which the elements of the machine are aligned during assembly, the rigidity of the load-bearing parts, and the adequacy of the attaching hardware.

CONTROL OF OFF-AXIS LOADS

The user can opt to design the system so as to eliminate or reduce off-axis loading on the load cells, even if the structure suffers distortion under load. In tension mode, this is possible by the use of rod end bearings with clevises.

Where the load cell can be kept separate from the structure of the test frame, it can be used in compression mode, which almost eliminates the application of off axis load components to the cell. However, in no case can off-axis loads be completely eliminated, because the deflection of load carrying members will always occur, and there will always be a certain amount of friction between the load button and the loading plate which can transmit side loads into the cell.

When in doubt, the LowProfile® cell will always be the cell of choice unless the overall system error budget allows a generous margin for extraneous loads.

REDUCING EXTRANEOUS LOADING EFFECTS BY OPTIMIZING DESIGN

In high-precision test applications, a rigid structure with low extraneous loading can be achieved by the use of ground flexures to build the measurement frame. This, or course, requires precision machining and assembly of the frame, which may constitute a considerable cost.

OVERLOAD CAPACITY WITH EXTRANEOUS LOADING

One serious effect of off-axis loading is the reduction of the cell's overload capacity. The typical 150% overload rating on a standard load cell or the 300% overload rating on a fatigue-rated cell is the allowed load on the primary axis, without any side loads, moments or torques applied to the cell concurrently. This is because the off-axis vectors will add with the on-axis load vector, and the vector sum can cause an overload condition in one or more of the gaged areas in the flexure.

To find the allowed on-axis overload capacity when the extraneous loads are known, compute the on-axis component of the extraneous loads and algebraically subtract them from the rated overload capacity, being careful to keep in mind in which mode (tension or compression) the cell is being loaded.

IMPACT LOADS

Neophytes in the use of load cells frequently destroy one before an old-timer has a chance to warn them about impact loads. We would all wish that a load cell could absorb at least a very short impact without damage, but the reality is that if the live end of the cell moves more than 150% of the full capacity deflection in relation to the dead end, the cell could be overloaded, no matter how short the interval over which the overload occurs.

In Panel 1 of the example in *Figure 11*, a steel ball of mass "m" is dropped from height "S" onto the live end of the load cell. During the fall, the ball is accelerated by gravity and has achieved a velocity "v" by the instant it makes contact with the surface of the cell.

In Panel 2, the velocity of the ball will be completely stopped, and in Panel 3 the direction of the ball will be reversed. All this must happen in the distance it takes for the load cell to reach the rated overload capacity, or the cell may be damaged.

In the example shown, we have picked a cell that can deflect a maximum of 0.002” before being overloaded. In order for the ball to be completely stopped in such a short distance, the cell must exert a tremendous force on the ball. If the ball weighs one pound and it is dropped one foot onto the cell, the graph of *Figure 12* indicates that the cell will receive an impact of 6,000 lbf (it is assumed that the mass of the ball is much larger than the mass of the live end of the load cell, which is usually the case).

The scaling of the graph can be modified mentally by keeping in mind that the impact varies directly with the mass and with the square of the distance dropped.

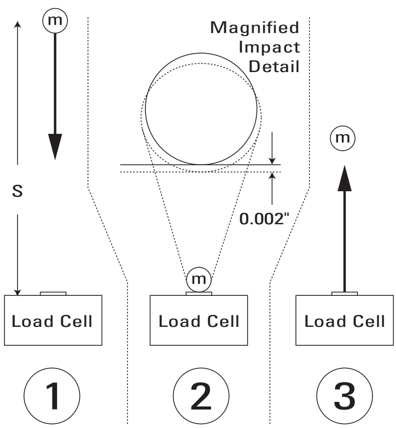

Figure 11. Peak Impact Load

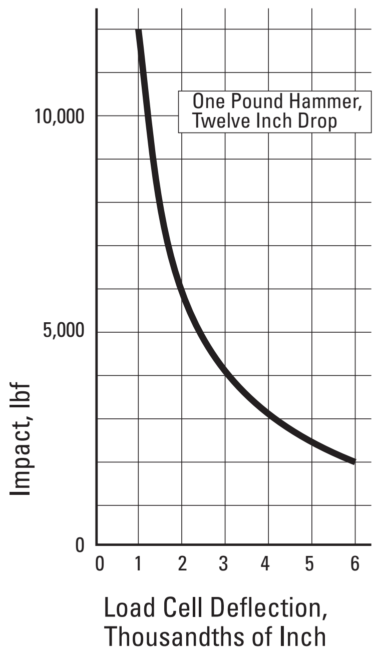

Figure 12. Typical Curve, Impact vs Load Deflection

Interface® is the trusted The World Leader in Force Measurement Solutions®. We lead by designing, manufacturing, and guaranteeing the highest performance load cells, torque transducers, multi-axis sensors, and related instrumentation available. Our world-class engineers provide solutions to the aerospace, automotive, energy, medical, and test and measurement industries from grams to millions of pounds, in hundreds of configurations. We are the preeminent supplier to Fortune 100 companies worldwide, including; Boeing, Airbus, NASA, Ford, GM, Johnson & Johnson, NIST, and thousands of measurement labs. Our in-house calibration labs support a variety test standards: ASTM E74, ISO-376, MIL-STD, EN10002-3, ISO-17025, and others.

You can find more technical information about load cells and Interface®'s product offering at www.*interfaceforce.com*, or by calling one of our expert Applications Engineers at *480.948.5555*.

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